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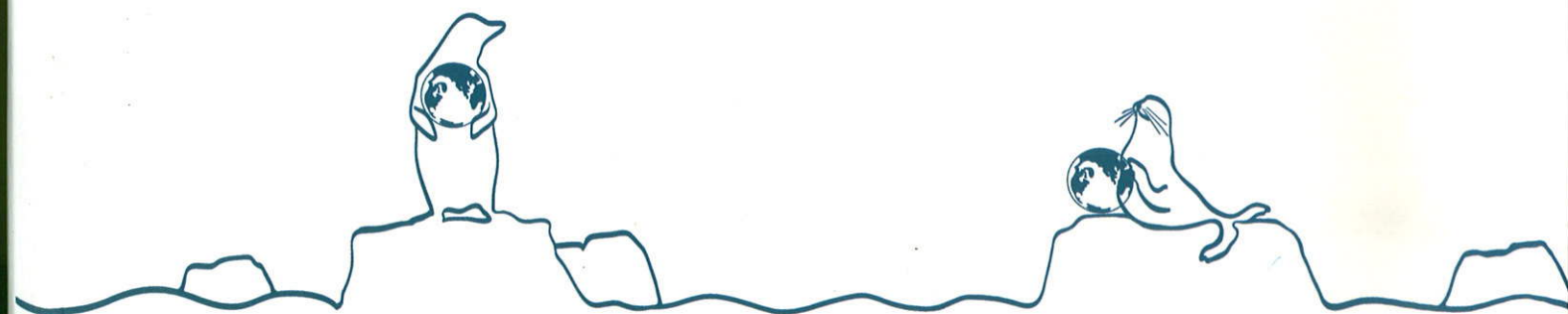


**Asia-Pacific
Economic Cooperation**

**APEC Conference on
Low Carbon Town and Physical Energy Storage
PROCEEDINGS**

Expert Group on
New & Renewable Energy Technologies (EGNRET)
of Energy Work Group (EWG)

Organized by
Hunan University
The HongKong Polytechnic University
University of Kansas



APEC · 2013

Changsha · Hunan · China

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PREFACE

All over the world society's foremost challenges this century are global warming, sustainability and security of energy supply. These issues are altogether linked with global stability, economic prosperity, and quality of life. At the 9th APEC Energy Ministers Meeting (EMM9), ministers pointed out the importance of developing renewable energy, which focused on the theme "Low Carbon Paths to Energy Security". They observed that the "Introduction of low-carbon technologies in city planning to boost energy efficiency and reduce fossil energy use is vital to manage the rapidly growing energy consumption in urban areas of APEC". Physical energy storage has garnered particular attention because of the large amount of heat absorption/release during the phase change processes with only small temperature change.

This proceeding is based on the APEC Conference on Low-carbon Towns and Physical Energy Storage. More than 120 papers from all over the world are included in this proceeding, mainly about advanced technology about how to build and reach Low-carbon Town and new physical energy storage technology for energy conversation with phase change material. This proceeding can provide a good communication platform and reference for technical cooperation for all researchers.

This proceeding is co-edited by Hunan University, Hong Kong Polytechnic University and University of Kansas. Thanks for Zhang Quan (Hunan University), Yang Hongxing (Hong Kong Polytechnic University), Mario A. Medina (University of Kansas), Liao Shuguang (Changsha Maxxom High-tech Co., Ltd.), Sun Xiaoqin (Hunan University) and Zhou Ping (Hunan University). The proceeding is partly financial support by research project, which mainly including APEC project (SEWG 16 12A), [KY2011-No.15] ,[K1104273-11], [KY2012-No.155], [11JJ1009], [2012AA052503].

This proceeding is inevitably inadequate; your advices and suggestions will be highly appreciated.

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A PANEL DATA PARAMETRIC FRONTIER TECHNIQUE FOR MEASURING ENERGY EFFICIENCY: APPLICATION TO JAPANESE REGIONS

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ABSTRACT

This paper, using the stochastic frontier analysis (SFA) model, estimates total factor-energy efficiency (TFEE) scores for 47 administrative regions in Japan during 1996-2008. For this purpose, we extend the cross-sectional SFA model proposed by Zhou et al. [Zhou, P., B.W. Ang and D.Q. Zhou (2012), Measuring Economy-wide Energy Efficiency Performance: A Parametric Frontier Approach. *Applied Energy*, 90, 196-200] to a panel data model. The results in this paper could provide the TFEE in which statistical noise is taken into account. In the previous studies to evaluate TFEE, including the authors, data envelopment analysis (DEA) has been mainly employed. DEA is a nonparametric, linear programming method and widely used to evaluate the performance of decision-making units (DMUs) such as firms, governments, agencies.

The empirical results show that there is still much room to save energy consumption in most Japanese regions. For the bottom three regions, Chiba, Okayama, and Oita, the TFEE scores are very low, less than 0.2, for all models. This implies that there is much room to save energy, larger than 80%, in each of the three regions. Our technical inefficiency effects models use broad and detail industry share variables as environmental variables which describe the determinants of inefficiency. The former results show that the share of higher share of manufacturing industry and wholesale and retail trade incur more inefficiency. The later results share of pulp, paper and paper products, chemicals, non-metallic mineral products, and iron and steel industries significantly depreciate efficiency. Share of wholesale and retail trade industry also remains incurring inefficiency but its coefficient drops to a lower value than the former.

Keywords: Stochastic frontier analysis (SFA); Total-factor energy efficiency (TFEE); Panel data; Distance function

1. INTRODUCTION

After the Fukushima Daiichi nuclear disaster in 2011, in Japan energy-saving has been an urgent issue. For political constraint for restart nuclear power plants, the Oi plant is now only operating nuclear power station after the Fukushima. Although there has been full operation of thermal power plants - some of them were inactive before the Fukushima for their out-of-date technology - and effort to save energy in firms and houses, electricity shortage remains serious. Moreover, Japan loses money on the international trade for importing fossil fuels after the Fukushima. Because the restart of nuclear power plants faces political difficulties and renewable energy cannot immediately meet electricity demand, the only way that no one has any objection is improving energy efficiency. For this purpose, a more accurate measurement of regional energy efficiency is needed.

There have been large studies on energy efficiency by using data envelopment analysis (DEA). DEA is a nonparametric linear-programming methodology to measure efficiency of multiple decision making units

(DMUs). Hu and Wang (2006) and Hu and Kao (2007) propose the total-factor energy efficiency (TFEE) index. It is applied for Japan by Honma and Hu (2008, 2013), for Chinese Taipei by Hu et al. (2013), OECD countries by Honma and Hu (2011). Although DEA is one of the powerful tools to evaluate efficiency, it has a drawback that its efficiency is suffering from statistical noises.

This paper, using the stochastic frontier analysis (SFA) model, estimates TFEE scores for 47 administrative regions in Japan during 1996-2008. For this purpose, we extend the cross-sectional SFA model proposed by Zhou et al. (2012). The results in this paper could provide the TFEE in which statistical noise is taken into account.

In our model, efficiency measurement is based on the Shephard energy distance function that is assumed as the Cobb-Douglas functional form. Our study employs three inputs, energy, labor, and capital stock. Regional GDP is the sole output. The distinctive feature of the estimation technique is that 'reciprocal of energy consumption' is deemed as an output and is produced by using labor, capital stock, and GDP as inputs. It enables us to estimate parametrically energy efficiency.

The maximum likelihood estimator is used to estimate parameters including inefficiency component. In our models, the determinants of inefficiency are also simultaneously estimated by employing the technical inefficiency effects model proposed by Battese and Coelli (1995).

The rest of the paper is organized as follows. Section 2 describes our methodology and data. Section 3 presents the TFEE results and determinants of inefficiency. Section 4 concludes with a brief summary.

2. METHODOLOGY AND DATA

2.1. The SFA Model for input efficiency

Zhou et al. (2012) apply the single-equation, output-oriented stochastic frontier analysis (SFA) model to estimate the total-factor energy efficiency. Their version is still a cross-section SFA model, which analyze 21 OECD countries in 2001. Combining Zhou et al. (2012) and Battese and Coelli (1992), this paper further extends a panel data SFA model to estimate the total-factor energy efficiency. Following Zhou et al. (2012), we assume that the stochastic frontier distance function is in the Cobb-Douglas functional form:

$$\ln D(K_{it}, L_{it}, E_{it}, Y_{it}) = \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_E \ln E_{it} + \beta_Y \ln Y_{it} + v_{it} \quad (1)$$

where $D(\cdot)$ is the distance function, K_{it} is the capital stock, L_{it} is labor employment, E_{it} is the energy input, Y_{it} is the real economic output, i indicates the region, and t indicates the time, and v_{it} is the statistical noise following the normal distribution. Since the distance function is homogeneous of degree one in the energy input, the above equation can be re-arranged as:

$$\ln D_E(K_{it}, L_{it}, E_{it}, Y_{it}) = \ln E_{it} + \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_E \ln 1 + \beta_Y \ln Y_{it} + v_{it} \quad (2)$$

which can be also arranged as

$$-\ln E_{it} = \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_E \ln 1 + \beta_Y \ln Y_{it} + v_{it} - \ln D_E(K_{it}, L_{it}, E_{it}, Y_{it}) \quad (3)$$

That is,

$$\ln(1/E_{it}) = \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_Y \ln Y_{it} + v_{it} - u_{it} \quad (4)$$

where u_{it} is the inefficiency term which follows a non-negative distribution and $v_{it} - u_{it}$ is the error component term of a stochastic production frontier. Eq. (4) fits in the panel data, stochastic frontier model proposed by Battese and Coelli (1992). The free software Frontier 4.1 kindly provided by Professor Coelli (1996) can be used to estimate Eq (4). The total-factor energy efficiency of region i at time t then is

$$TFEE_{it} = \exp(-u_{it}) \quad (5)$$

Therefore, we then can apply the panel data, stochastic production frontier approach to estimate the total-factor energy efficiency and are limited to use the input-

oriented data envelopment analysis (DEA) suggested by Hu and Wang (2006) and Hu and Kao (2007). Moreover, if we use disaggregate energy inputs, we can also change the logged inverse energy input on the left-hand side of Eq. (4) and keep other logged inputs on the right-hand side of Eq. (4), such that we can obtain the TFEE scores of different energy inputs.

Battese and Coelli (2005) further add up an inefficiency equation for simultaneous estimation with the stochastic frontier in the form of Eq. (4):

$$u_{it} = \delta_0 + \delta_1 z_{1it} + \dots + \delta_K z_{Kit} + \varepsilon_{it} \quad (6)$$

where the variables $z_1, \dots, z_K$ are environmental variables and ε are white noises following the normal distribution. As a result, we can simultaneously estimate Eqs (4) and (6) by applying the approach of Battese and Coelli (1995). The free software Frontier 4.1 kindly provided by Professor Coelli (1996) can be used to simultaneously estimate Eqs (4) and (6).

2.2. Data and variables

In our SFA model, we assumed as if reciprocal of energy consumption is produced using regional GDP, labor, and capital stock. Data on regional GDP, labor, and capital stock are taken from the Economic and Fiscal Model by Prefecture (the Cabinet Office, Government of Japan). All monetary values are 2000 year million yen. Labor is the number of employees. Data on energy are taken from the Energy Consumption Statistics by Prefecture (Agency for Natural Resources and Energy, Japan). The aggregated energy consumption is the sum of oils, gases, coal, electricity, and industrial heat and is presented by thermal unit, Tera Joule (TJ). The aggregated energy does not contain residential sector, transportation sector, and non-energy use. Because residential energy consumption such as cooking, air heating, hot-water supply does not generate any value added, it is excluded from the aggregated energy consumption. For the same reason, energy consumption by private vehicles is excluded. Energy consumption in business transportation sector is unavailable because fuels consumed beyond region borders cannot be accurately parted to each region in the statistics.

We employ industry shares as environmental variables in the technical inefficiency effects model¹. We constructed two models for this purpose. When the share of the manufacturing industry that is aggregated into one category is used as one of the environmental variables, we refer it as the *broad* industry share model. When shares of five

¹ There are various candidates for environmental variables except industry share, e.g., social variables such as population density, and natural variables such as average temperature and yearly precipitation. Because many of them are correlated with industry share (for example, service industry is likely located in highly populated areas), we refrain from adding more environmental variables in order to avoid multicollinearity.

representative energy-intensive industries are used, we refer it as the *detail* industry share model. Data on broad and detail industry shares are taken from the Annual Report on Prefectural Accounts (Cabinet Office, Government of Japan). Data on detail industry shares for Okinawa prefecture which is small islands lacks is unavailable. The sample period is from 1996 to 2008. Table 1 reports the summary of statistics of input, output, environmental variables.

3. RESULTS

3.1. TFEE scores

The maximum-likelihood estimates of TFEE scores are given in Table 2². To the space limitation, we show only the average TFEE scores and rankings in three models during 1996-2008³. Column 1 reports TFEE scores that are estimated without using environmental variables, following Zhou et al. (2012) and Hu (2013). It is only presented to compare with column 2 and 3. The results of the new technique proposed in this paper are columns 2 and 3, which report efficiency that is measured with broad and detail industry share. It should be noted that determinants of inefficiency are simultaneously estimated in columns 2 and 3. The top three regions, Yamagata, Nagasaki, and Tokyo, are the same between columns 2 and 3⁴. They obtain very high TFEE scores more than 0.95 except Tokyo in column 3. It means that there little rooms to save energy consumption, less than 5 percent, in these regions. The bottom three regions, Chiba, Okayama, and Oita, are also the same between them⁵. Their TFEE scores are very low, less than 0.2, for all models. This implies that there is still much room to save energy, larger than 80%, in each of the three regions.

While rankings between columns 2 and 3 are similar, those between column 1 and 2 and between columns 1 and 3 are not similar. There may be one reason why some regions vary their rankings between columns 1 and 2 and between columns 1 and 3. Because expected mean inefficiency terms in columns 2 and 3 varies among regions depending on their environment variables, the more advantageous environment region is located, the more efficient it would obtain. As Figure 1 shows, distributions of TFEE scores of the each model are over side ranges. The

distributions of (2) and (4) have two peaks around 0.625 and 0.9. However, inefficient scores can be observed the range less than 0.5.

Table 3 shows the estimated coefficients and determinants of inefficiency. In column 1, whereas the coefficient of $\ln Y$ is negative contrary to the expectation and insignificant, those of $\ln L$ and $\ln K$ are significantly negative at the 1% level, respectively. The parameter, η , presents the trend of time-varying inefficiency (Battese and Coelli, 1992). A negative (positive) η implies that the efficiency is decreasing (increasing) over time. In column 1, η is insignificant. This implies that we cannot observe inefficiency increases or decreases over time. It is consistent with stable TFEE scores in each year. In column 2, the coefficient of $\ln Y$ is significantly positive at the 10% level and those of $\ln L$ and $\ln K$ are significantly negative at the 1% level, respectively. The signs of the above variables are as expected. In column 3, the coefficient of $\ln Y$ is positive but insignificant. The coefficients of $\ln L$ and $\ln K$ are significantly negative, as expected, at the 1% level, respectively.

3.2. Determinants of inefficiency

In columns 2 and 3, the determinants of inefficiency are simultaneously estimated in the models. Using broad industry share as environmental variables, the estimated coefficients for the share of manufacturing industry and wholesale and retail trade, 15.531 and 11.332, respectively, are highly significant, but that of the service activities is insignificant (column 2). It implies that regions with a higher share of manufacturing industry and wholesale and retail trade are more inefficient, vice versa. Column 3 provides more in-depth analysis on the determinants of inefficiency. Except non-ferrous metals industry, four energy-intensive industries, pulp, paper and paper products, chemicals, non-metallic mineral products, and iron and steel industries, highly significantly depreciate efficiency. Share of wholesale and retail trade industry remains incurring inefficiency but its coefficient drops to a lower value than column 2.

Table 4 reports the correlations of the three models of TFEE scores. The three TFEE scores are highly correlated. Especially, the correlation coefficient between the two models with broad and detail industry shares is 0.999.

4. CONCLUDING REMARKS

This paper estimates TFEE scores for regions in Japan and the determinants of inefficiency for 1996-2008. We extend the SFA approach employed in Zhou et al. (2012) and Hu (2013), incorporating technical inefficiency effects model proposed by Battese and Coelli (1995). The empirical results show that there is much room to save energy consumption in Japanese regions. For the bottom three regions, Chiba, Okayama, and Oita, the TFEE scores are very low, less than 0.2, for all models. This implies that there is much room to save energy, larger than 80%, in each of the three regions. Our technical inefficiency effects

² The estimations are calculated by Frontier 4.1 which is kindly provided by Coelli (1996).

³ The efficiency values of each region are very stable during the sample period.

⁴ For only Tokyo, there is a significant divergence between TFEE scores in column 1 and those in columns 2 and 3. This may come from that Tokyo has more advantageous environment for models in columns 2 and 3.

⁵ Note that the number of the regions in columns 1 and 2 is 47 but that in column 3 is 46 because TFEE score of Okinawa cannot be calculated by data unavailability.

models use broad and detail industry share variables as environmental variables which describe the determinants of inefficiency. The former results show that the share of higher share of manufacturing industry and wholesale and retail trade incur more inefficiency. The later results share of pulp, paper and paper products, chemicals, non-metallic mineral products, and iron and steel industries significantly depreciate efficiency. Share of wholesale and retail trade industry also remains incurring inefficiency but its coefficient drops to a lower value than the former.

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Table 1 The summary of statistics of input, output, environmental variables

Variable	Unit	Average	Standard deviation	Minimum	Maximum	Observations
Regional GDP	million yen	11,267,620	14,899,507	2,070,534	100,982,870	611
Labor	person	1,358,697	1,437,445	300,652	8,746,255	611
Capital stock	million yen	36,051,479	34,440,744	7,662,999	230,327,688	611
Energy	TJ	203,883	221,225	28,331	1,181,999	611
Share of manufacturing industry		0.21239	0.07702	0.04031	0.43107	611
Share of wholesale and retail trade industry		0.11268	0.03134	0.05782	0.21673	611
Share of service activities		0.19281	0.02771	0.11619	0.29261	611
Share of pulp ,paper and paper products industry		0.00575	0.00591	0.00006	0.04200	598
Share of chemicals industry		0.01798	0.01867	0.00021	0.10745	598
Share of non-metallic mineral products industry		0.00854	0.00588	0.00124	0.03798	598
Share of iron and steel industry		0.01395	0.01466	0.00044	0.10167	598

Share of non-ferrous metals industry	0.01359	0.00945	0.00198	0.09322	598

Table 2 Average TFEE scores of Japanese regions (1996-2008)

Region	1 TFEE scores (rank) without industry share variables	2 TFEE scores (rank) under broad industry share variables	3 TFEE scores (rank) under detail industry share variables
Hokkaido	0.406 (33)	0.641 (25)	0.607 (25)
Aomori	0.528 (29)	0.574 (30)	0.567 (30)
Iwate	0.770 (14)	0.845 (14)	0.838 (13)
Miyagi	0.584 (23)	0.674 (22)	0.676 (21)
Akita	0.877 (9)	0.888 (9)	0.886 (10)
Yamagata	0.971 (3)	0.966 (1)	0.967 (1)
Fukushima	0.689 (17)	0.802 (16)	0.793 (15)
Ibaraki	0.181 (43)	0.225 (43)	0.223 (42)
Tochigi	0.564 (24)	0.614 (26)	0.623 (24)
Gunma	0.663 (18)	0.750 (17)	0.754 (16)
Saitama	0.530 (28)	0.727 (19)	0.731 (18)
Chiba	0.105 (47)	0.146 (45)	0.144 (44)
Tokyo	0.562 (25)	0.951 (3)	0.949 (3)
Kanagawa	0.233 (40)	0.345 (39)	0.344 (38)
Niigata	0.513 (30)	0.656 (24)	0.637 (23)
Toyama	0.530 (27)	0.533 (32)	0.532 (31)
Ishikawa	0.920 (8)	0.889 (8)	0.907 (8)
Fukui	0.785 (11)	0.747 (18)	0.745 (17)
Yamanashi	0.964 (6)	0.881 (12)	0.900 (9)
Nagano	0.777 (13)	0.914 (7)	0.914 (7)
Gifu	0.617 (22)	0.704 (20)	0.701 (20)
Shizuoka	0.434 (32)	0.585 (29)	0.580 (28)
Aichi	0.305 (38)	0.479 (34)	0.469 (33)
Mie	0.207 (42)	0.229 (42)	0.230 (41)
Shiga	0.543 (26)	0.514 (33)	0.527 (32)
Kyoto	0.734 (16)	0.814 (15)	0.835 (14)
Osaka	0.375 (34)	0.599 (28)	0.589 (27)
Hyogo	0.254 (39)	0.352 (38)	0.347 (37)
Nara	0.982 (2)	0.887 (10)	0.916 (6)
Wakayama	0.349 (35)	0.341 (40)	0.338 (39)
Tottori	0.800 (10)	0.690 (21)	0.704 (19)
Shimane	0.987 (1)	0.924 (5)	0.920 (4)
Okayama	0.111 (46)	0.125 (46)	0.125 (45)
Hiroshima	0.219 (41)	0.269 (41)	0.268 (40)
Yamaguchi	0.137 (44)	0.148 (44)	0.146 (43)
Tokushima	0.634 (20)	0.571 (31)	0.578 (29)
Kagawa	0.477 (31)	0.447 (36)	0.456 (34)
Ehime	0.337 (36)	0.363 (37)	0.361 (36)
Kochi	0.632 (21)	0.602 (27)	0.598 (26)
Fukuoka	0.328 (37)	0.452 (35)	0.448 (35)
Saga	0.967 (5)	0.919 (6)	0.917 (5)
Nagasaki	0.935 (7)	0.958 (2)	0.961 (2)
Kumamoto	0.785 (12)	0.882 (11)	0.881 (11)
Oita	0.117 (45)	0.115 (47)	0.116 (46)
Miyazaki	0.660 (19)	0.665 (23)	0.669 (22)
Kagoshima	0.758 (15)	0.853 (13)	0.844 (12)
Okinawa	0.970 (4)	0.938 (4)	na
Average	0.570	0.621	0.614
SD	0.273	0.259	0.259
Max	0.987	0.966	0.967
Min	0.105	0.115	0.116

Table 3 Maximum-likelihood estimates for parameters of stochastic frontier functions for Japanese regions.

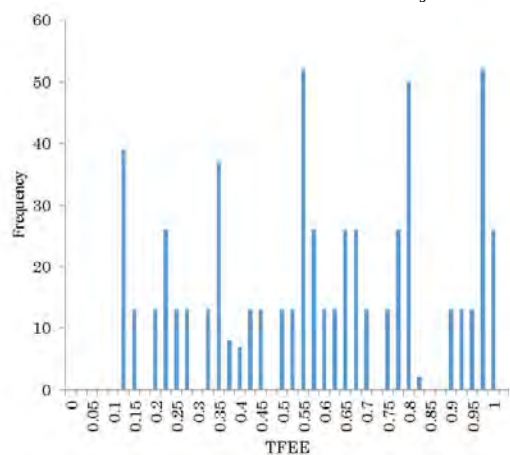
Variable	1 TFEE without industry share variables	2 TFEE with broad industry share variables	3 TFEE with detail industry share variables
Constant	-0.530 (-0.458)	3.491*** (6.94)	2.912*** (6.804)
lnY	-0.069 (-1.271)	0.306* (1.861)	0.181 (1.451)
lnL	-0.576*** (-6.41)	-0.971*** (-7.492)	-0.928*** (-9.109)
lnK	-0.084*** (-2.827)	-0.354*** (-3.735)	-0.239*** (-2.907)
Constant		-5.369*** (-2.389)	-1.64*** (-3.866)
Share of Manufacturing		15.531*** (4.187)	
Share of Pulp ,paper and paper			24.243*** (4.925)
Share of Chemicals			18.872*** (11.527)
Share of Non-metallic mineral			41.388*** (7.098)
Share of Iron and steel			35.694*** (15.276)
Share of Non-ferrous metals			-8.696** (-2.517)
Share of Wholesale and retail trade			7.359*** (6.392)
Share of Service activities			-0.785 (-0.453)
Share of Wholesale and retail trade		11.332** (2.504)	7.359*** (6.392)
Share of Service activities		-4.12 (-0.61)	-0.785 (-0.453)
$\sigma^2 = \sigma_v^2 + \sigma_u^2$	2.311 (0.986)	1.3969414*** (3.98208)	0.254*** (9.846)
$\gamma = \sigma_v^2 / \sigma^2$	0.999*** (1292.201)	0.9987426*** (1646.483)	0.994*** (353.468)
μ	-1.961 (-0.623)		
H	0.001 (0.956)		
Log likelihood	901.151	-287.998	-122.431
Number of observations	611	611	598
Number of regions	47	47	46
Number of years	13	13	13

Note) t-values are in parenthesis. The statistical significance at the one, five, and ten percent levels are indicated by ***, **, * respectively.

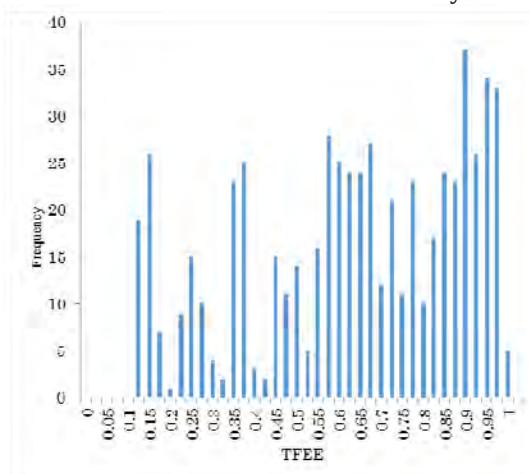
Table 4 Correlation coefficients among the TFEEs

Variable	TFEE without industry share	TFEE with broad industry share	TFEE with detail industry share
TFEE without industry share	1.000		
TFEE with broad industry share	0.931	1.000	
TFEE with detail industry share	0.937	0.999	1.000

(1) Distribution of TFEE scores without industry share variables



(2) Distribution of TFEE scores with broad industry share variables



(3) Distribution of TFEE scores with detail industry share variables

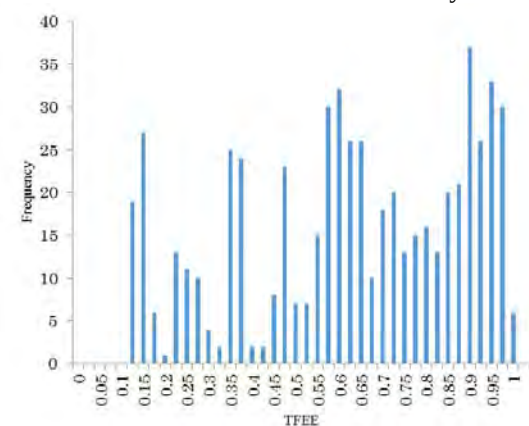


Figure 1 Distributions of TFEE scores

THE DEVELOPMENT EVOLUTION OF CHINA'S BUILDING ENERGY EFFICIENCY AND ITS INFORMATIZATION RELATED POLICIES, LAWS AND REGULATIONS, TECHNICAL GUIDELINES AND STANDARDS

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ABSTRACT

This paper introduces the general situation of energy utilization in China and points out the two prominent contradictions about it, which are the low utilization efficiency and high consumption of resources and a big gap in unit resource benefit compared with the international advanced level. From the macroscopic and microscopic angles this paper deeply analyses the background and situation in the present stage of China's building energy efficiency, also describes not only the situation of energy consumption belongs to different function buildings but also the energy consumption during the operation of buildings, and then several technologies and engineering applications related to building energy efficiency are summarized. Further, through analyzing China's building energy efficiency and its informatization related policies, laws and regulations, technical guidelines and standards during the period of the "10th five-year plan", the "11th five-year plan" and the "12th five-year plan", there are six main aspects in them, such as building energy efficiency planning and management, building energy efficiency design and construction, building energy management and its informatization, building energy efficiency reconstruction, the utilization of renewable energy, green building. This paper takes the form of radar chart to intuitively show the distribution of them in order to find out the development track and the development direction of the future in the process of urbanization of our country.

Keywords: Building energy efficiency and its informatization; Policy; Laws and regulations; Technical guidelines and standards; Development track.

1. INTRODUCTION

China is a major country refer to production and consumption of energy, energy production is only after the USA and Russia, has 1.00249 trillion tons coal reserves and 89.3 billion tons recoverable reserves, while basic energy consumption accounts for the total world consumption one of ten, ranking the world's second. Currently, China is in the phase of urbanization and industrialization, the sustained and rapid development of economy led to the sharp rise in energy consumption. Although we have 9 % of the world's GDP, energy consumption accounts for 19% of our GDP, keeping the average annual growth of 6.6%. Then, there are two prominent contradictions which are the low utilization efficiency and high consumption of resources and a big gap in unit resource benefit compared with the international advanced level.

In 2010, China's building energy total consumption was 677 million tce, accounting for 20.9% of national total energy consumption. From 1996 to 2010, the construction area keeps rapid growth and building energy consumption intensity was in the gradual rise.

As the two factors, total building energy consumption continues to increase, and presents the following characteristics. Primarily, although heating energy consumption intensity of northern town is bigger, keep decline in recent years, which shows the effect of energy saving work. Secondly, except the northern town heating consumption, almost all kinds of energy consumption intensity show a trend of sustained growth, while public building consumption is the fastest, so that it has been the focus of building energy efficiency. Thirdly, along with the increase of unprecedented height the amount of rural residential energy consumption goods, biomass energy using continues to rapidly reduce. China's urban population will increase to 900 million by 2020; urban construction area is likely to reach more than 30 billion square meters. If maintain current levels, the urban building energy consumption also will reach nearly 700 million tce, however coupled with the countryside, the number would be more than 900 million tce, which accounts for 20%~25% of total energy consumption. Under the severe situation, building energy efficiency has been promoted to an unprecedented height.

Abbreviation

BEEI	Building Energy Efficiency and its Informatization
MOHURD	Ministry of Housing and Urban-Rural Development
NDRC	National Development and Reform Commission
Symbols	
tce	Ton of Standard Coal Equivalent

In building energy efficiency work, by adopting appropriate technologies for different function buildings to dig up the potential, reduce the total building energy consumption and effectively relieve the contradiction between supply and demand of energy, at the same time, constantly improve people's living environment comfort. These technologies include building envelope heat preservation and heat insulation, heating energy saving, building energy conversion and energy supply, heat pump, large public building energy efficiency management, renewable energy. With the constant improvement of the level of science and technology, more and more efficient building energy efficiency technologies will appear constantly and be widely used.

To explore China's BEEI related policies, laws and regulations, technical guidelines and standards development evolution, firstly we need to sort out the definition and relationship among policies, laws and regulations, technical guidelines and standards. Policy is a reflection of the government will and a process concept, which the government takes a series of operational activities to achieve certain goals, also is dynamic and related to the history of the past and the future. Law is a general binding force social norm established or approved by the state and ensures implementation by national force to standardize the parties' right and obligations. General laws include laws, the interpretation of the law and regulatory documents that the administrative organ established for law enforcement. Technical guidelines and standards are the uniform rule to repetitive things, based on the comprehensive of science, technology and practical experience. As the rules abide by common people, it is approved by the competent authorities, in a specific form to release. Policies must be formulated within the framework of laws and regulations, and if some universal policies after long-time application are proved they are reasonable, can upgrade to laws and regulations. Technical guidelines and standards also are the summary of the experience and regulations, guide our formulation of policies and laws and regulations, while must be restricted by laws and regulations.

2. THE ANALYSIS TO CHINA'S BEEI RELATED POLICIES, LAWS AND REGULATIONS DEVELOPMENT EVOLUTION

From 2000 to 2010, China building energy efficiency developed by leaps and bounds. During this period, the promulgation and implementation of large numbers of important building energy efficiency policies, they have played a big role in promoting the development of China's energy efficiency technology

and smooth realization of energy conservation and emission reduction target. Also the policies developed from overall plan to the various specific aspects, from macro to micro, and closely followed the requirements of the Times. By searching, screening and analyzing a large number of policies, laws and regulations from the beginning of 2000 to the end of 2012, which are issued by MOHURD, NDRC and General Office of the State Council, we divide China's BEEI related policies into six main aspects: building energy efficiency planning and management, building energy efficiency design and construction, building energy management and its informatization, building energy efficiency reconstruction, the utilization of renewable energy, green building.

This paper takes use of the radar chart to indicate the development evolution of China's BEEI related policies during the period of "10th five-year plan", "11th five-year plan" and "12th five-year plan". According to the previously mentioned six directions of policies, we begin the quality and quantity analysis.

Six Angles of the radar chart represent six different directions, also different line-types and line-colors represent the cumulative number in different years. When comparing the two different color lines, the policies increment to different direction can be seen, so we can predict the future development trend.

Figure 1 show during the "10th five-year plan" the policies focus on building energy efficiency planning and management and building energy efficiency design and construction. This is because the urbanization process in China was still in its infancy, numerous new buildings appeared, and meanwhile building energy efficiency reconstruction work around the northern heating began to expand. However, building energy management and its informatization in the process of building operation has not been mentioned. The concepts such as Green building and the utilization of renewable energy did not appear.

Figure 2 presents not only the change of policies from the 10th five-year plan to the 11th five-year plan, but also points out the policies distribution about "11th five-year plan". During the period of the "11th five-year plan" building energy management and its informatization related policies grow rapidly. Building energy efficiency began to pay close attention to the energy consumption supervision and management in the building operation phase. The requirements of building energy efficiency planning and management were constantly improved as the expansion of building energy efficiency work. Both the new concepts such as green building and the utilization of renewable energy have not been mentioned, but they represent the future development direction of building energy. The existing

building energy efficiency reconstruction work was carried out steadily.

Figure 3 shows the track of BEEI policies in the "10th and 11th five-year plan" as well as the development trend of the BEEI policies in the "12th five-year plan". Our country gradually introduced a large number of building energy efficiency policies, which suggested that the government's emphasis on building energy efficiency work. Policy developed from the local to all-round, from the basic direction to the basic direction and emerging direction, the future new

buildings design and construction in accordance with the standards of green building and more renewable energy used in building areas. Reconstruction of existing building energy efficiency work in both residential building and public building will begin at the same time. Building energy management and informatization will be constantly developed with the booming of the Internet of things, also the application of information technology will be the highlight of building energy efficiency.

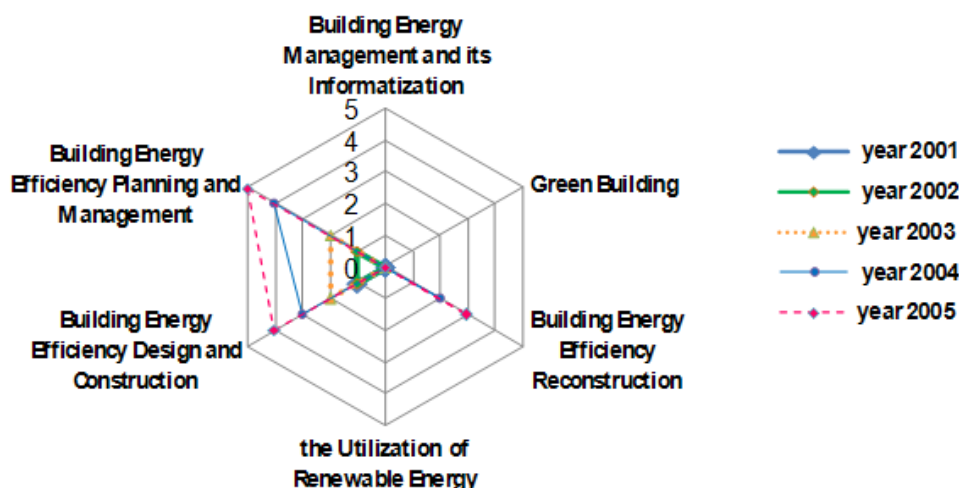


Figure 1 The distribution of BEEI related policies during the period of the "10th five-year plan"

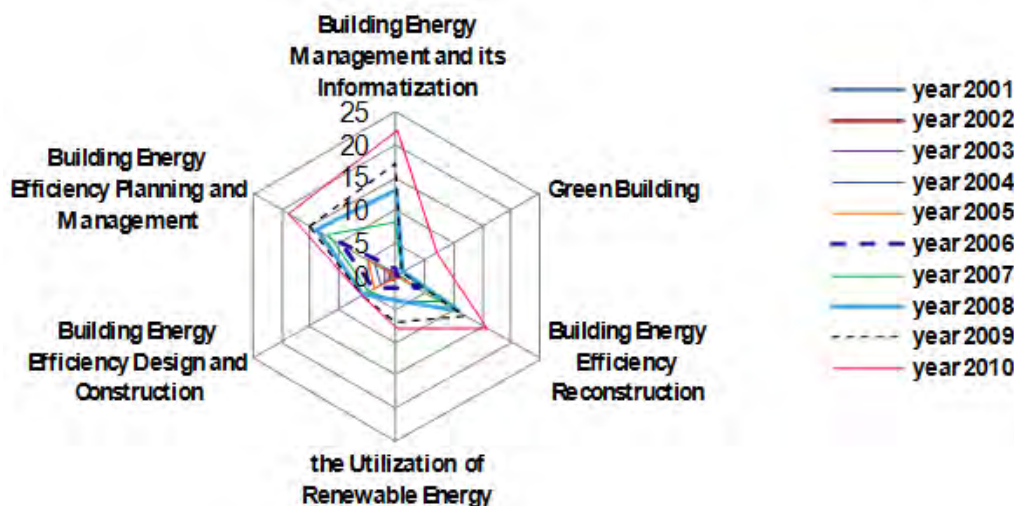


Figure 2 The distribution of BEEI related policies during the period of the "10th and 11th five-year plan"

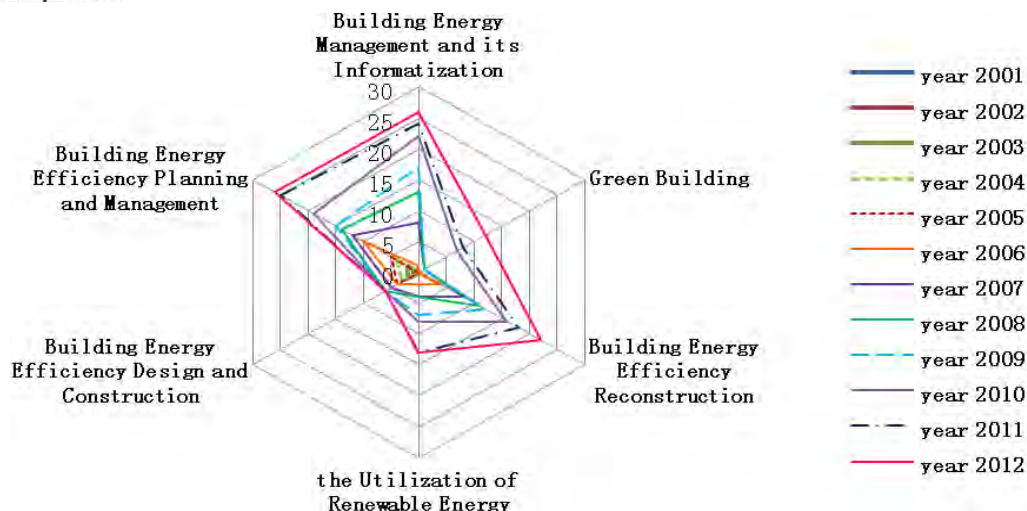


Figure 3 The distribution of BEEI related policies during the period of the “10th, 11th and 12th five-year plan”

3. THE ANALYSIS TO CHINA'S BEEI RELATED TECHNICAL GUIDELINES AND STANDARDS

According to the standard applicable scope, Chinese standards are divided into four levels: national standards, industry standards, local standards and enterprise standards. For example, code GB represents mandatory national standards, while code GB/T represents voluntary national standards. After searching, screening and analyzing the relevant standards have been issued from year 2000 to year 2012 and the upcoming standards, also in order to keep with the continuity of the policy analysis to China's BEEI, technical guidelines and standards based on the content can be divided into five main aspects, such as building energy efficiency examination and evaluation, building energy efficiency design and construction, building

energy management and its informatization, green building, the utilization of renewable energy.

Figure 4 presents the distribution of technical guidelines and standards during the "10th five-year plan", the standards referred to design and construction of building energy efficiency was significantly prominent, while building energy efficiency examination and evaluation took second place. The reason are the increase of new buildings with the beginning of China's urbanization, and then come up with some mandatory standards and requirements for building energy efficiency design and construction, the next need examination and evaluation to building energy efficiency. Further, these standards could promote the implementation of building energy efficiency work in our country. Building energy management and its information and green buildings due to specific applications nonexistent and relevant experience insufficient so that related standards have not been formed.

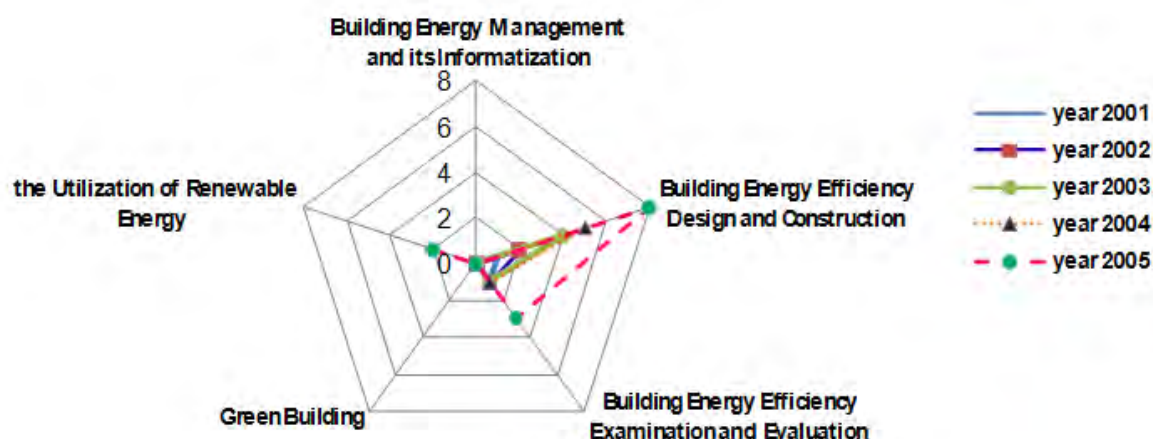


Figure 4 The distribution of BEEI related technical guidelines and standards during the period of the “10th five-year plan”

As is shown in figure 5, during the "11th five-year plan" building energy efficiency design and construction were still leading the development of

standards, this is because the country attached great importance to the building energy efficiency work. Every year there are a lot of new buildings appearing in

our country with urban development, which requires high, strict, and suitable for different climate zones standards, and these standards are intact and purposeful. Building energy management and its informatization related technical guidelines firstly appeared in 2008 and 2009, which indicated that the country began to focus on the operation phase energy consumption of building, meanwhile many information technologies were used such as Internet of things to accomplish online

monitoring, evaluation, diagnosis. The construction of the state organ office buildings and large public building energy consumption monitoring system and the construction of campus building energy efficiency supervision system have been promoted and implemented. There were several applications about renewable energy utilization and green building, but practical experience was not yet mature so that standards and technical guidelines are less.

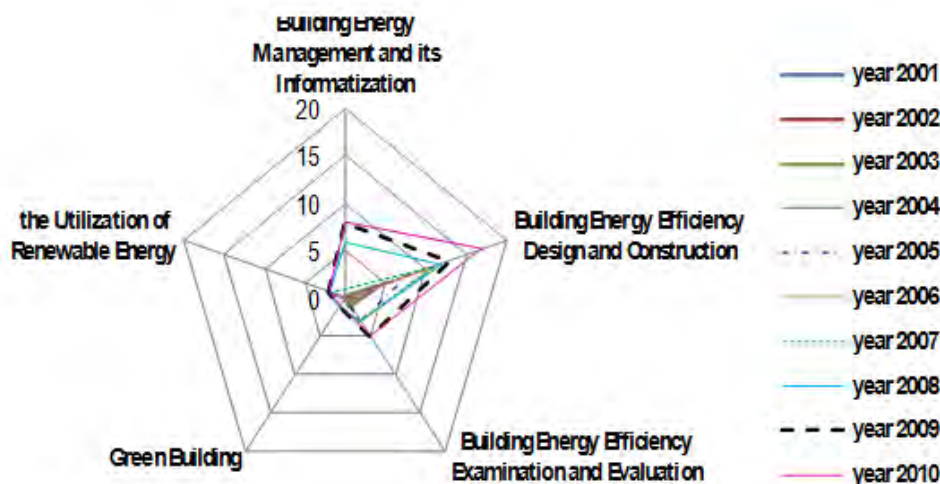


Figure 5 The distribution of BEEI related technical guidelines and standards during the period of the “10th and 11th five-year plan”

Figure 6 shows that during the period of the “10th, 11th and 12th five-year plan”, building energy efficiency design and construction related technical guidelines and standards have been maintained a stable growth momentum. It is due to energy efficiency requirements for new buildings are higher and higher. Buildings energy management and its information related technical guidelines and standards will also

continue to increase with the arrival of the age of the Internet of things. After a period of exploration, some practical experience and rules about renewable energy utilization have been summed up, then relevant standards will be burgeoning. Green building, this emerging concept in the field of construction will be a high frequency vocabulary, and related standards must be the hot spot of next phase.

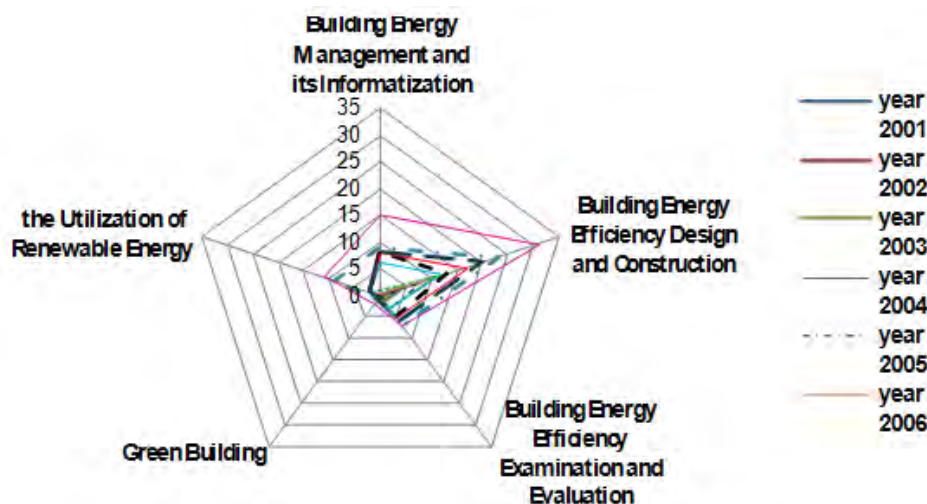


Figure 6 The distribution of BEEI related technical guidelines and standards during the period of the “10th, 11th and 12th five-year plan”

4. CONCLUSION

Through the above analysis, we can clearly realize the track of BEEI related policies, laws and regulations, technical guidelines and standards, it has experienced from the northern town to different climate zones, from residential building to public building, for existing buildings from building envelope modification to the operation and management of building energy consumption, for new buildings from high energy consumption to green building, for building energy utilization from conventional energy to the combination of conventional and renewable energy. Along the known track, we can predict the development direction of the BEEI related policies, laws and regulations, technical guidelines and standards in the future: Green building is the development direction of new buildings, and new buildings' design and construction of energy efficiency requirements will be higher; renewable energy will be a hot direction of building energy utilization; the existing building energy efficiency reconstruction work will be in full swing; building energy management will be widely applied in all kinds of high energy consumption buildings; with the advent of the Internet of things, information technology will run throughout the entire process of operation and management of building energy.

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THE INTEGRATED ENERGY TAXATION FOR GREEN FISCAL REFORM: A THREE-PART TARIFF APPROACH

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ABSTRACT

Various taxes and fees on energy products are collected in many countries for different purposes. In response to the critical challenges of climate change, energy crisis and green tax reform, the Energy Tax Act was first proposed in 2006 that allows the Ministry of Finance to collect energy tax from suppliers and importers of energy products. Since the first draft was not passed by the Congress, there were in the last seven years another seven drafts submitted to the Congress for approval. All versions are characterized by different objectives and tax rates. In particular, diversified views are expressed on how the tax revenue should be disposed in order to implement green fiscal reform. So far none of the drafts has been passed by the Congress and the debate continues. The debate is complicated in terms of policy harmonization and integration not only by the Greenhouse Gases Reduction Act initiated in 2005, that provides with legal foundation for emission trading system (ETS), but also by the decision made by EPA in May of 2012 that greenhouse gases are treated as air pollutants and, therefore, subject to carbon emission fee (or carbon tax) according to the Air Pollution Control Act. The key question is whether these three policy instruments (i.e., energy tax, carbon tax, and ETS) are competing or complementary?

In light of the European Commission's proposal, we critically review the legislation objectives of all versions of the Energy Tax and examine the linkage between tax rates and objectives, set up a multiple principals-agents model that consists of duopolistic energy suppliers, competitive consumers and four government agencies in charge of energy-related taxes and fees, and construct an alternative institutional regime as benchmark, in which all regulated firms, including the duopolistic energy suppliers, engage in permit trading in two permit markets of different nature, a competitive emission trading market and an auction permit market.

To integrate carbon tax and energy tax, we argue that the legislative objectives of the energy tax should be simplified and confined to the following three, namely, carbon reduction, energy efficiency improvement, and fiscal reform. Accordingly, a three-part tariff scheme is derived for energy tax

Keywords: Energy tax; Emission charge; Carbon tax; Green fiscal reform; Three-part tariff; Energy efficiency; Carbon reduction.

1. INTRODUCTION

Both environmental tax and energy tax of various types are playing important roles in many countries for different purposes such as revenue raising, internalizing external cost, implementing green fiscal reform, etc. In Chinese Taipei, for example, the Environmental Protection Administration (EPA) is authorized to collect Air Pollution Control Fee (APCF) and Water Pollution Control Fee, while the Bureau of Energy to collect Energy R&D Fund and Petroleum Fund. In addition, the Ministry of Transportation is responsible for collecting Vehicle Fuel Fee, while Ministry of Finance collecting such conventional taxes as excise tax, business tax, business income tax, and import tariff on energy

products. The revenue from every tax (or fee) will be disposed according to relevant regulations. Nevertheless, green fiscal reform has never become one of the purposes in Chinese Taipei until recently when the Energy Tax Act (ETA) was first initiated in 2006 as a response to the critical challenges of climate change, energy crisis and double-dividends hypothesis.

According to the 2006 draft of ETA, the Ministry of Finance will be authorized to collect energy tax from suppliers and importers of energy products, including gasoline, diesel, natural gas, aviation fuel oil, LPG, coal, kerosene oil, etc. Since the first draft failed to pass by the Congress, there were in the last seven years another seven versions submitted to the Congress for approval consideration. All versions are characterized by different legislative objectives, tax rates and how the

tax revenue should be disposed in order to implement green fiscal reform.

Despite the energy tax has been endorsed by consensus achieved in several national energy conferences, none has been passed by the Congress and debate and communication continue. Controversial questions are: (1) Should the energy tax be introduced to replace the prevailing excise tax on energy products since nothing is changed except that the tax rate will increase over time? (2) How is the tax rate determined? (3) How to allocate the tax revenue among alternative uses? (4) Whether the green fiscal reform will realize double dividends? (5) To what extent will the new tax affect the economy and industrial competitiveness?

The questions become more complicated not only because EPA initiated the Greenhouse Gases Reduction Act (GHGRA) in 2005 to provide with legal foundation for emission cap and trade system (ETS), but also because EPA announced in May of 2012 that greenhouse gases will be considered as air pollutants and, therefore, subject to APCF according to the Air Pollution Control Act (APCA). Consequently, the government is faced three distinct policy instruments to choose from, namely, energy tax, APCF for GHG emission, and ETS. As the legislations of energy tax and ETS are not yet completed while APCF for GHG emission is legally enforceable under APCA, the decision problem now is whether these three policy instruments are competing or complementary? More specifically, how could we make them harmonize with each other?

In light of the European Commission's proposal of revising the 2003 Energy Taxation Directive, we propose in this paper an innovative rule to decompose the energy tax rate into three components, with each component to meet the legislative objectives of the tax with respect to carbon reduction, energy saving, and green fiscal reform. To accomplish this idea, a theoretically sounded model is warranted to identify a socially optimal tax mix.

For this purpose, we critically review the legislation objectives of the Energy Tax Act, the linkage between tax rates and objectives, and the drawbacks underlying the EU's decomposition approach. To derive a socially optimal tax mix that consists of income tax and energy tax, we, taking into account the unique properties of the energy markets, set up a principal-agent model that describes the decision behavior of energy suppliers, competitive consumers and government agencies.

The paper is organized as follows. In section 2 we briefly review the EU's renovation on the 2003 Energy Taxation Directive, and identify the drawbacks underlying the EU's decomposition approach. The energy tax under contemplation in Chinese Taipei is also critically reviewed regarding the legislation objectives, as well as the linkage between tax rates and objectives. The validity of the double-dividends hypothesis is also discussed, with particular emphasis on the factors contributing to the existence of the dividends. In section 3 we present a comprehensive principal-agent model that allows for the determination

of a socially optimal tax mix. The institutional arrangement of the model is described in three dimensions, i.e., characteristics of the energy market, institutional arrangement, and decision behavior of energy consumer, supplier and the authorities. Accordingly, a three-part tariff is designed for the energy tax rate. Finally, simulation results are presented and policy implications addressed in section 4, followed by some concluding remarks.

2. THE EXSISTING PROBLEMS

2.1 EU's renovation on energy taxation

On 10 March 2010, the European Commission announced a plan of an EU-wide minimum tax on carbon. The Commission is looking at revising the 2003 Energy Taxation Directive, to bring energy taxation into line with the EU's 2020 emission reduction objectives (reducing greenhouse gas emissions by 20% by the year 2020, when compared to 1990 levels).

On 13 April 2011 the European Commission presented a proposal to renovate the rules on taxation of energy products in the EU. The new rules aim to restructure the way energy products are taxed to remove current imbalances and take into account both their CO₂ emissions and energy content. That is, existing energy taxes would be split into two components that, taken together, would determine the overall rate at which an energy product is taxed. This approach is hereafter referred to as the two-part tariff. That is, the energy tax rate is decomposed into two components as follows:

$$\tau_i(t) = c_i(t) + e_i(t), \quad (1)$$

where $c_i(t)$ and $e_i(t)$ represent, respectively, the carbon-content tariff and energy-content tariff.

With this measure, the Commission wants to improve energy efficiency and promote more environmentally friendly consumption as well as to avoid competitive distortion within the internal market by removing unjustified subsidies to certain energy products (such as diesel and coal), giving a consistent treatment to all sources of energy. It will also avoid distortions of competition in the EU market

The new minimum taxes would be introduced in stages until 2018. Tax for diesel and kerosene would increase to a higher per-volume rate than petrol (CO₂ emissions and energy content are higher per volume in those). Minimum taxes on LPG and natural gas would increase in a factor of four, and biofuels would be also taxed though there won't be a CO₂ tax for sustainable fuels because its emission factor is zero.

Fuels used to generate electricity would be mostly exempted from CO₂ tax because they are subject to ETS. But CO₂ tax would be applied in small installations exempted from ETS.

Electricity produced by nuclear energy would be now taxed as all other electricity, with no CO₂ tax because it does not produce CO₂.

On 20 April 2012 the European Parliament, with a majority of votes including the EU conservatives, has

voted against the draft Energy Taxation Directive stating that this is not a good moment to increment energy taxes, since it is a time of economic austerity and high fuel costs.

The vote of the European Parliament is not binding; it's now up to the EU countries to decide if they will follow the Parliament or the proposal from the EU Commission. As an example, the Climate Change Levy adopted by UK in 2001 is more like the energy-content component of energy tax, while the CRC Energy Efficiency Scheme Charge enforced in 2012 like the carbon-content component.

2.2 Energy tax in Chinese Taipei

The legislation objectives of energy tax identified in all drafts are summarized as follows:

- to reduce CO₂ emission,
- to promote energy saving,
- to improve energy efficiency,
- to develop renewable energy,
- to internalize external costs of energy use, and
- to implement green fiscal reform.

In fact, most of the energy products to be taxed under the Energy Tax Act are already subject to excise tax that, however, will be replaced by energy tax if the Energy Tax Act is approved. In addition, APCF, according to the Air Pollution Control Act, is a readily available tool to internalize externality arising from air pollutants emission, including GHG emissions. As such, the policy motivation of imposing a new tax, while discarding the prevailing policy tools, is obviously not well justified, particularly when the new tax has been experiencing certain barrier.

Despite the existing energy tax drafts are rather diversified in terms of tax rate and revenue disposal, the proposed tax rate ($\tau_i(t)$) is commonly characterized by a constant increasing rate over time up to certain years (see Equation 2).

$$\begin{aligned} \tau_i(t+1) &= \tau_i(t) + c_i t \quad \forall 0 \leq t \leq \hat{t} - 1, \\ &= \tau_i(t) \quad \forall t \geq \hat{t}, \end{aligned} \quad (2a)$$

$$\tau_i(0) = \bar{\delta}_i(0) + c_i, \quad (2b)$$

where $\bar{\delta}_i(0)$ and c_i represent, respectively, the prevailing excise tax rate of the i -th energy product and the annual increment of the tax rate.

Note that different values of c_i have been specified in all drafts without clear explanation. The linkage of c_i to the legislation objectives remains unjustified. Above all, the above tax rate is quite different from the two-part tariff and carries much less policy implication.

There are several practical problems associated with the two-part tariff approach as well. Firstly, when this approach is applied to keep the new energy tax identical to the existing excise tax rate, $e_i(t)$ will be negative for most of the energy products. The main reason lies in the fact that the existing excise tax is not based on the carbon content of the energy product. Secondly, there is lack of theoretical underpinning

about how the imposition of $e_i(t)$, even positive in value, would ensure energy saving or energy efficiency improvement. Thirdly, the overall rate of the energy tax ($\tau_i(t)$) is far from the socially optimal level. As a result, the double dividends from green fiscal reform would be less feasible. More critically, the value of the carbon-content component depends on carbon price that is exogenously determined. Finally, one can hardly achieve multiple objectives as mentioned above simply by relying on a single policy instrument such as energy tax.

We argue that the legislative objectives of the energy tax should be confined to the following three, namely, carbon reduction, energy efficiency improvement, and fiscal reform. Accordingly, a three-part tariff scheme could be derived for energy tax rate as follows:

$$\tau_i^*(t) = c_i(t) + e_i(t) + f_i(t),$$

where $\tau_i^*(t)$ is the socially optimal overall rate; the third component is the residual required to make the overall rate socially optimal (i.e.,

$$f_i(t) = \tau_i^*(t) - c_i(t) - e_i(t).$$

Following the EU's two-part tariff approach, it is straightforward to determine the value of c_i and the second component. It turns out that the determination of the third component requires a socially optimal level of energy tax.

2.3 Double-dividends hypothesis and green fiscal reform

Various factors had been identified in the literature addressing the existence of the dividends of environmental taxes through proper revenue disposal. Major determinants include product market structure (Barnett, 1980), tax-interaction effect (Bovenberg and Mooij, 1994a, 1994b; Goulder, 1995; Parry, 1995; Kahn and Farmer, 1999), tax rates (Parry, 1995; Bovenberg and Goulder, 1996; Fullerton, 1997), tax base (Bovenberg and Mooij, 1994b; Goulder, 1995b; Parry, 1995), consumer's preference (Kahn and Farmer, 1999; Schwartz and Repetto, 2000; Williams, 2002, 2003; Pang and Shaw, 2007; Chung-Huang Huang and Wan-Ling Chou, 2012), labor market imperfection (Carraro, Galeotti and Gallo, 1996; Bosello and Carraro, 2001), role perception about the environmental tax (Goulder, Parry and Burtraw, 1997; Bovenberg, 1999; Parry and Bento, 2000; Conrad and Löschel, 2005; Bento and Jacobsen, 2007), the properties of the health production function (Koç, 2007), and intertemporal welfare concern (Chiroleu-Assouline and Fodha, 2005, 2006). Nevertheless, much less analysis on the existence of double dividends associated with energy tax could be found, although its economic impact evaluation is relatively more abundant.

Note that a socially optimal energy tax rate is just a necessary condition, but not sufficient, for realizing the double dividends. Therefore, we provide a comprehensive model that would allow us to derive a

socially optimal energy tax rate, together with the prevailing taxes.

3. THE MODEL

3.1 Properties of the energy markets

In Chinese Taipei, the domestic markets of secondary energy (such as energy products and electricity) could be classified into energy product market and electricity market, according to the supply chain of energy (see Figure 1).

The energy product market is characterized by a duopoly, where there are only two firms producing all fossil energy products, one state-owned and the other private. The price of the energy products produced by the state-owned company (namely, Chinese Taipei CPC) is partially regulated by the authorities and its product outlet is domestic-oriented. In contrast, the private company, competing with the state-owned company by a price-competition game, is allowed to export without restriction, except under certain circumstances.

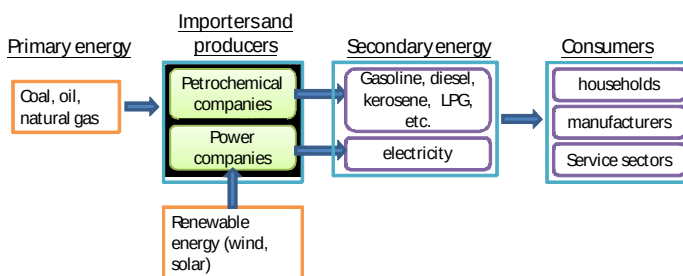


Figure 1 Supply chain of energy

As for the electricity market, it consists of a monopolistic power company (i.e., Chinese Taipei Power Company, Taipower) and several small independent power plants (IPP) that sell electricity generated from fossil energy as well as renewables to Taipower at feed-in tariffs. Taipower, also a state-owned enterprise, is regulated in pricing rule that is rather peculiar and far from profit maximization. The pricing scheme is currently under review in the hope of more reasonably reflecting fuel cost and energy mix used for power generation.

3.2 Institutional regime

The energy-product-related taxes and fees are rather complicated in Chinese Taipei since the payers are tax-dependent and revenues administered by different regulatory authorities (see Figure 2).

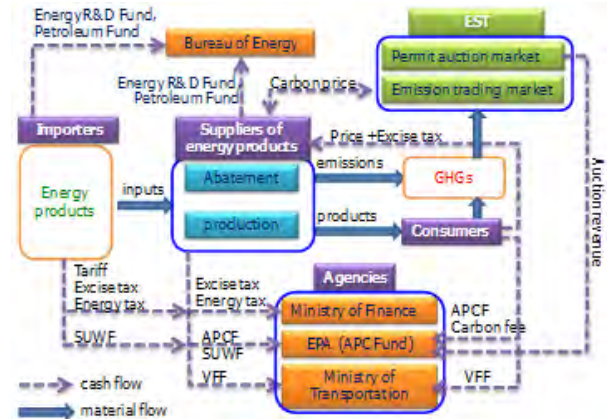


Figure 2 Institutional regime of environmental and energy tax

In Figure 2, energy tax and the carbon emission fee are new to the prevailing tax system. Energy tax is designed to replace the existing energy-product-related excise tax, while carbon emission fee to play the role of carbon tax. ETS remains uncertain since GHGRA still under debate for legislation approval.

The emerging issue is how to reconcile the energy tax and the carbon tax. Is there any way to integrate the two so that the original legislation objectives could be fulfilled? Here the three-part tariff approach does it all.

3.3 Decision behaviors

The Chinese Taipei General Equilibrium Model (TAIGEM-III), developed for the Council of Economic Planning and Development, is used in this paper for policy simulation. TAIGEM-III is a recursive type, single country CGE model with 52 sectors, 72 commodities (including ten taxable energy products) and 10 electricity generation technologies.

The model is featured with three market categories: primary input markets (including primary energy, labor, land, capital, etc.), commodity markets (including secondary energies and other commodities), and trading markets (including commodity trading and emission trading). A two-tier labor market is specified to properly capture the driving forces of unemployment in Chinese-Taipei. Duopoly is specified for the secondary energy product markets since there are only two suppliers, while electricity market is monopolized with several small IPPs and government intervention. The emission trading market determines the carbon price that provides us with a benchmark to estimate the carbon-content tax.

A social planning problem is designed here to solve for the socially optimal tax mix $((t_L^*, t_X^*))$. We modifies the health production function of the household by incorporating both medical treatment expenditure and illness prevention expenditure and tax deductibility, and redefines the marginal social damage of consuming dirty goods by incorporating both the disutility of illness and utility of leisure time.

Following the same decomposition approach as shown in literature, a new source of dividends with

environmental tax on dirty goods, named “prevention-based tax-interaction effect” (IEA), is derived, that, however, is negative in sign and, therefore, weakens overall second dividend. The mitigation-based tax-interaction effect (IEM) might be negative as well, provided that the medical treatment expenditure being normal. In sum, whether or not the medical treatment expenditure and illness-prevention expenditure are normal goods play crucial role in signing IEA as well as IEM. Our model contends that both are more likely to be negative, and thus tend to weaken the overall dividends of the energy tax.

4. RESULTS

The findings are summarized as follows:

1. With the energy tax, environmental quality is improved, confirming the first dividend.

2. The difference of welfare between the base case and the scenarios is positive and increases initially with energy tax, but begins to decline and eventually turns out to be negative when the tax rate is sufficiently high. Figure 3 indicates that there exists an optimal tax bundle (t_L^*, t_X^*) such that the welfare is maximized in the scenarios. This implies that the validity of the double dividends hypothesis to some extent depends on the tax bundle selected by the authorities.

3. Note that the marginal utility of income (λ) decreases with energy tax. Nevertheless, the conventional dividend decomposition approach usually assumes constant for λ . Consequently, this approach may generate estimate bias for the dividend components.

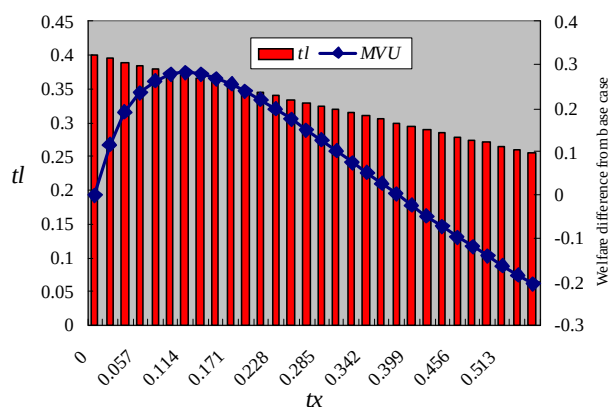


Figure 3 Effects of environmental tax on welfare: difference from base case

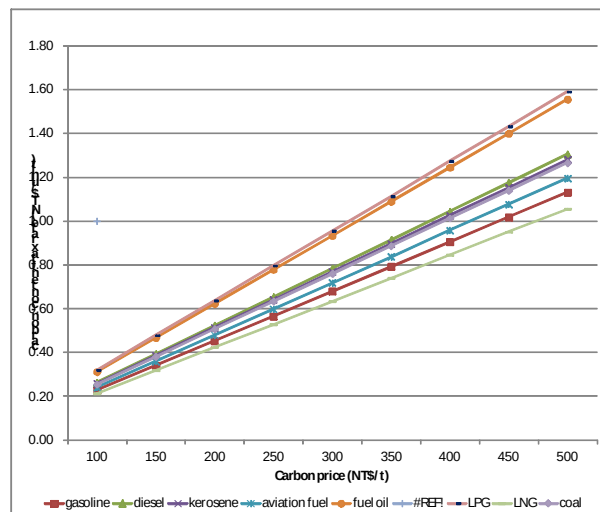


Figure 4 Carbon-content tax rate of energy products

5. CONCLUSION

Both medical treatment expenditure and illness prevention expenditure are pervasively observed in the real world. Their health effects are different to some extent and so is their income tax deductibility. Failure to incorporate both expenditures in addressing the dividends of environmental tax tends to be incomplete. The marginal social damage of consuming dirty goods was neither accurately defined nor correctly measured in literature since it ignores the fact that the consumer may value time for alternative uses differently. Assuming identical value for the time of alternative uses (work, leisure, health care, etc.) may also end up with biased estimates of the second dividend.

Simulation results indicate that an optimal tax bundle (t_L^*, t_X^*) may exist such that the welfare is maximized in the scenarios. This implies that the validity of the double dividends hypothesis to some extent depends on the tax bundle selected by the authorities.

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ENERGY EFFICIENCY AND ECONOMIC PERFORMANCE:A CASE STUDY ON EAST ASIA SUMMIT NATIONS

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ABSTRACT

In the current trend of earth's sustainable development, the role of energy efficiency is becoming more and more important. We employ the extended Cobb-Douglas production function to examine the impact of energy efficiency on economic performance among East Asia Summit Nations (EASN). The time span of our dataset is from 1997 to 2009. We conclude that under the optimal amount of energy usage, China can create an absolute improvement in GDP, whereas there is an absolute regression in GDP for India, South Korea, Thailand, and Vietnam. The condition that an improvement in energy efficiency creates an absolute GDP increase is a negative energy elasticity coefficient.

Keywords: sustainable development; energy efficiency; economic growth; EASN; ASEAN.

1. INTRODUCTION

Over the past two decades, some of the world's best economic performances have taken place in Asia where economies, including China, the Asian Tigers (Hong Kong, Singapore, South Korea, and Chinese Taipei), and the Association of Southeast Asian Nations (ASEAN), have exhibited fast economic growth and development. According to the population and vital statistics conducted by the United Nations, the world's largest population center is in Asia. Economic growth and environmental protection in such a densely populated area have always been noticeable.

The geopolitical and economic organization of ASEAN was established on August 8, 1967. The ten ASEAN members are Indonesia, Malaysia, Philippines, Singapore, and Thailand (as the five Founders) and Brunei Darussalam, Vietnam, Lao PDR, Myanmar, and Cambodia. The organization's main aim and purpose are to improve economic growth, social progress, and cultural development in the region. To develop friendly and mutually beneficial relations, ASEAN's external relations encompass Australia, China, India, Japan, South Korea, and New Zealand.

In order to improve the economic growth and development of ASEAN by disseminating the concept of energy efficiency, the ASEAN Center for Energy (ACE) was set up on January 1, 1999. Energy efficiency means that less energy is able to provide the same level of energy service or the same amount of energy provides more output. In July 2010 the 9th Regional Energy Policy and Planning-Sub Sector Network was held in Singapore, with the task set forth upon ACE to develop the ASEAN Energy Review and ASEAN Energy Indicators.

Two commonly used energy indicators estimating energy efficiency for ASEAN members are energy elasticity and energy intensity. Karki et al. (2005) pointed out that energy intensity in these countries has increased from 1983 to 1997. Balce et al. (2001) noted that high energy intensity is caused by poor energy efficiency. Hence, an effective way to decrease energy intensity and to save energy consumption is by increasing energy efficiency not only as an incentive for economic growth, but also as a direct path toward sustainable energy.

Keong (2005) suggested a sustainable energy policy through demand-side management and supply-side management. Demand-side management emphasizes energy efficiency, whereas supply-side management targets the deployment of other efficiency-driven strategies, including improvements in plant generation, transmission and distribution systems, and fuel substitution. According to the report by Lindula et al. (2007), Lao PDR and Vietnam put forth less effort than other ASEAN members in the demand-side management of energy efficiency.

The latest policy of Energy Efficiency and Conservation (EE&C) has already been set up in the ASEAN region under the 2009 ASEAN Plan of Action on Energy Cooperation (APAEC) 2010-2015. To reduce energy consumption without cutting the use of energy-consuming plant and equipment, energy efficiency means making better use of energy and achieving less energy intensity through the promotion of individual behavior and better working methods and manufacturing practices. The target of ASEAN is to lower energy intensity by at least 8% by 2015 based on 2005 levels. The ASEAN Energy Review and Energy Indicators aid the ASEAN members in formulating an appropriate energy policy. ACE estimates that energy

consumption in ASEAN will see a large increase from 280 million tons of oil equivalent (MTOE) in 2000 to 583 MTOE in 2020. Thus, it is important to achieve energy efficiency.

Energy no doubt is an essential and crucial factor in the production process (Stern, 1993, 2000; Oh and Lee, 2004a, b; Ghali and El-Sakka, 2004; Beaudreau, 2005). Numerous studies have examined the relationship between energy consumption and economic growth. For example, papers examining the relationship between electricity consumption and GDP include Abosedra et al. (2009), Altinay and Karagol (2005), Chen et al. (2007), Fatai et al. (2004), Ferguson et al. (2000), Ghosh (2002), Ho and Siu (2007), Jumbe (2004), Morimoto and Hope (2004), Mozumder and Marathe (2007), Narayan and Smyth (2005), Shiu and Lam (2004), Thoma (2004), Wolde-Rufael (2006), Yang (2000), Yoo (2005a, b, 2006), and Yuan et al. (2007). Some studies also investigated the relationship between GDP and other forms of energy, such as fossil fuel, coal, petroleum, coke, gas, total energy, etc., including Cheng and Lai (1997), Lee and Chang (2005), Lee (2006), Lise and Montfort (2007), Morimoto and Hope (2004), Sharma (2010), Stern (2000), Wolde-Rufael (2004), and Yuan et al. (2008). Oh and Lee (2004a) and Lee and Chang (2008) employed a multivariate model with input factors of real capital stock, labor force, and energy consumption and an output factor of real GDP to investigate the relationship between energy consumption and real GDP. Recent papers investigating the relationship between energy consumption and GDP present a good connection with environmental issues, such as Hooi and Smyth (2010) who looked at the relationship among carbon dioxide emissions, electricity consumption, and economic growth for five ASEAN members. According to our best knowledge, no other previous paper has examined the relationship between energy efficiency and economic growth and discussed the condition that energy efficiency creates economic growth.

In order to investigate our concerns, the present research employs the extended Cobb-Douglas (C-D) production function with real GDP as the output factor and real capital stock, labor force, and energy usage as the input factors for 8 ASEAN members and 6 countries having external relations with ASEAN. These 8 ASEAN members and 6 countries having external relations are all members of East Asia Summit Nations (EASN). Hereafter, we combine these countries and use the term “EASN members” for simplification. Figure 1 presents the geographic locations of the 14 EASN members as follows.

The remaining framework of the paper is organized as follows. In the following section we introduce the methodology. Section 3 implements the empirical analysis and discusses findings. In the final section we provide some concluding remarks.

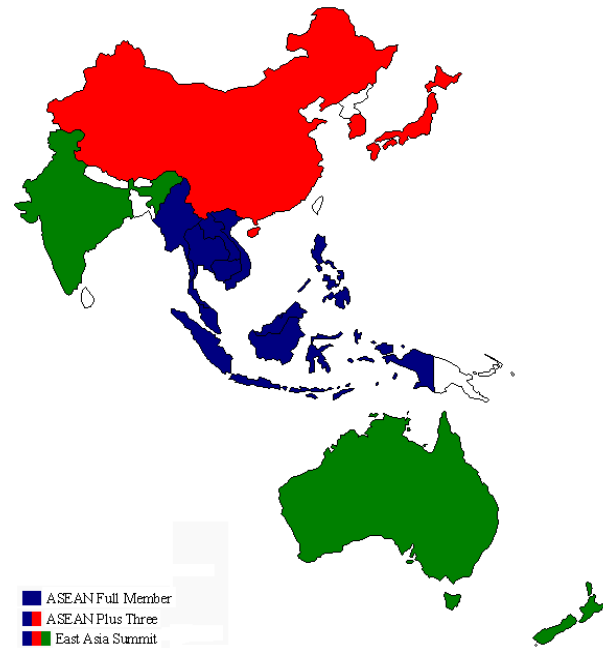


Figure 1 14 East Asia Summit Nations

2. METHOD

The data envelopment analysis (DEA) approach is a well-known mathematical linear programming technique to estimate the efficiencies of decision-making units (DMU). Hu and Wang (2006) used the DEA approach to calculate total factor energy efficiency (TFEE). In the TFEE estimation process, it is necessary to calculate the optimal amount of energy usage. Therefore, we also take up the TFEE concept to calculate the optimal amount of input factors.

According to the DEA model, if there are n DMUs and each DMU has s inputs and t outputs, then the efficiency score of the i th DMU can be estimated by the following linear programming problem:

$$\text{Min}_{\theta, \eta} \theta$$

$$\text{Subject to} \quad -y_i + Y\eta \geq 0,$$

$$\theta x_i - X\eta \geq 0, \quad (1)$$

$$I_n \eta = 1,$$

where θ is an efficiency score of the i th DMU; η is an $n \times 1$ vector of constants; x_i and y_i are $s \times 1$ and $t \times 1$ vectors of the input and output of the i th DMU, respectively; X and Y are $s \times n$ and $t \times n$ matrices of the input and output of all DMUs, respectively; I_n is an $n \times n$ identity matrix. Equation (1) is known as the input-oriented variable returns to scale (VRS) DEA model originally proposed by Banker et al. (1984).

Based on the input-oriented VRS DEA model, Hu and Wang (2006) established the TFEE index (E) as follows:

$$E_{i,t} = \frac{\text{Optimal amount of energy usage}_{i,t}}{\text{Actual amount of energy usage}_{i,t}} \quad (2)$$

which takes place in the i th DMU and in the t th year. The optimal amount of energy usage means that the energy input is minimized so that the DMU operates at the frontier position of energy efficiency.

We describe the optimal amount of energy usage as follows:

Optimal amount of energy usage = Actual amount of energy usage – Radical adjustment – Slack adjustment.

(3)

We obtain the radical adjustment and the slack adjustment by using DEAP 2.1, a DEA software. When the radical adjustment and the slack adjustment are equal to 0, it means that the optimal amount of energy usage is equal to the actual amount of energy usage. At this time, the TFEE index in Equation (2) is equal to 1, which implies energy usage efficiency reaches the optimum. If the radical adjustment or the slack adjustment is not equal to 0, then the optimal amount of energy usage should be smaller than the actual amount of energy usage. At this time, the TFEE index is smaller than 1, implying energy usage efficiency does not reach the optimum.

We take one output and two input factors as an example to draw Figure 2. Here, the horizontal axis and the vertical axis indicate that one unit of output needs how many units of x and e inputs. The variables x and e stand for the general input factor and the energy input, respectively.

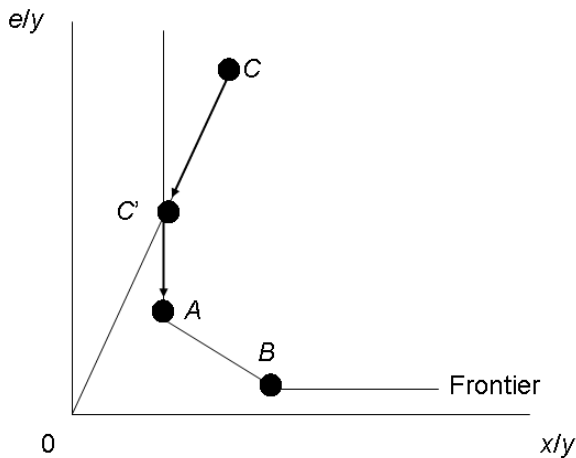


Figure 2 Radical Adjustment and Slack Adjustment

Since DMU A and DMU B have the best practice, both of them locate on the frontier. As DMU C does not locate on the frontier, it thus takes DMU A as a reference objective for reaching the best energy efficiency by reducing the amount of energy used. Point C' is the best practice for DMU C by radial adjustment $\overline{CC'}$. Point C' can maintain the same output level by slack adjustment $\overline{C'A}$. Hence, DMU C can reach the best energy efficiency, such as DMU A, by radial adjustment $\overline{CC'}$ and slack adjustment

$\overline{C'A}$. According to Equation (3), the optimal amount of energy usage = actual amount of energy usage – $\overline{CC'}$ – $\overline{C'A}$.

3. EMPIRICAL ANALYSIS

In this section we examine the influence of energy efficiency on economic growth.

3.1 Data

The research objectives in this paper are 14 EASN members: Brunei Darussalam, Cambodia, Indonesia, Malaysia, Philippines, Singapore, Thailand, Vietnam, Australia, China, India, Japan, South Korea, and New Zealand. Because 2 EASN members - Lao PDR and Myanmar - have incomplete data, we do not incorporate them into our research samples. We obtain the annual data for the 14 EASN members from the World Bank's website, and the empirical time span is from 1997 to 2009. The base year of annual data for the 14 EASN members is 2005. We set up the variables G , K , L , and U in the extended C-D production function as follows:

$$G_{i,t} = \text{Real GDP(US\$)}_{i,t}, \quad (4)$$

$$K_{i,t} = \text{Real capital accumulation(US\$)}_{i,t}, \quad (5)$$

$$L_{i,t} = \text{Amount of labor}_{i,t}, \quad (6)$$

$$U_{i,t} = \text{Energy usage(Kt of oil equivalent)}_{i,t}, \quad (7)$$

where $i = 1, 2, \dots, 14$, and $t = 1997, 1998, \dots, 2009$.

3.2 Optimal Amount of Energy Input

We use DEAP 2.1, a DEA software, to obtain the radical adjustment and the slack adjustment of energy usage. We then apply Equation (3) to solve the optimal amount of energy usage of the 14 EASN members as noted below.

3.3 Optimal Energy Usage and GDP

Guo et al. (2012) used an extended C-D production function to investigate the contribution rate of scientific and technological progress to GDP in regions of China. In order to examine the question about whether optimal energy usage can create GDP, we follow the idea of Guo et al. (2012) to employ an extended C-D production function as follows:

$$G_{i,t} = ZK_{i,t}^a L_{i,t}^b U_{i,t}^c, \quad (8)$$

where the variables $G_{i,t}$, $K_{i,t}$, $L_{i,t}$, and $U_{i,t}$ respectively stand for the output, the capital input, the labor input, and the energy usage of the i th DMU in the t th year. The parameters Z , a , b , and c are the technological level, the capital elasticity coefficient, the labor elasticity coefficient, and the energy elasticity coefficient, respectively. For estimating the parameters a , b , and c in the extended C-D production function, we take the logarithm on both sides of Equation (8) and obtain

Table 1 Optimal Amount of Energy Usage

	1997	1998	1999	2000	2001	2002	2003	2004	2005	2006	2007	2008	2009
Australia	101288	103935	65676	64902	106829	65206	62636	73335	80867	88064	124945	129401	79032
Brunei Darussalam	2394	2384	2430	2454	2280	2225	2579	2596	2539	3253	3320	3629	3124
Cambodia	3640	3816	3889	3978	2280	4296	2579	4623	2539	3253	3320	5193	3124
China	423056	435663	236535	242095	216752	289897	210713	227768	282386	329292	393632	423844	426020
India	412207	422258	183571	181150	122135	197077	81248	91489	134768	127517	146863	145975	137343
Indonesia	113163	141247	152888	155692	65925	166499	36833	38925	77898	62376	63231	90144	85006
Japan	512461	503256	512344	518946	510777	510376	506207	522465	520515	519778	515171	495549	471992
South Korea	118388	102028	82834	84767	123385	99300	85622	90330	126612	133982	118434	121439	106468
Malaysia	23631	25504	27954	25507	30148	31758	17745	21510	62552	63897	23352	73024	66826
New Zealand	16179	15880	16448	16820	16819	17163	13862	15597	14723	15336	13055	17257	16327
Philippines	34008	34357	39045	40424	38778	35612	15426	17525	35263	38849	38142	39605	37224
Singapore	15165	15296	14021	13622	19020	17090	15379	19417	18612	19268	15652	16705	18130
Thailand	48438	60828	56093	41362	45475	63795	22275	26021	64772	57617	41424	73471	72141
Vietnam	9207	21588	19015	15008	17062	26608	8278	9405	22814	18810	16999	22173	23749

Table 2 Results of Multiple Regression Analysis

Country	R^2	Prob (F)	lnZ		a		b		c	
			Techno logical level	Prob (T)	Capital elasticity coefficient	Prob (T)	Labor elasticity coefficient	Prob (T)	Energy elasticity coefficient	Prob (T)
Australia	0.991	0.000***	2.917	0.562	0.806	0.000***	0.815	0.387	-0.847	0.356
Brunei Darussalam	0.924	0.000***	-21.296	0.032**	0.055	0.736	2.819	0.005**	1.033	0.046**
Cambodia	0.944	0.000***	-40.131	0.355	0.280	0.101	4.165	0.276	-1.250	0.518
China	0.996	0.000***	-39.970	0.369	0.971	0.000***	2.230	0.312	-0.310	0.180
India	0.990	0.000***	24.646	0.007***	0.587	0.000***	-0.764	0.106	0.215	0.261
Indonesia	0.896	0.000***	-8.275	0.836	0.572	0.000***	1.683	0.541	-0.923	0.360
Japan	0.933	0.000***	73.318	0.000***	0.846	0.000***	-3.076	0.000**	-0.903	0.010**
South Korea	0.972	0.000***	18.800	0.250	0.657	0.000***	-1.077	0.331	0.807	0.004**
Malaysia	0.946	0.000***	9.594	0.634	0.266	0.051*	-0.280	0.845	1.299	0.035**
New Zealand	0.966	0.000***	0.277	0.970	0.664	0.001***	1.006	0.203	-0.582	0.652
Philippines	0.525	0.071*	-6.202	0.836	0.635	0.097*	0.998	0.089*	-0.112	0.961
Singapore	0.875	0.000***	-3.782	0.628	0.242	0.068*	1.809	0.004**	-0.339	0.280
Thailand	0.918	0.000***	102.551	0.040**	0.569	0.001***	-6.293	0.057*	1.804	0.020**
Vietnam	0.992	0.000***	11.369	0.402	0.447	0.001***	-0.032	0.969	0.306	0.128

Note: “*” represents 10% level of significance; “**” represents 5% level of significance; “***” represents 1% level of significance.

$$\ln G_i = \ln Z + a \ln K_i + b \ln L_i + c \ln U_i \quad (9)$$

By multiple regression analysis on the sample data of the 14 EASN members, Table 2 shows the estimation values of a , b , and c .

In Table 2 we find that the R^2 values of the 14 EASN members are higher than 0.5, meaning that the multiple regression equation for each member has a high goodness of fit. The F -value of the multiple regression equation for each country is significantly lower at the 5% level of significance, further implying that there is a good linear relationship between the dependent variable and the independent variables.

The result in Table 2 shows that the capital elasticity coefficients of the 14 EASN members, except Brunei Darussalam and Cambodia, significantly affect their GDP. It suggests that the capital factor affects the GDP of more EASN members. The labor factor has a significant influence on GDP of 5 EASN members: Brunei, Japan, Philippines, Singapore, and Thailand. However, Japan and Thailand have negative marginal productivity of labor. Similarly, the energy factor also significantly affects the GDP of 5 EASN members: Brunei, Japan, South Korea, Malaysia, and Thailand. Among these 5 EASN members, only Japan has a significantly negative marginal productivity of energy.

Table 1 presents the optimal amount of energy usage for the 14 EASN members at period t defined as $U_{i,t}^*$. In Table 2 we obtain the estimation values of the parameters a , b , and c . We replace them into Equation (9) and solve for G_i^* defined as the GDP under the optimal amount of energy usage for the i th EASN member. We then compare G_i^* and the real GDP (G_i) in order to examine the question of whether energy efficiency increases GDP. Figure 3 shows the comparison result.

In Figure 3 we find that China has an absolute GDP increase when an improvement in energy efficiency occurs, whereas India, South Korea, Thailand, and Vietnam have an absolute GDP decrease when an improvement in energy efficiency occurs. Table 2 shows that China has a negative marginal productivity of energy, and India, South Korea, Thailand, and Vietnam have a positive marginal productivity of energy. Hence, a negative (positive) marginal productivity of energy may create an absolute GDP increase (decrease) when energy efficiency improves. The economic intuition is as follows: According to the definition of Equation (2), the promotion of energy efficiency indicates that the actual energy input decreases given the target energy input. If the energy elasticity coefficient (c) in Equation (8) is negative, then the decrease in the actual energy input causes the output to increase. Hence, the condition that the promotion of energy efficiency causes the

output to increase is a negative energy elasticity coefficient. Figure 4 illustrates the transport process whereby the promotion of energy efficiency causes the output to increase.

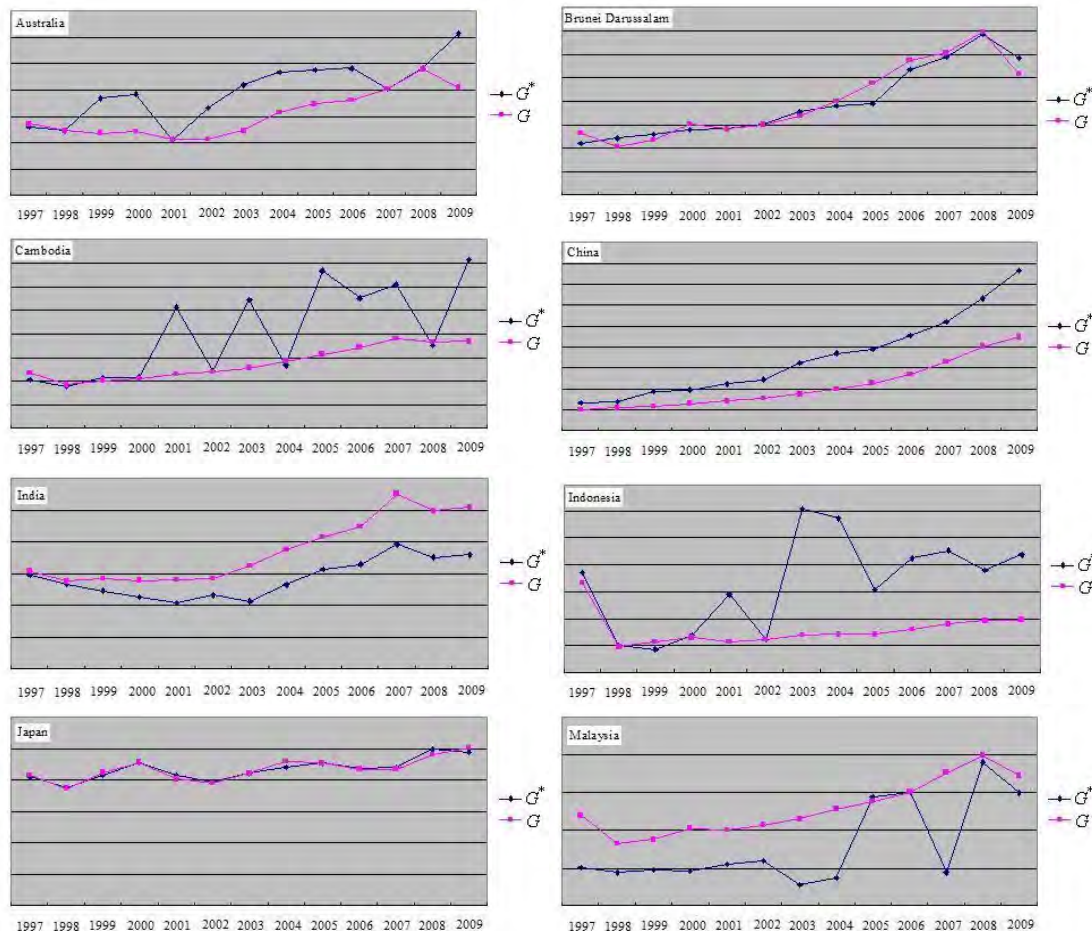
$$U_{i,t} \downarrow \Rightarrow E_{i,t} \uparrow \Rightarrow G_{i,t} \uparrow \text{ if } c < 0.$$

Figure 4 The Transport Process of the Change in Output

3.4 Energy efficiency and GDP

We use Equations (2) and (3) to calculate the energy efficiency of the 14 EASN members. Table 3 shows the calculation result and presents that the countries with the best energy efficiency in the data period are Brunei Darussalam and Japan, because their energy efficiency values are always equal to 1. China, India, and Vietnam have inferior energy efficiency values. The result is the same as Lindula et al. (2007) who argued that Lao PDR and Vietnam put forth less effort than the other EASN members in the demand-side management of energy efficiency.

The average annual GDP in this paper reflects the average degree of economic phenomena. We draw the relationship between the average annual GDP and the average energy efficiency of each of the 14 EASN members in Figure 5. Here, the total average of energy efficiency is 0.693, and the total average GDP is US\$683.1 billion.



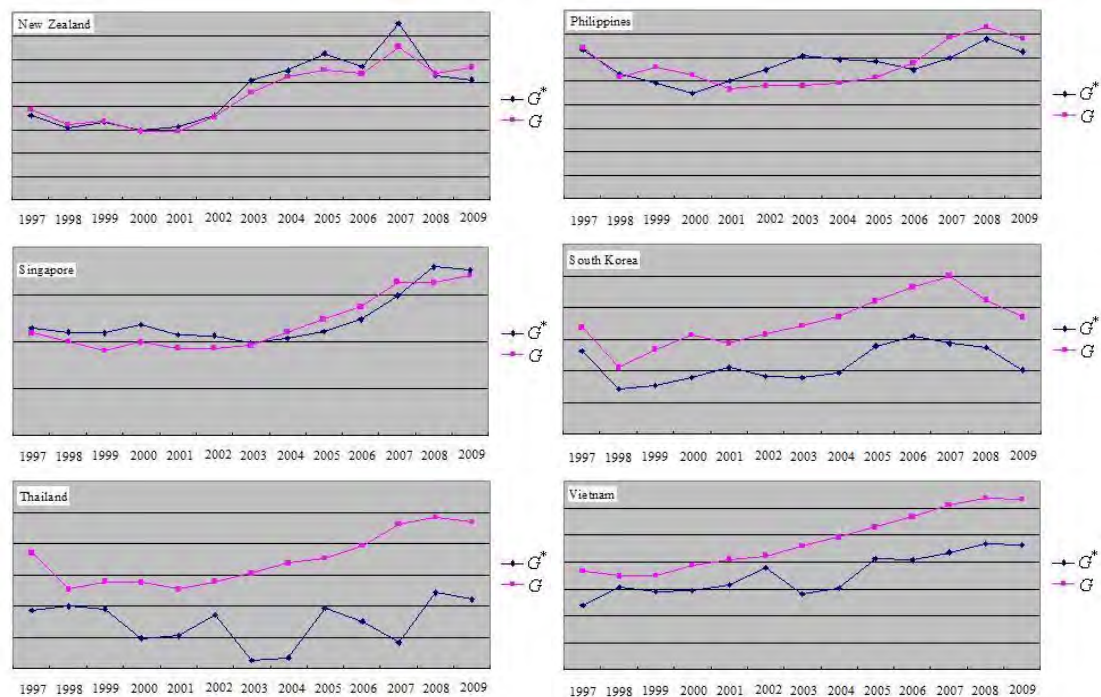


Figure 3 Comparison between Real GDP and Optimal GDP

Table 3 Energy Efficiency Value of EASN Members

Country	1997	1998	1999	2000	2001	2002	2003	2004	2005	2006	2007	2008	2009	Average
Australia	1	1	0.618	0.600	1	0.585	0.554	0.645	0.676	0.720	1	1	0.603	0.769
Brunei Darussalam	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Cambodia	1	1	1	1	0.548	1	0.581	1	0.534	0.659	0.650	1	0.603	0.813
China	0.389	0.401	0.218	0.221	0.199	0.245	0.157	0.145	0.166	0.178	0.200	0.200	0.189	0.224
India	1	1	0.409	0.396	0.263	0.413	0.166	0.176	0.251	0.226	0.246	0.236	0.203	0.383
Indonesia	0.783	1	1	1	0.412	1	0.221	0.218	0.429	0.332	0.336	0.470	0.421	0.586
Japan	1	1	1	1	1	1	1	1	1	1	1	1	1	1
South Korea	0.691	0.653	0.479	0.451	0.646	0.500	0.423	0.434	0.603	0.627	0.533	0.535	0.465	0.542
Malaysia	0.536	0.610	0.685	0.540	0.604	0.625	0.321	0.396	1	1	0.336	1	1	0.666
New Zealand	1	1	1	1	1	1	0.821	0.923	0.889	0.910	0.772	1	0.938	0.943
Philippines	0.917	0.902	1	1	1	0.906	0.392	0.448	0.900	1	1	1	0.958	0.879
Singapore	0.674	0.774	0.764	0.708	0.867	0.810	1	1	1	1	1	1	0.981	0.891
Thailand	0.687	0.920	0.795	0.572	0.599	0.786	0.254	0.275	0.675	0.587	0.405	0.691	0.698	0.611
Vietnam	0.271	0.605	0.533	0.406	0.438	0.633	0.189	0.189	0.448	0.359	0.303	0.375	0.371	0.394

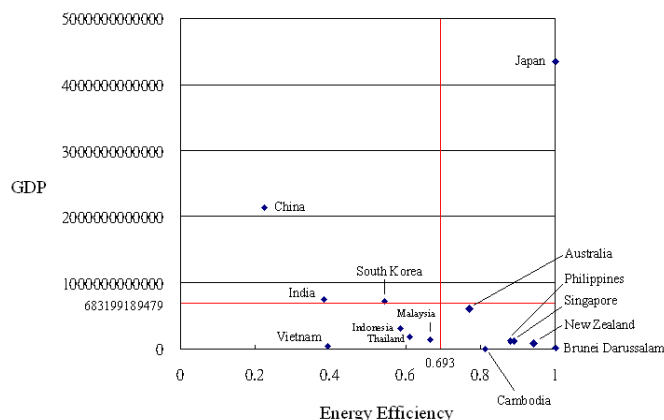


Figure 5 Relationship between Energy Efficiency and GDP

The total average energy efficiency and the total average GDP classify the 14 EASN members into four quadrants. A country in quadrant I means that it has excellent performances on energy efficiency and GDP, such as Japan. In order to reduce primary energy usage and slow down primary energy demand, Japan is actively researching and developing renewable energy technology.

A country in quadrant II has a poor performance in energy efficiency, but an excellent performance in GDP, such as China, India, and South Korea. China has one of the fastest economic growths in the world. The main reason for such fast economic growth there is capital accumulation, as Table 2 shows that capital has a significant effect on the production function of China. This conclusion is the same as in Guo et al. (2012), who investigated the contribution rate of scientific and technological progress to economic growth in regions of China.

A country in quadrant III has low energy efficiency and GDP, such as Indonesia, Malaysia, Thailand, and Vietnam. According to the record from the U.S. Energy Information Administration (EIA, 2007), Indonesia and Thailand consumed 52.6% of the total primary energy in EASN. In recent years, Indonesia has announced that it is reducing its energy exports, because its own oil and gas fields are beginning to decline (Thomson, 2006). Vietnam also has low energy efficiency, because it puts forth less effort than other EASN members in the demand-side management of energy efficiency (Lindula et al, 2007).

A country in quadrant IV has high energy efficiency, but low GDP. This type has the most EASN members belonging to it, such as Australia, Brunei Darussalam, Cambodia, New Zealand, Philippines, and Singapore. It is interesting to find what reason causes high energy efficiency in some developing countries. For example, Philippines ranks fourth in terms of global potential geothermal energy, as it uses about 2300 MW of geothermal-based power as a substitute for its primary energy usage. Many rural people in Cambodia use biomass as energy, with the proportion of biomass in energy usage as high as 73.8% (ACE, 2002). On the other hand, the per unit

electricity price in Cambodia is US\$0.17/kWh in 2002, which is much higher than US\$0.0055/kWh in Laos. Thus, the price variation in electricity is one thing that improves Cambodia's energy efficiency (ACE, 2002).

4. CONCLUSION

Since many newly industrialized countries and fast economic growing countries are located in East Asia and Southeast Asia, ASEAN is playing a bigger and more important role in the world. Some big countries in Asia, including China, Japan, South Korea, and India, as well as Australia and New Zealand, have also joined this international organization under the auspices of external relations. The 10 ASEAN members and these 6 countries with external relations have organized into EASN.

Many scholars are interested in economic performances within the EASN region. Two reasons for why we introduce energy efficiency into this study are: (1) the concept of earth's sustainable development is becoming more and more emphasized around the world; (2) ACE disseminates the concept of energy efficiency in order to improve economic growth and development in ASEAN. According to our best knowledge, previous papers in the literature have not investigated the relationship between energy efficiency and GDP in EASN.

We use the extended C-D production function to investigate and discuss three impact factors of economic performance: capital accumulation, labor force, and energy usage. We divide our sample countries into four quadrants, where energy efficiency and GDP are two indices, and find that some countries with abundant energy reserves, such as Indonesia and Malaysia, have low energy efficiency. China and India are two very fast economic growing countries, but they also have low energy efficiency. Moreover, under the optimal amount of energy usage, China can create an absolute improvement in GDP, whereas there is an absolute regression in GDP for India, South Korea, Thailand, and Vietnam. The condition that an improvement in energy efficiency creates an absolute GDP increase is a negative energy elasticity coefficient.

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A FRAMEWORK TO EVALUATE GREEN TRANSPORTATION POLICY : INTEGRATION OF DISCRETE CHOICE AND CGE MODEL

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ABSTRACT

In recent years, several new national policies have been shaped, channeling substantial investments into sustainable transportation in growing urban areas (Litman 2011). National governments have shown tremendous interest in sustainable transport because transportation is the key to achieving national targets on GHG emissions and addressing climate change. Specifically, the Asian Development Bank launched the Sustainable Transport Initiative in 2008. Eight multi-lateral development banks (MDBs) committed at the Rio+20 Summit to invest \$175 billion for sustainable transport over the next 10 years. The U.N. Decade of Action on Road Safety from 2011-2021 also placed great emphasis on the human benefits of sustainable transport. Nevertheless, many of the national policy instruments used to regulate transportation sector are distorted in various ways. For instance, fuel price is controlled by the government at a level much lower than the one that would prevail in the competitive market. The Vehicle Fuel Fee, collected in Chinese Taipei to finance infrastructure and road maintenance, is charged based on vehicle size rather than fuel consumption. To ensure the economic efficiency and cost effectiveness of transportation policies, the transportation authorities are required to evaluate the energy, economic and environmental (3E) effects of major policies such as imposing energy tax, full-cost recovering of fuel price, controlling vehicle growth, feebate program to reduce emission cost-effectively, offering subsidies to promote public transportation, etc.

In this paper we introduce the theoretical framework of the 3E Model for Green Transportation Policy Making in Chinese Taipei (GoTT). A policy scenario that covers such policy instruments as energy tax, energy efficiency improvement, the Vehicle Fuel Fee, and subsidy for public transportation is designed for simulation and the results with policy implications are reported. In addition, some research agendas are proposed to enhance the performance of the CGE approach.

Based on the proposed model, we found that (1) the baseline of CO₂ emission projected by GoTT is increasing at a decreasing rate; (2) the 3E effects are significantly different among policy instruments, and the optimal choice of instrument mix, therefore, depends on the political preference weights over the 3E effects. In other words, priority setting of potentially applicable instruments toward green transportation could be made based on the 3E effects of GoTT with diversified weighting combinations; and (3) individual's evaluation of time for alternative uses (e.g., leisure, work, and health production) are theoretically different. The model performance is sensitive to the specification of time values.

Although there are many policy instruments applicable for energy saving and emission reduction, very few are administered by the transportation authority. To ensure cost effectiveness, it is necessary to strengthen the institutional, policy and enforcement integrations. Besides, since discrete choice models are relevant to mode choice behavior, its integration to CGE model could enhance the model performance. Moreover, since time allocation among alternative uses is crucial in transportation economic models, more empirical works on the individual's valuation of time and the elasticity of substitution between uses are crucial.

Keywords: Green transportation policy, Emission reduction, CGE model, Discrete choice model, Policy integration,

1. INTRODUCTION

CO₂ emission from transportation sector accounts for more than 13 per cents of the total greenhouse gas emission worldwide and is the only sector exhibiting net increase in the last decade except during the world

financial crisis of 2008-2009. Consequently, transportation sector has been targeted by international community for mitigation. As an example, the European Union (EU) intended to extend its emissions trading scheme (ETS) to the aviation industry and began a plan to levy carbon emission surcharges by 1 January 2012 on international flights entering or exiting

the Continent. In the face of strong protests from the Obama administration, India, China and other nations that have protested the ETS, EU decided to back down the plan on 12 November 2012. Commissioner Connie Hedegaard, however, warned that unless the International Civil Aviation Organization, an agency of the United Nations, approves a plan to limit greenhouse gas emissions from aircraft by next year, the tax will be reinstated.

Instruments	Europe (continued)				North America										Latin & South America		
	Paris	Rome	Stockholm	Vienna	Brussels	Chicago	Denver	Houston	Los Angeles	New York	San Francisco	Montreal	Ottawa	Toronto	Vancouver	Mexico City	Sao Paulo
Planning	Land Use Planning				(✓)												
	Public Transport																
	Non-Motorized Modes																
Regulatory	Physical Restraint Measures																
	Traffic Management Measures																
	Regulation of Parking Supply																
	Low Emission Zone			(✓)													
	Speed Restrictions																
Economic	Road Pricing								(✓)								
	Fuel Tax Incentives																
	Parking Pricing			(✓)													
Information	Public Awareness Campaigns																
	Stakeholder Conferences																
	Driver Training																
Technology	Cleaner Technology																
Regulate	Emission Monitoring, Establishing a reliable database																

Source: GIZ (2011), 'Urban transport and climate change action plans: an overview.'

Figure 1 Instruments toward sustainable transportation

In recent years, several new national policies have been shaped, channeling substantial investments into sustainable transportation in growing urban areas (Litman 2011). National governments have shown tremendous interest in sustainable transport because transportation is the key to achieving national targets on GHG emissions and addressing climate change. Specifically, the Asian Development Bank launched the Sustainable Transport Initiative in 2008. Eight multi-lateral development banks (MDBs) committed at the Rio+20 Summit to invest \$175 billion for sustainable transport over the next 10 years. The U.N. Decade of Action on Road Safety from 2011-2021 also placed great emphasis on the human benefits of sustainable transport.

In Chinese Taipei, Green Transportation is one of the Ten Benchmark Programs under implementation by the central government. Several innovative policies and strategies had been proposed and implemented to achieve the ambitious targets of emission reduction and energy saving in transportation sector. Major strategies under implementation are grouped into five types: transportation service demand reduction, mode shift from private vehicles to public transportation, fuel switch and efficiency improvement, and advanced vehicle development.

Nevertheless, it seems that the efforts made by the transportation authority to save energy and reduce emission do not adequately achieve the planned targets for some reasons. Firstly, transportation service demand is a derived demand significantly driven by economic

growth, population growth, and city growth. The scale effect originating from such driving forces often outweighs the offset effects from the measures undertaken. Secondly, although there are many policy instruments applicable for energy saving and emission reduction, very few are administered by the transportation authority. Energy tax and energy price are typical examples. Thirdly, many of the national policy instruments used to regulate transportation sector are distorted in various ways. For instance, fuel price is controlled by the government at a level much lower than the one that would prevail in the competitive market. The Vehicle Fuel Fee, collected in Chinese Taipei to finance infrastructure and road maintenance, is charged based on vehicle size rather than fuel consumption. Finally, the measures taken to save energy and reduce emission might conflict with each other in outcomes. Fuel efficiency improvement, for example, might induce more energy consumption and, therefore, more emission (rebound effect).

Accordingly, it is indispensable to strengthen the institutional, policy and enforcement integrations to ensure cost effectiveness and policy harmonization. For such purposes, a comprehensive model for transportation policy evaluation is definitely warranted. Computable general equilibrium (CGE) model is considered one of the most appropriate approaches to evaluate the energy, economic and environmental (3E) effects of a wide-range of transportation-related policies such as energy tax, fuel pricing of full-cost recovery, controlling vehicle growth, feebate programs, subsidies to promote public transportation, etc.

Despite CGE model has been pervasively applied to evaluate 3E effects of fiscal policy, trade liberalization, environmental and climate policies, its application to sustainable transportation policy is rather limited. The CGE modeling of transportation policy evaluation is made more difficult for several reasons. (a) The linkage between individual discrete choice and transportation policy is crucial in policy evaluation but technically difficult to deal with, despite simultaneous incorporation of the discrete choice to CGE model could enhance the model performance. (b) Optimal rules of time allocation among alternative uses are crucial in transportation economic models. Nevertheless, conventional CGE models rarely incorporate the individual's valuation of time for alternative uses. Let's consider the uses of time for working, leisure, commuting, health care, and other required activities. One would expect that an individual's evaluation of the time for alternative uses may be different from one to the other. Nevertheless, value of time for alternative uses is usually assumed in literature constant and identical. This is neither realistic nor acceptable in transportation modeling. (c) The demand for transportation service is quite different from the consumption behavior of conventional market goods. It is produced by consumers by combining time, durable goods characterized by discrete choices and various attributes, and market goods (fuel, maintenance, etc.). This kind of nature complicates the service demand modeling within CGE framework.

In this paper we introduce the theoretical framework of a CGE model currently developed in Chinese Taipei, namely, the 3E Model for Green Transportation Policy Making (TaiGoTT). Based on the proposed model, we found that (1) the baseline of CO₂ emission projected by TaiGoTT is increasing at a decreasing rate; (2) the 3E effects are significantly different among policy instruments, and the optimal choice of instrument mix, therefore, depends on the political weights over the 3E effects. In other words, priority setting of potentially applicable instruments toward green transportation could be made based on the 3E effects of TaiGoTT with diversified weighting combinations; and (3) individual's evaluation of time for alternative uses (e.g., leisure, work, and health production) are theoretically different. (4) The model performance is rather sensitive to the specification of time values.

The paper is organized as follows. In section 2 we present the major development of transportation models, with particular focus on three subjects as follows: integration of the discrete choice model to CGE model, application of household production function approach to the demand modeling of transportation service, and allocation of time for alternative uses. In section 3, a dynamic CGE model developed for the Institute of Transportation in Chinese Taipei is introduced. Conclusions are reported in section 4.

2. RECENT DEVELOPMENT IN MODELING TRANSPORTATION POLICY

In light of recent development in transportation policy modeling, the following four aspects have attracted much attention than ever before:

(a) Integration of CGE model with discrete choice model, with particular focus on the choice of transportation modes (Van Steenberg, Vandresse, and Mayeres 2011; Truong and Hensher 2012);

(b) Application of the household production function approach (Huffman 2010);

(c) Benefit evaluation of transportation projects and policies (Wing, Anderson, and Lakshaman 2007);

(d) Value of time for alternative uses (Pollak 2011); and

(e) Effects of green transportation policy on income distribution and welfare (REFIT 2006; Schlör, Fischer, and Hake 2012; Nam, Selin, Reilly, and Paltsev 2010).

The integration of discrete choice model with CGE model is of particular important since mode choice is critical in transportation service demand. The value of time is also extremely important in transportation modeling. As one of the key factors of household production for transportation service, leisure, working, and health care, the traditional theory on value of time needs re-examination to effectively reflect the role of time allocation in household decision making.

2.1 Integration of CGE model with discrete choice model

In general, CGE model is characterized by numerous equations representing the behavior of representative decision makers as well as market equilibrium conditions. In practice, consumers are heterogeneous in terms of preferences over commodities and endowment of resources. Given the fact that vehicle, a durable commodity, is rather diversified in terms of its attributes (e.g., manufacturers, brand, size, equipments, safety features, power, etc.), the general consumption theory is no longer appropriate to characterize the demand of transportation services. Discrete choice over transportation modes or vehicle attributes becomes an important part of a complete theory.

There are a couple ways to integrate the discrete choice model with CGE model (see McFadden and Reid 1975, Magnani and Mercenier 2009, Truong and Hensher 2012). Figure 1 and Figure 2 are, respectively, a representation of the well-known model by Truong and Hensher (2012) and REFIT (2006). Figure 3 demonstrates how the discrete choice could be integrated with a CGE model, and Figure 4 represents a nested structure of the discrete choice over transportation modes proposed by Steenberg (2011).

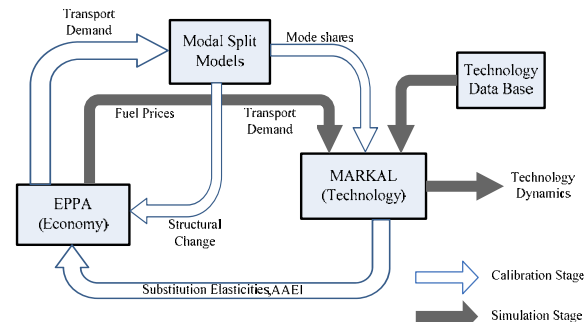
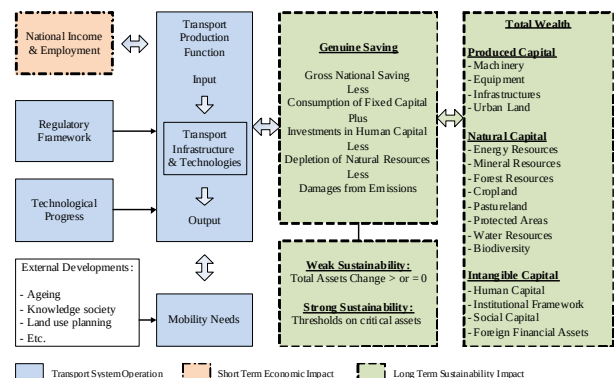
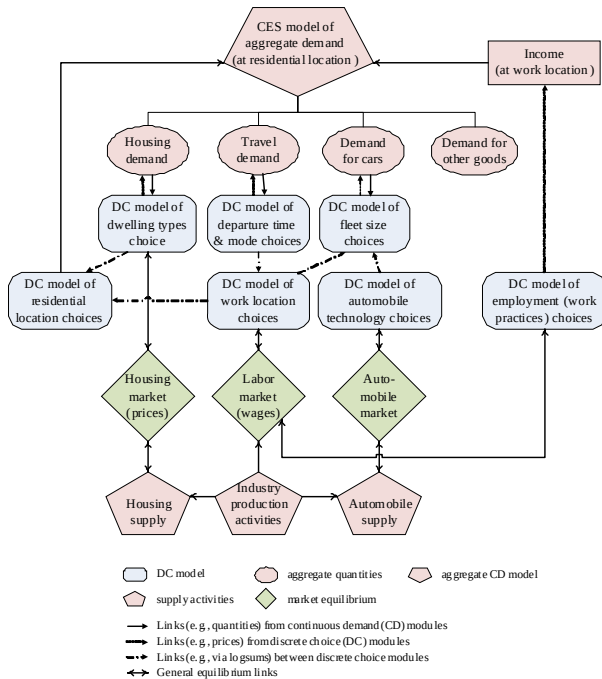


Figure 1 The framework of EPPA/MARKAL



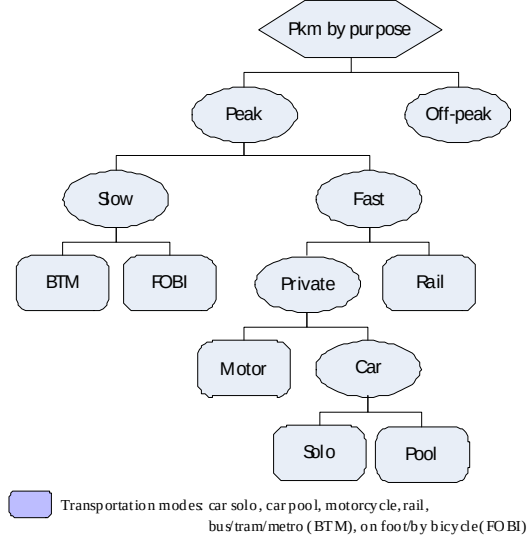
Source: REFIT (2006).

Figure 2 The framework of the EDIP model



Source: Truong and Hensher (2012).

Figure 3 Linking the discrete choice to CGE model



Source: Steenbergen (2011).

Figure 4 A nested structure of the discrete choice over transportation modes

To characterize the discrete choice, multiple Logit or Probit function is commonly specified and estimated using individual data on durable goods choice. The estimated probability is then used as an explanatory variable in the conventional demand function.

2.2 Definition of transportation demand and its marginal cost

In transportation model, the objects of demand must be carefully defined and measured. The relevant demands may be classified into three categories: demands of durable goods (e.g., vehicle), attributes of the vehicle, and transportation service (e.g., Pkm, Tkm,

etc.). The magnitude of the marginal cost associated with a particular type of transportation demand definitely differs from the one of different nature. Conrad (2000), for example, defines the user cost of a durable (i.e., vehicle) as the sum of user cost of capital, the fuel cost and other operating expenses. In contrast, the marginal cost of attribute demand and service demand is less concrete in concept.

Theoretically, the marginal cost of vehicle attributes could be estimated using hedonic price approach. The marginal cost of transportation service is so far not well defined. Further theoretical and empirical studies are warranted.

2.3 Service demand and household production function approach

In the real world, demand of transportation service is of great concern, even more important than demand for durable goods, but rarely specified in CGE models because of the following reasons:

- (a) Lack of data, especially those associated with private passenger cars;
- (b) The marginal cost of transportation service such as Pkm or Tkm is not well defined; and
- (c) Modeling of transportation service much more complicated, since it may be considered the output of a household production function that is not observable in reality.

In fact, the household production function (HPF) approach has been pervasively applied in modeling the demand for leisure and energy service. Similarly, a HPF could be specified to model the individual's demand for transportation service. The following section intends to address this issue.

3. THE MODEL

TaiGoTT, a dynamic CGE model developed for the Institute of Transportation in Chinese Taipei, aims at conducting impact evaluation of major transportation policies (e.g., vehicle fuel charge, vehicle growth control, etc.) and non-transportation policies (e.g., energy tax, energy price) as well.

The general framework of TaiGoTT is represented by Figure 5. One of the key features of TaiGoTT lines in the demand modeling of transportation services, of that the nested structure is represented by Figure 6.

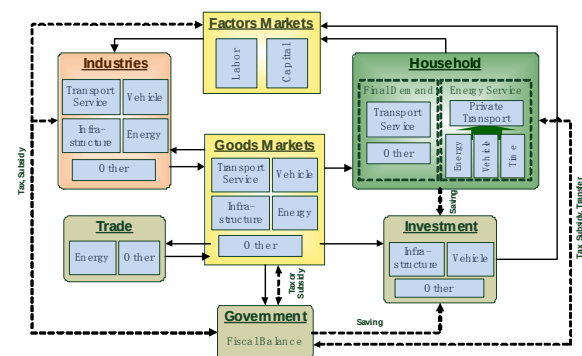


Figure 5 The framework of TaiGoTT

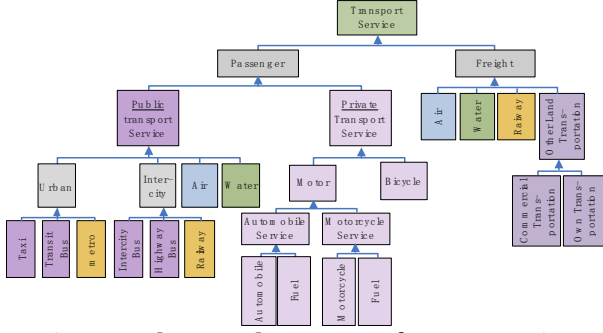


Figure 6 The nested structure of transportation service

The transportation service is produced by the household with a well-defined HPF (see Figure 7). Given the HPF, one could derive the cost function of transportation service with specific properties by solving the following cost minimization problem:

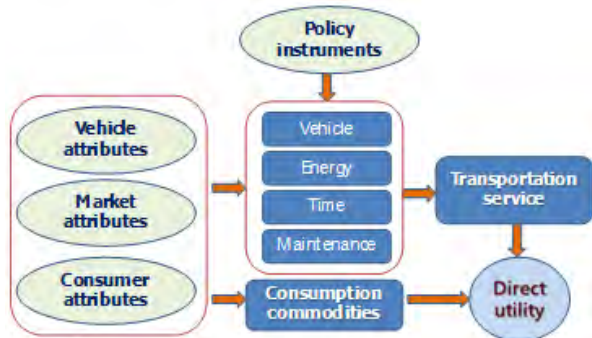


Figure 7 Production of transportation service and its linkage to consumption goods

$$C(P_E, w, \hat{t}_i, TS_i) \equiv \min_{\{Z_i, t_i\}} C_i = (r + \delta) \cdot P_i(Z_i) + P_E \cdot E_i + w_s \cdot t_i$$

$$\text{st. } TS_i = f_i(E_i, Z_i, t_i) \quad (1)$$

$$E_i = \delta_i \cdot \theta_i(Z_i) \quad (2)$$

$$t_i \geq \hat{t}_i \quad (3)$$

where P_i is the price of type the i -th vehicle, and r and δ are, respectively interest rate and depreciation rate. P_E and w_s represent the price of fuel and opportunity cost of commuting time, t_i . The transportation service of type i is produced through the HPF (Equation 2) by combining energy, attributes of the vehicle, and commuting time, bound from below by $\hat{t}_i$.

Accordingly, the marginal cost of transportation service of type i is given by

$$MUS_i = \partial C_i(P_E, w, \hat{t}_i, TS_i) / \partial TS_i.$$

The household is then assumed to solve the following utility maximization problem:

$$\max_{\{X, TS, t\}} U = U(X, TS, I)$$

$$\text{st. } \sum_j P_j \cdot X_j + \sum_i C_i(\cdot) \cdot TS_i = (1 - \tau) w_w \cdot t_w \quad (4)$$

$$t_w + I + \sum_i t_i = \bar{T} \quad (5)$$

where the individual derives utility from consumption commodity (X), transportation service ($TS = (TS_1, \dots, TS_n)$), and leisure (I). Equation (4) is the budget constraint, where τ , w_w and t_w represent income tax rate, wage rate and working time, respectively. Equation (5) is the time constraint. Note that here the time is allocated among three uses, working, leisure and commuting.

Solving the utility maximization problem, one gets the demand functions of all kinds of transportation services, and demand function of all consumption goods, and a bundle of time allocation, $(t_w, I, t_1, \dots, t_n)$.

4. CONCLUSION

There are several promising transportation models in literature, including ECLIPSE, EPPA/MARKAL, EDIP and ASTRA. TaiGoTT is distinct from others for embedding the following modules:

- A multiple discrete choice model of transportation modes that links to a CGE model;
- A new theory on the demand of transportation service and the marginal user cost;
- A complete model of time allocation for working, leisure and commuting.

The model could be improved in the following aspects:

- Integration of discrete choices and investment on durable goods with aggregate transportation service demand;
- Estimation of density functions characterizing demand structure of vehicles (e.g., by age, size, type, etc.);
- Phase-out mechanisms of durable goods consumption; and
- Projection of transportation service demand of private vehicles.

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ENERGY MANAGEMENT OF SMALL AND MEDIUM ENTERPRISES INDUSTRIES AND BUILDINGS FOR LOW CARBON ECONOMY IN THAILAND

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ABSTRACT

Thailand was an ordinary agriculture country and still in a growth rates of industrialization process of the national economy. The energy demand in Thailand has been increasing continuously. There was important to face a further challenge from the environmental impact caused by the use of energy and related to impact of climate change resulting from global warming. Relevant international agreements, coupled with the pressure from Thailand's trade dialogue partners that have stringent policies to reduce greenhouse gas (GHG) emissions, are major driving factors for Thailand to gear the national system towards a low-carbon energy system. This paper was focused on the energy management in several types of industries and buildings that could promote effectively function display of energy saving and emission reduction in 101 small and medium enterprises industries and buildings for low carbon economy of the policies and regulations about energy saving and emission reduction in Thailand.

Keywords: Small and medium enterprise, Low carbon economy, Greenhouse gas emission, Energy saving.

1. INTRODUCTION

The energy demand in Thailand has been increasing continuously, the energy consumption in Thailand continuously increased by an annual average rate of 4.4% at present. In the next 20 years of energy forecast, if there is no energy conservation or energy efficiency improvement measures or no significant reform of the industrial structure and others sectors, energy demand will increase from 71,000 ktoe/year to 151,000 ktoe. Focusing in the demand in the industrial and commercial sectors, there will still increase at a higher rate than other sectors. Hence, greenhouse gas emission from these energy sectors will trend to increase accordingly, an environmental impact and climate change resulting from energy production and utilization will be exacerbated in the future.

For response to this reason, energy conservation has been an important policy of the Thai government, particularly since the enforcement of the Energy Conservation Promotion Act, B.E. 2535 (1992). The Ministry of Energy has developed Energy Efficiency Project Plan (2011-2030) to provide the national policy framework and guidelines on energy conservation implementation in Industrials. The action plans to allocate funding support from the Energy Conservation Promotion Fund for energy conservation activities have been consecutively developed. The Promotion of

Energy Conservation in Small and Medium-Sized Industry and Commercial Building is the programme involved the study, R&D and support to bring about efficient use of energy in the industrial and commercial building sectors. The target was to increase energy efficiency and hence reduce commercial energy consumption. There were a total of 101 industries and buildings involved in the project, including 90 factories and 11 commercial buildings, respectively. The aim of this study was to evaluate were covered issues : results of energy saving measures of electricity and fuels amount; cost saving value, cost investment, payback periods and internal rated of return (IRR), respectively. According to an energy saving target has been set with a view to strengthening regional energy security and addressing climate change problems.

2. MATERIAL

The methodology to identify the evaluation and monitoring process of the project are monitored through direct data collection of 101 project participants. In each participant, the visiting included site survey, follow-up energy saving measures as well as interviewing the representative of the factories/buildings who involved in the project. The interviews are classified into 9 group of industries and building cluster, as shown in Table 1.

Table 1. Energy consumption by industries and buildings cluster

Groups	Type of Industry	Quantity	Energy consumption		toe/yr
			Electricity (kWh/yr)	Thermal energy (MJ/yr)	
1	Metallic	25	14,916,335.38	9,425,555.84	1,494.28
2	Food & beverage	22	23,850,803.58	100,263,403.90	4,405.98
3	Chemical	17	22,132,124.06	99,933,117.89	4,251.70
4	Non-metallic	15	18,329,763.79	5,469,605.86	1,691.52
5	Furniture	6	3,623,957.15	23,025,600.00	853.90
6	Textile	5	6,418,680.00	702,768.00	563.63
	Total (industries)	90	89,271,663.96	238,820,051.49	13,261.00
7	Hospital	4	6,505,692.00	-	554.41
8	Office	4	3,576,868.03	-	304.82
9	Hotel	3	1,690,674.00	-	144.08
	Total (building)	11	11,773,234.03	-	1,003.31
	Total	101	101,044,897.99	238,820,051.49	14,264.31

3. METHODS

The collaboration of expert teams and process engineers has been defined and proposed the guidelines, procedures and minimum standards for energy-saving in electrical and thermal systems for 101 participants of small and medium enterprises were as following :

1. Energy audit and investigated in energy saving measures of machines, equipment plant and system.
2. Evaluation and monitoring for process improvement and the success of reduced energy consumption.

4. RESULTS

4.1 Implemented of energy saving measures

The expert teams has been implemented the energy saving measures in electricity and thermal process, which were recommended to the industries and buildings owners. The purpose of measures is to achieve a realistic energy-saving before the after of the project by the expert teams and process engineers. The energy saving measures were purposed of measures is to achieve a realistic energy-saving before the after of project.

The classification of energy saving measures can be distinguished into 10 groups according to their systems and equipments/appliances, such as lighting, air condition, process, air compressor, electrical power, motor, boiler, chilled water, dryer and other system. The total implemented energy saving measures of all types of industries and buildings were the 560 energy saving measures, for each of the energy saving measure groups was shown as Figure 1.

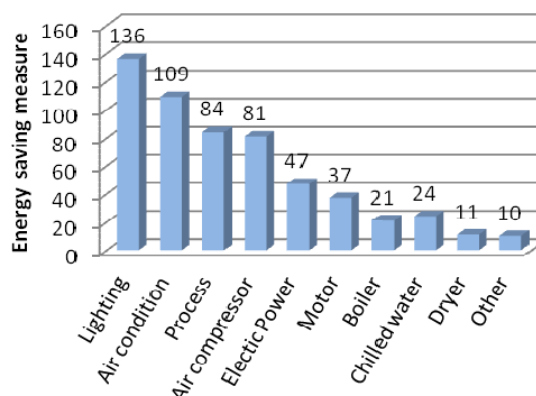


Figure 1. The number of energy saving measures

Figure 1. illustrated 10 groups of energy saving measures, there has been implement in the 101 industries and buildings. The group of energy saving measure in lighting system and air conditioning has the highest activities, because of it was very simple method and without investment. Next for process improvement and reducing air compress leakage for energy saving measures, there was lots of energy saving potentials but need some investment. The energy saving measures of electric motor, boiler, chiller and dryer were difficult to be done but lots of energy saving potentials.

Table 2 presented the result of energy-savings corresponding to the recommended measures by the experts, which classified by types of industries and buildings. These measures were already implemented.

Table 2. Summary of implemented of energy-savings measures by cluster

Type		Energy consumption	Energy-saving		Energy-saving in Baht	Investment cost	Payback period
		(toe/yr)	(toe/yr)	%	(Baht/yr)	(Baht)	(year)
1	Metallic	1,494.28	128.71	8.6	4,423,242.66	632,026.00	0.14
2	Food & beverage	4,405.98	261.50	5.9	5,135,753.35	1,226,286.00	0.24
3	Chemical	4,251.70	199.38	4.7	4,846,450.13	1,285,690.00	0.27
4	Non-metallic	1,691.52	104.26	6.2	4,267,681.39	369,714.00	0.09
5	Furniture	853.89	244.08	28.6	593,305.49	219,000.00	0.37
6	Textile	563.63	27.02	4.8	41,269,161.16	370,740.00	0.01
Total (industries)		13,261.00	964.95	7.3	60,535,594.18	4,103,457.00	0.07
7	Hospital	554.41	36.63	6.6	1,259,827.57	713,656.00	0.57
8	Office	304.82	31.95	10.5	1,254,907.95	1,671,060.00	1.33
9	Hotel	144.08	20.96	14.6	791,898.52	547,151.00	0.69
Total (building)		1,003.31	89.54	8.9	3,306,634.04	2,931,867.00	0.89
Total		14,264.31	1,054.49	7.4	63,842,228.22	7,035,323.00	0.11

4.2 Summary of energy saving

The assessment of energy saving for 90 industries was 964.95 toe/yr, which represented 7.3% of the total energy consumption. There have been translate in terms of an annual monetary saving of 60,535,594.18 Baht/yr, which has an investment cost of 4,103,457 Baht and simple payback period of 0.07 year.

And the energy saving for 11 buildings, there was 1,054.49 toe/yr, which represented 8.9% of the total energy consumption. There have been translate in terms of an annual monetary saving of 3,306,634.04 Baht/yr, which has an investment cost of 2,931,867 Baht and simple payback period of 0.89 year.

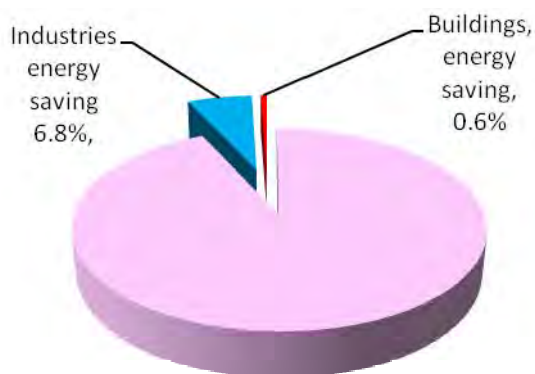


Figure 2. The proportion of energy-savings classified by cluster

As a results, the total energy saving measures in implement industries and buildings was 1,054.49 toe/yr, which an average 7.4% of the total energy consumption, the proportion of energy-savings by sectors was shown as Fig.2.

4.3 Cumulative avoided of low carbon economy

The final energy saving of 1,054.49 toe/yr were decreasing pollution emissions resulting from the use of fossil fuels for heat and electricity generation, and maintain cost of the environmental quality, avoided CO2 emission at about 3,766 tons/yr.

Table 3. Summary of GHG emission reduction

Sector	Energy saving (toe/yr)	GHG emission reduction (ton/yr)
Industries	964.95	3,446.25
Buildings	89.54	319.79
Total	1,054.49	3,766.04

5. DISCUSSION

From the survey and monitoring of the programme participation, the expert team has been monitoring the organisation energy management in each industries and building in order to analyse the improvement and follow-up the organisation energy management. This can be summarised as followings.

5.1 Energy management policy

There are energy management policy which is supported by the management level in most participants. But it is lack of the follow-up determined policy.

5.2 Organisation arrangement

Mostly participants are having person responsibility for energy management and energy committee, which is selected from different departments. However, the committee has no real power. Once the end of project, they are not playing an importance role in energy saving.

5.3 Motivation and convincing in energy saving

In most participants use the established energy committees for pursuing energy saving activities. But the management level should put more afford to support the energy conservation programmes.

5.4 Information system

There are found that in most participants' do not having a good energy information system. Generally, they have internal summary report, which is distributed to management level. This information does not disseminate widely to the employees.

5.5 Public relation

For the most participants, they do advertise regarding energy conservation programme to the employees. In addition, they arrange some training but it is during the project period. It should be encouraged more campaign and advertising even after the end of the project.

5.6 Investment

The participants are concern with the payback period as an indicator for investment in energy saving measures. Some participants are having budgets for the investment in the important of energy conservation project; however the rest of them is considered the measures that are having short payback period or lower investment.

6. CONCLUSIONS

The programme were evaluated through direct survey and interview the 101 participates, the small and medium-sized industries and buildings. The results of data analysis state that there were 560 recommended energy saving measures by the expert teams. The achievement of the project was reduced the energy consumption of 14,264.31 toe/year, which were equivalent to 7.4% of the total energy consumption; the monetary-savings of 63,842,228.22 Baht/year, the total investment cost were 7,035,323 Baht with the average payback period of 0.11 year. Regarding the continuing to implementation of energy saving measures after the end of the project.

The Assessment of the GHG emission impacts from this programme, the final energy saving of 1,054.49 toe/yr were decreasing pollution emissions resulting from the use of fossil fuels for heat and electricity generation, and maintain cost of the environmental quality, avoided CO₂ emission at about 3,766 tons/yr, which cause global warming and climate change.

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THE CHOICE OF THE SAME WAY OR THE DIFFERENT WAY

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ABSTRACT

Concretely,how large buildings fit for the reverse water system or the direct water system, now it has no clear limit. Take the fan-coil plus fresh air system as an example, when the imbalance in the rate of water is usually less than 15%, we analyze the construction scale fit for the reverse water system or the direct water system, and set up the model of construction scale-the reverse water system-the direct water system, to provide theory for air-conditioning design.

Keywords: construction scale; the reverse water system in the air-conditioning; the direct water system in the air-conditioning; the imbalance in the rate of water.

1. FOREWORD

The air-conditioning water system draws pipes laying into the reverse water system and the direct water system, in the reverse water system the water rate that flows over every terminal equipment is equal,but in the direct water system it is contrary[1]. As we all know,the hydraulic stability and the flow distribution uniformity of the reverse water system is better than that of the direct water system. In the reverse water system, the water flow that flows over every terminal equipment is equal(or approximative Equal),its strongpoint is that it is close to the current resistance of every terminal loop,it is favourable to the imbalance in the rate of water,it can reduce load of the beginning of debugging in the system. But the reverse water system expends more pipe materials than the direct water system, because the reverse water system uses return pipe, as the length of the pipeline increases,the water resistance increases,the energy consumption increases,investment in the direct water system,early increases. In the usually the terminal equipment is further from machine room,whose loop resistance is bigger. The major virtue that adopts the direct water system is saving pipes and its space(usually it can save a backwater pipe than the reverse water system),it is favorable to early investment. The direct water system for these buildings that adopt variable-water-volume system is a kind of economical pipe disposal mode.

the reverse water system divided into the vertical reverse water system and the horizontal reverse water system: the vertical reverse water system mainly figure out the resistance balance problem among the terminal equipment loop of every floor; but the horizontal reverse water system mainly figures out the resistance balance problem among the terminal equipment loop.

Certainly if in the building the air-conditioning water system may design annular waterway,the system should design the reverse water system

$$\text{According to } H = S \times Q^2 \quad (1)$$

H—— the hydraulic pressure drop of pipe

S——the impedance of pipe

Q—— the water flux

S only relate to pipe,the S is fixed in the test,when the Q changes, H varies inversely with Q^2 . It is obvious that in the variable-water-volume system,the hydraulic pressure drop is changing at all times,it don't represent the advantage of the reverse water system. So the choice of the reverse water system and the direct water system only is in the constant-water-volume system and the resistance of terminal equipment is about the same[3].

the reverse water system and the direct water system have their conditions of application,but they all have not touched that how large building accurately fit for the reverse water system or the direct water system,that the text can educe in the semi-centralized system of fan-coil plus the new wind.

It will increase the power consumption of return water pump, if the water resistance of pipe is large,it increases operational cost of system,the small resistance is not only increasing the cost of system but also stopping the system. Because the water does not care how to deal with,it must have a certain amount of impurity,the density of the most impurity is greater than that of water,these impurity will subside in the case of slow flow. So we can not select the dia. of pipes optionally, in the fact the water resistance is between 200PA/m—300PA/m[5]. In the literature [1], literature [2]and literature [6], the friction drag R varies a little. For calculating the frame of model, the model doesn't consider the infection of breadth, at present the

arranging of water system along the infection of building breadth can be considered negligible.

2. When the friction drag r take 200 pa/m, and when the vertical pipes lay in the perimeter of building, the frame of construction scale-the reverse water system-the direct water system model

In the air-conditioning system design, it is important to insure the terminal equipment of the worst loop achieving enough flow of water. Many people think it can adjust through the valve of fan-coil system, this point of view is right. But the valve of fan-coil system is mostly ball valve and gate valve, the scope of kind of valve is very limited. It is a good idea for an air-conditioning system to have some or over ten terminal equipments, but how about it has over a hundred even more? Adjusting one of the terminal equipments may affect the others, so in the design it must be taken into account. In the design only make resistance level off and can do fine-tuning with valve, there are several circulatory systems in the air-conditioning system, sometimes the length of primary pipes of several circulatory loops is equal, in that case it need not special attention. When the length of primary pipes of several circulatory loops is not equal, it should confirm the water resistance of every circulatory system's pipe according to the size of every circulatory system, the warp is no more than 15% among the water resistance of every circulatory system [4]. It is obvious that only when the imbalance in the rate of water is less than 15%, can ensure that the terminal equipment of the worst loop has enough water flow rate.

Suppose

X —the length of building

Y —the highness of building

n —the number of the root level of building

m —the number of room required air condition

in every floor of building

L_{plumb} —the length of vertical pipe of building

L_{return} —the length of level backwater pipe

L_{nm} —the length of branch pipe on the n th floor the m th primary pipe

L_{nm_0} —the length of the n th floor the m th primary pipe

P_1 —the resistance of the worst loop

P_2 —the resistance of the best loop

$(P_1 - P_2)/P_1$ —the imbalance in the rate of water.

$L_{plumb} \approx Y$

$L_{return} \approx X$

Thereinafter the friction drag R take 200 Pa/m, and the vertical pipes lay in the perimeter of building just can't design the annular water system, the local loss of the worst loop is one half of the friction loss in the system [7].

2.1 When the level pipe is the reverse water system, vertical pipe is the direct water system

According to the imbalance in the rate of water is less than 15%

viz. $(P_1 - P_2)/P_1 < 15\%$

the total length of return water system pipe of the worst loop is L

viz. $L \approx 2X + 2Y$

The resistance of water chiller: offered by unit factory manufactory, usually between 60~100kPa.

The resistance of water chiller takes $8mH_2O = 80kPa$ (the precision value refers to catalogue sheet)

The resistance of terminal equipment of air-conditioning, it is between 20~50kPa

The resistance of fan-coil takes 40 kPa (the precision value refers to catalogue sheet)

The sum of catchment device' resistance, water divider' resistance and catch-basin' resistance takes 60kPa

The following is the sketch of chilled water.

Usually if the floor of building is few, vertical pipe will devise the direct water system, if the floor of building is more, vertical pipe will devise the reverse water system.

So the friction loss of the worst loop = RL

$= 200 * 2 * L_{plumb} + 200 * (L_{n10} + L_{n20} + \dots + L_{nm0})$

$+ 200 L_{return}$ Pa

$L_{n10} + L_{n20} + \dots + L_{nm0} \approx X$

viz. $400 * Y + 200 * X + 200X = 400Y + 400X$ Pa

The local loss of the worst

loop = $0.5 * (400Y + 400X) = 200Y + 200X$ Pa

The resistance of the worst loop $P_1 =$ The friction loss of the worst loop + The local loss of the worst loop + $80 * 1000 + 40 * 1000 + 60 * 1000$ Pa

viz. $P_1 = 600Y + 600X + 180000$ Pa

The friction loss of the best loop

$= RL = 200 L_{return} * 2 = 400X$ Pa

The local loss of the best loop = 0.5 (The friction loss of the best loop) = $200X$ Pa

The resistance of the best loop

$P_2 = 400X + 200X + 180000 = 600X + 180000$ Pa

Because the imbalance rate of water system is less than 15%

$(P_1 - P_2)/P_1 = (600Y + 600X + 180000 - 600X - 180000) / (600Y + 600X + 180000) < 15\%$

viz. $10200Y < 1800X + 540000$

viz. $5.6Y < X + 300$

2.2 When the level pipe is the direct water system, vertical pipe is the direct water system, too.

The friction loss of the worst loop is

$RL = 200L$ Pa

The local loss of the worst loop is

$0.5 * 200L = 100L$ Pa

So the resistance of the worst loop

$P_1 = 300L + 80 * 1000 + 40 * 1000 + 60 * 1000$

$= 300L + 180 * 1000$ Pa

The friction loss of the best loop is
 $RL=200*2L110+200L111=400L110+200L111$ Pa
 The local loss of the best loop is
 $0.5*(200*2L110+200L111)$
 $=200L110+100L111$
 So the resistance of the best loop is

$P_2=600L110+300L111+180*1000$ Pa
 According to the imbalance in the rate of water is
 less than 15%
 $(P_1-P_2)/P_1=300L+180*1000-$
 $(600L110+300L111+180*1000)/300L+180*1000<15\%$
 viz. $1.7(X+Y)<90+2L110+L111$

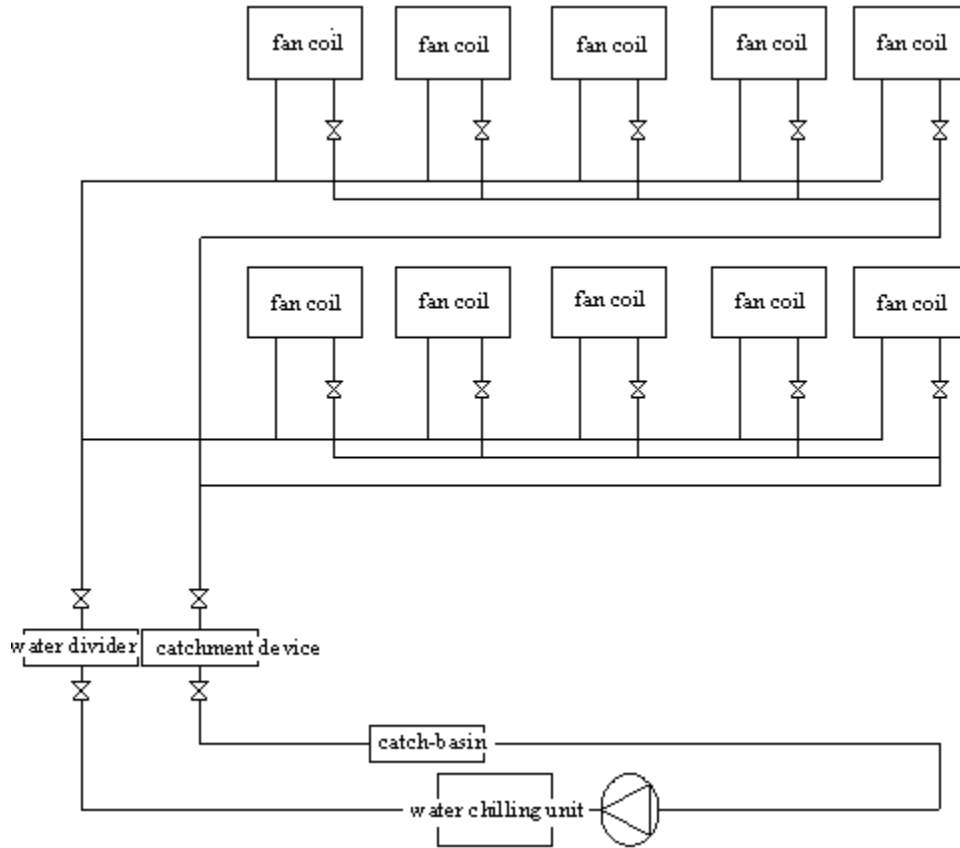


Figure 1 the sketch of chilled water

2.3 When the level pipe is the direct water system, vertical pipe is the reverse water system

There are two forms of disposal when the vertical pipe is reverse water system, the article selects the form of that the bearing of equipment and pipes are small[8]. In the summer, the difference in temperature of return water of water chiller is usually five degrees centigrade, in the winter, the difference in temperature of return water of water chiller is usually ten degrees centigrade, consider the physical process pressure, it almost regards the vertical pipe as the imbalance in the rate of water.

So the friction loss of the worst loop= RL
 $=200*2(L110+L120+...+L1(m-1)0+L1mm)$ Pa
 The local loss of the worst loop
 $=0.5*200*2(L110+L120+...+L1(m-1)0+L1mm)$ Pa
 The resistance of the worst loop
 $P_1=200*2(L110+L120+...+L1(m-1)0+L1mm)$
 $+0.5*200*2(L110+L120+...+L1(m-1)0+L1mm)$

$+80*1000$ (The resistance of water chiller)
 $+40*1000$ (The resistance of fan-coil)
 $+60*1000$ (The sum of catchment device' resistance, water divider' resistance and catch-basin' resistance) Pa
 viz. $P_1=300*2(L110+L120+...+L1(m-1)0+L1mm)+180000=600X+180000$ Pa
 The friction loss of the best loop
 $=200*2L110+200L111=400L110+200L111$ Pa
 The local loss of the best loop
 $=0.5*(200*2L110+200L111)$
 $=100*2L110+100L111$
 $=200L110+100L111$ Pa
 The resistance of the best loop
 $P_2=600L110+300L111+180*1000$ Pa
 The imbalance rate of water system
 $(P_1-P_2)/P_1=600(L110+L120+...+L1(m-1)0+L1mm)+180000-(600L110+300L111+180*1000)/600(L110+L120+...+L1(m-1)0+L1mm)+180000$

viz. $(P1-P2)/P1=600(L120+\dots+L1(m-1)0+L1mm)-300L111/600(L110+L120+\dots+L1(m-1)0+L1mm)+180000$

viz. $600X-600L110-300L111/600X+180000$

Because the imbalance rate of water system is less than 15%

$600X-600L110-300L111/600X+180000<15\%$

viz. $1.7X<90+2L110+L111$

Sum-up: When the level pipe is the direct water system, vertical pipe is the direct water system, too. The condition is that

$1.7(X+Y)<90+2L110+L111$

When the level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that

$1.7X<90+2L110+L111$

When the level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that

$5.6Y<X+300$

When the level pipe is the reverse water system, vertical pipe is the reverse water system, too.

2.4 Establish the system model of the construction scale- the reverse water system - the direct water system

When it may ignore the length of branch pipe of terminal equipment, viz. $L=0$

The level pipe is the direct water system, vertical pipe is the direct water system. The condition is that $X+Y<53$

The level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that $X<53$.

The level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that

$5.6Y<X+300$

The level pipe is the reverse water system, vertical pipe is the reverse water system.

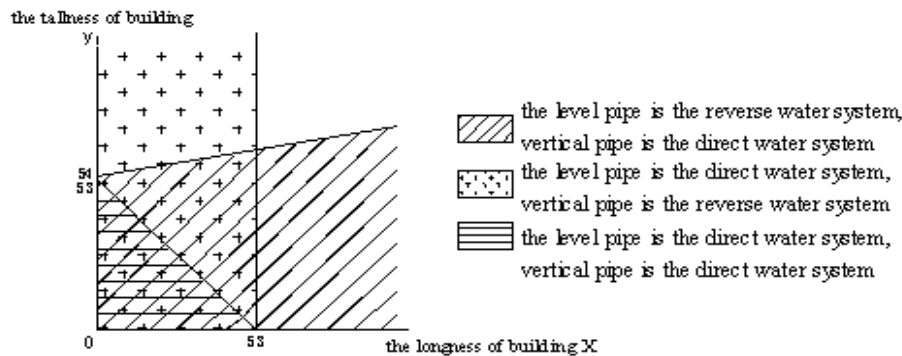


Figure 2 the model of construction scale-the reverse water system the direct water system

According to the formula above, it may find the system model of the construction scale-the reverse water system-the direct water system

In the reference frame, it all may devise that the level pipe is the reverse water system, the vertical pipe is the reverse water system too.

According to figure 2. When the friction drag R take 200 Pa/m, and the vertical pipes lay in the perimeter of building just can't design the annular water system.

(1). For fan-coil air-conditioning water system, it doesn't always use the reverse water system in any case. When the level length of building is less than 53 meter, the level pipe may devise the direct water system.

(2). The longer the building' level length is, the higher the building' vertical length, the building fits for the reverse water system more. Contrarily it fits for the direct water system.

(3). The vertical pipe devises the direct water system, the level pipe devises the reverse water system. The vertical pipe devises the direct water system, the level pipe devises the direct water system. The vertical pipe and the level pipe are both the direct water system. They all have the scope of themselves.

3. when the friction drag r take 300 pa/m, and when the vertical pipes lay in the perimeter of building, the frame of construction scale-the reverse water system -the direct water system model

When the friction drag R take 300 Pa/m, and the vertical pipes lay in the perimeter of building just can't design the annular water system. The method is the same as above.

3.1 When the level pipe is the direct water system, vertical pipe is the direct water system, too

$P1=450L+180000$ Pa

$L=2X+2Y$

$P2=900L110+450L111+180000$ Pa

According to the imbalance in the rate of water is less than 15%

viz. $(P1-P2)/P1<15\%$

$(P1-P2)/P1=\{450L+180000-(900L110+450L111+180000)\}/450L+180000<15\%$

viz. $1.7(X+Y)<60+2L110+L111$

3.2 When the level pipe is the direct water system, vertical pipe is the reverse water system

$$\begin{aligned} P1 &\approx 900X + 180000 \\ P2 &= 900L110 + 450L111 + 180000 \\ (P1 - P2)/P1 &= 900X - 900L110 - \\ 45000L111/900X + 180000 &< 15\% \\ \text{viz. } 1.7X &< 60 + 2L110 + L111 \end{aligned}$$

3.3 When the level pipe is the reverse water system, vertical pipe is the direct water system

$$\begin{aligned} P1 &= 900X + 900Y + 180000 & Pa \\ P2 &= 900X + 180000 & Pa \end{aligned}$$

According to the imbalance in the rate of water is less than 15%

$$\begin{aligned} (P1 - P2)/P1 &= (900Y + 900X + 180000 - 900X - 180000)/ \\ (900Y + 900X + 180000) &< 15\% \\ \text{viz. } 1.7Y &< 3X + 600 \end{aligned}$$

Sum-up: When the level pipe is the direct water system, vertical pipe is the direct water system, too. The condition is that

$$1.7(X + Y) < 60 + 2L110 + L111$$

When the level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that

$$1.7X < 60 + 2L110 + L111$$

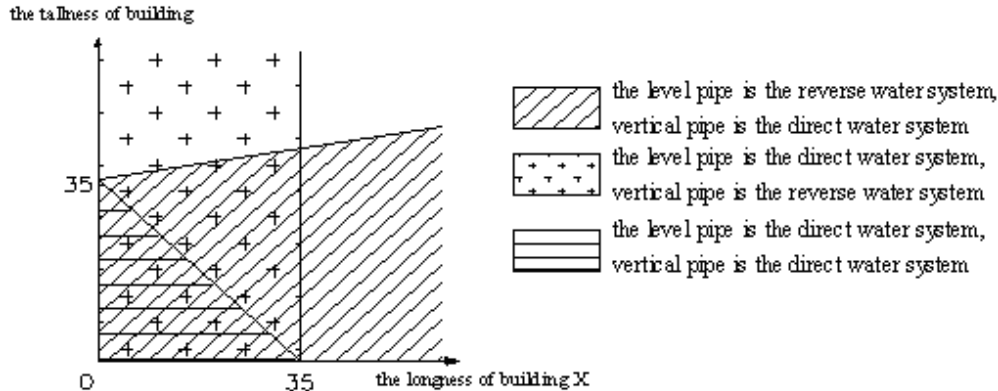


Figure 3 the model of construction scale-the reverse water system -the direct water system

According to Figure 3. When the friction drag R take 300 Pa/m, and the vertical pipes lay in the perimeter of building just can't design the annular water system.

For fan-coil air-conditioning water system, it doesn't always use the reverse water system in any case. When the level length of building is less than 35 meter, the level pipe may devise the direct water system.

The vertical pipe devises the direct water system, the level pipe devises the reverse water system. The vertical pipe devises the direct water system, the level pipe devises the direct water system. The vertical pipe and the level pipe are both the direct water system. They all have the scope of themselves.

When the level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that

$$1.7Y < 3X + 600$$

When the level pipe is the reverse water system, vertical pipe is the reverse water system, too.

3.4 Establish the system model of the construction scale- the reverse water system - the direct water system

When it may ignore the length of branch pipe of terminal equipment, viz. $L=0$

The level pipe is the direct water system, vertical pipe is the direct water system. The condition is that $X + Y < 35$.

The level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that $X < 35$.

The level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that $1.7Y < 3X + 600$.

The level pipe is the reverse water system, vertical pipe is the reverse water system. The condition is nothing

According to the formula above, it may find the system model of the construction scale-the reverse water system-the direct water system.

4. when the friction drag R take 200 Pa/m, and when the vertical pipes lay in the middle of building, the frame of construction scale-the reverse water system -the direct water system model

When the friction drag R take 200 Pa/m, and the vertical pipes lay in the middle of building. The method is the same as above.

4.1 When the level pipe is the direct water system, vertical pipe is the direct water system,too

$P_1 = 300L + 180000$ Pa
 $L = X + 2Y$
 The friction loss of the best loop = RL
 $= 200 \times 2L110/2 + 200L111 = 200L110 + 200L111$ Pa
 The local loss of the best loop
 $= 0.5 \times (200L110 + 200L111) = 100L110 + 100L111$ Pa
 The resistance of the best loop
 $P_2 = 300L110 + 300L111 + 180 \times 1000$ Pa
 Because the imbalance rate of water system is less than 15%
 $(P_1 - P_2)/P_1 = \{300L + 180000 - (300L110 + 300L111 + 180000)\} / (300L + 180000) < 15\%$
 viz. $0.85(X + 2Y) < 90 + L110 + L111$

4.2 When the level pipe is the direct water system, vertical pipe is the reverse water system

$P_1 \approx 300X + 180000$ Pa
 $P_2 = 300L110 + 300L111 + 180 \times 1000$ Pa
 According to the imbalance in the rate of water is less than 15%
 $(P_1 - P_2)/P_1 = 300X + 180000 - 300L110 - 300L111 - 180000 / 300X + 180000 < 15\%$
 viz. $0.85X < 90 + L110 + L111$

4.3 When the level pipe is the reverse water system, vertical pipe is the direct water system

$P_1 = 300L + 180000$ Pa
 $P_2 = 300X + 180 \times 1000$ Pa
 According to the imbalance in the rate of water is less than 15%
 $(P_1 - P_2)/P_1 = (300L + 180000 - 300X - 180000)$

$$/(300L + 180000) < 15\%$$

$$\text{viz. } 34Y < 3X + 1800$$

Sum-up: When the level pipe is the direct water system, vertical pipe is the direct water system,too. The condition is that

$$0.85(X + Y) < 90 + 2L110 + L111$$

When the level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that

$$0.85X < 90 + 2L110 + L111$$

When the level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that

$$34Y < 3X + 1800$$

When the level pipe is the reverse water system, vertical pipe is the reverse water system,too. The condition is nothing.

4.4 Establish the system model of the construction scale- the reverse water system - the direct water system

When it may ignore the length of branch pipe of terminal equipment, viz. $L = 0$

The level pipe is the direct water system, vertical pipe is the direct water system. The condition is that $X + Y < 106$

The level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that $X < 106$.

The level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that $34Y < 3X + 1800$.

The level pipe is the reverse water system, vertical pipe is the reverse water system. The condition is nothing

According to the formula above, it may find the system model of the construction scale-the reverse water system-the direct water system.

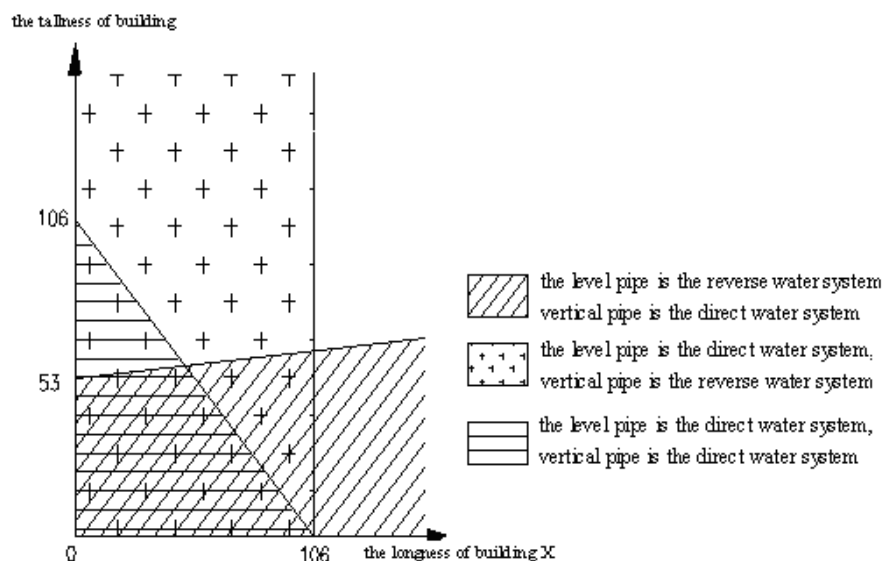


Figure 4 the model of construction scale-the reverse water system -the direct water system

According to Figure 4. When the friction drag R take 200 Pa/m, and the vertical pipes lay in the middle of building.

(1) For fan-coil air-conditioning water system, it doesn't always use the reverse water system in any case. When the level length of building is less than 35 meter, the level pipe may devise the direct water system.

(2) The vertical pipe devises the direct water system, the level pipe devises the reverse water system. The vertical pipe devises the direct water system, the level pipe devises the direct water system. The vertical pipe and the level pipe are both the direct water system. They all have the scope of themselves.

5. when the friction drag r take 300 pa/m, and when the vertical pipes lay in the middle of building, the frame of construction scale-the reverse water system -the direct water system model

When the friction drag R take 300 Pa/m, and the vertical pipes lay in the middle of building. The method is the same as above.

5.1 When the level pipe is the direct water system, vertical pipe is the direct water system, too.

$$P1=450L+180000 \quad \text{Pa}$$

$$L=X+2Y$$

$$P2=450L110+450L111+180*1000 \quad \text{Pa}$$

According to the imbalance in the rate of water is less than 15%

$$(P1-P2)/P1 = (450L+180000-450L110-450L111-180*1000) / (450L+180000) < 15\%$$

$$\text{viz. } 0.85(X-2Y) < 60+L110+L111$$

5.2 When the level pipe is the direct water system, vertical pipe is the reverse water system

$$P1 \approx 450X+180000 \quad \text{Pa}$$

$$P2=450L110+450L111+180*1000 \quad \text{Pa}$$

According to the imbalance in the rate of water is less than 15%

$$(P1-P2)/P1 = (450X+180000-450L110-450L111-180000) / (450X+180000) < 15\%$$

$$\text{viz. } 0.85X < 60+L110+L111$$

5.3 When the level pipe is the reverse water system, vertical pipe is the direct water system

$$P1=450L+180000 \quad \text{Pa}$$

$$P2=450X+180*1000 \quad \text{Pa}$$

According to the imbalance in the rate of water is less than 15%

$$(P1-P2)/P1 = (450L+180000-450X-180000) / (450L+180000) < 15\%$$

$$\text{viz. } 34Y < 3X+1200$$

Sum-up: When the level pipe is the direct water system, vertical pipe is the direct water system, too. The condition is that

$$0.85(X+Y) < 60+2L110+L111$$

When the level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that

$$0.85X < 60+2L110+L111$$

When the level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that

$$34Y < 3X+1200$$

When the level pipe is the reverse water system, vertical pipe is the reverse water system, too. The condition is nothing.

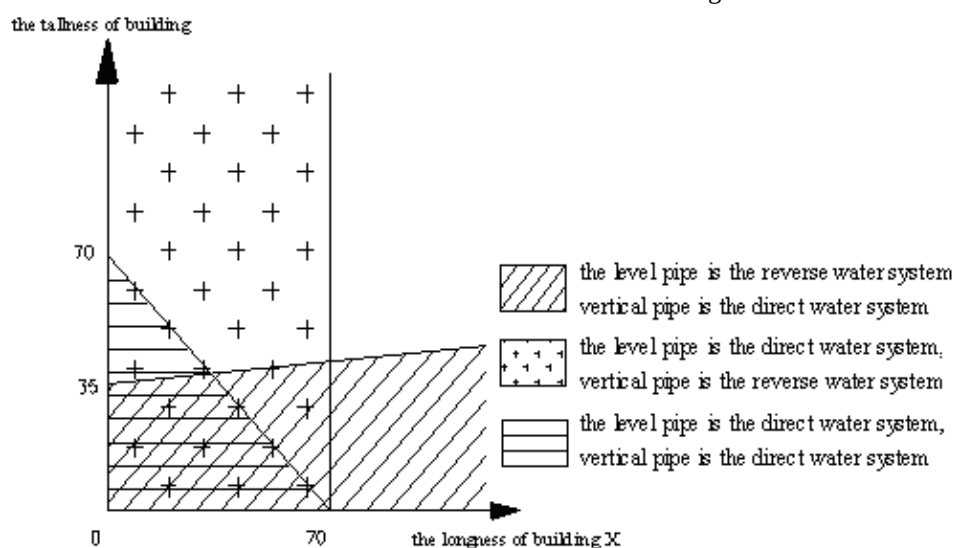


Figure 5 the model of construction scale-the reverse water system -the direct water system

5.4 Establish the system model of the construction scale- the reverse water system - the direct water system

When it may ignore the length of branch pipe of terminal equipment, viz. $L=0$

The level pipe is the direct water system, vertical pipe is the direct water system. The condition is that $X+Y<70$.

The level pipe is the direct water system, vertical pipe is the reverse water system. The condition is that $X<70$.

The level pipe is the reverse water system, vertical pipe is the direct water system. The condition is that $34Y<3X+1200$

The level pipe is the reverse water system, vertical pipe is the reverse water system. The condition is nothing

According to the formula above, it may find the system model of the construction scale-the reverse water system-the direct water system.

According to chart 4. When the friction drag R take 300 Pa/m, and the vertical pipes lay in the middle of building.

(1) For fan-coil air-conditioning water system, it doesn't always use the reverse water system in any case. When the level length of building is less than 70 meter, the level pipe may devise the direct water system.

(2) The vertical pipe devises the direct water system, the level pipe devises the reverse water system. The vertical pipe devises the direct water system, the level pipe devises the direct water system. The vertical pipe and the level pipe are both the direct water system. They all have the scope of themselves.

6. THE REVERSE WATER SYSTEM AND THE DIRECT WATER SYSTEM TO COMPREHENDED

(1) No matter selecting the reverse water system or the direct water system, their aims all make terminal equipment get enough chilled water flux, the difference of the reverse water system and the direct water system is only the distinctness of pipe disposal, the direct water system may achieve similar impact with the reverse water system by control the friction drag R of pipe. Using the reverse water system or the direct water system, the deciding means of the friction drag R of pipe are absolutely different, but nowadays, most people choose the pipe diameter of the direct water system according to the reverse water system's method. Correspondingly, confirming the friction drag R of the direct water system is more complex, if it can't deal with properly, it will not bring enough water of terminal equipment.

(2) The essential of the reverse water system and the direct water system is the imbalance of resistance, making part of terminal equipments not having enough water. Though it may make terminal equipments get enough water by increasing system

flux, it will increase energy consumption of pump. This method only fits for that the water resistance of terminal equipments is similar, the operational costs of system will not increase more.

(3) By the method of installing electromotion two way valve, static state balancing valve or using variable-frequency pump, it may avoid that the part of terminal equipments can not reach the effect of refrigeration (heating) because of the lopsided waterpower.

(4) The reverse water system greatly fit for that the water resistance of terminal equipments are almost the same or using balancing valve make the resistance of terminal equipments are almost the same in the constant-water-volume system. The distribution according to need of the reverse water system based on all the terminal equipments are almost the same. Without that it is false to discuss the reverse water system's excellency.

(5) No matter selecting the reverse water system or the direct water system, in most cases balancing valve is still integrant[3]. Because the balancing valve with closed function is a kind of diagnosis tool, inspect the system whether under design or not.

7. CONCLUSION

To fan-coil air-conditioning water system, it doesn't always use the reverse water system in any case. The vertical pipe devises the direct water system, the level pipe devises the reverse water system. The vertical pipe devises the direct water system, the level pipe devises the direct water system. The vertical pipe and the level pipe are both the direct water system. They all have the scope of themselves.

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AN ANALYSIS ON INPUT-OUTPUT MODEL AND CURRENT CONDITION OF HOTEL INDUSTRY ECONOMY-ENERGY- ENVIRONMENT SYSTEM

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ABSTRACT

Aiming at intermediate production of Chinese hotel industry's main products, material resources devotion, energy depletion, and the output of environment wastes in final service process, and having made use of combining of input-output analytical method and the analysis method of energy dissipation life spanning period, we built up input-output synthesize calculation model for economy-energy-environment system. The domestic eight representative hotels and their main product service systems have been carried on input-output calculation and synthesize analysis. The result shows that the input-output index of hotels signs differ over 200%, therefore the management of hotel industry has a huge improvement potential. In addition, the hotel's internal dining service system is the outstanding representative of economy development mode of the traditional liner open type. For this reason, it should be carried on business scale adjustment, installed energy category and consuming proportion reasonably, implemented cleaner production emphatically.

Keywords: hotel economy-energy-environment system, input-output, energy consumption coefficient, pollutant generation coefficient; cleaner production.

1. FOREWORD

According to the perspective of sustainable development theories, the hotel industry product service system should embody these three greatest characteristics: safety, comfort health, and environmental protection. They are also the basic characteristics of a green hotel. It has been gradually studied and applied by the hotel industry of our country since promulgating series standards of "ISO14000 environment management" in 1996. The hotel is constructed for provide high quality of safety, comfort, healthy indoor environment, sanitary product, and high level of service. Because of belongs to a traditional "resources-product-wastes" linear open economic development mode, the main product service system consumes a great deal of energy resources such as water, electricity, and fuel. It becomes an outstanding representative[1-3] of high energy consumption buildings. It also brings problems [4-6] of environment pollution, waste of resources etc. Besides, it displays the characteristics of "high investment, high consumption, and high pollution". This is flat opposite from the strategic requirement for the sustainable development theory. The problem of economy-energy-environment coordination and improvement is crucial. Base on the theoretic, we mostly analyze and evaluate the design of building enclosure and energy saving of equipment; however,

the lack of the research on economy-energy-environment internal influence relationship of the hotel operating process, especially need of quantitative analysis and evaluation of the whole processes of product of the hotel production and service which is due to energy depletion and atmospheric pollutant. So we need to build up an input-output calculation model, economic production value of the hotel service system, the energy of the product to consume amount, and pollutant creation amount that should relate to. We can find out the economic structure problem of hotel industry and respective hotel product service system by calculating the input-output proportion of the middle production and final service process of this service system and it helps a lot to adjust those product service systems which is economic low yield, consuming large resource, producing large environmental waste. And finally we can make a balance of hotel industry economic-environment-resource and support reference for the industry having a cleaner production, transferring to cycle economical pattern of the whole nation.

2. HOTEL'S ECONOMY-ENERGY- ENVIRONMENT SYSTEM FOR INPUT OR OUTPUT MODEL FRAMEWORK

The hotel's each item of product service usual need semi-production process and final service the

process, It will consume the human resources, the material resource, the energy resources and the water resources in these two processes, at the same time It will produce the environment waste. Therefore this computation model frame needs to manifest each item of product's semi-production process and the final structure of the service and integration between the input and output volume. Because the different of the scale of the hotel , business level, the use level , and also the product service system disposition energy, therefore each kind of energy all will have the different quantity environment load in the operational phase or in the entire life cycle.

when Analyzing the relations of the hotel economy pneumatic power supply-environment system puts into production from the sustainable development angle , It not only analyze the operational phase the material resource, the energy resources and water resources consumption quantity, but also analyze the pollutant and the greenhouse gas which produces because of the resources consumption to the environment the influence. With this viewpoint we Establish the computation model of input or output economy pneumatic power supply - environment system for the hotel production and the service of the entire process , But in this article It doesn't consider the special government measure and the human resources consumption, the water resources consumption of life cycle environment load question temporarily, It only Considers (1) the stoichiometric relation of the product for semi-production and final services in the material resource consumption, the energy consumption and the waste produce. (2) The product for semi-production and final service's the stoichiometric relation in the life cycle energy consumption and the waste produce.

Suppose that : x_{ij} is the Consumption of material resource quantity which is the i th product service for hotel and has j semi-production process;

y_i is the quantity of consumption of material resources in the final service of the process;

Q_i is the total consumption of material resources of the i th intermediate services in the production process and ultimate services of process for the hotel ;

Y_i is the hotel operating income in the additional products and services systems;

E_{ij} is the i th energy resources consumed in the j th kinds of middle production processes;

y_{Ei} is the i th energy resources consumed in process of the final services ;

Q_{Ei} is the i th total consumption of middle of the production process and ultimate services of process;

$Q_{Ei, tro}, Q_{Ei, pro}, Q_{Ei, fee}$ is 1KJ the energy consumption in transportation, manufacturing, energy exploitation;

$Q_{Ei, LC}$ is the whole life-cycle process of various energy resources consumption volume for the production of intermediate and final products services;

P_{ij} is the i th the volume of waste among the j th process of the production;

y_{pi} is the i th the volume in waste among the process of the final service;

Q_{Pi} is the total of waste producing in the middle of the production process and ultimate services of process;

$Q_{Pi, T, use}$ is the consumption of 1KJ energy of emissions in the production of intermediate and final services(using phase emission factor);

$Q_{Pi, T, tro}, Q_{Pi, T, pro}, Q_{Pi, T, fee}$ is the 1KJ energy for the emissions in transportation, production and exploitation (transport, production, mining emission factor);

$Q_{Pi, T, LC}$ is the entire process of life-cycle emissions of various energy in production of intermediate and final products service;

$Q_{Pi, C, use}$ is the urban waste emissions when consumption 1KJ energy for the production of intermediate and final (using urban stage emission factor);

$Q_{Pi, C, tro}, Q_{Pi, C, pro}, Q_{Pi, C, fee}$ is the city waste emissions for consuming 1KJ energy for transportation, production and exploitation (transport, production, mining cities emission factor);

$Q_{Pi, C, LC}$ is the energy consumption of various cities emissions production for the entire process life cycle in intermediate and final products;

Establish the relationship Following:

1)The consumption of material resources in the all process of intermediate services in the production process and ultimate services of process for hotels.

$$Q_i = \sum_{j=1}^n x_{ij} + y_i \quad (i=1,2, \dots, n) \quad (1)$$

2)The various consumption of material resources in the intermediate services of the production process and ultimate services of process for hotels.

$$Q_{Ei} = \sum_{j=1}^n E_{ij} + y_{Ei} \quad (i=1,2, \dots, n) \quad (2)$$

3) The whole life-cycle process of various energy resources consumption volume for the production of intermediate and final products in service for hotels;

$$Q_{Ei, LC} = \sum_i Q_{Ei} (1 + Q_{Ei, tra} + Q_{Ei, pro} + Q_{Ei, fee}) \quad (i=1,2, \dots, n) \quad (3)$$

4)The entire process of emissions of various energy in production of intermediate and final products in service for the hotel;

$$Q_{Pi} = \sum_{j=1}^n P_{ij} + y_{Pi} \quad (i=1,2, \dots, n) \quad (4)$$

5)The energy consumption of emissions production for the whole process life cycle in intermediate and final products(The discharge of main pollutant and the discharge of greenhouse gas)

$$Q_{Pi, T, LC} = \sum_i Q_{Ei} (Q_{Pi, T, use} + Q_{Pi, T, tra} + Q_{Pi, T, pro} + Q_{Pi, T, fee}) \quad (5)$$

6)The pollutant of city's emissions the hotel's semi-production and finally serves which the entire process life cycle energy consumption production.

$$Q_{Pi,C,LC} = \sum_i^n Q_{Ei} (Q_{Pi,C,use} + Q_{Pi,C,tra} + Q_{Pi,C,pro} + Q_{Pi,C,fee}) \quad (i=1,2,---,n) \quad (6)$$

Then

$a_{ij} = \frac{x_{ij}}{Y_i}$, It expressed the consumptive material resource's quantity of the output value which production ith service unit in jth semi-production process, which is called the consumption coefficient of semi-production process of material resource.

$a_{iy} = \frac{y_i}{Y_i}$, It expressed the consumptive material resource's quantity of the output value which production ith service unit in the finally serving process, which is called the consumption coefficient of finally serves of process resource.

$e_{ij} = \frac{E_{ij}}{Y_i}$, It expressed the quantity of ith energy resources which consumes by a unit output value of semi-production process of jth, which is called the consumption coefficient of semi-production process of energy resources.

$e_{iy} = \frac{y_{Ei}}{Y_i}$, It expressed the quantity of energy resources of ith which consumes by one unit output value of the final product service process, which is called consumption coefficient of finally serves process of energy resources.

$p_{ij} = \frac{P_{ij}}{Y_i}$, It expressed the quantity of pollution which drained by the final product service process when it make one unit output value, which is called the pollutant coefficient of semi-production process.

$p_{iy} = \frac{y_{pi}}{Y_i}$ It expressed the pollutant quantity of the ith kind which produces in the final product service process when it make one unit output value, which is called pollutant coefficient of finally serves process.

Put $a_{ij}, a_{iy}, e_{ij}, e_{iy}, p_{ij}, p_{iy}$ into formular1, formula 2 and formula 4, we will get:

$$Q_i = \sum_j^n a_{ij} Y_j + a_{iy} Y_i \quad (i=1,2,---,n) \quad (7)$$

$$Q_{Ei} = \sum_j^n e_{ij} Y_j + e_{iy} Y_i \quad (i=1,2,---,m) \quad (8)$$

$$Q_{Pi} = \sum_j^n p_{ij} Y_j + p_{iy} Y_i \quad (i=1,2,---,n) \quad (9)$$

Formula 7 and formula 9 reflect the integration connection between production of intermediate, economic output of final production services and energy consumption, the number of environmental waste.

3 THE OUTPUT VALUE, CONSUMPTION OF ENERGY AND QUANTITIES OF POLLUTION IN THE SYSTEM OF HOTELS' MAIN SERVICE.

3.1 The partition of production services system in hotel

The production services include hotel room, banquet and dinner, this is the three revenue sources of hotel. The other departments without income, such as security services, engineering services, finance and human resources management is not the main hotel products and services system. Refer to form 1.

Table 1 The partition of production services system in hotel

service item	content
tenement service system	lobby reception, room service, laundry service, clean, recreational services, business centers
banquet service system	multifunction conferences, dinning, shows, exhibitions services
dinning service system	
	food storage, processing, production, table service

3.2 confirmation of the environmental waste material

We investigated eight representative hotels' (Dalian four, Beijing one, Nanjing one, Shanghai two) average economical output value, and the energy consumption carries from 2003 to 2005, in which the hotels of Dalian A, Dalian D, Nanjing E and Shanghai F use steam which is made by fuel oil boiler production themselves, mostly uses the 0# diesel oil. The steam used in kitchen and washhouse is made by fuel-boiler in

Dalian B, Dalian C, Beijing G, the steam used in other aspect is made by thermoelectricity, by coal-boiler. The steam is made by fuel-boiler and natural gas-boiler in Shanghai H. Every kitchen used civilian gas. We should pay attention to the consumption of mineral fule, the discharge of contamination and Greenhouse gas. In allusion to this characteristics of consume, we could ensure the most contamination is SOx、NOx、CO、PM10(the grain whose diameter is less than 10cm), the Greenhouse gas is CO2. So, the hotels' garbage is one

kind of environmental waste material in allusion to the characteristics of the production-service.

3.3.Hotel economy –energy sources - environmental system

3.3.1 The introduction of calculation about input and output

It is the first that investigate the service system of the most production and the whole economic production value and energy consumption in a hotel .And then, you can use the correlative calculation model that have been given before and calculation method of every phase's environmental load in the energy period of the total life(H.S. Carslaw and Jaeger 1959), calculate the total of environmental in period of life .The amount of waste bases on the everyday statistic of managing section .Every production value of service system and the whole production value of the hotel can be received from the statistic of financial section; All hotel have installed the various gage of energy consumption, the great mass of dining service system has installed the various gage of energy consumption, but the others hardly any have installed the gage of energy consumption, so the actual energy statistic can not be received. Therefore, aimed at the actuality that the whole economic-energy consumption-

environmental system of eight hotels input and output and the proportion that dining service system input and output, firstly, we must make a calculation and a diastrophic analysis and compare; afterward, switch the difference of input and output value of the whole hotel and dining service system into the all of tenement service, epulary service and logistic section's, analyze and compare the difference among input and output value of tenement service, epulary service and dining service system, accordingly, we can know the emphases that central production service system adjust configuration of input and output.

3.3.2 Eight hotels' annual average about economic value, energy consumption and the quantity of environmental waste.

Eight hotels' investigative result of the whole economic production value, energy consumption and the quantity of productive garbage; and the calculative result of environmental waste' output quantity in the period of energy life, followed the form 2.According as the data of form 1, calculated the energy consumption coefficient of eight hotels and the environmental waste' productive coefficient, the calculative result followed the form 3

Table 2 annual average of economic value、 energy consumption and the quantity of environmental waste in eight hotels

cm		units	DalianA	dalianB	dalianC	dalianD	nanjing E	shangha iF	beijingG	shanghai H
All architectural areas		m2	43580	51994	120018	107424	67524	66628	39873	92439
economy	Producti on value	Ten thousan ds	6388.6	11293. 9	18775.5	19964.2	9742.3	14387.6	9754.9	41899.3
Energy-consump tion	Water quantity	m3	157678	259089	365856	299667	363996	368992	230061	536330
	Steam quantity	kg103	14472.3	20635. 9	24917.8	25789.2	13592.9	10832.4	18234.5	21823.7
	Convert into joule	GJ103	32.67	46.58	56.25	58.21	30.68	24.10	41.16	49.26
	Oil consum ption of boiler	kg103	829.9	352.6	346.7	1478.9	793.4	615.7	288.7	146.6
		GJ103	38.22	16.24	15.97	68.11	36.54	28.35	13.30	23.21
	Convert into joule									
	Coal consum ption of boiler	kg103	0	3304.5	4398.4	0	0	0	1416.3	
		GJ103	0	112.5	149.75	0	0	0	65.13	
	Convert into joule									

Environ- mental - waste	Natural gas consumption of boiler	m3 GJ103							462003 32.81	
	Convert into joule									
	electricity	kwh103 GJ103	5574.48 20.05	8927.9 32.05	11889.9 42.69	10379.9 37.27	12338.8 44.30	12359.9 44.38	8985.8 32.26	20984.5 75.33
	Convert into joule									
	Coal gas quantity	m3 GJ103	197563 7.7	212782 8.3	303098 11.82	397823 15.52	381566 14.88	386591 15.08	199848 7.79	568994 22.19
	Convert into joule									
	Energy in all	GJ103	60.42	86.93	110.76	111	89.86	83.56	81.21	146.78
	Proportion of energy sources in all	%	858.72 13.44	1318.3 8 11.67	1734.4 9.24	1591.9 7.97	1350.8 13.87	1731.1 12.03	1314.9 13.48	2870.2 6.85
	Garbage quantity	kg103	494.8	852.9	606.9	949.6	388.7	1415.8	581.9	1002.9
	PM10	kg103	11.38	88.1	116.82	19.87	16.2	15.21	58.68	21.44
	SOx	kg103	81.49	236.56	316.31	117.5	117.81	116.59	173.59	183.27
	NOx	kg103	40.31	156.25	228.86	64.74	84.39	75.35	119.14	123.93
	CO	kg103	43.59	87.07	119.9	19.39	24.57	18.68	57.38	30.28
	Sum total	kg103	176.77	567.98	781.89	221.5	242.97	225.83	408.79	358.92
	CO2	kg103	11825.2	27010.4	35502.8	21876.2	20737.3	18883.52	24409.4	29193.3

Table 3 direct energy consumption coefficient,direct productive waste coefficient in eight hotels

content	units	dalianA	dalianB	dalianC	dalianD	nanjin gE	shangha iF	beijingG	shangha iH	av era ge
Water consumption coefficient	m3/ Ten thousand yuan	24.68	22.94	19.49	15.01	37.36	25.65	23.58	12.80	22.69
Sort order Energy consumption coefficient	GJ/s Ten thousand yuan	6 9.45	4 7.69	3 5.89	2 5.56	8 9.22	7 5.80	5 8.33	1 3.5	6.93
Sort order Mineral fuel consumption coefficient	GJ/ Ten thousand yuan	8 19.47	5 23.42	4 25.13	2 11.39	7 21.22	3 13.54	6 25.46	1 7.96	18.45
Sort order		4	6	7	2	5	3	8	1	

Productive garbage coefficient	kg/ Ten thousand yuan	77	76	32	48	39	57	59	24	51.5
Sort order		8	7	2	4	3	5	6	1	
Productive waste coefficient	kg/ Ten thousand yuan	27.67	50.29	41.64	11.09	24.94	15.69	41.9	8.56	27.72
Sort order		5	8	6	2	4	3	7	1	
CO2Product coefficient	kg/ Ten thousand yuan	1850.9	2391.6	1890.9	1095.7	2128.6	1312.5	2502.3	696.8	1733.7
Ranking		4	7	5	2	6	3	8	1	
Total ranking		6	7	4	2	5	3	8	1	

Followed form 3, in eight hotels, the difference between maximum and minimum of energy consumption coefficient 、productive waste coefficient is about three to four times, the difference between maximum and minimum of water consumption coefficient is approximate two times, all coefficient of shanghai H hotel is the minimum, there is a remarkable difference of proportion and structure among the economy-energy sources-environment system input and output of hotel.

3.3.3 average production value of year,energy consumption,quantity and

proportion of environmental waste in dining service system

The result showed in the form 4.in the eight hotels, the average proportion of water quantity is 25.1%; average proportion of all energy consumption is 51.6%;average proportion of environmental waste is 40.95%; average proportion of CO2 is 43.23%; average proportion of garbage is 36.2%; average proportion of production value is 29.2%.the result of the direct material consumption coefficient 、 direct energy consumption coefficient、 productive coefficient of environmental waste 、 productive coefficient of garbage about complete process of dining service in eight hotels, followed form 5.

Table 4 average production value of year、energy consumption、 quantity and proportion of environmental waste about dining service system in eight hotels

content	units	Dalian A	dalianB	dalianC	dalianD	nanjing E	shangha iF	beijing G	shanghaiH	ave ra ge
econom y	productio n value	2300.05	3989.17	4280.43	4518.79	3710.52	4105.78	2975.46	8379.8920	
	Proportio n	36	35.3	22.8	22.6	38.1	28.5	30.5		29.2
	Proportio n									
Energy-consum ption	wateruan tity	50489	75464	87325	85674	77554	73623	66080	92542	
	proportio n	32	29.1	23.8	28.6	21.3	20	28.7	17.3	25.1
	Steam quantity	6633.7	9239.8	13123.1	13658.6	5921	5619.6	8752.6	8662.5	
	proportio n	45.8	44.8	52.7	52.9	43.6	51.8	48	39.6	47.4
	Consump tive quantity of electricit y	2374.5	4487.9	5442.4	4327.9	4034.9	4994.7	3857.7	5460.3	

Environ- mental- waste - output	proportion	%	40.8	50.3	45.7	41.7	32.7	40.4	42.9	26	40.1
	Coal gas quantity	m3	197563	212782	303098	397823	381566	386591	199848	568994	100
	proportion	%	100	100	100	100	100	100	100	100	100
	All energy	GJ103	39.34	52.4	61.01	61.92	42.75	45.72	41.42	61.38	53.3
	proportion	%	65.12	55.71	55.06	55.76	47.55	54.45	50.97	41.77	49.53
	Energy sources expenditure	Ten thousand yuan	387.55	635.49	806.79	734.34	785.33	780.27	786.75	1355.75	42.34
	proportion	%	45.13	48.2	46.52	46.13	58.14	45.07	59.83	47.24	36.2
	Mineral fuel	GJ103	59.83	97.04	126.22	116.4	85.43	94.96	75.71	114.62	40.95
	proportion	%	48.2	36.68	47.7	51.19	41.32	48.74	30.48	34.39	43.23
	garbage	kg103	130.2	236.9	277.4	365.6	167.4	382.3	184.1	496.1	
	proportion	%	26.3	27.8	45.7	38.5	43.1	27	31.6	49.5	
	PM10	kg103	6.56	21.7	30.38	13.1	10.03	10.36	13.46	13.02	
	SOx	kg103	33.42	97.55	132.19	64.53	53.99	60.28	86.7	72.25	
	NOx	kg103	19.94	48.58	64.39	38.24	23.93	36.19	35.95	43.12	
	CO	kg103	6.12	8.74	30.48	12.02	6.43	10.82	14.72	14.61	
	total	kg103	66.04	233.99	257.44	127.89	94.38	117.65	150.83	143	
	proportion	%	37.36	41.20	32.93	57.74	38.84	52.1	36.91	39.84	
	CO2	kg103	5911.1	10073.9	13141.3	11455.9	8888.9	9917.4	7975.02	12007.1	
	proportion	%	49.98	37.29	37.05	52.37	42.86	52.52	32.67	41.13	

Table 5 Dining service material resources consumption coefficient,energy consumption coefficient,environment waste formation coefficient of the eight rummeries.

content	unit	Dalian A	Dalian B	Dalian C	Dalian D	Nanjing E	Shanghai F	Peking G	Shanghai H	Average value
Supplies consumption coefficient	ten thousand yuan/ten thousand yuan	47.32	48.16	49.08	46.28	45.81	40.66	43.57	34.21	44.39
order		6	7	8	5	4	2	3	1	

Water consumption coefficient	m ³ /ten thousand yuan	21.95	18.90	20.4	18.96	20.9	17.93	22.21	11.04	19.04
order		7	3	5	4	6	2	8	1	
Energy resource coefficient	GJ/ten thousand yuan	13.56	13.13	14.25	13.7	11.52	11.13	13.92	7.32	12.32
composiror		5	4	8	6	3	2	7	1	
Mineral fuel consumption coefficient	GJ/ten thousand yuan	26	24.33	29.49	25.76	23.02	23.13	25.44	13.68	23.86
order		7	4	8	6	2	3	5	1	
Garbage formation coefficient	kg/ten thousand yuan	56.6	59.38	64.8	80.9	45.1	93.1	61.88	59.2	65.12
order		2	4	6	7	1	8	5	3	
Environment waste formation coefficient	kg/ten thousand yuan	28.71	58.66	60.1	28.3	25.43	28.65	50.69	17.06	37.2
order		5	7	8	3	2	4	6	1	
CO2 formation coefficient	kg/ten thousand yuan	2570	2525	3070	2535	2396	2415	2680	1433	2453
order		6	4	8	5	2	3	7	1	
Comprehensive order		6	5	8	4	2	3	7	1	

As is shown in form 5 , the dining service material resources consumption coefficient, energy consumption coefficient, environment discard creation coefficient of the H rummery in shanghai are all the minimum and the H rummery rows the first synthetically in despite of the garbage creation coefficient rowing the third. If the usage of each kind of energy in H rummery dining in Shanghai is improved further to minimize its environment impact, its comprehensive input-and-out frame work would be the best model of the eight rummeries.

3.3.4 The material resources consumption coefficient, energy consumption coefficient, environment waste formation coefficient of the tenement, the party service and the logistic section.

According to the former calculation and explanation, the consequence shows the average

proportion of water consuming in the tenement, the party service and logistic section(Apart from the occurrence of the dining service system) is 74.9%, the average proportion of total energy consuming 48.4%, the average proportion of environment waste 59.05%, the average proportion of CO2 56.77% and the average proportion of garbage is 5.9%; The material resources consumption coefficient, energy consumption coefficient, environment discard creation coefficient of the eight rummery can be obtained by these data and are recorded in form 6.The average input-output parameter of the main product service system of the eight rummeries are included in table 7.

Table 6 The resource consumption coefficient, energy consumption coefficient and environment waste formation coefficient of services such as tenement, banquet and so on in the eight rummeries.

contend	unit	Dalian A	Dalian B	Dalian C	Dalian D	Nanjing E	Shanghai F	Peking G	Shanghai H	Average
Production value of tenement and banquet	ten thousand yuan	4088.7	7307.15	14494.68	12334.84	6030.48	10287.13	6779.66	29512.82	
Material	ten	10.55	11.48	13.27	11.32	9.45	10.42	13.71	8.03	11.

resource	thousand									03
s	yuan/ten									
deletion	thousand									
coefficient	yuan									
Water	m3/ten	26.22	25.49	19.23	17.34	47.5	28.69	24.19	15	25.
consumption	thousand									46
coefficient	yuan									
Energy	GJ/ten	7.14	5.27	3.43	3.82	7.82	3.7	5.87	2.9	4.9
consumption	thousand									9
coefficient	yuan									
Mineral	GJ/ten	15.76	22.9	17.03	8.99	20.1	9.7	25.47	7.41	15.
fuel	thousand									92
deletion	yuan									
coefficient										
Garbage	kg/ten	5.94	4.82	2.7	6.43	6.33	4.39	6.96	1.56	4.8
formation	thousand									9
coefficient	yuan									
Environment	kg/ten	27.1	45.7	36.17	7.59	24.64	10.51	38.04	7.32	24.
waste	thousand									63
formation	yuan									
coefficient										
CO2	kg/ten	1447	2318	1542	845	1965	872	2424	582	149
formation	thousand									9
coefficient	yuan									
nt										

Table 7 Comparison of average input-output parameter of main product service system of the eight rummeries

parameter	unit	Dining service system	Services such as tenement banquet	rummery
Specific gravity of production value	%	29.22	Tenement 49.24%, banquet 21.54%	
Material resource consumption coefficient	%	44.39	11.03	15.76
Energy deletion coefficient	GJ/ten thousand yuan	12.32	4.99	6.93
Mineral fuel deletion coefficient	GJ/ten thousand yuan	23.86	15.92	18.45
Water consumption coefficient	m3/ten thousand yuan	19.04	25.46	22.69
Environment waste formation coefficient	kg/ten thousand yuan	37.2	24.63	27.22
Garbage formation coefficient	kg/ten thousand yuan	65.12	4.89	51.5
CO2 formation coefficient	kg/ten thousand yuan	2453	1449	1734

The material resources consumption coefficient specific value of tenement, party...etc and the dining service system is 0.248;The energy consumption coefficient specific value is 0.41;The water consumption coefficient specific value is 1.34;The mineral fuel consumption coefficient specific value is 0.56;The environment waste formation coefficient specific value is 0.66;The CO2 formation average coefficient specific value is 0.61;The garbage formation

coefficient specific value is 0.075.The rummery tenement service is the uppermost source of the income, the proportion of the total production value of the tenement service and the party service is over 70%. Compared with the dining service system, the rummery tenement service system possesses a more preponderant input-output frame work and has been the most important service system of rummery industry.

model and the case example:[The thesis of doctor].shanghai:Tongji University, 2003

4. CONCLUSION

(1)Among the eight rummeries, the material consumption coefficient, the direct consumption energy coefficient, the mineral fuel consumption coefficient, the direct water resources consumption coefficient, the environment waste formation coefficient, the C_{O_2} formation coefficient, and the direct garbage produce coefficient are quite different, and the biggest value and the least one are worth to differ 2-3 or so, which shows that in our country ,the whole of rummery profession process is considerably potential in the economy-energy-environment system improvement.

(2)The direct material consumption coefficient, the direct energy consumption coefficient ,the mineral fuel consumption coefficient ,the environment waste formation coefficient ,the CO_2 formation coefficient and the direct garbage formation coefficient of the dining service system all are the maximal in the main product service system, the coefficient proportion is nearly 50% averagely , however, the average production value proportion is not more than 30%, remain a typical model of "the traditional line open type" economy development model and its input-output frame work is badly maladjusted.

(3)The scale of rummery dining service system should be adjusted , the energy category consumption proportion also need to be optimized, and we should establish a systematic sweep production scheme and carry out performance evaluation policy.

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EXERGY COST ANALYSIS OF AIR SOURCE HEAT PUMP SYSTEM DRIVEN BY PHOTOVOLTAIC

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ABSTRACT

Based on the study of the exergy analysis and economic theories, an exergy cost model was established by using the method of exergy cost analysis. Under the background of the existed experimental system, it was calculated and analyzed comparably to study the difference between the air source heat pump system driven by photovoltaic and that driven by city power grid, applying the exergy cost model established before and the energy cost analysis method. And also the analysis of sensitivity was conducted in the exergy cost analysis of the air source heat pump system driven by photovoltaic, based on the system driven by city power grid. At last, the exergy cost characteristics of the heat pump system driven by photovoltaic were figured out, which was introductive to the practical application and development of the technology that the air-source heat pump system was driven by photovoltaic.

Keywords: Photovoltaic; Air source heat pump; Exergy cost; Sensitivity analysis.

1. INTRODUCTION

Due to the series of advantage such as that it can be used to cool in summer or heat in winter, the air source heat pump systems with simple construction are applied in residential and commercial buildings. However, most of the air source heat pump systems are driven by the city power grid. The huge electricity load increases the pressure on the city grid and also limits the promotion in some areas with the serious power gap, and the property of environmental protection or economy is also questioned. Thus, the evaluation of the economic and technical index of the air source heat pump system is particularly important.

At present, there are two methods used most commonly for the analysis of heat pump systems, energy analysis and exergy analysis. The method of energy analysis is mainly used to analyze the energy quantity of the system, which makes it more conveniently and intuitively to make the comparable analysis among different systems. However, the cooperation of the energy quality is neglected. The method of exergy analysis is based on the exergy balance of the energy, and the energy quantity and quality are also taken fully account^[1] so that it can be used to calculate the exergy loss and exergy efficiency^[2-4].

T.L Tsaros and R.A.G aggioliA used the exergy analysis method to calculate the exergy loss and exergy efficiency of every part of the air source heat pump system, and they also pointed out the direction to improve the system based on the calculated results[5]. It should be admitted that the exergy analysis method was just a kind of method based on the thermodynamic, and the economical indexes were not taken into account so that economics and thermodynamics were not unified. In order to solve this contradiction, the concept of exergy cost (thermal economics) was introduced. Gong Guangcai and Zeng Wei built the exergy cost model of cold and heat source system and pointed out that the used energy exergy cost was the key factor in evaluating heat pump systems [6]. Gong Guangcai and He Jun developed the software based on the used energy exergy cost to design and optimize the cold and heat source systems [7]. Gong Guangcai and Feng Qihui established the cost analysis exergy model of soil source heat pump system and pointed out that exergy cost analysis method was more reasonable and more scientific [8].

The method of exergy cost analysis was used in this article to calculate the used energy exergy cost of the air source heat pump driven by solar photovoltaic and the city power grid. In order to make a compare with energy analysis method, the energy analysis was also carried on in this paper.

NOMENCLATURE

Abbreviation		Subscript	
PV	Photovoltaic	-c	Cooling condition
ASHP	Air-source Heat Pump	-h	Heating condition
Symbols		Symbols	
Ex_g	Exergy of the PV, KJ	T_{su}	Outdoor temperature, K
C_g	Exergy cost of the unit PV output, 0 Yuan/kJ	T_c	Average temperature of refrigerant, K
C_e	exergy cost of unit city grid power, Yuan/kJ	T_h	Average temperature of heating medium, K
Ex_e	Exergy of the annual power consumption for the heat pump system, KJ	Q_c	Cooling capacity, KJ
C_a	Unit input exergy cost of the heat pump exchanger, Yuan/KJ	Q_h	Heating capacity, KJ
Ex_a	Input exergy of the heat pump exchanger, KJ	C_0	initial cost, Yuan
Ex	Annual cooling exergy, KJ	S_v	Salvage value of economic life period, Yuan (assumed as 10% of the initial cost)
C_1	Annualized initial cost of the system, Yuan	$PWF(i,n)$	Coefficient of cash, assumed as 10%
C_2	Annual maintenance cost of the heat pump system, Yuan	$CRF(i,n)$	Capital reclamation ratio
n	life cycle, Year	i	Annual interest rate, assumed as 10%
C_n	Unit cold(heat) load cost, Yuan/kJ	C_3	Operation cost, Yuan
Q	Cold(heat) load, KJ	Δn	Change of the lifecycle, Year(assumed as 5)
W'	Electric energy production of PV in different areas	ΔW	Change value of the energy production, ,assumed as 100kw·h

2. MODEL AND SAMPLE

2.1 Exergy cost model of heat pump system

The Algebraic exergy cost model was applied in this paper to build the exergy cost model of the air source heat pump system driven by the solar energy, which was shown in the Fig.1 and Fig.2. It could be divided into a system which included the solar plates, storage cells, power inverter and the air source heat pump unit, which could be called gray box model and there were only the energy input, cold (heat) output and the total energy loss in such a model, analyzed by the method of energy. While analyzed by the method of exergy, there were only exergy input, cold (heat) exergy output and the total exergy loss, thus the exergy cost analysis equations could be established according to the unit cost of input exergy and thermodynamic economic theory.

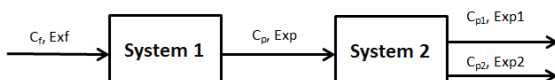


Figure.1 The Algebraic model of exergy economics

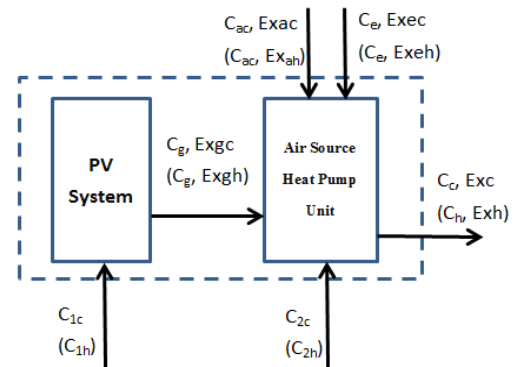


Figure. 2 Exergy cost analysis model of air source heat pump system driven by PV

2.1.1 Exergy cost analysis of the air source heat pump system driven by PV

Some exergy cost equations could be established according to the thermodynamic economic theory:

$$C_g Ex_g + C_e Ex_e + C_a Ex_a + C_1 + C_2 = CEx \quad (1)$$

The exergy cost of the unit cold and heat quantity could be indicated:

$$C_c = \frac{C_g Ex_{gc} + C_e Ex_{ec} + C_a Ex_{ac} + C_{1c} + C_{2c}}{Ex_c} \quad (2)$$

$$C_h = \frac{C_g Ex_{hc} + C_e Ex_{eh} + C_{ah} Ex_{ah} + C_{1h} + C_{2h}}{Ex_h} \quad (3)$$

According to the concept of exergy, the electric energy could be transferred into the exergy absolutely so that the amount of the input electric exergy equaled to the actual electric consumption. According to the concept of the cold (heat) exergy--the largest available work in the heat transfer process because of the temperature difference between the system and the environment, it could be regarded as that the exergy from the environment was NULL.

The exergy of cold could be indicated as:

$$Ex_c = \left(\frac{T_{su}}{T_c} - 1 \right) Q_c \quad (4)$$

The exergy of heat could be indicated as:

$$Ex_h = \left(1 - \frac{T_{su}}{T_h} \right) Q_h \quad (5)$$

The time value of the initial cost should be taken into consideration to calculate the system exergy cost or energy cost for that the life of heat pump system was as long as 20 years^[9]. What should be done was to make annual calculate and make equal division for the initial cost based on the whole life.

The calculation equation was shown as:

$$C_1 = [C_0 - S_v \times PWF(i, n)] \times CRF(i, n) \quad (6)$$

$$CRF(i, n) = \frac{i}{1 - (1+i)^{-n}} \quad (7)$$

$$PWF(i, n) = \frac{1}{(1+i)^{-n}} \quad (8)$$

According to the equations (2),(3),(4) and (5), the exergy cost of unit energy load was related to the input or output exergy as well as to the system initial cost and maintenance cost. As a result, it could be made a conclusion that the unit cooling or heating exergy cost could express the energy utilization and economic efficiency of the system comprehensively.

2.1.2 exergy cost analysis of the air source heat pump driven by the city grid

Compared with the air source heat pump driven by PV, there was no PV system in the one driven by city grid so that the whole air source heap pump unit could be considered as an individual system to build a gray analysis model. In this model, the calculation principle for exergy cost was the same to that in the model with PV. The model was shown in the Fig.3.

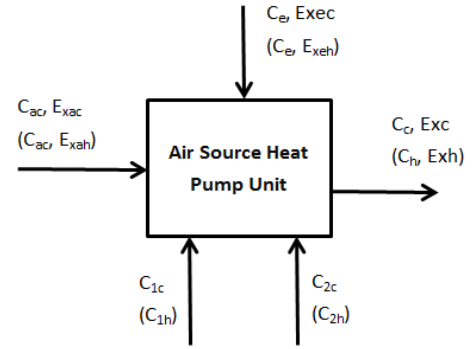


Figure. 3 Exergy cost analysis model of air source heat pump driven by urban power grid

2.2 Energy cost model of heat pump system

The output cold (heat) energy from the heat pump was regarded as the base of the cost analysis and the total number of the annualized initial cost and the operation cost was made energy unitization. The common calculation equation was shown as:

$$C = \frac{C_1 + C_2 + C_3}{Q} \quad (9)$$

2.3 Sensitivity analysis

The life and output power of PV systems would be various in different areas, while the indexes of the city grid power were relevant stable so that the air source heat pump system driven by PV was selected to make sensitive analysis. The life of the heat pump driven by PV could be given by the following equation:

$$n' = n_0 + \Delta n \quad (14)$$

The electric energy production of PV could be elected according to the follow equation:

$$W' = W_0 + \Delta W \quad (15)$$

2.4 Sample

There was a demonstration building, Hongye Building, in Hunan University and the related indexes and the schematic were shown in Tab.1 and Fig.4.

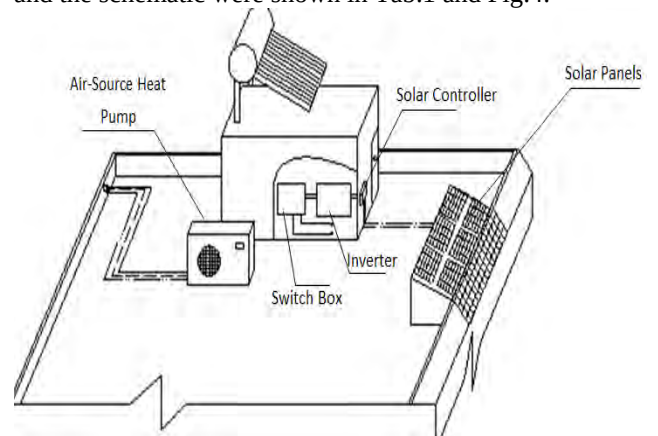


Figure.4 Schematic of the demonstration system

Table. 1 Indexes of the demonstration building

PV System	Amount of Solar Panels	10
	Rated Efficiency of Solar Panels	13%
	Total Installed Power (w)	1700
Air Source Heat Pump Unit	Air Conditioning Area (m ²)	44.5
	Rated Refrigerating Capacity (w)	7120
	Rated Refrigerating Power(w)	2232
	Rated Heating Capacity (w)	7900
	Rated Heating Power (w)	2240

Table.2 The cost and electricity consumed of the both heat pump system

Item	Case1	Case2
Total Initial Cost (Yuan)	41000	8000
Initial Cost for Cooling (Yuan)	22140	4320
Initial Cost for Heating (Yuan)	18860	3680
Annual Power Consumption for Cooling(PV) (kw·h)	600	0
Annual Power Consumption for Cooling (City Grid)(kw·h)	1245	1845
Annual Power Consumption for Heating (PV)(kw·h)	300	0
Annual Power Consumption for Heating (City Grid)(kw·h)	2148	2448
Annual Maintenance Cost for Cooling (Yuan)	664.2	129.6
Annual Maintenance Cost for Heating (Yuan)	565.8	110.4

According to the Tab.3, the unit cooling exergy cost in Case1 was 2.17 of that in Case2, the cooling energy cost was 2.16, the unit heating exergy was 2.01 and the unit heating energy was also 2.01.

The results were same when the two methods were taken to analysis the energy saving characteristics so that the exergy cost analysis method and the energy cost analysis method were accordant in the selection of the cold and heat source schemes. The results form Tab.3 also showed that the unit heat exergy cost was 5.0 of the unit cold exergy cost in Case1 and the unit heat exergy cost was 5.4 of the unit cold exergy in Case2. However, the unit cold energy cost and heat energy cost were similar in Case1 and Case2.

Table.3 Calculated result of energy cost, exergy cost and annualized cost of the two cases

Item	Case1	Case2
Annualized Cost for Cooling (Yuan)	2561.9	499.9
Unit Cooling Cost (Yuan/GJ)	164.4	76.1
Unit Cooling Exergy Cost (Yuan/GJ)	45.7	21.1
Annualized Cost for Heating (Yuan)	2182.4	425.8
Unit Heating Cost (Yuan/GJ)	185.9	92.3
Unit Heating Exergy Cost (Yuan/GJ)	229.0	113.7

Due to the exergy cost analysis, it was illustrated that the cold capacity was more valuable than the heat capacity in air-source heat pump systems. Compared with the system driven by city grid, the percentage of unit cold exergy cost from the total unit exergy cost was higher in the system driven by PV so that to develop and research the system driven by PV would create more cooling economic value.

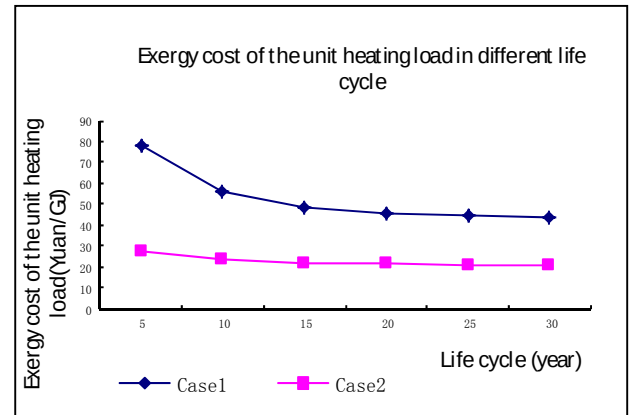


Figure 5 Sensitivity analysis of unite cooling exergy cost in different life cycle

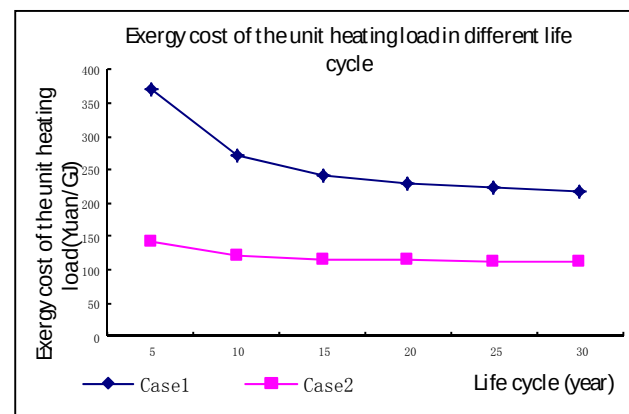


Figure.6 Sensitivity analysis of unite heating exergy cost in different life cycle

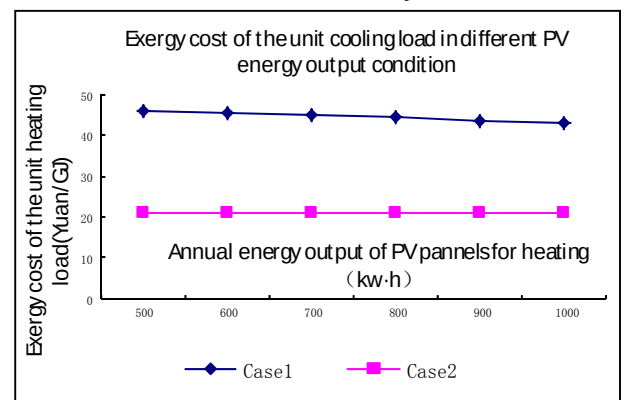


Figure.7 Sensitivity analysis of unite cooling exergy cost under the condition of different photovoltaic power generation

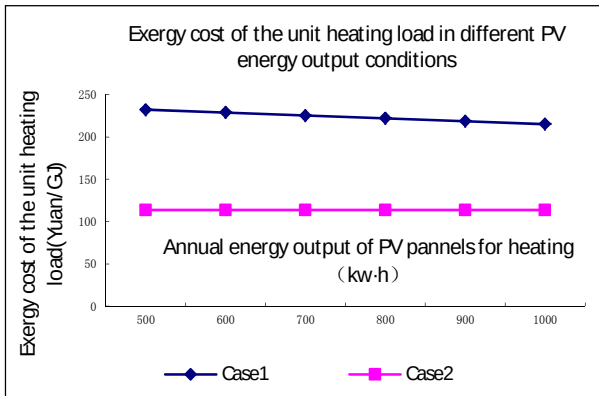


Figure.8 Sensitivity analysis of unite heating exergy cost under the condition of different photovoltaic power generation

In the Fig.5 and Fig.6, the unit cold (heat) exergy cost was reducing along with the increase of life cycle and when the life cycle was up to 20 years, the downtrend was distinct weakening to settle out. In the Fig.7 and Fig.8, the sensitivity of the system to areas with different PV power generation was very very low. It meant that the property design of life cycle would be more help to reduce the unit energy exergy cost and increase the economic benefit. However, the unit energy exergy cost of the system driven by PV was still more than twice as that of the system driven by city grid in current marketing environment even if the life cycle was designed as 30 years

3. CONCLUSIONS

Main conclusions could be presented as follows:

1. It was further confirmed that the exergy cost analysis method was very different from the energy cost analysis method for that the results from the exergy cost analysis method would more reflect the energy utilization.
2. Although the unit cold (heat) exergy cost was higher in the system driven by PV, the percentage of the unit cold exergy was also higher than that in the system driven by city grid, which meant that to develop and research the system driven by PV would create higher cooling economic value.
3. In terms of the air-source heat pump system driven by PV, the property design of life cycle would be more efficient to cut down the unit exergy cost.
4. Although the solar energy the renewable and green energy was used in this system, the unit energy exergy cost was still higher than that of city power grid in current marketing environment, which meant that something more should be done to reduce the exergy cost.

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THE TOTAL-FACTOR ENERGY EFFICIENCY OF REGIONS IN CHINESE TAIPEI: AN APPLICATION OF THE STOCHASTIC FRONTIER ANALYSIS

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ABSTRACT

This research applies the panel data stochastic production frontier to estimate total factor-energy efficiency (TFEE) scores for 23 administrative regions in Chinese Taipei during 2004-2008. The panel data stochastic frontier analysis (SFA) model to estimate TFEE is extended from the cross-section version of Zhou et al. (2012). This SFA panel data model can be empirically estimated by using that proposed by Battese and Coelli (1992) and hence the free software kindly provided by Professor T.J. Coelli (1996) can be used for estimation. This paper is an effort to complement the work of the TFEE index proposed by Hu and Wang (2006); moreover, it is also a very good experiment to further try to extend the SFA modeling for estimating energy efficiency. There are six inputs (the employed population, amount of productive electricity power consumed, amount of electricity consumed for household electric light, amount of electricity consumed for non-household electric light, amount of gasoline sales, and amount of diesel sales) and one output (the total real income in the base year of 2005) in the SFA models. Note that the rankings in TFEE scores of the same energy source, obtained by the SFA models, are all stable over time. Most administrative regions in Chinese Taipei are not efficient in almost all kinds of energy use. The average TFEE scores in various energy sources during 2004-2008 are: productive electricity power (0.378566), household lighting (0.751216), non-household lighting (0.606920), gasoline (0.754872), and diesel (0.478735). The efficiencies of using productive electricity power and diesel are needed to improve very much. productive electricity power usage and non-household lighting showed increasing inefficiency (i.e., worsening TFEE scores) over time. The efficiencies of using productive electricity power and diesel are needed to improve very much. It is a priority for administrative regions where manufacturing and high-tech industries to concentrate to achieve more efficient use of productive electricity. It is a priority for metropolitan regions to improve the efficiency of both household and non-household lighting. It is a priority for all administrative regions to improve the motor-vehicle energy efficiency in order to improve gasoline efficiency. It is a priority for non-metropolitan regions to improve the diesel use efficiency.

Keywords: Stochastic Frontier Analysis (SFA), Total-Factor Energy Efficiency (TFEE), Panel Data

1. INTRODUCTION

Chinese Taipei is insufficient in natural resources and constrained by a limited environment-carrying capacity (Ministry of Affairs 2008). Since 97.9% of Chinese Taipei's total energy consumption is imported (Bureau of Energy 2008), the foreign market fluctuations or policy changes can easily affect its energy supply and security. Because energy has great impacts on economic development as well as ecological environment, whether or not the energy use is efficient is an important issue for Chinese Taipei.

Since Hu and Wang (2006) and Hu and Kao (2007) first proposed the total-factor energy efficiency (TFEE) index, the follow-up empirical papers have been applying the data envelopment (DEA) approach to compute the energy input target (or the total energy input slack). By referring to Hu and Wang (2006), Zhou, Ang and Zhou (2012) derive an econometric

equation for estimating TFEE via re-arranging a single-equation stochastic production function. The current version of Zhou, Ang and Zhou (2012) is still for only cross section data and analyzes 21 OECD countries in the year 2001. This paper further extends the cross-section SFA model of Zhou, Ang and Zhou to a panel data, stochastic frontier analysis (SFA) model. The total-factor energy efficiency scores of 23 administrative regions in Chinese Taipei from 1999 to 2008 will be estimated by this extended panel data SFA model. Fortunately, the panel data SFA approach of Battese and Coelli (1992) can be used to estimate by this extended panel data SFA model.

The application of the total-factor energy efficiency index to Chinese Taipei's regions can be found in Hu et al. (2012). Their study applies data envelopment analysis (DEA) to compute the efficient energy-saving ratios for 23 administrative regions in Chinese Taipei from 1999-2005. One output (total

income) and seven inputs (local government expenditure, employment, processed trash, household and commercial electricity consumption, industrial electricity consumption, gasoline sales volume, and diesel sales volume) are considered in the DEA models. It is found that most of the 23 administrative regions do not efficiently use household and commercial electricity, industrial electricity, gasoline, and diesel, even with respect to Chinese Taipei's own efficiency frontier. Their results suggest that improving energy efficiency in household and commercial electricity use is a priority for non-metropolitan regions. There is still much room to improve the efficiency of electricity use for industries, especially for regions where manufacturing and high-tech industries concentrate, by means of cleaner production, energy-saving technology and equipment, etc. Motor vehicle energy efficiency is the key factor for saving gasoline. The energy efficiency of farming machines and carrying equipment should be continuously improved, especially for rural regions.

Hu et al. (2011) applies the four-stage DEA procedure to calculate the energy efficiency of 23 regions in Chinese Taipei from 1998 to 2007. After controlling for the effects of external environments, only Taipei City, Chiayi City, and Kaohsiung City are energy efficient. Kaohsiung City reaches the efficiency frontier due to the adjustment via partial environmental factors such as higher education attainment and transport vehicles. They also find a worsening trend for Chinese Taipei's energy efficiency. Not only is there a gap of energy efficiency between Chinese Taipei's metropolitan and non-metropolitan regions, but the gap has also widened in recent years. Those inefficient counties should be given priority and the savings potential. Except for road density, the evidence indicates that each environmental factor has partial incremental effects on input slacks. As more cars and motorcycles are unfavorable externalities affecting partial energy efficiency, the central government should help local governments retire inefficient old motor vehicles, encourage energy-saving vehicle models, and provide convenient mass transportation systems. Besides, people with higher education cause industrial energy inefficient in Chinese Taipei. The consciousness of effective energy saving is necessary to schools, communities, and employee accordingly.

This paper is a follow-up work which is most similar to Hu et al. (2012) by applying the panel data stochastic production frontier to estimate TFEE scores for 23 administrative regions in Chinese Taipei. Instead of the sample period of 1999-2005, this paper uses the data during 2004-2008. Similar patterns with those found in Hu et al. (2012) are found by the SFA approach extended from Zhou et al. (2012). This paper is an effort to complement the work of the TFEE index proposed by Hu and Wang (2006); moreover, it is also a very good experiment to further try to extend the SFA modeling for estimating energy efficiency.

2. RESEARCH METHODOLOGY

2.1 Research methodology

Zhou et al. (2012) apply the single-equation, output-oriented stochastic frontier analysis (SFA) model to estimate the total-factor energy efficiency. Their version is still a cross-section SFA model, which analyze 21 OECD countries in 2001. Combining Zhou et al. (2012) and Battese and Coelli (1992), this paper further extends a panel data SFA model to estimate the total-factor energy efficiency.

Following Zhou et al. (2012), we assume that the stochastic frontier distance function is in the Cobb-Douglas functional form:

$$\ln D(K_{it}, L_{it}, E_{it}, Y_{it}) = \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_E \ln E_{it} + \beta_Y \ln Y_{it} + v_{it} \quad (1)$$

where $D(\cdot)$ is the distance function, K_{it} is the capital stock, L_{it} is labor employment, E_{it} is the energy input, Y_{it} is the real economic output, i indicates the region, and t indicates the time, and v_{it} is the statistical noise following the normal distribution. Since the distance function is homogeneous of degree one in the energy input, the above equation can be re-arranged as:

$$\ln D_E(K_{it}, L_{it}, E_{it}, Y_{it}) = \ln E_{it} + \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_E \ln 1 + \beta_Y \ln Y_{it} + v_{it} \quad (2)$$

which can be also arranged as

$$-\ln E_{it} = \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_E \ln 1 + \beta_Y \ln Y_{it} + v_{it} - \ln D_E(K_{it}, L_{it}, E_{it}, Y_{it}) \quad (3)$$

That is,

$$\ln(1/E_{it}) = \beta_0 + \beta_K \ln K_{it} + \beta_L \ln L_{it} + \beta_Y \ln Y_{it} + v_{it} - u_{it} \quad (4)$$

where u_{it} is the inefficiency term which follows a non-negative distribution and $v_{it} - u_{it}$ is the error component term of a stochastic production frontier. Eq. (4) fits in the panel data, stochastic frontier model proposed by Battese and Coelli (1992). The free software Frontier 4.1 kindly provided by Professor Coelli (1996) can be used to estimate Eq. (4). The total-factor energy efficiency of region i at time t then is

$$TFEE_{it} = \exp(-u_{it}) \quad (5)$$

Therefore, we then can apply the panel data, stochastic production frontier approach to estimate the total-factor energy efficiency and are limited to use the input-oriented data envelopment analysis (DEA) suggested by Hu and Wang (2006) and Hu and Kao (2007). Moreover, if we use disaggregate energy inputs, we can also change the logged inverse energy input on the left-hand side of Eq. (4) and keep other logged inputs on the right-hand side of Eq. (4), such that we can obtain the TFEE scores of different energy inputs.

2.2 Data and variables

The data are regional and annual for 23 administrative regions in Chinese Taipei during 2004-2008. There are six inputs (the employed population, amount of productive electricity power consumed, amount of electricity consumed for household electric light, amount of electricity consumed for non-

household electric light, amount of gasoline sales, and amount of diesel sales) and one output (the total real income) in the stochastic frontier analysis (SFA) models.

The data of productive electricity power, electricity of household light, non-household light are from Chinese Taipei Power Company. The productive electricity power usage refers to non-lighting electricity usage including most of the production and service operations. The data of gasoline and diesel sales are from Bureau of Energy, Administrative Yuan. The data of regional income is obtained by multiplying the average household income and the total number of households, from the National Statistics maintained by Directorate-General of Budget, Accounting, and Statistics, Administrative Yuan. The nominal total income is converted into the real income in 2005 prices by dividing the GDP deflator. Therefore, this is a panel data set of 23 administrative regions in Chinese Taipei from 2004 to 2008.

The definitions and units of these input and output variables can be found in Table 1. Table 2 lists the correlation coefficients among these input and output variables. The isotonicity property that an output cannot decrease as an input increases is satisfied.

2.3 Results

Tables 3-7 list the SFA regression results on each logged inversed energy input. As the methodology section shows, the efficiency score obtained from the SFA model depicted in Eq. (4) is exactly the TFEE score for region i at time t .

The parameter γ is the proportion of variance in the inefficiency term (u) to the total variance of the error component term ($v - u$). As Tables 3-7 show, all estimated γ values are significantly different from zero and more than 0.9, indicating that the stochastic frontier analysis model is more appropriate than the ordinary least square method by taking into account the inefficiency term.

As Battese and Coelli (1992) define, the parameter μ is the parameter for a non-negative truncated distribution. An estimated μ which is not significantly different from zero implies that the inefficiency term u should follow the half normal distribution. On the contrary, an estimated μ which is significantly different from zero implies that the inefficiency term u should follow a truncated normal distribution. Tables 3-7 show that most of the inefficiency terms should follow truncated normal distributions.

According to Battese and Coelli (1992), the parameter η indicates the trend of time-varying inefficiency. A negative (positive) η implies that the efficiency is decreasing (increasing) over time. Therefore, productive electricity power usage and non-household lighting showed increasing inefficiency (i.e., worsening TFEE scores) over time. However, other kinds of energy usage showed no significant change in TFEE scores over time.

Note that the rankings in TFEE scores of the same energy source, obtained by the SFA models, are all stable over time. Most administrative regions in Chinese Taipei are not efficient in almost all kinds of energy use. The average TFEE scores in various energy sources during 2004-2008 are: productive electricity power (0.378566), household lighting (0.751216), non-household lighting (0.606920), gasoline (0.754872), and diesel (0.478735). It can be seen that the efficiencies of using productive electricity power and diesel are needed to improve very much.

The five most productive electricity power-efficient regions in Chinese Taipei are: Chiayi City (0.780437), Taitung County (0.781623), Keelung City (0.743171), Nantou County (0.569994), and Penghu County (0.561261). The five most productive electricity power-inefficient regions in Chinese Taipei are: Hsinchu City (0.088394), Tainan County (0.134991), Hsinchu County (0.145448), Taoyuan County (0.149119), and Yilan County (0.167073). Among them Hsinchu City and Tainan County have science parks while Hsinchu County, Taoyuan County, and Yilan County have large industrial parks. These regions are where many manufacturing industries locate.

Using the average TFEE scores, the five most household lighting-efficient regions in Chinese Taipei are: Tainan County (0.985176), Chiayi County (0.964069), Nantou County (0.903394), Yilan County (0.887618), and Yunlin County (0.885096). All of these top five lighting-efficient regions are non-metropolitan areas. Using the average TFEE scores, the five most household lighting-inefficient regions in Chinese Taipei are: Penghu County (0.534357), Taipei City (0.546529), Taichung City (0.569089), Chiayi City (0.590065), Taipei County (0.609240). These five most lighting-inefficient regions in Chinese Taipei include both metropolitan and non-metropolitan regions.

2.4 Conclusions

This paper applies the panel data stochastic production frontier to estimate TFEE scores for regions in Chinese Taipei, using the panel data during 2004-2008. The SFA approach extended from the cross-section SFA of Zhou et al. (2012) to a panel-data SFA. This paper is an effort to complement the work of the TFEE index proposed by Hu and Wang (2006) and Hu and Kao (2007).

Most administrative regions in Chinese Taipei are not efficient in almost all kinds of energy use. The average TFEE scores in various energy sources during 2004-2008 are: productive electricity power (0.6444), household lighting (0.875134), non-household lighting (0.895671), gasoline (0.846243), and diesel (0.672207). That is, the efficiencies of using productive electricity power and diesel need to improve very much.

Although here we use a different sample period and a different methodology, the empirical findings are generally consistent with Hu et al. (2012) who use the DEA approach to compute TFEE scores. Based on the

empirical findings, we come up with the following energy efficiency policy suggestions:

1. It is a priority for regions where manufacturing and high-tech industries concentrate to achieve more efficient use of productive electricity. Firms should be encouraged with incentive measures and administrative schemes to engage in cleaner production and adopt energy-conserving technology and equipments. The industrial sector can be restructured towards a high value-added and low energy intensive structure.
2. It is a priority for metropolitan regions to improve the efficiency of both household and non-household lighting. For these regions, energy-saving facilities such as LED bulbs should be especially encouraged.
3. It is a priority for all regions to improve the motor-vehicle energy efficiency in order to improve gasoline efficiency. Both of the central and local governments should put more efforts on retiring inefficient old motor vehicles, encouraging energy-saving vehicle models, providing convenient mass transportation systems to reduce the usage of private vehicles, and building the user-oriented and green-oriented municipal transportation environment.
4. It is a priority for non-metropolitan regions to improve the diesel use efficiency. Energy efficiency of farming machines and diesel trucks should be continuously improved especially for these regions. The local administrations of these regions should encourage these equipments to change into hybrid energy systems with financial incentives such as tax reductions and subsidies.

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EFFECTS OF WATER CONFIGURATION MODE ON THE FLOW PATTERN OPTIMIZATION OF ANAEROBIC DIGESTER

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ABSTRACT

The biogas plant is applicable to exploit renewable energy and carbon emission reduction technology to solve rural energy and environmental pollution, but there are defects such as poor liquidity, cause low biogas production efficiency, bad discharge, which limit the use widely. The key to solve this problem is to master the internal flow field of fermentation, optimize the flow pattern, and prevent the uneven flow distribution. In this paper, taking a 3L anaerobic digester as case, research the two water configuration modes of bottom agitation and dispersed exports by the software Computational Fluid Dynamics (CFD), analyzing on the influence of the optimal design on the flow pattern. It is surveyed that the digester can be optimized by velocity, pressure and flux design.

Key words: Biogas Engineering; CFD; Fluid pattern analysis.

1. INTRODUCTION

Biogas plants product high-quality energy thought digesting urban and rural organic wastes, meanwhile, collect the powerful greenhouse gas CH₄, is greatly benefit to environmental protection, energy and emission reduction. It is a well and practical technology to solve rural energy shortage and environmental pollution. According to the statistics of the Ministry of Agriculture of China, by the end of 2012 in China, 40,270,000 household anaerobic digesters were built up, and energy of 1,760,000 families are guaranteed by centralized supply plants, the annual total gas production of 13.08 billion cubic meters. But the biogas plants take the static fermentation process lacking enough manual intervention and effective mixing, which cause poor liquidity, low efficiency fermentation and difficult discharge, limit the wide popularization and format industrialization chain. The key to solve this problem is to analyze on the fermented liquid flow field, eliminate the static fermentation, optimize the flow field (Lai Z. M., 2011). This paper takes a cylindrical biogas fermentation tank whose bottom diameter is 0.8m and height is 1.25m as a case, simulate and calculate with the Computational Fluid Dynamics (CFD) software, describe the effect of bottom agitation and dispersed outlets on eliminating static fermentation and redistributing flow model, then discuss on and take prospect of the research and application of this technology.

2. SIMULATION OBJECTS AND MATHEMATICAL MODEL

2.1 Simulation object and condition

At first, a closed loop system is constructed, with each a hole in the bottom and side, the power is provided by a pump, forming a hydraulic agitation. The study object of this article is limited to the liquid portion of the biogas fermentation tank, without considering the rest part of gas and the closed loop system, so only the object material liquid part of 1m high as the simulation calculation, the model is simplified as a liquid column 1m high and two 0.1m diameter circular holes, as shown in fig. 1.

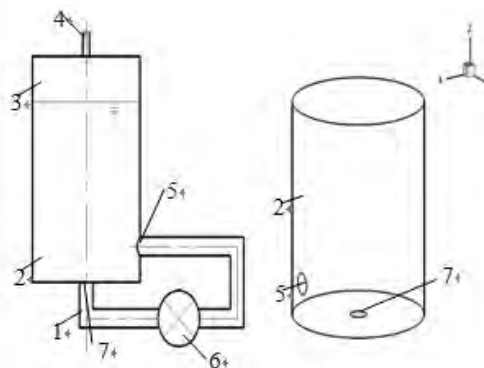


Figure 1 Biogas fermentation device closed cycle system and liquid model diagram

1. liquid pipe 2. liquid zoom 3. gas zoom 4. gas pipe 5. outlet 6. pump 7. inlet

2.2 Mathematical model

Using ordinary water as the medium, simulate the basic flow patterns in anaerobic and analysis on the rule of flow fields in diversified digesters.

The fluid flow continuity equation is as follow.

$$\frac{\partial(\rho u_x)}{\partial x} + \frac{\partial(\rho u_y)}{\partial y} + \frac{\partial(\rho u_z)}{\partial z} = 0 \quad (1)$$

The momentum equation is as follow.

$$\nabla \cdot (\rho \mathbf{u}) = -\frac{\partial \rho}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + \rho f_x \quad (2)$$

$$\nabla \cdot (\rho \mathbf{u}) = -\frac{\partial \rho}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{yz}}{\partial z} + \rho f_y \quad (3)$$

$$\nabla \cdot (\rho \mathbf{u}) = -\frac{\partial \rho}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \rho f_z \quad (4)$$

In the formula, the ∇ is Hamilton differential operator:

$$\nabla = i \frac{\partial}{\partial x} + j \frac{\partial}{\partial y} + k \frac{\partial}{\partial z} \quad (5)$$

2.3 Numerical methods

The control volume method is set up according to the grid, 1,000 grids on each side, total 3,383,839 grid elements are generated, the grid model as shown in fig. 2.

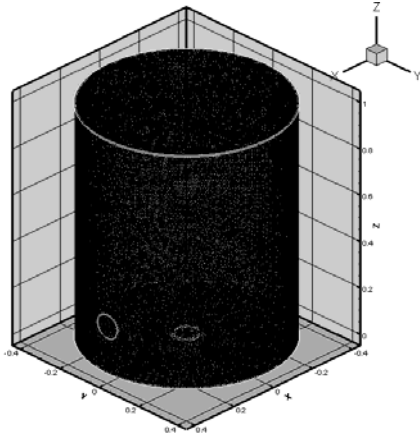


Figure. 2 Schematic diagram of fermentation liquid part of the grid

3. SELECTION OF VELOCITY ENTRANCE DIRECTION

3.1 Agitating mode choose

According to research of Qiu (2005) at Northwest Agriculture and Forestry University, annual biogas production of dynamic fermentation increased by 31.0%~72.4% contrast to static fermentation.

According to the research overview of Yang et al (2010), mixing is clearly beneficial to biogas anaerobic digestion. But on the agitating mode, there is some controversy. Karim et al (2005) compared the impeller mechanical agitation, hydraulic agitation and pneumatic agitation in three ways, pointed out that while the concentration was low, the effect of each agitating modes on enhancing the biogas production rate was not obvious, but while the concentration of TS (Total Solid) reached 10%, the biogas production rates of three agitating modes were respectively higher than no agitating by 22%, 29% and 15%, illustrated the hydraulic agitation is perhaps the best agitating mode. About the agitating position, it mainly exists a controversy of two opinions: lateral inserted agitation or bottom agitation (Cai et al, 2001; Chen 2004). In this paper, the velocity entrances of hydraulic agitation are respectively set to the lateral and bottom, simulate the fluidization effect with CFD, and further test the optimal model in order to make the comparative analysis.

3.2 The pattern of lateral inflow and bottom outflow

Set the boundary conditions for the inlet velocity as 1m/s. Set the gravity acceleration as 9.81m/s², through the simulation results, shown in fig. 3, the velocity distribution is not uniform seriously, the hydraulic path between the inlet and outlet is very obvious, the velocity of remaining zoom especially high zoom is very low.

Intercept two sections of 0.2m and 0.8m height in order to compare, through the Fig. 4, the velocity of high zoom is far less than the low. Perhaps it is due to the acceleration of gravity is perpendicular to the initial velocity, so the deviation of acceleration is very serious.

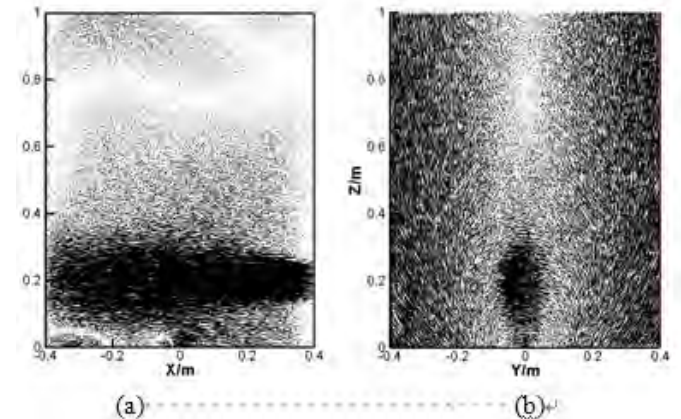


Figure. 3 lateral inflow, the velocity vector diagrams of y=0 and x=0 sections

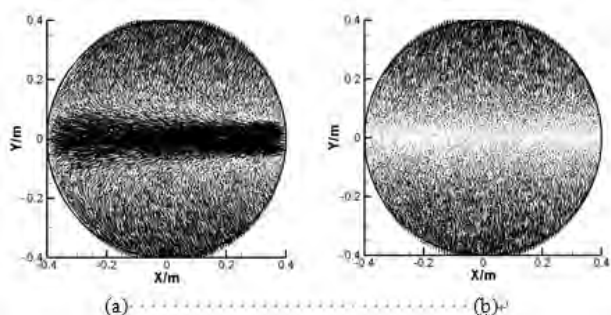


Figure. 4 lateral inflow, the velocity vector diagrams of $z=0.2$ and $z=0.8$ sections

3.3 The pattern of bottom inflow and lateral outflow

Exchange the velocity inlet and pressure outlet, which form the bottom inflow, lateral outflow. Set boundary condition, take the acceleration of gravity and the other operating conditions as the same as 2.2. Through the simulation results, shown in Figure 5, the velocity distribution is still not very uniform, but significantly improved than the last agitating mode, especially in the high zoom. Through the simulation results, shown in fig. 6, the velocity gap between two sections height of 0.2m and 0.8m is far less than the last agitating mode, perhaps it is due to the acceleration of gravity directs to initial velocity in the vertical direction, so doesn't cause serious deviation of flow in one side of a direction.

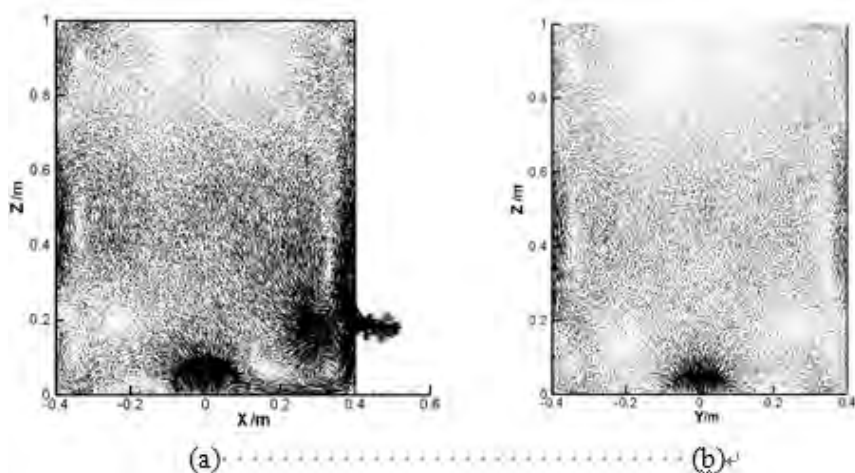


Figure. 5 bottom inflow, the velocity vector diagrams of $y=0$ and $x=0$ sections

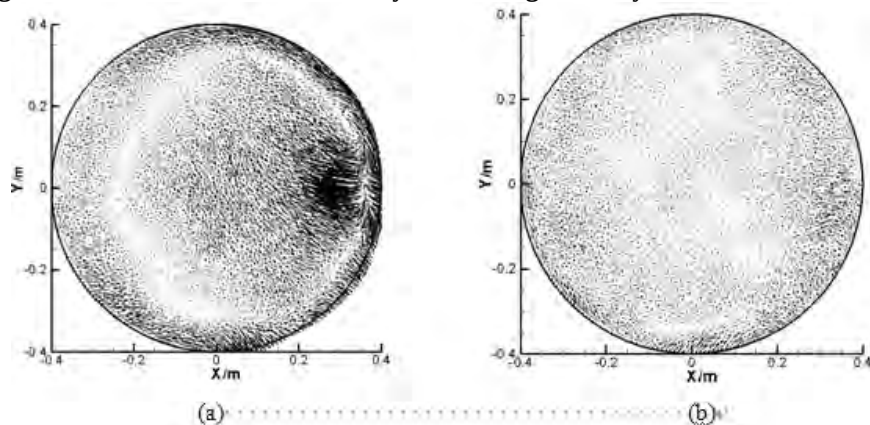


Figure. 6 bottom inflow, the velocity vector diagrams of $z=0.2$ and $z=0.8$ sections

3.4 Bottom inflow is the best agitating pattern

Through the comparison of the above two kinds of water inflow direction, obviously, the flow field distribution of bottom inflow is more uniform, the influence range is larger. So, about the choice of

agitating mode, bottom agitation is better than lateral inserted agitation.

4 OPTIMAL DESIGNING OF PRESSURE OUTLETS

4.1 Design of dispersed exports

Through the simulation results in the previous, inlet is too close to outlet, easy to form a short single hydraulic path, resulting in liquid retention time is short, washing zoom is little, flow velocity of the zoom higher than the lateral pipe reduce sharply, most close to 0, indicate that the fluidization effect in the total zoom is not ideal. So, change a new design, set the water outlet pipe from bottom up to the 0.8m high, and design 4 exports, so it is expected to form a plurality of circulation loop in the zoom, redistribute velocity, pressure and flow, as shown in figure 7.

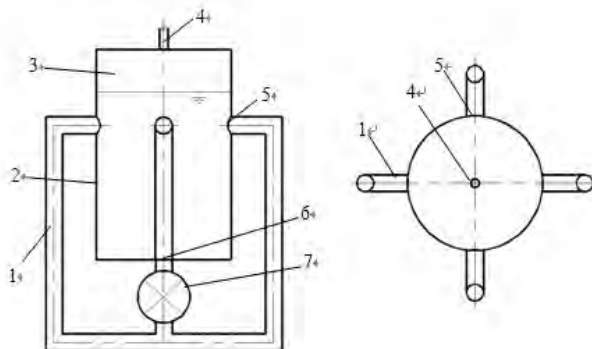


Figure. 7 distributed pressure outlet design schematic diagram

1. liquid pipe 2. liquid zoom 3. gas zoom 4. gas pipe 5. outlet 6. inlet 7. pump

4.2 Basic flow field simulation and optimal design

Simulate the flow field of bottom inflow model according to the optimal design. The control volume method is still set up according to the grid, 300 grids on each side, total 2,921,029 grid elements are generated, the grid model as shown in figure 8.

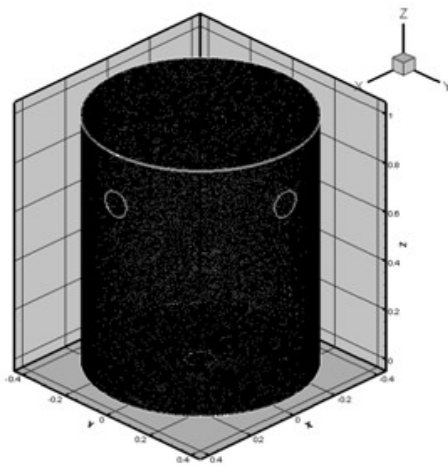


Figure. 8 schematic diagram of dispersed exports model grid

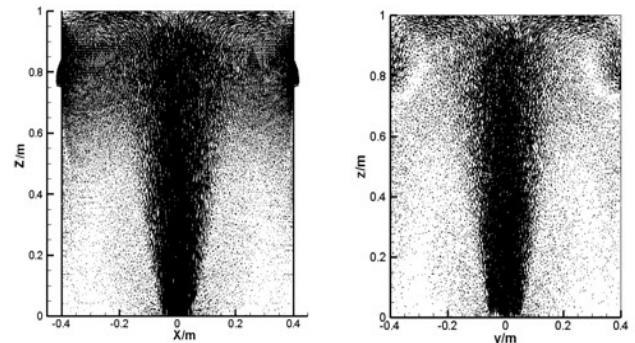


Figure. 9 dispersed exports, velocity vector diagrams of $y=0$ and $x=0$ sections

Set the initial condition for the inlet velocity as 1m/s, acceleration of gravity as 9.81m/s². Through the simulation results, shown in Figure 9, the higher zoom acquires more kinetic energy than the original design, the velocity distribution is much more uniform.

Through the simulation results in Figure 10, the velocity of the 0.8m high section is larger than 0.2m section, and their distribution is very uniform. In this case, while agitation is stop, liquid will sink along the direction of gravity; while agitation restarts, it rises up again, repeats like this, form tumbling, is extremely beneficial to the uniform mixing.

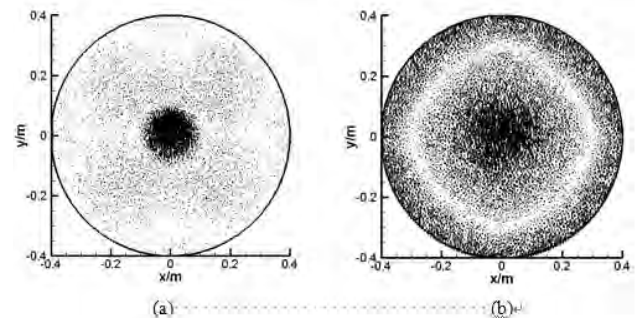


Figure. 10 dispersed exports, velocity vector diagrams of $z=0.2$ and $z=0.8$ sections

Through the simulation results in Figure 10, the height is 0.8m cross section flow ratio of height 0.2m section is higher, and their distribution is very uniform. In this case, stop stirring, liquid will sink along the direction of gravity; restart up again after mixing, so repeatedly, form tumbling, are extremely beneficial to the uniform mixture.

5 CONCLUSION

As hydraulic agitation model, the direction of the inflow can't parallel with the gravity such as vertical that will cause the acceleration deviates from the vertical direction and result in uneven distribution of flow field. On the one hand, it could avoid the unparallel of the inflow and the gravity, on the other hand, it can form tumbling at the time of intermittent. Therefore, it is reasonable to design the velocity of entrance oppositely to the direction of the gravity acceleration.

In general biogas plant, the outlet is designed to close the bottom of the anaerobic digester in order to discharge residua unhindered. As hydraulic agitation model, the pressure outlet should be designed higher in order to avoid flow field uneven and expand the influence range of hydraulic agitation, and promote mixture of all zoom.

It can form a plurality of pressure outlets by discharging pipe designing, and form a plurality of hydraulic circulation pathways in the anaerobic digester which cross each other. It can greatly optimize the flow field in digester by redistributing velocity, pressure and flux.

ACKNOWLEDGEMENTS

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A WASTE HEAT UTILIZATION TECHNOLOGY OF DEEP COAL MINE RETURN AIR

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ABSTRACT

Mine ventilation system is an important guarantee for the safety production. By ventilating, not only the fresh air would be supplied to the miner underground, but also the residual heat and harmful gas underground would be discharged into the atmosphere. As the temperature is about 15°C-25°C and the air flow rate is relatively higher and more stable, mine return air (MRA) is a high quality waste heat resource to the WSHP. The working principle of a combined system which consists of a MRA heat recovery system and a water source heat pump (WSHP) system is presented. In the MRA heat recovery unit, heat and mass would exchange between MRA and water droplets atomized by nozzles installed in the MRA heat exchanger. It is found that there are four primary differences between the vertical spray MRA heat exchanger which is the key equipment of the combined system and the water spray chamber used in textile air-conditioning in the aspects of the application objective, the moist air handling processes, the parameters of inlet air and the structure requirements of the heat exchanger unit. An actual application has been presented and the parameters of air and water in MRA heat recovery unit in two operation modes of WSHP have been tested. It is found that the heat transfer efficiency E and the enthalpy efficiency X of MRA heat recovery unit in heating mode are higher than that in the cooling mode and the contact coefficient E' are all equal to 1 in the two modes. Moreover, the ratios of sensible heat transfer rate to the latent heat transfer rate are also different in the two modes. Finally, the application effects in a typical mine when the WSHP is running in a standard mode show that the total heat transfer rates of the MRA heat recovery unit are remarkable, 5479 kW in cooling mode and 3823 kW in heating mode respectively.

Key words: Mine return air; Waste heat recovery; Heat and mass transfer; Efficiency; Application

1. INTRODUCTION

Mine ventilation system is an important guarantee for the safety production. By ventilating, not only the fresh air would be supplied to the miner underground, but also the residual heat and harmful gas underground would be discharged into the atmosphere. Figure 1 shows a typical mine ventilation system^[1]. As shown in Figure 1, the fresh air is sent to working face underground along the downcast shaft meanwhile heat and humidity are exchanged among airflow, rock wall, equipment and workers, and then the air at higher temperature laden with dust particles is discharged into atmosphere through the diffuser installed on the outlet of return air shaft. The function of a diffuser is to enhance the main fan's effective power and reduce kinetic energy loss.

In general, the temperature and humidity of MRA is stable all the year and the airflow rate is high (about 50-500m³/s). MRA is a high-quality waste heat source. The temperature range of MRA is about 15°C-25°C, the Relative Humidity (RH) is near 100%^[2], and the parameters is relatively stable throughout the year,

which contributes MRA to be a proper heat source for the Water Source Heat Pump (WSHP) system.

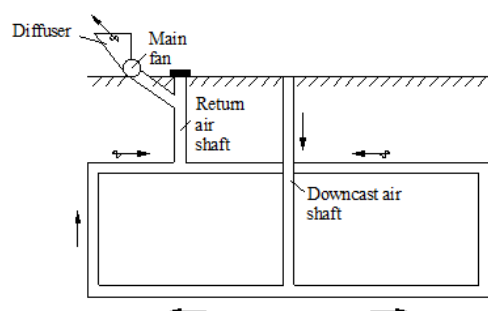


Figure 1 A typical MRA ventilation system

There have been a few examples of the successful use of mine water in WSHP systems^[3-5]. However, at present, there are limited examples and researches of extracting heat from the exhaust mine air. In 1994, a mine air-assisted heat pump system which was used in greenhouse heating was firstly presented^[6]. In the following years, various types of heat exchangers such

as the plate type, the heat pipe or the spray chamber have been proposed for MRA heat recovery^[7,8]. Furthermore, the application feasibility of MRA heat recovery system combined with WSHP system was theoretically analyzed^[2, 9]. By absorbing heat from MRA or releasing heat to MRA, the WSHP system would meet heating and cooling demands of the buildings on the ground, and supply heat for mine shaft anti-freezing in winter and hot water for miner bathing in all year. Sbarba HD^[10] et al. have developed computer software to evaluate the economics of MRA heat recovery system for heating fresh air intake during cold periods.

The heat exchanger is the key equipment of the MRA heat recovery system due to its heat transfer efficiency between the MRA and the water from WSHP would influence the application effect of MRA heat recovery system. Compared with the wall type heat exchanger, the direct-contact heat exchanger possesses higher heat transfer efficiency which would contribute to its wide application in water spray chamber, cooling tower and evaporative cooler, etc.. However, considering the security of mine ventilation system, the direct-contact heat exchanger cannot be filled with packing otherwise the pressure loss of heat exchanger would be higher. Moreover, the air velocity in diffuser may be more than 25m/s which would lead to serious drift water loss; therefore, a remodeling diffuser has been designed by the authors^[11]. The authors also designed a vertical spray MRA heat exchanger which installed on the remodeling diffuser^[12], and the working principle is illustrated in “Section 2”.

Until now, there are little studies on the thermal performance of MRA heat exchanger and the actual application effects of MRA heat recovery system combined with WSHP system and etc.. Firstly, the paper presents a working principle of the combined system, four differences between the vertical spray MRA heat exchanger and the traditional water spray chamber, and the evaluation indexes on heat and mass transfer efficiency. Additionally, an actual application case of MRA heat recovery system is presented; the thermal performances of MRA heat exchanger and the application effects are tested and analyzed. Finally, the application effects in a typical mine in a standard mode of WSHP are analyzed.

2. SYSTEM OVERVIEW

2.1 Working principle of MRA heat recovery system

A basic MRA heat recovery system consists of a MRA heat exchanger and a diffuser (called MRA heat recovery unit), the power equipment, a collecting basin, a storage water tank and accessories. In most cases, a MRA heat recovery system would be combined with a WSHP system.

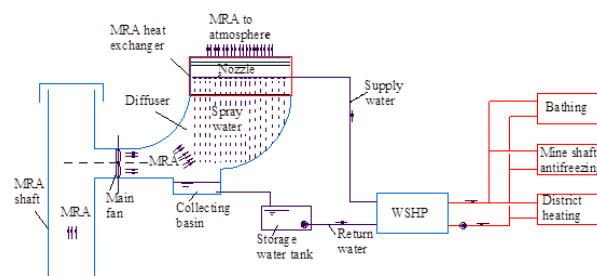


Figure 2 Schematic diagram of the MRA heat recovery combined system (in heating mode)

The basic scheme of the MRA heat recovery system combined with a WSHP system is presented in Figure 2. MRA is delivered to the remodeling diffuser from the MRA shaft driven by the main fan; in the remodeling diffuser, lower velocity, well-distributed and steady airflow is obtained. Then, in the MRA heat recovery unit, heat and mass would exchange between MRA and water droplets atomized by nozzles installed in the MRA heat exchanger. Ultimately, the MRA is discharged to the atmosphere and the water droplets drop into the collecting basin and then the collected water would be pumped to the WSHP. When the WSHP is running in heating mode shown in Figure 2, the chilled water is conveyed to the MRA heat exchanger and absorbs heat from MRA. Oppositely, when the WSHP is operating in cooling mode, the cooling water is delivered to the MRA heat exchanger and rejects heat to MRA.

2.2 Differences between the vertical spray MRA heat recovery unit and the spray chamber

A vertical spray MRA heat recovery unit is similar to a vertical spray chamber and a shower cooling tower^[13] in which both moist air and water droplets are in counter-flow and exchange heat and mass mainly in vertical direction. However, there are several remarkable differences in aspects of application objectives, moist air handling processes, parameters of inlet air, and structure requirements of heat exchanger unit between them.

2.2.1 application objectives

In a textile mill, the function of a spray chamber in textile air-conditioning is mainly to cool, dehumidify or humidify the return air or the mixture of return air and fresh outdoor air. The outlet air's RH of the spray chamber is about 90% which would meet the temperature and high humidity requirements of textile mill.

In MRAHR system, the function of a MRA heat recovery unit is to absorb heat from the cooling water of WSHP which is operating in cooling mode or reject heat to the chilled water in heating mode so as to obtain the proper water supply parameters for WSHP.

2.2.2 the moist air handling processes

Generally, in a spray chamber, there are two typical moist air handling processes—air isenthalpy

humidifying process in winter and air cooling and dehumidifying process in summer.

However, in MRA heat recovery unit, there are also two primary moist air handling processes corresponding to the two operating modes of WSHP. In heating mode, the moist air handling process in MRA heat recovery unit is similar to the air cooling and dehumidifying process in a spray chamber in which heat is ultimately removed from the warm air to the water. Oppositely, in the cooling mode, air is heated and humidified while water is cooled in MRA heat recovery unit, and both sensible heat and latent heat which is removed through evaporation from the water to the air are transferred from the water to the air.

2.2.3 parameters of inlet air

The inlet air's RH of the spray chamber is less than 75%, after heat and mass exchanging, it would reach to nearly 90%^[14]. In most cases, the inlet air is the mixture of outdoor air and return air, therefore the mixture temperature is near the return air's temperature (about 15-30°C) because the ratio of the fresh air flow rate to the return air flow rate is small^[14]. Therefore, the cooling load of spray chamber will increase along with the increasing of temperature and RH of inlet air in order to obtain the designing parameters of outlet air.

The temperature of inlet air in MRA heat recovery unit is about 15-25°C which depends on the mining depth and geological condition of the mine, and the RH is higher than 90% and close to 100%. Unlike in the spray chamber where the objective of heat and mass exchanging is to obtain the proper parameters of outlet air, the target in a MRA heat recovery unit is to obtain the water with the proper temperature by exchanging heat and mass with MRA. In the MRA heat recovery unit, much larger heat capacity exchanged is expected. Therefore, in heating mode, the heat capacity absorbed by water from MRA increases along with the increase of MRA temperature and RH on the condition that the water temperature is constant.

In cooling mode, the MRA heat recovery unit operates as a cooling tower where the cooling water rejects heat to the air. However, the impetus of heat and mass transfer is larger in MRA heat recovery unit than in a cooling tower. Taking Beijing city as an example, the outdoor air design dry-bulb and wet-bulb temperature for summer air conditioning is respectively 33.2°C, 26.4°C (the RH is about 57%), supposing MRA temperature is 20°C, RH is 90%, and the temperature of cooling water t_w is 35°C, the representation of moist air is shown in Figure 3. Figure 3 shows that, the enthalpy difference between saturated air boundary layer around the water droplet and the MAR (Δh_{w-MRA}) is larger than that between the saturated air boundary layer and the outdoor air (Δh_{w-a}). Obviously, the heat exchange capacity in MRA heat recovery unit is higher than that in the cooling tower as the heat exchange capacity will increase along with the decreases of RH and temperature of the inlet air.

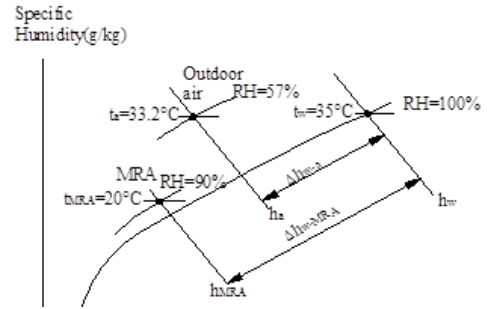


Figure 3 The enthalpy differences in the psychrometric chart

2.2.4 structure requirements of heat exchanger unit

When designing the structure of a vertical spray chamber, it is mainly to calculate its section size and height considering airflow rate and nozzle arrangements. However, it is more complex to design a MRA heat recovery unit. Besides the elements when a spray chamber is designed, the dynamic performance of the diffuser where heat and mass transfer conducts must be taken into account, such as kinetic energy recovery performance, pressure drop, uniformity of outlet air velocity distribution, etc.^[15].

2.3 Evaluation indexes on the performance of heat and mass transfer

In China, the thermodynamic calculation of a spray chamber is based on two semi-empirical formulas for overall heat transfer efficiency E and contact coefficient E' which is generally used to evaluate the heat and mass transfer performance of a spray chamber^[16].

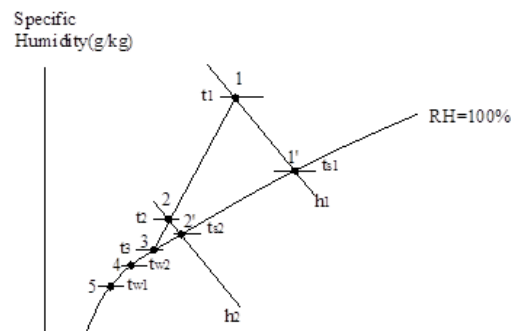


Figure 4 The cooling and dehumidifying process of moist air in psychrometric chart

Take moist air cooling and dehumidifying process as an example. As shown in Figure 4, the definition formulas of E and E' are presented as following:

$$E = \frac{(1'2' + 45)/1'5}{1'5} = 1 - (t_{s2} - t_{w2}) / (t_{s1} - t_{w2}) \quad (1)$$

$$= 1 - (h_2 - h_{w2}) / (h_1 - h_{w1})$$

$$E' = \frac{12/13}{1'2'/1'3} = 1 - 2'3/1'3 \quad (2)$$

$$= 1 - \frac{22'1}{11'1} = 1 - (t_2 - t_{s2}) / (t_1 - t_{s1})$$

Where t_1 and t_2 are the dry bulb temperatures of inlet and outlet air; t_{s1} and t_{s2} are the wet bulb temperatures of inlet and outlet air; t_{w1} and t_{w2} are the temperatures of inlet and outlet water; h_1 and h_2 are the enthalpies of inlet and outlet air; h_{w1} and h_{w2} are the equivalent enthalpies of saturated air at t_{w1} and t_{w2} .

If the inlet air approaches to saturation, probably, the outlet air would reach saturation in the MRA heat recovery unit. According to Eq. (2), the contact coefficient E' might equal 1, which would be negative to evaluate the heat and mass transfer efficiency. Therefore, the enthalpy efficiency which is defined as the ratio of actual change over ideal maximum change between the inlet and outlet air enthalpy^[17] is used to describe heat and mass transfer performance in the American, Japan and Switzerland etc.^[18]. The expression of enthalpy efficiency is shown in Eq. (3).

$$X = (h_1 - h_2) / (h_1 - h_{w1}) \quad (3)$$

3. AN ACTUAL ENGINEERING APPLICATION

Here an actual application of MRA heat recovery system in Shuguang coal mine in Shanxi province where the MRA temperature is lower is presented. The purpose of application testing is to investigate the efficiency of heat and mass transfer and the total heat transfer capacity of MRA heat recovery unit in cooling mode and heating mode.

3.1 The project overview

The central heating area of buildings on the ground in Shuguang coal mine is about 36500m². In addition, the whole year heating load for miner bathing is 678kW. 6 WSHPs whose nominal heating capacity is 652.2kW are installed, and one of them supplies heat water for bathing all year round. Two main fans are installed, one of which is the standby. Correspondingly, two diffusers and two MRA heat exchangers are installed.

3.2 Methods

It is found that the MRA flow rate of the mine, the MRA temperature and RH are stable all day even a whole month and not adjustable. Consequently, the testing is conducted while both the air and the water flow rate are constant. The MRA flow rate (G) of main

fan has been monitored automatically per minute which can be obtained directly. Dry and wet bulb temperatures of MRA are measured by dry and wet bulb thermometers. Mercurial thermometers are used to measure the temperatures of the inlet and outlet water. The distributions of measurement points (t_1 t_2 t_{s1} t_{s2} t_{w1} t_{w2}) are shown in Figure 5. The testing results are required to be recorded when the system runs stably.

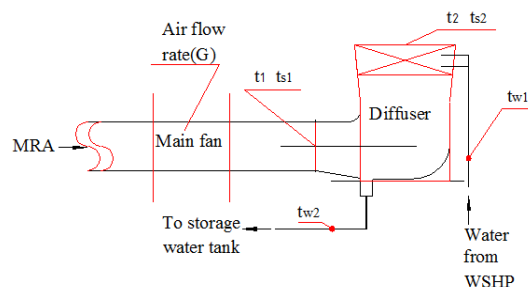


Figure 5 The distributions of measurement points

3.3 Results and discussion

3.3.1 parameters of MRA

On August 24th, 2011, the heat and mass transfer performance of MRA heat recovery unit was tested when the MRA heat recovery system was running in cooling mode. The testing results show that the outdoor air dry bulb temperature and wet bulb temperature are respectively 20.6°C, 18°C at 9:42 o'clock a.m. The monitoring data shows that the MRA flow rate is about 123.7m³/s that day, and the deviation between the maximum and minimum is less than 1%. Similarly, the dry bulb temperature t_1 and wet bulb temperature t_{s1} of MRA almost have no variation which are, respectively, 19.6°C, 19.3°C.

Accordingly, a similar test was conducted on December 20th, 2011, when the MRA heat recovery system was running in heating mode. The testing data is shown in Table 1.

Compared with the MRA parameters in summer, the air flow rate in winter is increased by 25.6m³/s and the temperature is decreased by about 5°C. However, both of them have changed little all day long. Thus, the inlet air parameters of MRA heat recovery unit can be supposed constant when testing is conducted on the day.

Table 1 The parameters of MRA in the MRA heat recovery unit (in heating mode)

Outdoor air		MRA (the inlet air of MRA heat recovery unit)		
Dry bulb temperature(°C)	Wet bulb temperature(°C)	Dry bulb temperature(°C)	Wet bulb temperature(°C)	Air flow rate (m ³ /s)
0.20	-2.30	14.93	14.67	149.29

3.3.2 the testing results of t_2 , t_{s2} , t_{w1} , t_{w2}

As the outdoor air temperature is low, the cooling load of buildings at the mine may be small and the WSHP may be closed. However, in order to test the heat and mass transfer performance of MRA heat recovery unit, three WSHPs are launched. When the

MRA heat recovery combined system ran stability, we begin to record the data of t_2 , t_{s2} , t_{w1} , t_{w2} , which are shown in Figure 6. As shown in Figure 6, the average values of t_2 , t_{s2} , t_{w1} , t_{w2} are, respectively, 20.8°C, 20.8°C, 22.86°C, 20.9°C. Compared with the inlet air, the dry bulb temperature of outlet air (t_2) increases due to the

MRA absorbs heat from the hot water from WSHP. Because the MRA is humidified in heat exchanger, the outlet MRA almost reaches saturation, the t_{s2} line is basically coincident with the t_2 line, as shown in Figure 6. Correspondingly, the hot water from WSHP is cooled by releasing heat to the MRA, thus the t_{w2} is decreased by about 2°C.

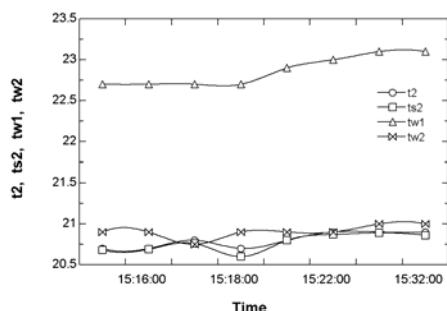


Figure 6 The testing results of t_2 , t_{s2} , t_{w1} , t_{w2} in cooling mode

Accordingly, the testing was conducted when the WSHP are running in heating mode in the winter of 2011. The average values of t_2 , t_{s2} , t_{w1} , t_{w2} are, respectively, 13.2°C, 13.2°C, 11.5°C, 13.17°C. As shown in Table 2, t_1 of inlet air equals 14.93°C, t_{s1} of inlet air equals 14.67°C, then the dew point temperature of the inlet air can be obtained, which equals 14.58°C. As t_{w1} equals 11.5°C which is lower than the dew point temperature of the inlet air, the moist air is dehumidified and cooled, which is verified by the outlet air parameters. The testing results also show that the outlet air reaches to saturation whose dry bulb temperature t_2 is equal to its wet bulb temperature t_{s2} . Simultaneously, due to absorb heat from the moist air, the temperature of outlet water increases from 11.5°C to 13.17°C.

3.4 E, E', X

Table 2 The E, E' and X of MRA heat recovery unit in the two processes

	E	E'	X
The air heating and humidifying process	0.972	1	0.41
The air cooling and dehumidifying process	0.987	1	0.481

According to the testing results analyzed in "Section 3.3.1" and "Section 3.3.2" and the Equations (1) to (3) shown in "Section 2.3", the E, E' and X are calculated and the results are shown in Table 2. From Table 2, it can be seen that, in air cooling and dehumidifying process, the E and X are higher than that in the heating and humidifying process. The E's in the two processes are all equal to 1, which verifies the previous analysis in "Section 2.3".

3.5 Heat transfer rate

The total heat transfer rate Q_t in the MRA heat recovery unit is the sum of the sensible heat transfer rate Q_s and the latent heat transfer rate Q_l . Sensible heat transfer will occur when temperature difference exists. Q_l is the heat absorbed by the water when it evaporates or the heat rejected by the water vapor when it condenses. Q_t , Q_s and Q_l can be calculated as follows:

$$Q_t = G(h_2 - h_1) \quad (4)$$

$$Q_s = Gc_p(t_2 - t_1) \quad (5)$$

$$Q_l = Q_t - Q_s \quad (6)$$

Where G is the air flow rate and c_p is the specific heat at constant pressure of the air.

Based on the Equations (4) to (5) and the testing results shown in "Section 3.3.1" and "Section 3.3.2", Q_t , Q_s and Q_l between MRA and water in the two modes can be calculated which are shown in Table 3. Table 3 shows that Q_t in the heating mode is lower than that in the cooling mode although G is higher. This can be explained as the enthalpy difference of cooling mode is larger than that in the heating mode if the temperature differences between the inlet air and the inlet water in the two modes are basically the same. In the cooling mode, Q_l occupies a larger proportion of Q_t which is about 75.3%. However, there is small difference between Q_s and Q_l in heating mode; Q_s/Q_t and Q_l/Q_t are 44.80% and 55.20%, respectively. Moreover, Q_s in the two modes are relatively small. The main reason is that the enthalpy difference is small as the heating load or the cooling load is small on the test-day, which leads to the inlet water temperature being too low in cooling mode or too high in heating load.

Table 3 Q_t , Q_s and Q_l in the two modes

	Q_t (kW)	Q_s (kW)	Q_s/Q_t (%)	Q_l (kW)	Q_l/Q_t (%)
Cooling mode	727.36	179.91	24.70	547.45	75.30
Heating mode	698.68	313.03	44.80	385.65	55.20

4. APPLICATION ANALYSIS IN A TYPICAL MINE

It is found that the MRA temperature in the province such as Hebei and Henan province is generally higher than that in Shanxi province. According to the analysis in "Section 3.3.4", the higher temperature of MRA would contribute to increasing the heat transfer rate in the heating mode which would be adverse in cooling mode. Take Suncun mine of Xinwen Mining Group in Shandong province whose MRA parameters are shown in Table 1 as an example. Supposing that the MRA heat recovery combined system was installed in the mine and the X of the MRA heat recovery unit was equal to the calculated results shown in Table 3, according to the Equations (3) and (4), Q_t can be calculated as follows:

$$Q_t = G(h_{w1} - h_1)X \quad (7)$$

Where $G=167\text{m}^3/\text{s}$, $X=0.41$ in cooling mode, $X=0.481$ in heating mode, $h_1=62.32\text{kJ}/(\text{kg K})$ ($t_1=22^\circ\text{C}$, $\text{RH}=95\%$), $h_{w1}=129\text{kJ}/(\text{kg K})$ ($t_{w1}=35^\circ\text{C}$ in cooling mode), $h_{w1}=22.66\text{kJ}/(\text{kg K})$ ($t_{w1}=7^\circ\text{C}$ in heating mode). After calculated, Q_s in cooling mode and heating mode between MRA and water in MRA heat recovery unit, respectively, equal 5479kW and 3823kW .

CONCLUSIONS

In this paper, three conclusions were obtained as followed:

- (1) In the cooling mode and the contact coefficient E' are all equal to 1 in the two modes.
- (2) The ratios of sensible heat transfer rate to the latent heat transfer rate are also different in the two modes.
- (3) The application effects in a typical mine when the WSHP is running in a standard mode show that the total heat transfer rates of the MRA heat recovery unit are remarkable.

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APPLY OF LOW-CARBON TECHNOLOGY IN BUILDING ENERGY CONSERVATION IN HOT SUMMER AND COLD WINTER AREA

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ABSTRACT

Building energy conservation is the basis for carbon emission reduction, through elaborating the relationship between low carbon and energy efficient building, pointing out that the construction of energy-saving emission reduction is an effective means to solve the problems of high energy consumption of the building, and it is also an important measure for China's carbon emission reduction. According to the climate characteristics in hot summer and cold winter area, low carbon technology suitable for the construction of energy-efficient hot summer and cold winter area is proposed which is based on the analysis of the current main building energy-saving technical measures.

Keywords: building energy conservation, low-carbon building, renewable energy

1. INTRODUCTION

December 19, 2009, "the United Nations Framework Convention on climate change" conference, which is held in Copenhagen, shows that although countries all over the world hold different opinions on strategies in response to climate change and the distribution of reduction of greenhouse gas emission, the subjects to reduce greenhouse gas and energy conservation are same when facing the rigidity reality of global warming, considering the average global temperature rise of 2 °C as an acceptable "safe baseline". The Chinese government solemnly promised that, by 2020, carbon dioxide emissions decreased by 40% ~ 45% per unit of GDP compared with in 2005. To reduce large-scale carbon emissions in such a short period of time, hard work and efforts are necessary, and it requires different fields to positively participating in this task. In reducing emissions, people should be aware that the building emissions also account for a considerable role. According to statistics, the building energy consumption (including building energy consumption, heating and air conditioning energy consumption, life energy consumption etc.) accounted for about 30% of the total energy consumption of the whole society, of which the most important is the heating and air conditioning, and it accounted for 20%. Thus, the construction of energy-efficient buildings can achieve low carbonization. That is, employing low-carbon technology to realize energy-saving in the entire life cycle of a building [1].

2. BUILDING ENERGY CONSERVATION AND LOW-CARBON BUILDING

Building energy conservation refers a way, through adopting the plane form and the overall layout suitable for local climate conditions in the architectural design process, to achieve the most conducive method for energy-saving, comprehensively considering a rational use and disposal of resources in the building design, construction and material selection, reducing the waste of resources, which aims at renewable resources. The low-carbon building means that it can diminish the amount of emission of carbon dioxide through reducing the use of fossil fuel and improving its efficiency in the process of building designing, equipment manufacturing, construction and its entire life cycle. Compared with the other green residential concept such as green real estate and other sustainable real estate, low carbon residential more emphasizes on emissions of greenhouse gas - carbon dioxide caused by energy consumption.

To achieve a substantial reduction of low carbon buildings is mainly reflected in two aspects: the first one is to reduce emission of carbon through decreasing the total housing construction materials and choosing specific kind of materials, for example, the production process of wood emits less carbon dioxide than it is by steel; the other one is conservation of water, for example, running water production and waste water treatment will increase the carbon dioxide emissions, thus it is necessary for water conservation and recycling. The low carbon building has gradually

become the main trend of international architecture industry, and low carbon building will also become one of the mainstream of Chinese architecture. Now China is moving in this direction. The low carbon building mainly includes two aspects: 1. low carbon materials; 2. low carbon building technology [2].

3. IMPORTANT ROLES OF BUILDING ENERGY CONSERVATION IN CARBON EMISSION REDUCTION

As statistics show, about 0.8 tons of carbon releases from every 1 square meters of housing in China. In addition, the real estate development process involves building heating, air conditioning, ventilation, lighting and other forms of the energy, and results in a great carbon emission. Therefore, to effectively control and reduce building carbon emission and establish a recycling sustainable format, we should introduce more low-carbon and green building project, realize innovation in building energy-saving technology and establish a low building carbon emission system. Finally, the energy saving buildings effectively and meet the corresponding standards, China's real estate industry is on the route one must take the healthy development, but also developers are duty-bound responsibility.

According to statistics, carbon emissions from energy consumption of the building are located only in the industry and traffic, in the third row. According to the management consulting company McKinsey analysis report, the construction field of building energy efficiency in reducing carbon emissions, is the most easy to implement and is a negative cost measures. In the McKinsey Co completed a research report "greenhouse gas emission reduction cost curve" of a proposed carbon reduction measures, more than half of the measures are building energy-saving technology or construction belongs to negative cost measures, through the construction of energy-saving and carbon reduction is a good measure. So, should give priority to the development of building energy-saving in carbon reduction measures, one of the basic building energy conservation and the development of low carbon city.

4. LOW CARBON TECHNOLOGY SHOULD APPLY IN ENERGY-SAVING BUILDING IN HOT SUMMER AND COLD WINTER ZONE.

Building energy-saving measures in hot summer and cold winter area many of which are low carbon emissions or zero emissions, such as ground source heat pump underground heat collection, rainwater collection and utilization of photovoltaic system, solar lighting, through the light pipe will introduce indoor sunlight lighting etc. Following to introduce of the building energy-saving measures for low carbon emissions or zero carbon emissions in hot summer and cold winter zone.

4.1 The energy-saving technology in building enclosure structure

At present, China's housing unit area heating and air conditioning energy consumption is 2 ~ 3 times the consumption in developed countries which have similar climatic conditions, building envelope thermal performance is poor, building energy conservation has become the first step in improving the efficiency of energy use in the whole society. Therefore, vigorously promote the construction of energy-efficient is conducive in improving energy efficiency and energy conservation, and it is an important measure to ensure national energy security and build a low carbon economy society.

4.1.1 Window energy-saving technology

In hot summer and cold winter zone in China, building doors and windows is an important channel to absorb solar radiation in winter, and they are also one of the main causes of indoor overheating, thus, the point is to reduce the permeability and structure of heat transfer if we struggle to meet the needs of the premise and energy-efficient doors and windows and improve the insulation performance. Reduced infiltration can resolved mainly by setting the seal, the volume of cold infiltrating into indoor space is determined by window pressure and thermal pressure on both sides of doors and windows. The key of controlling cold air infiltration is to set the seal and increase the air tightness, and it can help to save energy by 10% ~ 15% of setting the seal. For heat transfer in enclosure structure, glasses listed below are used more popularly: hollow glass, coated glass (including reflective glass, heat absorbing glass) of high strength LOWE fireproof glass (high strength low radiation coated fire-resistant glass using magnetron sputtering method), vacuum plating metal silver layer of glass and the most special smart glass.

4.1.2 Roof energy saving technology

The energy-efficient constructions are achieved by intelligent technology and ecological technology, such as solar energy roof and controlled ventilation and insulation roof. Solar energy with its unique advantages is the ideal alternative energy, its development and utilization is the ultimate solution to resolve problems like energy shortages, environmental pollutions and greenhouse effect brought out by conventional energy, especially fossil fuel. Its development and utilization will get considerable development in twenty-first Century, playing an important role in the transfer of energy structure in the world mission, and becoming the dominant energy in the late twenty-first Century.

4.1.3 wall energy saving technology

External wall energy saving technology: wall with composite technology involves three types which are insulation layer, the insulating layer and the insulation layer. In European, most countries use the externally foamed polystyrene block, in Germany, the outer insulation building construction accounts for 80% of the total, of which 70% are used polystyrene foam board.

Currently, external wall insulation method in our country has 3 kinds, namely, insulation, thermal insulation and exterior wall insulation sandwich, but the inner and exterior thermal insulation is used more popularly. Internal insulation is to attach wall with thermal insulation material, and the heat insulating layer is formed between the structure layer of air interlayer insulation materials to prevent damp; external insulation is to install thermal insulation material in external wall, which is covered with protective layer; the exterior wall insulation sandwich is arranged or insulation material filled in or between the exterior masonry wall panel in.

4.2 Building equipment energy-saving

Heating, cooling and lighting are major parts of building energy consumption, to achieve energy reduction in low carbon buildings, it is necessary to improve the energy utilization rate in demand sides (terminal), which will be considered as an alternative resource, and forming virtual energy for zero-carbon emission. It changed the traditional thought which seeks to meet the increasing demand through increase the supply of resources. In the course of long-term research, many experts and scholars have bravely done a lot attempt in research of low carbon buildings, and having achieved fruitful results. In the selection of construction equipment, we should use those of high energy efficiency. In addition, it is necessary to strengthen the management of the equipment operation and energy-saving monitoring.

4.3 The development of low carbon building materials

According to statistics, energy of building materials that are used in production process accounted for 16.7% of total social consumption, if the development of low carbon building materials, it will largely reduce carbon dioxide emissions. The low carbon building material sets the goal to reduce greenhouse gas emissions with a basis of low energy consumption and low pollution, paying more attention to green ecological design and recycling of usable resources. Low carbon building materials focus on reducing emission in the process of material production. In order to achieve energy conservation and emission reduction, our country should select new technology and materials to replace the materials which are highly related with high energy consumption and high emission industry (such as cement, steel and electrolytic aluminum). For example, we can highly use timber structure markets with ecological, energy-saving market, regeneration characteristics and extensive use of low carbon mortar.

4.4 The use of renewable energy

Buildings in hot summer and cold winter zone, based on its local climatic characteristics, can use a lot of renewable energy. Such as the use of new energy technologies which are fully developed, For example, solar water heaters, photoelectric roof panels, wall

panels, photoelectric photoelectric sunshade board, photoelectric wall between windows, skylights and photoelectric photoelectric glass curtain wall etc.

4.4.1 Solar low-temperature radiant floor heating

The low-temperature radiant floor heating refers to a way using low temperature for heating through arranging pipelines underground which take hot water below 60 °C as heating medium. It heat the floor through using the hot pipelines underground, uniformly generating heat radiation to the interior, involving characteristics like thermal stability, uniform temperature, comfort, energy saving, maintenance free, convenient management, thus it is an extremely rural residential heating ideal. Since the heat medium temperature is low in floor radiation heating, the heat source can be selected on a wide range, use solar energy as the heat source of floor heating, is one of the optimum ways of combining solar energy utilization technology and radiant floor heating technology.

4.4.2 Ventilation and heat integration

In hot summer and cold winter region southern China, summer is extremely hot, it is necessary to apply ventilation in rural residential, and people need take a shower with hot water at night. It can help to strengthen the ventilation and preparation of hot water if use solar water heater and building natural ventilation at the same time, and in this case the energy we choose is solar energy which is inexhaustible. The author of literature [3] mentioned a scheme in which the waste heat is considered as a secondary resource in solar water heater, in which the solar energy use waste ventilation heat to improve the heating process, removing the reheating device of traditional, saving energy, protecting environment, and achieve the purpose of energy-saving.

4.2.3 The application of ground source heat pump in the construction of the

In our country, shallow geothermal resources are widely distributed and largely reserves, the expert estimates that the low-temperature contained in soil and underground water in China's shallow strata within hundreds of meters is 3750 times of the electricity generator's installed capacity which is 400 millions KW, moreover, low temperature available in underground water within 100 is also 200 millions kW each year. Ground source heat pump operating costs relatively low energy consumption, low maintenance cost, flexible operation, durable, and it relies on a single system, which works well in low-story building in rural, to heating in winter and cooling in summer[4].

4.4.4 Application of wind power in village house in

Wind energy has many advantages like large reserves, widely distributed, no pollution and others, and it is also an inexhaustible and renewable sources of energy. The use of wind power, without fuel, does not produce radiation or air pollution. Rational utilization of wind energy can both reduce the environmental pollution and ecological destruction and alleviate the

pressure of energy shortage. In region which has a abundant content of wind power, it can to provide energy for the farmer's production and life through transforming wind power into electric power by mechanical devices.

4.4.5 building integrated photovoltaic

Building integrated photovoltaic combines solar photovoltaic and building, supplying electricity through installing photovoltaic matrix on external side of the wall in building. Building integrated photovoltaic, according to various way in combining matrix and building, has mainly two types: The first type is a combination of matrix and building. In this combination, photovoltaic matrix is attached on building, which works as a supporter to support the matrix; The other type of solar photovoltaic is an integration of building and photovoltaic matrix, and it comes as a kind of construction material, in which the matrix is indivisible with building. Kind of this integration has been showed like photovoltaic curtain wall, photovoltaic roof, photovoltaic visor and photovoltaic lighting roof. In these two ways described above, the first combination is generally more popular and, especially, the combination of photovoltaic matrix and roof can make use of space on roof and barrier thermal radiation from the sun in summer in cold winter and hot summer regions. Since it does not require extra space, this electricity supply system, which has highly attracted attention, has been widely spread and installed in city [5].

4.4.6 solar building photovoltaic/thermoelectric system

In the practical application of building integration photovoltaic, 80% of energy from solar radiation absorbed in solar cell cannot be effectively transformed into electric energy, instead, most of solar radiation has been transformed into thermal energy, some of which help to increase the temperature of cell and weaken its efficiency in generating electricity. It can improve efficiency of generating electricity through reducing heat by endothermic media if we set channel to release the temperature. If we have ways to absorb and store these heat for electricity generating, the system can supply electricity both photoelectrically and thermoelectrically, which is so-called photovoltaic/thermoelectric system (PV/TV). The efficiency of the entire system equals the efficiency increased by 5 to 15 percent if a single solar cell works [3].

4.5 Passive energy-saving technology

Passive energy-saving technology refers a technical way to reduce energy consumption in building through employing non-mechanical and non-electric means. Specifically, it means that, while designing and planning building, we can reduce energy consumption in HVAC system through properly setting building orientation, placement of visor, thermal insulation technology in building enclosure structure and designing building outlet that is favorable for natural ventilation. Non-carbon passive energy-saving

technology must be highly utilized if we reduce energy consumption in low-carbon building. For example, daylight illumination and natural ventilation are very helpful [6]. Daylight illumination diminishes the energy consumption in lighting, since there is no need to turn on the light in daytime; Natural ventilation is a significant way in building energy-saving, since it can, if is utilized properly while designing the building, realize free cooling in summer and transition seasons, improving IAQ and saving energy in air conditioning system in building. Some so-called zero-energy or zero-carbon buildings in Europe are realized by passive technology and renewable energy. We should, however, take a serious consideration on climate and environment features in certain regions when applying passive technology.

5. CONCLUSION

With a drastic process of urbanization, carbon building will definitely have a high building density and plot ratio, however, one important feature of low-carbon building is that it is also energy-saving building. Energy-saving in building is a basic way to develop low-carbon building, it can better develop low-carbon and green building only if we combine tightly the low-carbon and energy-saving technology in building. In hot summer and cold winter region, climate condition is bad and it results in high energy consumption in HVAC system. It must be a significant contribution in energy-saving and emission-exhausting if we largely develop low-carbon building. Low-carbon technology mentioned in this paper works well in this climate region, and it may bring to us an effective energy-saving affect if we apply it to building design process.

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APPLYING EARTH TUBE AND SOLAR CHIMNEY FOR SUMMER COOLING

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ABSTRACT

Geothermal energy, as a type of renewable energy, is widely used in various forms for air-conditioning applications, such as ground source heat pump, ground water heat pump, and earth to Air Heat Exchanger (EAHE). Without complex mechanical components, an EAHE can be utilized to condition the outdoor air for space cooling purpose. In this paper, we present a systematic study of a coupled cooling system with an EAHE and a solar collector enhanced solar chimney. Experiments were conducted with an existing test facility in a summer to evaluate the performance of the system, in terms of the passive cooling capability, active cooling capability, and soil recovering capability. The testing period of about one and a half months was divided into three time portions with three different tests. In the passive cooling period, the outdoor air was naturally pulled by the solar chimney into the tube and facility for cooling. In the active cooling period, a fan was installed to drive the air through the EAHE for space cooling. In the soil recovering evaluation period, the system was restored back to a passive mode. The results show that the indoor air temperature and humidity during the forced test was more drastic than that of the natural airflow test. The thermal sensation analysis based on Predicted Mean Vote (PMV) indicates that the indoor air condition under the natural airflow stage was more comfortable than that of the forced airflow stage. The cooling capacity of the coupled system reduced drastically after the forced airflow test due to the underground soil temperature increase. It took the soil over two weeks to fully recover from thermal saturation after the forced air test. In addition, the underground soil temperature measurements indicate that the underground heat dissipation in the horizontal level was higher than that in the vertical. The findings in this paper suggest that, while the coupled geothermal system is capable to provide summer cooling, a minimum level of control is needed to maintain the soil cooling capability and improve the space thermal condition stability.

Keywords: Geothermal energy; Earth tube; Solar chimney; Free cooling; Energy savings.

1. INTRODUCTION

The building sector is one of the biggest non-renewable energy consumers that are not utilizing much of the renewable energy available in the U.S. About 50 percent of this energy is used for heating, cooling, and ventilating buildings. At the same time, about 86 percent of energy used in buildings is from the combustion of fossil fuel [1]. Accordingly, research has been conducted to improve the operation efficiency on the building sector [2-4] and find ways of utilizing renewable energy sources and saving non-renewable energy [5-6]. In order to utilize such a renewable energy source with the great heat capacity and high thermal inertia underground, many techniques have been developed in the last decades such as ground source heat pumps (GSHPs), earth-to-air heat exchangers (EAHEs) and so on [7-8].

In the ambit of bioclimatic architecture, where a building has a direct connection with nature, EAHEs have been assigned an increasingly important role in cooling primary air free for buildings during hot weather [9]. Air passing through the buried pipes in an

EAHE system can be either from outdoor or indoor. To assess the performance of EAHEs, Krarti and Kreider [10] proposed a simplified analytical model. The enthalpy transfer problem was treated as a transient heat transfer problem without condensation, assuming a periodic variation of both air source and ground temperatures. They validated the model against measured data and applied it for parametric analysis on cooling capacity and air temperature with the pipe diameter and air velocity. Without considering the moisture transfer in the air, EAHE is like the other GCHXs, where soil is utilized as a heat sink/source. The heat transfer rate of a GCHX is critical since it determines the overall performance of a system which couples a GCHX [11-13]. Akpinar and Hepbasli [11, 12] and Hepbasli et al. [12] investigated the performance of a GCHX coupled heat pump system. They concluded the heat extraction rate of the system was impacted by a GCHX in terms of the depth below grad, pipe size, spacing, and soil type beside other system parameters. Nam et al. [13]

NOMENCLATURE

Abbreviations/Subscripts

air, a	Air
cooling	Cooling
system	System
t	Total
ACP	Above cooling pipe
EAHE	Earth to air heat exchanger
GCHX	Ground-couple heat exchanger
NCP	Next to cooling pipe
OA	Outdoor air
PMV	Predicted mean vote

RA	Room air
RAT	Room air temperature
SA	Supply air
SAT	Supply air temperature
Symbols	
h	Enthalpy, kJ/kg
Q	Rate of heat transfer, kW
T	Temperature, °C
V	Volume flow rate of air flow, m ³ /s
Greek Letters	
ρ	Specific density, kg/m ³

Just like other GCHXs, an EAHE is a passive device that provides a thermal coupling between the medium fluid and soil. Additional components are needed to provide the driving force for the medium fluid. In a pure passive design of an EAHE conditioning system, a solar chimney may be used to pull the air through the tubes. A solar chimney is a vertical or inclined air channel where its bottom end is connected to the building to be cooled and ventilated [14]. This air channel generates air movement by buoyancy forces and stack ventilation, drawing cooler air through the building in a continuous cycle in which hot air rises and exits from the top of the chimney.

So far, very few experimental studies have been carried out to systematically evaluate the performance of an EAHE-solar chimney coupled geothermal system in natural and forced airflow conditions. In addition, research about the soil temperature saturation and recovering of such a system, which could strongly impact the operation and performance, was not reported.

2. TEST SETTINGS

2.1 Introduction of the test facility

A test facility for an EAHE-solar chimney coupled geothermal system was constructed in Omaha, Nebraska, USA. A culvert steel EAHE 57-meter long with 0.45m diameter was buried at a depth of about 3-meter below grade. The test facility is a one-story building with external dimensions of 19.4 m length, 4.9 m width, and 3m height. It includes a main testing room with dimensions 15.2 m length, 4.52 m width, and 2.4 m height. The design cooling load was 2813 W based on the TRACE 700 software. A solar collector was designed and constructed with an absorbing surface area of 20m² considering that the solar irradiation is 400W/m² with a heat loss coefficient of 8.5W/(m² °C) and an absorptivity of 0.8. The solar collector was constructed by wood and a special type of solar glass that traps the solar irradiation inside the collector. The solar chimney height and diameter were selected as 12.2 m and 0.457 m respectively to provide an available draft of more than 5 Pa at a temperature of 57°C [15].

2.2 Specs of sensors

The main measurements that had been recorded for analyzing the performance of the coupled system are: the indoor relative humidity (%), outdoor relative humidity (%), supply air relative humidity (%), supply air flow rate (m³/h), solar collector airflow rate (m³/h), average indoor temperature (°C), supply air temperature (°C), outdoor air temperature (°C) and underground temperatures above and next to the EAHE.

The temperature sensors are of an accuracy ± 0.1 °C. The installation information is collected below:

- Soil temperature: As in Fig.1;
- Indoor air temperature: Between the EAHE outlet and the solar collector inlet;
- Outdoor temperature: Near the EAHE tube inlet;
- Supply air temperature: At the EAHE outlet;
- Exhaust air temperature: At the chimney inlet;

The air relative humidity sensors are calibrated within $\pm 2.5\%$ accuracy. They are installed for measuring:

- Indoor air temperature: Between the EAHE outlet and the solar collector inlet;
- Outdoor temperature: Near the EAHE tube inlet;
- Supply air temperature: At the EAHE outlet;

The airflow rates are measured mainly on the supply air and the relief air with instruments with accuracy $\pm 3\%$:

- Supply air flow: The EAHE outlet inside the house;
- Relief air flow: The solar collector inlet.

Fig.1 provides the sectional view of the buried soil temperature sensors.

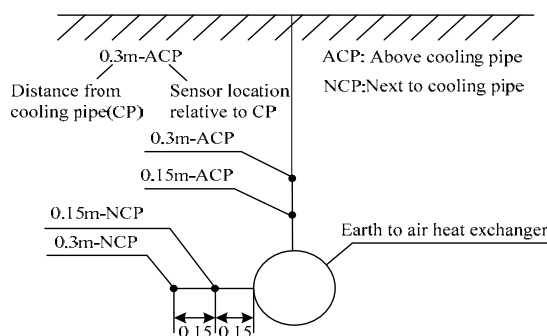


Figure 1 Underground soil temperature sensors

3. EXPERIMENT PROCEDURE

The experiments were conducted in summer, from July to September, 2009. In the 43-day testing period, both the natural and forced cooling modes were carried out to analyze the performance of the coupled geothermal system. The effect of the air on the underground soil temperature and the recovery ability of soil temperature were investigated. This testing period has been divided into three time portions. The first test was conducted in a passive/natural cooling mode and lasted for 14 days. The second test was carried out in an active cooling mode for 9 days. The third one was conducted in passive cooling mode and lasted for 20 days.

All the experiments were well monitored with measurement instruments connected to the data collection modules. Program residing in data collection software package automatically calibrate, monitor and log the data from the measurement instruments (e.g. thermocouples, humidity sensors, etc.). All the data collection modules and the data loggers were synchronized to log and store an average measurement every 15 minutes.

4. EXPERIMENTAL RESULTS AND ANALYSIS

4.1 Room air condition analysis

The EAHE and solar chimney coupled system can be a purely passive system or equipped with a simple control system for space air conditioning. In order to analyze the performance of the coupled geothermal system under different types of airflow, which were natural and forced airflow mode, an assisted constant airflow fan was installed. Fig.2 shows the trended airflow rates in three different testing stages. During the natural cooling mode, the airflow rate varies from $0\text{m}^3/\text{h}$ to maximum $500\text{m}^3/\text{h}$. The peak value of the ventilation air occurred during the daytime when high solar intensity was strong. The peak airflow was close to a desired value in design condition for a conventional forced air conditioning system, assuming 12.78°C supply air temperature in summer. This demonstrates that the solar collector enhanced solar chimney is capable to provide a significant pulling force and save fan power for the coupled geothermal energy system. The second test was in a forced airflow mode with the constant speed fan turned on continuously. The airflow rate kept a constant value at $2750\text{m}^3/\text{h}$.

Fig.3 shows the comparison of indoor and outdoor air temperature and Fig.4 shows the comparison of indoor and outdoor air humidity for the first two test periods, respectively. It can be seen from Fig. 3 and Fig. 4 that, under the natural airflow condition, the indoor air temperature fluctuated in a narrow range of $21\text{--}24^\circ\text{C}$, and the indoor air relative humidity was also maintained in between $50\text{--}70\%$. For most of the time in the natural passive cooling mode, the indoor air environment

satisfied the ASHARE standard 55-2004 for acceptable thermal comfort [16].

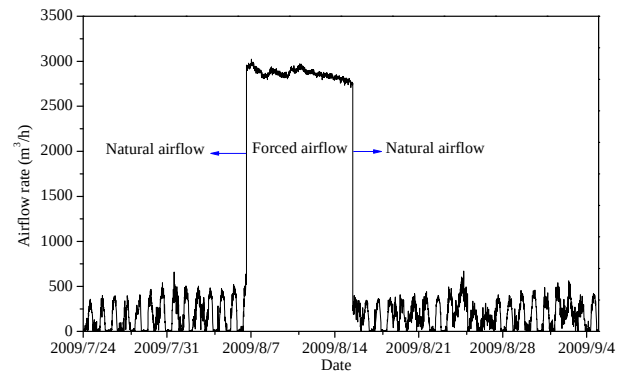


Figure 2 Airflow rate during the test

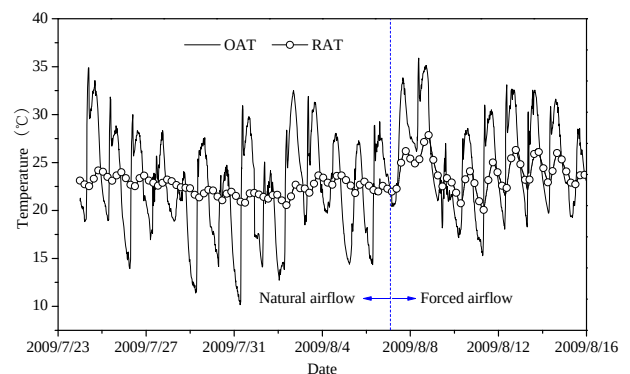


Figure 3 Indoor and outdoor air temperatures

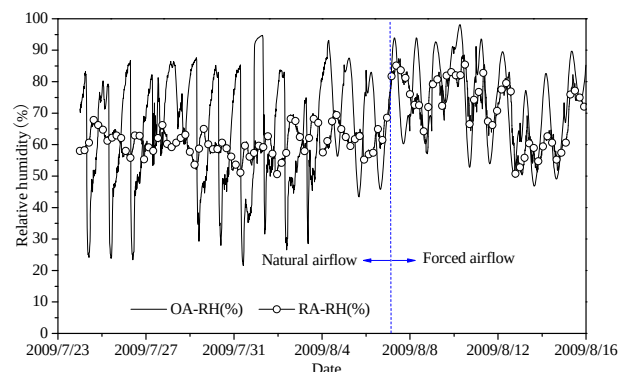


Figure 4 Indoor and outdoor air humidity comparison

This pattern of room air temperature and relative humidity changed after the active cooling mode was initiated. When the assisted fan was turned on from August 7th and the EAHE was utilized with a forced airflow, the indoor air temperature had a significant fluctuation in the range of $20\text{--}27.5^\circ\text{C}$ and the indoor air humidity fluctuated sharply with a higher value and in a bigger range. The reason is that, during the forced airflow stage, the supply air temperature climbed up with a constant high airflow and got close to the room air temperature and therefore lost its useful cooling capacity. For most of the day time, the supply air temperature was found higher than 22°C . Sometime the temperature even exceeded the designed value which resulted in the thermal comfortable problem. As a 100

percent outdoor air system, the high EAHE capacity does not necessary lead to high cooling capacity.

4.2 Thermal comfort analysis

Predicted mean vote (PMV) can be used to evaluate the indoor thermal comfort. PMV represents the predicted mean vote of a large population of people exposed to a certain environment. PMV is derived from the physics of heat transfer combined with an empirical fit to sensation. The range of PMV with the satisfied thermal sensation scale referred from ASHRAE is from -0.5 to +0.5 [16].

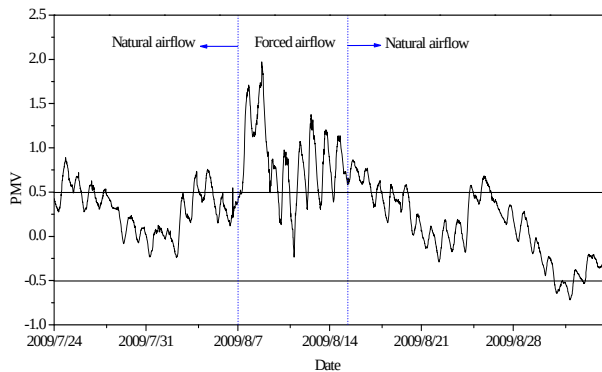


Figure 5 PMV of the indoor condition during the tests

In this study, the thermal resistance of clothing was set as 0.5 clo and the human activity as 45W. The other variables, including the air velocity, were based on real measurements or calculated values. The final results for PMV are plot in Fig. 5. It can be seen the PMV of indoor condition in the natural airflow stage almost all fell in the range of -0.5~+0.5. The PMV in the forced airflow stage rarely satisfied the standard. The main thermal sensation was slightly warm. When outside air temperature was high the thermal sensation appeared to be warm. The results show that the indoor air condition under the natural airflow condition was more comfortable than that of the forced airflow condition.

4.3 Thermal capacity analysis

With a fixed cooling load, the indoor air condition of the coupled system is determined by the total cooling capacity and the supply air condition, which is mainly driven by the outdoor air condition and the airflow rate, rather than the extracted energy from the EAHE. This is not difficult to understand since this system is a 100 percent outdoor air system. When the supply air temperature is higher than the desired room air temperature, any high cooling capacity will be useless and actually serve against the cooling purpose. More discussion of this feature can be found in [15].

The total cooling capacity of any system or subsystem that utilizes air as the heat transfer medium can be formulated as shown in equation (1):

$$Q_{cooling, t} = V_{air} \rho_{air} \Delta h \quad (1)$$

Consequently, total cooling capacities of the coupled system and EAHE can be obtained based on the airflow rate, density, and enthalpy difference. The coupled system total cooling capacity is formulated as:

$$Q_{system, t} = V_{air} \rho_{air} (h_{RA} - h_{SA}) \quad (2)$$

The EAHE total cooling capacity is formulated as:

$$Q_{EAHE, t} = V_{air} \rho_{air} (h_{OA} - h_{SA}) \quad (3)$$

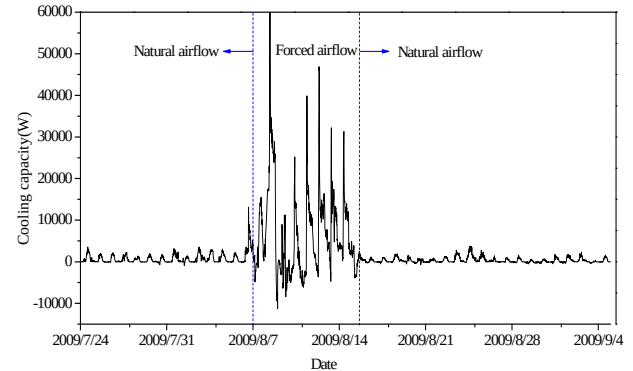


Figure 6 EAHE total cooling capacity

Fig.6 and Fig.7 show calculated results for the EAHE and coupled system total cooling capacity. It can be seen from Fig.6 and Fig.7 that EAHE and coupled system total cooling capacity in both the first and third testing stages had a smaller value than that of the second testing stage. The forced air from the fan to some extents enhanced the overall cooling capacity and might be able to remove a higher cooling load from the space. However, the coupled system sometimes provided negative cooling to the space in the forced airflow mode. This is because the EAHE could not accommodate the high flow rate with the given outdoor air condition. During the forced cooling test, the average supply air temperature was moved up and therefore the cooling capacity as shown in Fig. 8 becomes invalid to bring the room air to the desired conditions. The higher room air temperature was also partially resulted by the high outdoor air temperature during the forced air test period.

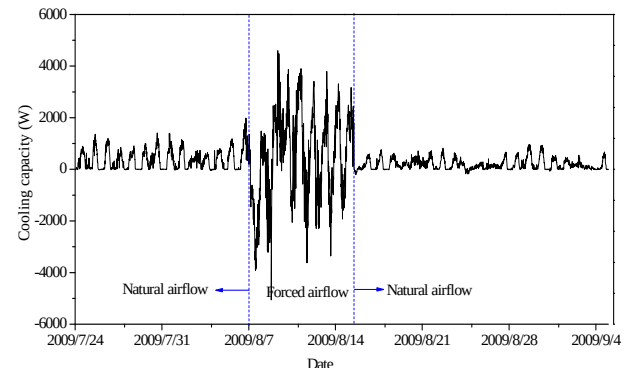


Figure 7 Coupled system total cooling capacity

4.4 Underground soil temperature analysis

Fig. 8 shows the underground soil temperature profiles at 9 m away from the test facility building at three different heights above the EAHE during the entire testing period. It is clear from the figure that, regardless the depths, the soil temperature increased drastically during the forced airflow testing period. From Fig. 8, it can be seen that the change of soil temperature of location closer to the EAHE tube was more severe than that farther from the tube. For example, the underground soil temperature at 0.3m above the EAHE preserved at a temperature of about 17.8°C during the first passive cooling period then it rose up to 21.1°C during the forced airflow test and even crossed the temperature curve of 1.2m above the tube. Afterwards it gradually settled down at a temperature about 18°C in two weeks. The soil temperature at 1.2 m above the EAHE tube was also elevated by about 0.8°C after the forced air test.

Figs. 9 and 10 show the underground soil temperatures at two different directions to EAHE. As seen from these two figures, when the airflow entered the forced airflow stage from natural airflow, the temperatures increase in the horizontal level was higher than that in the vertical level. The results indicate that the underground heat dissipation in the horizontal level was higher than that in the horizontal level. This might be because that the soil temperature at horizontal level of the EAHE tube is generally lower compared to that at vertical level above the tube. Therefore, a higher gradient leads to higher heat dissipation in that direction.

From the combined information, it can be deduced that, in order to improve the cooling capacity from the geothermal system and avoid thermal saturation around the EAHE tube, enhancement on heat transfer around the tube and along the horizontal direction might be necessary. Since the main resistance comes from the soil side, a possible measure is to weld metal rods around the tube and extend the supporting volume of soil to the system.

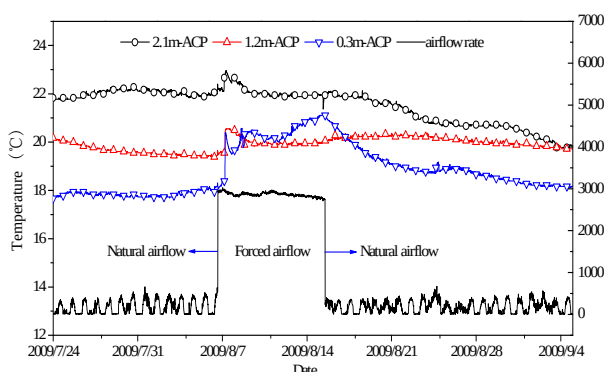


Figure 8 Underground soil temperatures at three different heights above EAHE

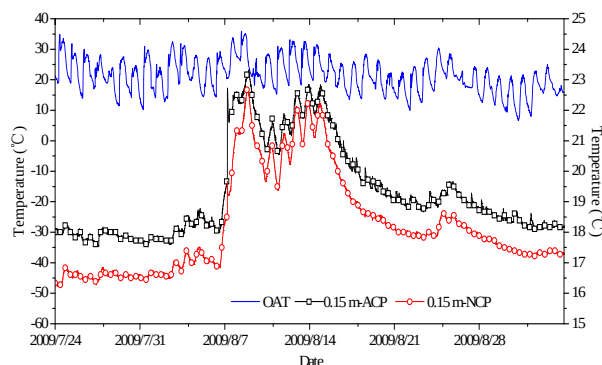


Figure 9 Underground soil temperatures at two different directions with 0.15 m away from EAHE

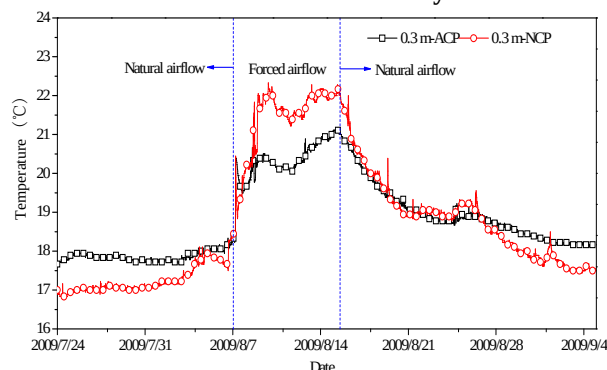


Figure 10 Underground soil temperatures at two different directions with 0.3 m away from EAHE

5. DISCUSSIONS AND CONCLUSIONS

In this paper, an experimental study of a coupled geothermal cooling system is presented. In this system, an earth-to-air tube is coupled with a solar collector enhanced solar chimney to achieve free space cooling in summer. In order to analyze the performance under both the natural and forced airflow/cooling conditions, the effect of the air on the underground soil temperature, and the recovery ability of soil temperature by itself, the paper carried out three continuous field tests with an existing facility.

Testing period was divided into three time portions with three different tests. The testing period lasted for 43 days from July 24th, 2009 till September 4th, 2009. The three tests were a passive cooling test with natural airflow, an active test with forced airflow, and another passive cooling test. The data from the experiments on the outdoor air conditions, indoor environment, soil temperature at various locations and depths were analyzed. In addition to the previous observations, following conclusions can be made based on the analysis:

The coupled geothermal system is feasible to provide cooling to the facility in natural operation mode free without using any electricity. The solar collector enhanced solar chimney can utilize the solar energy and provide more airflow to the system during the daytime with a stronger solar intensity. For most of the time, the indoor air temperature, humidity ratio, and thermal comfort were in an acceptable range;

The indoor air condition is more stable in passive cooling mode than in active cooling mode. As a 100 percent outdoor air system, when a forced air mode is used to improve the overall cooling capacity, a preliminary control is needed on the fan to ensure that the operation can maintain an acceptable supply air condition with an increased cooling capacity;

The underground heat dissipates faster in the horizontal level than the vertical level. The soil around the tube in the horizontal level may saturate sooner than the vertical direction above the tube due to the higher temperature gradient;

The surrounding soil of the earth tube may saturate and takes long to recover if the heat is over-extracted from the soil in a forced air mode. The design of the earth tube can be further improved by, for example, welding rods around the tube and extend the effective volume of soil to the system.

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RESEARCH ON ENERGY CONSUMPTION MEASUREMENT METHOD OF RURAL HOUSES BASED ON KANG

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ABSTRACT

In north China in winter, rural houses have its unique characteristics as follows, having big thermal inertia and big indoor temperature fluctuation, having wide variety of heating fuels, having a number of rooms however few are heated, having limited allowed operating current with residential electric cable and meter, which bring difficulties to building energy consumption measurements. The study aims to put forward a novel method of energy consumption measurements in rural houses in order to deal with above existing problems, it regards the building as a whole, treating the Kang, firewall and supporting stove as primary heat sources, treating the electric heating device as auxiliary heat source, and thus building energy consumption measurements can be conducted in condition that all heat sources are in stable operation and the indoor temperatures are maintained within the permitted level scope. Compared with the method that utilize electric heaters alone as the heat source for energy consumption measurement, measurement error for unit heat consumption is 2.52% under standard meteorologic conditions. The overall results of the research shows that energy consumption measurement of rural houses in China has its special characteristics. Using electric heaters alone to measure the rural residential unit heat consumption has the strengths of stable indoor temperatures and an easy testing method, together with reliable measurement results, however, it is only suitable for occasions with high-capacity electric wire and electric energy meter in houses. The novel method combining the Kang and stove with auxiliary electric space heaters for heat consumption measurement is more feasible for rural residential houses under actual operating conditions.

Keywords: Energy consumption measurement; Rural houses; Chinese kang; Electric space heater.

1. INTRODUCTION

Building unit heat consumption is an important index for evaluating building energy consumption, according to China 'Standard for energy efficiency test of residential buildings', calculation of building unit heat consumption $q_{ha}(W/m^2)$ is given by[1]

$$q_{ha} = \frac{Q_{ha}}{A_0} \cdot \frac{278}{H_r} \quad (1)$$

Where Q_{ha} is the accumulated supplied heat for a building within test duration(MJ), H_r is the test duration(h) and A_0 is total building floor area(m^2).

For urban central heating systems, Q_{ha} can be detected with meters installed at building thermal entrance. However for dispersed heating systems of rural residential houses in China, many difficulties do exist to obtain Q_{ha} , which is listed as follows:

a)In rural houses, Chinese kang or firewall together with its supporting stove are main heating sources. Both combustion equipment and heat dissipating equipment send out heat to indoor environment, which relate with fuels consumed and energy efficiency of equipments.

b)Current test method for the performance of civil kang[2] treats stove dissipated heat as heat loss, whereas it do disperse heat to indoor space.

c)Current test method of chinese kang[2] is under thermal unsteady condition, kang's surface average temperature keeps rising up during the test. And the result of kang's thermal efficiency can not reflect its inherent performance, which is affected by fuel addition and personnel operation.

d)Building heat energy consumption measurements should be conducted in condition that indoor temperatures are kept stable. But in rural houses, in order to save fuel consumption, intermittent heating is very common and only partial rooms are heated in winter. Thus it is rather difficult to ensure rural houses' indoor temperatures within the allowed fluctuation range because of big thermal inertia and big indoor temperature fluctuation[3].

e)The easy method to obtain Q_{ha} is using electric heating devices with temperature controller, by metering electric power consumption during test period while maintaining indoor temperatures within allowed range. But in rural houses it is usually infeasible to use such method due to limited allowed operating current with residential electric cable and meter.

The article aims to put forward a novel method of heat energy consumption measurements in rural houses in order to deal with above existing problems.

2. PRINCIPLES AND METHODS

The method regards the building as a whole, treating the Kang, firewall and supporting stove as primary heat sources, treating the electric heating device as auxiliary heat source, and thus building energy consumption measurements can be conducted in condition that all heat sources are in continuous stable operation and the indoor temperatures are maintained within the permitted level scope within test duration. Start and stop mode of an electric heating device is controlled by temperature controller.

The method and formulars are described in detail as follows:

a) total accumulated supplied heat of a rural house when the rural house meets thermal steady condition, in other words indoor temperatures are maintained within permitted level scope, Q_{ha} is described as:

$$Q_{ha} = Q_d + Q_k \quad (2)$$

Where Q_d is the accumulated electric power consumption within test duration(MJ), $1\text{MJ}=1\text{ kW} \cdot \text{h} \times 3.6$, Q_k is the kang and stove's accumulated supplied heat within test duration(MJ).

b) accumulated supplied heat of a kang and a stove

$$Q_k = \eta_k \cdot B_k \cdot Q_{dw,k}^y \cdot 10^{-3} \quad (3)$$

Where η_k is kang's thermal efficiency including the supporting stove under thermal steady condition, B_k is fuels consumed within test duration(kg), $Q_{dw,k}^y$ is net calorific value as received basis of fuels(kJ/kg)

c) kang's thermal efficiency including the stove under thermal steady condition

In northern rural houses, both kang and the supporting stove dissipate heat to indoor space, especially the stove is part of heating source and should not be neglected.

η is defined by the counter balancing method [2]:

$$\eta = 100 - (q_2 + q_3 + q_4 + q_5 + q_6) \quad (4)$$

Where η is thermal efficiency of independent kang, q_2 is smoke exhaust heat loss, q_3 is gaseous imperfect combustion heat loss, q_4 is solid imperfect combustion heat loss, q_6 is heat loss due to cinder(usually 0.5%), and q_5 is heat loss of stove (usually 8%) which should be treated as heat contribution in the method. Thus η_k can be concluded as

$$\eta_k = \eta + q_5 = 100 - (q_2 + q_3 + q_4 + q_6) \quad (5)$$

d) unit heat consumption of a rural house

Based on test data and above formulas, total accumulated supplied heat of a rural house Q_{ha} can be obtained during test period, substituting for equation (1) and thus building unit heat consumption q_{ha} (W/m^2) in actual meteorologic condition can be calculated. q_{hm} (W/m^2) in standard meteorologic condition is given by

$$q_{hm} = q_{ha} \cdot \frac{t_i - t_{eh}}{t_{ia} - t_{ea}} - q_r \quad (6)$$

Where t_i is design indoor temperature($^{\circ}\text{C}$)(normal value 18°C), t_{eh} is local outdoor mean air temperature during heating period($^{\circ}\text{C}$), t_{ia} and t_{ea} are actual indoor mean air temperature and outdoor mean air temperature respectively within test duration($^{\circ}\text{C}$), q_r (W/m^2) is heat gain inside the house including heat release of personel, lighting and household electrical appliances etc.

3. TEST CASE STUDIES

3.1 Test building introduction

The test rural house is located in rural area of Harbin near Shui Yuan Road. It is single one-layer family house(Figure 1), total floor area is 33.6m^2 , building exterior walls are 370 mm solid clay brick with cement sand plaster on both sides without insulation, windows are single-layer double glass of plastic-steel frame, sloping roof is roof board hanging tiles with sawdust and rice hull filled in for insulation, the floor is crushed stone concrete with cement sand finishing without insulation, exit door is double-layer wooden with door socket.



Figure 1 Test building photograph

Inside the building, there is a traditional ground kang(length 2.4m, width 1.8m), with U-shape kang cave layout with the stove and chimney on one side(Figure 2). During the test period, no other persons lived inside except one who was in charge of the stove and kang's operation. Each room had one electric space heater(rated power 2 kw) with temperature controller.

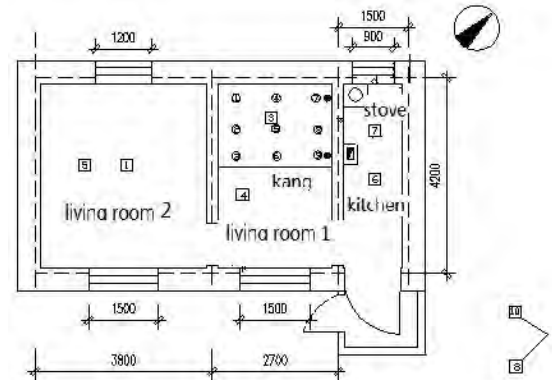


Figure 2 Floor plan of the test building and sensors layout

□-□ representing indoor & outdoor temperature sensor,

○-○ representing temperature sensor on kang's surface

● Representing kang's inlet and outlet smoke temperature sensor

3.2 Testing instruments

Building indoor & outdoor temperatures were recorded by BES-02 automatic collection of temperature and humidity(precision 0.5°C)(see Figure 3), log interval is 10 min. Kang's surface temperatures were recorded by BES-C portable multi-channel data acquisition instrument(precision 0.3°C)(see Figure 4), log interval is 10 min.

Kang's inlet and outlet smoke temperatures were measured by K-thermocouple ((precision 1%)and 4-channel data acquisition instrument(Figure 5). Gas components were analyzed by a portable flue gas analyzer(Figure 6)



Figure 3 BES-02



Figure 4 BES-C



Figure 5 K-thermocouple and 4-channel instrument



Figure 6 Portable flue gas analyzer



Figure7 Electric power meter

Building electric power consumption was measured with an ordinary electric energy meter(precision 1st grade), elementary current 10 A/rated current 20A(Figure 7). Fuels(coal) consumed were weighed by a platform scale, sensibility 0.05kg. proximate analysis of coal was finished in lab.

3.3 Results and discussion

In order to verify the novel method, test case studies were conducted in two phases. First phase test was through Mar. 3 to Mar.8 in 2011, combined heat sources included kang ,stove and electric space heater, test duration of thermal steady condition was 120 hours. Second phase test was through Mar. 9 to Mar.14 in 2011, only electric space heaters were used as heat source, test duration of thermal steady condition was also 120 hours. Two tests' data are given in Table 1, also Figure 8 and Figure 9. Indoor air temperature of the test house at every test moment is calculated from room area weighted mean air temperature, and on the basis t_{ia} is the average of the whole during the test period[4].

Measuring points on kang's surface were placed in order to get kang's mean surface temperature at each measuring time, which is used to discriminate thermal steady state from unsteady state. In-depth discussion can be referenced in author's another article having been submitted to a journal named' Discussion on evaluation indices and test method for civil Kang's thermal performance'.

For the 1st phase test, single kang's thermal efficiency under thermal steady condition was 78.9% according to equation (4) the counter balancing method, which was not explained in detail in the article, it can be referenced in [2]. Therefore comprehensive thermal efficiency of kang and stove was valued as $78.9+8=86.9\%$.

Table 2 shows the specific comparative calculated data of two tests according to previous methods and equations, where outdoor mean air temperature during heating period of Harbin was valued as -8.5°C, and q_r was valued 0. It shows that the relative error of calculated unit heat consumption of two methods $\Delta = (67.5 - 65.8) / 67.5 = 2.52\%$.

Due to difficulty of accurate calculation of the stove dissipated heat into indoor environment, 8% is selected as stove estimation according to literature[2], which may cause unexpected error of final value 65.8 w/m². During the test period, the kang's mean surface temperature should be observed and maintained carefully, because rate of change of kang's mean surface temperature 2% is recommended as criterion for thermal steady state of kang, which is discussed in author's journal article to be published .

Table 1 Test data of two phases

Phase	fuel consumed (kg)	electric power consumed (kW·h)	Test duration (h)	Temperature mean value	
				t_{ia} (°C)	t_{ea} (°C)
1 st phase	39.2	50	120	25.6	-5.9
2 nd phase		214	120	18.7	-2.2

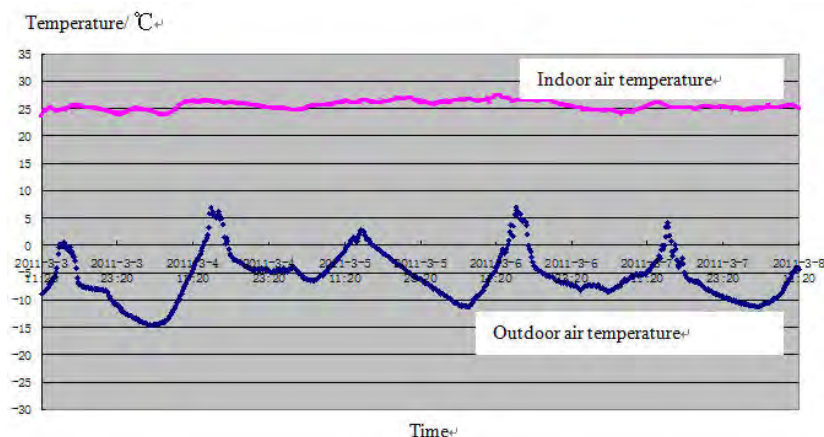


Figure 8 Indoor and outdoor temperature variation in 1st phase

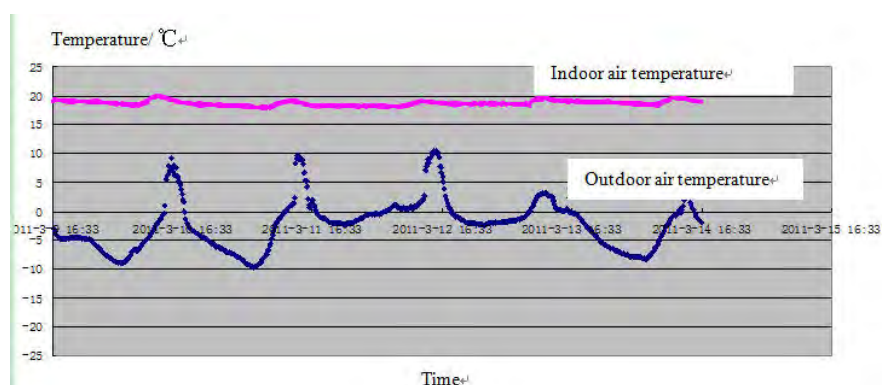


Figure 9 Indoor and outdoor temperature variation in 2nd phase

Table 2 Two phase comparative data between calculations of heat consumption

Item	Combined heating (1 st phase)	Electric space heater only(2 nd phase)
Accumulated electric power consumption (kWh)	50	214
Accumulated electric power consumption converted to heat supply Q_d (MJ)	180	770.9
Accumulated heat supply of a kang and a stove Q_k (MJ)	954.41	0
Accumulated supplied heat for the building Q_{ha} (MJ)	1134.41	770.4
building unit heat consumption in actual meteorologic condition q_{ha} (w/m ²)	78.2	53.1
building unit heat consumption in standard meteorologic condition q_{hm} (w/m ²)	65.8	67.5
Relative error Δ (%)		2.52%

4. CONCLUSIONS

Energy consumption measurement of rural residential houses in China has its special characteristics and difficulties. Compared with the method that utilizes electric space heaters alone as the heat source for energy consumption measurement , which is only suitable for occasions with high-capacity electric wire and electric energy meter in houses, the novel method combining the kang and stove with auxiliary electric space heaters for building heat consumption measurement is more feasible for rural residential houses under actual operating conditions.

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RESEARCH ON INCENTIVE MECHANISM OF VOLUNTARY AGREEMENT OF ENERGY CONSERVATION IN BUILDING SECTORS

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ABSTRACT

Voluntary agreement of energy conservation plays an important role in improving the energy conservation and achieving sustainable development in industry sectors. In addition to energy savings, Voluntary agreement of energy conservation may also raise awareness regarding energy efficiency among industrial managerial and technical staff ultimately leading to more future investments in sustainable energy efficiency as well as the promotion of further research and development into efficient products and services. However, the development speed of Voluntary agreement of energy conservation in China is very slow, especially with building sectors. This paper analyzes incentive mechanism of Energy Conservation Voluntary Agreement from the macro to micro levels, and proposes corresponding measures so as to offer theoretical references for the development of Voluntary agreement of energy conservation in building sectors. These measures should be effectively coordinated to form a network structure of an organic whole which can produce the best results.

Key words: Voluntary agreement; Building energy conservation; Incentive mechanism.

1. INTRODUCTION

The use of Voluntary Agreements of Energy Conservation (VAEC) as a policy for increasing energy-efficiency in industry, which has been a popular approach in many industrialized countries since the early 1990s, is being tested for use in industry sector in China [1]. However, up to present, no attempt is tried for VAEC in building sector, although China's building sector consumes over 30% of the nation's total energy each year and is heavily dependent on the country's abundant, yet polluting, coal resources.

Voluntary agreement of energy conservation can help to overcome resistance to investing in improved energy efficiency in building sectors. Voluntary agreement of energy conservation is based on contracts negotiated between building owners, serving companies and governments, which will often include targets, commitments and time frames. Voluntary agreement of energy conservation has already resulted in significant increases in efficiency within industry, and has also been successful in encouraging a change in attitude and awareness of managerial and technical staff regarding energy efficiency.

There are many international examples of Voluntary agreement of energy conservation, and many have proven to be very successful in motivating industry and improving energy efficiency. Over 125 Voluntary agreements of energy conservation programs currently exist world-wide, including in New Zealand,

Australia, Canada, the Netherlands, Germany, Korea and Chinese Taipei.

In addition to energy savings, voluntary agreement of energy conservation may also raise awareness regarding energy efficiency among industrial managerial and technical staff ultimately leading to more future investments in sustainable energy efficiency as well as the promotion of further research and development into efficient products and services.

Voluntary agreement of energy conservation plays a key role in improving the energy conservation and achieving sustainable development. However, the development speed of voluntary agreement of energy conservation mechanism in China is very slow, especially with building sectors[2]. Energy conservation service companies are not paid much attention and their enthusiasm is not brought into play.

The paper analyzes the main obstacles from three aspects of capital, policy and technology, and proposes corresponding countermeasures so as to offer theoretical references for the development of voluntary agreement of energy conservation in building sectors.

2. VOLUNTARY AGREEMENT OF ENERGY CONSERVATION

Agreements to meet specific energy-use or energy efficiency-targets are used in the industrial sector in many countries around the world. Such agreements can be viewed as a tool for developing a long-term strategic plan for increasing industrial energy efficiency that

fully engages not only the engineers and management at industrial facilities, but also includes government, industry associations, financial institutions, and others. An agreement or target can be formulated in various ways. Two common methods are those based on specified energy-efficiency improvement targets and those based on absolute energy use or greenhouse gas emissions reduction commitments. Either an individual company or an industrial subsector, as represented by a party such as an industry association, can enter into such agreements [3].

Voluntary Agreements typically have a long-term outlook, covering a period of five to ten years. The agreements focus the attention of all actors on energy efficiency or greenhouse gas emissions reduction goals. The key elements of Voluntary Agreement programs are the assessment of energy-efficiency potential of the participants as well as target-setting through a negotiated process with all parties. Supporting programs and policies, such as audits, assessments, benchmarking, monitoring, information dissemination, and financial incentives, all play a role in assisting the participants in meeting the target goals.

Overall, Voluntary Agreements are viewed as an innovative and effective means to motivate industry to improve energy efficiency and reduce greenhouse gas emissions. Through review of 350 voluntary actions and programs, it can be found that past and present experiences with voluntary actions show that, properly designed and implemented, they can achieve stated objectives, sometimes even exceeding those of minimum regulatory standards, and help integrate economic and environmental goals. Another analysis of seven Voluntary Agreement programs found that the programs could be credited with about 50% of the observed energy-efficiency improvement or emissions reductions. In addition to these so-called direct effects of the programs, there are also important medium- and long-term impacts including: changing attitudes and awareness of managerial and technical staff regarding

energy efficiency; addressing market, institutional, and regulatory barriers to technology adoption and innovation; fostering market transformation to establish greater potential for sustainable energy-efficiency investments; promoting positive dynamic interactions between different actors involved in technology research and development, deployment, and market development; and facilitating cooperative arrangements that provide learning mechanisms within a sector or industry to combine knowledge and develop new competencies in industry[4].

Based on experience to date, the “Seven Golden Rules” for these type of agreements are: 1) make sure they are negotiated agreements based on assessments of energy efficiency potentials that are more than “business-as-usual”, 2) set clear, well-defined targets and specific timetables for achieving those targets, 3) ensure long-lasting government support in the form of policies and programs that assist industries in implementing energy-efficiency improvements, 4) focus on large, energy-intensive industries to start with because this is where the greatest savings are found, 5) establish clear monitoring guidelines, 6) evaluate progress using physical energy intensity measurements, and 7) provide for independent verification of progress.

The voluntary agreement system is a non-regulatory measure wherein businesses or business groups producing, supplying, or consuming energy conclude agreements with the government[5].

Businesses present and achieve energy saving and/or greenhouse gas emission reduction goals, implementation schedules, implementation methods, etc.

The government provides support in the form of funds and tax incentives as well as performs monitoring and evaluation to achieve the goals in collaboration with businesses[6].

The mechanism of Voluntary Agreements of Energy Conservation (VAEC) is depicted as Figure 1.

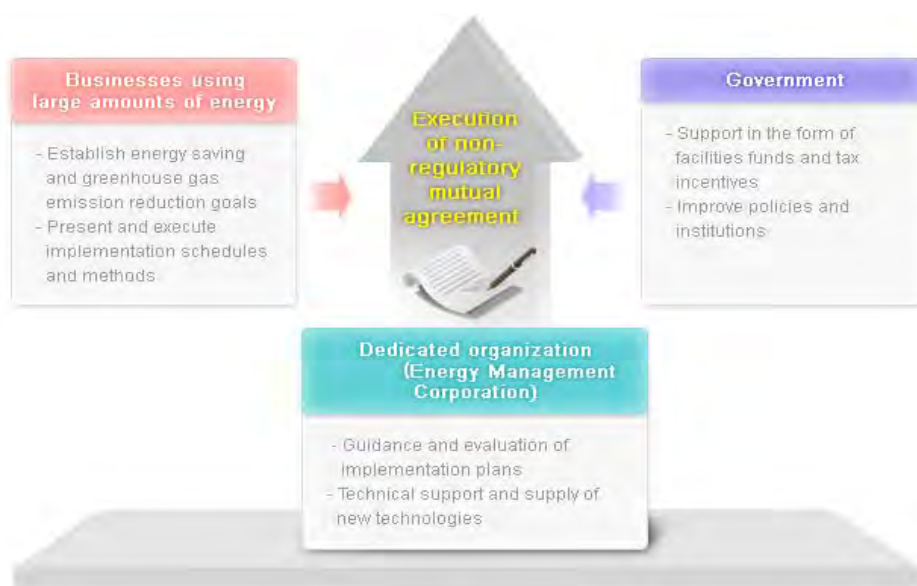


Figure 1 Mechanism of voluntary agreements of energy conservation

3. INCENTIVE MECHANISM OF ENERGY CONSERVATION IN BUILDING SECTORS

3.1 Energy conservation voluntary agreement incentive policies

Currently Energy saving and emission reduction policies, which are basically based on “top-down” administration mode, can not meet the requirements of the Energy Conservation Voluntary Agreement. Therefore, energy-saving incentive policies in finance, taxation and other aspects must be introduced in order to satisfy the requirements of the rules of market economy. Besides, a conducive corporate voluntary energy conservation policy system should be gradually established.

3.1.1 Financial assistance

The Government should establish a voluntary energy saving special funds which can be derived partly from the financial allocation of the national energy saving and partly from the municipal financial support. The amount and growth must be definite and budget support long-term mechanism of the Energy Conservation Voluntary Agreement should be established. In order to encourage enterprises to participate in the Energy Conservation Voluntary Agreement, the initial pilot of the voluntary agreement of all those involved in the pilot enterprises should make the appropriate financial subsidies. Building owners or enterprises which realize the energy-saving target after the completion of the pilot should be give certain financial awards to honor the Government's commitment.

3.1.2 Tax relief

The role of incentives and constraints of the existing tax policy should be strengthened. Energy Conservation Voluntary Agreement enterprises should be given a certain amount of income tax relief in order to attract more enterprises to join the ranks of the Energy Conservation Voluntary Agreement. Besides, the energy tax and carbon tax system should be developed as soon as possible ,with the taxed relieved for those enterprises to participate in the Energy Conservation Voluntary Agreement .

3.1.3 Financing support

By the government to guarantee, the business or the size of the financing of energy service companies to participate in the Energy Conservation Voluntary Agreement could be expanded. In addition, the arrangement of concessional loans of international financial organizations and foreign governments could be prioritized to support the project of the Energy Conservation Voluntary Agreement.

3.1.4 Audit and evaluation policies

With the audit and evaluation of transaction support for voluntary agreements, the government can take two forms: free audit and assessment services, bearing partly the cost of audit and evaluation.

3.1.5 Technical support and information dissemination

In the process of the Voluntary agreement signing and implementation, the government should be responsible for the preparation of the agreement, provide professional technical support and related data to the business units, and disclose information on the implementation of the project to the public.

3.1.6 Government recognition and public acceptance

Government may grant building owners participating in voluntary agreements protection flag, honor, or publicize and promote the enterprise, improve the public to participate in the recognition.

The above mentioned incentives of Energy Conservation Voluntary Agreement, from the macro to micro levels, play a different role. These measures should be effective coordinated to form a network structure of an organic whole which produce the best results.

3.2 Energy conservation voluntary agreement regulatory regime

3.2.1 The supervision and management of government departments

Energy Conservation Voluntary Agreement relates to a number of departments and economic entities. It could not be successfully implemented without the supervision and management of the relevant government departments. The Municipal Energy Agency must track the enterprises in the entire process to determine whether the participating companies fulfill the commitments stipulated in the agreements. Capacity building of energy monitoring, law enforcement should be strengthened, and the energy monitoring system at all levels should be gradually established and improved.

3.2.2 Self-supervision and management

The building owners should be in accordance with the relevant provisions and requirements of the Energy Conservation Voluntary Agreements. And They should implement the most economically feasible energy conservation measures and projects, report regularly the statements of the energy consumption to the government authorities and publish the energy consumption regularly.

Energy consumption publicity should be in accordance with the laws and regulations of relative laws and polices. Through the establishment of the

energy consumption of fixed fare system, energy consumption statistics, energy efficiency audits could be carried out. The results should be published in the official website of the enterprise as well as local mainstream medias.

3.2.3 Media and public scrutiny and recognition

It is needed to play the role of the media and the public, to promote enterprise quality and quantity to complete the voluntary energy conservation agreement defined objectives and targets. The role of the media and the public is reflected in the supervision and recognition, and the public rely mainly on information disclosure, public participation and other means. It is important to establish the Energy Efficiency Voluntary agreements on information disclosure system and reporting system. The media and the public are encouraged to report irregularities in the implementation of energy conservation voluntary agreement projects in order to establish a corporate image of energy saving and environmental protection in public and to guide market behavior .

3.3 Online monitoring system of building energy consumption

It is important to take full advantage of modern computer, the Internet, mobile communications, and automated control of high-tech to construct energy consumption online monitoring system which can achieve real-time monitoring of water, electricity, gas, heat and energy consumption involved in the Energy Conservation Voluntary Agreement. The system can conduct scientific and standardized appraisal management, and provide advanced technical support and security for enterprise energy-saving goals.

Online energy consumption monitoring system should include energy management, communication layer and data monitoring layer.

4. CONCLUSIONS

Voluntary agreement of energy conservation can help to overcome resistance to investing in improved energy efficiency in industry or building sectors. Voluntary agreement of energy conservation is based on contracts negotiated between building owners, serving companies and governments, which will often include targets, commitments and time frames. Voluntary agreement of energy conservation has already resulted in significant increases in efficiency within industry. In addition to energy savings, Voluntary agreement of energy conservation may also raise awareness regarding energy efficiency among industrial managerial and technical staff ultimately leading to more future investments in sustainable energy efficiency as well as the promotion of further research and development into efficient products and services.

The paper analyzes the main obstacles from three aspects of capital, policy and technology, and proposes

corresponding countermeasures so as to offer theoretical references for the development of Voluntary agreement of energy conservation in building sectors. Besides, this paper constructs theoretical model of building energy efficiency under market mechanism, and expounds its defining mechanism under market mechanism, and function mechanism of mandatory policy, encouraging policy and technology advance theoretically. finally, by combining the theoretical study and particular practice of China, and using the experience of other countries for reference, this paper puts forward both the overall train of thought and implementation tactics of China building energy efficiency, and also puts forwards some policy suggestions on its implementation.

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EVALUATION METHODOLOGY OF GREEN CAMPUS CONSTRUCTION IN CHINESE UNIVERSITIES

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ABSTRACT

With the expansion of the enrollment in recent years, energy use in Chinese higher education institutions are increasing, and there is a great energy saving potential due to the low-level operation in campus facilities. Meanwhile, as a place for knowledge transformation, sci-tech talents cultivation and technical innovation, universities are of great importance in the development of the sustainable society. Thus, the development of an evaluation methodology of sustainable campus is significant for deep and persistent construction of green campus in China. In this paper, the definition and connotation of the sustainable campus are worked out. On this basis, a set of evaluation indices system of sustainable construction and operation of green campus is developed. The evaluation indices system refers to five aspects of resource-saving, environment-friendly, campus culture, organization and management, and social services. Quantitative and qualitative indices are proposed for each field above. As for the evaluation methodology, the indicators in these fields consist of the control terms and the general terms. Control terms are mandatory to be realized, and the scores of all the general terms in each field are accumulated to get the total score, which is used to evaluate the construction and operation of green campus.

Keywords: Green campus construction, Evaluation indices, Evaluation methodology

1.INTRODUCTION

There are more than 2700 colleges and universities in China, with more than 700 million square meters of school buildings [1]. Campus energy consumption is very large, with energy and water consumption per student higher than per capita residential energy consumption in China [2]. There are a great number of campus facilities, and their operation is still in the stage of low performance. Loose management is also a common phenomenon of the university campus. Meanwhile, university is also a base to inherit knowledge, and to cultivate talents of science and technology. The formation of green humanities and the advocacy of green concept in campus, play an important role in enhancing the students of sustainable development conception and relevant scientific knowledge, and forming the green campus culture and low-carbon lifestyle. In addition, the combination of green campus construction and the application of green technology innovation can drive the development of related social green industry. Therefore, through a full range of green campus construction, making the university campus a best base of professional sustainable education is of great significance to the building of a sustainable society.

It is very necessary to develop an appropriate green campus evaluation method, so as to reasonably

lead the energy conversation and carbon reduction on campus, and assess the achievement of green campus construction and operation by a long-term mechanism. There has been a lot of international research works on green campus assessment, and a series of related evaluation methods have been developed. However, most of the evaluation method is evaluated for a special field of green campus. For example, the Sustainability Assessment Questionnaire of Association of University Leaders for a Sustainable Future, and Auditing Instrument for Sustainability in Higher Education issued by AISHE are used to evaluate the sustainability of higher education [3,4]; LEED for school for the evaluation of campus building [5]; There are some other evaluation methods, such as Penn State Indicators Report, Sustainability Tracking, Assessment & Rating System, making a comprehensive evaluation from all fields of Green campus construction: energy, water, food, transportation, the built environment, community, research, education, etc [6,7]. However, even these more mature comprehensive evaluation systems also have some problems: hard to make the comparison among the evaluated institutions, the uniform weight of different indicators, etc. Meanwhile, many evaluation indices are not suitable for the Chinese universities, due to the big difference of management mechanism and operation between foreign and Chinese universities. Aiming at these problems, a green campus evaluation

system is developed for Chinese universities, considering the characteristics of green campus construction and operation in China.

2.THE CONNOTATION OF GREEN CAMPUS

Green campus construction is a major project which faces to the teachers and students and covers all kinds of school activities. Firstly, management agencies and a complete set of mechanism system should be established to guarantee the smooth operation of green campus. Secondly, green campus needs the sustainable development in all aspects--campus planning, built environment, infrastructure operation, teaching, research, student activities and so on. Students and faculty are the main part of sustainable teaching and research, and the green campus construction has connected both the campus residents and the outside world; under the cooperation and resource integration of each department of the school, the campus will be made as a living laboratory. Therefore, Green campus construction is very extensive, involving the following seven areas: (1) Green campus organizations and institution building; (2) utilization efficiency and management of energy and resources; (3) Campus ecological environment; (4) Green Teaching; (5) Green scientific research; (6) Green Humanities; (7) Social services.

3.THE DEVELOPMENT OF GREEN CAMPUS EVALUATE METHODOLOGY

According to the connotation and covered fields of the green campus, combined with the characteristics and current situation of China's green campus construction, an evaluation methodology suited to the green campus construction and operation of Chinese

universities is developed. The indices are set through five aspects: resource-saving, environment-friendly, campus culture, organization and management, and social services. The control item must be met. The general item takes the form of scoring, and the green campus construction and operation among different colleges and universities can be compared according to the total value of the five categories of indices.

3.1 Evaluation indices of energy and resource saving and their indices scores

Indices of resource saving include five aspects of energy, water, land, materials and waste, and transport. Control item has eight terms, and general item has total score of 95 points. Currently, with the stimulation of the national funding of energy saving campus construction, colleges and universities have carried out the energy efficiency retrofit of existing buildings, the applications of renewable energy, the application of water-saving appliances and non-conventional water resources, energy-saving lighting and other special energy-saving projects. Therefore, some indices are set to assess the progress and effectiveness of these measures. Energy saving rate and water saving rate are set to evaluate the effect of energy and resource conservation on the whole campus. Carbon emission inventory is required to list in order to understand the school's carbon footprint. In the evaluation of materials and waste, it is important to assess the effects of the use of renewable materials, the purchase of energy-saving products, waste electronic equipment and furniture recycling, as well as campus hazardous waste management. As for green transportation, enough bicycle parking spaces, separate bike paths, convenient public transport network, are encouraged to set, and clean fuel vehicles are also suggested for the public transport.

Table 1 evaluation indices of energy and resource saving and their indices scores

Categories	Indices	Scores
Energy (4 control term and 54 points)	1. There is any bldg. with green building label for operation stage	2 points
	2.1 the implementation of building retrofit	control term
	2.2 the ratio of floor area of retrofitted or retrofitting building to the total building area	>20%, 5 points; >10%, 3 points; >5%, 2 points; >1%, 1 point
	2.3 the ratio of floor area of retrofitted buildings with 50% energy reduction to the total building areas	>10%, 5 points; >5%, 3 points; >3%, 2 points; >1%, 1 point
	3.1 the ratio of renewable energy use to total energy use $\geq 3\%$	control term
	3.2 the ratio of renewable energy use to total energy use	$\geq 13\%$, 5 points; $\geq 8\%$, 3 points
	4.1 the ratio of energy efficient lamps in exterior lighting	2 points is multiplied the actual ratio to get the final score
	4.2 the ratio of energy efficient lamps in interior lighting	ditto
	4.3 the ratio of lighting with smart control in interior lighting	ditto
	4.4 smart control system for street lamps.	1 point
	4.5 campus IC card for water and electricity usage management	2 points
	5.1 establishment of campus energy management system (CEMS)	control term
	5.2 the rate of the buildings monitored by CEMS	>80%, 5 points; >50%, 3 points; >20%, 1 point;
	5.3 the rate of building with end use monitoring	>10%, 1 point; >30%, 2 points; >50%, points; >80%, 5 points.
	5.4 the percentage of main water use units (such as kitchen,	100%, 5 points; >70%, 4 points; >50%, 3

	centralized bathroom, hotel etc.) with the online monitoring	points, >30%, 2 points; >10%, 1 point
	6.1 the daylight factor of the living room in dormitory must conform to the national standard of GB/ T 50033.	2points
	6.2 the daylight factor of the main spaces in other buildings must conform to the national standard of GB/ T 50033.	2points
	7. Energy reduction before and after energy saving campus construction ¹ .	The score of each term below is multiplied by its weight, and then sum them to get the final score.
	7.1 Energy saving rate (weight of 0.2)	< 5%, 1 point; <10%, 2 points; <15%, 3points; <20%, 4points, >20%, 5 points.
	7.2 Energy saving rate per floor area (weight of 0.3)	ditto
	7.3 Energy saving rate per student(weight of 0.3)	ditto
	7.4 Energy saving rate per research output.(weight of 0.2)	ditto
	8.1 Annual carbon emission inventory	control term
	8.2 Carbon emission amount of air travel of staffs	1 point
	8.3 other carbon emission amount	1 point
Water (1 control terms and 7 points)	1. the utilization of nontraditional water source $\geq 20\%$	control term
	2. the usage rate of the water saving appliances	2 points is multiplied the actual ratio to get the final score
	3. water reduction ¹	
	3.1 The rate of water reduction on campus(weight of 0.6)	Decrease of < 10%, 1point; < 20%, 2 points; <30%, 3points; <40%, 4points; > 40%, 5points;
	3.2 The rate of per student water reduction (weight of 0.4)	Ditto
Land(1 control term+1 point)	1. The per capita building floor area should not exceed the limit values of the indicators of floor areas of architecture planning of ordinary high education institutes.	control term
Material and waste (2 control terms+20 points)	2. The usage rate of underground space is larger than 5%	1 point
	1. the usage of the reusable building material $\geq 5\%$	2 points
	2. the ratio of double-side printed paper	2 points is multiplied the actual ratio to get the final score
	3. the ratio of the use of recycled paper	ditto
	4. the ratio of electronic products with energy efficiency label to the total electronic products used on campus	ditto
	5. The percentage of the garbage bins with separate collection is larger than 75%.	control term
	6. Develop and implement the kitchen waste compost program	2points
	7. the ratio of garbage used for composting and recycling to total campus waste.	2 points is multiplied the actual ratio to get the final score
	8. the rate of per capita garbage reduction ¹ .	Ditto
	9. The ratio of recycled electronic products to the total number of waste electronic products.	Ditto
	10. The ratio of reused furniture	Ditto
	11.1 The rate of separate collection of wasted battery	1 point
	11.2 The rate of separate collection of fluorescent tube	1 point
	11.3 Take the strict management on the collection, temporary storage, transshipment, and process of dangerous chemicals.	control term
Transportation (13 points)	1. the ratio of buildings with barrier-free facilities is over 75%.	2points
	2. Split flow of people and vehicles are considered.	2points
	3. Campus should have separate bicycle lanes	1 point
	4. enough bicycle parking space	1 point
	5. The distance between the public transport site and campus entrance is suggested to be less than 500 meters.	1 point
	6. Use clean fuel and high efficient fuel driving vehicle to serve the campus public transportation and other purposes.	2points
	7. The ratio of staffs with green transport (such as by non-motorized vehicle and taking public transportation).	2 points is multiplied the actual ratio to get the final score
	8. Provide measures to encourage carpool among staffs.	2points
1. As for energy/water/garbage reduction, the annual amount of the year before the energy saving campus construction is taken as the baseline, and the value of the year before the evaluation is taken for the comparison.		

3.2 Evaluation indices of friendly environment and their indices scores

Indices of friendly environment can be used to assess campus site, ecology landscape, and micro climate environment, with totally five control items and general item altogether 25points. The Indices of ecology

landscape mainly consider the ratio of green space on campus, the proportion of native plants, as well as landscape waste composting program. The indices of micro-climate environmental covers regional heat island, the campus thermal environment, wind environment, air quality, and light environment evaluation.

Table 2 evaluation indices of friendly environment and their indices scores

Categories	Indices	Scores
Campus site (2 control terms and 2 points)	1.Campus construction should not destroy the local cultural relics, natural water, natural wetlands, basic farmland, forests and other reserves.	control term
	2.Campus site should not have the potential threat of floods, landslides and soil containing radon. Within the construction site, there is no harm of electromagnetic radiation and fire, explosive, toxic substances and other hazards	control term
	3. Outdoor permeable pavement area ratio should be more than 30%.	Two points
ecological landscape (1 control term and 4 points)	1. campus greening rate is more than 30%	control term
	2. the native plant ratio is more than 75%	Two points
	3. Develop and implement the landscape waste composting programs.	Two points
micro climate (1 control term and 19points)	1. average value of heat island intensity is less than 1.5℃	Two points
	2.measured wet global temperature in summer is less than 32℃ on campus	Two points
	3.All the buildings can meet the indoor thermal comfort requirement in the design specification of heating, ventilation and air conditioning.	Three points
	4. the wind power amplification factor at pedestrian area is less than 2	Three points
	5. the number of days with air pollution index less than 100 is more than 90%	control term
	6. All the campus buildings can meet the indoor air quality requirements in the national standard of indoor air quality.	control term
	7. the campus is the non-smoking campus	Three points
	8. All the campus area can meet the requirement of the national standard of acoustic environment quality.	Three points
	9. the ratio of campus area meeting the requirement in the related standards of lighting environment quality to the total campus area is larger than 70%	Three points

3.3 Evaluation indices of campus culture and their indices scores

The indices of campus culture is used to assess the progress and implement of green teaching, green scientific research, and green humanities, with general item a total of 80 points. Green teaching is mainly to measure the efforts and effectiveness of the setup of sustainable courses and programs, and the construction of

green education base. Green scientific research is mainly to assess the research achievement in the areas of sustainability, and the policies set to support the research. Green humanities assessment involves the evaluation of green campus training, seminars and other promotional measures, green movement and behavior of energy-saving incentive mechanisms, and related student association activities and student competitions.

Table 3 evaluation indices of campus culture and their indices scores

Categories	Indices	Scores
Green education (30 points)	1. The ratio of undergraduate courses in sustainability to total courses	<5%, 1 point; <10%, 2 points; <15%, 3 points; <20%, 4 points; >20%, 5 points.
	2. The ratio of graduate courses in sustainability	ditto
	3.the ratio of the schools with the undergraduate courses in sustainability to total number of schools	<20%, 1 point; <40%, 2 points; <60%, 3 points; <80%, 4 points; >80%, 5 points.
	4. the ratio of the schools with the graduate courses in sustainability to total number of schools	ditto
	5. Establish green campus demonstration bases.	1 point for one base, top 5 points
	6. Have the initiatives to encourage teachers to set sustainable courses and make curriculum reform.	2 points
	7.have the initiatives to set the discipline or program in sustainability	3 points
green scientific research	1.the ratio of research project numbers in sustainability to total number.	<10%, 1 point; <20%, 3 points; <30%, 5 points; <50%, 8 points.
	2.the percentage of the research funding in sustainability	ditto

green humanities (29 points)	(21 points)	3.1 Have initiatives to encourage teachers to do the research in sustainability.	3 points
		3.2 Have initiatives to encourage students to do the research in sustainability.	2 points
		1. Provide the green campus trainings	2 points
		2. Schools provide the periodic lectures and workshops in sustainability.	Once for each semester in each school, 1 point; twice, 3 points, above 3 times, 5 points.
		3. Incentive mechanisms for green movement and behavior	Establish and fully implement, 5 points; Establish and partly implement, 3 points; Just establish, 1 point.
		4. Posters or brochures for green campus	0.5 point for each, 2 points for more than 4.
		5.1 the number of student associations in sustainability	0.5 point for each, top 2 points
		5.2 the number of activities held by student associations.	Once per semester for each association, 0.5 points; top 2 points
		6. The student competition award above the municipal level in sustainability each year.	International competition, 4 points; National award, 3 points; provincial award, 2 points; municipal award, 1 point; top 10 points.

3.4 Evaluation indices of organization and management, and their indices scores

The indices of organization and management mainly involve the construction of organization and system; with one control item and general item altogether 45 points. This part is mainly to evaluate the foundation of management committee, professional committee, and

executive office, the designation of the person in charge in each school and department, and professional personnel with skill qualification in the specific position of management and logistics. Meanwhile, a set of regulations should be set up to grantee and stimulate and efficient operation and management.

Table 4 evaluation indices of organization and management, and their indices scores

Categories	Indices	Scores
Organization (1 control term and 18 points) mechanism construction (27 points)	1. establish the management committee	control term
	2. Establish the executive office, responsible for the concrete work	5 points
	3. Designate one person in each school or department to charge the implementation of all the tasks set by university.	5 points
	4. Have professional staffs in the campus energy management and logistics.	3 points
	5. Establish the expert committee to provide technical support	5 points
	1. Make medium and long term plans for the sustainable development.	3 points
	2. Set up and perform inspection regulations for green campus construction projects.	3 points
	3. Establish and perform the annual audit for campus building energy consumption	3 points
	4. Establish and perform the campus energy quota regulation	3 points
	5. Compose and publish the annual report of the campus sustainable development.	3 points
	6. Establish and perform the regulations for energy efficiency operation of facilities	3 points
	3 Establish and perform the green procurement regulations.	3 points
	8. Establish and perform the waste recycling management regulations	3 points
	9 Property management department pass the ISO14001.	3 points

3.5 Evaluation indices of social service, and their indices scores

Evaluation of social services mainly relates to green technology training and consulting provided to the

community, sustainable policy support, the project among firms, universities and research institutes, as well as students participating in community sustainable practices, with general items a total of 25 points.

Table 5 evaluation indices of social service, and their indices scores

Categories	Indices	Scores
social service (25 points)	1. Offer green technology consulting and training for community.	1 point each; top 5 points
	2. Support policy making for green energy, green building, green cities.	ditto
	3.1 Provide support to encourage staff and students to set up the company in the field of sustainability	5 points
	3.2 projects among firms, universities and research institutes	1 point each; top 5 points
	4 the times of students' participation in sustainable activities in community or city.	1 point for 5 times, top 5 points

4. CONCLUSIONS

Aiming at the current situation and characteristics of green campus construction in China, an evaluation methodology for the Chinese green campus construction is put forward. The indices cover five aspects of resource-saving, environment-friendly, campus culture, organization and management, regional collaboration and social services, so as to fully evaluate the initiatives and effectiveness of green campus construction and operation. The indices include control items and general item. The control item must be met. The general item takes the form of scoring, and the green campus construction and operation in different colleges and universities can be compared according to the total value of the five categories of indices.

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RENEWABLE ENERGY AND APPROPRIATE TECHNOLOGY PERCEPTION IN RURAL COMMUNITY IN THAILAND

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ABSTRACT

It is now widely acknowledged that the Thai government needed to increase renewable energy capacity particularly in rural communities. The study aims to explore the use of renewable energy in a community in Thailand covering their knowledge on renewable energy, renewable energy use behavior, and public participation. The survey was carried out through the sample of 120 households from 12 subdistricts located in a self-help settlement of the Nan river, Uttaradit province, Thailand. A questionnaire survey and in-depth interview provided quantitative and qualitative data. The findings state that most of people in the community know and regularly use renewable energy in many ways particularly charcoal for cooking. Even they have both liquid petrol gas (LPG) and charcoal to use for cooking, they prefer to use LPG only in case of urgent works. There is only one subdistrict that people realize and perceive to use renewable energy. One of the key success factors is the role of their community leaders and the government agency who have sufficient budget to manage and promote to use renewable energy. One of the weaknesses in promoting renewable energy is the lack of public relation from government agency through community. Furthermore, people use solar energy to dry fresh foods such as cashew nuts, and fishes. Other renewable energy is scraps of wood from small furniture factory to make charcoal. Actually, community has used renewable energy for more than ten years but has just been supported by the government agency for the past 5 years. The example of supporting solar energy is granting solar energy equipments to the households locating in the area without electricity. However, the lack of training the members of community to maintain the solar energy equipments cause more than a half of these households unable to use solar energy as they desired. Regarding to the technology perception, it is shown that high efficiency charcoal stove, solar drying equipments, bio-gas stove, and solar cell for lighting are in requirements. In conclusion, the results lead to policy decision maker to promote renewable energy in this area through the three significant parties i.e. community leaders, supporting budget, and public relations.

Key words: Renewable Energy; Rural Community; Public Perception.

1. INTRODUCTION

The problem of energy imports is one of the most important problems in many countries particularly for Thailand. It was reported that, in 2011, Thailand imported fuel energy 80% of the country consumption and will import more and more in the future [1]. The major government agency such as the ministry of energy who monitors the situation of Thailand energy consumption then determined the policy with strategies to promote the efficient use of energy. One goal in policy is to promote the use of renewable and alternative energy to achieve the target of 25% within the next ten years (Alternative Energy Development Plan : AEDP, 2012-2021). The policy also aims to succeed in sustainable energy use both at community and country level. A crucial strategy of energy management that the government agency uses is to support the use of renewable energy at community level

[2]. Because the communities have distinct facility structure, environment, and energy sources, the agency has to account for the characteristics of each community [3]. These factors can lead to the appropriate potential evaluation of various communities. For example when considering the energy sources to use in household, biomass (e.g. fuel wood, energy crops, agricultural residues, food waste, and industrial waste) may be required to be considered as an important source of energy for household use [4].

The self-help settlement of the Nan river, Uttaradit province, was selected to be the sample under study because of its area and management. This community is a new area that the government provided and it locates near the Sirikit hydro power plant. The area can be used as a pilot area of renewable energy management in Thailand.

The objectives of this study are to investigate the knowledge of people in community, energy use

behavior and technology requirements of people in community, and the problems concerning technology use in community.

2. METHOD

2.1 Respondents

The community used as the sample in this study is the self-supporting settlement of the Nan river, Utharadit province, Thailand. The area of the community was provided by the government to help people who used to live near the Sirikit dam [5]. This community contains 12 sub-districts, so in order to make the sample covers all of sub-districts, 10 households of each sub-district were randomly selected, by simple random sampling method. The sample size is then 120 households from 12 sub-districts of the community. The respondents are the representative of each household in the sample.

It was found, about general characteristics of the respondents, that most respondents (72%) are farmers, and their incomes vary highly. Almost all (94%) of the respondents have their incomes not greater than 8,000 baht/month. Most of the respondents (43%) finished only primary school and 30% of the respondents finished 3-year of secondary school.

2.2 Data Collection and Measures

This study used questionnaires and interviews as the tools in data collection. The respondents are the household representatives. The content of survey covers 4 aspects : knowledge (of community) on renewable energy, behavior of community involving renewable energy use, requirements of community to be supported, and problems of community in renewable energy use. The data from both questionnaire and interview were analyzed together. The analysis of content focused on the promoting of renewable energy use in community.

3. RESULTS AND DISCUSSION

In overall, the knowledge on renewable energy of the community is at low level, even many households have used renewable energy regularly in some ways, for example they use charcoal in daily cooking. The types of renewable energy that the community know most are biomass specially for hydro power and solar energy (as shown in Fig. 1). The community gets benefits from renewable energy in such a way that this type of energy provides convenience to their daily lives. It was also found, from the interview, that more than a half (62%) of the people in this community realize that renewable energy is useful for their lives at community level and for the country.

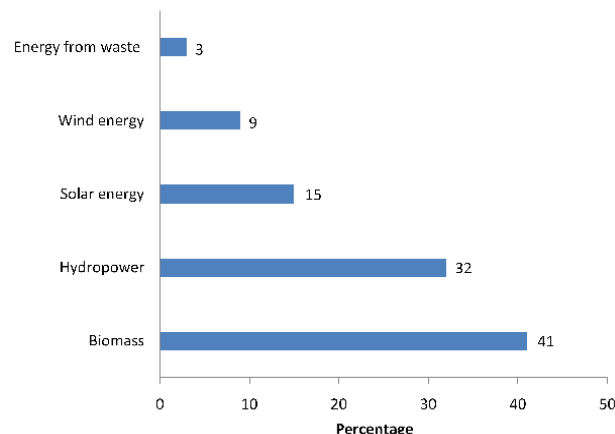


Figure 1 Recognition of renewable energy in the community

On the behavior of community on renewable energy use, it was found that most (57%) of the renewable energy are used for cooking, the second most (24%) are for drying agricultural and fishery products, the third most (13 %) are for burning to be charcoal, and the few (4%) are used for solar cell for lighting (as shown in Fig. 2).

For the requirements of community to be supported, it was found that most (65%) of the people in community require the efficient cooking stove that can save energy by use of less charcoal and wood as fuels. The second most (15%) of the community desire solar dryer for agricultural and fishery products which support their occupation. The third most (13%) of the respondents require the technology to produce biogas stove, and the rest (7%) of the community need the solar photovoltaic technology to produce electricity for in-community use (as shown in Fig. 3).

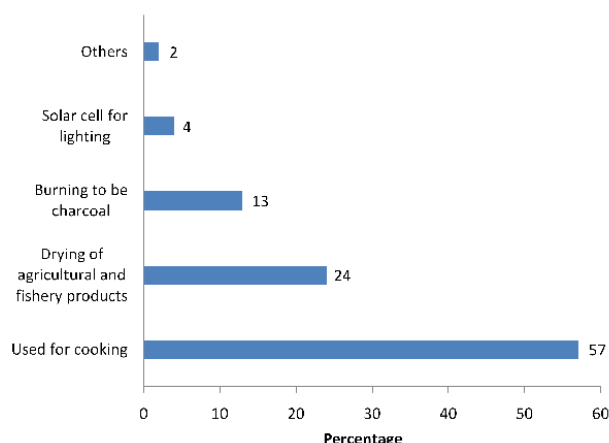


Figure 2 Use of renewable energy classified by activities

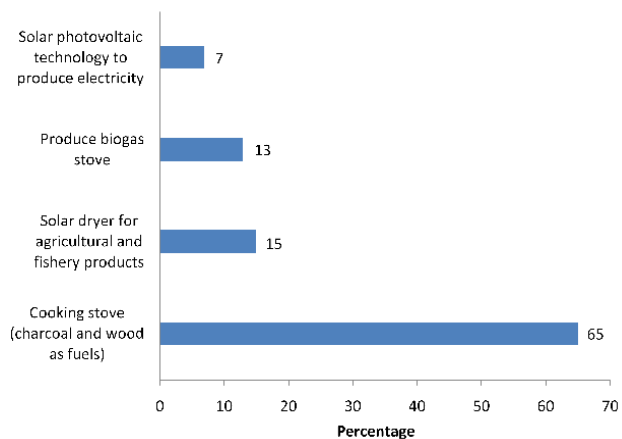


Figure 3 Community requirements of renewable energy technology



Figure 4 Charcoal cooking stove



Figure 5 Fuel wood cooking stove



Figure 6 Biogas for cooking



Figure 7 Solar cell for lighting



Figure 8 Solar dryer for agricultural products



Figure 9 Solar dryer for fishery products

In addition, it was found, from the interview, that more than a half (58%) of the people in community do not know that there is the promoting of renewable energy use. But for the ones who know, the significant sources are the leaders of the community such as sub-district head and village head. These persons play important role in supporting renewable energy use. The other sources are local government agencies and mass media such as television, radio, and published document (as shown in Fig. 10).

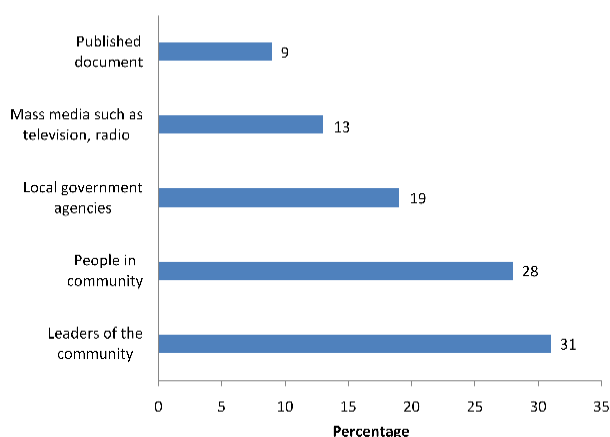


Figure 10 Sources of promoting renewable energy use

On the problems of community in renewable energy use, it was found that most of the people (65%) face the problems after use of renewable energy whereas the rest (35%) do not have any problem. The problems can be grouped as the high cost of investment in renewable energy equipment (43%), the academic knowledge on renewable energy (37%), and the problem in maintenance the equipment used (as shown in Fig. 11).

On restrictions of renewable energy use, it was found that most of the people in the community (48%)

do not have enough fund to obtain the equipment involved. The second most (38%) of the group do not have enough knowledge and experience in operation. In addition, the two influential obstacles in supporting renewable energy use are the community leaders do not firmly support renewable energy use and the lack of the cooperation of people in community (as shown in Fig. 12).

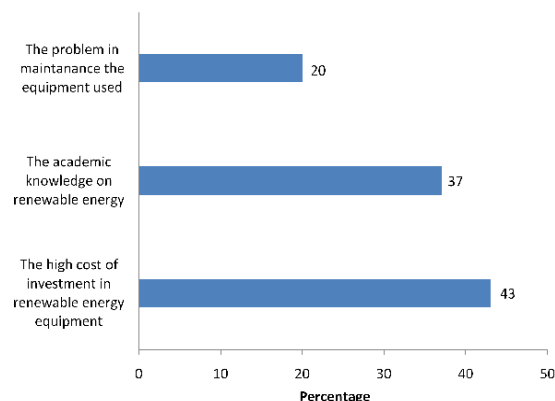


Figure 11 Problems in renewable energy use

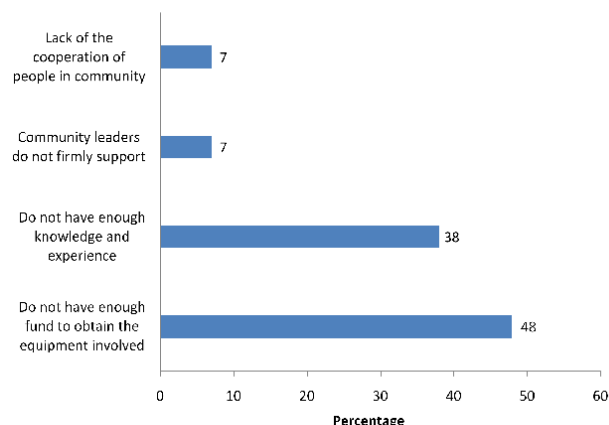


Figure 12 Restrictions of renewable energy use

4. CONCLUSION AND RECOMMENDATIONS

The self-supporting settlement of the Nan river, Utharadit province is the community under the responsibility of government agency. This area can be a pilot of community to promote renewable energy use. The findings, from the study, have identified the problem on lack of knowledge of people in community. Because of their agricultural occupation, the people do not have enough time to learn further knowledge on renewable energy. However, there are a lot of people in community who started to use renewable energy but they still need many supports such as the information or technical knowledge on renewable energy use and the firm supporting from community leaders [6]. These supporting factors do important affect for future study.

The study of requirements for renewable energy technology, that are suitable for the environment, found that the most needed subject is the technology to produce efficient cooking stoves with saver energy.

This kind of stove is very useful for community in daily cooking. The second most needed subject is the biogas stove. The community requires to produce the equipment involved the biogas stove by themselves to save cost of buying the new one. This kind of stove can be extended to mass production and developed to community business [7]. In addition, the community also needs assistance from the government agencies in supporting technical knowledge and funding [8].

About the current problems in renewable energy use, it was found that local agencies do not thoroughly communicate the use of renewable energy to the people in community. The other problem is the high price of the equipment involved. Therefore the promoting of renewable energy use should be done with the supporting in financial resource.

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EXPLORATION ON THE CONSTRUCTION OF LOW-CARBON CAMPUS FROM THE PERSPECTIVE OF GREEN CONCEPT

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ABSTRACT

Green resources utilization is related to the sustainable development of colleges and universities, and the green society calls for conservation-oriented campus construction. In the course of the construction of low-carbon campus, we faced a lot of difficulties and problems, such as solving instructional resources sharing, non-renewable resources of land, and the seriously exceeded use of water, electricity, coal (gas). Colleges and universities should actively convert into a low-carbon, energy-saving environmental protection way, establishing the new patterns of low-carbon life. The key to construct low-carbon campus lies in changing ideas and adhering to the low-carbon concept. A series of measures are adopted, including management of daily operations, life philosophy and life style of teachers and students.

Key words: green concept; economy; low-carbon campus; social responsibility; construction

1. INTRODUCTION

The "green movement" in Western countries at the end of nineteen sixties means "ecology" or "environmental protection", and "green" has become "sustainable development" instead of the word[1]. The basic idea is harmony, inclusion and sustainable development. The construction of low-carbon campus is an important part of implementing the scientific outlook on development, and also part of constructing a harmonious society. So, Construction of low-carbon campus is not only an efficient practice of the construction of Harmonious low-carbon Society, but also it is needed for colleges and universities to undertake the social responsibilities. The construction of low-carbon campus is micro-embody of low-carbon society construction[2]. In order to set up and well implement "green" concept, it is an effective choice for colleges and universities to practically construct low-carbon campus.

2. BACKGROUND OF LOW-CARBON CAMPUS CONSTRUCTION

2.1 Low-carbon campus construction is the inevitable requirement to build a conservation-minded harmonious society

In order to implement the scientific outlook on development and let people live in vigorous worlds, it is required to rebuild a green world for low-carbon society construction. This can establish harmonious human-society-nature relationship, realize the optimization of social ecological system and sustainable development,

and achieve the greatest social & economic benefits and ecological benefits

2.2 The construction of low-carbon campus is a rational choice for China

The green concept is not only a change of energy consumption system, but also the future direction and mode of a circulation system[3]. Green, environmental protection, energy saving, low carbon and low emissions as a symbol, the foundation is to build green education system and build a low-carbon energy support system. The core is to increase new energy technology application in the university's energy support system and promote innovation of energy-saving management system and mechanism.

3. PROBLEMS AND CONTRADICTIONS WIDESPREAD IN THE LOW-CARBON EDUCATION OF CURRENT COLLEGES AND UNIVERSITIES

3.1 Resources waste of water, electricity and coal (gas)

Power consumption has been a primary source of resource consumption[4]. However, these phenomena were particularly prominent in colleges and universities. The canteen left the tap running; bedroom, corridor and teaching buildings left the light on; scientific research institutions and offices' water and electricity without control; and air conditionings and fans were normally under way. This situation brought the economic burden to the colleges and universities.

3.2 The waste of land resources

Due to unreasonable situations in the development and exploitation of land resources in the higher education, the waste of land resources was prominent, and it has caused the destruction of the ecological environment to a great extent[5]. Especially when the campus' scale crazed to vie with each other in some places, the bigger campus' scale, the more idle land. With the new campus built and students abroad, the old campuses have been various degrees of "empty". Furthermore, China's land resources are very scarce. Campus students' average land standards were set higher. So it obviously does not meet the requirements of the scientific outlook on development, and it is not conducive to the sustainable development of education. Some colleges and universities cannot make full use of the new land. They have been in debt because of blindly expansion of new campus. Therefore, it is necessary to promote the sustainable utilization of the land resource.

3.3 The waste of Teaching resources

Every year, colleges and universities waste a lot of resource books. Whenever the students graduated, we could see a large number of books to be sold as waste paper, no reuse. In fact, some public course books can be used by the same professional grades. Colleges and universities purchased and issued the same version of the book each year, resulting in a low repeated utilization rate of books. In addition, the sharing of equipment resources did not realize between colleges and universities, between personnel in different departments, which formed a certain recessive waste.

4. STRATEGIES AND METHODS OF CONSTRUCTING LOW-CARBON CAMPUS

4.1 Renewing the old concept of resources and establishing a low- carbon and green idea

Firstly, establish a low-carbon investment idea. Because the current inputs to energy saving are main for energy saving long time. As some of the basic, long-term infrastructure construction should properly be advance the concept, and it is inefficient to transform a few years later. We should pay attention to the campus culture construction and create a green culture.

Secondly, improve the environmental protection and the consciousness among sustainable development of the students by strengthening the green education. Colleges and universities should fully play a leading role in talents, science & technology and culture, advocate popular civilization, vigorously promote green education, cultivate of teachers and students' awareness of environmental protection and ecological concept, and guide the students to form a good habit of green consumption and civilized behavior.

Finally, establish a low-carbon lifestyle by assiduously propagating the green concept. Through the

University environmental education and demonstration, Colleges and universities promote simple lifestyle and advocate a positive practice of green travel and low-carbon life. Inculcate water saving, energy saving and recycling use of resources through influencing character by environment. It has been an important means to strengthen the cultivation of the students' awareness of low-carbon and ecological civilization construction and sustainable development of the construction of low-carbon campus. The establishment of consciousness "Conservation of resources is everyone's responsibility" is much more important than the technology. Only teachers and students make the saving awareness into conscious action can we really implement effective energy saving, and then realize the unification of individual development and social development.

4.2 Integration of resources and resources sharing

Energy is limited in a certain range, and it is lacked of balance[6]. The integration and sharing between the school and the society, between or among the school and the school, can not only save initial resources, but also save operation during the process of resource consumption.

Using the special intellectual resources to strengthen scientific research innovation on new energy-saving technology and new energy, it also helps us better conserve resources. Solar energy and other new energy should become a new generation of energy. We can give priority to the promotion of new energy in Colleges and universities in order to the progressive realization of "self-reliance" and establish the real low-carbon universities.

4.3 By means of innovation, make the sustainable development through the construction of Low-Carbon Campus come true

To realize the sustainable development through the construction of low-carbon campus, we must always adhere to innovation.

First of all, develop and popularize the technology of energy-saving and emission reduction, giving full play to the advantages of scientific research. Higher education has the advantages and great potential for development in scientific research of energy-saving emission reduction and energy efficiency. Therefore, first of all, during the construction of low-carbon campus, universities should take energy conservation and improve resource utilization as the core, and take engineering construction of energy-saving, water-saving and emission reduction as the focus, advocating low-carbon life. Secondly, colleges and universities should vigorously reduce carbon emissions by the development and promotion of energy-saving air-conditioning technology, green lighting technology, wastewater treatment and comprehensive utilization technology. During development and promotion to applicable technologies of conservation, substitution,

recycling, solar and other clean energies are especially actively made use of.

Secondly, strengthen carrier innovation in construction of a conservation-oriented campus. Vigorously promote the construction of a logistics information platform, make full use of the network information characteristics of strong sharing and wide coverage, expanding the channels and means of construction of a conservation-oriented campus. Such information needed to share and summary statements as an equipment condition of service, warehouse goods, details of the financial situation, and personnel details can query to the information platform system in unity. At the same time, the office network can economize recessively, inspection and supervision departments on the campus without walking back and forth. Some manual accounting department will be reduced, and even some departments can be reduced with.

Finally, actively promote and utilize new technology and equipment. Do well in the construction of low-carbon campus, colleges and universities should take campus planning, implementation of the system, measures to promote the use of new technology.

There are mainly two measures: one is the appliances use of water-saving and energy-saving, such as campus greening water using hose sprinkler irrigation system; the other is the introduction to new equipment and technology, such as bathing, washing water using a solar heater. It can save energy and money, reducing coal burning pollution. However, the extension of the use of these new technologies and equipment requires prophase investment of the government and relevant departments. Give full play to the role of teachers and students, and create a low-carbon campus atmosphere

Colleges and universities carry out various forms of energy-saving and emission reduction activities within the whole school, culture the concept of advocating conservation for teachers and students, so that energy-saving emission reduction reflected in the university teaching, management, campus culture and another aspects. And start from the little things and details, develop good saving habits, consciously to eliminate the use of waste of resources products, make thrift a traditional virtue of consciousness through the teachers' and students' daily life and the consumption behavior, and promote the all-round development within the individual. For the construction of low-carbon campus, promoting social development and promoting low-carbon is our inescapable responsibility[7].

Create a low-carbon campus atmosphere and build a low-carbon education carrier. Campus culture is a recessive carrier for ecological civilization education of College Students. It not only affects people's thoughts and their feelings and life, but also purifies the soul.

Low-carbon campus environment requires the harmonious campus culture atmosphere. Therefore, through the creation of a harmonious campus culture, make the students feel the importance of ecological civilization of the campus environment. Restrain their words and deeds according to the requirements of the

standard needed by the environment. And put it into a habit of moral sentiments and their behavior. And at last establish the concept of ecological civilization, which can have the lasting impact on their future work and life.

PROSPECT

As China's economic growth mode is gradually perfect, construction and development of Low-Carbon Campus in Colleges and universities will be paid more and more attention. Bearing significant social responsibility, universities must strengthen the sense of hardship and take a good start in the construction of low-carbon society[8]. Constructing a conservation-oriented campus is not in a short period of time to complete, it needs widespread support and participation for teachers and students in colleges and universities and all social strength. This is a long-term, arduous, and historic work.

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EVALUATION AND OPTIMIZATION OF SUN-SHADING PERFORMANCE FOR AEROGEL GLASS IN HOT SUMMER AND WARM WINTER ZONE

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ABSTRACT

Based on the library building of Guangzhou University, the integrated building virtual environment simulation software IES<VE> was used to calculate the nine components of building cooling loads for five types of glass such as common clear glass, insulating glass and newly developed aerogel glass. It can be observed that window solar component is one of the greatest cooling load components for all the five types of glass. The results show that the shading coefficient of glass is the key design consideration of exterior building glass for building energy efficiency in hot summer and warm winter zone. Furthermore, the glass sun-shading mechanism was analyzed relevant to the inferred and visual light transmittance, and the effects of the inferred and visual light transmittance on the shading coefficient of glass and the building cooling loads were evaluated. The shading coefficient of glass reduces by 53.8% and the window solar component reduces by 43% as the inferred transmittance decreases from 1 to 0 and the visible transmittance keeps at 0.7. The research results indicate that decreasing the inferred transmittance of glass is an effective way to reduce the shading coefficient and building cooling load. It also can provide references for the optimization of aerogel glass sun-shading performance and the building cooling energy efficiency in hot summer and warm winter zone.

Keywords: sun-shading, aerogel glass, infrared transmittance, cooling energy efficiency, hot summer and warm winter zone

1. INTRODUCTION

In 2010, China's building sector accounted for 20.9% of China's total national energy consumption [1]. Of the annual energy use of public buildings, approximately 50-60% is for HVAC (Heating, Ventilating, and Air-Conditioning), and 20-30% is for lighting [2]. As an indispensable element of the exterior building envelope, glass windows meet the requirements such as daylight, ventilation, view and esthetic. However, the poor thermal performance of common building glass widely used in China leads to a high heating and cooling energy demand. Therefore, the study on the thermal performance of the exterior building glass has attracted increasing interests aiming to building energy efficiency.

Very recently, aerogel has become a promising material for application in highly energy efficient window glass owing to its extremely low thermal conductivity and high visual transmittance [3]. Schultz et al. [15] developed a newly evacuated aerogel glazing of with the area of $55 \times 55 \text{ cm}^2$ and the aerogel thickness of 15 mm. The aerogel glazing had the heat transfer coefficient of $0.66 \text{ W/(m}^2 \text{ K)}$ and the solar transmittance of 76%. The few literatures [3-6] have

shown that the aerogel glazing is the ideal material which has the excellent combination of the high solar and light transmittance and the low thermal conductivity for colder climate zones. However, it is the shading coefficient (SC) of glass that is the key design consideration in warmer climate zones.

In China, large quantity of research focused on the effect of shading coefficient (SC) of glass on the building energy consumption [7-9]. However, little work has been directed towards the effect mechanism of the inferred and visual light transmittance on the shading coefficient (SC) of glass and the building cooling energy consumption. In this paper, based on the library building of Guangzhou University, the integrated building virtual environment simulation software IES<VE> was used to calculate the nine components of building cooling loads for five types of glass such as common clear glass, insulating glass and newly developed aerogel glass. Furthermore, the glass sun-shading mechanism was analyzed relevant to the inferred and visual light transmittance, and the effects of the inferred and visual light transmittance on the shading coefficient of glass and the building cooling loads were evaluated. It is expected that this research can provide references for the optimization of aerogel

glass sun-shading performance and the building cooling energy efficiency in hot summer and warm winter zone.

2. SELECTION OF MODEL BUILDING AND FIVE TYPES OF GLASS

2.1 Model building

The library of Guangzhou University is selected as a model building in this study. The library is a 5-storey building with the first floor height 4.2 m and other four floors height 4.5 m, and the glass ceiling height 9 m. The total floor area is 41160.35 m². The azimuth of the model building is south. The heat transfer coefficients of outer-wall and roof of the model building are 1.47 W/(m²·K) and 0.89 W/(m²·K), respectively, which conform to the rules in the “Design standard for energy efficiency of public buildings (GB 50189-2005)” formulated by Ministry of Construction of P. R. China. Based on the characteristics of the building, the drawing of IES<VE> model of the library building is shown in Figure 1

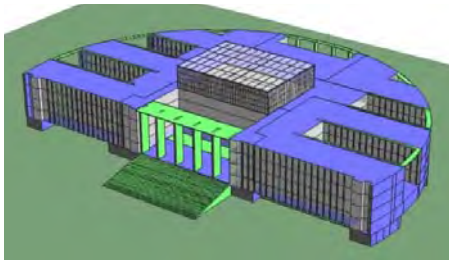


Figure 1 The IES<VE> Model of library building in Guangzhou University

The indoor design parameters include the occupancy density of 10 m²/person, the lighting power density of 18 W/m² and the electric facility power density of 5 W/m². The indoor design cooling temperature and the relative humidity are set at 26 °C and 40-65%, respectively. The infiltration rate is designed according to the number of ventilator of 0.45 per hour. The service period of the HVAC and the lighting system in the building is 8:30 am to 9:30 pm from May 1st to July 16th and September 1st to October 31st during the year.

2.2 Five types of glass

Five types of exterior building glass are chosen to determine the hourly building cooling loads in Guangzhou so as to evaluate the feasibility of different

glass in hot summer and warm winter zone. The glass #1 is common clear glass, the glass #2 and #3 are insulating glass. The glass #1-3 are selected according to the conventional exterior building glass providing in the Chinese industry standard JGJ/T151-2008 [10]. The glass #4 and #5 are newly developed aerogel glass with super-thermal insulation performance [4, 5]. The aerogel glass #4 and #5 have the same heat transfer coefficient (U) but different shading coefficient (SC). The detailed structure and thermal characteristics (e.g. heat transfer coefficient (U) and shading coefficient (SC)) of the five types of exterior building glass are shown in Table 1.

3. BUILDING COOLING LOADS FOR FIVE TYPES OF GLASS

The hourly cooling loads of the model building are calculated by using the integrated building virtual environment simulation software IES<VE>. The building cooling load comprises of nine parts. Four parts of the building cooling load are owing to the heat loss/gain through building envelopes (e.g. wall conduction, roof conduction, window conduction and window solar). Three parts of the cooling load are owing to the interior heat loads (e.g. occupants, electric lighting and office equipment). The other two parts are owing to the heat through infiltration and the heat conduction of the interior surfaces [11].

Figure 2 shows the composition of the component cooling loads for five types of exterior building glass in Guangzhou which is a typical city of hot summer and warm winter zone, it can be observed that for the clear glass #1, insulating glass #2 and aerogel glass #4 the largest cooling load component is the window solar component with 1306, 1283 and 1271 MWh, respectively. It is because that the three types of glass (viz. #1, #2 and #4) have higher shading coefficients of 0.84, 0.86 and 0.85, respectively, which means greater solar heat gain and accordingly higher cooling load. For the other two types of glass (viz. #3 and #5) with lower shading coefficients of 0.62 and 0.71, respectively, the window solar components are decreased and become the second largest cooling load components slightly smaller than the lighting components. On the other hand, the window conduction components owing to the U values are extremely small for all the five types of glass compared to the window solar component owing to SC values. It indicates that lowering the SC value of glass would be an efficient way to reduce the building cooling load.

Table 1 Detailed structure and thermal characteristics of exterior building glass

Glass type	Structure	Heat transfer coefficient U (W/(m ² K))	Shading coefficient SC
Clear glass #1	12 mm clear glass	5.5	0.84
Insulating glass #2	6 mm clear glass outer / 12 mm air interlayer / 6 mm clear glass inner	2.8	0.86
Insulating glass #3	6 mm low-emissivity glass outer / 12 mm air interlayer / 6 mm clear glass inner	1.9	0.62

Aerogel glass #4	4 mm low-iron glass outer / 15 mm aerogel interlayer / 4 mm low-iron glass inner	0.72	0.85
Aerogel glass #5	4 mm clear glass outer / 15 mm aerogel interlayer / 4 mm clear glass inner	0.72	0.71

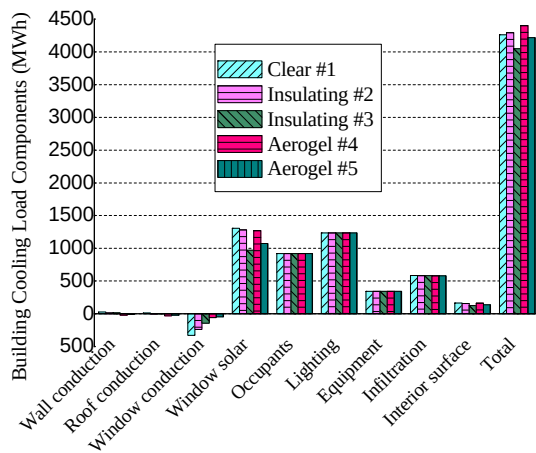


Figure 2 Building cooling load components for the five types of glass

4. OPTIMIZATION OF SHADING COEFFICIENT OF GLASS FOR ENERGY EFFICIENCY

It is the shading coefficient (SC) not the heat transfer coefficient (U) of glass that is the key design consideration of exterior building glass in warmer climate zones. Some authors [8, 9] have studied the effects of shading coefficient on the building cooling energy consumption for the existing common glass types. They found that the reduction of the SC value of glass can effectively decrease the cooling energy consumption. However, for the common glass, the decrease of the SC value has to be achieved by simultaneously reducing the inferred and visual light transmittance. This will degrade the building indoor natural lighting and increase the artificial lighting energy consumption. Recently developed aerogel glass can get high visible light transmittance and block the solar inferred radiation under building environmental temperature besides the extremely low thermal conductivity. These provide great possibility to obtain high energy performance building glass with high visual light transmittance and low inferred transmittance. But little work has been directed towards the effect mechanism of the inferred and visual light transmittance on the SC value of glass and the building energy consumption up to now.

The glass sun-shading mechanism was modeled and analyzed briefly relevant to the visual light and inferred transmittance [12]. From the analysis and calculation on the SC value of glass, it can be found that the direct transmittance of solar radiation is the key factor influencing the SC value of glass.

The direct solar heat radiation mainly distributes in the visual light wave bands (43.0%) and the inferred

wave bands (48.3%). To reduce the entrance of the solar heat radiation into the indoor space, the exterior building glass requires decreasing the transmittance in the inferred and visual light wave bands. However, the exterior building glass requires a higher transmittance (e.g. 0.7) in the visual light wave bands to meet the natural lighting of building. Therefore, the sun-shading performance of the ideal exterior building glass could be illustrated in Figure 3 which shows that the visual light and inferred transmittance of the ideal exterior building glass are 0.7 and 0.0, respectively.

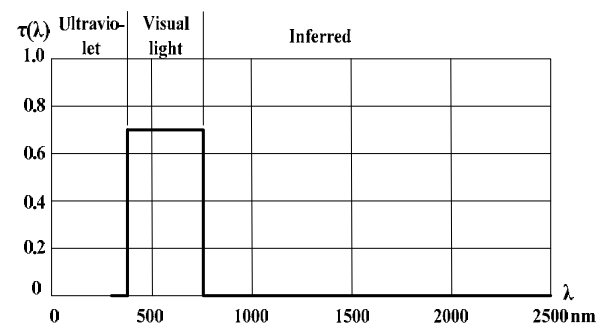


Figure 3 Sun-shading performance of ideal external building glass

Figure 4 shows the shading coefficients for different inferred transmittance calculated according to Eq. (3) and Eq. (4) as the visual light transmittance of the exterior building glass is 0.7. It can be found that the SC value of glass significantly reduces from 0.916 to 0.423 (reduces by 53.8%) as the inferred transmittance decreases from 1.0 to 0.0 and the visual light transmittance keeps at 0.7 that could provide sufficient natural lighting for the building indoor space. It means that decreasing the inferred transmittance of exterior building glass can significantly reduce its shading coefficient.

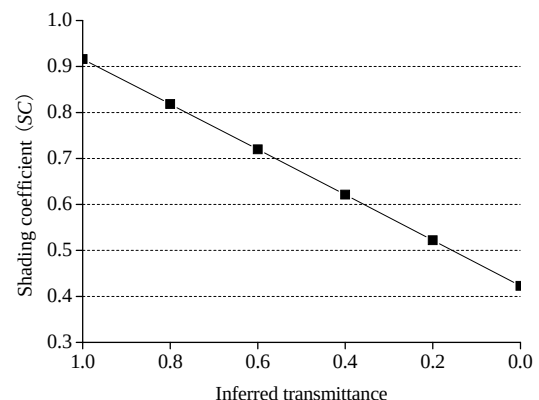


Figure 4 Effect of inferred transmittance on the shading coefficient

The major components of building cooling loads are simulated to analyze the effects of the inferred and visual light transmittance of glass in the hot summer and warm winter zone.

Figure 5 shows the major components of the building cooling loads as the SC value of glass varies from 0.916 to 0.423 and the U value of glass keeps at 3.5 W/(m² K). It can be observed that when the SC value of glass varies from 0.916 to 0.423 which means the inferred transmittance decreases from 1.0 to 0, the building cooling load reduces 528 MWh, correspondingly. The greatest cooling load component - the window solar component reduces 597 MWh and reduces abruptly by 43%. It means obviously that decreasing the inferred transmittance of exterior building glass can distinctly reduce the building cooling load.

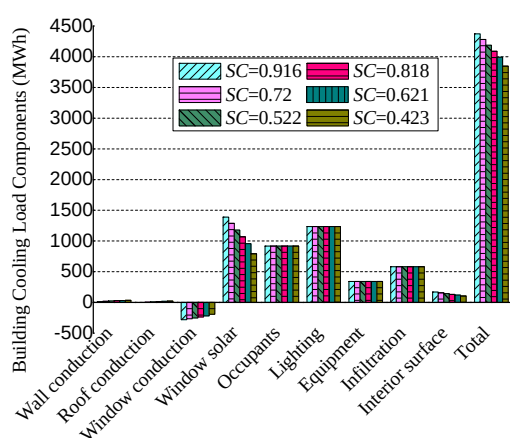


Figure 5 Building cooling load components for different shading coefficients

5. CONCLUSIONS

The component cooling loads of a model building for five types of glass (e.g. clear glass #1, insulating glass #2 and #3, and aerogel glass #4 and #5) are calculated in Guangzhou as a typical city of hot summer and warm winter zone by using the integrated building virtual environment simulation software IES<VE>, the following conclusions can be obtained:

(1) The results indicate that the window solar component relevant to the SC value of glass are the greatest one in the nine cooling load components as the SC value is larger. It is the shading coefficient (SC) not the heat transfer coefficient (U) of glass that is the key design consideration of exterior building glass in hot summer and warm winter zone.

(2) The shading coefficient (SC) of glass can be significantly reduced from 0.916 to 0.423 (reduces by 53.8%) as the inferred transmittance decreases from 1.0 to 0.0, and the visual light transmittance remains constant at 0.7, which can provide sufficient natural lighting to the indoor space and not increase the artificial lighting energy consumption. It indicates that lowering the SC value of glass would be an efficient way to reduce the building cooling load.

(3) It can be observed that when the SC value of glass varies from 0.916 to 0.423 which means the inferred transmittance decreases from 1.0 to 0, the greatest cooling load component - the window solar component reduces by 43%. The findings of the effect mechanism of the SC value of glass on the building cooling load could be helpful for evaluate the feasibility of the five types of glass #1-5 in hot summer and warm winter zone and further optimize the thermal characteristics of glass aiming to develop new high-performance glass for building energy efficiency.

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STUDY ON LOW-CARBON ENERGY PROBLEMS AT RURAL AREAS IN CHINA

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ABSTRACT

With the progress of the era, the word “low carbon” has become one of the hottest topics at home and abroad. The China, as the biggest developing agricultural country in the world, the carbon emissions in rural areas should not be underestimated, no matter it is come from the farmer’s traditional way of life or the way to producing crops. This paper through some effective ways to elaborate the energy-saving emission reduction, and the most important aspects is the way of energy using and the crop irrigation farming system. Only by establishing the perfect system of science and technology, the farmers only believe the fact. Only in this way, the low-carbon project and the low-carbon economy can be carried out smoothly in the rural, so this paper mainly describes some detail data and low carbon agricultural methods are currently fitted for the rural areas in China.

Keywords: low-carbon; low-carbon crop; biogas; energy conservation

1. INTRODUCTION

When the human civilization has developed during the period of the industrial revolution, the resources of the earth had increasingly decreased, and the concentration of carbon dioxide and other greenhouse gases in the atmosphere had a significant increase. The change with the concentration of carbon dioxide can cause the greenhouse effect which has big influence on the daily life of the human beings, and what’s worse, it will destruct the home of the human. As one of the signatory countries of Kyoto Protocol, China public the "China National Climate Change Program", which clearly pointed out that the development and utilization of the low-carbon energy sources in rural. The development of biogas in rural areas, cannot only solve the problem of the shortage of the rural energy, but also improve the agricultural ecological environment and the rural landscape, it played a very important role in developing the rural economy and reducing the greenhouse gas. In the rural energy, the proportion of coal played the most important role, and the coal releases a large number of carbon dioxide. On the other hand, biogas also has a great high calorific value, which can basically replace some fossil fuels, and it can reduce greenhouse-gas emissions. In China, most of the population is in the rural, and about 66.7% of the residents rely on the biomass energy [1]. Based on data, when the straw burned, the number of carbon dioxide emissions just under the coal. So it is a large space to the development of the biogas. Besides, the low-carbon crop production also makes some differences on the environmental protection and reducing the greenhouse gas. Low-carbon crop productions conclude the cultivation and promotion of the low-carbon row crop

varieties, low-carbon farming systems, low-carbon fertilization and low-carbon cultivate technology. During the "Eleventh Five-Year" period, the target of the China's low-carbon energy is a 20% reduction in energy consumption per unit of GDP and the major pollutants reduced by 10%. And to realize this goal, the work of rural low-carbon energy can't be ignored.

The first part of the article has discussed the background of the low-carbon economy, and the significance of the low-carbon agriculture development and some measures has to be taken. And in the second part, it takes a brief discussion of the biogas, including its concept, characteristics, functions and meaning. At the third part, from four aspects to explore the low-carbon crop production technologies and some measures, the aspects include low-carbon crop varieties, low-carbon farming systems, low-carbon fertilization raise and low carbon cultivation techniques. At the same time, it had better to take close connection with the current situation at home and abroad. It has new information and perspective to the development of the low-carbon economy, on the other hand, the low-carbon agriculture and low-carbon crop production has been realistic guiding significance and reference value.

2. TO DEVELOP BIOGAS IS ONE OF THE MOST IMPORTANT WAYS OF LOW-CARBON ENERGY IN RURAL

Compared with the city, the rural has the diversified carbon source. It is decided by the eco system of the rural. If it is used by an unsuitable way, it will be a large way to release the greenhouse gases, such as CH₄ and CO₂.

Low-carbon energy saving is based upon the concept of the sustainable development, as much as possible to reduce the consumption of high-carbon energy, to stave off global warming, and protect the homeland of the human beings. According to the state of the rural in our country, the realization of the low-carbon concept can't ignore to promote the use of biogas in rural. Currently, the potential new energy in rural mainly includes solar, small hydroelectric plant, biogas, wind energy, and so on, but a series of technical problems to use solar energy, it is difficult to solve in a short time, and the small hydropower stations rely on the water resources within the region, what's more, the early investment is relatively higher than others. And the use of wind is very limited in the region, and some technical problems and funding issues have not been resolved, but to the utilization of biogas, his raw material is rich in materials, rice and wheat straw, grass weeds, those materials are almost everywhere in the rural areas, and in the rural, every family feeds chicken and pigs, all these will become the raw material to produce biogas [2-5].

In Chinese rural areas, the majority of the crop is directly burned after harvest, the heat utilization rate is only 10% -20%, not only will it produce a lot of smoke but become an important pollution to the environment. Here is a part of statistics of the biogas energy consumption.

If it is fermented as biogas to burn, the utilization of the geothermal energy can reach 40% -60%, so there is a great difference between the utilization methane heat and direct burning. According to some statistics, China has about four hundred million tons of crop straw has been directly burned, which is a huge waste to use

agricultural resources. Secondly, when use the agricultural resources, you can also pay attention to use the bio-energy heat, to do this, will make full use of the organic matter, for example, take a part of the straw to feed the livestock to digestion and utilization, so that the nutrients can be converted into the meat, eggs, milk and other products. Then take use of the livestock's manure and the other part of the straw weed is dropped into the septic tank to produce biogas, the biogas can be used, and it can replace some of the farm household energy.

According to the statistics of Ministry of Agriculture in 2006, show that the farmers used approximately 900 million tons standard coal, and the commercial energy is about 600 million tons standard coal, which is almost a quarter of the national commercial energy consumption. Judging from these data, the discharge of carbon dioxide is not a small number. If the use of biogas in rural areas can be vigorously developed, the heat of the use of biogas is equal to 185 million tons of standard coal, it can reduce 45 million tons carbon dioxide emissions. Per cubic meter biogas value 5,600 kcal, which equivalent to 0.55 kg full released by diesel or 0.8 kg coal. It is consistent with the energy-saving initiatives. In addition, the popularity of the biogas, the phenomenon of the smoke in rural land will not exist, the garbage and the weed in front and behind the house will initiative to be cleaned up. The open-air pit will become a source of biogas, on this way, the rural environment will be improved. Some mosquito disease will be significantly reduced, and the health standards of the rural farmers will be greatly improved.

Table 1 The Energy Consumption in Rural Areas in 2000 Mtce

	Commercial Energy				Non-commercial Energy			
	Coal	Fuel oil	Electricity	Total	Stalk	Firewood	Total	Total
For production	175.27	45.55	64.69	285.51		14.96	14.96	300.47
For livelihood	118.01	7.57	34.44	160.02	123.60	80.52	204.12	370.00
Total	293.28	53.12	99.13	455.53	123.60	95.48	219.08	670.47

Values in table 1 are calculated to the figures from Energy Policy Research, 2002(1)(published by Chinese Energy Research Society). (2) Include liquid petrol gas.

3. LOW-CARBON CROP PRODUCTION IS ANOTHER IMPORTANT WAY OF LOW-CARBON ENERGY

The soil, as big carbon storage is a carbon source, the function of the carbon in soil makes a great influence on the climate and the greenhouse-gas emissions. Moreover, the soil is the converter of the carbon form. In agricultural production, to protect soil is very important. The proportion of the organic manure in rural areas has increased year by year, these years it had increased to 150 pounds per acre, as the increase of

the organic fertilizer, the soil, which survived many kinds of microbes has been damaged, and a variety of mineral elements in the soil have seriously disordered. Moreover, if the fertilizer is continual used also lead to denitrification, which increased N_2O emissions. Over time, the farmland soil will be compaction, and the water will be polluted.

The excessive fertilizer has made the soil deteriorate, and the unsuitable way of farming let the soil nutrients run off. To develop the low-carbon crop production has no time to delay.

Table 2 The Increase in Fertilizer use in China

Year	1957	1978	1990	2000	2007
Usage amount(ten thousand ton)	7.8	884.0	2590.3	4146.4	5107.8
Nitrogen fertilizer usage(ten thousand ton)			1638.4	2161.5	2297.2

Data source, China Rural Statistical Yearbook (2005-2008)

With the development of the science and technology, the low-carbon technologies on crop production of China include the soil cultivation, fertilization and irrigation technology. Currently, the most respected conservation is the conservation tillage technology, to promote the use of minimum subsoiling or the tillage of no-till technology. That is, after the harvest of crops, keep the straw residue cover the surface, it is in order to prevent the soil from erosion, and to improve the fertility of the soil. According to the experiments of Reicosky et al, when soils is cultivating for 80 hours, it will produce three times of CO₂ than the no-till soil, and the application effect in Beijing and other places also shows that it cannot only increase the food production, but also save much energy. First, the improvement of fertilization techniques is to test on the soil and then to fertilize it, so that to avoid excessive fertilization, which would cause the excessive emissions of N₂O. Secondly, we'd better promote the use of slow-release fertilizer and slow-release fertilizer, and then, to use the nitrification inhibitors can also reduce the emissions of N₂O from the agricultural soil. In addition, the application of the biogas digesters' residue as manure can reduce 55% of the methane emissions [6-8]. Thirdly, strengthen the cultivation of new varieties so that to improve the planting techniques. The low-carbon technologies in the irrigation, some studies suggest that the field if take use of the intermittent irrigation technology will reduce the emissions of methane by 46%, so the reasonable cultivation and fertilization technology is very important to the improvement of the low-carbon energy saving.

4. ENHANCE THE PEASANTRY' AWARENESS OF LOW-CARBON ENERGY ALSO CANNOT BE OVERESTIMATED

The accounts of Chinese rural population is more than half of the total population, according to the Chinese national conditions, rural people generally have a low level of education, lack the understanding to protect and improve the living environment. What's worse, some of the leading cadres don't have the concept of the environmental protection and energy conservation, what they think more important is the economic development, and on the other hand, in their eyes, environmental protection often is not important than the economic development, so the quality of the living environment in rural areas have still stay in the same stage. So, we should use some channels or other media to broadcast the importance of the environment and to protect the environment is not only being benefited for the lives of contemporary people, but related to generation after generation of descendants' life. For example, some farmers have the consciousness of developing the biogas, but they still think that it's has a lot of trouble to clean the septic tanks, and do not get full use to, it can be instead of the fertilizer, so they turn back to use the biomass energy or coal-fired. In

addition, the education must be strengthened as well, and to allow the villagers to realize the harm of the deterioration of the living environment, as well as its dangers. The farmers should change the traditional style of life, from the rough transformation of resource-saving to low-carbon lifestyle. In rural, low-carbon energy, low-carbon jobs and low carbon farming are not only conducive to the improvement of the living environment, but also improve the food production and animal husbandry, and take better protection of arable land, so that to prevent the loss of soil carbon. A good job in low-carbon energy for rural areas can greatly promote the ecological balance, improve the quality of consumption, promote harmony consumption and be better to make a harmony community.

5. CONCLUSIONS

Green low-carbon sustainable development is the only way to develop the agriculture. To develop and promote the use of biogas and explore high efficient of energy and the methods are the newest way to develop the modern rural. The farmers not only have to plant high-yielding and healthy food, but also to protect the soil structure and the soil and water, so that the people may achieve the sustainable development of the agriculture. As a new generation of the intellectual, we must study hard, and continuously work and explore.

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THE RESEARCH AND ANALYSIS OF TECHNOLOGY IN CHINA'S URBAN BUILDING ENERGY UTILIZATION AND ITS INFORMATIZATION

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ABSTRACT

This paper introduces the development of China's building energy utilization and its informatization technology in the process of urbanization since the "11th five-year plan". There are five main technology lines in this paper: the energy utilization and its informatization technology related to civil building, industrial building and urban infrastructure, the application of renewable energy technology in urban building energy and the Internet of things technology in building energy efficiency. Further, it divided into several technical directions in each technical line, then research and analysis the development in the field of longitudinal project and national demonstration projects, For example, the National 973 program, 863 program, National Natural Science Foundation and National science and technology support plan, etc. Through technical analysis meanwhile combining the current policy orientation, this text got the latest achievements of the application and development direction of the latest technology in each of the technology line. Through the analysis and study of national longitudinal project amount and density for the "11th five-year plan", this paper finally got the key technology direction of the energy utilization and its informatization and potential commercial projects in national urbanization development strategy, and providing to policy support and technical support for China's urbanization development in the next step.

Key words: Urbanization; Urban construction; Energy utilization; Informatization; Technology.

1. INTRODUCTION

From 2000 to 2010, our country's construction area has increased from 27.7 billion square meters to 45.3 billion square meters, the total energy consumption of buildings in the nation has increased from 289 million tce to 677 million tce(ton of standard coal equivalent), building energy consumption accounts for the proportion of the total energy consumption in China has increased from 10% in 1978 to 27.6% in 2010. By the end of 2012, China's urbanization rate reached 52.6%,The requirements on urban building energy utilization become more and more important with the rapid development of urbanization. Since the "11th five-year plan", the national ministry of science and technology and the national natural science foundation have carried out a number of major projects and key projects in the field of building energy utilization and its informatization technology, it consist of the civil building, industrial building and urban infrastructure, the application of renewable energy technology in urban building energy and the Internet of things technology in building energy efficiency, meanwhile, to carry out the implementation of a series of demonstration project construction. This parer aims to study and analysis the above areas, and summarized the territory of each classification technology development.

2. The research of china's urban building energy utilization and its informatization technology during the "11th five-year plan"

2.1 The research of civil building energy utilization and its informatization technology

During the period of "11th five-year plan", China's civil building energy utilization and its informatization projects are mainly distributed in the field of national natural science foundation of China and national science and technology support plan. Among them, the research of the national natural science fund item number is 14, the project capital is 6.91 million RMB; National science and technology support plan of the item number is 57, the project funds is 369 million RMB.

The research has divided it into seven class technology direction in the survey of civil building energy utilization and its informatization, such as building ontology energy saving technology (tech-1), heating and air conditioning energy saving technology (tech-2), heat pump technology (tech-3), the performance evaluation of the public building energy

consumption (tech-4), new buildings energy saving technology (tech-5), energy efficiency reconstruction of existing buildings (tech-6) and building energy efficiency and information technology (tech-7), the following will be counted as technology 1-7.

As shown in the figure 1, the new buildings energy saving technology and energy efficiency reconstruction technology of existing buildings accounts for a bigger share of the total amount, this is due to the speeding up

of urbanization in China during the "11th five-year plan", the demand for new buildings continue to increase; At the same time, with the rapid development of domestic economy, the contradictions increasingly prominent between comfort housing needs of people with backward building using function, make the energy efficiency reconstruction technology of existing buildings becomes more and more important.

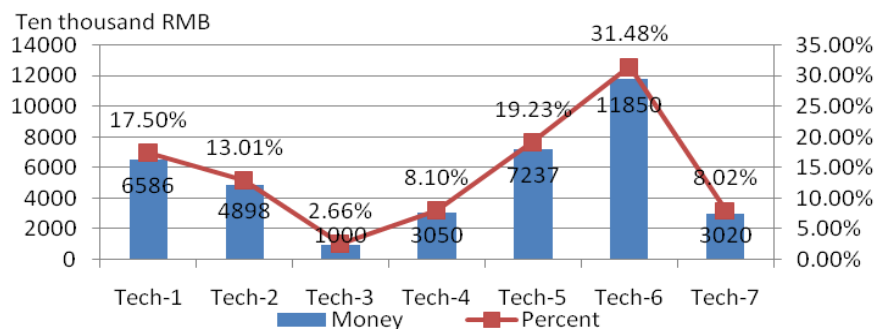


Figure 1 The category technology money in civil building energy utilization and its informatization

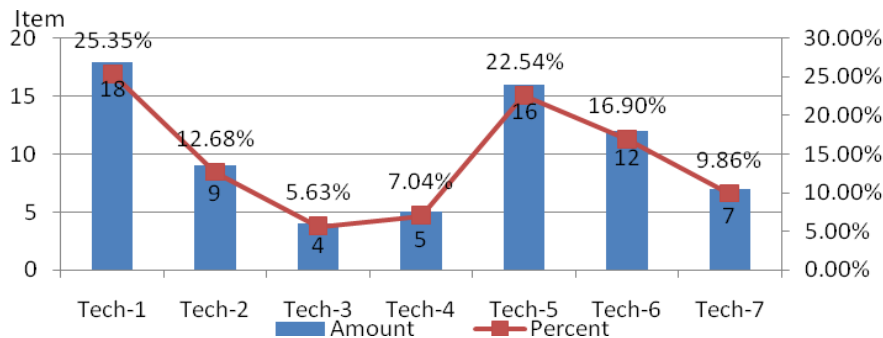


Figure 2 The category technology amount in civil building energy utilization and its informatization

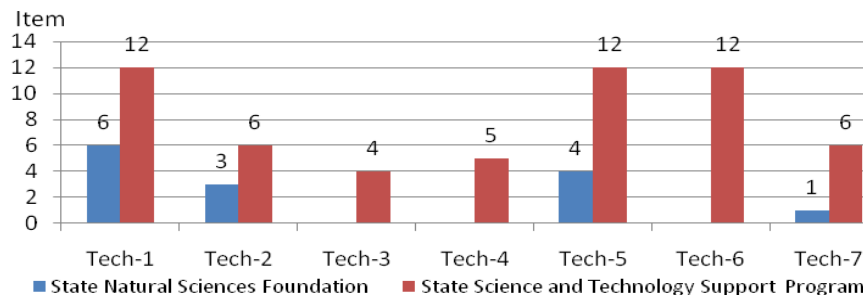


Figure 3 National science and technology support and natural science fund classification information graph during the period of "11th five-year plan"

In figure 2, The building ontology energy saving technology and new buildings energy saving technology accounts for a big share of the total number, and it is mainly due to the country attaches great importance to building heat preservation and heat insulation technology in hot-summer and cold-winter zone; In addition, with the rapid development of urbanization in our country, the demand for new buildings continue to increase, making new buildings energy saving technology project number share is higher.

We can classify the total 79 national science and technology support subjects and 10 national natural science funds into seven class technology directions as the above showed, then we can get the classification information in figure 3.

Combined with the concluding reports of national science and technology support project and national natural science foundation during the "11th five-year plan", we got seven technical representative technical directions and their typical demonstration project, as shown in table 1.

Table 1 Civil building energy utilization and its informatization technology demonstration project classification

Technology	Representative direction	Demonstration project
Tech-1	Outside wall and window heat preservation and heat insulation new material application technology	Hebei Province construction information service center
Tech-2	Household heat metering technology, air-conditioning frequency conversion control energy-saving technologies	Daqing heating power company new village
Tech-3	Water source heat pump technology in air-conditioning	Shanghai minhang District archives centre
Tech-4	Public building energy consumption data acquisition technology Green building	Guangzhou pearl river building
Tech-5		Wuxi residential area
Tech-6	External wall insulation technology	Construction science research institute building of Jiangsu Province
Tech-7	BIM technology in building	The national stadium

As shown in table 1, direction of building energy conservation technology has been changed from the traditional energy-saving buildings to intelligent building and green building, building energy conservation information technology play a more important role in modern large public buildings design, intelligent building energy efficiency design will become a building energy-saving hot technology in the future.

2.2 The research of industrial building energy utilization and its informatization technology

Currently, the main way of industrial construction energy conservation is contract energy management. From 2006 to 2010, the number of energy conservation

service company that using contract energy management mechanism to implement energy-saving projects has increased from 76 to 782; Energy saving service amount increased from 4.73 billion RMB to 83.629 billion RMB.

The industrial building energy efficiency service and their contract energy management were divided into six classes, all kinds of industrial building energy-saving projects in energy conservation service company distribution information and typical cases are showed in table 2. As shown in table 2, common methods of industrial building energy saving transformation mainly adopts computer intelligent control technology, control method has developed from simple part of the PID control to the system linkage control and fuzzy control.

Table 2 The first industrial building energy saving energy conservation service company information

Category	Corporation number	Typical case
Energy regulatory system	7	Weichai power energy monitoring system
HVAC energy saving optimization	1	Shenzhen Tianma microelectronics co., LTD
Green lighting system	1	Changzhou Starr Wright building energy-saving renovation project
Improvement of mechanical and electrical equipment and systems	26	The east of Inner Mongolia energy saving renovation project
Industrial equipment modification	8	Tianjin ironworks industry boiler energy-saving transformation projects
Industrial waste heat utilization	17	The Pearl River steel furnace flue gas waste heat power station

2.3 The research of urban infrastructure energy utilization and its informatization technology

Distribution information of urban infrastructure energy utilization and its information technology in 863 program, the national natural science fund and the national science and technology support projects showed in figure 4.

During the period of "11th five-year plan", the national support for urban infrastructure energy utilization and its informatization reached 73.4 million

RMB, covering 13 projects, the monomer project amount is about 5.65 million RMB.

The city infrastructure energy utilization and its informatization technology is divided into five types, respectively, central heating energy conservation technology (tech-A), concentration cooling technology (tech-B), heat metering technology (tech-C), pipe network heat balance technology (tech-D) and urban public lighting system (tech-E), the following will be counted as tech A – tech-E, respectively. Figure 5 is the money and number of each category technology of national projects.

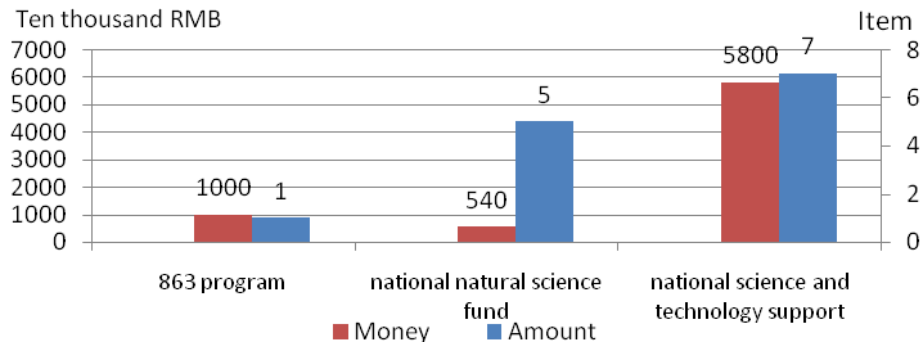


Figure 4 Distribution of urban infrastructure energy utilization and its informatization technology

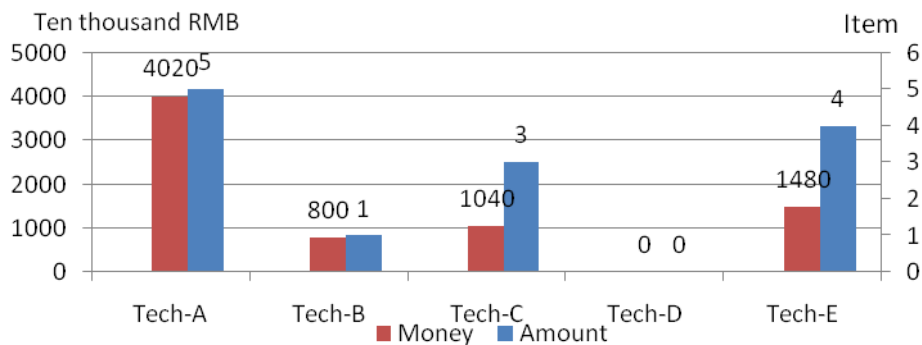


Figure 5 Urban infrastructure energy utilization and its information technology classification

As shown in figure 5, the northern town of heating energy consumption is still an important part of total energy consumption in our country, from 1996 to 2008, the building heating energy consumption of northern town increased from 72 million tce to 153 million tce, therefore, the central heating energy conservation technology is of a great responsibility.

2.4 The research of the application of renewable energy technology in urban building energy utilization

The application of renewable energy technology in urban building energy utilization is mainly composed of solar energy utilization and wind energy utilization, among them, the state items for solar energy utilization

is 55, total amount is 129.8 million RMB; the state items for wind energy is 10, total amount is 32.4 million RMB. Figure 6 is the distribution information of the application of renewable energy technology in urban building energy utilization.

At present, the amount of solar energy utilization technology accounts for 80% of the total project amount of renewable energy. Current solar utilization technology mainly includes the solar hydrogen production technology, solar cells, solar photovoltaic technology, solar heating water systems, solar thermoelectricity, solar energy air conditioning refrigeration technology and solar building integration. Wind energy technology is mainly applied to wind power grid-connected systems.

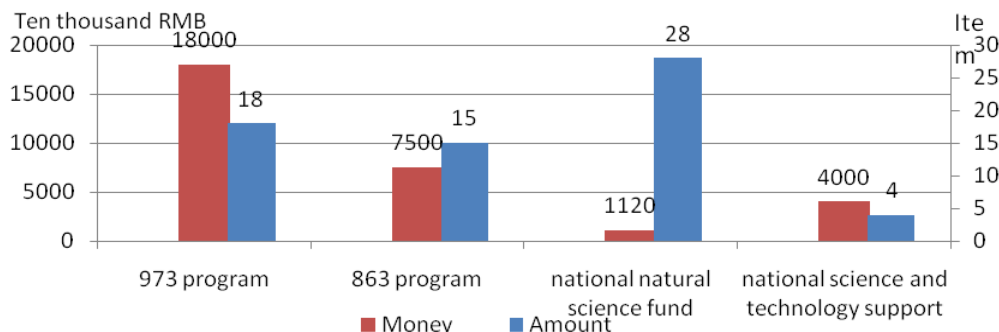


Figure 6 The distribution of the application of renewable energy technology in urban building energy utilization

2.5 The research of Internet of things technology in building energy efficiency

Jili Zhang, professor of Dalian university of technology, is proposed for the Internet of building energy system (iBES) the first time in 2010 and defines the concept of technology system, iBES refers to online awareness of all kinds of physical quantities in the building energy system, through the heterogeneous network integration, information gathering and decision making diagnosis and feedback control, the implementation of the Internet of things of building energy system monitoring, control and management. Research analysis the iBES engineering application on building energy consumption monitoring platform in A province in the northeast of China.

The number of buildings in all cities of A provincial is 50, a total construction area is 2.0082 million square meters; installation of the monitoring module (heat, water, electricity) 1901 sets; Through energy management method and energy consumption quota system in the building energy consumption monitoring platform, owner on the premise of zero cost

input, can achieve the aim that saving electricity 15% a year, saving water by 10% a year, saving heat by 12% a year.

3. The research of china's urban building energy utilization and its informatization technology during the "12th five-year plan"

Promulgated by the state council "the comprehensive work plan of energy conservation and emissions reduction during the "12th five-year plan"", the construction energy conservation quantity accounted for about 17% of the total energy saving task in the whole society during the "12th five-year plan", that is to say to save 120 million tce.

The investigation method of urban building energy utilization and its informatization technology of China during the "12th five-year plan" in accordance with the "11th five-year plan", We can get the five main line in representative technology direction during the "12th five-year plan", as shown in table 3.

Table 3 Five main lines in representative technology direction during the "12th five-year plan"

The main technology line	Representative technology direction
Civil building	Energy conservation materials, Heat metering heating, Heat pump technology, The public building energy efficiency supervision, Green buildings
Green industrial building	The identity of the green building, Intelligent control of HVAC system
Urban infrastructure	System heat metering technology
The application of renewable energy technology	Solar energy building technology
Internet of things technology	Energy consumption monitoring platform

4. Project information analysis between the "11th five-year plan" with the "12th five-year plan"

Table 4 is the project line and the project number information diagram according to 5 main technology lines in China urban building energy utilization and its

informatization technology during the "11th five-year plan" and "12th five-year plan".

Table 4 Chinese urban building energy utilization and its informatization technology classification information

Category	Money during the "11th five-year plan"/Ten thousand RMB	Money during the "12th five-year plan"/Ten thousand RMB	Amount during the "11th five-year plan"/Items	Amount during the "12th five-year plan"/Items
Civil building	34641	24139	71	93
Urban infrastructure	7340	—	13	—
The application of renewable energy technology	30620	13160	65	41

As shown in table 4, there are only 3 technology lines in the statistics, during the period of "11th five-year plan", National longitudinal project amount has implemented is about 726 million RMB; during the "12th five-year plan", so far amount of national longitudinal project which have project or are implementing is about 373 million RMB; Plus local

matching funds (1:1) and self-raised funds (1:2), since 2006, the aggregate amount is about 4.396 billion RMB; According to the measuring standard of market, Social programs amount is about 1:10 relations with the matching funds in the field of building energy consumption monitoring, according to the proportion,

since 2006, the total amount of social programs is more than 50 billion RMB.

8.The national natural science fund committee:
<http://www.nsf.gov.cn/Portal0/default152.htm>

5. CONCLUSION

This paper through survey and analysis for China urban building energy utilization and its informatization technology during the "11th five-year plan" and the "12th five-year plan", finally got 6 parts potential business projects: Energy contract management project, North central heating heat metering and energy saving control project, Public building energy consumption monitoring and energy-saving renovation integration projects, Large-scale renewable energy building integration project, The green building certification and assessment consulting projects, Building energy-saving materials production and application project. The direction of Chinese urban building energy utilization and its informatization technology in the future is mainly manifested in the following seven directions: New building envelope material research, Based on the heat measurement of central heating technology in existing buildings, Large double temperature heat pump and high temperature heat pump technology, Public building energy consumption monitoring and data analysis technology, Large public buildings energy system promotion of energy efficiency technology, Building information model (BIM) technology, Internet of things technology for building energy systems, Efficient solar energy thermal and photovoltaic technology, Based on renewable energy supply technologies of cooling, heating and electrical cogeneration technology.

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DEVELOPMENT OF A NEW MULTI-CHAMBER DOUBLE-LAYER ADSORBER IN ENERGY EFFICIENT SOLAR ADSORPTION COOLING SYSTEMS

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ABSTRACT

In recent years, global warming and energy shortage have become more and more serious due to the rapid economic development all over the world. The energy consumed on cooling system accounts for 30-40% of total energy consumption in buildings. The vapor compression (VC) system is the most common used technology for cooling generation but it consumes a large amount of electricity. The refrigerants utilized in the VC system cause strong ozone depletion and global warming. Studies of adsorption cooling systems using solid adsorbents have been ongoing since 1970s, but have not received enough attention compared to the systems using liquid absorbents. Adsorption cooling systems require a heat source and little other electricity supply. Integrating adsorption thermal systems with waste heat or solar energy can substantially reduce the dependence on fossil fuels. In this study, a new multi-chamber double-layer adsorber is designed and manufactured in order to achieve a higher cooling performance of the adsorption cooling system. The optimized performance can be achieved when seven cylindrical shape chambers are integrated into one hexagonal shape adsorber. The amount of adsorbent filled in the new adsorber is calculated, showing a 40% improvement. The total contact surface area between the adsorbent (silica gel) and the adsorbate (water vapor) is also recorded and it shows 100% improvement over the original adsorber. In addition, a prototype of the adsorption cooling system has been built at The Hong Kong University of Science and Technology. Because of manufacturing limitation, only five chambers with fin tubes were installed into the adsorber in the experiment. The coefficient of performance (COP) and the specific cooling power (SCP) are addressed. Under the experimental condition of about 35°C adsorption, 75°C desorption, 30°C condenser and 15°C evaporator temperatures, the COP of the adsorption cooling system is recorded at about 0.26 while the SCP is about 3.4W/kg. Most importantly, it is predictive that the COP and SCP will have a significant improvement after utilizing the new adsorber which installs all 7 chambers in the adsorption cooling system. In conclusion, the results demonstrate that the newly developed adsorber can be a suitable candidate to be used in the adsorption cooling systems.

Keywords: Adsorber, Adsorption Cooling System, Coefficient of performance, Energy efficiency ratio

1. INTRODUCTION

Global warming and potential shortage of fossil fuel have become more and more serious because of the rapid economic development all over the world. In Hong Kong, for example, the energy consumed on buildings accounts for about 90% of the total energy consumption, and the energy consumed on cooling system accounts for 30-40% of the consumption on buildings [1]. Under this condition, a reliable, pollution

free and low energy cost cooling system has drawn much attention. Adsorption cooling system is a type of refrigeration system driven by thermal energy, such as low grade solar energy or waste heat. Compared with a traditional vapor compression system, adsorption cooling system has the following advantages. It can reduce the carbon dioxide emission and electricity consumption. It also uses a natural working fluid such as water or ammonia which has zero ozone depletion potential instead of Freon [2, 3].

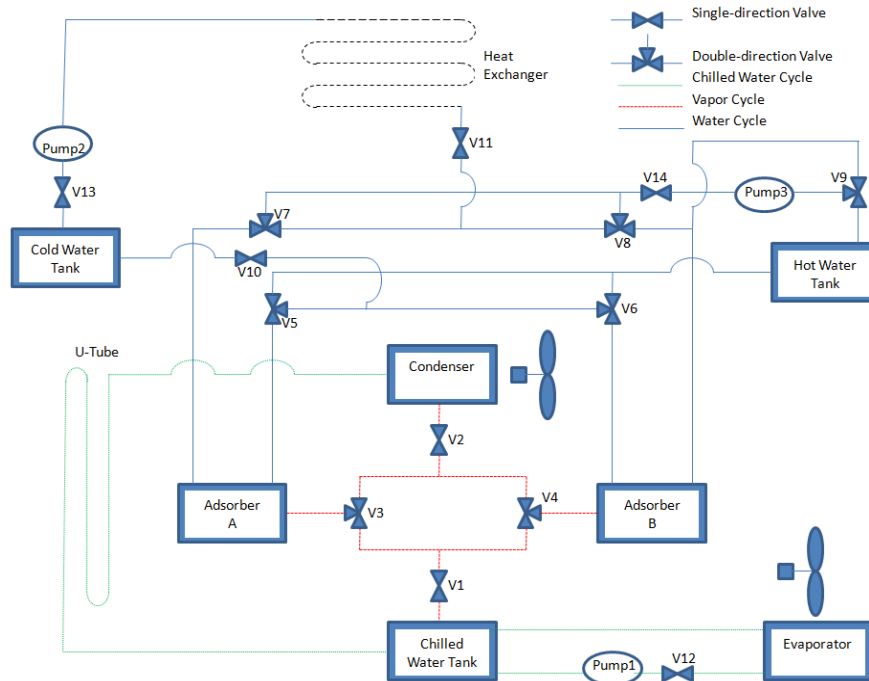


Figure 1 Schematic Design of the Laboratory Adsorption Cooling System Prototype

Figure 1 shows the schematic design of the prototype of adsorption cooling system. The basic principle is that the refrigerant (water) would evaporate rapidly in a low pressure condition. The temperature of water would decrease by continuous evaporation. The water vapor would be adsorbed by the adsorbent in the adsorber. After the adsorbent fully adsorbs, it would be heated up by hot water and start to desorb. After the desorption phase, the adsorbent could be used to adsorb water vapor again. Desorbed water vapor would condense in the condenser and flow back into the chilled water tank. Both the refrigerant and adsorbent could be regenerated.

Although the adsorption cooling system has many attractive advantages, vapor compression systems still dominate almost all application areas because adsorption cooling system has shortages which need to be improved, such as: i) long adsorption/desorption time; ii) low coefficient of performance (COP); iii) low specific cooling power (SCP). These disadvantages make the system bulky and lower the performance. Since the adsorber has a prominent influence on the COP and SCP of an adsorption cooling system, a new adsorber with multiple double-layer chambers was designed to improve the performance by increasing the amount of adsorbent which could be filled in the adsorber and the total surface area between the adsorbent and water vapor.

In the literature, there are many studies which have been performed to improve the performance of the adsorber. Li and Wang [4] used multi-adsorber to create a higher internal heat and mass transfer as well as enhanced heat transfer between the adsorber and heat transfer fluid. Maier-Laxhuber et al. [5] used coated adsorber and described an adsorber coating process for use on a metallic surface. The adsorption speed of water

vapor under 20mbar pressure increases by more than 5% within 30 minutes. In order to prevent the hydrogen gas production in the aluminum casing which would increase the heat transfer resistance on the heat exchanger surface, Nagashima et al. [6] created and patented a method to prevent it. The case in their adsorber is made of stainless steel instead of aluminum. It is used to seal the refrigerant in the chamber inside the adsorber, so that the vacuum condition could be kept in the chamber. For the other parts of heat exchangers or adsorbers, SiO₂ films are formed on the surfaces of them to prevent the chemical reaction of the aluminum with the refrigerant. Heat pipe technology is used for solving corrosion problem in ammonia-seawater systems which are usually used in boats. Xia et al. [7] invented a compound alternative heat pipe type adsorber. In this adsorber, seawater circuit is separated from the adsorbent, refrigerant and ammonia. There are also other technologies to improve the performance of the adsorber such as: adsorbent container improvement [8]; filling efficiency of a suction agent [9]. Different adsorbers of different adsorption cooling systems need different technologies. No technology is omnipotent and suitable for all kinds of systems.

In this paper, a new adsorber with new adsorbent container technology was proposed for achieving a high performance in water vapor adsorption and desorption. The adsorbent would be filled in a chamber and several chambers were integrated into the adsorber. The optimized design is a 7-chamber adsorber. The performance of the system with 5-chamber adsorber would be tested because of the manufacturing limitation. Silical gel was used as the adsorbent and water was used as the adsorbate in the testing. The amount of adsorbent which could be filled in the adsorber was recorded. The total surface area between

the adsorbent and water vapor was estimated. Finally, the values of the COP and SCP for an adsorption cooling system after installing the new adsorber were estimated as well.

2. EXPERIMENT

2.1 Design Concept

A prototype of adsorption cooling system was built at the Hong Kong University of Science and Technology. When the prototype was set up, a 4L chilled water tank was used and two stainless steel cuboid tanks were used as the adsorbers. The adsorber and the copper tube of the inner chamber are shown in Figure 2. The size of the adsorber is 400×200×250mm. The chamber inside the adsorber including two parts: copper tube and bag-shape metal mesh. The copper tube is covered by the metal mesh and around 2.5kg adsorbents could be filled in the mesh. The surface that is between the adsorbent and water vapor is around 0.226m².

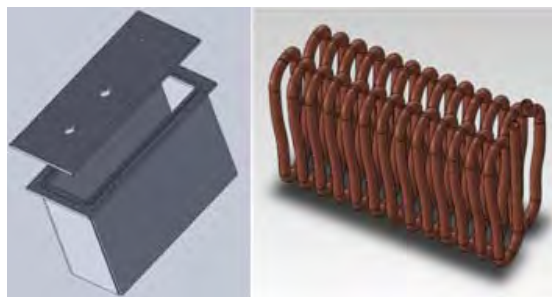


Figure 2 The Original Adsorber and Copper Tube

For experimental purpose, four pressure sensors were installed in the adsorbers, chilled water tank and condenser to monitor the pressure differences. Several temperature sensors were installed in the water tanks, chilled water tank, adsorbers and evaporator to monitor the temperature differences. In the preliminary experiment, de-ionized water was used as the adsorbate and silica gel was used as the adsorbent. Under the experimental condition of about 35°C adsorption, 75°C desorption temperatures, the system failed to produce cool effect. The pressure in the chilled water tank and the adsorber under adsorption phase increased slowly in the beginning and started to increase rapidly after 5 minutes from 6.7×10^{-2} Pa. It shows that the water vapor adsorption rate is much lower than the chilled water evaporation rate and the adsorption capacity is too low as well. There are several possible reasons. I) The amount of the adsorbent was not enough. II) The thickness of the adsorbent in the chamber was too thick so that the inside adsorbent could only adsorb water vapor slowly. III) The contact surface area between the adsorbent and water vapor was not larger enough. IV) The heat transfer between the adsorbent and working fluid is not fast enough. Developing a new adsorber with higher capacity and larger adsorption surface area may solve these problems.

2.2 New Adsorber Design

There are two adsorbers in the prototype. In the optimized design, each adsorber consists of seven cylindrical shells (chambers). Each chamber has two layers and a helix copper tube. The adsorber design is shown in Figure 3.

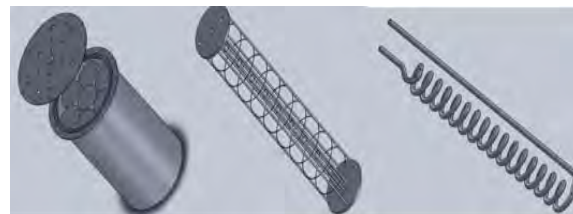


Figure 3 Optimized Design of Adsorber, Chamber and Helix Copper Tube

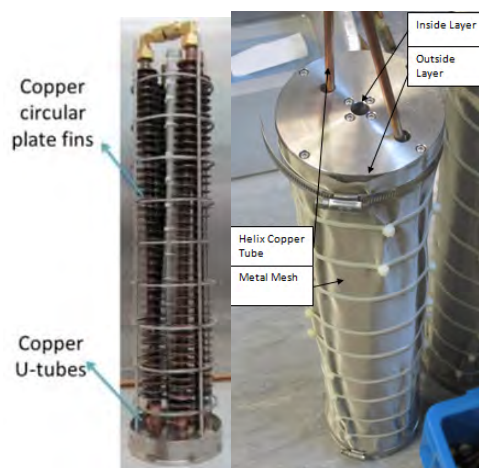


Figure 4 Chambers in the Adsorber

The helix copper tube is used to increase the heat transfer characteristics. Because of the manufacturing limitation, U-shape copper tube with fins was used to replace the helix copper tube. The outer layer represents a large cylindrical shell and inner layer represents a small one. The two layers are covered with metal meshes with a pore size of about 74 microns, which is shown in Figure 4.

The adsorbent is filled in the space between the two layers. Each chamber could be filled in about 3.5kg adsorbent. Both two layers are used for passing the water vapor through so that the water vapor could be adsorbed by the adsorbent. The seven chambers are integrated into one hexagonal shape adsorber. Totally, about 24.5kg adsorbent could be filled in an adsorber.

In Figure 5, it shows the adsorber which was used in the experiment to collect data. Only five chambers were installed into the adsorber and U-shape copper tube with fins was used to replace the helix copper tube to simplify the manufacturing procedure. The five chambers were connected in parallel. The adsorbent is filled in the space between the two layers. Totally, about 17.5kg adsorbent could be filled in an adsorber.

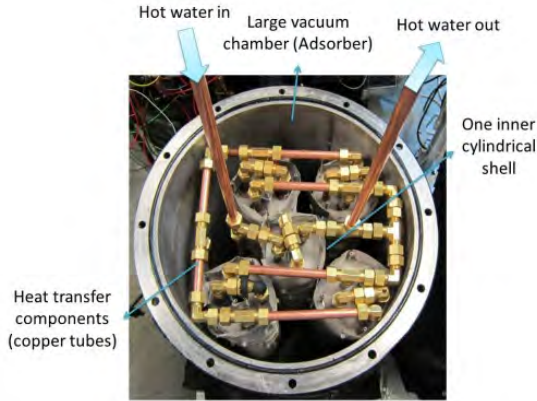


Figure 5 The Adsorber Used in Experiment



Figure 6 The Adsorbers

Figure 6 shows the two adsorbers. The chambers which are filled in with adsorbents are installed into the adsorbers. The two adsorbers would be covered with heat insulating cotton before they were installed into the prototype.

2.3. Test Unit

The prototype is shown in the Figure 7. The new adsorbers were installed into the prototype of adsorption cooling system which was built at the Hong Kong University of Science and Technology. Silica gel was used as the adsorbent and water was used as the adsorbate in the testing. Thermocouples were installed in the system to monitor and record the temperature of the adsorber and evaporator. Pressure sensors were also installed to monitor and record the pressure of the adsorber and chilled water tank. The hot water used for desorption was heated up by an electrical heater and controlled at about 75°C; the cold water used for adsorption was cooled down by a small heat exchanger and cooled to about 35°C. The temperatures of water in the condenser and evaporator were recorded as well. There is 4L chilled water in the chilled water tank and 10L hot water in the hot water tank.

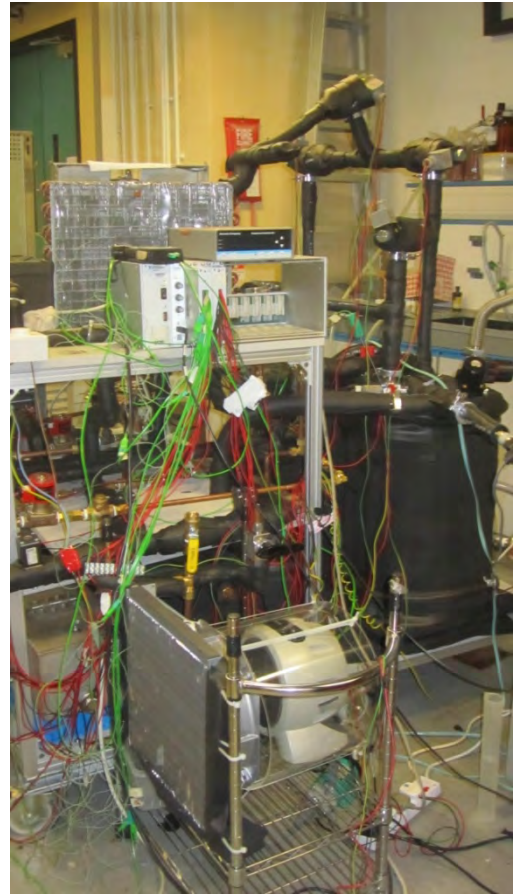


Figure 7 Prototype of Adsorption Cooling System

3. RESULT AND DISCUSSION

3.1 Theory

The coefficient of performance (COP) was estimated with the following equation:

$$COP = \frac{Q_{chill}}{Q_{hot}} \quad (1)$$

where Q_{chill} represents the cooling output power and Q_{hot} represents the thermal input power. They could be calculated from the Equation (2) and (3):

$$Q_{chill} = c_{water} \times \dot{m}_{chill} \times (T_{chill,in} - T_{chill,out}) \quad (2)$$

where c_{water} represents the heat capacity of water, $\dot{m}_{chill}$ represents the mass flow rate of chilled water passing through the evaporator which is 2L/min, $T_{chill,in}$ and $T_{chill,out}$ represent the temperature before and after the chilled water passing through the evaporator; and

$$Q_{hot} = c_{water} \times \dot{m}_{hot} \times (T_{hot,in} - T_{hot,out}) \quad (3)$$

where c_{water} represents the heat capacity of water, $\dot{m}_{hot}$ represents the mass flow rate of hot water passing through the adsorber which is 14L/min, $T_{hot,in}$ and $T_{hot,out}$ represent the temperature before and after the hot water passing through the adsorber.

The specific cooling power (SCP) was estimated with the following equation, Equation (4):

$$SCP = \frac{Q_{chill}}{W_{sg}} \quad (4)$$

where W_{sg} represents the weight of silica gel which was used as the adsorbent in the prototype.

3.2 Results and Discussion

Figure 8 shows the temperature profile of the heat transfer fluid of the adsorption cooling system prototype during the experiment. The cycle time of the experiment is about 45mins.

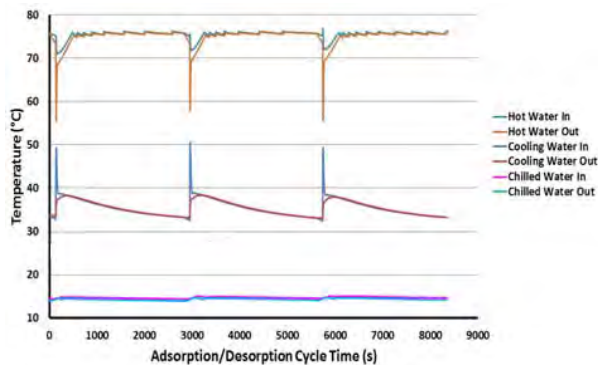


Figure 8 Temperature Profile of Heat Transfer Fluid

The water in the hot water tank was heated up by an electrical heater and controlled at about 75°C. The temperature decreased a little bit after the hot water which was used for desorption flowed back into the hot water tank. The temperature of the water in the cold water tank kept decreasing because of the heat exchanger, while it increased a little bit after it was used to cold down the adsorbent and flowed back into the cold water tank. The temperature of delivered outlet chilled water is shown as the chilled water outlet which is always lower than the temperature of the inlet chilled water which is shown as the chilled water inlet in the figure. The temperature of the chilled water is about 13°C. The outlet temperature of the chilled water is about 0.45°C lower than the inlet temperature. It shows that the prototype has a steady cooling effect.

The COP and SCP were estimated according to the data and equations. The COP is 0.26, SCP is about 3.4W/kg and the cooling power is about 60.1W. Compared to the old adsorbers which failed to produce cooling effect, the performance of the new adsorber is much better. It is predictive that the COP and SCP will have a significant improvement after utilizing the new adsorber which installs all 7 chambers in the adsorption cooling system instead of 5 chambers in the current prototype. The improvement of performance was caused by the following reasons. I) The capacity of adsorber increases by 40%. Totally about 24.5kg adsorbent can be filled in one adsorber which has a volume of around 0.1m³. II) The total contact surface area between the adsorbent and the adsorbate is about 1.45m² and it shows 100% improvement over the original adsorber.

Although the performance has a great improvement, the cooling power of the prototype is still unsatisfactory, which is less than 100W. The main

reasons are listed in the followings. I) The heat transfer area between working fluid (hot water/cold water) and the adsorbent is still not large enough and the adsorbent (silica gel) has a low thermal conductivity, so that it takes a long time to heat up/cold down the adsorbent. II) The adsorption capacity of the adsorbent (silica gel) is low. Only small amount of water vapor could be adsorbed even though the adsorbent was cold down to a low temperature. III) The mass transfer rate is slow. During the adsorption phase, only little amount of water vapor could flow into the adsorber and be adsorbed. It leads to a small temperature difference between the chilled water inlet and outlet.

4. CONCLUSION

In this study, a new multi-chamber double-layer adsorber was developed in order to achieve a higher cooling performance of the adsorption cooling system. A double-adsorber silica gel-water prototype of adsorption cooling system was used to test the performance of the new adsorber. Under the experimental condition, a cooling power of 60.1W, a COP of 0.26 and a SCP of 3.4W/kg could be achieved. There is a significant improvement on the total capacity of the adsorber and the contact surface area between the adsorbate and adsorbent. Although the performance of the prototype improves, there are still a lot of research work required for increasing the heat and mass transfer ability. Firstly, the pipes connecting the chilled water tank and adsorbers should be enlarged so that the mass transfer of water vapor from the chilled water tank to the adsorbers could be increased. Secondly, although some fins were added to the copper tube, the ability of heat transfer between the working fluid and adsorbent should be improved. Lastly, silica gel and water pair could be replaced by an adsorbent and an adsorbate pair with higher adsorption capacity and heat transfer ability.

ACKNOWLEDGMENT

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AN APPLICATION OF WIND POWER GENERATION ENERGY SYSTEM FOR COASTAL BUILDING IN XIAMEN

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ABSTRACT

Wind power utilization can play an important role in increasing energy supply, improve the energy structure, energy security, reduce greenhouse gases, to protect the environment and the construction of a harmonious society, etc. Take reasonable way, wind power and other new energy and renewable energy together is an important way to develop a low carbon economy. Wind power will be utilization to contribute the implementation of scientific, green, low carbon energy strategy. Special regional characteristics of Xiamen have provided Xiamen a good and rich wind energy resources, water resources and solar resources; Therefore, research and utilization of these resources for the benefit of mankind, the use of 'renewable energy' to realize building energy efficiency, the use of 'natural cold' can realize air conditioning energy conservation in summer, is the primary task of the building energy efficiency in the area for xiamen. Xiamen' wind environment resources and solar resources features are introduced in this paper, distributed wind-light combined energy system in coastal construction can be used to combined with the actual of xiamen energy resources. The application is discussed for building ventilation energy saving technology and renewable energy utilization way.

Keywords: Wind power generation; Solar energy; Energy system; Building ventilation.

1. INTRODUCTION

The proportion of taking the total energy consumption of energy consumption of the building has become higher and higher. The energy consumption of the building includes those such as air conditioner, heating, light, cook consumption, hot water of daily life, etc., and air conditioner and heating consumption is main part[1]. The investigation indicates that the possession rate of home air conditioner in the south of China is increasing at the speed of 20% for every year. In China, the purpose of the energy-conservation building is that the energy consumed by air conditioner and heating system for comfortable thermal environment in the room can be reduced.

Special regional characteristics of Xiamen have provided Xiamen a good and rich wind energy resources, water resources and solar resources; Therefore, research and utilization of these resources for the benefit of mankind, the use of 'renewable energy' to realize building energy efficiency, the use of 'natural cold' can realize air conditioning energy conservation in summer, is the primary task of the building energy efficiency in the area for Xiamen. Distributed wind-light combined energy system in coastal construction can be used to combined with the actual of Xiamen energy resources. The application is discussed for building ventilation energy saving technology and renewable energy utilization way.

2. CHARACTERISTIC OF WIND ENERGY RESOURCES

Xiamen is located in the southeast of our country north latitude 24°27', longitude 118°04', 63.2m above sea level. The climate here has the typical climate characteristic in the southern coastal city, such as damp, warm summer, larger speed of the outdoor wind, the summer outdoor average wind speed up to 2.5 m/s, and the southeast wind is in the majority.

Xiamen is located in the southeast coast and is one of the regions abundant of wind resource. According to the opinion on China's wind power division suggested by the experts working in weather bureau, the southeast coastal and its islands area China's largest wind energy resources region. The area of effective wind energy density is greater than or equal to 200w/m² contour parallel the shorelines and the wind energy density of the coastal islands is more than 300w/m². The timing of the appearance of effective wind is up to 80%-90%.The time of the wind, speed of which is larger than or equal to 6m/s, throughout the year is about 7000h-8000h. Time of the existence of wind ,the speed of which is greater than or equal to 8m/s, is about 4000h. The coastal wind is strong in the winter half year and the wind power is relatively weaker in summer half year with typhoon occurs from time to time.

Inland from the coastal areas due to the extension and hills constantly,it is difficult for the strong cold air

in winter to enter the south directly. In the half year of summer when typhoon occurs, the winds decay very quickly. From the coast 50 km the wind speed will be reduced by 68%. This forms bigger difference of the wind in Xiamen city and Xiamen coastal.

3. CHARACTERISTICS SOLAR ENERGY RESOURCES

Lots of power, about 50×10^{18} kJ, is radiated from the sun to the surface of the earth every year[2]. It is equivalent to 13000 times of the annual energy consumption of the world. With great potential, the use of solar energy will not affect the earth's heat balance, will not cause environmental pollution. So the solar energy is inexhaustible, inexhaustible green energy. With vast territory, there are great solar energy resources can be used. The sunshine time in most areas of China is more than 2000h one year, some places is even up to 2800h-3300h, belonging to the region where the solar energy utilization is advantageous. Xiamen has strong solar radiation in summer, and its annual sunshine time is 2233.5 hours, 51% of the total hours for years.

4. APPLICATION TECHNOLOGY OF WIND ENERGY

4.1. Principle of wind power generation

Transform wind energy into electrical energy is one of the most basic way in wind energy utilization. The working principle of wind dynamo is that the wind turbines rotates under the affection of the wind power. The kinetic energy of the wind is transformed into mechanical energy of the shaft of wind turbine. Electricity will be generated by the rotational generator driven by the wind shaft.

4.2. Distributed wind-light power system

There are vast of wind power on the surface of the earth. In theory, 1% of the wind power can meets all energy demand of human. Although there is abundant of wind resources on the surface of the earth, but the its time and space distribution is uneven. The wind resource available area accounts for more than half of the total land area and mainly distributed in coastal and three line areas in China. Regional wind resources and the Local wind environment is the prerequisite of building wind power utilization. Building wind power generation system can work effectively on condition that both preconditions are satisfied.

Wind and solar resources have the characteristics of randomness that cause their limitations during the actual use. The stability of wind and solar energy resource utilization system could improved if they are combined together.

In winter of Xiamen coastal the wind is strong and the solar energy is weak, but summer is in contrast. In this way, wind and solar power just complementary on

the season. Therefore, 'distributed wind - light united power system', combining wind and solar energy together, is better than single wind or solar power supply system in performance. The continuity, reliability and economy of the power will be greatly increased. Distributed wind-light united power system can have different forms, as shown in figure 1 is one of them[3].

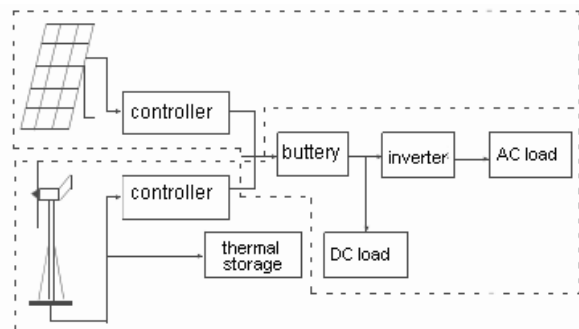


Figure 1 wind heating power supply, solar power supply system diagram

The power system is mainly composed of wind turbines, solar photovoltaic cells, control circuit, storage battery, inverter, regenerative energy storage device.

4.3. Application of wind-light united power system in littoral buildings

Based on the status of wind energy resource and the environmental conditions of Xiamen, the wind - light united power system is considered to be applied to the littoral buildings. The application method is shown in figure 2.

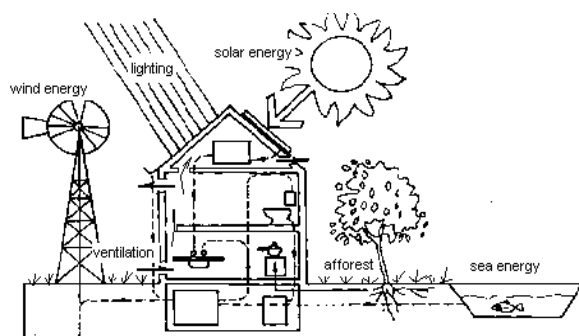


Figure 2 wind - light united power system applied in the construction

The energy produced by permanent magnet ac motor and solar panels. Most of the Power from wind turbines is directly supplied to the heat energy storage system through reasonable distribution. Electrical energy from the wind turbine is stored in the form of heat energy, and spare parts is stored in the battery. Electricity produced by solar panel is stored in the storage battery directly and through the inverter it can

be transformed into living power available to families.

One of the main climatic characteristics in Xiamen is that in summer the southeast and southwest wind is prevailing from April to September. The ground wind speed at 3 m/s or faster, the sea wind is bigger, more in 4-6 m/s. Therefore, wind power resources can be made full use of and effective ventilation technology should be adopted for coastal construction to cool in summer, improve the micro climate of the residential area and ameliorate the thermal environment inside the building.

Such increase the body's thermal comfort and the freshness of air, to provide more possibility and time for the occupants to live in 'natural' indoor air environment, embody people-oriented, healthy building design. The ventilation opening time will also be saved on condition of good and effective ventilation, achieving energy saving purpose.

The ventilation technology mainly include natural ventilation, mechanical ventilation, intermittent natural ventilation and mixing ventilation, etc. several forms. They can be flexibly used and adjusted by the occupants according to specific situation.

Due to the excellent outdoor air circulation and the good air quality, the intermittent natural ventilation is adopted as the basic measures that improve thermal environment of Xiamen city in the building[4] (shown in figure 2 and figure 3).

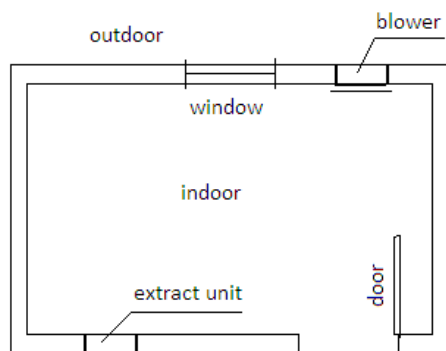


Figure 3 room ventilation

As shown in figure 3 ventilation device is mainly consisted of air supply and exhaust air device. The supply device is made up of air duct (making different frequencies of noise attenuation rapidly), air filter, blower, phase change energy storage body, absorption material, remote control receiver, the wind shutter and can bring up the wind shutter, etc. The device power is mainly supplied by low noise centrifugal fan with little energy consumption, whose power is supplied by the 'wind-light united power system'.

When summer indoor thermal environment can't meet the requirements, air conditioning technology must be used to improve the thermal environment of the building. Heat recovery fresh air ventilator is a kind of effective energy-saving equipment in the air conditioning system.

Not only can it ensure enough indoor fresh air and strengthen the fresh air and exhaust air filter, but also be equipped with heat recovery device, improving the

utilization rate of energy and resource utilization, reducing energy consumption. It has the advantage of two-way ventilation, clean and efficient energy saving, easy, safe and reliable application, etc. The equipment does not need cold source and heat source, reducing equipment investment and usable floor area, saving energy consumption more than 70% of the fresh air. Meanwhile the equipment has strong ability to purify the indoor air and improve the quality of indoor air. Also the power is supplied by the 'wind-light united power system'.

5. CONCLUSION

From the preliminary analysis above it is acknowledged that the wind energy resources around Xiamen coastal buildings is rich with the wind usually up to 4-6 m/s and also have abundant solar resources. So the wind-light united power system is expected to be used in the area for the building energy conservation.

ACKNOWLEDGEMENTS

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ECONOMIC ANALYSIS OF SERIES HYBRID GROUND SOURCE HEAT PUMP SYSTEMS IN A COMMERCIAL BUILDING

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ABSTRACT

For cooling load-dominated buildings, if the peak load is supplied only by ground heat exchanger, the initial investment of the ground source heat pump system is higher and the system coefficient will get lower and lower in contrast to hybrid systems, whose cooling load is partly fulfilled by cooling towers. Energy consumption simulation is done for a seven-floor commercial centre in Zhuzhou city, Hunan Province, PR China. Then the initial cost, operating cost, total cost and the life cycle cost of the series hybrid system, in which the cooling tower and ground exchanger undertake different ratio of cooling load, are calculated. The result analysis shows that the improper configuration cases will result in diseconomy of hybrid ground source heat pump systems.

Keywords: ground source heat pump (GSHP), cooling tower, ground heat exchanger, economic analysis

1. INTRODUCTION

The first ground source heat pump (GSHP) system could trace back to 1946, with its successful application in a commercial building in Portland, Oregon, USA (Beier, Smith et al. 2011). The emergence of hybrid GSHP system with auxiliary cooling or heating equipment has undoubtedly promoted the wide application of GSHP systems since 1990s for its lower first cost and alleviating heat accumulation around ground loops [2].

In the hot summer and cold winter region of central China, building cooling load generally is much bigger than heating load in winter, especially in commercial buildings. Therefore, hybrid GSHP system with cooling tower is one of the best choices. Comparing to traditional GSHP system, the initial cost of ground heat exchanger in hybrid system is less but it increases the expense of cooling towers. Furthermore, the operation cost of hybrid systems with bigger cooling towers and shorter buried pipes will reduce the energy consumption of heat pumps due to the auxiliary cooling of cooling towers in summer but elevate the running cost because of the decrease of buried pipe length. Consequently, detailed economic analysis is needed to confirm an optimized cooling load ratio undertaken by cooling towers [3].

In this paper, the cooling and heating load of a seven-floor commercial centre in Zhuzhou is calculated and hybrid GSHP systems serving this building are designed. Then the initial cost, operating cost, total cost

and the life cycle cost of the series hybrid system, in which the cooling tower and ground exchanger undertake different ratio of cooling load, are calculated.

2. COOLING AND HEATING LOAD OF A COMMERCIAL BUILDING

2.1. Description of the building

A seven-floor commercial centre in Zhuzhou, Hunan, China, whose gross area is 9200 m², business hours 8 a.m.—10 p.m. is chosen as the object (see Figure 1).

Major building envelopes material and their thermodynamic performance properties are: (1) 300 mm thick aerated concrete external wall, 600 kg/m³ density aerated concrete block, total heat transfer coefficient 1.493 W / (m² · °C); (2) 100 mm thick 30 kg/m³ density polystyrene board insulated reinforced concrete roof panel, total heat transfer coefficient 0.926 W / (m² · °C); (3) low-emissivity double panes heat resistance aluminum alloy hollow window, total heat transfer coefficient 6.26 W / (m² · °C).

NOMENCLATURE

Abbreviation		Subscripts	
GSHP	Ground Source Heat Pump	-c	cooling mode
COP	Coefficient Of Performance	-h	heating mode
LCC	Life Cycle Cost	-bp	buried pipe
Symbols			
Q	cooling or heating load or capacity (kW)	i	discount rate /%
W	power (kW)	S	salvage ratio /%
C	cost or investment /RMB yuan	n	equipment service life /a

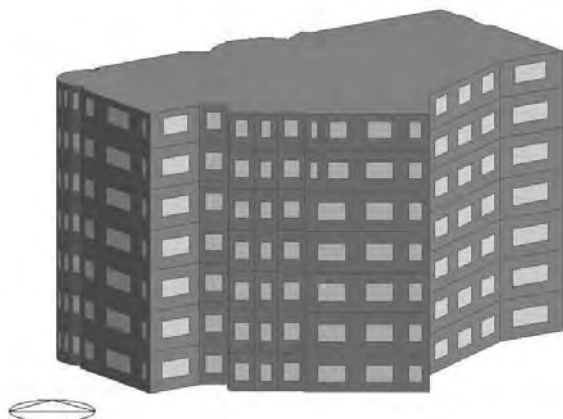


Figure 1 Perspective model of a commercial building

2.2. Cooling and heating load

After the buildup of the typical meteorological year, hourly cooling and heating load of the building is calculated in Energyplus© (see Figure 2).

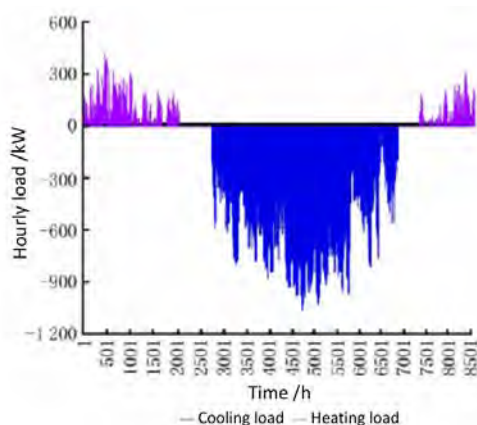


Figure 2 Hourly heating and cooling load

Thus the building design cooling load is 1,062 kW, heating load 420 kW. Annual cooling requirement is 1,167,891 kWh, heating requirement 114,639kWh. Obviously, cooling load is much bigger than heating load, so additional cooling equipments are crucial to achieve annual heat balance in the ground heat exchanger.

3. HYBRID GHSP SYSTEM DESIGN

Referring to the building cooling and heating requirements, heat pumps, whose total nominal cooling capacity Q_c and input power W_c are 1062.5 kW and 230.1 kW, and total nominal heating capacity Q_h and input power W_h are 947.7 kW and 231.2 kW respectively.

Subsequently, buried pipe length, inlet and outlet water temperature of heat pumps, et al. are simulated in a GSHP system design software, GLD (see Figure 3) based on the input of cooling and heating requirements, heat pump performance parameters, and typical soil types and thermodynamic parameters.



Figure 3 Operation window of GLD software

In the simulation software, cooling load taken on by buried pipes Q_{cbp} is calculated using the following equation

$$Q_{cbp} = \frac{Q_c(1 + COP)}{COP},$$

while heating load undertaken by buried pipes Q_{hbp} is obtained by the following equation

$$Q_{hbp} = \frac{Q_h(COP - 1)}{COP}.$$

From the series hybrid GSHP system simulation results,

1. When there is no auxiliary cooling equipment, buried pipe length sums 27,879 m. During the cooling period, COP of heat pump averages 4.1, total electricity consumption is 0.318 GWh, and heat dissipation by buried pipes sums 1.485 GWh. Corresponding average COP of heat pump is 4.3, total electricity consumption

is 27 MWh, and heat drawn by buried pipes sums 88 MWh during the heating season.

2. When 10% cooling load is fulfilled by cooling towers, total buried pipe length decreased to 25,160 m. Cooling season average heat pump COP increases to 4.4, and heat dissipation by buried pipes sums 1.450

GWh. Heating period average heat pump COP decreases to 4.1, and heat drawn by buried pipes sums 86 MWh.

Data under other ratio of cooling load is listed in Table 1 and Table 2.

Table 1 Initial investment of the series hybrid GSHP system

Cooling load ratio taken on by cooling towers	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%
Buried pipe length /m	27879	25160	22424	19670	16898	14108	11300	8475	6230	4867
Cost of buried pipe /thousand Yuan	2230	2013	1794	1574	1352	1129	904	700	486	300
Cost of additional cooling equipments /thousand Yuan	0	107	214	321	428	535	641	748	855	962
Total initial investment /thousand Yuan	2230	2120	2008	1894	1780	1663	1545	1448	1341	1262
Initial investment per floor area /Yuan/m/m	242.2	230.2	218.0	205.7	193.2	180.6	167.8	157.2	145.5	135.8

Table 2 Running cost of the series hybrid GSHP system

Cooling load ratio taken on by cooling towers	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%
COP during the cooling season	4.1	4.4	4.88	5.12	5.27	5.51	5.66	5.82	6.05	6.28
Heat pump energy consumption during the cooling season /MWh	317.5	289.9	273.3	259.5	247.0	236.2	230.0	223.7	215.2	207.3
HP running cost during the cooling season /thousand Yuan	254.0	231.9	218.6	207.6	197.6	188.9	184.0	178.9	172.1	165.8
COP during the heating season	4.3	4.17	3.92	3.7	3.46	3.25	3.0	2.74	2.52	2.33
Heat pump energy consumption during the heating season /MWh	26.9	27.8	29.5	31.3	33.5	35.6	38.6	42.2	45.9	49.7
HP running cost during the heating season /thousand Yuan	21.5	22.2	23.6	25.0	26.8	28.5	30.9	33.8	36.7	39.7
Auxiliary cooling equipments electricity consumption /MWh	0	16.7	33.3	50.0	66.7	83.3	99.8	116.7	133.4	149.9
Auxiliary cooling equipments running cost /thousand Yuan	0	1.33	2.67	4.0	5.33	6.68	7.99	9.32	10.67	12.0
System energy consumption /MWh	344.4	334.3	336.2	341.0	347.2	355.2	368.5	382.5	394.4	406.9
Annual system running cost /thousand Yuan	275.5	267.4	268.9	273.0	277.7	284.1	294.8	306.0	315.5	325.6
Running cost per floor area /Yuan/m/m	30.0	29.0	29.2	29.6	30.1	30.9	32.0	33.2	34.3	35.4

4. ECONOMIC ANALYSIS

4.1. Initial investment and annual running cost[4]

Assuming the price of buried pipe and its fitting is 80 Yuan/m, and electricity price is 0.8 Yuan/kWh, initial investments and running costs of the series hybrid GSHP system are figured out in Table 1 and Table 2 respectively.

From Table 1, total investment decreases with the increase of cooling load ratio fulfilled by cooling towers because the fall range of buried pipe cost is bigger than

the increase range of investment in auxiliary cooling equipments.

From Table 2, COP of heat pump (HP) during the cooling period elevates and energy consumption of HP decreases with the increase of cooling capacities of auxiliary cooling equipments and their electricity use. However, COP of HP decreases during the heating season due to shorter buried pipes, so running cost in winter increases. In sum, annual system running cost AC_r rise along with the increase of cooling load ratio fulfilled by cooling towers. The AC_r of hybrid system is less than that of traditional system without additional cooling devices when auxiliary cooling load ratio is smaller than 40%.

4.2. Static investment analysis

Assuming the life span of the major equipments in the hybrid GSHP system is 20 years, total static costs of the system are calculated by the following equation

$$C_t = C_i + nAC_r,$$

where C_t is total static cost of the system, C_i is the initial cost, n is life span of major devices (20 years), AC_r is the annual running cost.

Figure 4 illustrates the tendency of the change of total static cost with the service time of the hybrid GSHP systems with different additional cooling capacities.

Because of the reduction of initial investment with the increase of additional cooling capacity, hybrid systems with more auxiliary cooling capacity are inclined to reduce more system static cost C_t in the beginning seven years. But C_t booms with service time for the increase of running cost with the decrease of buried pipe length. Eventually, the 80% heating load taken on by cooling towers case has the biggest total static cost. 60% and 80% additional cooling capacity systems surpass traditional GSHP system with no auxiliary cooling in total static cost in the service time of 16th year and 12th year respectively.

In conclusion, system running cost is the crucial part of system total cost. Hybrid systems are not cheap if the increase of running cost exceed their save in initial investment.

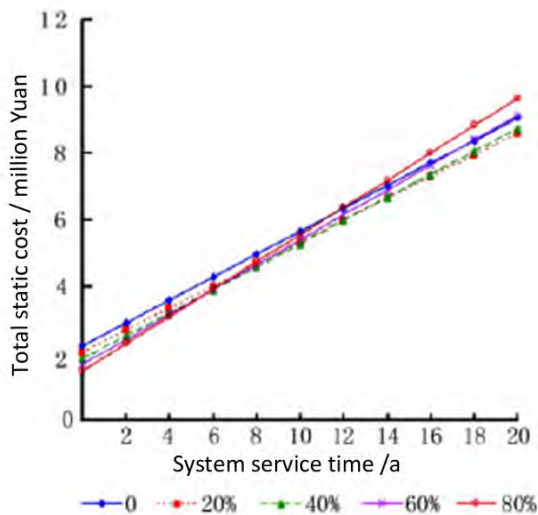


Figure 4 Tendency of total static cost with system service time

4.3. Dynamic investment analysis[5]

Annual dynamic cost and life cycle cost (LCC) are popular since 1990s for their integrity and rationality in account of the change of capital value and price with the time. Annual dynamic cost AC_t consist of initial investment C_i , annual running cost AC_r , annual management cost AC_m ($5\%C_i$), discount rate i (10%)

and salvage ratio S ($4\%C_i$), and it can be worked out as follows

$$AC_t = AC_r + AC_m + \frac{i}{1-(1+i)^{-n}}(C_i - S).$$

LCC further corrects running cost and salvage ratio

$$LCC = C_i + AC_r \frac{(1+i)^n - 1}{i(1+i)^n} + S(1+i)^{-n}.$$

Figure 5 shows the change of LCC, annual dynamic cost and its constituents in different ratio of cooling load fulfilled by cooling towers.

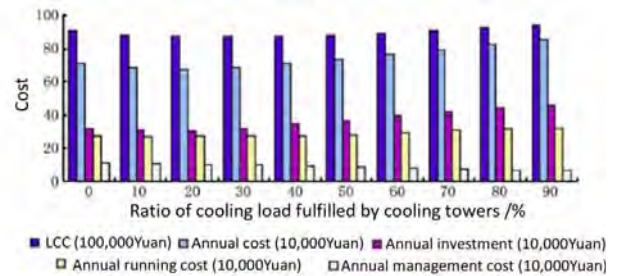


Figure 5 Costs in different additional cooling load ratio cases

Generally speaking, annual dynamic cost and LCC increase with the augmentation of auxiliary cooling capacity. However, the 20% additional cooling load ratio case, whose annual cost and LCC cut down 5.2% and 4.6% compared to no additional cooling case, is the most economic case in Figure 5. 90% ratio of cooling load case, whose annual cost and LCC rise 14.1% and 8.9% referring to no auxiliary cooling case, undoubtedly is the most expensive case. Subsequently, optimization is necessary in the design of auxiliary cooling devices.

5. CONCLUSIONS

Taking a commercial building in Zhuzhou city as the object, its annual cooling load is much more than heating load, the adoption of hybrid GSHP system with auxiliary cooling equipment is inevitable nevertheless. Detailed static and dynamic economic analysis is carried out among different ratio of additional cooling GSHP system cases.

As far as the static investment analysis is concerned, initial investment reduce with the increase of cooling load ratio fulfilled by cooling towers. Meanwhile, annual running cost booms, and system running cost of more than 40% auxiliary cooling case surpass no additional cooling case. In its 20 year life span, total static cost is less in more additional cooling cases during the first 7 years; but total static cost elevates so swiftly in latter years that hybrid GSHP systems with less auxiliary cooling capacities are more economic compared to more additional cooling cases.

In the dynamic economic analysis, hybrid GSHP systems with less auxiliary cooling capacities are also cheaper. 90% ratio of cooling load case, whose annual cost and LCC rise 14.1% and 8.9% referring to no

auxiliary cooling case, is the most expensive case, while 20% ratio of cooling load case is most economic case.

Caution: owing to the many factors affecting hybrid GSHP system's economic analysis, the results are not easy to simply extend to other buildings in optional cities.

ACKNOWLEDGEMENTS

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PERFORMANCE COMPARISONS OF A HEAT PUMP SYSTEM FOR ENERGY RECOVERY USING AQUEOUS SOLUTIONS OF LITHIUM AND CALCIUM CHLORIDES

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ABSTRACT

Heating capacity of an air-source heat pump (ASHP) system often decreases due to frost on the air-side heat exchanger (evaporator) when the air temperature drops in winter. To improve the coefficient of performance (COP) of the ASHP system, defrosting mode has to be operated periodically. This will not only increase the energy consumption but also makes users uncomfortable for the fluctuation of temperature. This paper proposes a ASHP system to recover energy from air using liquid desiccant in winter. The thermodynamic calculation model was established to analyze the effects of ambient air temperature and relative humidity, inlet solution temperature in solution tower, mass fraction of solution in solution tower, solution-to-refrigerant mass flow rate ratio in evaporator, air-to-solution mass flow rate ratio in solution tower, air-to-solution mass flow rate ratio in regenerator, total heat transfer efficiency of solution tower and total heat transfer efficiency of the regenerator on the performance of ASHP system when the desiccant solutions of calcium chloride and lithium chloride are adopted, respectively. Results indicate that this system can operate much more efficiently than the conventional air-source heat pump in winter. In addition, the new system does not need to run in defrosting mode periodically, which improves the indoor thermal comfort. More, the comparisons results can help choose suitable work fluid according to both the working conditions and economic problem, for there is a higher COP when lithium chloride is used as work fluid, but it is more expensive. On the contrary, calcium chloride is cheaper than lithium chloride though the COP is lower when it is used as work fluid.

Keywords: heat pump, energy recovery, solution

1. INTRODUCTION

Air source heat pump (ASHP) system can be used to provide cooling and heating for building heating, ventilating and air conditioning (HVAC) systems [1]. But heating requirements will represent a challenge for air-source heat pumps when the ambient air temperature is subzero in winter. During the heating season, the ASHPs extract heat from the outside cold air and release the heat inside the living space. Under certain weather conditions, frost will form on the outdoor heat exchanger surface that is the main problem with ASHP system, which will degrade the ASHP's performance.

Researchers introduced solutions into the traditional heat pump air-conditioning system to improve its performance [2-7]. To improve the performance of ASHP evidently, energy recovery using solution is proposed in this paper. In the system, solution is used to achieve the process of energy recovery from air in solution tower. Otherwise, in traditional air-source heat pump system, evaporator contact with outdoor air directly. In this new system, extracting heat process from environment has two steps:

the first is heat transfer between solution and environmental air, the second is that between evaporator and solution, avoiding frosting on the evaporator surface [8].

There are several liquid desiccants are available: organic solutions such as triethylene glycol, diethylene glycol and ethylene glycol; and brines such as calcium chloride, lithium chloride and lithium bromide. These organic desiccants have a lower vapor pressure than that of the brines at similar temperatures [9]. However, the viscosities of the glycol are much higher than that of the aqueous salt solutions and the pumping cost is higher also, and the glycol is easier to evaporate into the air. To these brines, the vapor pressure of lithium bromide is higher than those of the other two desiccants. So, calcium chloride and lithium chloride are two good candidates to be as work solution in this system.

2. HEAT PUMP FOR ENERGY RECOVERY USING SOLUTION

The principle of the system proposed in the paper is shown in Fig.1. The system is composed of three subsystems: a compression refrigeration subsystem, a

energy recovery subsystem and a solution regeneration subsystem. The compression refrigeration subsystem is composed of a compressor, a condenser, a subcooling-regenerating heat exchanger which couples with the energy recovery subsystem, an expansion valve and an evaporator. The working fluid is refrigerant. The energy recovery subsystem is composed of a solution tower, a solution pump (solution pump A), an adjusting valve (adjusting valve 1) and the subcooling-regenerating

which couples with the compression refrigeration subsystem. The working fluid is solution. The solution regeneration subsystem is composed of two adjusting valves (adjusting valve 2 and 3), a solution heat exchanger, a pre-heat exchanger, a first regenerator, a second regenerator, two solution pumps (solution pump B and C) and the subcooling-regenerating heat exchanger which couples with the compression refrigeration subsystem. The working fluid is solution.

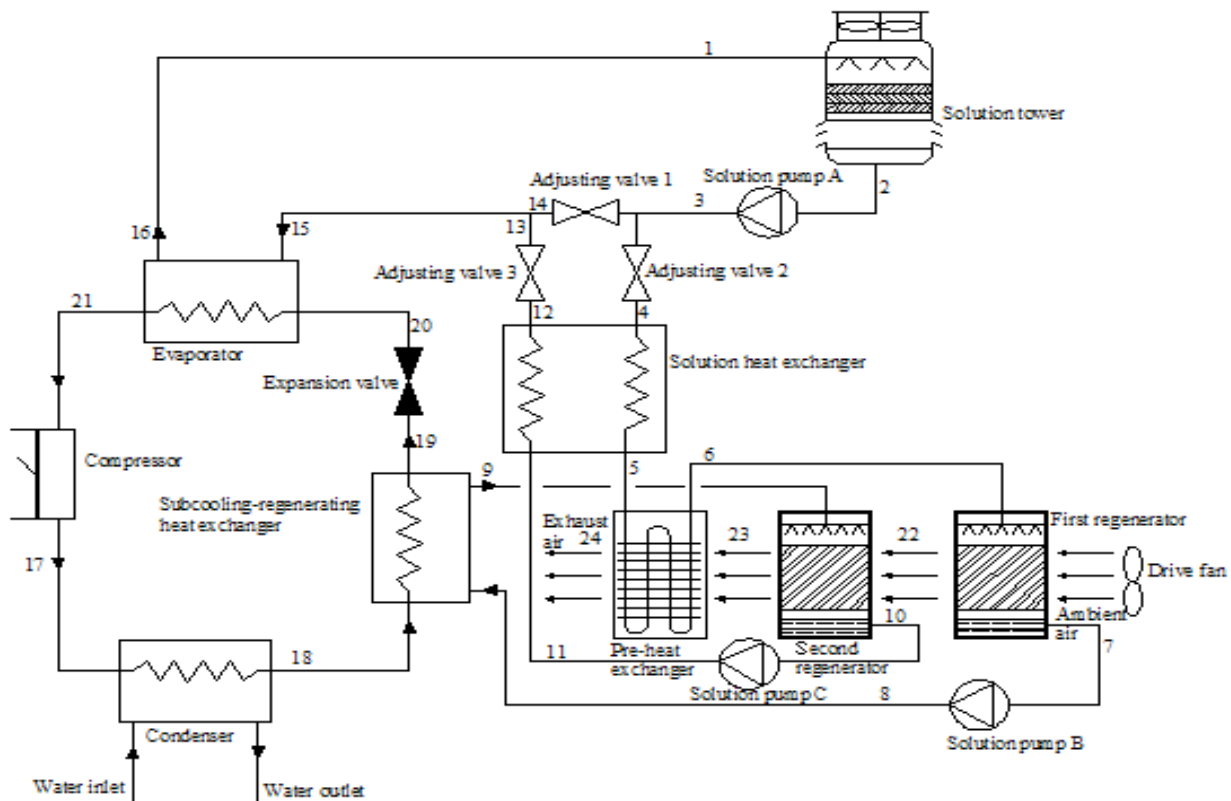


Figure.1. Schematic view of the frost-free air –source heat pump system

3. CONDITIONS OF ANALYSIS

The refrigerant R410A are used in this study, the effect of several key parameters on the performance in winter have been analyzed using calcium chloride and lithium chloride as working solution, respectively. Analysis is performed as varying one of these

parameters while others keeping their designed values. The designed values of parameters are shown in Table 1. The thermodynamic properties of solution were calculated according to literature [10-13]. The results are compared to detail the difference of using calcium chloride and lithium chloride as working solution in the ASHP system.

Table 1 The designed values of parameters

Parameter	Value	Parameter	Value
Inlet solution temperature in solution tower (°C)	-15	Ambient air temperature (°C)	-15
ms/mr in the evaporator (kg/kg)	1.2	Ambient air relative humidity	80%
Mass fraction of solution	0.3	ma/ms in the regenerator (kg/kg)	1.2
ma/ms in the solution tower (kg/kg)	2.5	Condensing temperature (°C)	45

4. RESULTS AND DISCUSSION

4.1 Comparison on effect of inlet solution temperature in solution tower

The effects of inlet solution temperature in solution tower on COP of the ASHP system are analyzed within -35°C — -15°C but keeping other parameters constant and condensing temperature for 45°C when calcium chloride and lithium chloride are used as working solution, respectively.

From Fig.2 it can be seen that COP of the ASHP system have a similar trend: increasing rapidly and then declining as the inlet solution temperature in solution tower increases. This case can be explained as follow: in winter, humidity ratio and temperature of ambient air is low, so temperature rise of solution in solution tower for sensible and latent heat exchange are both finite, this lead the lower inlet solution temperature, the lower evaporating temperature in the evaporator. Accordingly, with the temperature increment of inlet solution temperature in solution tower, the evaporating temperature and the COP increase evidently. When the inlet solution temperature reach a certain value, the COP get a peak and then decline slightly. This is because higher inlet solution temperature makes the temperature difference between solution and ambient air decreases, simultaneously, higher inlet solution temperature rise the vapour pressure of solution, which decrease the drive force of mass transfer, that means the heat absorbed from environment by solution in solution tower decrease, so COP decreases. Fig.2 also shows: using lithium chloride as working solution can get a higher COP than using calcium chloride.

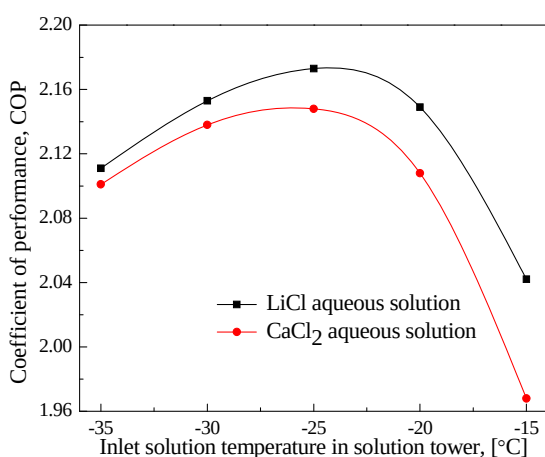


Figure.2 Comparison between inlet solution temperature in solution tower and COP of ASHP system

4.2 Comparison on effect of ambient air temperature

The analysis is done by changing the ambient air temperature but keeping other parameters constant and condensing temperature for 45°C .

Fig.3 describes that COP increase dramatically with the increase of ambient air temperature. ASHP system extracts heat from the ambient air and releases it to meet the demand for heat in winter. Obviously, if the ambient air temperature is higher, the system can extract more heat from the environment, so COP rise dramatically as ambient air temperature increase.

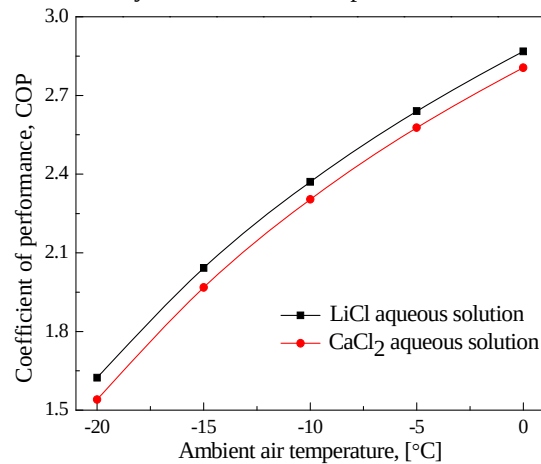


Figure.3 Comparison between ambient air temperature and COP of ASHP system

4.3 Comparison on effect of ambient air relative humidity

The analysis is performed by changing the ambient air relative humidity but keeping other parameters constant and condensing temperature for 45°C .

From Fig.4 it can be known system COP increase slightly with the ambient air relative humidity increase. As we all know, higher air relative humidity means higher air humidity ratio, so solution can absorb more water from air at a high air relative humidity level, that's the reason of COP increase as ambient relative humidity increase. But humidity ratio is low even at high relative humidity in winter, so the water vapour absorbed by solution is quite scarce, which is the just reason that COP merely climb a bit.

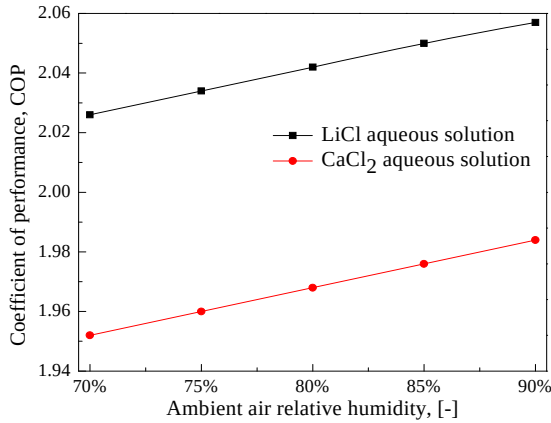


Figure.4 Comparison between ambient air relative humidity and COP of ASHP system

4.4 Comparison on effect of solution-to-refrigerant mass flow rate ratio in the evaporator

The effect is analyzed when changing the mass flow of solution in evaporator but keeping other parameters constant and condensing temperature for 45°C. The solution-to-refrigerant mass flow rate ratio m_s/m_r in evaporator varies from 1.0 to 1.4 in this paper.

Fig.5 shows system COP increase obviously as m_s/m_r increase. The more solution flows into solution tower the more heat can be extracted from environment, so there will be more heat enter into evaporator, then heat exchanging will be enhance in evaporator, and more heat will transfer to refrigerant, going up the evaporating temperature. That is the reason why COP of the system increase as m_s/m_r increase.

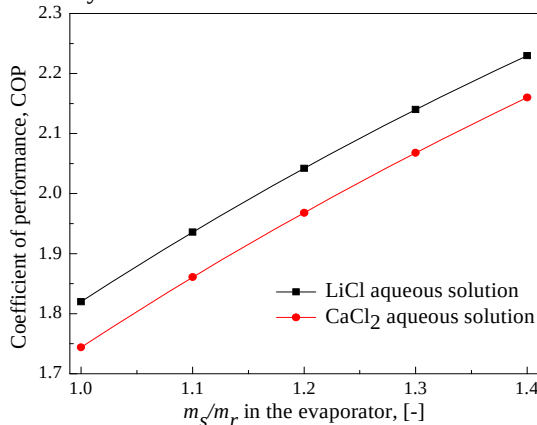


Figure.5 Comparison between solution-to-refrigerant mass flow rate ratio in evaporator and COP of ASHP system

4.5 Comparison on effect of air-to-solution mass flow rate ratio in solution tower

The variation trend of system COP is analyzed by changing mass flow rate of air in the solution tower and condensing temperature for 45°C. The air-to-solution mass flow rate ratio m_a/m_s in solution tower vary from 1.5 to 3.5 in this paper.

Fig.6 describes system COP increase slight as m_a/m_s increase. In solution tower, specific thermal capacity of air is much less than solution, so when temperature of solution and air are equal, mass flow rate of air influence the outlet temperature of solution faintly.

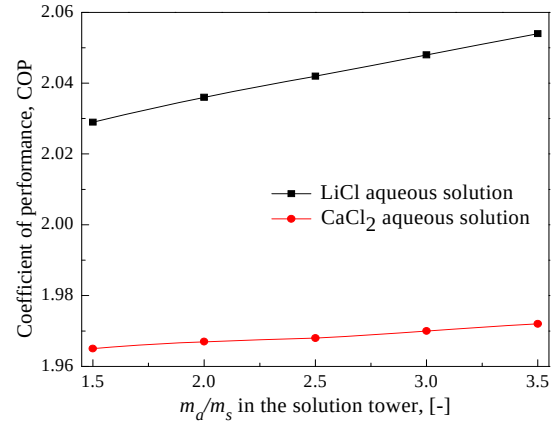


Figure.6. Comparison between air-to-solution mass flow rates ratio in solution tower and COP of ASHP system

4.6 Effect of air-to-solution mass flow rate ratio in the regenerator

The effect of changing mass flow rate of air in the regenerator when keeping other parameters constant and condensing temperature respectively for 40°C, 45°C and 50°C is analyzed, the results are shown as Fig.10. The air-to-solution mass flow rate ratio m_a/m_s in solution tower vary from 1.0 to 1.4 in this paper.

Fig.7 shows system COP decrease as m_a/m_s increase. In regenerator, the capability of solution absorbing water vapour will be recovered by separate out water from solution; the water is taken away by air as water vapour. Simultaneously, the air will take away heat of solution too. So the temperature of solution will fall. Furthermore, more mass flow rate of air, more heat will be taken away and lower outlet temperature of solution, these make the system COP decrease.

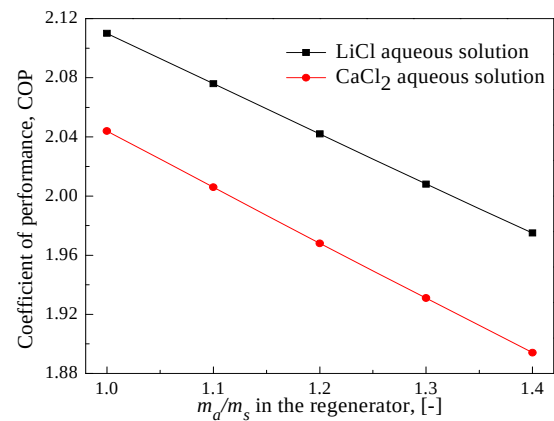


Figure.7. Comparison between air-to-solution mass flow rate ratio in regenerator and COP of ASHP system

4.7 Effect of total heat transfer efficiency of solution tower

The effect of total heat transfer efficiency of solution tower is analyzed keeping other parameters constant and condensing temperature for 40°C, 45°C and 50°C, respectively.

It can be seen from Fig.8 that system COP climb as total heat transfer efficiency of solution tower increase. Evidently, we can know that total heat transfer rate is directly proportional with total heat transfer efficiency. So higher total heat transfer efficiency means higher total heat transfer rate and the higher total heat transfer rate shows the system can extract more heat from environment, this cause the system COP climb.

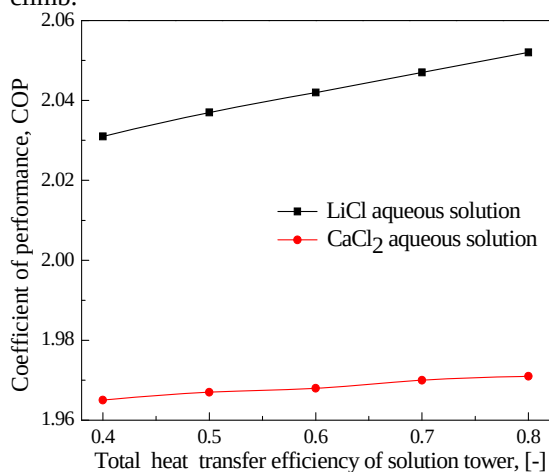


Figure.8. Comparison between total heat transfer efficiency of solution tower and COP of ASHP system

4.8 Effect of total heat transfer efficiency of the regenerator

The effect of total heat transfer efficiency of the regenerator is analyzed keeping other parameters constant and condensing temperature for 40°C, 45°C and 50°C, respectively.

Fig.9 shows the effect of total heat transfer efficiency of the regenerator on COP of ASHP system. From this figure, it can be seen that system COP deteriorate as total heat transfer efficiency of the regenerator increase. For higher total heat transfer efficiency of the regenerator, air which flow into regenerator will take away more heat of solution, it's that to say the heat extract from environment will decrease, so system COP will deteriorate.

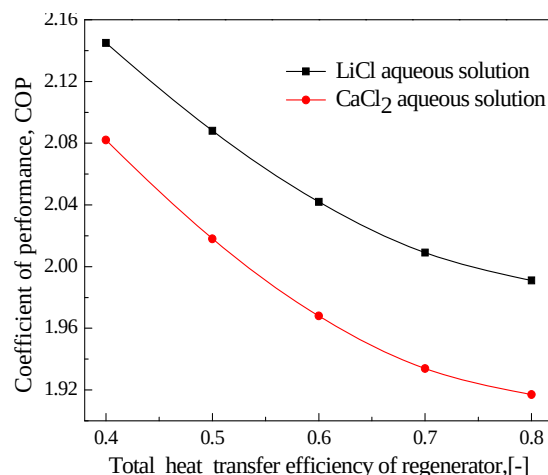


Figure.9. Comparison between total heat transfer efficiency of regenerator and COP of ASHP system

5. SUMMARY AND CONCLUSIONS

The conventional ASHP system can not operate efficiently in winter for frosting on evaporator surface, so defrosting technology have been developed widely. But all these measure have shortcomings, which can not solve this problem in essence.

This paper propose using aqueous solution to recovery energy, this system can realize not only cooling in summer but also heating which needs not defrost in winter. The operating principle is illustrated, and then the effects of several key parameters on COP of the ASHP system are analyzed. When keep temperature difference between inlet solution temperature and ambient air at a certain level, increase inlet solution temperature and mass fraction in solution tower will both improve the system COP, but they also be restricted by ice and crystallization range of the solution. Ambient air conditions also affect the system's performance. Increasing temperature or relative humidity of air means there is a higher humidity ratio; it can enhance the heat and mass transfer between solution and air, and then raise COP of the system. To solution tower, the higher air flow rate and total heat transfer efficiency will improve the COP of the ASHP. On the contrary, the higher air flow rate and total heat transfer efficiency of regenerator will deteriorate the COP of the ASHP. The appropriate mass flow rate of working fluid in evaporator will improve COP of the system.

According to the analysis results it can be seen that even the ambient air temperature arrive to -20°C, the system still operate at an acceptable performance. In addition, the ASHP system can be operated continuously and needs not to run in defrosting mode periodically, which will widen the application of ASHP system.

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EXPERIMENTAL STUDY ON ANTI-FREEZING OF SOLAR ENERGY SYSTEM IN SEVERE COLD AREAS

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ABSTRACT

As a kind of sustainable and promising energy, solar energy is getting increasing concern due to its cleanliness and renewability. However, freezing of the fluid in solar systems employed in severe cold areas is a troublesome issue which limits the widespread application of solar energy technologies. To solve this issue, an anti-freezing strategy based on antifreeze fluid was developed in this paper. To test the effectiveness of this strategy, an experimental solar system with all-glass vacuum tube collectors were built in the northeast city of China, Harbin. A series of tests on this novel operation strategy were carried out at continuous sunny and rainy weather respectively. Test data suggested that the proposed anti-freezing strategy was viable and effective at different weather. It can reduce the concentration of antifreeze fluid from 50% to 30%, even to 20% in some cases. In addition, this anti-freezing strategy was demonstrated more reliable, safe and economic than the traditional anti-freezing measures. The optimal concentration of antifreeze fluid is also discussed in the paper, suggesting that the concentration of antifreeze fluid should be determined through comprehensively considering some parameters, such as solar system form, thickness of insulation layer and the anti-freezing operation strategy, not only based on the lowest outdoor air temperature like the traditional method. The research in this paper can provide a key reference for the designing and operating as well as the determination of antifreeze fluid of solar systems in severe cold areas in further.

Keywords: solar system, anti-freezing strategy, antifreeze fluid, heating, severe cold areas.

1. INTRODUCTION

Solar is a kind of renewable energy which can be exploited and replace the conventional energy. Solar thermal collection system has been utilized widely over the world. However, solar systems are mostly limited to the intensity and angle of solar radiation, variation of outdoor air temperature and type of solar collector [1-4].

In severe cold areas, the outdoor air temperature is very low which causes a high heat loss, and in turn freezing of the circulating water happens in solar systems, which is a main issue faced by the users. To solve this issue, some researchers focused on the development of anti-freezing measures and strategies. Toth [5] presented one kind of solar collector fabricated from plastic transparent insulation materials (TIM) saving auxiliary anti-freezing measures. Therefore, it can result in significant cost reduction in solar water heating. Zhao [6] analyzed the performances of five familiar anti-freezing methods used in solar water pipeline in winter. All these anti-freezing strategies and measures only can be used in small buildings and suitable to small domestic hot water systems. For large-

scale solar systems, such as the solar heating system, the conventional measures utilized by users to prevent freezing includes electric tracing band, antifreeze fluid and heating the outdoor devices and pipes by the hot water in thermal storage tank. As known, the method of anti-freezing by electric tracing band was expensive and dangerous. The anti-freezing measure, heating by hot water required extra power consumption of pump and hot water heating. In real projects, the use of antifreeze fluid in solar system was much popular in severe cold areas. However, because of the viscosity, corrosivity, and expensiveness of antifreeze fluid, this method required special equipments, pipes and system types, resulting in a high cost. Actually the cost rose with the increase of concentration of antifreeze fluid. Hence, to cut the cost of this method, reducing the concentration of antifreeze fluid required by the system is a research field for the use of solar system in severe cold areas.

In severe cold areas, the concentration of antifreeze fluid in solar systems was determined according to the local ambient air temperature. The freezing point of antifreeze fluid should be higher than the ambient air temperature at a certain range. In fact, the required design concentration of antifreeze fluid was much higher than that needed. Taking the city of

Harbin (in severe cold areas) for example, the suitable concentration of antifreeze fluid was 50% (the freezing point was -33.8°C) in consideration of the calculated temperature of the exterior heating (-26°C) in winter. In winter nights, with the cooling of air temperature, the fluid temperature in solar collector and pipes decreased gradually. As the water capacity was small in pipes, the fluid temperature can decrease from 30°C to 0°C , or even lower than 0°C in a few hours even though the insulation was sufficient. However, the all-glass vacuum tube collector has larger water capacity, better insulating property and higher fluid temperature than the pipes.

A new anti-freezing strategy was proposed in the paper and a series of experiment was performed in severe cold areas, Harbin. The purpose of this study was to analyze the fluid temperature changing in solar system and to research the economic and reliable methods of anti-freezing. The suitable concentration of antifreeze fluid was also discussed in the paper which can provide a key reference for the designing of solar

systems as well as the determination of antifreeze concentration.

2. DESCRIPTION OF THE SYSTEM AND ANTI-FREEZING STRATEGY

2.1 Experimental system description

According to *Technical code for solar heating system* (GB50495-2009), the suitable type of solar system for severe cold areas was indirect cycle system filled with antifreeze fluid or air in solar collectors, and the terminal heating ways were floor radiation or hot-air heating. Therefore, an experimental platform was built in terms of the recommendation in above references, as illustrated in Figure 1. The experimental system was to research the effect of anti-freezing and solar collecting performance in winter, so the equipment types and the installation slope of solar collector were considered in winter conditions.

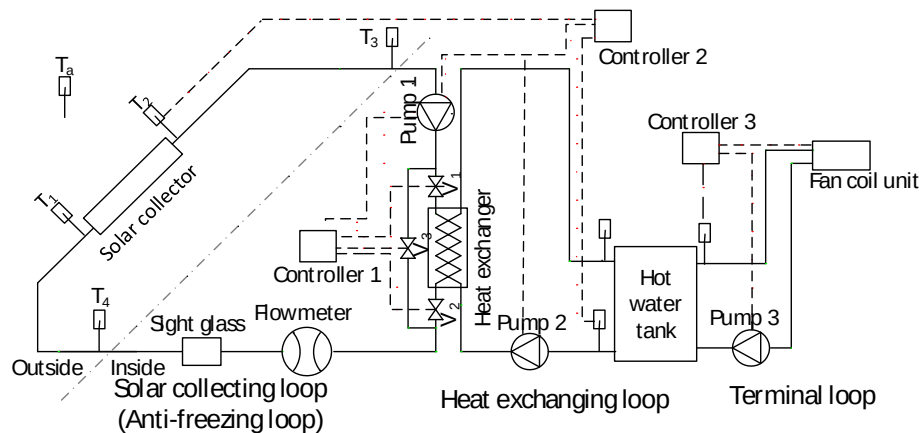


Figure 1 The schematic diagram of the system

The experimental system consisted of solar collecting loop (anti-freezing loop), heat exchanging loop and terminal heating loop, as well as data acquisition and control system. In this system, a glycol-water mixture was used as the antifreeze fluid and the solar collecting loop was an open-loop subsystem.

The gross area of solar collector was 3.5 m^2 in the system. The out and inner diameters of collector tube were 58 mm and 47 mm respectively, and the total length was 2 m, arranging in south-north direction. The total area of heat exchanger was 0.3 m^2 , and the capacity of thermal storage tank was 150 L. Controller 1 was used to realize the control of pump and valves in anti-freezing loop realizing by a 16-channel serial relay. Controller 2 was a differential temperature controller and controller 3 was a thermostatic controller, both of which were used in solar collecting mode. The temperatures were measured by a T-type thermocouples and the flow rate was tested by a glass-tube rotameter. There was a sight glass in the pipeline before the inlet of the collector in order to observe whether there are impurities or freezing in the working fluid.

2.2 The proposed anti-freezing strategy

Two operation modes can be realized in this system: solar collecting mode and anti-freezing mode. In sunny weather, the solar collector can absorb the heat from solar radiation and raise the fluid temperature in it. When the temperature of T_2 (T_6 for system 2) increased to the set value, the pumps of solar collecting loop and heat exchanger loop were turned on and the solar collecting mode began to operate. In the nights and rainy days, as there is no solar radiation, the fluid temperature in solar system was low. The anti-freezing mode must be operated to prevent freezing of pipes and devices. Because the length of this paper was limited, the system performance at antifreeze mode was presented, and the system performance at solar collecting mode will not be covered in this article, which will be reported in follow-up papers.

In anti-freezing mode, the by-pass loop was set beside heat exchanger. Valves 1, 2 were closed and valve 3 was opened. The high temperature fluid in solar collector was circulated by pump to the outdoor pipes

where the fluid temperature was lower and much easier to freezing. The power energy consumption (by pump) was low in anti-freezing mode and no additional thermal energy required. It can ensure the fluid temperature in outdoor pipes was higher enough than the ambient air temperature.

Two identical systems (system 1 and system 2) were built to compare the anti-freezing effectiveness in different concentration or flow rate of the fluid. The parameters setting are shown in Figure 2 and Figure 3. T_3 , T_4 , T_7 and T_8 were located at the points where freezing was easy to happen inside the outdoor pipes.

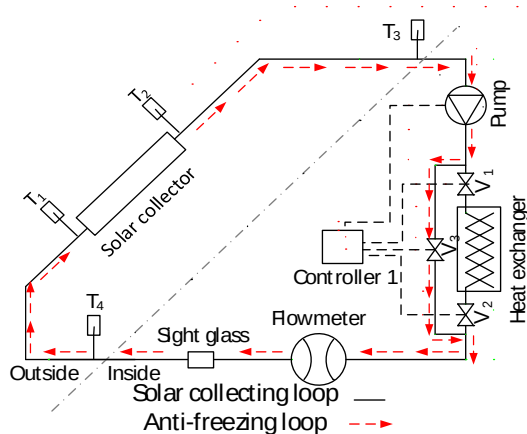


Figure 2 The parameters setting and anti-freezing loop of system 1

For system 1, it was obvious that the temperature of T_1 and T_2 were higher than T_3 and T_4 . Thus, in controller 1, the pump and valves were controlled according to the value of T_3 and T_4 , which is similar to controller 4 in system 2.

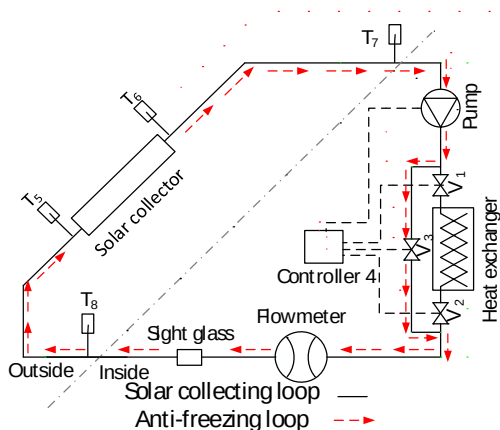


Figure 3 The parameters setting and anti-freezing loop of system 2

3. RESULTS

The anti-freezing strategy was operated based on the outdoor environmental conditions and the fluid temperature in the pipes. Here, the anti-freezing effectiveness and the temperature changing in system 1 and 2 are discussed in sunny days and continuous rainy

weather. The experiment was carried out from December 1, 2012 to February 28, 2013 in Harbin. The daily ambient air temperature (T_a) of the city during the testing period is shown in Figure 4.

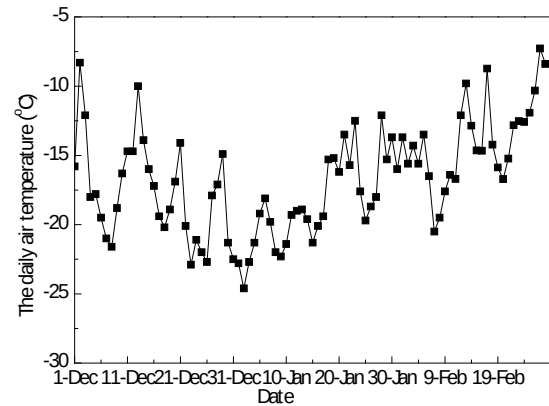


Figure 4 The daily ambient air temperature of Harbin (from December 1, 2012 to February 28, 2013)

3.1 Sunny days

For sunny days, the solar collecting loop was operated in the day, and the temperature of fluid in collector was much higher. It was advantageous for anti-freezing in the night because the thermal capacity was large and the fluid temperature was high enough. Usually, solar collecting mode lasted from 9:30 am to nearly 5:00 pm in sunny days, and the fluid temperature was above 20°C at 6:00 pm. In the experiment, the mass concentration of antifreeze fluid were 50% and 30% respectively for system 1 and 2, with the freezing point were -33.8°C and -14.1°C. The system 1 was static at night as a reference test while the system 2 were operated by anti-freezing mode.

The fluid temperatures at different measuring points of systems 1 and 2 were tested, together with the ambient air temperature, T_a . All the data are shown in Figure 5.

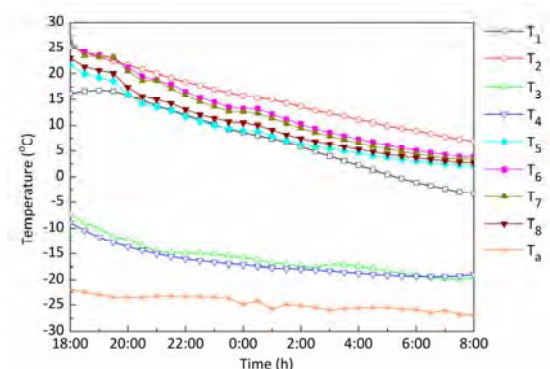


Figure 5 The temperature contrast of system 1 and 2 on sunny nights (1)

For system 1, the fluid temperatures in the collector and pipes differ significantly. T_1 and T_2 were the fluid temperature at collector inlet and outlet. Because the solar collecting mode was operated in the

day and the thermal capacity of collector was large, T_1 and T_2 were much higher than T_3 and T_4 at the time of 18:00. Until 8:00 of the next day, T_1 and T_2 were about 0°C while T_3 and T_4 decreased to nearly -20°C . In this condition, the high concentration of antifreeze fluid must be used to prevent freezing.

For system 2, as the novel anti-freezing mode was operated in the night, the temperatures of T_5 , T_6 , T_7 and T_8 were almost the same and the difference were less than 4°C . Until 8:00 am of the next day, the temperatures were higher than 0°C . In this situation, the low concentration of antifreeze fluid or even pure water can be used as the heat transfer medium. The temperature difference was related to the fluid flow rate. The fluid flow rate in system 2 was about 40 L/h in Figure 1. When the fluid flow rate increased to 80 L/h, the temperature difference was almost less than 1°C . However, with the increasing of flow rate, temperature difference decreased and the heat dissipation and thermal losses increased. Consequently, the recommended fluid flow rate for system 2 was 60 L/h ~ 80 L/h.

When the solar radiation intensity was not enough, the running time of solar collecting mode was short or almost zero in the day. Then the temperatures of the fluid were lower than those in sunny days, as shown in Figure 6. The variation of temperature was similar to the values as indicated in Figure 5. The temperature in system 2 was about 0°C at 6:00 pm and the useful thermal capacity in solar collector was less than sunny days. The lowest temperature in system 2 (T_7) was -8°C at 7:00 am. Owing to the cold sky radiation, T_3 and T_4 were even less than T_a at times.

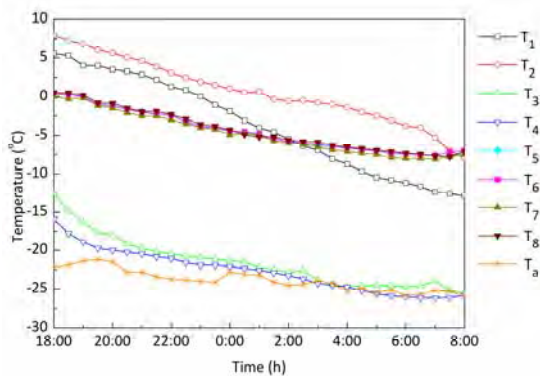


Figure 6 The temperature contrast of system 1 and 2 on sunny nights (2)

3.2 Continuous rainy weather

In continuous rainy weather, it was adverse for the application of solar apparatus. A auxiliary heat source was used to provide heat to the outdoor pipes and devices of solar systems to solve the issue of freezing in rainy weathers. Therefore, the extra energy would be consumed and the initial investment was vast.

The proposed anti-freezing mode was also tested in continuous rainy weather to research the anti-freezing effectiveness, as shown in Figure 7.

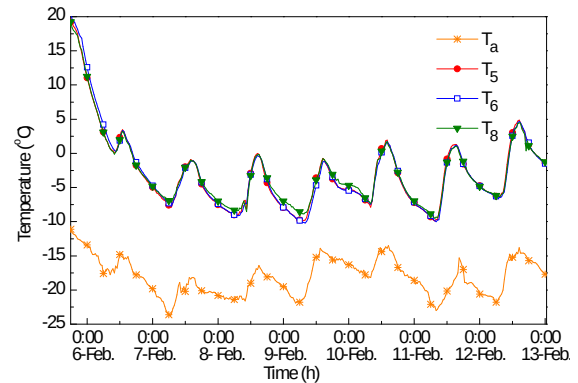


Figure 7 The temperature changing of system 2 in continuous rainy weather

The test was lasted from February 5, 2013, 18:00 to February 13, 2013, 0:00. During the test period, the concentration of antifreeze fluid of system 2 was 30%. It can be seen that the testing temperature in system 2 changed regularly. Generally, the temperature increased in the daytime and decreased in the night, and the daily lowest temperature appeared at 7:00 am ~ 8:00 am. By the proposed anti-freezing mode, the fluid temperature in system 2 was above -10°C while the ambient air temperature was -24°C . The concentration of 30% (the freezing point was -14.1°C) in system 2 was high enough to prevent freezing in the outdoor pipes. However, not all the days from February 5 to February 13 were rainy. The solar collector was shaded to avoid the sunshine during testing period and the solar radiation was zero even in the daytime to simulate the continuous rainy weather. In fact, there was still some amount of solar radiation and the fluid temperature can rise to a certain range in rainy days. So the temperatures were higher in real situation than that of tested conditions and the anti-freezing effectiveness was better, that is, the fluid in outside pipes can't get freezing by the operation of anti-freezing mode even in continues rainy weather.

3.3 Discussion

On the basis of experiments introduced above, the concentration of antifreeze fluid was changed from 30% to 25% and 20%. The temperature changing and the anti-freezing effectiveness were also tested and analyzed. The temperature changing has similar trends regardless of the different concentration of antifreeze fluid of 30% or 20%. It was important to note that even the lowest temperature of the system was -9°C when the concentration of the fluid was 20% (the freezing point was -7.8°C), the fluid was still able to flow in the pipes. When the temperature in the pipes was 2 degrees lower than that of freezing point, it influenced the flow rate in the system and the pipes could potentially freeze in this situation. As a result, the suitable concentration of antifreeze fluid was 30% or 25% in Harbin, if the proposed anti-freezing mode was operated in winter nights.

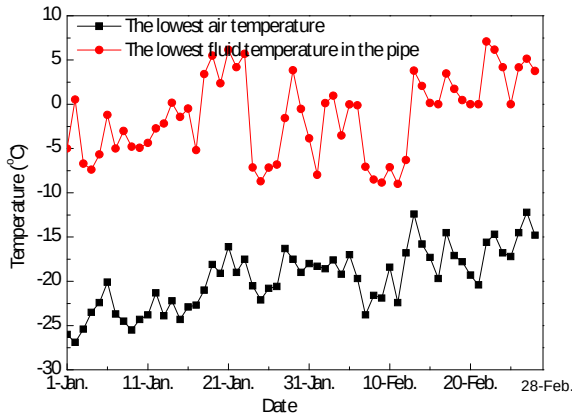


Figure 8 The daily lowest temperature of the fluid under the anti-freezing mode

As is shown in Figure 8, when the anti-freezing mode was operated during the nights, the lowest testing temperature was -9°C . From February 15, another control strategy of intermittent cyclic was used to set the lowest temperature of 0°C . In general, the lowest fluid temperature in the pipes was almost above 0°C when the lowest air temperature was above -20°C , which can provide a reference to the design of solar systems in freezing zone.

4. CONCLUSIONS

The current method that determining the concentration of antifreeze fluid according to the ambient air temperature in solar systems was problematic, which often led to a higher concentration with a higher expense than that needed. The suitable concentration of antifreeze fluid should be determined through comprehensive consideration of solar system form, thickness of insulation layer and the anti-freezing control strategy, as well as the ambient temperatures.

A novel anti-freezing operation mode was proposed to prevent freezing in this study. Test data suggested that the proposed anti-freezing strategy was viable and can reduce the concentration of antifreeze fluid from 50% to 30%, even to 20% in some cases without freezing. Therefore, it can cut the initial cost of solar system. The study in this paper can be taken as a reference for the designers and users of solar system in severe cold areas.

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DIFFERENT HEATING SYSTEMS OF FLAT PLATE SOLAR COLLECTOR EFFICIENCY IN LHASA

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ABSTRACT

The flat-plate solar collector's thermal performances are tested in different working conditions as direct heating model and water tank heating model in Lhasa. Compared to existing norms and commonly used solar collector efficiency calculation methods, and comparing solar radiation intensity, import & export water temperature of solar collector, effective heating of solar collector, import & export water temperature of user side, the instantaneous efficiency curve is given in this paper. Comparing experimental data with nameplate marked efficiency, the measured value often can not reach the nameplate manufacturer calibration value, and the average instantaneous efficiency of the solar collector shows that the data of direct heating model is up to that of water tank heating model. Experimental result shows that the models each have their own advantages and disadvantages.

Keywords: Lhasa, Flat plate solar collector, Heat collecting efficiency, Direct heating model, Water tank heating mode.

1. INTRODUCTION

The steady state heat collecting efficiency of solar collector is the key factor to evaluate collector's thermal performance. In this method, it are given a set of test conditions, and draw the instantaneous efficiency formula with stable working conditions and then come to the average collecting efficiency. Even though the steady state test model is easy, test conditions are highly required. The rigorous experimental conditions are set in the steady state testing of the flat plate solar collector's instantaneous efficiency in "Solar collector thermal performance testing methods" (GB/T4271-2007)[1]. The conditions include the level of solar irradiance and its variation, the air velocity in the environment less than 4m/s, the mass and flow of the working fluid, the angle of incidence of the sun.

On the other hand, due to the difference between the testing conditions and the flat plate solar collector's usual working conditions, the test results are often inconsistent with the thermal performance of the actual working conditions. To solve the problem, the solar dynamic testing method is developing, which can test the thermal performance in the frequently changing weather conditions. Hot main test methods are mainly divided into one-node method, multiple-nodes method and function response method. Amar developed a transient test method to characterize the dynamic behavior of flat plate solar collectors based on a one-node model[2]. A survey on different methods for rating the thermal performance of solar collectors in transient conditions has been made. Two inexpensive methods

(Saunter's and Excel's) have been selected for an extensive comparative study[3]. Also, a new procedure for testing solar flat plate collectors under unsteady weather conditions has been developed[4]. Among the most representative one-node model are Saunier model, Excell model [5], Perers model[6], QDT model[2], Wijeyundera model and NDM model[4]. DSC model [7] is on behalf of multiple nodes model. Rogers model [8] and Filter model[9] are the representation of function response method.

In China, the dynamic testing method only remains in the research stage. However, the steady state test of the flat plate solar collector's instantaneous efficiency is conducted as the specification[1] in the market. In the actual project, the intensity of solar radiation is largely different in different regions, and obviously there is a big error from the design if the system design is decided solely on the basis of the steady-state calibration coefficient of thermal performance of solar collectors. A transient thermal performance of flat plate solar collector was

tested in Lhasa. The calculation result was compared to factory calibration efficiency in order to guide the design and selection of solar collectors more accurate in locality.

NOMENCLATURE

Symbols

a_1, a_2	coefficient, as a reference, $W/(m^2 \cdot ^\circ C)$	F	mixed solution flow in collector panels, m^3/s
A	the total collector area, m^2	G	solar radiation intensity of the lighting surface of the solar collector, W/m^2
A_n	effective collector area in each collector panel, m^2	G_d	instantaneous solar radiation intensity, W/m^2
C	the specific heat capacity of liquid mixture of water and antifreeze in collector panel, $J/(kg \cdot ^\circ C)$	J_T	the designing month average solar total radiation with a certain inclination in locality, kJ/m^2
c_f	corresponding to the average working fluid temperature t_m heat transfer working fluid specific heat capacity, $(J/kg \cdot ^\circ C)$	K_0	incidence angle correction factor
m	mass flow of the working fluid, kg/s	n	the number of collector panels
Q	useful power, KW	Q_v	effective set of heat of solar collector, KW
SY	average day solar irradiation hours in designing month, h	ΔT	difference between the outlet and inlet temperature of the working fluid $=t_o - t_i$, $^\circ C$
T^*	normalized temperature difference, when using the inlet temperature $T_i^* = \frac{t_i - t_a}{G}$, $(m^2 \cdot ^\circ C)/W$	t_a	environmental or ambient air temperature, $^\circ C$
t_m	the medium temperature of the collector, $^\circ C$	t_{in}	the designing import temperature of the water tank, $^\circ C$
t_{out}	the designing export temperature of the water tank, $^\circ C$	t_o, t_i	outlet temperature and inlet temperature in the mixed solution of the collector, $^\circ C$
U	solar collector heat loss coefficient, $W/(m^2 \cdot ^\circ C)$		
Greek Letters			
η	instantaneous heat collecting efficiency	η_0	$T_i^* = 0$, the value of η
ρ	mixed solution density in collector plates, kg/m^3		

2. THE ANALYSIS OF DYNAMIC TESTING

The measured building is a multi-storey office building in Lhasa, and the construction area of which is 7,500 square meters. The cold and hot sources are mainly solar panels, supplemented by air-cooled heat pump. Collector Specifications are as follows: Model FKS-1S, effective collector area of 2.25m², calibration instantaneous efficiency of 0.77, working pressure 0.6MPa, water capacity of 86kg, recommended operating flow block 50L/h, the collector voltage drop. 250Pa work inclination 45°, 88 pieces of panels. Experimental circuit is shown in Figure 1.

2.1 Test conditions

The system is divided into two modes of operation, one is: Solar collector hot water goes directly into the

plate heat exchanger and exchange heat with air conditioning return water, referred to as the direct heating mode, the other is: Solar collector hot water goes into the solar storage tank, and exchanges heat with tank water, storage tank hot water goes into the plate heat exchanger and exchanges heat with air conditioning return water, referred to the water tank heating mode. Low-temperature air-cooled heat pump units are connected in series between the plate heat exchanger and air-conditioning supply and return water, when solar collectors or heat storage tank through plate for the air conditioning return water's temperature still cannot meet the heating temperature, air-cooled heat pump reheat.

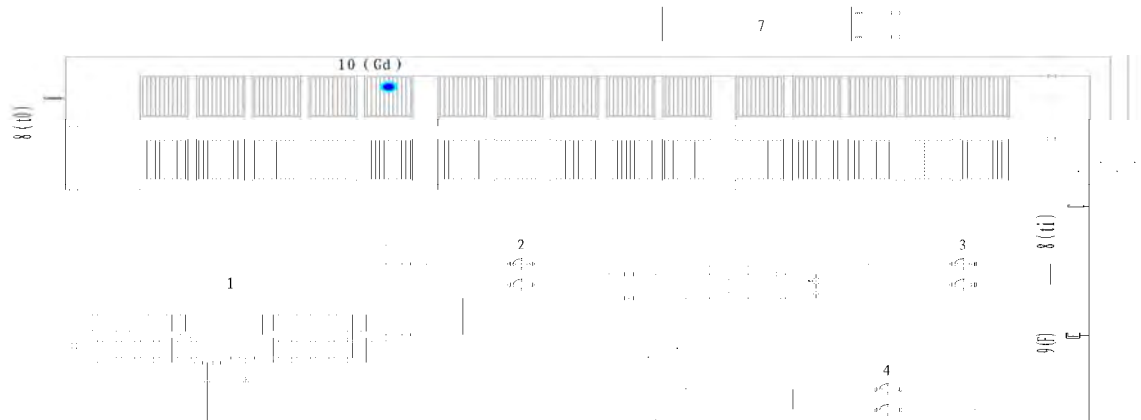


Figure 1 Dynamic thermal performance testing diagram of solar collector heating system

1 Solar heat storage water tank, 2 First round circulation pump of water tank, 3 The heating circulation pump, 4 Solar glycol circulating pump, 5 Air conditioning water supply, 6 Air conditioning water, 7 Low temperature type air-cooled heat pump units, 8 temperature sensor, 9 flowmeter, 10 Solar radiation tester.

2.2 Test cycle and parameters

Make $\Delta s=10\text{min}$ for the step length. The main measured physical quantities are as follows: instantaneous solar radiation intensity, mixed solution flow in the heating collecting plate F, outlet temperature and inlet temperature in the mixed solution of the collector. Results should be clear and concise t_o , t_i .

3. TEST PRINCIPLE

3.1 Dynamic test method of solar panels in this experiment

$$\eta = \frac{Q_v}{nA_n G_d} = \frac{CFp(t_o - t_i)}{nA_n G_d} \quad (2)$$

3.2 Standard steady-state instantaneous efficiency of the collector conclusions

According to the specification[1], the instantaneous efficiency of solar collectors running under steady state conditions η can be defined as the ratio of useful power of collector to received solar radiation power the surface of the collector gains [10]. The collector instantaneous efficiency equation was shown in equation (2) [11].

$$\eta = \frac{Q}{AG} = \frac{mc_f \Delta T}{AG} \quad (2)$$

Instantaneous efficiency normalized function of the temperature difference T^* graphically represented, which was expressed as follows.

$$\eta = \eta_0 K_0 - a_1 T_i^* - a_2 G(T_i^*)^2 = \eta_0 K_0 - a_1 \frac{t_i - t_a}{G} - a_2 G \left(\frac{t_i - t_a}{G} \right)^2 \quad (3)$$

The test conditions are as follows.

Table 1 Experimental parameters scope

Parameters	Range	Allowed deviate range
solar irradiance	$\geq 700\text{W/m}^2$	$\pm 50\text{W/m}^2$
ambient air speed	$\leq 4\text{m/s}$	—
ambient air temperature	—	$\pm 1^\circ\text{C}$
mass flow rate of the working fluid	$0.02\text{kg}/(\text{m}^2 \cdot \text{s})$	$\pm 1\%$
direct solar radiation incidence angle	—	angle of incidence $\pm 2.5\%$
collector inlet working fluid temperature	$\leq 70^\circ\text{C}$	$\pm 0.1^\circ\text{C}$

3.3 The mean monthly collection efficiency η_{cd} of solar collector

η_{cd} could be assigned a value range of 0.40~0.50 when no specific test data. When calculating the

monthly average collection efficiency, the medium temperature can be calculated as formula (4)[12].

$$t_i = \frac{t_{in}}{3} + t_{out} \times \frac{2}{3} \quad (4)$$

The total radiation intensity of illumination G is valued by formula (5).

$$G = J_T / (S_Y \times 3600) \quad (5)$$

Substituting the above data into instantaneous efficiency curve equation (3), the mean monthly collection efficiency can be got.

4. THE ANALYSIS OF DYNAMIC TESTING

According to the formula (1), the direct heating mode and tank heating mode parameters of the solar collector system are tested and the dynamic efficiency of the collector is calculated. Test parameters: instantaneous solar radiation intensity G_d , mixed solution of the collector outlet temperature and inlet temperature t_o , t_i , collector plates mixed solution flow F . The major computed quantities: the effective set of the heat Q_v , collector instantaneous efficiency η_d .

4.1 Direct heating mode

Using an ultrasonic flowmeter, the measured collector plate mixed solution flow rate of 21.95m³ / h,

and other measurement parameters are shown in Figure 2. From Figure 2 we can see that the intensity of solar radiation collector and working fluid inlet and outlet temperature have the similar trend, the peaks appear 15:00 to 15:30. Solar radiation intensity is up to 1200W/m².

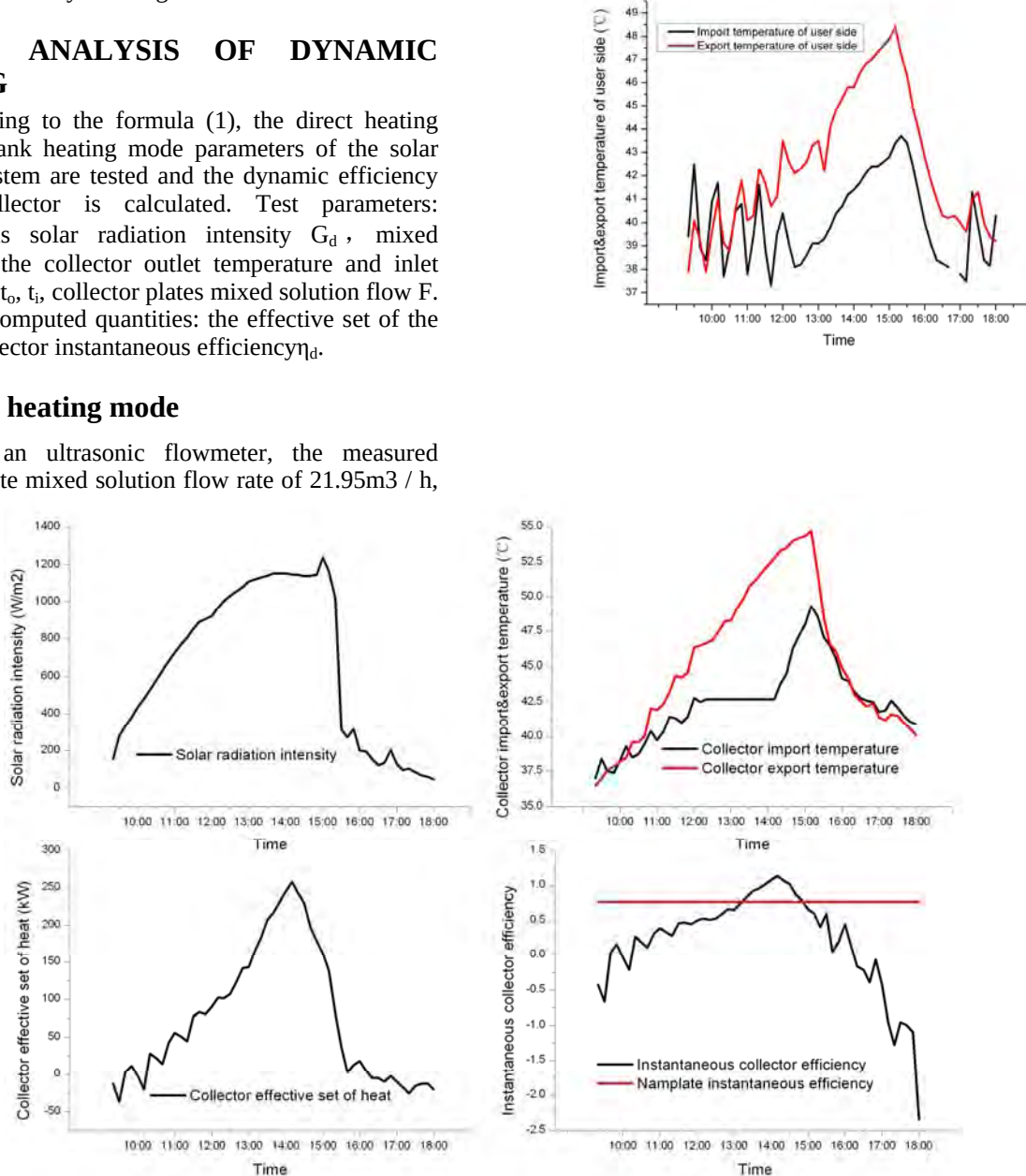


Figure 2 Direct heating mode each parameter dynamic changes curves

After 15:00, solar radiation intensity rapidly decays. Collector working fluid inlet and outlet temperature trends also will be performed very consistent, the collector working fluid inlet temperature reached a maximum of 54.35 °C at 15:00, 10 minutes later outlet temperature reached the maximum of 49.3 °C. Collector working fluid inlet and outlet temperature difference at 14:10 reached the maximum of 10.1 °C. According to the formula (1), effective set of the heat and the instantaneous efficiency reached the

maximum value, were 257.78KW and 1.135. It can be seen from the instantaneous efficiency of the collector, the collector dynamic efficiency most of the time a day is less than the factory calibration value of the instantaneous efficiency. It can be seen from the import & export temperature of user side that, before 11:30, the heat exchange between the medium of solar collector and of user is not obvious. In some moment on the plate heat exchanger sides, the back-water temperature of user side is greater than that of solar collector outlet

water, which means that the solar collector outlet water absorbed the heating of the user side back-water. The result leads air-cooled heat pump assisted heating for the user side back-water to rated water supply temperature. The kind of situation caused a certain

amount of energy waste. when the import & export temperature difference maintain between 4~5°C, the heating exchange effect is better.

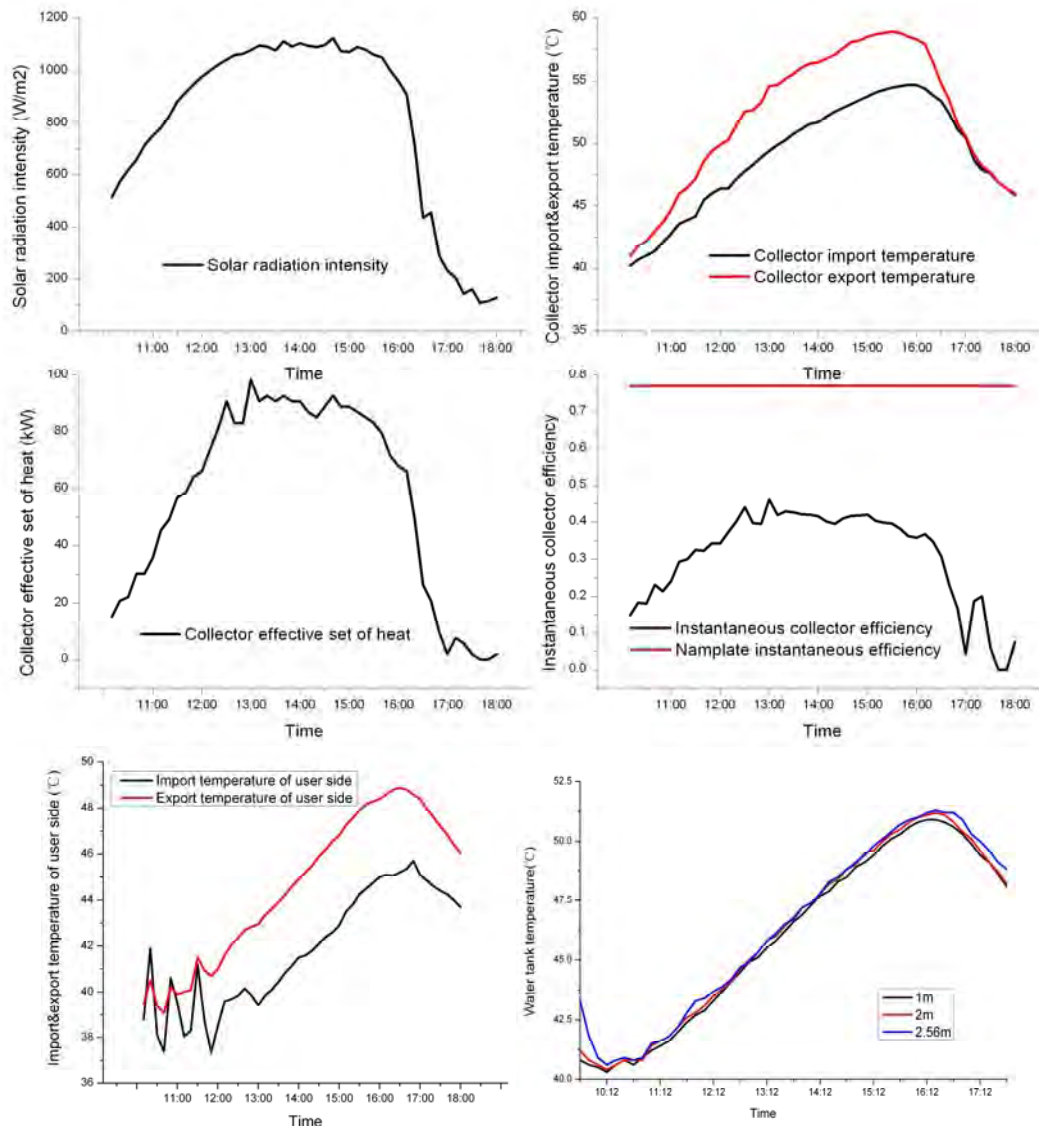


Figure 3 Water tank heating mode each parameter dynamic changes curves

4.2 Tank heating mode

Using an ultrasonic flow meter, the measured collector plate mixed solution flow rate of 16.25m³ / h, and other measurement parameters is shown in Figure 3. As can be seen in Figure 3, from 12:10 to 15:50 solar radiation intensity is above 1000 W/m², the working fluid inlet and outlet temperature of the collector is into a parabolic variation trend, collector working fluid inlet temperature reached at 15:30 maximum of 58.5 °C, the outlet temperature reached the maximum of 54.7 °C at 15:50. Collector working fluid inlet and outlet temperature difference at 13:00 reached the maximum of 5.2 °C. Collector effective set of heat is 98.26KW the instantaneous efficiency 0.46. It can be seen from

the collector instantaneous efficiency the day dynamic efficiency of the collector is less than the factory calibration value of the instantaneous efficiency. As can be seen from the user side board in the import and export temperature, compared with the direct heating mode, in the tank heating mode, the plate heat exchanger of user side, the export temperature is mainly greater than the import water temperature. The import and export temperature difference maintain between 3 ~ 4 °C, so the heat exchange efficiency is poor than that of direct heating mode.

5. CONCLUSION AND ANALYSIS

Comprehensive compare Figure 2 and Figure 3, total amount of radiation during the day of the direct

heating mode is less than the tank heating mode, but the value of dynamic efficiency of the collector of the former is larger than the latter. The dynamic efficiency of the collector is not only related to the radiation intensity, but also has a great relationship with the collector inlet and outlet temperature of the working fluid. When other factors are constant, the greater inlet and outlet the temperature difference is, the greater the collector effectively set the heat. For direct heating mode, the working fluid heated in the collector exchange heat with the end of the heating system through plate heat exchanger. For water tank heating mode, the medium of solar collector exchange heating with water tank water through the coil, and then the tank water transfer heat to the end of the heating system. The tank mode experienced two indirect heating exchange, such the system overall efficiency is declined. In addition, the initial water tank temperature is more than 40°C, so that comparing direct heating mode, the import & export temperature difference is much smaller, and the collector exchanged less heat.

Compared to the national standard of "The solar collector thermal performance testing method" (GB/71-2007) the steady-state collector instantaneous efficiency test method, in engineering applications, the collector dynamic efficiency value is usually different from nameplate value. The standard [1] defines settled test conditions, the allowable deviation range is very small, but in engineering application, the outdoor environment parameters are variable, different type of solar heating system result distinction of the solar collection efficiency. In solar heating system design, the regional climate characteristics and suitable system type are both the key of influencing factor for solar collector efficiency parameters.

As seen from figure 2 and figure 3, the solar collector instantaneous efficiency curve and solar radiation intensity curve, in most cases the testing solar collector instantaneous efficiency cannot reach the nameplate calibration value. The average collector efficiency is about 0.52 in the type of direct heating mode, while the water tank heating mode is about 0.3. Using standard [1], the mean monthly collection efficiency η_{cd} of solar collector is more accurate when doing the design calculation.

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PERFORMANCE OF FLAT-PLATE SOLAR THERMAL COLLECTING SYSTEM IN COLD PLATEAU REGIONS

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ABSTRACT

The cold plateau regions are abundant in solar energy resource. In regards to the widely used flat-plate solar thermal collecting system, this paper presents a field test of the operation of the flat-plate collectors applied in a heating project for the Construction Department of Lhasa. The performance of flat-plate solar thermal collecting system has been analyzed with consideration of the unique climate characteristics of the cold plateau regions, such as low air pressure, low temperature, strong solar radiation, etc. Results show that during normal testing periods, the maximum heat collecting efficiency of the collector is 0.62 and 0.46; the maximum effective heat gain per unit area is 710.9W/m^2 and 493.1W/m^2 ; average effective heat gain is 411.1W/m^2 and 363.7W/m^2 ; daily average heat collecting efficiency is 0.475 and 0.372.

Keywords: Cold plateau regions; solar energy; solar thermal collector; system performance.

1. INTRODUCTION

The plateau cold area has the characteristic of rich solar energy resources. Solar thermal collector (STC) systems are widely used in the heating engineering in the plateau cold area such as Lhasa. Except the characteristic of abundant solar energy in Lhasa, the climate in Lhasa also possesses the characteristic of low atmospheric pressure, low air temperature, big temperature difference between day and night, big wind speed, good air quality and so on. The unique climate characteristics in plateau cold area would lead to a particular operating effect on the heat collecting efficiency and effective heat absorption of STC systems. Many researchers have studied the thermal performance of STC and proposed relative thermal process model such as transient 3-D mathematical model for solar flat plate collectors, transient model used to evaluate the dynamic test conditions of STC, exergetic evaluation model of flat plate solar collectors and so on. However, the actual operating conditions of STC systems in the plateau cold area have not been proposed.

Baccoli Roberto [1] targets the solar collector with evacuated pipes and developed the mathematical description that supervises the functioning of the solar collector in transient conditions using the equation of the energy balance, with the aim of determining the order and architecture of the two models (Graybox Identification Model and a Dynamic Adaptive Linear Neural Network (ALNN) model). The input and output vectors of the two models are constructed, considering the measures of 4 days of solar radiation, flow mass,

environment and heat-transfer fluid temperature in the inlet and outlet from the thermal solar collector. The efficiency curves derived from the two models are detected in correspondence to the test and validation points. N. Molero Villar, etc. [2] presented a transient 3-D mathematical model for solar flat plate collectors which considers the effect of temperature on the thermal properties of the materials. The performance of a parallel tubes collectors with non-uniformity flow is analyzed and compared with experimental data from literature with good agreement. A.C. Mintsu Do Anjo etc. [3] presents numerical simulations aimed at optimizing the design of polymer flat plate solar collectors that improves solar collectors economic competitiveness compared to solar collectors which absorbers are usually made of copper or aluminum. Weiqiang Kong etc. [4] developed a numerical solutions of the mathematical transfer function model and test procedure used for dynamic solar collector test method and the results show that the transfer function method can accurately predict the fluid outlet temperature according to analysis of the errors between the predicted and measured fluid outlet temperatures for various test conditions.

Maatouk Khoukhi etc. [5] presented a rigorous theoretical approach of a flat-plate solar collector with a black absorber considering the glass cover as an absorbing-emitting media and a suitable semi-gray model is proposed for rapid calculation. The results show that the profile of the efficiency curve obtained in the present study was found to be not linear in shape due to the heat loss from the collector is a combination of convection and radiation and highly non linear.

NOMENCLATURE

Abbreviation		Subscripts	
STC	Solar thermal collector	STC	Solar thermal collector
		θ	Inclination angle of STC
		f	Fluid in the STC systems
		out	The outlet of STC systems
		in	The inlet of STC systems
Symbols		Greek Letters	
Q	Effective heat absorption (W/m^2)	ρ	Density (kg/m^3)
	Specific heat ($\text{J}/(\text{m}^3 \cdot ^\circ\text{C})$)	η	Heat collecting efficiency
C	Mass flow (m^3/h)		
M	Temperature ($^\circ\text{C}$)		
t	Effective collector area of STC		
A	systems (m^2)		
	Solar radiation intensity (W/m^2)		

Damir Dović etc. [6] presents the results of experimentally supported numerical analysis performed in order to investigate the possibilities to improve the thermal efficiency of plate solar collectors and the results show that two designs of collector without tubes, with parallel flat and corrugated absorber plates of chevron type is shown to be an effective way to increase the thermal efficiency of solar energy conversion beyond that of commercial glazed and unglazed solar water heaters. S. Farahat [7] developed an exergetic optimization of flat plate solar collectors to determine the optimal performance and design parameters of these solar to thermal energy conversion systems and a simulation program is developed for the thermal and exergetic calculations. The results of this computational program are in good agreement with the experimental measurements noted in the previous literature. Farzad Jafarkazemi etc. [8] presented a theoretical and comprehensive model for energy and exergy analysis of flat plate solar collectors through which the effect of the entire design parameters on performance can be examined. Ziqian Chen etc. [9] tested the efficiencies for the collectors at different flow rates and the efficiencies for the collectors as functions of flow rate are obtained on the basis of the measured efficiencies and the calculated efficiencies are in good agreement with the measured efficiencies.

The plateau cold area has the characteristic of rich solar energy resources. In view of the widely used heating project of flat-plate collector, the actual operating effect of the flat-plate solar collector heating source system in terms of the heating project in Construction Department of Lhasa is analyzed, based on the unique climate characteristics in plateau cold area.

2. SOLAR THERMAL COLLECTING SYSTEM

Fig. 1 presents the schematic of a solar thermal collecting system in Lhasa. This system mainly consists of solar collectors, hot water storage tank, plate heat exchangers, etc. The system has two modes of operation: (1) Direct solar heating. In this mode, Valve

V1,V2 are open and valve V3, V4 are closed; (2) Indirect solar heating. In this mode, valve V3, V4 are open and valve V1, V2 are closed.

This solar thermal collecting system consists of 96 paralleled solar thermal collectors and the effective thermal collecting area of one collector is 2.25m^2 . The solar thermal collectors is divided into 12 groups and each collector group is made up of 8 collectors. One of the 12 groups is not in operation due to failure. In actual operation, the total effective solar collecting area is 198m^2 . The installation angle θ of soalar collectors is 45° .

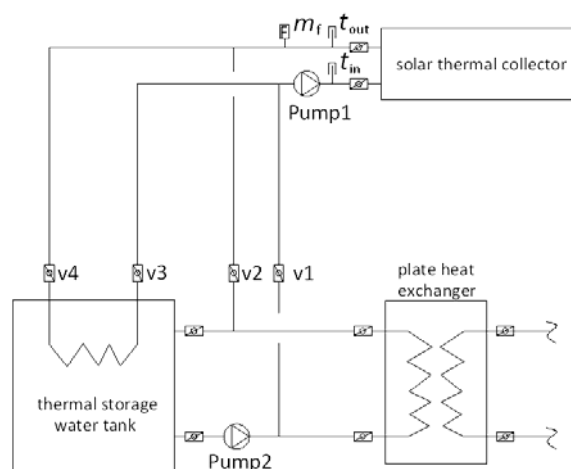


Figure1 The schematic of solarthermalcollecting systems

3. TESTING PRINCIPLES AND INSTRUMENTS

To evaluate the performance of the solar collector in cold plateau regions, field test of a solar thermal collecting system in Lhasa has been carried out to obtain operating parameters and related meteorological parameters. The heating effects and thermal performance of solar thermal collectors in cold plateau regions can thus be evaluated through calculation with these testing parameters. The effective heat gain and heat collecting efficiency are two key indices of the performance of solar thermal collectors. The effective

heat gain per unit area of the thermal collecting system $Q_{STC,\theta}$ can be calculated as:

$$Q_{STC,\theta} = \frac{\rho_f C_p m_f (t_{out} - t_{in}) / 3600}{A} \quad (1)$$

In equation(1), $Q_{STC,\theta}$ is the effective heat gain per unit area of solar collectors with an installation angle of θ , W/m²; t_{out} and t_{in} are the inlet and outlet temperature of the solar thermal collecting system, °C, acquired by the mercury thermometer; m_f is the system flow rate, m³/h, acquired by the ultrasonic flow meter.

The heat collecting efficiency η can be calculated as:

$$\eta = \frac{Q_{STC,\theta}}{I_{STC,\theta}} \quad (2)$$

In equation (2), η is the heat collecting efficiency of the collectors; $I_{STC,\theta}$ is the solar radiation intensity that the collectors accepted per unit area at an installation angle of θ , W/m², tested by solar radiation tester.

The testing apparatus used during the test and their corresponding specifications are listed in Table 1.

Table 1 The used apparatus and technical specifications

Apparatus	Measurement range	Measurement accuracy
Mercurial thermometer	0~50℃	±0.1℃
Ultrasonic flow meter	0~12m/s 0~2000W/m ²	±1%
Solar radiation survey meter	m ² ; 280~3000nm	<2%
Humiture self-recording apparatus	10~95%RH; -40~85℃	±0.5℃; ±3RH(25℃)

Fig. 2 shows the test sites and experimental apparatus. To acquire the solar radiation intensity on inclined solar thermal collectors, the solar radiation acquisition is laid on the cover glass of the solar thermal collectors directly. The solar radiation intensity of north-facing collectors at an angle of inclination of 45° has been tested.

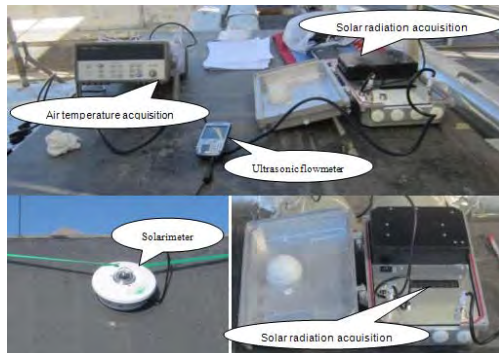


Figure2 test sites and relative used apparatus

4. RESULTS AND DISCUSSIONS

Typical meteorological characteristics of cold plateau regions are low outdoor temperature, strong solar radiation and high wind speed, etc. We took Lhasa, Tibet as an example and tested the solar radiation intensity, outdoor air temperature and performance of the solar thermal collecting system on December 25 and 27, 2012.

4.2 Outdoor meteorological characteristics of the cold plateau regions

Fig. 3 and 4 present the variation of solar radiation intensity and outdoor air temperature during the test. According to the figures, obvious correlation lies between outdoor air temperature and solar radiation intensity. The outdoor air temperature rises when the solar radiation intensity rises, and falls when the solar radiation reduces, with a certain lag. From 9:40 to 14:00, solar radiation intensity keeps increasing continuously and reaches the peak at about 14:00, with a value of 1100W/m² ~ 1200W/m². Then the solar radiation decreases, and approximates to 0 after 18:00; It can be seen in Fig. 3 there is a sudden weather change at about 15:20 on Dec. 25, the solar radiation drops, resulting in a difference between the total solar radiation of Dec.25 and 27. The total solar radiation of is 20.58 MJ/m² and 23.75 MJ/m², and the average solar radiation intensity is 686W/m² and 791.54 W/m², respectively for Dec. 25 and 27. Based on the above analysis, it can be concluded that in cold plateau regions solar radiation intensity received by collectors is high and radiation intensity changes continuously, due to the good air quality and high transparency of atmosphere.

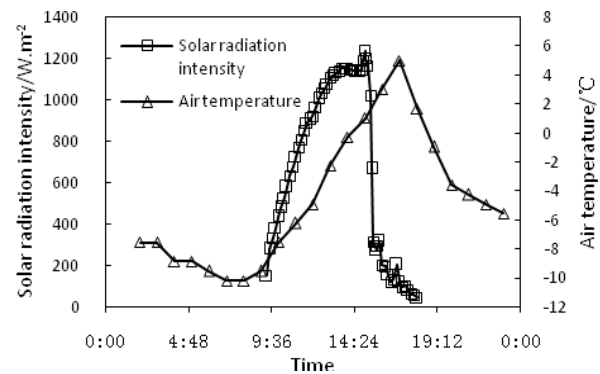


Figure3 The air temperature and solar radiation intensity on Dec. 25, 2012

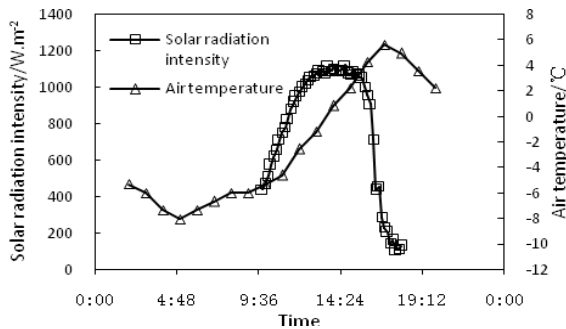


Figure4 The air temperature and solar radiation intensity on Dec. 27, 2012

Comparison of the outdoor temperature on Dec.25 and 27 show big fluctuation as well as daily range of temperature in the cold plateau region. On Dec. 25, the highest temperature, the lowest temperature and daily range of the temperature are 5.02°C, -10.19°C, 15.21°C, respectively; while on Dec. 27, the highest temperature, the lowest temperature and daily range of the temperature are 5.63°C, -8.03°C, 13.67°C, respectively; In addition, analysis shows that during the test, temperature in Lhasa in the morning remains lower than 0°C; the temperature rises above 0 °C at around 14:00. Field test found that the time when the temperature is below 0°C adds up to 20h and 16 hr respectively for Dec. 25 and 27. The analysis above shows that the outdoor temperature of Lhasa is low, and for most of the time the outdoor air temperature is below 0 °C.

4.2 Outdoor meteorological characteristics of the cold plateau regions

4.2.1 Inlet and outlet water temperature of the thermal collecting system

Fig. 5 and 6 show the inlet and outlet water temperature of the collector and their difference on Dec. 25 and 27. According to the figure, the system keeps operating continuously for 24h. Before 10:00, the collectors received little solar radiation due to low solar radiation intensity, while the inlet and outlet water temperature of the collector is high and outdoor air is low (-5.3 to ~10.2°C). As a result, the outlet water temperature of the collector is lower than that of the inlet because the heat loss of the collector is greater than the effective heat gain of the collector. After 10:00, the solar radiation increases fast and the outdoor temperature gradually increases. The inlet and outlet water temperature of the collector return normal and increase gradually. The temperature difference increases gradually as well. At about 15:10 to 15:30, the inlet and outlet temperature of the collector reach the highest value. On Dec. 25 and 27, the highest inlet and outlet water temperature are 49.3/54.7 °C and 54.5/58.9°C, respectively; then after 15:30, the inlet and outlet temperature of collector reduce rapidly due to the decrease of solar radiation and air temperature. On Dec.

27, when there was no sudden changes in weather, after 17:00, the temperature difference between the inlet and outlet water temperature of the collector approximates to 0°C, which means the collectors have lost their thermal collecting ability.

On Dec.25, the collectors are connected directly to the user-side plate heat exchangers without hot water storage tank, while on Dec. 27, the collectors are linked to the hot water storage tank. Hot water is collected into the storage tank, which is linked to the user-side plate heat exchanger. On the morning of Dec. 25 and 27, as the solar thermal collecting system operates in two different modes during the test, the piping impedance of the system changes, causing difference in system flow rate. The flow rate of the thermal collecting system on Dec. 25 and 27 are 21.95 m³/h and 16.25 m³/hr respectively. During the operation, the inlet and outlet water temperature of the collector is high. On Dec. 25, the average water inlet and outlet temperature are 44.2/46.7°C, and on Dec. 27, average water inlet and outlet temperature are 49.2/52.4 °C.

At around 15:20 on Dec. 25, the weather suddenly overcasts and solar radiation intensity drops. During 15:20~15:30, the solar radiation intensity reduces from 1020W/m² to 313W/m², and the heat gain of the collectors can no longer meet the needs of heating, then the sensible heat of high temperature water in the equipment and piping of the system are applied for heating. As a result, during 15:20~15:30, the average inlet and outlet water temperature of the collector reduces from 50.65°C to 47.8°C, with a decline rate of 0.285°C/min. Whereas, on Dec. 27, due to a sudden change in weather, the decline rate of the average water temperature of the collector is only 0.01 ~ 0.045°C/min.

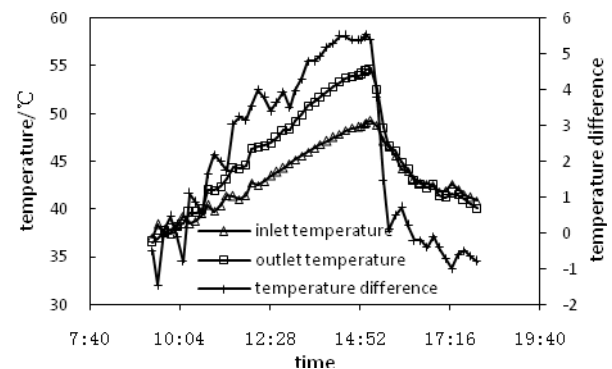


Figure5 the inlet and outlet temperature of STC on Dec. 25, 2012

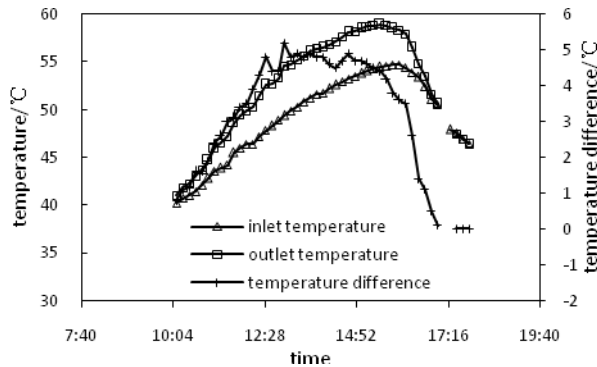


Figure 6 The inlet and outlet temperature of STC on Dec. 27, 2012

4.2.2 Effective heat gain and thermal collecting efficiency

The effective heat gain and thermal efficiency are the key factors that characterize the performance of solar thermal collectors. Fig. 7 and 8 show the variation of heat gain per unit area and thermal collecting efficiency over time on Dec. 25 and 27.

Analysis show that the effective heat gain and thermal collecting efficiency first increases and then decreases over time. Compared to the variation of solar radiation and outdoor air temperature throughout the day, the changes of the effective heat gain and thermal collecting efficiency are similar to the changing characteristics of solar radiation and outdoor temperature, which means the solar radiation intensity and outdoor temperature have significant effect on the effective heat gain and thermal collecting efficiency. Take 15:20~15:40 on Dec. 25 as an example, solar radiation decreases during this period of time, and accordingly the effective heat gain and thermal collecting efficiency decreases sharply, or even with a negative value.

On Dec. 25, before 10:10, the heat gain of the collectors is low due to the low intensity of solar radiation, while the ambient air temperature is low and the water temperature inside the collectors is high, resulting in a large amount of heat loss of the collector. The amount of heat loss is even more than the effective heat gain, which means the values of effective heat gain and collecting thermal efficiency are negative. After 16:20, due to above reasons, the effective heat gain and thermal collecting efficiency also turn to be negative. Whereas, on Dec. 27, because the weather remains in good condition without abrupt changes, the trends of effective heat gain and heat collecting efficiency of the collectors show better regularity. Due to an accident, on Dec. 27, the data of the period before 10:10 is lost, but the trend is supposed to be consistent with the data of Dec. 25. On Dec. 27, there is only a small mutation in solar radiation intensity at 17:20, resulting in negative values of the heat gain and heat collecting efficiency.

Comparison of the effective heat gain and thermal collecting efficiency of Dec. 25 and 27 shows that during normal testing periods, the maximum heat collection efficiency of the collector is 0.62 and 0.46;

the maximum effective heat gain per unit area is 710.9W/m^2 and 493.1W/m^2 ; daily average effective heat gain is 411.1W/m^2 , and 363.7W/m^2 ; daily average collection efficiency is 0.475 and 0.372, respectively for Dec. 25 and 27th.

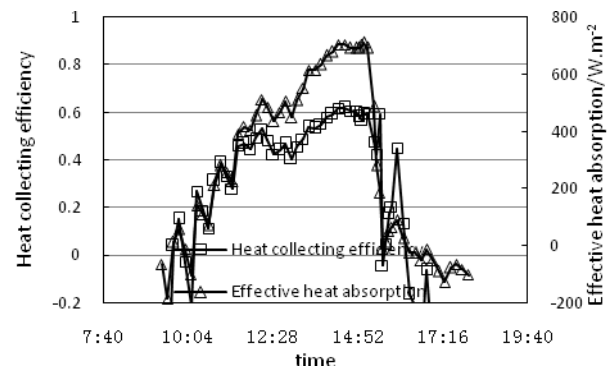


Figure 7 The heat collecting efficiency and effective heat absorption on Dec. 25, 2012

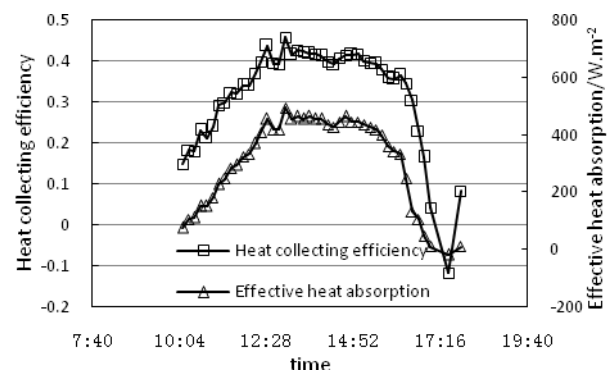


Figure 8 The heat collecting efficiency and effective heat absorption on Dec. 27, 2012

In order to analyze the comprehensive effects on effective heat gain and thermal efficiency caused by such parameters as intensity of solar radiation, outdoor air temperature, thermal collecting system flow rate and inlet and outlet water temperature of the collector, Table 2 gives the operation parameters and performance indicators during 10:20~16:10 on Dec. 25 and 27, when the performance indicators are positive. Analysis shows that the total flow rate of the solar thermal collecting system on Dec. 25 is 35.1% higher than that of Dec. 27, which means the heat transfer is more efficiency on Dec. 25; the difference between average temperature of water within the collectors and ambient air temperature is 48.8°C and 52.01°C , respectively for Dec. 25 and 27th, which means the heat loss due to temperature difference on Dec. 25 is less than that of Dec. 27. For comprehensive consideration of system flow rate and temperature difference, although the average radiation received by the collecting system is low on Dec. 25, the heat collecting efficiency of the system as well as effective heat gain per unit area are higher than that of 27th. As a conclusion, the effective heat gain and heat collecting efficiency are significantly affected by system flow rate and temperature difference between the water within the collectors and the surrounding air.

Table 2 The operatingparameter and performance index

Date	Average solar radiation intensity/W.m ⁻²	Average air temperature /°C	Water flow /m ³ .h	Average temperature of STC/°C	Heat collecting efficiency	Effective heat absorption /W.m ⁻²
Dec. 25	865	-2.45	21.95	46.35	0.475	411.1
Dec. 27	977.1	-0.91	16.25	51.10	0.372	363.7

5. CONCLUSION

To explore the heat collecting efficiency as well as the effective heat gain of the solar thermal collecting system in cold plateau regions, and thus provide guidance for solar heating project in cold plateau regions, a heating project in Lhasa is taken as an example to test the performance of the solar thermal collecting system, and to analyze the effects on system performance caused by such parameters as intensity of solar radiation, outdoor air temperature, flow rate of the thermal collecting system and inlet and outlet water temperature of the collectors. Some conclusions could be obtained as the following:

(1) Lhasa has high solar radiation intensity, and during the test, the solar radiation intensity of north-facing collectors at an inclination angle of 45° ranges from 686 to 791.54 W/m². The outdoor air temperature is low, and the daily range of temperature is big.

(2) Within normal testing periods, the maximum heat collecting efficiency of the collector is 0.62 and 0.46; The maximum effective heat gain per unit area is 710.9W/m² and 493.1W/m²; daily average effective heat gain is 411.1W/m² and 363.7W/m², and daily average thermal collecting efficiency is 0.475 and 0.372, respectively for Dec. 25 and 27.

(3) The effective heat gain and heat collecting efficiency are significantly affected by system flow rate and temperature difference between the water within the collectors and ambient air. The higher the system flow rate is and the smaller the temperature difference between the water within collectors and ambient air is, the higher the heat collecting efficiency as well as the effective heat gain will be.

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A CASE STUDY OF PERFORMANCE OF HYBRID SOLAR COLLECTOR SYSTEM AS COMPARED TO SIDE-BY-SIDE PHOTOVOLTAIC AND WATER-HEATING SYSTEMS

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ABSTRACT

An experimental side-by-side photovoltaic and solar water-heating collector system was set up at one office building of the university campus. The electricity output was used to support the exit sign lights via battery storage and changeover circuit devices for switching between the power company supply and the photovoltaic system on site. The hot water generated was consumed in selected pantries of the offices. System performance and difficulties encountered were recorded for investigation. Numerical models were then developed and successfully validated by the experimental results. The computer models were used to examine the year round system performance with the use of typical weather data of Hong Kong. Furthermore, hybrid photovoltaic/thermal (PV/T) collector systems were modeled to support the same functional areas of the building through two different design principles: (i) the emphasis on similarity in collector material usage and energy outputs, and (ii) the emphasis on similarity in overall collector area. It was found that based on the first design principle the PV/T system does not show much advantage in electricity and thermal energy outputs over the side-by-side system. However based on the second design principle, the PV/T system is much superior in energy output intensity on unit collector area basis. Comparatively, the second design principle is found more important to cope with the increasing demand of zero-carbon buildings, since the renewable energy output has to be maximized within the limited building space. This article summarizes the findings of this case study by comparing the energy performance of the related solar collector systems based on the year round simulation runs. Also reported are the key factors in the services of the related solar collector systems based on the difficulties encountered in real building application.

Keywords: Renewable energy resources, Advanced energy technologies, Energy conservation in buildings, Energy systems for power generation.

1. INTRODUCTION

A photovoltaic/thermal (PV/T) system is a combination of photovoltaic (PV) and solar thermal (ST) devices that generate both electricity and heat energy from one integrated collector system [1]. With solar cells as (part of) the thermal absorber, the hybrid design is expected to have higher energy yield from an allocated space reserved for solar application. Those with flat plate collectors meet well the low temperature air and/or water heating system requirements. They are also ideal for preheating purposes when hot fluid at higher temperature is in need. In the warm climate area, buildings have year round demands on domestic hot water rather than hot air. An office building was selected for evaluating the system performance in Hong Kong. Both the experimenting solar thermal sub-system and electrical sub-system were installed at the flat roof level of the building. The water-heating system is to serve the tap-water supply at selected pantries. The electrical sub-system is to serve a group of LED exit signs of the building. The energy performance of these systems was monitored and the field data were used to validate the self-developed computer program for long-

term energy performance evaluation. In addition, the above validated models were integrated to form a hybrid solar PV/T system model. This facilitated the use of computer simulation in the prediction of long term environmental performance of hybrid solar water-based PV/T system as compared to the side-by-side PV and ST systems.

2. PV ELECTRICAL SUB-SYSTEM

Fig. 1 shows an indicative circuit diagram of the PV sub-system. Four numbers of PV panels were used. The basic parameters of each PV panel are shown in Table 1. The charging controller regulates the PV generated electricity to the battery bank. The two lead-acid batteries have a full capacity of 6 kWh stored electric power, which is first discharged to a 24V DC to 220V AC power inverter for matching the nominal voltage of the LED exit sign lighting. Through a changeover switch, the 24-hour operated LED lights were either powered by the PV system or the main grid supply of the building depending on the availability. Listed in Table 2 are the physical parameters of the storage battery, inverter and charging controller.

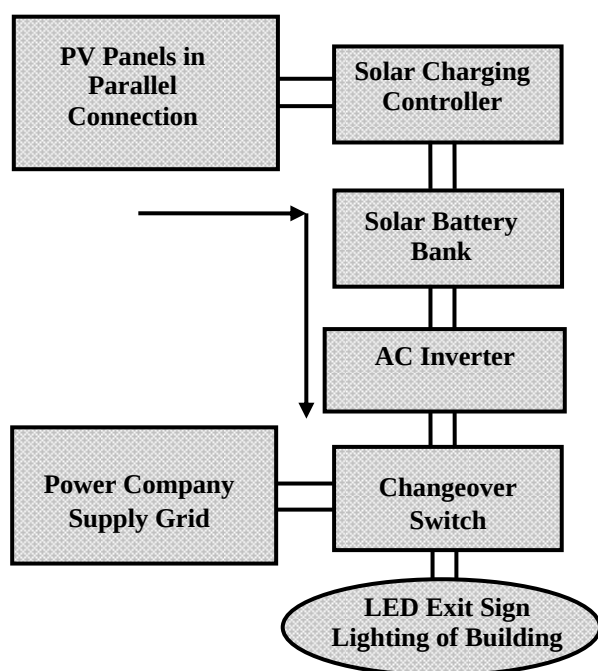


Figure 1 Brief circuit diagram of the PV sub-system

3. SOLAR THERMAL SUB-SYSTEM

Figure 2 shows an indicative schematic diagram of the solar hot water system. There are two water circuits: the solar thermosyphon circuit and the potable water circuit. The thermosyphon system operates by natural buoyant flow. Here no circulation pump is required for the flow of working fluid between the ST collector array and the heat coils in the stratified water tank. On the other hand, potable water from the water supply main flows into the insulated water tank, which serves as a heat exchanger and for hot water storage. Subsequently the hot water was delivered to the pantries for serving the occupants. At this end a re-circulation water pump is provided, so that if there is no demand of domestic hot water (DHW) for an extended period, water in the cold leg will be sent back to the water tank and the DHW pipework refilled with hot water. Four numbers of flat-plate sheet-and-type type solar water heating collectors were installed. Table 3 shows the physical parameters of the ST system.

Table 1 Physical parameters of the PV panels

Solar cell type:	pc-Si cell
Maximum power:	220W
Maximum power voltage:	29V
Maximum power current:	7.57A
Open circuit voltage:	36.4V
Short circuit current:	8.19A
Number of cells:	60 connected in series
Size of the modules:	1.643 x 0.99m (L x W)
Size of each solar cell:	0.156 x 0.156m
Cell efficiency:	13% at STC

Table 2. Physical parameter of the storage battery, inverter and charging controller

Battery: Type	lead-acid
Nominal voltage	12V
Nominal capacity	250AH
Inverter: Type	24DC to 220AC
Conversion efficiency	90%
Solar charging controller:	
Maximum current load:	45A
System voltage:	12-48V
Self-consumption current:	<20mA

Table 3. Design parameters of the thermosyphon solar hot water system

Solar collector	
Front glazing	Material: low-iron glass Effective area: 1.91m ² Thickness: 0.0032m Air gap underneath: 0.025m
Thermal absorber	Material: copper Dimension: 1.84 x 0.95m (LxW) Base plate thickness: 0.002m Number of riser tubes: 8
Back thermal insulation	Material: fibre glass Thickness: 0.05m
Water tank	
	Type: horizontal cylindrical Water storage capacity: 670L Thermal insulation: Polyurethane 0.05m thick

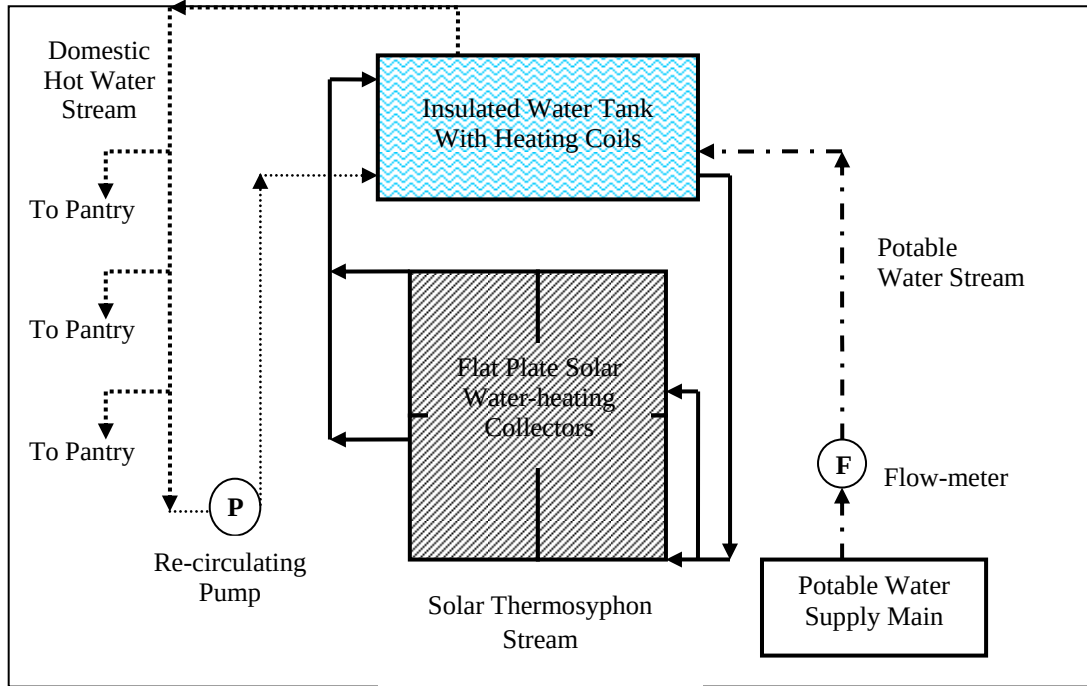


Figure 2 Brief schematic diagram of the solar water heating system

4. SIMULATION MODEL

The basic mathematical models of PV cell can be referred to the work of Raushenbach [2]. In this study a modified version by Clarke et al. [3] was adopted for its better representation of the I-V characteristic of a practical photovoltaic array. Through this, the electric current can be expressed as

$$I = I_{pv} - I_o \left[\exp \left(\frac{V + R_s I}{V_t a} \right) - 1 \right] - \frac{V + R_s I}{R_p} \quad (1)$$

where I_{pv} and I_o are respectively the photovoltaic and saturation current of the array; a is the diode ideality constant; $V_t = N_s k T/q$ is the thermal voltage of the array with N_s cells connected in series, in which k is the Boltzmann constant, T is the temperature of the p-n junction, and q is the electronic charge. At the maximum power point, the output PV power P_{max} is given by

$$P_{max} = V_{max} I_{max} = V_{max} I_{pv} - V_{max} I_o \left[\exp \left(\frac{V_{max}}{V_t a} \right) - 1 \right] \quad (2)$$

In modeling the solar thermal collector (and the PV/T collector as well), the finite difference control volume method was adopted. This is by dividing the collector into a number of sub-components each of uniform physical properties and working conditions [4]. The mathematical model of the thermosyphon water loop is based on the work of Koffi et al. [5]. The flat plate collector efficiency can be expressed as

$$\eta = F_R (\tau\alpha)_\theta - F_R U_L (T_{fi} - T_a) / I_{t\theta} \quad (3)$$

where F_R is the collector heat removal factor; $(\tau\alpha)_\theta$ is the transmittance-absorptance product; U_L is the

upward heat loss coefficient; T_{fi} and T_a are the temperatures of entering fluid and the ambient air respectively; and I_t is the incident solar irradiation.

Fig. 3 shows the comparison of the predicted PV outputs from the simulation results against the experimental data from 7am to 6pm. Fig. 4 shows the comparison of simulation results of the water temperature at the upper portion of the stratified water storage tank against the measured data. Good matching can be observed in these graphs showing good quality of the system models in use.

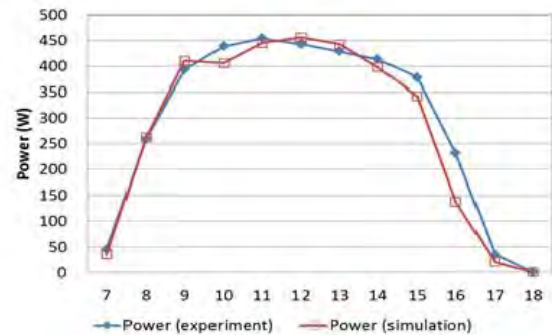


Figure 3 Electrical model validation – comparison of hourly electricity output of the PV system

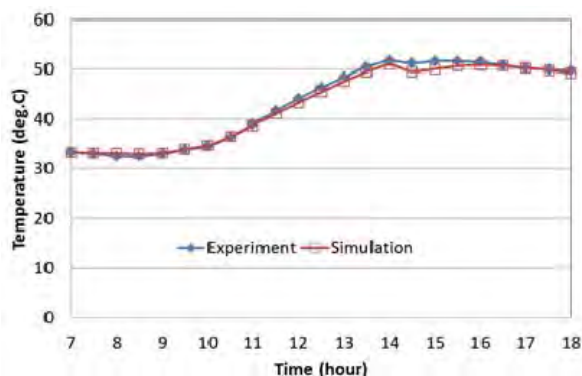


**Figure 4 Thermal model validation –
comparison of daily variation of upper layer
water temperature in storage tank**

5. ANNUAL ENERGY PERFORMANCE COMPARISON

After the successful model validation, computer simulation was conducted for the following 5 cases based on the typical weather conditions of Hong Kong:

- Case 1: a PV system consisted of 4 PV panels (with 13% cell efficiency at STC and 6.5 m² overall panel area);
- Case 2: a solar hot water system consisted of 4 ST collectors (with 7.6m² overall collector area);
- Case 3: a water-based PV/T system consisted of 4 PV/T collectors (with 7.6m² overall collector area);

Case 4: a water-based PV/T system consisted of 8 PV/T collectors (with 15.2m² overall collector area); and
Case 5: a water-based PV/T system consisted of 8 PV/T collectors using higher-quality sc-Si solar cells (with 17.8% cell efficiency at STC and 15.2m² overall collector area).

The balance-of-system (BOS) provisions of the first three cases 1 to 3 are making reference to the experimental case. The occupied floor space of 4 PV or ST collectors at the roof top is relatively the same for these three cases. As for Cases 4 and 5, the overall collector areas are relatively the same as the total collector area of the combined PV and SC systems (i.e. Cases 1 and 2 together). The corresponding changes to the BOS in these two cases are then:

- i) The number of solar battery increased from 2 to 4;
- ii) The total number of LED exit sign lights increased from 36 to 83; and
- iii) The total number of pantries in service increased from 5 to 10.

In the above water heating cases, the water tank size remained unchanged. For cases 1 and 3, the daily water consumptions are 240L, which is 37.5% of the water tank capacity. By increasing the services to 10 pantries in Cases 4 & 5, the daily water consumption is 75% of the tank capacity. Electric water heater will be operated for final water heating so that the overall system electricity saving (lighting plus water heating) can be readily compared in these cases.

Table 4 Summary of system energy performance and electricity saving in typical weather year

Case	Nos. of PV panels	Nos. of ST collectors	Nos. of PV/T collectors	Annual PV output (kWh)	Annual LED electricity saving (kWh)	Annual thermal gain (MJ)	Annual electrical saving for water heating (kWh)	Overall electricity saving (kWh)
1	4	0	0	799.5	617.7	0.0	0.0	617.7
2	0	4	0	0.0	0.0	5277	1543	1543
3	0	0	4	694.8	547.3	4795	1402	1949
4	0	0	8	1335	1076	9546	2791	3867
5	0	0	8	1834	1391	8709	2546	3937

6. RESULTS AND DISCUSSIONS

Table 4 summarizes the energy performance of the five cases based on the simulation results.

(i) Cases 1 and 2

With the 4 PV panels incorporated with lower-quality solar cells, the annual electrical output from the panels is 799.5kWh (Case 1). The monthly averaged conversion efficiency is ranged from 9.41% to 10.17%. The servicing hour of the exit sign lighting by the PV sub-system is 4540 hours in a year. This is equivalent to 617.7kWh of electricity saving. The efficiency of the electrical sub-system is therefore 77.2%, considering the BOS losses.

By using the solar hot water system with 4 ST collectors, the thermal energy saved in one year is 5277MJ (Case 2). Taking an electric heater efficiency of 95%, the electricity saving is then 1543kWh for water heating purpose.

(ii) Case 3

The annual electrical power output with 4 PV/T collectors is 694.8kWh, and the monthly averaged conversion efficiency is ranged from 8.51% to 8.81%. These are lower than in Case 1 because of the lowered electricity generating efficiency as a consequence of the reduced solar availability in the presence of the front glazing and air gap in the PV/T collector. The corresponding annual thermal energy gain is 4795MJ, which is lower than in Case 2. This is because of the PV

electricity conversion that reduces the heat absorption at the thermal absorber. For an electric heater efficiency of 95%, 1402kWh of electricity can be saved per year for water heating purpose. The overall electricity saving is then dropped by 9.8%, compared with Cases 1 & 2 together.

The similarity between Cases 1 & 2 together (side-by-side) and Case 3 is that they use similar amount of collector materials. It is found that although the electricity output of Case 3 is less than Case 1 and the thermal energy output of Case 3 is less than Case 2, the overall electricity saving per unit collector area in Case 3 is better than Cases 1 and 2 together by 102.2 kWh/m². The percent improvement in overall energy yield is 66.8% on unit collector area basis.

(iii) Case 4

The annual electrical energy conversion by the PV/T system with 8 PV/T collectors is 1335kWh and the monthly average efficiency is ranged from 7.72% to 8.71%. This is slightly lower than in Case 3 owing to the reduced ratio of collector surface area to water storage capacity. This affects the PV cooling performance when the water temperature in tank becomes higher most of the time. The supporting service-hour of LED lights is 3431 hours, which is equivalent to 1076kWh of electricity. The annual thermal energy gained by this PV/T system is 9546MJ. For an electric heater of 95% efficiency, the electricity saved for water heating by this PV/T system is 2792kWh.

The relationship between the Cases 1 & 2 together with this Case 4 is that they are occupying relatively the same flat roof space. It is found that the overall electricity saving of Case 4 is much higher than Case 1 (3867 vs. 2161kWh); the percentage improvement is 79%. This confirms the advantage of PV/T system as compared to the side-by-side systems in full utilization of the available space for solar system installation. The overall electricity saving per unit collector area in Case 4 is better than Cases 1 and 2 together by 100.1 kWh/m². The percent improvement is 65.5% on unit collector area basis. This is comparable to the 66.8% of Case 3, though the value drops slightly.

(iv) Case 5

With the use of higher-quality solar cells (STC cell efficiency 17.8% vs. 13% in Case 4), the annual electricity output can be increased to 1834 kWh, which is 37.4% better than Case 4. However, because of the improved electricity generation, the annual thermal gain is reduced to 8709MJ, which is only 91.2% of Case 4. The net improvement in annual overall electricity saving is only 1.8% compared with Case 4. The above pin-points one important consideration in the optimal design of PV/T system: to match as far as possible the actual demand profile of the occupants with the energy mix of the system outputs.

7. CONCLUSION

Numerical models of the side-by-side photovoltaic and solar water-heating collector systems have been successfully validated by the measured data obtained

from the experimental systems installed at an office building. Then the replacement of the individual PV panel and solar water heating collector models by the PV/T model made possible the energy performance evaluation of the difference individual and hybrid system designs. The findings support that the use of PV/T system is able to improve the overall energy output intensity for all the three PV/T cases studied. Based on the design principle of having similarity in collector material usage, the PV/T system gives 66.8% increase in overall electricity saving per unit collector area installed. However, the PV/T system does not have absolute advantage in electricity and thermal energy outputs over the side-by-side system. Based on the design principle of having similarity in overall collector area, however, the PV/T system is much superior in overall collector energy output and overall system energy saving. In real applications, the appropriate system design should tailor for the actual mix of hot water and electricity demands of the building. Towards the common goal of developing zero-carbon building technology, the increasing R&D activities on PV/T technology becomes an obvious trend.

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THE STUDY ON HEATING RUNNING SCHEMES OF SOLAR ENERGY COMBINED WITH MUNICIPAL HEATING NETWORK

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ABSTRACT

The research, based on combination system of solar energy and municipal heating network, studied the running scheme of combination system of solar energy and municipal heating network through ways of experiments and theoretical calculation of the mathematical model. Firstly, the research established mathematical model of heat equilibrium of the combination system of solar energy and municipal heating network; secondly, keeping the indoor heat load steady, compared the dynamic heat load provided by solar collector with that needed by heated room at the same time, determined time spans of the network stopping service and the heat load added by the network within 24 h. In the research we took a heating period in a typical room in Shenyang for example, under the conditions of low temperature floor radiant with heating index standard of 50W/ m² and 50°C of the average supplied water temperature, 40 °C of the return water temperature, calculated and verified the established mathematical model. The results include: when the heat load provided by solar collector is greater or less than the heat needed by the room, how many heat amount should be supplied by the running network and the amount of water flow within the network pipes correspondingly, and how long the network runs or stops. The final conclusions include: whether the running scheme of the combination system can save energy? the conditions of energy saving in running a combination system, and whether the system can bring environmental benefits.

Key words: solar energy; municipal heating network; combined heating; mathematical model; heat equilibrium

1. INTRODUCTION

In recent years, with the comfort requirements of the people on the indoor environment increased, energy consumption also increase. China energy consumption has been mainly fossil fuels such as oil, gas and coal, but the limited reserves of fossil fuels and over-exploitation makes fossil fuels more scarce. The combustion of fossil fuels produces large amounts of carbon dioxide, sulfur dioxide, dust and other harmful substances, causing serious pollution to the environment. Lots of people try to find a new, clean, economic renewable energy. Solar energy is one of these conditions of the energy. The previous research focuses on solar energy itself or central heating network itself, only few people study the integrated heating system. Mingzhao Zhang, based on a demonstration project in the application of solar energy - air source heat pump - boiler heating on the joint use of technology in the city of Tangshan, analysed the technical feasibility of the three joint operations and highlighted the necessity of comprehensive use for renewable energy^[1]. After an experimental study, Ye Liu and Zhiguo Zhang confirmed that the solar floor radiation heating applied in cold region was feasible, could save energy and could recover the

investment in the past four years^[2]. In 2003, Guoqing Yu chose 100 m² room as the research object, simulated solar energy heating system, analysed the initial investment of solar heating system, total operation cost and synthetic heat price, compared with conventional energy, considered the climatic conditions at the same time and the solar collector price, etc. They would affect the economic of the solar heating system^[3]. In 2010, ShiLin Qu, under different operating conditions, had carried on the theory of modeling and the experimental study, for solar energy - water source heat pump - floor radiant heating system, analyzed the supplied water temperature and the return water temperature of solar collector and floor radiant heating system, the heating performance of the system and on this basis put forward the control schemes and operation strategies of the system^[4]. Zhenhua Zhao chose the demonstrative project of solar heating as the object, designed a set of solar low-temperature hot water floor radiant heating system, introduced solar heating system designed process and the design calculated method, and carried on the theoretical and experimental

Symbols		Symbols	
A	collector area, m ²	$Q_{(w)}$	pipe network heat , w
Q	heating index, 50w/ m ²	S	area ,m ²
$J_{(T)}$	irradiation ,w / m ²	$\eta_{(cd)}$	heat collecting efficiency, 0.45-0.6
$\eta_{(L)}$	heat loss rate,0.2-0.25	G	pipeline flow of traffic, kg.

research^[5]. P.J.M artinez analysed the performance for solar radiant floor heating system and the data of performance on solar heating system in murcia residential region , compared the recorded data with the data obtained from the figure and concluded that ,the January and February in 2004,the rate of guarantee on the solar floor radiant heating system was less 20% than figure^[6]. Peng Chen presented the basic method of central heating running adjustment, analysed possible problems and provided the solutions and measures^[7].

Heating running schemes of solar energy combined with municipal network has the following three contents: when the solar radiation heating capacity can satisfy heat needed by the room , the network will stop running; when the heat load provided by solar collector is less than the heat needed by the room needed, the network will run; when the heat load provided by solar collector is greater than the room needed heat,we still can consider ether to run the network and store the rest of the quantity of heat supplied by solar collector or shut down the network , at the same time store redundant energy through heat transfered to water tank.It is selected according to the needs of our life. The heating running curves on solar energy combined with municipal network will be set up in accordance with the solar radiation energy , so as to meet requirements of our life. This paper will use the theoretical mathematical models,actual measurement and VB programming language ,to study the above mentioned problems and provide the joint running schemes. the purpose is to provide reference for integrated heating system.

2. SIMULATION CONDITIONS

2.1 The mathematical model of the heating running schemes of solar energy combined with municipal network.

we took a typical room in heating period in Shenyang for example, under the conditions of low temperature floor radiant with standard heating index of 50W/ m² and 50°C of the average supplied water temperature, 40°C of the return water temperature, established mathematical model, and assumed that solar radiation energy changed in every hour was used as unit and the indoor heat load was steady. Meanwhile,The

energy and water temperature supplied by municipal network was steady extraordinarily.

Firstly, the establishment of the mathematical model of heat equilibrium of the combination system of solar energy and municipal heating network : according to the finished experiment data, the typical room area and heating index, the ideal hypothesis of energy consumption, etc, we established the mathematical model of heat equilibrium.Based on energy conservation and numerical method, we could calculate how much heat amount provided by solar collector.

Secondly, establishing the mathematical model of municipal pipeline flow: according to the caculated heat , the basic principle of flow calculation and the ideal hypothesis of energy consumption,etc, we established the mathematical model of flow rate. Based on energy conservation and numerical method, we could learn that how much the amount of water flow supplied by the running network. $Q_{(w)}$

The establishment of the mathematical model of heat equilibrium:

$$AJ(T)\eta(cd)(1-\eta(L))+Q(w)=QS \quad (1)$$

The establishment of the mathematical model of municipal pipeline flow:

$$|0.86\{QS-AJ(T)\eta(cd)(1-\eta(L))\}/10|=G \quad (2)$$

3. RESULT AND ANALYSIS

3.1 The heating running curves of solar energy combined with municipal heating network

The operation curve was set up by VB:

The operation curve was divided into 2 parts. One was the solar operation curve.the other one was municipal network operation curve. Northern heating time was about 5 months.Taking solar radiation intensity of any sunny day in each month as the standard, we set up operation curve based on this standard, and everyday operation curve of the solar energy and the heat pipe network was derived respectively.

According to the premise conditions: room area was 100 square meters, the heating index was 50 w / m², the average supplied water temperature was 50°C, the return water temperature was about 40°C , the temperature supplied by municipal heating network was the constant , there were plenty of solar energy in the

day ,the following figure was operation curve of heating running schemes of solar energy combined with

municipal heating network set up through VB programming langua

(heat:w)

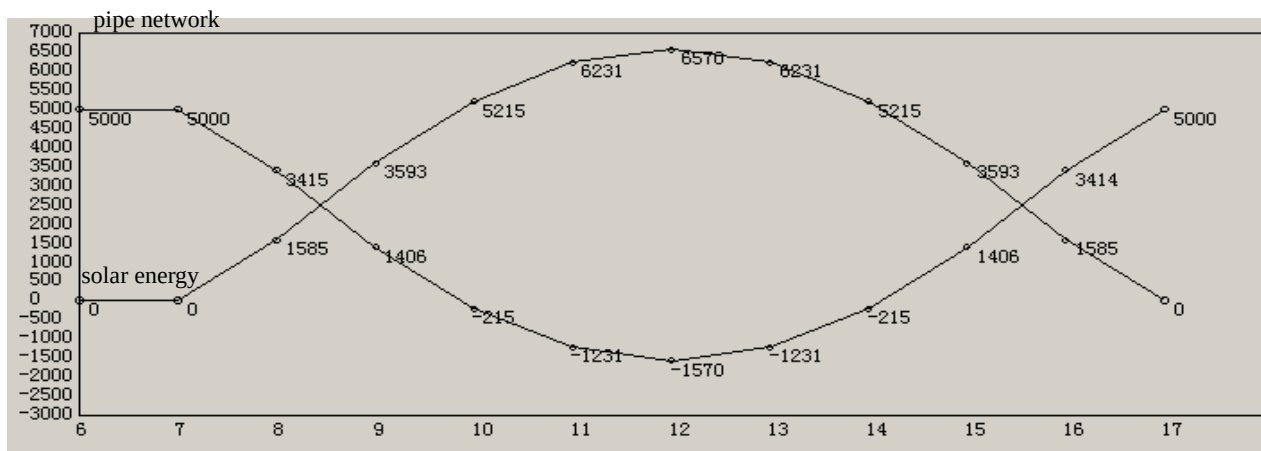


Figure.1 In November, everyday heating curve

(hourly time:h)

(heat:w)

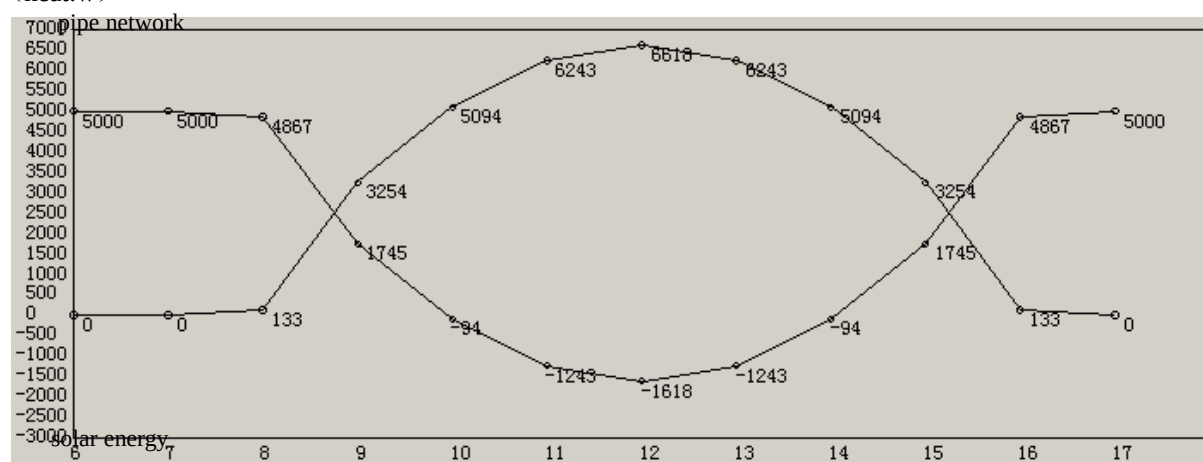


Figure.2 In December, everyday heating curve

(hourly time:h)

(heat:w)

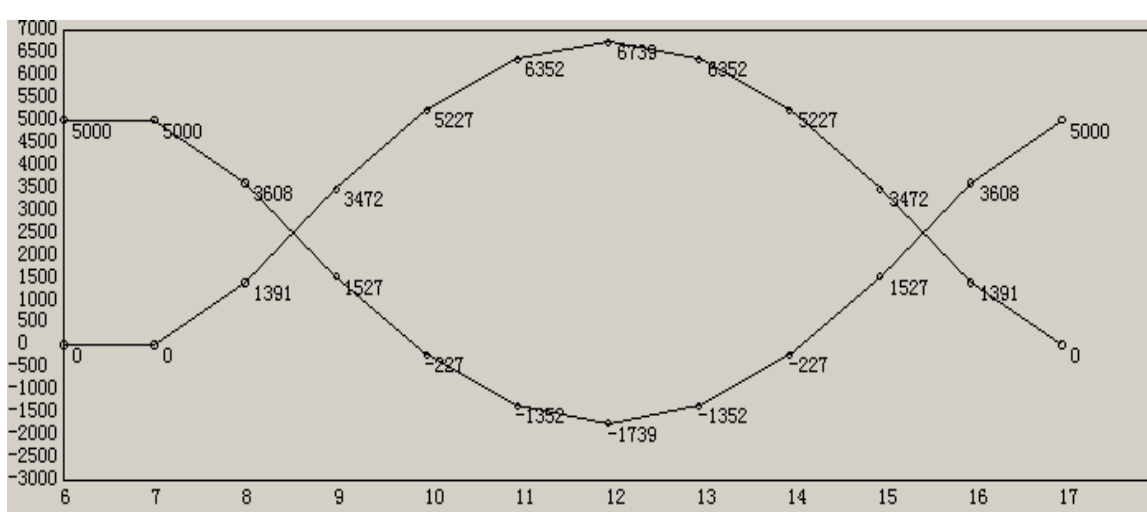


Figure.3 In January, everyday heating running curve

(hourly time:h)

(heat:w)

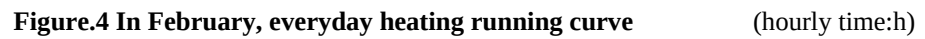


Figure.5 In March, everyday heating running curve (hourly time:h)

could bear parts of the heat load of the room .Therefore, under the conditions, the combination system can save lots of energy, and reduce burning the coals, which emits a large amounts of carbon dioxide, sulfur dioxide, and dust. So,the combination system could bring environmental benefits to us. However, for solar radiation intensity changing with the weather, we should take measures to improve the running schemes of the combination system,to facilitate running. Based on above heating curves and provided technical parameters, we provide a table of complete heating operation data that includes how to combine solar energy with municipal heating network in many different ways in a typical room in Shenyang.

Table 1 Operation table of solar energy combined with municipal heating network

operation data month	hour													su m
		6h	7h	8h	9h	10h	11h	12h	13h	14h	15h	16h	17h	
november	solar energy(w)	0	0	158 5	359 3	521 5	623 1	657 0	623 1	521 5	359 3	158 5	0	39 81 8
	the heat of network(w)	500 0	500 0	341 5	140 6	- 215	- 123 1	- 157 0	- 123 1	- 215	140 6	341 5	500 0	
	the amount of water flow(kg)	430	430	294	120	18	106	135	106	18	120	294	430	25 01 36 06 6
	solar energy(w)	0	0	133 4	325 4	509 4	624 3	661 8	624 3	509 4	325 4	133 4	0	
december	the heat of network(w)	500 0	500 0	486 7	174 5	-94	- 124 3	- 161 8	- 124 3	-94	174 5	486 7	500 0	
	the amount of water flow(kg)	430	430	419	150	8	106	139	106	8	150	419	430	27 95 37 62 4
	solar energy(w)	0	0	139 1	347 2	522 7	535 2	673 9	535 2	522 7	347 2	139 1	0	
	the heat of network(w)	500 0	500 0	360 9	152 8	- 227	- 135 2	- 173 9	- 135 2	- 227	152 8	360 9	500 0	
january	the amount of water flow(kg)	430	430	310	131	19	116	149	116	19	131	310	430	25 91 44 83 9
	solar energy(w)	0	12	214 1	415 0	575 9	678 8	715 1	678 8	575 9	415 0	214 1	12	
	the heat of network(w)	500 0	498 8	285 9	850	- 759	- 178 8	- 214 1	- 178 8	- 759	850	285 9	498 8	
	the amount of water flow(kg)	430	429	245	73	65	153	184	153	65	73	245	429	25 44 45 03 4
february	solar energy(w)	0	847	248 0	411 4	545 7	631 6	660 6	631 6	545 7	411 4	248 0	847	
	the heat of network(w)	500 0	415 3	252 0	886	- 457	- 131 6	- 160 6	- 131 6	- 457	886	252 0	415 3	
	the amount of water flow(kg)	430	357	217	76	39	113	138	113	39	76	217	357	21 72
	solar energy(w)	0	847	248 0	411 4	545 7	631 6	660 6	631 6	545 7	411 4	248 0	847	
march	the heat of network(w)	500 0	415 3	252 0	886	- 457	- 131 6	- 160 6	- 131 6	- 457	886	252 0	415 3	
	the amount of water flow(kg)	430	357	217	76	39	113	138	113	39	76	217	357	21 72
	solar energy(w)	0	847	248 0	411 4	545 7	631 6	660 6	631 6	545 7	411 4	248 0	847	
	the heat of network(w)	500 0	415 3	252 0	886	- 457	- 131 6	- 160 6	- 131 6	- 457	886	252 0	415 3	

By the above table, we observed that heating running times of solar energy combined with municipal heating network ranged from 7 o'clock to 17 o'clock in a day in different month.

Under normal circumstances, regardless of the heat load provided by solar collector was greater or less than the heat needed by the room, we may run network all the time. However, when the heat load supplied by solar collector was greater than the heat needed by the room, the network may be stopped ,to store energy through transferring heat to water tank. In most cases, we should send the excess heat to municipal network, in order to facilitate running and management.

Certainly, solar radiation intensity could not meet the heat needed by the room,the network must run.

According to the operation table of solar energy combined with municipal heating network, when the heat of network was positive, it meant that network should run and give off heat ; when the heat of network was negative, it suggested that network may stop or run and absorb heat. Taking any sunny day in the first month for example, at 9 o 'clock, the output of heat supplied by the running network was 1528 w and the amount of water flow was 131 kg correspondingly; At 12 o 'clock, the heat of network was – 1739w and the amount of water flow within the network pipes was 149 kg correspondingly ,with excess heat send to municipal network. Then, in this day, solar energy could provided 37624w that accounted for thirty-one percent of the heat load in the room .Of course, the heat of network may stop 5 hours, for the heat of network was negative from

10 o'clock to 14 o'clock. Based on the above table and analysis, we could infer operation condition in a heating period in a typical room in Shenyang.

4 .CONCLUSIONS

In this study we can see that the technical difficulty of the demand on running scheme of the combination system is great . Combined system flow distribution must be proper to meet the requirements of indoor comfort. Clearly ,the requirements of automatic control system and staff technology level also are high ,in order to better running a system . If the above conditions are met, energy saving in running a combination system must be able to achieve. Meanwhile, with solar energy used and burning the coal reduced, The air and the environment around us will be improved. It means the combination system could bring environmental benefits .More importantly, the operation table of solar energy combined with municipal heating network can improve the running scheme of the combination system and guide us how to run.

However, the installation of combined system not only takes up a lot of space, but also affects local landscape . In addition ,there is certain difficulty in the installation process of system . Whether the running scheme of the combination system can be adopted, we need to consider many factors and study further .

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INFLUENCE OF INTERMITTENT OPERATION ON GROUND THERMAL BALANCE IN GCHP SYSTEM

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ABSTRACT

The ground thermal balance is an important factor to ensure the GCHP operation with high efficient for a long time. Taking an office building with GCHP system as the research object, on the condition of inputting the load characteristics which takes load intensity and operation time as physical parameters, this paper establishes GHE heat transfer model including the single U-tube 3-D model adopted to the short time-span coupled response and the tube-group line heat source model adopted to the long time-span coupled response. The different intermittent and continuous operation modes are simulated and analyzed, and the numerical calculation results are compared with the practical operating data. Ground temperature distribution of different intermittent and continuous operation modes is put forward. Taking the excess ground temperature and the GHE outlet temperature as the judgment standard, the influence on the ground thermal balance of the different intermittent and continuous operation modes is get.

Keywords: GCHP System, intermittent operating, thermal imbalance, GHE, heat capacity, excess temperature

1. NTRODUCTION

In recent years, ground thermal balance has become the biggest hindrance of the popularization and application of GCHP systems [1-4].

Shang [5] and Li [6] studied the various influence factors on the ground heat transfer of GCHP system in intermittent operation by experiment. Li [7] and Zhao [8] researched the influence of intermittent operation on the ground temperature recovery characteristics by the simulation of the ground temperature distribution around the GHE in intermittent operation of GCHP system. At present, few of researches about the influence of intermittent operation on the ground thermal balance are conducted. Taking an office building with GCHP system as the research object, the simulation of different operation modes is carried out to analyze the influence of intermittent operation on the ground thermal balance in this paper.

2. MATERIALS AND METHOD

2.1 Materials

The project described in this paper is an office building in hot summer and cold winter zone in Chongqing, China. It has 4 floors with 3273.17m² in covered area and 1369.61m² in air conditioning area. Using GCHP system, it has 48 vertical single U-tube GHEs in buried tube region. Each borehole is 0.13m in

diameter, 80m in depth and the distance between the boreholes is 4m. Using PE pipe, the U-tube is 0.032m in outside diameter, 0.026m in inside diameter and the distance between the tubes is 0.05m. Table 1 shows the physical parameters of various materials.

DEST program is available to establish the building load model and calculate the hourly load. The geometric model and computational mesh of the GHE heat transfer model are generated in Gambit. Fluent (CFD) is used for numerical simulation, where the load input of different operation modes is achieved by compiling C language function in UDF-a secondary development function of Fluent.

2.2 Building load model

Fig. 1 shows the building load model. The calculated one-year hourly load and the chosen one-week (July 1st-July 7th) hourly load are shown in Fig. 2 and 3. By analyzing the load characteristics we can see that: for a year, the maximum cooling/heating load is 258.78/130.20kW, the annual cooling/heating load accumulation is 219240/52094kWh, and the imbalance rate reaches 4.2:1. Long-term operation may cause excess cooling/heating quantity accumulation which results in the ground thermal imbalance; for a week, there is load in all 7 days and the peak and valley load intensity respectively appears in the 3rd-4th days and the 5th-7th days; for a day, there is load only during 7:00-20:00, except that the load is 0.

Nomenclature

Abbreviation		Subscripts	
GCHP	Ground coupled heat pump system	x,y,z	direction of axis
GHE	Ground heat exchanger	Greek Letters	
EER	the cooling/heating energy efficiency ratio of SWHP unit (using “+” when in cooling period and “-”when in heating period) (W/W)		
Symbols		μ_t	the turbulent viscosity(Pa·s)
C_p	specific heat(J/kg·K)	G_k	the generation item of turbulence
$\dot{Q}$	inner heat source(W/m ³)		kinetic energy k
		τ	shear stress on the water surface (Pa)
		ρ	density (kg/m ³)
		λ	thermal conductivity(W/m·K)

1) 3-D model

2.3 GHE heat transfer model

In this paper, the single U-tube 3-D model is used when taking one week as the operation cycle, while the tube-group line heat source mode for one year. The following assumptions are made to enable a simplified analysis: (1)The influence of the surface temperature fluctuations on the ground temperature is ignored. (2) The physical parameters of the ground are constant in the whole heat transfer process.(3)Only consider the pure thermal conduction. (4)Ignore moisture transfer influence on the heat transfer.

2.3.1 physical model

Fig. 4 and 5 show the geometric model and computational mesh of the GHE heat transfer model.

Geometric model: 3-D model is a cylinder with 3m in diameter and 85m in height which including heat-transfer fluid, U-tube, backfill and ground, and the bottom bend of the U-tube is instead of a small cylinder because of its weak heat transfer. Line heat source mode is a 24m×20m 2-D plane, which is a symmetric quarter of the buried tube region, and the inlet/outlet tubes and boreholes are considered as circular section, while the ground as rectangular section. Using hexahedral element, the total node number of the 3-D model and the line heat source model are respectively 92068 and 151913.

2.3.2 mathematical model

(1)Governing equations

The heat transfer model of the ground is unsteady, whose governing equations can be expressed in the formulas (1)-(2).

The ground is without inner heat source, so $\dot{Q}=0$.

The realizable $k-\varepsilon$ two-equation model is adopted to the U-tube GHE due to the turbulence of water, whose governing equations can be expressed in the formulas (3)-(7).

They can be expressed in the formulas (8)-(9). Where:

$$C_\mu = 0.09, C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92, \sigma_k = 1.0,$$

$$\sigma_\varepsilon = 1.3, \sigma_T = 0.9, Pr_t = 0.85.$$

(2)Boundary and initial conditions

The inlet is set as velocity-inlet: its velocity is 0.43m/s calculated by the flow of a U-tube 0.82m³/h; its temperature is input by UDF, where the inlet temperature t_1 is calculated by the formula (10) (using “+” when in cooling period and “-”when heating) according to the hourly load Q in DEST and the outlet temperature t_2 achieved by iteration. The outlet is set as pressure-outlet: its pressure is 0; its temperature is achieved by iteration. The top surface is taken as no-slip solid wall and uses the third boundary condition: the free stream temperature is 32°C (outdoor calculated temperature); the surface heat transfer coefficient of the boundary is 1.2 W/m²·K. The bottom and distant surface are taken as no-slip solid wall and use the first boundary condition with constant temperature which is the initial ground temperature 21°C. The borehole and U-tube wall are settled as no-slip solid and coupled wall with 0 and 3mm in each thickness, so that the coupled heat transfer simulation can be performed between the heat-transfer fluid, U-tube, backfill and ground. For the initial condition, the initial temperature of the ground and backfill is set as the annual average ground temperature 21°C, and the velocity and pressure of all the boundaries are 0.

2) Line heat source model

The U-tube wall is set as no-slip solid and coupled wall with 3mm in thickness so as to carry out the coupled heat transfer simulation between the U-tube and backfill. Using the second boundary condition, the unit length heat flux is input by UDF, where it is achieved by the hourly load Q in DEST. The borehole wall is set as no-slip solid and coupled wall without thickness so as to carry out the coupled heat transfer simulation between the backfill and ground. The distant surface is taken as no-slip solid wall and uses the first boundary condition with constant temperature which is the initial ground temperature 21°C. The symmetric surface is settled as symmetry. The initial condition is the same as the 3-D model.

(3)Operation mode and time step

Different intermittent and continuous operation modes are designed: operation mode 2 and 3 are different in system start-stop rate (respectively 1.0 and 0.7), operation mode 3 and 4 in cooling/heating load

intensity (intermittent operation respectively at the time of peak and valley load intensity), and operation mode 5 and 6 in cooling/heating load imbalance rate (respectively 4.2:1 and 1:1).

Operation mode 1: in a week, continuous operation for 7 days. Operation mode 2: in a week, continuous operation for 6 days and intermittent operation for 1 day. Operation mode 3: in a week, continuous operation for 5 days and intermittent operation for 2 days. Operation mode 4: in a week, continuous operation for 2 days, intermittent operation for 2 days and then continuous operation for 3 days. Operation mode 5: in a year, continuous operation during cooling period Jun.-Sep., intermittent operation during Oct.-Nov., continuous operation during heating period Dec.-Feb., and then intermittent operation during Mar.-May. Operation mode 6: in a year, some time intermittent operation during cooling period Jun.-Sep., intermittent operation during Oct.-Nov., continuous operation during heating period Dec.-Feb., and then intermittent operation during Mar.-May.

3. RESULTS AND DISCUSSION

3.1 Model validation

Fig. 6 shows the comparison of simulation and experiment results from the thermal response experiment (the condition is that continuous operation for 5 days and intermittent operation for 2 days, and the measuring point is at z axis 20 m of the outlet tube). As shown in Fig. 6, there is a little difference between the simulation and experiment results, but the basic trend is consistent. Among them, the 3-D model simulation results are about 2°C higher than the experiment results. The reasons of difference may be as follows: ①the model simplified assumptions do not match with the actual situation; ②the meteorological condition of the simulation is set as the local typical meteorological parameters; ③the outlet tube has been made the insulation treatment in the experiment. A larger difference exists between the line heat source mode simulation and the experiment results, but it becomes smaller and smaller, then reaches about 2°C after a week. Except for the above reasons, the line heat source model is mainly adopted to the long time-span simulation of several months or years.

3.2 Excess ground temperature

The excess ground temperature is defined as the difference between the current ground temperature and the initial ground temperature. The smaller it is, the better the ground temperature recovery is.

For the one-week simulation in the 3-D model, the ground temperature recovery is analyzed by comparing the excess ground temperature in different depths, different radiuses and different operation modes, whose results can be seen in Table 2, Table 3 and Fig. 7.

As shown in Table 2, Table 3: along the depth direction, the excess ground temperature is basically unchanged with a little decrease, the same as different

radiuses and operation modes. It is because the ground temperature below the depth of 5m is almost constant in a year, only the region near ground surface can be affected by the surface temperature fluctuations except the region below a certain depth. Along the radius direction, the excess ground temperature gradually reduces from the near wall to the distant surface, similarly as different depths and operation modes. It is due to the effect of delay and attenuation when the temperature wave transmitting. When starting/stopping the system, the flowing/still state of the heat-transfer fluid will orderly lead to the temperature change of the U-tube, backfill and ground, the thermal response of the near wall is quick and with large amplitude and oppositely as the distant surface so as to transfer the cooling/heating quantity to the distant surface. As shown in Fig. 7: for different operation modes, the excess ground temperature in operation mode 1 is the largest and reaches 2.99°C after a week; in operation mode 2, it reduces a little when intermittent operation in the 7th day and reaches 1.75°C after a week; in operation mode 3, it reduces much when intermittent operation in the 6th and 7th day and reaches 1.18°C after a week; in operation mode 4, it reduces the most when intermittent operation in the 3rd and 4th day, in the following 3 days it rises a little but is still lower than the continuous operation, finally reaches 2.42°C after a week. By analyzing we can see that the ground temperature recovery is better in intermittent operation than in continuous operation, and the smaller the system start-stop rate and the larger the intermittent cooling/heating load intensity are, the better the ground temperature recovery is.

As shown in Table 4, Table 5: along the X-axis ($x=0, 8, 16, 20\text{m}$) direction, the excess ground temperature reduces from the tube-group center to the distant surface when the system releases heat during the summer cooling and its recovery period, and it continuously reduces or reduces firstly then rises slightly when the system absorbs heat during the winter heating and its recovery period. The heat transfer condition of the tube-group center is the worst, and the more far from the tube-group, the better. The reason is that the relatively slow ground thermal diffusion makes the cooling/heating quantity can not adequately and timely transfer to the distant surface but accumulate in the tube-group which is a heat retainer. As shown in Fig. 8: for different operation modes, the excess ground temperature in operation mode 5 is higher, which reaches 6.49°C before the heating period (early winter) and 5.10°C before the cooling period (early summer); in operation mode 6, it is smaller, which reaches 1.47°C before the heating period (early winter) and -0.18°C before the cooling period (early summer). By analyzing we can see that the ground temperature recovery is better in intermittent operation than in continuous operation, and the smaller the cooling/heating load imbalance rate by intermittent operation, the better the ground temperature recovery is.

3.3 Ghe outlet temperature

The GHE outlet temperature of the 3-D model can be directly monitored, as shown in Fig. 9. While the GHE outlet temperature of the line heat source model only can be achieved by the monitored borehole wall temperature [9,10], as shown in Fig. 10.

As shown in Fig. 9 and 10: the curves of GHE outlet temperature and excess ground temperature have the similar trend [9-10]. The ground temperature change can be directly reflected in the heat-transfer fluid temperature change, so the GHE outlet temperature change law can be achieved by studying the ground temperature recovery characteristics. In operation modes 1-6 the GHE outlet temperature respectively reach 24.22°C, 22.79°C, 22.20°C, 23.55°C, 24.65°C and 20.81°C after a week or a year. By analyzing we can see that the GHE outlet temperature recovery is better in intermittent operation than in continuous operation, and the smaller the system start-stop rate, the larger the intermittent cooling/heating load intensity, and the smaller the cooling/heating load imbalance rate by intermittent operation are, the better the GHE outlet temperature recovery is. Thereby, it can ensure the GCHP system operating with high efficiency for a long time by reducing the operation energy consumption of system, improving the operation efficiency of heat pump unit, and increasing the EER of system.

4. CONCLUSIONS

In this paper, three conclusions were obtained as followed:

- (1) The line heat source model can be instead of the 3-D model in the long time-span heat transfer process, and the simulated results are similar.
- (2) The ground thermal balance is better in intermittent operation than in continuous operation, and the smaller the system start-stop rate, the larger the intermittent cooling/heating load intensity, and the smaller the cooling/heating load imbalance rate by intermittent operation are, the better it is.
- (3) Intermittent operation can improve the ground temperature recovery and enhance the GHE heat transfer capacity, so as to ensure the GCHP system operating with high efficiency for a long time.

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Investigation And Analysis Of Operation Energy Consumption Of Green Office Buildings In Shanghai

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ABSTRACT

Although the amount of green building is increasing, research of their operation performance is few. In order to research the actual energy saving rate, an on-site investigation about energy consumption of 8 green office buildings and 7 traditional office buildings has been carried out in this paper. After comparison, the average of energy consumption of green office buildings and traditional office buildings is about $1.028\text{GJ/m}^2\cdot\text{a}$ and $1.196\text{GJ/m}^2\cdot\text{a}$ respectively, the energy saving rate is about 14.09%, but few green office buildings show bad energy consumption performance. On the other side, the energy consumption of air conditioning system in green office buildings and traditional office buildings is about $0.378\text{GJ/m}^2\cdot\text{a}$ and $0.364\text{GJ/m}^2\cdot\text{a}$ respectively, which illuminate that the energy efficiency of air conditioning system in green office buildings in Shanghai is bad currently.

Keywords: Energy consumption investigation, Green building, Office building, Energy saving rate.

1. INTRODUCTION

The energy consumption of building in China mainly come from air conditioning, heating and lighting, which accounts for about 20.7% of the total energy consumption in our country[1], however, it accounts for 40% of the total energy consumption in western country[2]. Thus, with the development of our country, the energy consumption of building would be larger.

The paper [3] analyzed the status of energy consumption of public buildings in Shanghai based on investigation of some office buildings, hotels and multi-functional complex buildings, and concluded that the energy consumption of building in Shanghai is large, which have good energy saving potential. Through the comparison of energy consumption of public buildings between Shanghai and foreign countries which are in the same climate in paper [4], the author found that the energy consumption of building in Shanghai is larger than in foreign countries. In paper [5], the author investigated three buildings which are green building certified, and found that their operation energy consumption is low, but their occupancy rate are low either, thus, it is difficult to determine the energy efficiency of green buildings.

In order to decrease the energy consumption of building and the affect to environment of it along with, many countries are promote the work of green building, such as ASHRAE came up with the ASHRAE90.1-2007, and became the standard of America; our country began to implement the 《Design standard of energy saving in public building》 (GB50189-2005) in 2005.

There are also some evaluate standards of green building in many countries, such as the LEED of America, DGNB of Germany, BREEAM of England, GREEN MARK of Singapore and 《Evaluate standard of green building》 (GB/T50378-2006) in our country.

However, as some statistics of American show that, it isn't all of the green buildings are energy saving. Since the operation data of green building is short of in our country, and in order to analysis the energy efficiency of green buildings, this paper make a comparison of energy consumption between 8 green office buildings and 7 traditional office buildings based on the operation data from investigation (note: traditional office building in this paper stand for building which haven't won any green building certified), and as the basic work of energy saving of green buildings.

2. METHODOLOGY AND PROCESS OF INVESTIGATION

The investigation began in 2011, buildings to be investigated are characteristic of green building certified, LEED certified or doesn't award any green building certified, which include 8green office buildings and 7 traditional office buildings. Questionnaire survey is the main pattern of the investigation, through consulting property management to obtain information of building, and do some measurement based on the preliminary investigation.

The content of this investigation include structural style, building area, built time, grade of green building, occupancy rate, lighting, air conditioning system,

heating and cooling source, power of pump and monthly energy consumption of recent years. Basic

information of buildings from investigation can be seen in Table1.

Table 1 Basic information of buildings

Building	Grade	Built time	Occupancy rate (%)	Area (million m ²)
GA	LEED platinum	2008	100	4.0
GB	Green building three star	2008	100	6.0
GC	LEED gold	2010	100	3.3
GD	LEED certified	1997	100	3.6
GE	LEED certified	1997	80	3.5
GF	LEED gold	2012	98	4.8
GG	LEED gold	2011	100	2.4
GH	Green building two star	2009	90	7.5
TA	Traditional building	2005	100	4.2
TB	Traditional building	1998	90	7.1
TC	Traditional building	1996	100	1.9
TD	Traditional building	1995	100	4.5
TE	Traditional building	2000	100	5.1
TF	Traditional building	1996	100	2.4
TG	Traditional building	1994	100	1.1

3. ANALYSIS OF INVESTIGATE RESULT

energy saving technologies and materials, which can be seen in the table 2.

3.1 Energy saving technologies used in green buildings

In order to decrease the energy consumption of building, lots of green buildings have applied many

Table 2 Energy saving technologies and materials applied in the green office buildings (“√”stands for applied)

Technology	GA	GB	GC	GD	GE	GF	GG	GH
Ground source heat pump			√	√				
Water source heat pump	√				√			√
Air source heat pump		√				√		
Solar PV							√	
Low-e glass	√					√	√	√
Energy saving light	√	√	√	√		√	√	√
Sunshading	√		√					√
Envelop insulation	√			√			√	
Green roof	√	√	√	√	√	√	√	√
Natural ventilation	√	√	√	√	√	√	√	√
VAV	√			√			√	

3.2 Comparison of total energy consumption of buildings

In order to compare the energy consumption of different buildings, this paper convert all the energy consumption unit to kgce/m²·a. The convert coefficient can be seen in Table3.

Table 3 Energy convert coefficient

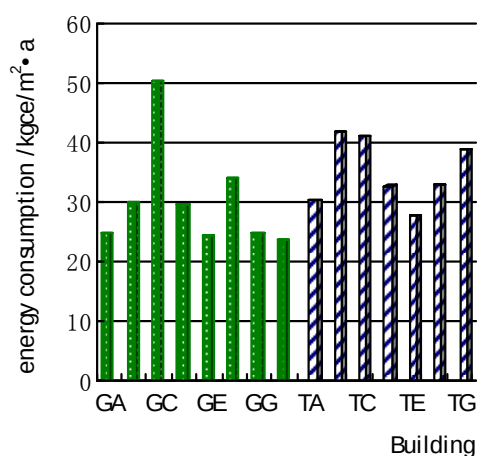
Energy	Convert coefficient
Electricity	10256kJ/(kWh)
Natural gas	38972kJ/m ³
Fuel oil	42697kJ/kg

Through calculation, the total energy consumption (including the energy consumption of heating) of 8

green office buildings and 7 traditional office buildings can be seen in the Table4.

Table4 Comparison of total energy consumption of buildings

Building	Cooling source	Heating source	Energy consumption			
			Electricity	Natural gas	Fuel oil	Total
			(kWh/m ² ·a)	(m ³ /m ² ·a)	(kg/m ² ·a)	(GJ/m ² ·a)
GA	Compression chiller + WSHP	WSHP	105.7	—	—	0.85
GB	ASHP	ASHP	100.4	—	—	1.03
GC	Screw chiller +GSHP	GSHP+ gas fired boiler	140.6	6.4	—	1.69
GD	Compression chiller	GSHP	98.8	—	—	1.01
GE	Screw chiller	WSHP	82.1	—	—	0.84
GF	ASHP	ASHP	113.25	—	—	1.16
GG	Screw chiller	Gas fired boiler	67.3	3.4	—	0.82
GH	Screw chiller +WSHP	WSHP	78.9	—	—	0.81
TA	ASHP	ASHP	101.0	—	—	1.04
TB	ASHP	ASHP	139.3	—	—	1.43
TC	ASHP	ASHP	137.0	—	—	1.41
TD	Screw chiller	Oil fired boiler	104.1	—	1.1	1.11
TE	Screw chiller	Oil fired boiler	85.0	—	1.6	0.94
TF	Split	Split	109.7	—	—	1.12
TG	Split	Split	129.4	—	—	1.33



■ green building ▨ traditional building

Figure.1 Comparison of total energy consumption of buildings

From the table 4, we can get the average energy consumption of green office buildings and traditional office buildings in Shanghai is 1.028GJ/m²·a and 1.196GJ/m²·a, respectively, and the energy saving rate is 14.09%, which can illuminate that the energy

consumption of green office buildings in Shanghai is lower. However, the energy consumption of green building GC is bigger, which is 1.44 times of traditional office buildings

3.3 Comparison of the energy consumption of air conditioning

Since the energy consumption of air conditioning accounts for big parts of the total energy consumption, and in order to make a better comparison of energy consumption between green office buildings and traditional office buildings, this paper also make a comparison of the energy consumption of air conditioning system.

Although there is a requirement of classified measurement should be used in green buildings, there are still many buildings doesn't reach this requirement. According to the methodology of paper [2], this paper treats the lowest energy consumption of two month in all year round as the benchmark load, and the up load of other months as the energy consumption of air conditioning. According to this methodology, energy consumption of air conditioning system can be calculated and seen in the Figure2.

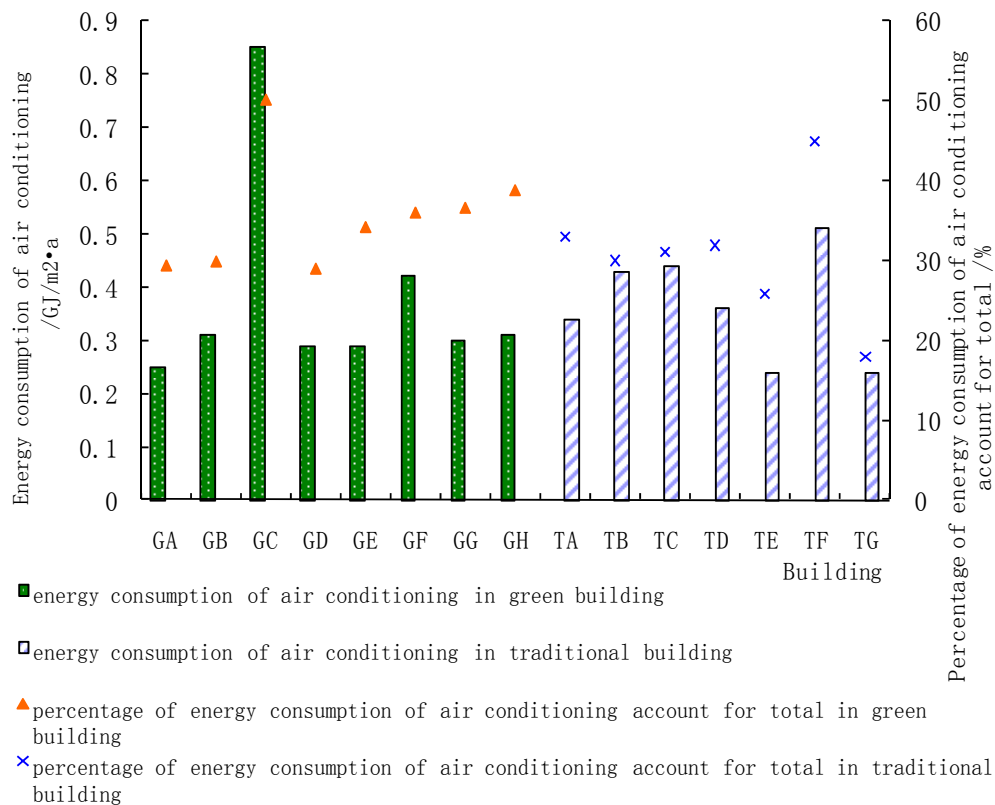


Figure 2 Energy consumption of air conditioning system in green office building and traditional office building

Through calculation, the average energy consumption of air conditioning system in green office building in Shanghai is $0.378\text{GJ/m}^2\cdot\text{a}$, accounts for about 35.51% of the total energy consumption; and the average energy consumption of air conditioning system in traditional office building is $0.364\text{GJ/m}^2\cdot\text{a}$, accounts for about 30.67% of the total energy consumption. Thus, the energy efficiency of air conditioning system in green office buildings in Shanghai is bad currently.

4. COCLUSIONS

In order to analysis the energy efficiency of green office buildings in Shanghai, this paper investigated 8 green office buildings and 7 traditional office buildings, and obtained their operation energy consumption data. Through calculation, the average of energy consumption of green office buildings and traditional office buildings is $1.028\text{GJ/m}^2\cdot\text{a}$ and $1.196\text{GJ/m}^2\cdot\text{a}$ respectively, the energy saving rate is about 14.09%, but few green office buildings show bad energy efficiency.

On the other side, the average of energy consumption of air conditioning system in green office buildings and traditional office buildings is $0.378\text{GJ/m}^2\cdot\text{a}$ and $0.364\text{GJ/m}^2\cdot\text{a}$, which account for about 35.51% and 30.67% of total energy consumption respectively. Thus, the energy efficiency of air conditioning system in green office buildings in Shanghai is bad currently.

ACKNOWLEDGEMENTS

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PILOT PROJECT OF ZERO ENERGY BUILDING IN THE UNIVERSITY OF TOKYO

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ABSTRACT

The concept of a zero energy building (ZEB) is getting popular all over the world, more so in Japan. The Ministry of Economy Trade and Industry (METI) is supporting to develop and promote the idea of net zero energy buildings. Associated with METI, New Energy and Industrial Technology Development Organisation (NEDO) initiated a project to subsidise some experimental pilot projects of ZEB. The University of Tokyo designed and constructed ZEB buildings in its campus on receiving their subsidy.

The present paper describes the latest technologies applied in this experimental pilot project. Some of these techniques discussed are the double skin cladding system, low exergy radiation cooling/ heating panel system, LED lighting system, AI control based integrated building operation system etc. We also present how the technology is integrated to enhance energy use performance, while reporting the actual data on the energy performance.

Keywords: zero energy building, heat pump, double skin, radiation cooling/heating, energy management system

1. INTRODUCTION

The idea of zero energy building is getting to be popular concept in various countries all over the world. Japan is not exceptional; The Ministry of Economy, Trade and Industry of Japanese Government is supportive to develop and diffuse the idea of net zero energy building. The research committee organized by METI [1] defined net zero energy building as;

“A building that consumes zero or almost zero energy on annual net primary energy consumption basis by enhancing the energy efficiency performance of buildings and equipment, Area Energy Network, using the renewable energy, etc. on site”.

Associated with METI, New Energy and Industrial Technology Development Organization (hereafter NEDO) initiated the project to subsidize to the experimental pilot projects of ZEB. The University of Tokyo had the opportunity to get the subsidy for design and construction of ZEB buildings in its campus. This paper introduces technologies applied in this experimental pilot project. It also presents how technologies are integrated to enhance higher energy use performance.

2. PROJECT DESCRIPTION

2.1 Outline of the building

The building was constructed in Komaba campus, one of the three major campuses of the University of Tokyo. The building is called as “The building for

innovative education”. It has five floors and one underground level floor. The total floor area of the building is 4,477m². It includes several studios and convention rooms such as halls and meeting rooms. Figure 1 shows the exterior view of this building.



Figure.1 Exterior view of the building

2.2 Design team for ZEB

Initially, in the conceptual design of the building ZEB idea was not included. However, The University of Tokyo, as a client, decided to introduce the idea of ZEB after the first version of conceptual design has been completed. Consequently, design process of ZEB is a series of change orders from client side.

The design team of the ZEB is inter-disciplinary and was composed of architects, building service

engineers, users and administrators, contractors and researchers in Institute of Industrial Science, the University of Tokyo. They are committed to ZEB technologies and transformed the building from an ordinary building to ZEB through the design development process.

Various design issues were mutually interdependent, and thus, change. A modification relating to some issue drew bunches of changes of other issues. The interdependency required tough design management process. Consequently, design process includes not only success but also failures.

Members of the team held periodic meetings where information has been exchanged and the design has been discussed and coordinated based on design documents made by team members. Thus, the periodic meeting provided the opportunity for design integration

2.3 A list of technologies applied

We conceived, developed and introduced many technologies respecting on specific contextual

requirements of the building. Figure 2 presents technologies installed and integrated in the building, which includes;

- 1) smart double skin cladding system by adaptively movable louvers
- 2) heat pump system by ground source/water utilization
- 3) radiant cooling/heating panel system
- 4) thermal mass utilization
- 5) natural ventilation by chimney effect
- 6) desiccant dehumidifier by exhaust heat from heat pump
- 7) LED lighting system
- 8) Potable water saving including rain water utilization
- 9) PV system
- 10) integrated building operation system by AI based control

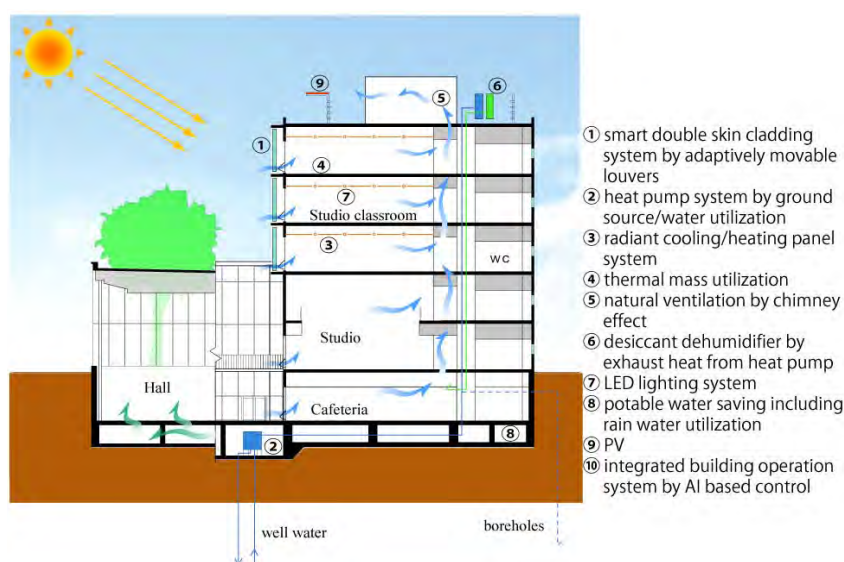


Figure 2 Technologies applied to the building

3. TECHNOLOGIES DEVELOPED AND APPLIED

This chapter presents outline of technologies developed and used in the building through collaboration of the interdisciplinary design team.

3.1 Heat Source System in this Building

Figure 3 shows heat source diagram in this building. There are three heat pumps, which are two air source heat pumps (ASHP-1A and ASHP-1B) and one water source heat pump (GSHP). The property of each heat pump is described in Table 1. The water source heat pump (GSHP) utilizes ground heat and ground water as heat source.

For the ground heat source, we drilled 10 boreholes with 100m depth and installed a single type

of U tube made of cross-linked poly-ethylene. Its heat source ability roughly estimates to 50 kW. For the ground water source, we installed two pairs of wells: one for lifting water and the other for injection. They lifted water at a rate of about 100 l/min. This translated to a roughly estimated heat rate extracted from this ground water system to be 70 kW with the assumption of a temperature difference of 5°C.

The water table is 10m below the ground surface. The wells measured 150 mm in diameter, 20 m in depth, and were protected by inner casing made of vinyl chloride. The capacity of lifted pump was 1.9kW per one well. ASHP-1A is a heat-recovery type heat pump which can supply chilled water (5-10 °C) and hot water (55-70 °C) simultaneously. This hot water was used to recover absorption material used in desiccant outdoor

dedicated system. Chilled water for cooling and heated water for heating from all heat pumps are usually stored in thermal storage tank which is installed under the basement and the mass of which is 300 ton. Furthermore, cooling heat from ground water can directly be used for radiation cooling panel without using a heat pump.

Table 1 Properties of Heat Pump

Heat Pump	GSHP	ASHP-1A	ASHP-1B
Location	Basement Floor	Rooftop	Rooftop
Heating ability (Rated Value)	121kW	99 kW	132 kW
COP (Rated Value)	4.4	3.0	3.0

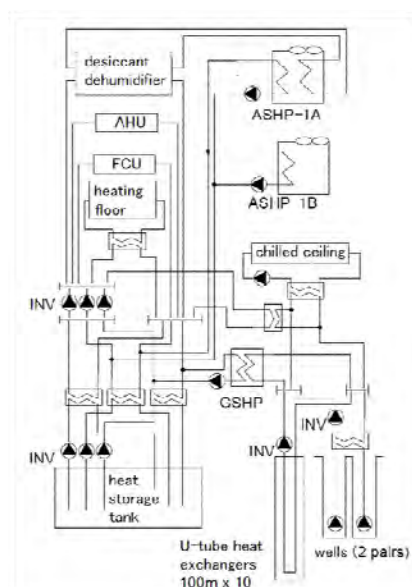


Figure 3 Heat Source System

3.2 Smart double skin cladding system with adaptively movable louvers

The west and east elevations of the building have considerable thermal load as well as an opportunity to utilize natural ventilation, natural light and heat radiation from sunrays in winter. In order to solve the contradict requirements on selective control of light heat and airflows, design team invented and developed smart double skin cladding system with adaptively movable louvers. In case of fixed windows, we installed triple glazing sashes with inlet for natural ventilation together with external sun shading.

In case of openable windows, we invented a double skin system with movable louvers and applied; this system consists of movable vertical louvers and their frames. Each vertical louver unit is composed of double layered slat panels and a double glazed panel. (Bearing on that, this double skin is installed in the west and east elevation, slats are installed vertically.) Fig. 5 presents detailed mechanism and the working principle of the double skin system in summer and in winter. The unit can be rolled around vertical axis. Consequently, as

it is shown in Figure.4, both slat panels and glazed panel can face outside/inside selectively depending on outside conditions. In addition, white coloured side of slat and black coloured side of slat are also turned around, thus, both sides of slat can face outside selectively as well. Exterior of this louver is shown in Fig.4.

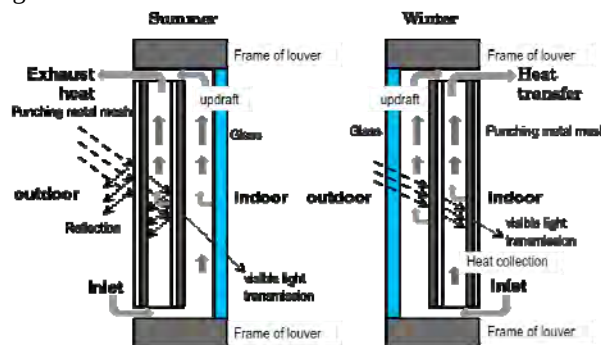


Figure 4 Mechanism of smart double skin cladding system with adaptively movable louvers

3.3 Radiant cooling/heating panel system

Conventional air conditioning method changes the temperature of air to warm or cool human's body. This indirect heating and cooling via air includes waste and inefficiency in energy use. Contrarily, heating and cooling by radiant panel is a method to warm or cool the occupant directly. It does not need the energy for bringing air and for activating fans. It also avoids needless draft as well as upgrades comfort. The installed ceiling panels have function of radiant panels. Figure 5 shows the structure of cooling/heating panel and ceiling finish.



Figure 5 the structure of cooling/heating panel and ceiling finish

However, it is expected that the ability of this radiant panel become short at the peak load or at the start up. Thus authors combined Fan Coil Unit with radiant panel as shown in Figure 6.

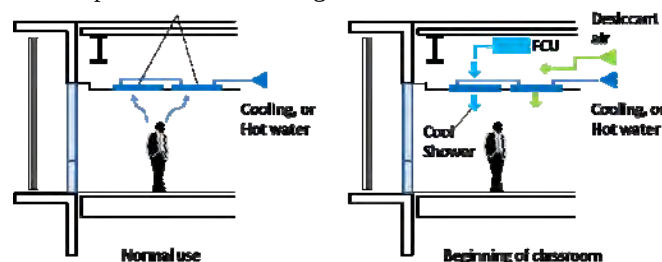


Figure 6 Combination of Fan Coil Unit with Radiant Panel

3.4 LED lighting system and integrated system

Needless to say, the use of LED has advantage in energy conservation compared to conventional lighting. This building aimed at more higher efficiency than simple, stand alone use of LED.

The building embodies integrated control system that maximizes use of natural light and facilitates complementary use of artificial light by LED with dimmer function.

3.5 PV (photovoltaic) system References

We installed a photovoltaic system of 300 m² of thin-type multi-crystalline Si solar modules on the roof of the building as shown in Figure 7. The maximum capacity of generation reaches to 30kw which is about 10% of anticipated maximum peak load demand of electricity in summer. Annual total electric power generated by installed PV is estimated as 300GJ on primary energy basis. Generated electricity is designed to be used in the building (i.e. PV here is separated from network). The building also embodies continual monitoring system to measure site basis real efficiency of the PV.



Figure 7 Thin type of multicrystalline Si PV panel

3.6 Integrated building operation system by AI based control

The building embodies energy monitoring system based on the idea of information embedded building [3]; the monitoring system consists of sensors of energy

4. ENERGY PERFORMANCE OF THE TARGET BUILDING

Figure 9 shows annual energy consumption of this building in primary energy. While averaged energy consumption of buildings in the University of Tokyo except for buildings for experiment is 1830 MJ/m²a, 508kWh/m²a, the measured energy consumption of this

use, temperature, humidity, indoor air quality, illumination intensity etc. The real time basis data is transferred from sensors as digital data to server which is located off-site.

Furthermore, building, lighting, and air conditioning are totally integrated and controlled through artificial intelligence (AI). Figure 8 shows outline of the integrated system. Previously mentioned louvers in double skin cladding system moves based on evaluation of the data from pyranometer; In case pyranometer would suggest that it is raining or cloudy, louvers moves to introduce maximum quantity of day light to inside. In case it would suggest bright and no direct sun rays, louvers also settle to the direction to introduce natural day light as much as possible. In case of sunny day with direct sun rays that could increase cooling load such as in summer, the louvers are settled to prevent inclusion of sun rays to inside of the building. In case indoor space is warmed artificially, louvers are settled to proactively introduce heat gains. Together with and harmonised with adaptively controlled louver system, artificial light by LED is controlled. Motion sensors works to switch off and on LED lighting which is controlled to assure sufficient illumination intensity based on illuminometer.

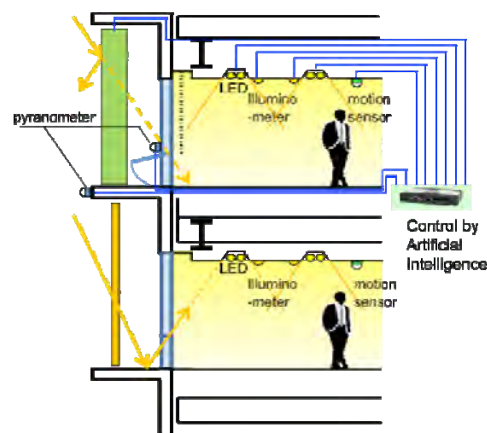


Figure 8 Integrated use of natural light and LED

building is 666MJ/m²a, 185kWh/m²a, which corresponds to 37% of the averaged value. With considering the energy production of PV power, the net energy consumption is 612MJ/m²a, 170 kWh/m²a, which corresponds to 34% of the averaged value. Although this is not still net zero energy in the strict sense, we can see energy performance of this building is very high.

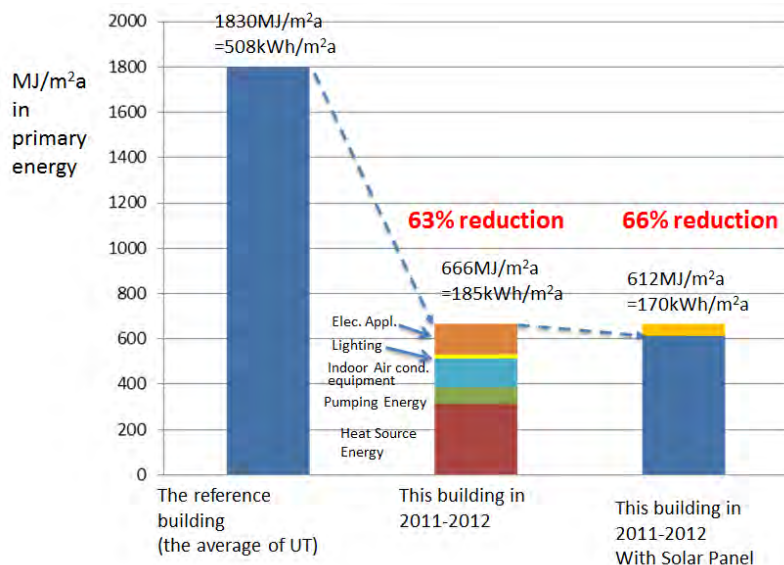


Figure 9 Annual energy performance of this building (2011. ~ 2012 Oct.)

5. CONCLUDING COMMENT

We developed brand new technologies and methods that would enhance the prospect of fructifying the idea of a real ZEB. Although we conceived these specific to the current building context, we strongly believe that, these could be successfully extended productively, to any other ZEB oriented project. In addition, the design integration method we adopted can be a valuable forerunning prototype for all the subsequent projects of this genre.

The building was fully completed in May 2011. An actual energy performance evaluation of the building against the pre-design ZEB targets is completed fully. We operate and maintain the building with a commitment to cut-down the energy levels. The research on this building is ongoing. We would utilize every opportunity to report and disseminate the knowledge obtained from this exercise and to propagate the message of ZEB.

ACKNOWLEDGEMENTS

The authors appreciated all collaborators of the reported experimental building projects and funding agencies including NEDO for their support and contribution.

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LOW CARBON ECONOMY-ISSUES IN BUILDING ENCLOSURE TECHNOLOGY SEEN FROM THE BUILDING PHYSICS POINT OF VIEW

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ABSTRACT

In the previous conferences on Low Carbon Economy in Nanjing, 2011 and the ISHVAC 2011 in Shanghai we have highlighted that the developments in building physics brought us to better understanding the differences between design of building enclosures in mixed climate of Shanghai or Nanjing and those in the cold and hot regions of the country. In mixed climate both the switchable thermal mass and switchable thermal resistances are needed. To this end we postulated integration of heating, cooling and ventilation and building enclosure (Bomberg, 2010). Since this Low Carbon Economy conference is focused on thermal storage it is appropriate to discuss hygrothermal upgrade of building enclosure as proposed for this climatic region. We do so in two parts; Part 1 reviewing progress in North America (US and Canada), Part 2 dealing with the Chinese situation.

The paper is presented here in a short version; a full version will be submitted as technical note to the Frontiers of Research in Architecture.

Keywords: Thermal upgrade of existing buildings, Renewable energy resources, Phase change, Energy conservation in buildings

1.INTRODUCTION

There is a lot of good will towards achieving a carbon neutral future. American Institute of Architects placed the date 2030. Yet experts realize that new construction is only a small fragment of the problem because the majority of highly inefficient buildings of today will be with us well beyond these deadlines.

We agree with a UN report that stated:

“The good news is we have got a huge source of alternative energy all around us. It is called energy conservation, and it is the lowest cost new source of energy that we have at hand. Since 1973 alone, improvements in energy efficiency resulted in a 50% reduction of our daily energy use, which is the same as discovering 25 extra million barrels of oil equivalent every single day. Clearly saving energy is like finding it”.

Based on past successful NA (North America includes USA and Canada) programs, we know that to achieve the required change it we must use the systems' approach. Every building, old or new, needs to be treated as a system in which every component is piece of a large puzzle. Quick fix of one component in the building envelope, at best, may not achieve

enough, and at worst, may cause damage to other components of the building. This multidisciplinary approach requires advice from practitioners with different expertise because the green value of this approach is determined by the resulting building performance under service conditions not by the perception that the actions were green.

For convenience, in this review we will consider the following topics.

- A measure for energy use in building
- Progress in the fenestration
- Progress in sustainable housing
- Progress in thermal upgrade of buildings
- Methods of thermal upgrade for these buildings
- Can the sustainable performance be standardized?
- Sustainability need to be the basis for all decisions
- Innovation and product quality management cannot be separated
- Bridge the gap between Universities and industrial Innovation

PART 1: THE SITUATION IN NORTH AMERICA

We restrict the discussion to NA as this is the only area all three.

2. A MEASURE FOR ENERGY USE IN BUILDING

Traditionally, each component of the building was designed and evaluated separately. The opaque parts of building enclosure: roof, walls and floor in contact with ground, all have specified limits for the heat transmission coefficient, so called U-value (an inverse of thermal resistance, R-value used in NA). Today, in cold climate we use, R10 (Rsi 1.75) for basement walls, R20 (Rsi 3.5) for the above ground walls and R30 (Rsi 5.3) for roofs. In typical design, however, one does not consider many effects such as thermal bridges, fenestration, air and moisture transfer etc and therefore using the total energy in relation to the surface area of the building is a better energy measure.

Average energy use of NA commercial buildings in 1990 was 315 kWh /m². Since 1990 energy use in the commercial buildings steadily declined, reaching 250 kWh /m² in 2002 (Finch et al., 2012). Note, however, that this is equivalent to the energy use of commercial buildings in 1920. So, a masonry building without insulation built nearly one hundred years ago consumed as much energy as a shiny, glass-clad building constructed recently! This of course says nothing about the increased functions that current modern buildings are now fulfilling compared with what was needed in 1920.

In contemporary office buildings, the office equipment and computers use 10% of total energy but lighting uses 28%. For a layman who believes that 1920 people also used lights (and probably less efficiently than now) and who knows that today we have thermal insulation, thermal mass, air barriers and many other energy saving measures it is difficult to understand why we do not use at least 50% less energy than used in the 1920's. Furthermore, we do not realize that from 1950 to 2000 there has been a 3-fold increase in emissions attributable to buildings.

3. PROGRESS IN THE FENESTRATION AREA

It is estimated that windows account for about 10 % of the total building energy use. The high negative impact of windows in buildings contrasts with the impressive progress in window technology that has taken place in recent years. Research issues that are being currently resolved are heat transfer through low

conductance spacers and insulated frame systems, and the installation of high performance windows in walls.

Today, 95% of the average current windows in NA are double-glazed, and 50% have a low-E coating and an average R-value of 2.2 (ft² °F hr)/Btu. This is a significant progress from 1973 when single pane windows with R 1.2 (ft² °F hr)/Btu were used. Comparing energy costs for a typical two-story house with no windows with a heating cost of \$1,000, use of double glazed windows with low-E coating would increase the bill by \$120, double glazed but without low-E coating windows increase the bill by \$218 and single glazed windows w/storm on the north side increase the yearly energy cost by \$310. This simple comparison stresses the impact that today's technology involving low emissivity coatings and gas fill is already an acceptable solution when the window to wall ratio is a reasonable percentage of the whole; however it is not acceptable to build a building with R 2.2 exteriors that are all glass!

To achieve the goal of making windows energy suppliers in cold climates one must reduce the U value (improve the thermal performance) of the glazed part sufficiently that the solar gain can exceed the heat loss. This implies that windows with a high R-value and a moderate solar heat gain coefficient (SHGC) should be used. In hot climates the situation is reversed but the static control of heat flow is not suitable because of large temperature oscillations between day and night. So in hot climates, much like in the mixed climate, the static control needs to be replaced by a dynamic control. This is the technology for the near future. In the more distant future, for windows to become the real energy supplier they may be integrated with photovoltaic films.

Those highly insulated window systems will include such technologies as aerogels, vacuum glazing, low-E, gas fill and improved thermally resistive frames. Using dynamic control involves operation of an optical switch from high to low transmission; and suitable technologies include electro chromic (that requires wiring) windows, thermo chromic (reacting to glass temperature), photo chromic (reacting to sunlight intensity), etc. Field tests with electro chromic windows were started in 1999. The 2030 window is expected to be R10 (ft² °F hr)/Btu (Rsi 1.7) and will provide a net winter gain and 80% saving in cooling.

4. PROGRESS IN SUSTAINABLE HOUSING

In a 2007 US survey, 41% of all respondents defined a green building as one with a specified percentage of green materials and 46 % stated that green buildings must follow criteria established by a national program. While the term "green buildings" has become a buzzword for environmentally driven impulses, it is not

equivalent with the term sustainability. The term “high performance building” was defined in the US Energy Policy Act of 2005 as follows:

The term “high performance building” means a building that integrates and optimizes all major high-performance building attributes, including energy efficiency, durability, life cycle performance, and occupant productivity.

Yet, for the technical people, every building must address all those aspects of performance. Unless the statement of building objectives states otherwise, every building should be treated as the high performance building.

4.1 Evaluation of systems not materials

It is important to place emphasis on performance of the whole building and built assemblies instead of emphasis on materials used in those assemblies. We realize that dealing with materials is easier. Building codes and standards always ascribe a specific function to a specific material because this is the only way that a prescriptive code can work. We have WRB, water vapor retarder, air barrier, thermal barrier (fire), rain-screen, where functions and materials are mentally coupled. But one material, e.g. closed-cell spray foam, can also function as insulation, a rain-screen, WRB, water vapor retarder or air barrier.

The outcome of an architectural design is modified by interactions between different materials and the trades involved in installing them in an assembly. Architectural design and construction are holistic processes that involve highly specialized people – how should they collaborate during this process? This aspect of design is so important that we stress the importance on mock-up testing and the continuous QA and/or commissioning as separate activities in the construction process. This is to ensure that the design concept is buildable and that all the trades learn what they must do for ensuring the required building performance.

So far we have established that the future belongs to green buildings. Let us now review the critical components of the complex called, for simplicity, the “green value” of a building.

4.2 Key components of “green value” during the design and construction of buildings

The list of typical components of “green value” includes:

1. Design for durability
2. Design for energy efficiency and efficient use of materials
 - a. Separation between ventilation/air distribution and heating/ cooling systems
 - b. Integrated hot water systems

- c. Review and if possible increased use of day lighting
- d. Improved indoor environment (with view to occupant health and productivity)
- e. Greater design flexibility, i.e., lower costs associated with changing space configurations
- f. Design from cradle to cradle, i.e., considering if the existing components can be used in the next generation buildings and re-use of materials in building enclosure systems
3. Design to the passive house standards (design must be efficient enough to justify economic use of renewable resources)
 - a. Use of adequate tools for performance evaluation of building enclosures
 - b. Improve control of inter-zonal and interstitial air flows
4. Building and testing the mock-up of building enclosures for commercial buildings
5. Using the commissioning process as a part of the design and construction process (from design objectives, through the construction and post occupancy tests)
 - a. Trouble shooting study of design drawings is the first step in commissioning
 - b. Testing air flows during construction
 - c. Testing air quality of occupied space (after occupancy on working level)

Number one in the green value complex is the issue of durability (long-term performance). If one can extend the service life of a building, say by 20% longer than a typical construction, one reduces the annual costs. One can postpone the use of materials needed for replacement of the building so that materials and energy can be used for other productive means and for rehabilitation of existing building stock. In this process the direct savings on replacement materials and energy provides a multiple effect that may be compared to injecting 3 to 5 times more money into the local economy.

The second most important consideration is to increase all possible passive energy efficiency measures that lead to energy savings before progressing to (active) energy generating measures. The passive measures that are often neglected, even though they offer the most value for the invested money, are (Pope⁶):

1. Simple building shape and mass placement integrated with air re-distribution measures (saves capital and energy)
2. Increase airtightness (cost little, saves lots)
3. Increase insulation values and reduce thermal bridging (costs but saves energy)
4. Reduce window area (saves capital and saves operating cost)

⁶ Steve Pope– presentation from Nat. Resource Council of Canada

5. Improve windows and wall-window interface (costs capital but saves operating cost)

The above may serve to produce a list of additional requirements, as follows:

- 1) Use solar preconditioning for ventilation air
- 2) Use geothermal preconditioning for ventilation air
- 3) Use solar hot water heating

These are requirements are followed by other, more complex technical measures ranked for their energy saving potential:

- radiant or hydronic heating and cooling
- heat and energy recovery ventilators
- diagnostics for malfunctioning of a system or component in service
- dedicated ventilation system (direct outdoor air and air redistributing system)

We also have positive experience with

- brushless DC motors
- small centrifugal compressors
- micro-channel heat exchangers

In short we stress that highly insulated, airtight building enclosures are a pre-requisite for the next generation of HVAC and lighting systems. Those measures can dramatically reduce thermal loads which in turn would encourage the use of the distributed HVAC. Locally operated small exhaust fans in each room are becoming popular in some European countries.

One can observe a trend for building enclosures to become multi-functional. Dynamic envelopes can be used to pre-heat or pre-cool indoor air; and by using filters and dehumidifiers these enclosures can modify the indoor environment. Advances in glass and window technology permits use of increased day lighting. With reduced thermal loads several technologies previously discarded in research are becoming more economically viable. Those include effects of thermal mass and phase change materials – even though these effects are

climate dependant – they are coming back as significant improvements in the technology mix.

Effectively we can distinguish three stages in design of low energy buildings:

- 1) Follow the passive house standards (development started by Prof. Feist and Adamson more than two decades ago)
- 2) Incorporate geothermal or water tank storage and solar thermal measures (the thrust of this conference)
- 3) Decide what fraction of PV (photovoltaic) will be constructed initially (the remainder will be done later during the service life of the building)

5.PROGRESS IN THERMAL UPGRADE OF BUILDINGS

Since the first energy crises in 1973 both Germany and Scandinavia continued slow but steady progress in the energy conservation, while NA quickly abandoned the energy considerations for almost two decades before NA re-joined Europe in this endeavor. Perhaps the most important technology developed in Germany is interior rehabilitation of masonry with capillary active thermal insulations. Interior thermal insulation systems are less effective than the exterior systems and will not reduce the effect of thermal bridges. Yet they may be successfully used and in many instances are often recommended as a complement to the exterior insulation.

Figure 1 presents the concept of capillary active insulation, as presented by Haeupl et al (1994?). The layer shown on the interior side of the masonry wall is shown to have a high capillarity and redistributes the moisture condensed on the back side of this thermal insulation through the larger volume of the material. In this way it can perform both the function of thermal and moisture protection.

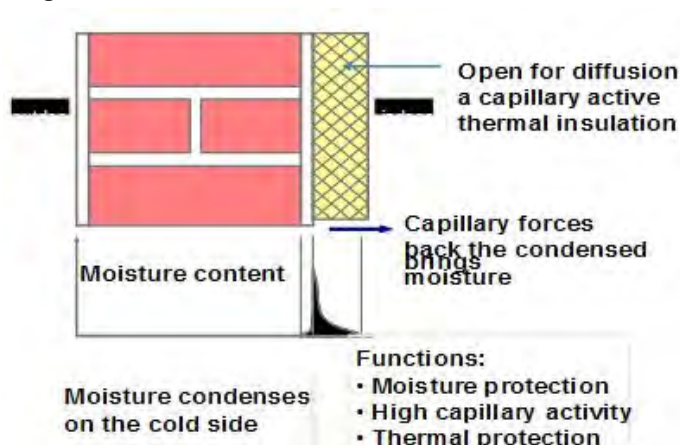


Figure 1: Principles of the capillary active insulation

To understand this concept we go back to a technique developed by Bomberg and Shirtcliffe (1978) that uses a sealed moist specimen placed in a heat flow

meter apparatus to be exposed to the specified thermal gradient. Fig. 2 shows moisture content redistribution in the same type of experiment performed by Kumaran

(1986) on the glass fiber and sprayed cellulose fiber specimens and gamma-ray spectrometer (Kumaran and Bomberg, 1985).

Thermal gradient drives moisture in vapor phase toward the cold side. This process goes fast in the glass fiber insulation (Fig. 2(a)), and moisture is carried to cold plate of the equipment already in 3 to 4 h and the final dynamic equilibrium develops in the span of 9 to 10 h.

The process of redistribution goes much slower in the cellulose fiber insulation (Fig. 2(b)). At the period 3 to 4 h a significant fraction of moisture is still

on the warm side of the specimen and the final dynamic equilibrium develops in the period of 17 to 18 h. We use term “dynamic equilibrium” because it is a composite effect of two counteracting each the other transport processes. Water vapor is driven toward the cold side and condensing there comes to a very high level of relative humidity at which liquid flow develops. As liquid viscosity is not strongly depended on temperature and the liquid flow tries to equalize moisture content i.e., drives moisture backward. How far backward moisture can be pushed depends on amount of moisture and liquid flow component in the total moisture conductivity.

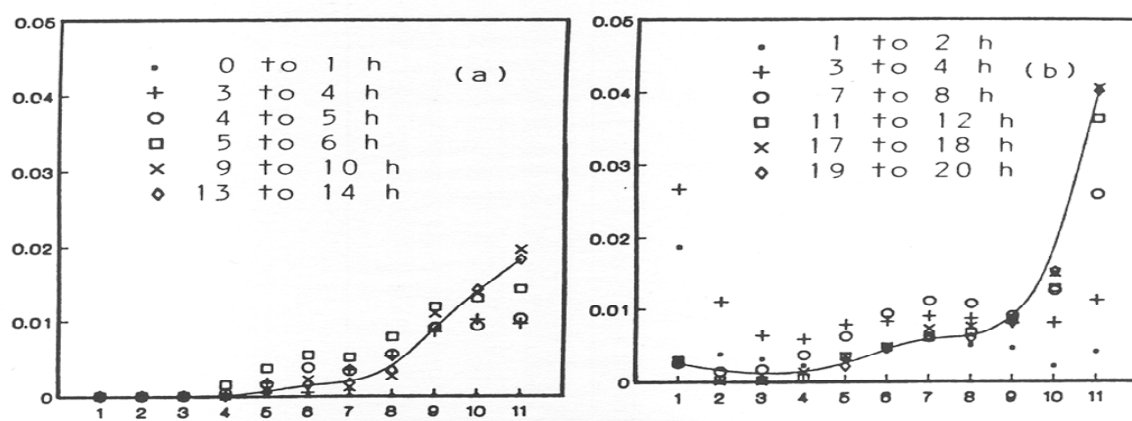


Figure 2: Moisture redistribution in test on (a) glass fiber (b) Sprayed cellulose fiber insulation

The concept of capillary active insulations was applied to thermal rehabilitation of masonry buildings where the critical consideration was the elimination of mold. After successful demonstration for several German buildings, this technique was also used in a few world known monuments such as the church of Our Lady in Dresden, some cultural properties in Japan and Rijksmuseum (Rembrandt) in Amsterdam.

Perhaps most visible technology new rehabilitation technology developed in GS is that of double walls or exterior protecting zone built around existing concrete buildings. We have also learned about the importance of air tightness in protecting exterior mineral fiber insulation and in the NA about "Water Resistive Barrier (WRB)" protecting the exterior of the walls from rain entry.

From Canada we have learned the concept of the house as a system. Introduced in 1980's program called R2000 was a blueprint of Total Quality Management applied to heating, cooling, ventilation, indoor air quality and durability of the building; something that later used in Building America program and more recently in the popular US Energy Star program. These programs are voluntary and based on social demands for sustainable construction.

Not all the innovations were readily accepted. For instance, realizing the significance of air control, the

model code in Canada7 introduced the requirement of air barriers in 1985. Yet, not much has changed until 1995 code defined and quantified specific requirements of this system. This was another lesson namely that technical requirements must be supported by the enforceable measurement technology and legal framework enforcing them.

A major lesson from the past is that the mere existence of technology is not sufficient to influence the market place. For instance, windows were always tested in laboratory as an independent product, until major water leakage damages in coastal regions of North Carolina and lower mainland of British Columbia forced the codes to require testing windows mounted in the wall in the same manner as manufacturer recommends their installation in the wall. By this requirement the responsibility for design of wall-window interface was placed on window manufacturer and the linkage between window field performance and the widow design was finally established. Another example relates to the fact that building scientists knew about air leakage effects on indoor environment for decades but only increase of the so called “sick building syndrome” cases introduces code requirements for mechanical ventilation in houses.

⁷ This code must be accepted by individual Canadian provinces.

6. CAN SUSTAINABLE PERFORMANCE BE STANDARDIZED?

Walls reflect culture.

Confucius

Putting it in the contemporary language, the way we built is a mixture of technology and social values. In the first period of urban development in China, similarly to other countries, the focus was on quantity not quality. Now we are talking about a switch to low carbon economy that implies innovation with a minimum of the cost increase. We invoke the wisdom of Confucius because we face the fundamental issue of a judgment.

On one extreme we can build to last for the next 100 years and while it may be expensive to build it saves a lot of money at the later time because of the

“Select appropriate test method to evaluate performance under service conditions”.

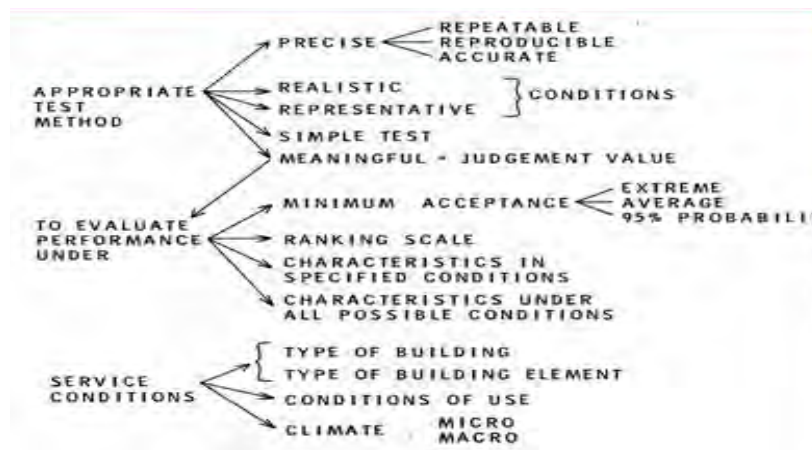


Figure 3 The options involved in a performance test

Furthermore, the term “performance” as much as the term “quality” depends on the acceptance criteria set by the “clients”. While building codes list requirements for the health and safety and for anything else related to building performance it is only at the minimum level.

So, the conclusion from the above discussion is three-fold:

- 1).we must educate the client and the best way to do it is through teaching an architect to be capable of assistance in this respect
- 2).we must provide adequate quality (performance) assurance methodology that will assist in making an intelligent decision in the realm of performance requirements.
- 3).The products delivered by construction materials manufacturing industry must have established service life span and assurance of performance for this period of service.

building durability. On the other extreme we can build for 10 years only, making it fast and inexpensive and be prepared to make repairs after one decade of use. The sustainable approach will be somewhere in between these extremes as the equilibrium between ecology, economy and society needs.

The scientist will say to establish this equilibrium perform a cost benefit analysis for a reasonable period of long-term investment. This period typically it varies between 13 to 18 years, say 15 years. Having selected period for optimization and using net present value method we can analyze the cost of the measurable performance aspects. While the reader would ask about the non-measurable performance attributes - we highlight that even dealing with the measurable attributes of performance is not simple.

Figure 3 shows the variables included in the simple sentence:

7. SUMMARY

Because of the delay in entering the world of energy efficiency, the problem solving approaches were developed in the US. Most important are two programs related to new housing:

1).The concept of shared innovation was introduced to the Building America program after observation that development of superior integrated technology (R2000 project in Canada) is not enough to rapidly change the market situation

2).Energy rating system initially introduced for a very limited scope for cost comparison was stepwise increased to include the systems approach of R2000 and more. After more than a decade it became a real driving market force

The Building America program splits the cost of innovation into two equal parts 50% is paid by the

builder and 50% is a government as a subsidy to the team of preselected experts assisting the builder with introducing innovation. The innovation comes from elsewhere in the private sector but is transferred to the construction industry in a shared manner.

The government funding for innovation is primarily restricted to strategic, national impact issues. The second program, called energy star program was initially developed by the Environmental Protection Agency and with input of leading building science consultants came now to the level of integrated multidisciplinary check list to ensure that all rated new housing are comparable on most of the aspects of performance. It resulted in creation of new specialization called “certified energy rater”. On the side of large commercial and institutional buildings the market forces created literally hundreds of web and local learning opportunities for new technology and the collaboration of American Institute of Architects with National Institute of Building Science provided a framework for infusion of applied science into the changing world of architectural design. A part of this support is development of whole building guide and building commissioning practice guide

Yet all this programs did little to improve the spread of affordable thermal upgrade in the existing buildings – tall or small.

Yet all this programs did little to improve the spread of affordable thermal upgrade in the existing buildings – tall or small.

PART 2: SITUATION IN CHINA

The close linkage between buildings and society requires that China follows own equilibrium criteria

2.1 Sustainability is the basis for all decisions

From building physics point of view the assessment of building technology should be based on sustainability not on its separate parts such ecology or materials used in the system. This may happen when the environmental component of sustainability is based on good intentions and implemented without adequate rigorous scientific examination. This type of action, in a long-run may hurt the society.

We provide two examples. In 1990’s world-wide ban of CFC (chloro-fluoro-carbons) compounds from both refrigeration and manufacturing of thermal insulating foams was scientifically justified. When doing research on the transition to the new blowing agents (HCFC) we have learned that the bulk of the blowing agents did not leave polyurethane foam but became dissolved back in the polymer. This is presented in Figure 4 (for CFC in PU foam) as a difference between curve 3 and 4. Four curves are shown in Figure 4. Curve 1 shows the aging of a 25 mm- (1 inch) thick

polyurethane specimen, fully encapsulated on all sides. The encapsulation prevents entry of air into the foam, but has no effect on the redistribution of the BA (BA) within the encapsulated foam. Part of the BA enters and saturates the polymer matrix, reducing the concentration of the BA in cell gas. This, however, does not change the thermal performance of the fully encapsulated specimen. The thermal conductivity (k-factor) does not change⁸ as long as air has not entered the cells of the foam, despite the change in pressure caused by the cell gas redistribution.

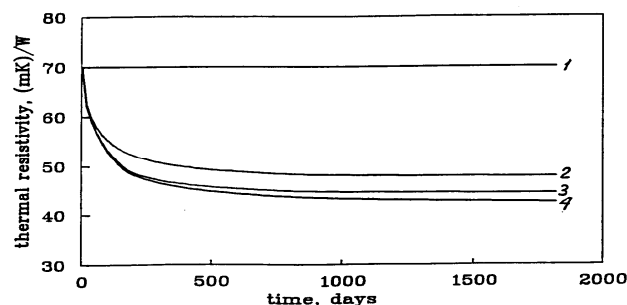


Figure 4: Aging of 25-mm thick

Curve 2 shown in Figure 4 relates to the hypothetical aging of the same specimen when only air is allowed to enter the foam. In this computer simulation, the BA redistribution is eliminated by using zero values for the effective diffusion and solubility coefficients of the BA; the effective diffusion coefficients for oxygen and nitrogen are measured. Curve 3 in Figure 4 shows the increase in the aging process when the effects of BA solubility in the polymeric matrix are included in the calculation. Finally, Curve 4 adds one more mechanism of the aging process, namely that of the BA outward diffusion. Figure 4 show that the difference between curves 3 and 4 is small. This implies that aging is primarily caused by the ingress of air into the foam and the solubility of the BA.

Now let us recapitulate the facts. CFC losses from PU foam were very small in CFC blown foam. The solubility of HCFC in the polymeric matrix of PU is much higher than that of CFC. The ozone depleting potential is 10-time smaller than that of CFC. From building science view point, one should observe the law of diminishing returns and do not pursue elimination of HCFC from PU foams but address the BA emission during the foam manufacturing and let this BA stay on the market place. Since China and other developing countries are still using HCFC that provides better performance for lower cost we may not see the reason for HCFC elimination. While the elimination of CFC from leaky refrigerators and foams addressed the ecological part of sustainability with a minimum effect

⁸ It is known that the thermal conductivity of the cell gas does not depend on the pressure of the gas, as long as the gas pressure does not fall below 1/100 of the atmosphere.

on the society and economy, further reduction of already miniscule emission of the ozone depleting substance from PU foams is out of equilibrium on which the concept of sustainability is built. In our opinion a rigorous scientific examination should be carried before HCFC is banned in China and we should spend our efforts somewhere else where all three components of sustainability can benefit.

Another example relates to ban of polystyrene in China on the exterior thermal insulation composite systems. While the authors support the reason for the action we would like to see that this ban is reformulated in such language that it assists in the technology improvement. Instead of using polystyrene in the exterior thermal insulation composite system (ETICS) one can use a different system, so called DEFS (direct applied finishing system) that can be made with a required fire protection because insulation is placed behind the fire protective layer. Again, if the concept of sustainability is involved, we achieve a good fire protection but we also continue using inexpensive material.

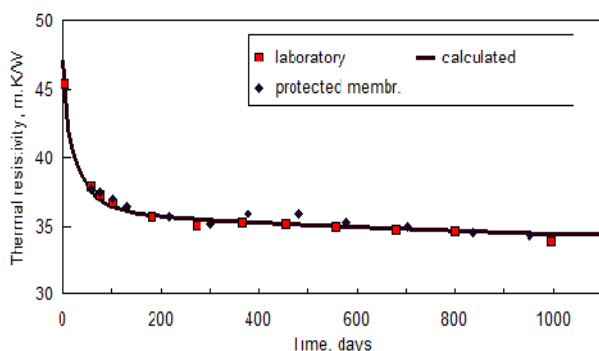


Figure 5: One of the verification tests of the DIPAC model

These two examples highlight that instead of the traditional approach of using materials to standardize performance, regulations should formulate performance requirements and the industry should be able to develop appropriate technology.

2.2. Innovation and product quality management cannot be separated

Most of our testing standards were developed for comparing different products for an identical use. For instance, to select an interior finish we compare properties of different types of gypsum boards. We are dealing with known application and known performance requirements. When a new area of concern arises, e.g., mold growth on the surface of building partitions and someone calls their product a "mold resistant board", we scramble how to test mold resistance of materials. In doing so, we make a number of assumptions that may or may not relate to condition of the field application. For instance, we start with a clean material and study the

mold growth over 28 days. In real life, however, mold can appear after 3 hours, 3 days or 3 weeks depending if the dormant mold spores were present or not. So, while the test may shed some light on the propensity of material to contribute to the mold growth it does not describe the risk of mold growth.

Field exposures involving variable environmental conditions are represented as exposures with a constant temperature and humidity. A good example of difficulty in relating field performance to the laboratory test are viscoelastic materials e.g. sealants. A cohesive damage in those materials is defined by so called stress-strain envelope that includes slow rate of elongation and high temperature or high rate of compression at low temperature. Furthermore, large size of panels is used (because of stress concentration behavior in thermal expansion) and except for very specialized research the standard testing either uses constant environment or variable environment and constant mechanical loads. Neither ASTM nor Agreement Board (Europe) considers any interaction of mechanical and environmental loads.

The last example highlights the problems with durability testing because deterioration processes cannot be accelerated. Using more severe exposure conditions seldom produces correlation valid for more than the tested specimen. For instance there is no real correlation between the UV lamps and actual exposure. Needless to say that most of the damage in the field relates to a combination of temperature, moisture and movements.

This is a general background and lacking adequate methods for evaluating field performance, we need information from the two sides: (1) product manufacturers and (2) contractors responsible for the process of construction. On the manufacturer side we have (1.1) quality assurance of the components and materials and (1.2) report on technical equivalency from the national code compliance center. For instance developed on the request of one company, the Evaluation Service of the International Construction Code (US) produced the Structural and Durability test (based on Canadian Construction Materials Center public protocol) ES395 that includes temperature oscillation, positive and negative air pressure and spray of water on the surface of the cladding material. In addition to the surface observations, cohesive and adhesive strength of specimens cut before and after the cycling exposure is examined. On the contractor side one follows (2.1) commissioning protocols and (2.2) performs testing on the mock-up assemblies. While traditionally mock-ups were built only to train the workers in construction sequence and details, they are now used to do testing that resembles field exposure, often performed by a qualified personnel from a testing agency.

In the Canadian national evaluation program all manufacturers of innovative materials are also requested

to provide their description of the quality assurance (QA) program in plant and periodically this QA program is checked by a third party testing. The QA program becomes a critical part of the manufacturing capability.

In China, it appears that, except for concrete industry, the construction manufacturing sector often does not have adequate requirements for ensuring quality (performance) of the manufactured products. Sometimes a larger company requests change of the product characteristics and pay premium for better performance. Incidentally, this is the single most important obstacle to the invention in the field of construction products and could be overcome by a policy of keeping QA log-books in parallel to the financial books.

2.3. Bridge the gap between universities and industrial innovation

When we look at mechanisms of academic – industrial collaboration in the world we can see everything between two extremes. On one hand we have a seamless, close collaboration on the other we have a dispute over the intellectual property resulting from the collaborative work. For financial gains, many Universities want to share of the intellectual property belonging to the private sector. When one of the authors worked in Sweden and run out of the grant money, collaborating industry would simply send a check to assist the research, when he worked in the US he had to rent the space in own laboratory and support the students (who were doing PhD at the University) from the industrial money to avoid the University claim to the intellectual property on the manufacturing process of an industrial product.

The authors who are involved in University-Industry collaboration on innovative materials use the following principles:

1. There is a steering committee that includes involved faculty and industrial representatives
2. The industrial members agree to introduce a quality assurance program for the products that is discussed by both parties
3. University assigns at least one graduate student to research in the area agreed between the University and the industrial member
4. Intellectual property (IP) related to the manufacturing technology remains with the industry
5. Intellectual property (IP) related to the performance evaluation of materials and systems remains with the University except that the involved company is automatically granted use of IP developed jointly. This IP is typically used in the QA program
6. Except for the collaboration issues agreed nothing else constitutes any relationship between the University and the companies involved in the consortium.

As you may see, we have included in the scope of Industry University Innovation Team (IUIT) not only

the support to the innovation but also degree of the support to the industry quality assurance program.

2.4. Concluding remarks

Building enclosures – their energy efficiency, durability and indoor environment – are today at the cross-roads. On the one hand there is a large amount of knowledge available; on the other hand the old approaches are not as valid as before. The society requires consideration of sustainability and this brings us to a new vision and holistic design with a view to improving energy efficiency and occupant comfort but without increasing the cost. To this end we must find where trade-offs can be made. Typically more efficient building enclosure reduces cost of HVAC.

We talk about three steps in the design process: (1) all passive measures, (2) geo-solar energy capture and storage, (3) renewable energy sources, primarily photovoltaic. In this process, one must simultaneously develop new solutions and new methodology of field performance evaluation.

To this end we need innovation. Yet innovation cannot be perceived as the concept of new product, it must be considered jointly with industrial manufacturing capability and as such is non-separable from quality assurance in the product manufacturing. Universities have a very important role to play in the whole continuum of innovation.

This conference is focused on a very important aspect of performance, namely bridging the passive design and the applications of renewable energy. Because the renewable energy is still very expensive, not only all the passive house characteristics but also the low grade energy (low exergy) sources must be well used before application of renewable energy. One must also remember that thermal storage does not necessarily mean heavy walls but it could be achieved with airtight, well insulated walls or in mixed climate with switchable thermal mass and thermal insulation. Yet, to this end one needs design air redistribution systems or integrate ventilation and building enclosure.

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AN ADAPTIVE RESEARCH ON GROUND SOURCE HEAT PUMP SYSTEM FOR RESIDENTIAL SPACE HEATING BASED ON MEASURED DATA

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ABSTRACT

China's southern residents are increasingly concerned about the space heating in winter. The chief aim of the present work is to find a cost-effective way of residential space heating in Shanghai, one of the biggest city in south China. Economic and energy efficiency of three residential space heating ways, including ground source heat pump (GSHP), air source heat pump (ASHP) and wall-hung gas boiler (WHGB), are assessed based on Long-term measured data. The results show that the heat load of the building is 120 kWh/m²/y over the heating season, and the seasonal energy efficiency ratio (SEER) of the GSHP, ASHP and WHGB systems are 3.27, 2.30, 0.88 respectively. Compared to ASHP and WHGB, energy savings of GSHP during the heating season are 6.2 kgce/(m².y) and 2.2 kgce/(m².y), and the payback period of GSHP are 13.3 and 7.6 years respectively. The sensitivity analysis of various factors that affect the payback period is carried out, and the results suggest that SEER is the most critical factor affecting the feasibility of ground source heat pump application, followed by building load factor and energy price factor. These findings of the research have led the author to the conclusion that ground source heat pump for residential space heating in Shanghai is a good alternatives, which can achieve significant energy saving benefits, and a good system design and operation management is the key initiatives that shorten the payback period.

Keywords: Adaptive, Ground source heat pump, Residential building, Measured data, Energy efficiency, Economic Analysis

1. INTRODUCTION

With a continuous large-scale snow and freezing weather in Southern China at the beginning of 2013, the debate over whether the government should provide central heating in the south of China is becoming more intense. Up to now, the central heating systems covers about 70% of the urban building area in Northern China (Yellow River basin and its north) but less than 5% of the urban building area in Southern China (Yangtze River basin, Hunan, Guizhou, and Yunnan etc.). During the winter season, the indoor temperature of residential buildings without central heating systems are fluctuated as a function of the change of outdoor temperature in China's hot summer and cold winter region. And the average indoor temperature is only 10-17 °C, which is far from ASHRAE comfort zone^{0,0}. Therefore, space heating in winter is becoming the urgent needs for resident in those areas.

For the purpose of finding a cost-effective way of residential space heating in China's hot summer and cold winter region, energy efficiency and economic feasibility of three residential space heating ways, including ground source heat pump (GSHP), air source heat pump (ASHP) and wall-hung gas boiler (WHGB), are assessed based on Long-term measured data in the present study.

2. SYSTEMS DESCRIPTION

2.1. Building

The testing building is located in Shanghai (31.20°N, 121.21°E). It is a three-storey private residential house, with a total gross floor area of 280m² and air-conditioned area of 180m². Figure 1 shows the second floor plan of the building. The average low temperatures in the area in January are about 4.3°C. There are approximately 1648 heating

NOMENCLATURE

Abbreviation	SEER	Seasonal energy efficiency ratio [kWh/kWh]
A The total heating space area [m^2]	SPER	Seasonal primary energy ratio [kWh/kWh]
AC The annual operating costs [Yuan]	t	Temperature [$^{\circ}\text{C}$]
ASHP Air source heat pump	WHGB	Wall-hung gas boiler
COP The coefficient of performance [kWh/kWh]	i	interest rate [%]
EC Energy consumption [kWh]	τ	instantaneous time [h]
FP Fuel price [Yuan/kWh, Yuan/ Nm^3]	$\Delta\tau$	time interval [h]
GSHP Ground source heat pump	Subscripts	
IC The initial cost [Yuan]	a	ambient
m Mass flow rate [kg/h]	pump	pump
MC Maintenance costs [Yuan]	ra	return air
n Number of interest periods [year]	rw	return water
N Electricity consumption [kWh]	unit	unit
ϵ The power generation efficiency [%]	s	the heating season
PEC Primary energy consumption [kgce]	sa	supply air
PP The payback period [year]	sw	supply water
PW The present worth [Yuan]	sys	system
Q Heat consumption [kWh]	w	water
q Heat consumption per unit area [kWh/ m^2]		

degree days (18°C base). The peak value of the heating load was calculated as 20 kW, equals to 110 W/m^2 . The occupancy time of the building is from 17:00 to 08:00 next day for the working days and 24h for weekend.

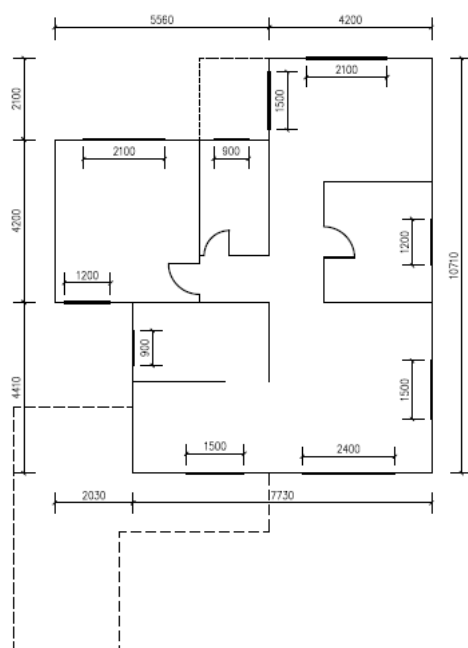


Figure 1 The second floor plan of the building

2.2. Space heating systems

2.2.1. GSHP heating system

The GSHP heating system installed in the testing building is mainly composed of three parts of ground heat exchanger (GHE), mechanical equipment (including heat pump and circulating pumps), and radiant floor heating terminal units. The detailed specifications and main technical parameters of the GSHP heating system were given in Table 1.

Table 1 The main components characteristics of the GSHP heating system.

Ground heat exchanger (GHE)	
Type of GHE	Vertical single U-bend
Material of U-bends	HDPE high density polyethylene
Size of U-bends	$\Phi 32\text{mm} \times 3.2\text{mm}$
Depth of boreholes	$5 \times 90 \text{ m}$
Mechanical equipment	
Heat pump	Rated heating capacity 16kW /inverter / DAKIN EEHS-6P
circulation pumps	$N_{\text{max}}=0.27\text{kW}$, $H_{\text{max}}=12.8\text{m}$, $Q_{\text{max}}=4\text{m}^3/\text{h}$ GROUNDFOSS 25-125 180

2.2.2. reference systems

Two conventional heating systems, which are the most commonly used for space heating in Shanghai, were selected as reference system in this study. One is a wall-hung gas boiler (WHGB) with a heating capacity of 20kW, and the other is a heating system composed by eight split air source heat pumps (ASHP). The heating capacity of a single ASHP unit is about 3.5 kW.

2.3. Data acquisition system

A network data-acquisition system was designed to monitoring the key operating parameters of GSHP system and ASHP system. The following data were regularly recorded in every 5 minute for GSHP system and ASHP system, respectively.

The parameters monitored for GSHP system as follows:

- Electrical power input to the compressor and circulating pump by using watt-hour meters;
- Supply /return water temperature and water flow rates of the GHE by using an ultrasonic calorimeter with resistance thermometers (Pt100).
- Supply /return water temperature and water flow rates of the radiant floor heating system by using an ultrasonic calorimeter with resistance thermometers (Pt100).

- Ambient and indoor air temperatures by using resistance thermometers (Pt100);

The parameters monitored for ASHP system as follows:

- Electrical power input to the compressor and fans by using a watt-hour meter;
- Supply /return air temperatures of outdoor and indoor unit by using resistance thermometers (Pt100);
- Circulating air flow rates of outdoor and indoor unit by using hot-wire anemometers.

3. ANALYSIS METHODS

3.1. Energy consumption and efficiency

3.1.1. heat consumption

A calorimeter was used to monitoring the heat energy supplied to the building. Therefore, the heat consumption of the building in the $\Delta\tau$ interval starting at τ_0 time was integrated by the following formulas.

$$Q = \int_{\tau_0}^{\tau_0 + \Delta\tau} Q(\tau) d\tau \quad (1)$$

$$Q(\tau) = m_{w,\tau} c_{p,w} (t_{sw,\tau} - t_{rw,\tau}) \quad (2)$$

Where $Q(\tau)$ is the instantaneous heating power, $m_{w,\tau}$ is the instantaneous mass flow rate of water, $c_{p,w}$ is the specific heat capacity of water, $t_{sw,\tau}$ and $t_{rw,\tau}$ represent the instantaneous temperature of supply water and return water, respectively. And the heat consumption per unit area over the heating season was calculated by using Eq(3)

$$q_s = \frac{Q \Big|_{\tau_{start}}^{\tau_{end}}}{A} \quad (3)$$

Where A is the total space heating area, $Q \Big|_{\tau_{start}}^{\tau_{end}}$ is the total heat consumption over the heating season,

τ_{start} and τ_{end} represent the space heating start and stop times, respectively.

3.1.2. GSHP system performance calculation

The energy consumption of the GSHP heating system consists of the electricity consumption of the heat pump units and water circulating pumps. The electricity consumption is monitored by electricity meters through integrating instantaneous power consumption ($N(\tau)$) by using Eq (4)

$$N = \int_{\tau_0}^{\tau_0 + \Delta\tau} N(\tau) d\tau \quad (4)$$

The coefficient of performance (COP) of the GSHP unit ($COP_{GSHP, unit}$) and the whole GSHP heating system ($COP_{GSHP, sys}$) were calculated by Eq (5) and Eq (6), respectively.

$$COP_{GSHP, unit} = \frac{Q}{N_{GSHP}} \quad (5)$$

$$COP_{GSHP, sys} = \frac{Q}{N_{GSHP} + N_{pump}} \quad (6)$$

Where N_{GSHP} is the electricity consumption of GSHP unit, and N_{pump} is the electricity consumption of water circulating pumps, including circulating pumps of ground coupling loop and circulating pump of radiant floor heating loop.

The seasonal primary energy ratio of GSHP heating system ($SPER_{GSHP, sys}$) was calculated by

$$SPER_{GSHP, sys} = \varepsilon \cdot SEER_{GSHP, sys} \quad (7)$$

Where ε is the power generation efficiency, according to statistics [4][5], the power generation efficiency in the present study is set to 0.31.

$SEER_{GSHP, sys}$ is the seasonal energy efficiency ratio of GSHP heating system, which was calculated by Eq (8).

$$C_{Di} = \frac{C_i}{S_i}, (i = 1, 2, 3)$$

$$C_A = CT_A \times G_A = (\overline{CT_A} \times 1.2) \times G_A$$

$$CT_A = \overline{CT_A} \times 1.2 \quad (8)$$

$$\overline{CT_A} = \frac{\overline{C_A}}{G_A}$$

$$COP_{GSHP, unit}$$

$$SPER_{SHP, sys} = COP_{GSHP, sys} \Big|_{\tau_{start}}^{\tau_{end}}$$

3.1.3. reference system calculation

A split air source heat pump (ASHP) heating system was investigated experimentally. The following parameters were monitored for a long period with 5-minute intervals, including the power consumption of the system (N_{ASHP}), supply air temperature ($t_{sa,\tau}$), return air temperature ($t_{ra,\tau}$), circulating air flow ($m_{a,\tau}$), ambient temperature ($t_{a,\tau}$), etc. The instantaneous COP of the ASHP heating system was calculated using the following formula:

$$COP_{ASHP, sys}(t_{a,\tau}) = m_{a,\tau} c_{p,a} (t_{sa,\tau} - t_{ra,\tau}) / N_{ASHP} \quad (3)$$

Considering the impact of ambient temperature on the COP of the ASHP heating system, the seasonal energy efficiency ratio of ASHP heating system ($SEER_{ASHP, sys}$) was estimated as weighted average of $COP_{ASHP, sys}$ on different ambient temperature conditions, using the following formula:

$$SEER_{ASHP, sys} = \sum \omega(t_{a,\tau}) \cdot COP_{ASHP, sys}(t_{a,\tau}) \quad (10)$$

Where $\omega(t_{a,\tau})$ is the probability of ambient temperatures which are between $t_{a,\tau}$ and $t_{a,\tau} + 0.5$

during the whole heating season. The seasonal primary energy ratio of ASHP heating system ($SPER_{ASHP,sys}$) was calculated by

$$SPER_{ASHP,sys} = \varepsilon \cdot SEER_{ASHP,sys} \quad (4)$$

The energy efficiency ratio of wall-hung gas boiler (WHGB) heating system is equal to the primary energy ratio of WHGB heating system at about 0.85-0.90⁰, and it does not vary with changes of the ambient temperature. The seasonal energy efficiency ratio and the seasonal primary energy ratio of WHGB heating system ($SEER_{WHGB,sys}$ and $SPER_{WHGB,sys}$) were set to 0.88.

We assumed that both the building heat consumption over the heating season by using reference heating systems (ASHP or WHGB) and GSHP heating system are the same. So the energy consumption (EC) and primary energy consumption (PEC) of three heating systems were calculated by Eq(12) and Eq(13), respectively.

$$E = Q \big|_{\tau_{start}}^{\tau_{end}} / SEER \quad (5)$$

$$PEC = Q \big|_{\tau_{start}}^{\tau_{end}} / (29.3 SEER) \quad (6)$$

Where value of 29.3 is the calorific value of coal equivalent.

3.2. Economic analysis

To determine whether GSHP would produce any economic benefits for home owners, an economic analysis was carried out to compare GSHP with ASHP and WHGB heating system.

3.2.1. initial cost

According to actual situation of the project construction and estimated using data from local contractors, equipment suppliers and construction estimating software, the investment cost, including material, labor, and taxes, for the three heating systems compared are given in Table 2.

Table 2 Initial investment of three heating systems

Systems	Item	Cost (Yuan)
GSHP	Ground heat exchanger	19,000
	Equipment	25,000
	radiant floor heating system	28,000
	Total	72,000
ASHP	Equipment	38,000
	installation and debugging	14,000
	Total	52,000
WHGB	Equipment	20,000
	radiant floor heating system	28,000
	Total	48,000

3.2.2. annual operating cost

The annual operating costs of a space heating system includes fuel cost and maintenance of equipment. The annual operating costs (AC) was given as:

$$AC = EC \times EP + MC \quad (7)$$

Where FP is fuel price, MC is equipment maintenance costs. Values of FP and MC are given in Table 3 and Table 4, respectively.

Table 3 Fuel prices

Fuel type	Unit cost
Electricity	0.67 Yuan/kWh
Natural gas	2.9 Yuan/m ³

Table 4 System maintenance costs

Systems	Maintenance costs
GSHP	100 Yuan/y
ASHP	400 Yuan/y
WHGB	200 Yuan/y

3.2.3. economic comparison

To study the economic feasibility of a system, different methods could be used to evaluate the different figures of merit of the systems. Some examples are: Net present value method, the internal rate of return method, the annual cost and the other methods⁰. In this study, the present worth (PW) method has been used to compare the cost effectiveness of the three heating systems.

The PW method refers to all the events in an economy, both incomes and expenditures, as one figure at one point in time. When comparing different alternatives for a given revenue or cost saving, the alternative with the least present worth of all expenditures or annual charges was to be preferred. Therefore, the payback period (PP) of alternative A compared with alternative B would be the year that the present worth of B over that of A. The present worth (PW) was given as:

$$PW = IC + AC \times \frac{1}{i} [1 - (1+i)^{-n}] \quad (8)$$

Where IC is the initial cost, AC is the annual operating cost, i is interest rate, n is number of interest periods.

3.2.4. sensitivity analysis

The PW method can be better used for assessing the economic feasibility of different heating systems, but the result is subject to many uncertainties. Such as the building heating load, the system energy efficiency is uncertain due to the impact of the human behavior and test instruments accuracy, and fuel price also has some uncertainty. Thus, analysis of these uncertainties impact on the economy of the heating system is also very important.

Sensitivity analysis is very useful when attempting to determine the impact the actual outcome of a particular variable will have if it differs from what was previously assumed. By creating a given set of scenarios, the analyst can determine how changes in one variable(s) will impact the target variable. In this respect, the sensitivity analysis of the key factors that

affect the payback period is carried out. For sensitivity analysis, three key parameters, including building heating load (q_s), the seasonal energy efficiency ratio of GSHP heating system ($SEER_{GSHP,sys}$) and fuel price (FP), were to be increased and decreased by 20% of their original value.

4. RESULTS AND DISCUSSIONS

4.1. Variation of ambient temperature and heating load

The ambient temperature is a key factor for building heating load and performance of heating systems. In order to observe and find out the interaction of energy consumption and coefficients of heating systems performance with ambient temperatures, the ambient temperature was measured continuously during the heating season. The measured data show that the ambient temperature was determined to change in the range of $-2 \sim 20^\circ\text{C}$. Figure 2 represents the distribution probability of ambient temperatures over the heating season. As we can see from the figure, the outdoor

temperature is mainly distributed in the range of $2 \sim 13^\circ\text{C}$, accounting for more than 90%. And the weighted average value of the ambient temperature during the whole heating season is 7.96°C .

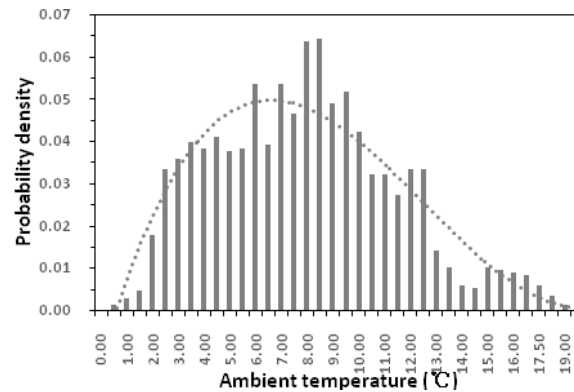


Figure 2 Probability density of ambient temperature during heating season

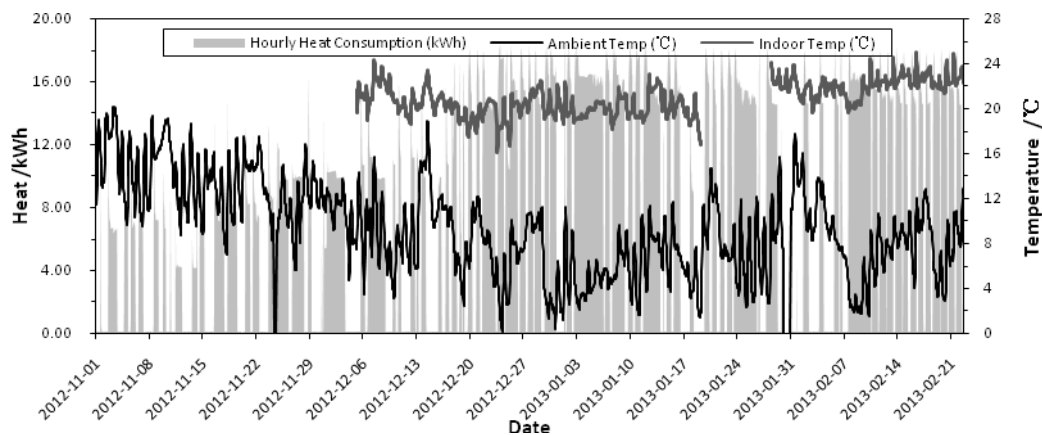


Figure 3 Variation of building heat consumption with ambient and indoor temperature

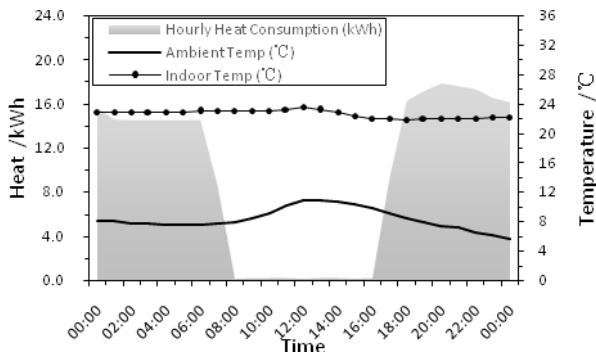


Figure 4 Variation of building heat consumption in atypical day

The variations of building heat consumption, ambient temperature and indoor temperature during the

heating season are shown in Figure 3. And Figure 4 shows the hourly variation of building heat consumption in a typical day. As these two figures suggested that the indoor temperature is mainly maintained at $19 \sim 23^\circ\text{C}$, and the building heat consumption and outdoor temperature has a very significant correlation. Figure 5 indicates that the average values of heat load is a liner function with ambient temperatures. During the heating season, the maximum heat load is about 80 W/m^2 when the ambient temperature reached 0°C , and the heat consumption per unit area over the heating season calculated by Eq (3) is 120 kWh/m^2 .

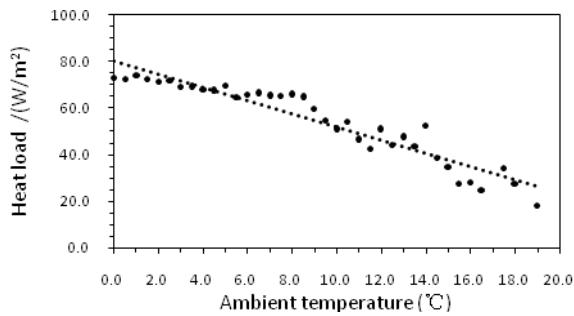


Figure 5 Average heat load versus ambient temperature

4.2. Performance of GSHP heating system

The energy performance of a GSHP heating system can be influenced by three primary factors: (a) the heat pump machine, (b) the circulating pump and (c) the ground coupling. The heat pump is the largest single energy consumer in the system. Its performance is a function of the efficiency of the machine and the water temperature produced by the ground coupling⁰.

Based on the measured data, the COP of the heat pump unit $COP_{GSHP,unit}$ and the whole system

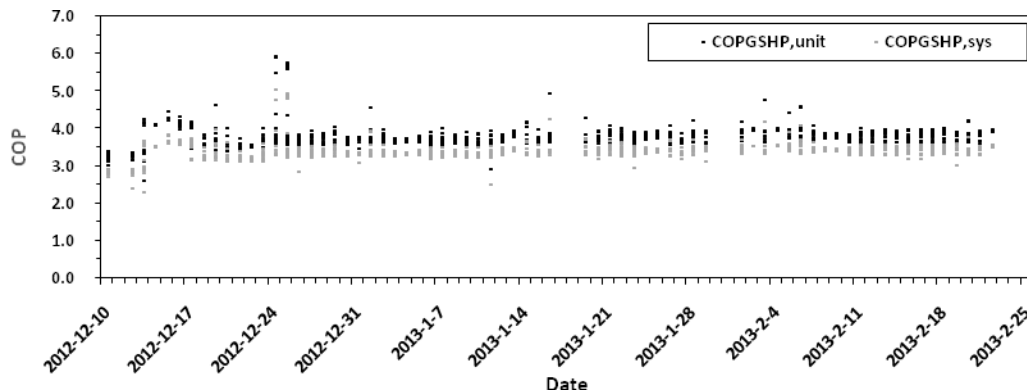


Figure 6 Unit and system COP values of GSHP during a heating period

4.3. Performance of ASHP heating system

To determine the actual heating performance of an air source heat pump, a split ASHP was investigated experimentally under different weather conditions. The COP of the ASHP heating system was calculated using the Eq(9), and Figure 7 shows the average values of COP_{ASH} under different ambient temperatures at 0.5°C interval. As the figure shows, the COP_{ASH} with ambient temperature is a linear relationship. The COP_{ASH} reached 3.57 when the ambient temperature get 12.5°C, but only 1.50 when the outdoor temperature dropped to 2 °C.

$COP_{GSHP,sys}$ during the heating season were calculated by using Eqs.(5) And Eqs (6) .Figure 6 shows the hourly average values of $COP_{GSHP,unit}$ and $COP_{GSHP,sys}$ during a long-term heating period, from Dec. 2012 to Feb. 2013. It is clear from this figure that the values of $COP_{GSHP,unit}$ and $COP_{GSHP,sys}$ are relatively stable except some outlier points. The highest COP values were observed as 4.65 for the heat pump and 4.16 for the system at the beginning of the heating season, when the water temperature produced by the ground coupling reached a relatively high value at 17.2°C. About one month later since the starting time of heating, the water temperature produced by the ground coupling stabilized at 12 °C to 13.5 °C and the $COP_{GSHP,unit}$ and $COP_{GSHP,sys}$ varied between 3.51-4.25 and 3.2-3.6 respectively. The value of the seasonal energy efficiency ratio of GSHP heating system ($SEER_{SHP,sys}$) and the seasonal primary energy ratio of GSHP heating system ($SPER_{SHP,sys}$) which calculated by Eq(8) and Eq(7), is 3.27 and 1.01, respectively.

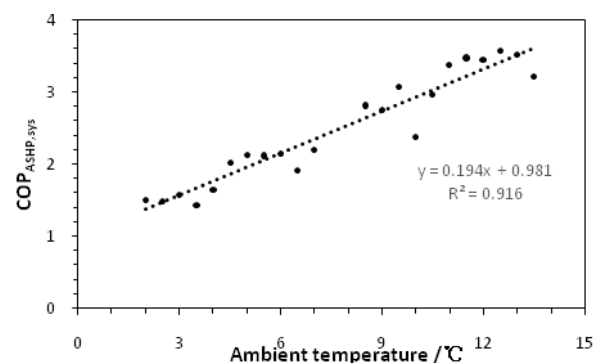


Figure 7 Variation of $COP_{ASH,sys}$ versus the ambient temperature

According to this function between $COP_{GSHP,sys}$ and ambient temperature, and the above-mentioned distribution probability of ambient temperatures over the heating season, the seasonal energy efficiency ratio of ASHP heating system

($SEER_{AS}$) and the seasonal primary energy ratio of ASHP heating system ($SEER_{AS}$) calculated by Eq(10) and Eq(11) is 2.30 and 0.71, respectively.

4.4. Energy efficiency comparison

Based on the above-calculated results, the comparison of energy efficiency of three heating systems are given in Table 2. As the table shows, the GSHP heating system has the highest seasonal primary energy ratio (SPER), and followed by WHGB. Compared with ASHP and WHGB, the annual energy saved per unit heating area by GSHP is 6.2 kgce and 2.2 kgce, respectively. This has confirmed that GSHP is the most energy efficiency way for space heating in Shanghai.

Table 2 Energy efficiency comparison

Item	GSHP	ASHP	WHGB
(kWh/m ²)	120	120	120
SEER	3.27	2.30	0.88
SPER	1.01	0.71	0.88
EC			
(kWh(e)/m ² , Nm ³ (g)/ m ²)	36.7(e)	52.2(e)	15.8(g)
PEC (kgce/m ²)	14.5	20.7	16.7

4.5. Economic feasibility evaluation

The present worth (PW) of three heating systems has been calculated by using Eq. (9) with a lifetime of 25 years and an interest rate of 5.6%. Figure 8 illustrates the variation of the present worth (PW) of three systems versus the operation life. As can be seen from the figure, at the twenty-five years, PW value of the GSHP system is the minimum and that of WHGB is the maximum. This result proves that the GSHP heating system is to be preferred among three heating systems. Compared with ASHP and WHGB heating system, the payback period of GSHP system is 13.3 and 7.6 years.

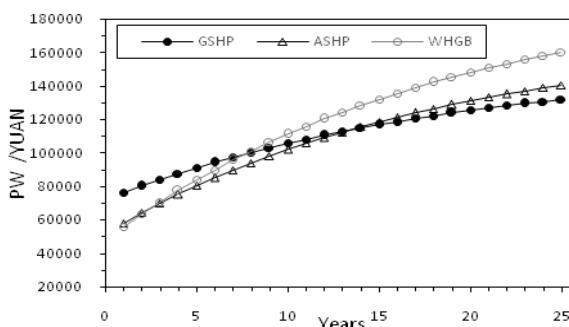


Figure 8 Variation of the present worth versus the operation life of the system

To determine which parameters are the key drivers of the payback period of GSHP system, a sensitivity analysis is carried out. Three key parameters, including building heating load, the seasonal energy efficiency ratio of GSHP heating system ($SEER_{GS}$) and fuel price (FP), were to be increased and decreased by 20% of their original value. The calculated results are shown in

Figure 9. It is clear from this figure that the payback period of GSHP compared with WHGB and ASHP decreased with one of the parameters ($SEER_{GS,sys}$ and FP) increased. Among these three key parameters, the $SEER_{GSHP,sys}$ is the most important parameter affecting the payback period of GSHP both compared with WHGB and ASHP.

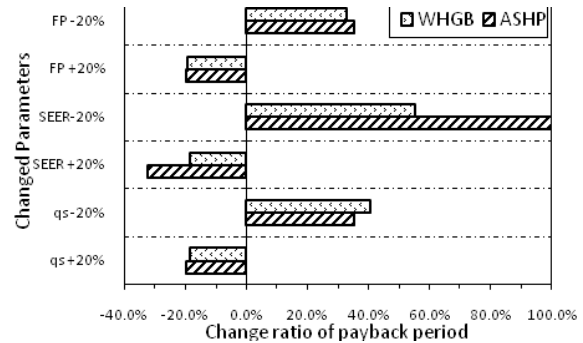


Figure 9 Sensitivity analysis of the payback period of GSHP compared with WHGB and ASHP.

5. CONCLUSIONS

Energy efficiency and economic feasibility analysis of three heating systems for residential building in Shanghai was carried out in the present study. From the results of the experimental study, the following conclusions may be drawn:

1) The average ambient temperature during the whole heating season is 7.96°C. The peak heat load of the residential building is about 80 W/m², and the total heat consumption is 120 kWh/m²/y.

2) The seasonal energy efficiency ratio (SEER) of the GSHP, ASHP and WHGB heating systems are 3.27, 2.30 and 0.88, respectively. And the corresponding seasonal primary energy ratio are 1.01, 0.71 and 0.88, respectively. Compared to ASHP and WHGB, energy savings of GSHP during the heating season are 6.2 kgce/(m².y) and 2.2 kgce/(m².y). Those results indicate that the GSHP heating system is the most efficiency way for residential space heating in Shanghai.

3) Comparing with ASHP and WHGB, the present worth (PW) value with 25 years lifetime of GSHP is the minimum, and the payback period is 13.3 and 7.6 years, respectively. This proves that the GSHP heating system is to be preferred among three heating systems.

4) The sensitivity analysis results suggested that the seasonal energy efficiency ratio of GSHP heating system is the most important parameter affecting the payback period of GSHP heating system. Therefore, a good system design and operation management is the key initiatives that shorten the payback period.

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A METHOD FOR OPTIMIZING THE DESIGN OF A SOLAR-ASSISTED GROUND COUPLED HEAT PUMP SYSTEM

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ABSTRACT

A solar-assisted ground coupled heat pump (SAGCHP) system is a technique to fully utilize sustainable energy. The matching of each component is important for the efficiency of the system. This paper proposes a method for optimizing the design of the system components. Using the method, the optimized pair of the length of borehole(L) and the area of solar collector(A) is determined for the objective of less energy consumption with the satisfaction of the cooling and heating loads. A demo case is designed and the dynamic characteristics in a whole year is simulated by the commercial software TRNSYS. The results show the pair of L and A with the less energy consumption can be found using the method based on the building loads.

Keywords: Renewable energy source, Heat pump, Solar energy, Ground source, Optimization.

1. TRODUCTION

Because building energy accounts for about 25.5% of commercial energy consumption in China[1], some techniques to integrate renewable energy sources are developed in buildings for energy saving. GCHP was used for cooling and heating in buildings. But when the heating load is much larger than the cooling load, the earth temperature will dropped after a running turn so that the heat pump system cannot work in a high efficiency. Solar-assisted ground coupled heat pump (SAGCHP) systems for building heating demands were proposed to improve the efficiency of the ground coupled heat pump (GCHP) and make a full use of solar energy[2,3]. Though the application of the solar collector can reduce the depth of borehole, and decrease the energy for heating to help the earth temperature recovery, it will maybe increase the energy for cooling because the potential rise of the earth temperature. Therefore it is important for the design of the system to determine the matching between the area of the solar collector (A) and the depth of the borehole (L). This paper proposes a design method for optimizing the pair of A and L .

2. HODOLOGY

2.1 Ematical description of the system

A SAGCHP comprises mainly of a solar collector, ground heat exchanger and heat pump in various of connections. In this paper, the system shown in Figure 1 is studied. In the heating mode, the solar collector is on while it is off in the cooling mode.

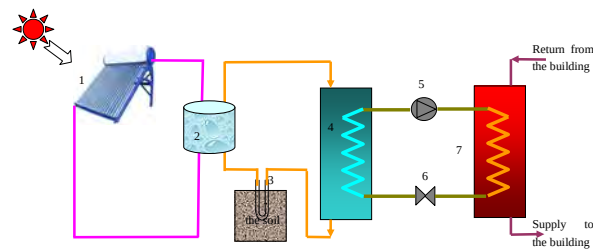


Figure. 1. Schematic shown of a SAGCHP system

1. The solar collector; 2. water tank; 3. Soil heat exchanger; 4. Evaporator (in winter) or Condenser (in summer); 5. Composer; 6. Throttling device; 7. Condenser (in winter) or Evaporator (in summer)

For the solar collector, the energy gain can be expressed by Eq. (1)

$$Q_c = A_c I \eta_c = C_p \dot{m} (t_3 - t_2) \quad (1)$$

Here, I is dependent of the location and can be determined by the meteorological information. η is determined by the type of the collector.

The heat transfer between the fluid inside GHE and the soil can be expressed as Eq. (2)

$$Q_e = kL(t_{soil} - \bar{t}_{f,GHX}) = C_p \dot{m} (t_2 - t_1) \quad (2)$$

where k is an equivalent coefficient and is determined the average temperature of the fluid can be simplified as Eq. (3):

$$\bar{t}_{f,GHX} = \frac{t_1 + t_2}{2} \quad (3).$$

For the heat pump, the energy balance exists as Equation (4) and (5) for heating and cooling mode.

NOMENCLATURE

Abbreviation		Subscripts	
GCHP	Ground coupled heat pump	c	The solar collector
SAGCHP	Solar-assisted ground coupled heat pump	co	condenser
COP	Coefficient of Performance	cooling	Cooling case
GHE	Ground heat exchanger	e	Soil heat transfer
		ep	evaporator
		s	Summer
		w	winter
		heating	Heating case
		1	The inlet of GHE
		2	The outlet of GHE
		3	The inlet of the evaporator (condenser)
		re	Heat rejection to the soil
		ad	Heat adsorption from the soil
		Greek	
		η	The efficiency of solar collection
Symbols			
A	the area of the solar collector(m ²)		
L	the depth of the borehole (m)		
Q	Heat transfer rate (kW)		
t	Temperature (°C)		
$\dot{m}$	Flow rate (kg/s)		
C _p	The capacity of the fluid (kJ/(kg °C))		
E	The electrical consumption (kJ)		

$$Q_{ad} = C_p \dot{m} (t_3 - t_1) = Q_{heating} (1 - 1 / COP_{heating}) \quad (4)$$

$$Q_{re} = C_p \dot{m} (t_1 - t_2) = Q_{cooling} (1 + 1 / COP_{cooling}) \quad (5)$$

COP of the heat pump is provided by the manufacturer and is affected by the fluid temperature.

2.2 Procedure

When the characteristics of the building is known, the hourly loads can be predicted based on the meteorological data and the settings of indoor environment. From the maximum value of the heating and cooling loads, the range of L can be obtained through Eq. (6) [4]:

$$kL_{min} (\bar{t}_{max} - t_{soil}^{set}) = Q_{cooling} (1 + 1 / COP_{cooling, set}) \quad (6-1)$$

$$kL_{max} (t_{soil}^{set} - \bar{t}_{min}) = Q_{heating} (1 - 1 / COP_{heating, set}) \quad (6-2)$$

And the relation between L and A exists as Eq. (7) for satisfaction of the heating loads :

$$kL(t_{soil} - \bar{t}_{min}) + I_{set} A \eta_{set} = Q_{heating} (1 - 1 / COP_{heating, set}) \quad (7)$$

Then for the different L and A within their ranges, the energy consumption estimation in a year is estimated by Eq. (8).

$$E = E_s + E_w = \sum_{i=1}^{M_w} Q_{heating}^i / COP_w^i \Delta \tau_w^i + \sum_{i=1}^{M_s} Q_{cooling}^i / COP_s^i \Delta \tau_s^i \quad (8)$$

where, i is the index of the heating or cooling month. Q^i is the average load in the i^{th} month; $\Delta \tau^i$ is the duration for heating or cooling in the i^{th} month; and COP_w^i and COP_s^i are the average COP of heat pump, which can be obtained by Eq.(9) determined by the difference between the temperature of the water supplied to the evaporator (in the winter) or the condenser (in the summer) and the setting average temperature of the water supplied to the building.

$$COP = a + b(t_3 - \bar{t}_{load}^{set}) \quad (9)$$

here, a and b is the fitting parameters, which is determined through fitting the data provided by the heat pump manufacturer into Eq. (9).

Thus, through Eqs. (1)-(9), the yearly electricity consumption can be estimated and the optimized pair of L and A will be corresponded to the minimum E

In summary, the design procedure can be illustrated as Figure. 2.

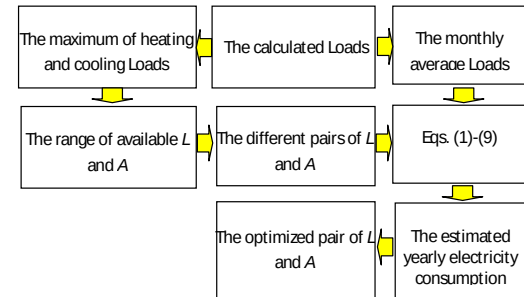


Figure 2 The design procedure

3.CASE STUDY

3.1 Case description

A building with 100m² located in Shenyang, China is used as the demo building. The heating period is from Nov. 15th, to Mar. 15th in the next year. And the cooling period is from May 1st to Sept. 30th. The heat pump is open from 7 A.M. to 10 P.M in the weekday.

A commercial software TRNSYS is employed to simulate the dynamic characteristics of the system. The system model is set up as shown in Figure 3.

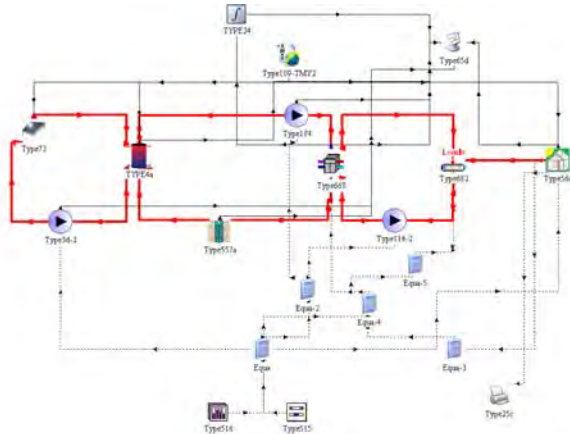


Figure 3 the setup of TRNSYS model

Based on the weather data, the hourly loads for the whole year are predicted as shown in Figure 4 with setting indoor room temperature as 18°C for the winter and 28 °C for the summer. From that, it is seen that the overall heating loads are much larger than the overall cooling loads. Therefore, when only GCHP is used for heating and cooling, the adsorbed heat will be much more than the rejected one, that is, it will lead to an unrecoverable drop of the earth temperature. The average monthly heating and cooling loads are also calculated for the system design shown in Figure 5.

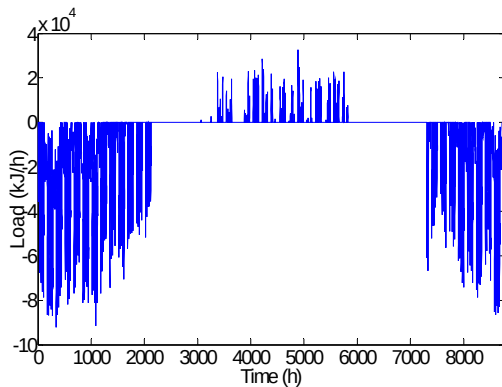


Figure 4 the hourly loads for a whole year

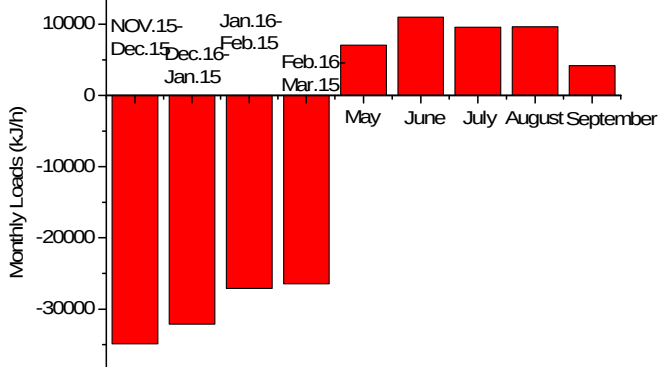


Figure 5. The monthly average loads

3.2.Optimization of L and A

For the design of the SAGCHP, the related parameters in the case are listed in Table 1.

Table 1 The parameters for the system design

$Q_{heating}^{max}$ (kJ h ⁻¹)	$Q_{cooling}^{max}$ (kJ h ⁻¹)	L_{max} (m)	L_{min} (m)
92148	32718	1146	146
η_c (%)	$COP_{heating}$	$\bar{t}_{max}$ (°C)	$\bar{t}_{min}$ (°C)
80	4	35	5
$\dot{m}$ (kg h ⁻¹)	$COP_{cooling}$	$\bar{t}_{load,summer}^{set}$ (°C)	$\bar{t}_{load,winter}^{set}$ (°C)
3000	5	7	35
a_{summer}	b_{summer}	a_{winter}	b_{winter}
6.47	-0.14	5.32	-0.061
I_{set} (kJm ⁻² h ⁻¹)	t_{soil}^{set} (°C)	C_p (kJ kg ⁻¹ °C ⁻¹)	k
1000	10.7	4.19	3.046

According to the procedure shown in Figure 2, the optimization of the pair of L and A is conducted and the optimized A and L are 60 m² and 346 m respectively. Through TRNSYS simulation, the electricity consumption in a whole year is compared for different pairs of L and A as shown in Figure 6. It is found that the energy consumption with the optimized pair of L and A is lower. That indicates the method can work well for the SAGCHP system.

The proposed method can be used for the design the depth of the borehole and the area of the solar collector based on the known building loads.

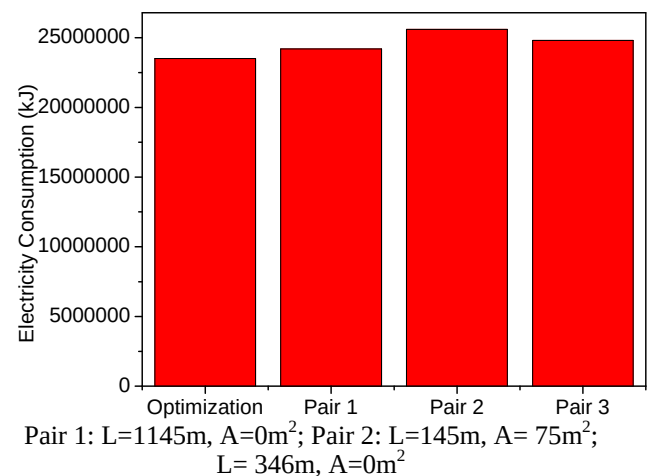


Figure 6. The comparison between different pairs of L and A. Conclusions

ACKNOWLEDGEMENTS

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A CASE STUDY OF BUILDING RETROFIT USING GROUND SOURCE HEAT PUMP TECHNOLOGY

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ABSTRACT

This paper presents a case study of ground water heat pump system (GWHP) for a commercial hotel building. The original residential air-conditioning and coal fired boilers are replaced by ground water heat pump in this hotel in order to save operating energy consumption and electricity fees. Based on the experimental results, the operation performance is simulated by the unit modeling. The results show that the GWHP system is particularly suitable for this hotel. In the present study, the experimental efficiency of ground water heat pump maintains 4.0-4.5 most of the time during the test two days under the retrofit technology. It could promote energy conservation and sustainability for buildings.

Keywords: Ground water heat pump; Energy efficiency; Retrofit technology.

1. INTRODUCTION

As one of the largest developing country in the world, China's rapid economic growth is accompanied with serious environmental challenges such as pollution, energy shortage and climate change [1]. Energy shortage is the most serious problem at present. The structure of energy consumption in China is dominated by coal consumption, however clean energies only represent a very small percentage. The GWHP's high energy efficiency and low environmental impact characteristics have already drawn a fair amount of attention as one of the clean energies in China. GWHP provides a new and clean way of heating or cooling buildings in the world.

It was estimated that, heating, ventilation and air-conditioning (HVAC) accounted for about 65% of the energy consumption in the Chinese building sector [2-3]. The HVAC system accounted for about 44% for the electricity consumption in the hotel in China, as it shown in fig.1 [4]. It is strongly desired to reduce energy consumption in HVAC systems. Professionals and policymakers have been making great efforts in this aspect.

The overall process of a building retrofit can be divided into five major phases [7]. The first phase is the project setup and pre-retrofit survey. The second phase comprises an energy audit and performance assessment (and diagnostics). The third phase is the identification of retrofit options. The fourth phase is site implementation and commissioning. The final phase is validation and verification of energy savings.

The retrofit technologies can be categorized into three groups, including supply side management,

demand side management [8-12], and change of energy consumption patterns, i.e. human factors. The retrofit technologies for supply side management include building electrical system retrofits and the use of renewable energy, such as solar hot water, solar photovoltaic (PV), wind energy, geothermal energy, etc., as alternative energy supply systems to provide electricity and/or thermal energy for buildings. In the last five years, there has been an increasing interest in the use of renewable energy technologies as building retrofit solutions due to the increased awareness of environmental issues. The use of renewable energy technologies may bring more benefits for commercial office buildings where a utility rate structure includes time-of-use differentiated electricity prices and demand charge is applied. Onsite generation of electrical and thermal energy is one area of this project, and the use of water source heat pumps falls into this category.

GWHP technology is regarded as an effective technology for this purpose. Chen et al. [5] investigated an underground water source heat pump system installed in a tall apartment building in Beijing, China. By analyzing this system for two years, operation methods and a controlling algorithm for the system were developed. Nam et al. [6] investigated the performance of a ground water source heat pump. This system depends on the temperature and depth of the water and its efficiency is much higher than that of the air source heat pump. However, the operation efficiency of GWHP is not satisfactory in some buildings due to many factors, including inefficient equipment, improper control schemes and many malfunctions happened in the building operation.

The project targets and client's concern for the environment have a significant impact on the selection

of retrofit technologies. This project aims at analyzing the energy consumption and energy performance of GWHP system. Then the electrical consumption and heat pump's COP were analyzed based on the test data. A simple economic analysis will also be performed, based on typical capital costs and energy tariffs, for replacing original residential air-conditioning and coal fired boilers with GWHP system.

2. SYSTEM DESIGN AND ANALYSIS

2.1 Overview

This hotel is located in Wuhan (latitude 30.52° N, longitude 114.32° E) and the area is 2200 m². This hotel has a total air-conditioning area of 1862m². The cooling load and heating load of this hotel is 208kW, 170kW, respectively, estimated by eQuest software.

In the original air-conditioning system, the residential air-conditioning is used as cold source of air-conditioning and rated COP of refrigerators is 3.8. The coal fired boilers are used as heat source supplying air-conditioning hot water and hotel hot water as well and the design efficiency of boiler is 78%. Fan coil and fresh air handling unit are used as terminal of air-conditioning.

The energy efficiency of the original air-conditioning is not very high. And this system is not environmental friendly. In order to save electrical consumption and reduce the CO₂ emissions, the original system is replaced by renewable and sustainable energy technology.

The hotel owners need pre-retrofit between ground source heat pump system, air-to-water heat pump system and VRV system. The cycle simulation of the ground water source heat pump was conducted to investigate the system characteristics, and the test data of air-to-water heat pump and VRV unit were also conducted for the comparison with the GWHP. Table 2 shows the performance comparison between the three different types of air-conditioning systems, including total energy consumption, electricity fee and heat pump's COP. The performance assessment is employed to benchmark building energy use by using green building rating systems. Based on the data in table 1, the GWHP system is selected as retrofit technology for this project.

This system mainly consists of the ground water heat pump system, an indoor air-conditioning system and a data acquisition system. A general view of the GWHP system is shown in fig.2. The specifications of the equipments appeared in Fig.1 are listed in Table 2. The GWHP system includes two MWH-020 type ground water heat pump units and two circulating pumps. The rated cooling and heating capacity of heat pump unit are 70 kW and 78 kW, and the corresponding input powers are 13.5 kW and 19.5 kW, respectively. It could operation in two modes, cooling and heating. The hot water GWHP system includes one MWH-020 type ground water heat pump unit and one circulating pump. The rated cooling/heating capacity and the

corresponding input power is the same with the GWHP system, and it could operation in three modes, supplying hot water, cooling and heating.

There are 28 rooms in this hotel, and there are average two beds in each room. The hot water demand quality for each bed per day is about 0.2m³. Hot water in this hotel is available 24 h a day all over the year. Hot water supplying mode is prior to cooling/heating mode.

The underground water source for the air-conditioning system comes from three wells, around the building, 23.6 m depth and 450 mm diameter. The length of vertical pipes is 70.8 m and the diameter is 219 mm, and the length of horizontal pipes is 100 m and the diameter is 80 mm. The water volume provided by each well is 30 T/h. In this project, the grit capacity in the well-water is 1/10,000.

The indoor air-conditioning system includes four types fan coil units (FP-51, FP-68, FP-102, FP-136) and fresh air handling units. Besides the ground temperature sensors, data acquisition system also includes several calorimeters. Each calorimeter consists of one flow meter with the minimum flow rate of 0.05 L/h and two Pt1000 temperature sensors with $\pm 0.1^{\circ}\text{C}$ accuracy. All power consumptions are recorded by the wattmeter.

2.2 Operation modes

Based on the requirements of residential owners, five operation modes are designed in the present GWHP system, including 1)Cooling and supplying hot water mode; 2)Cooling mode; 3)Heating and supplying hot water mode; 4)Heating mode; 5)Supplying hot water mode. Operation modes are switched by the control on different valves. The water pumps control strategies could be summarized as follows:

(1) Summer operation modes

i.Hot water heating circulating pump

a) When the hot water temperature in the tank reach at 50°C , opens the electromagnetic valve one and closes the electromagnetic valve two, then closes the hot water heating circulating pump, and the chiller unit supplies cooling normally;

b) When the hot water temperature in the tank lower than 42°C , opens the electromagnetic valve two and closes the electromagnetic valve one, then opens the hot water heating circulating pump, and the chiller unit supplies cooling normally;

ii.Submerged pump in ground source side

The submerged pump should be started up firstly no matter which chiller operates;

iii.Circulating pump in user side

The three water pumps could parallel operate simultaneously and operate separately.

(2) Winter operation modes

i.Hot water heating circulating pump

a) When the hot water temperature in the tank

reach at 50°C, closes the hot water heating circulating pump. The inside conversion valve of the chiller converses to terminal side and stops supplying hot water, and the chiller unit supplies heating normally;

b) When the hot water temperature in the tank lower than 42°C, the inside conversion valve of the chiller converses to hot water heating side and closes the terminal side supplying water, then open the hot water heating circulating pump, and the chiller unit heats water;

ii. Submerged pump in ground source side

The submerged pump should be started up firstly no matter which chiller operates;

iii. Circulating pump in user side

The three water pumps could parallel operate simultaneously and operate separately.

(3) Transitional season operation modes

i. Hot water heating circulating pump

a) When the hot water temperature in the tank reach at 50°C, closes the hot water heating circulating pump and submerged pump in ground source side;

b) When the hot water temperature in the tank lower than 42°C, opens the hot water heating circulating pump and submerged pump in ground source side.

ii. Submerged pump in ground source side

The submerged pump and the hot water heating circulating pump closes simultaneously.

iii. Circulating pump in user side

The circulating pumps in user side close all.

A major objective of the present GWHP system is to obtain an improved integration between the renewable sources available and residential thermal requirements, while guaranteeing a satisfactory level of comfort and quality of use under all situations.

2.3 Analysis of results

The performance of the water heat pump system was evaluated by COP. Eq. (1)-(2) is the definition of the unit and system COP. The unit COP is defined by the power consumption of only the heat pump system, which includes the compressor and fans for the heat exchangers. The power consumption for a heat pump comes from the compressor and the fans as shown in Eq. (3). For the system COP the power consumption, including the pumping power, is defined in Eq. (4). The cooling and heating capacity of the system is calculated using Eq. (5)-(6). Because the capacity is only measured in outdoor units, the heating and cooling capacity was calculated by adding or extracting the compressor power. The average uncertainty is $\pm 16.8\%$ [13].

In the supply side, more detail data for this air-conditioning system were recorded for two day, including inlet and outlet water temperature of the evaporator and condenser, the water flow mass of well,

the water temperature difference between inlet and outlet of well and energy consumption. In the demand side, test data included supply and return air temperature in some rooms on each floor. The performance of this GWHP system was test on Nov 26th 2012 to Nov 27th 2012. This could indicate the heating performance of this retrofit system. The cooling performance of this system is expected to test in July or August this summer in 2013.

The heating period for this hotel was from Nov 15th, 2012 to Jan 15th, 2013 (60 days). The energy consumption data was recorded in winter in 2012. From the data, it could conclude that the energy consumption of chiller and water pump is average 316.6 kW·h and 235.9 kW·h per day in winter, respectively. The total energy consumption of the GWHP unit is average 552.5 kW·h per day and the electricity fee is average CNY 541.43 per day based on the electricity price unit is 0.98 CNY/kW·h.

The test data for this GWHP system were analyzed from Nov 26th 2012 to Nov 27th 2012. Three GWHP units were opened in these two days. Energy consumption and system COP of GWHP system for two days are shown in Fig.2 and Fig.3, respectively.

Fig.3 shows that the energy consumption of GWHP system from Nov 25th to 26th is varied from 8:00 am to 20:00 pm during a day. The minimal energy consumption period is between 8:00 am to 12:00 am in the morning, and the maximal energy consumption period is between 18:00 pm to 20:00 pm in the evening. This result agrees well with the characteristic of the hotel.

Fig.4 shows the COP of GWHP system in these two days. It was also found that the heat pump's COP was 6.8 in maximum at the beginning of operation in Nov 26th but was later decreased to 3.9 at the end of day. The COP was 3.1 in minimum in the 19:00 pm and 20:00 pm in Nov 25th.

Fig.5 and 6 show the inlet and outlet water temperature flowing through the condenser and evaporator, respectively. Regardless of the temperature before the beginning of the night time hours, the final temperature of condenser outlet water increased to 50°C during night time hours. This implies that the heat pump has a sufficient design capacity.

During the heating operating conditions, water temperature changes vary with time, which implies that changes in COP vary widely over time. This conclusion could be validated in the COP of GWHP shown in Fig.3. The water temperature difference flowing through the condenser vary between 3°C and 10°C. The water temperature difference flowing through the evaporator vary between 4°C and 7°C. The temperature fluctuation in the evaporator was lesser than that in the condenser.

2.4 Conclusions

In this study an underground water heat pump system for cooling/heating purpose and instant hot water supply was designed for original air-conditioning retrofit in a hotel. An energy analysis was performed in

order to evaluate the efficiency of the system. The system was designed using a simple method which does not control the temperature of hot water and wastewater flowing into the heat pump's condenser and evaporator. Energy analysis was performed by test. The analysis results are as follows.

(1) This system is more economic and saves more energy compared with the original air-conditioning system, which needs both a cold and a heat-source. The estimate total electricity fee is less than air-to-water heat pump system and VRV system based on the comparative data in performance assessment phase.

(2) The operating COP of the heat pump was 4.0-5.0 most of the time during a day which has higher value than that of conventional heat pump system using ambient air heat source.

(3) The heat pump's COP varies widely, which has a maximum value 6.8 at the beginning of operation during morning time hours and a minimum value 3.1 at

the end of night. Therefore, the system needs to be re-configured to control the temperature of water flowing into the heat pump.

(4) The water temperature difference flowing through the condenser and evaporator is 7°C and 3°C, respectively. The important thing that should be considered for the system design is to decrease the temperature difference between condenser and evaporator working fluids during the heat charging process by the heat pump.

It is therefore concluded that this system is particularly suitable for this hotel in Wuhan city with a correction of temperature control flowing into the evaporator and condenser, respectively. The operating cost saving compared with the original air-conditioning system is an additional advantage of the system.

2.5 Tables

Table 1 Performance comparison between three different air-conditioning systems

Air-conditioning type	Ground Water Source Heat Pump Unit (with hot water)		Air-to-Water Heat Pump Unit (without hot water)		VRV Unit (without hot water)	
	Winter	Summer	Winter	Summer	Winter	Summer
Season						
Air-conditioning area(m ²)	1862	1862	1862	1862	1862	1862
Heating (cooling) load (kW/ m ²)	234	210	234	210	234	210
Supplying heating (cooling) days (day)	60	180	60	180	60	180
Operate hours (h)	24	24	24	24	24	24
Average heating (cooling) load coefficient	0.739	0.664	0.739	0.664	0.739	0.664
Season heating (cooling) load (kW·h)	248976	602438	248976	602438	211630	481950
Electricity consumption of season heat pump (kW·h)	62244	115324	102203	173937	105815	185365
Energy collection、release consumption (kW)	22.32	22.32	7.45	7.45	7.72	7.72
Season energy collection、release and water pump consumption (kW·h)	23753	64042	7927	21373	8218	8218
Season total energy consumption (kW·h)	85997	179366	110130	195310	114033	193584
Season equivalent energy consumption per square meters (kW·h)	46.19	96.33	59.15	104.89	61.24	103.97
Electricity unit price (CNY/ kW·h)	1.000	1.000	1.000	1.000	1.000	1.000
Season equivalent electricity fee per square meters (CNY)	46.19	96.33	59.15	104.89	61.24	103.97
Season electricity fee (CNY)	85997	179366	110130	195310	114033	193584
Total electricity fee (CNY)	265362		305440		307617	
Initial investment (CNY)	525000		456200		532500	
COP	5.22		2.3		2.6	

Table 2 Equipments list of GWHP system

No	Type	Specification	Number	Remark
1	GWHP unit	MWH-020, Qcooling=70kW,Ncooling=13.5kW, Qheating=78kW,Nheating=19.5kW	2	10/15℃, 40/45℃;25/30℃, 12/7℃
2	Hot water GWHP unit	MWH-020, Qcooling=70kW,Ncooling=13.5kW, Qheating=78kW,Nheating=19.5kW	1	10/15℃, 40/45℃;25/30℃, 12/7℃
3	Terminal circulating pump	Q=15m ³ /h,H=35m,N=4kW	4	one for standby
4	Hot water circulating pump	Q=12.5m ³ /h,H=12.5m,N=1.1kW	2	one for standby
5	Expansion tank	1.0m×1.0m×1.0m	1	
6	Full auto water treatment instrument	40t/h	1	
7	Desander	XLC-40B	1	
8	Water meter	40t/h	1	
9	Hot water tank	2.0m×1.2m×1.7m	1	
10	Hot water supply pump	3.3kW	2	
11	Water collector	DN100	1	
12	Water distributor	DN100	1	
13	Submerged pump	Q=30m ³ /h,H=40m,N=5.5kW	2	

2.6 Figures

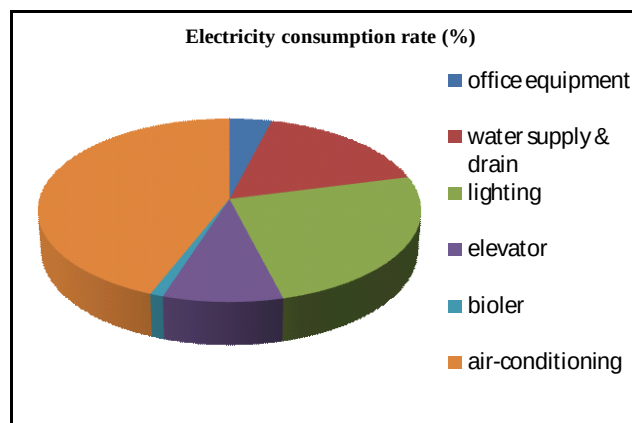
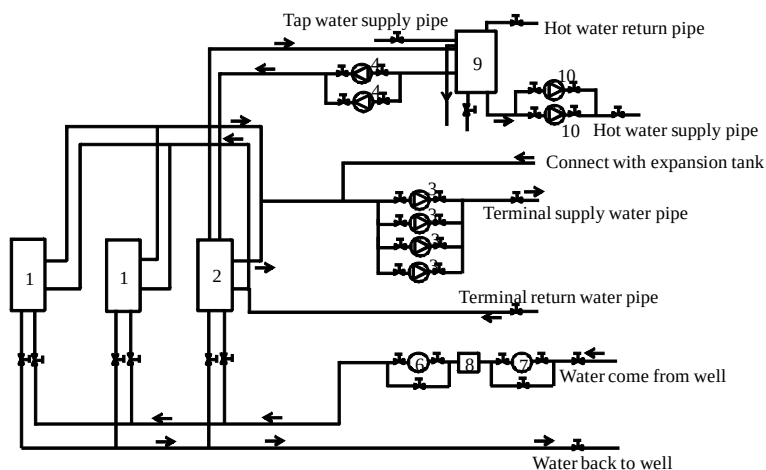


Figure.1 The electricity consumption rate of hotel building [1].



1:GWHP unit;2: hot water GWHP unit;3: terminal circulating pump;
4: hot water circulating pump;6: full auto water treatment instrument;
7: desander;8: water meter;9: hot water tank;10: hot water supply pump
Figure.2. Principle diagram of GWHP system.

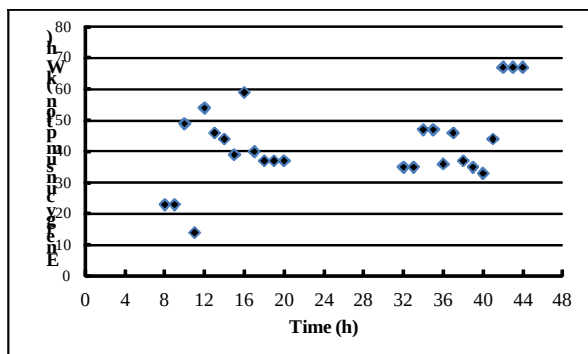


Figure.3. Energy consumption of GWHP system for two days.

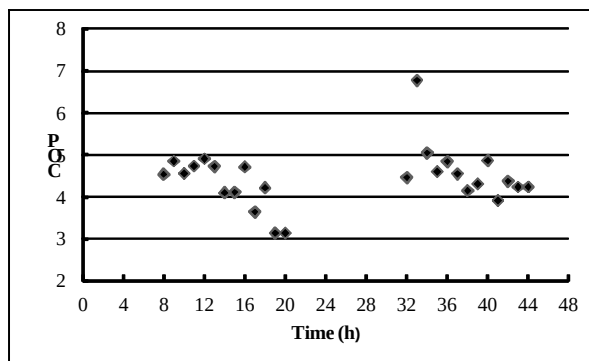


Figure.4. COP of GWHP system for two days.

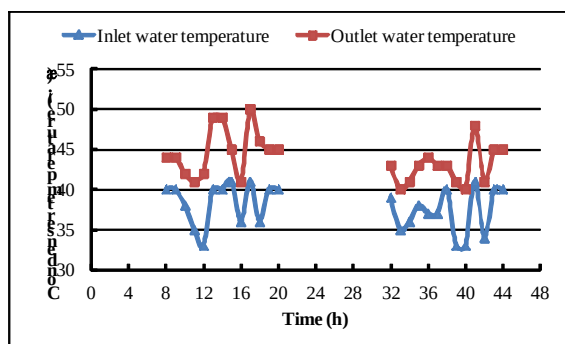


Figure.5. Condenser inlet and outlet water temperature.

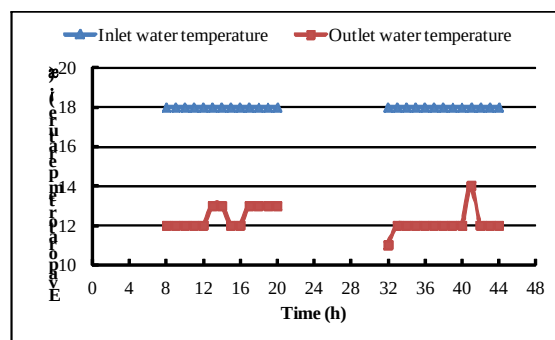


Fig.6. Evaporator inlet and outlet water temperature.

4.1. Formula

$$COP_{unit} = \frac{q}{w_{hp}} \quad (1)$$

$$COP_{sys} = \frac{q}{w_{sys}} \quad (2)$$

$$w_{hp} = w_{comp} + w_{fan} \quad (3)$$

$$w_{sys} = w_{hp} + w_{pump} \quad (4)$$

$$q_c = \dot{m} \cdot c_p \cdot \Delta T - w_{hp} \quad (5)$$

$$q_h = \dot{m} \cdot c_p \cdot \Delta T + w_{hp} \quad (6)$$

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THE IMPACT OF ENERGY SAVING TECHNIQUES ON A SMALL ECONOMY- A STOCHASTIC ANALYSIS

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ABSTRACT

This paper investigates the impact of Energy Saving Techniques on the economic performance of a small economy. Both the effects of unanticipated shocks and changes in the mean and variance of the probability distribution generating these disturbances are discussed. In all cases, the key element determining the response of the economy is the effect on the rate of growth of real wealth, to which all other real quantities are directly tied in equilibrium. Conditions for the Harberger-Laursen-Metzler effect to hold are discussed. The impact of these changes on economic welfare, as measured by expected discounted utility of the representative agent is also addressed.

Keywords: Energy Saving Techniques; Harberger-Laursen-Metzler effect.

1. INTRODUCTION

From the literature on technological change, the direction of technological change will be affected to energy-saving by the price of energy relative to other goods rises, government's sustainable energy policies and regulations, research and development (R&D) effort. IEA(2012) found a reduction in global energy intensity (energy share of GDP) of 0.5% per year over the last decade, but 1.8% to 2035. Narayanan and Sahu (2010) found the energy intensity of the aggregate manufacturing industries is decreasing from 1990 to 2008. Hooker (2002) proved the declining intensity (energy-saving) of U.S. for 1949-1999, and the intensity was 1 in the first quarter of 2000.

For the price of energy relative to other goods rises, the production function or capital goods would contain more energy-saving technologies, so the direction of technological change will be affected to energy-saving. International Energy Agency (IEA, 2012) set electricity prices annual increasing rate at 15% (by 2035) for higher fuel prices, increased use of renewables and, CO₂ pricing, etc.. Firms will use labor to produce on energy-saving time, and replace their capital goods (like cars) with more energy-saving models available on the market. (Newell, et al., 1999)

Government's sustainable energy policies and regulations have affected the energy-saving technologies available on the market. Many governments have used several kinds of policy tools to fund basic R&D, support the initial deployment, reduce and overcome the market barriers, and provide their suppliers and users the right signals of energy-saving technologies. The policy tools are subsidising R&D, technology and performance standards, subsidising dissemination, taxes and cap-and-trade systems,

voluntary agreements, policy mixes. (IEA, 2003) Jaffe, et al (2005) reviewed U.S. federal climate technology initiatives (Based on the budget of 2004), and found that energy conservation is related to 41% of R&D spending which is about \$1.3 billion/year, and 34% of adoption spending (\$1.0 billion/year) goes for state energy efficiency grants. IEA (2012) proposed energy efficiency increasing (energy-saving) policies which are increasing the visibility and affordability of energy efficiency, prioritizing energy efficiency, making energy efficiency mainstream, real, realizable.

R&D effort is a key element of technical change (more energy-saving), and it could drive their costs down, as International Energy Agency (IEA, 2000) use experience curves and the progress ratios of the power-generating sector to understand the reason why the new technology become cost-saving over time, and the progress ratio is the reduction ration of its installed price (=cost) as the doubling of cumulative installed energy technology. The progress ratio for photovoltaic modules on the world market in the period 1976-1992 was 82% (reduced the price by 18%); it for Danish wind turbines in the period 1982-1997 was 96%; and it for United States (European Union) wind power in the period 1985-1994 (1980-1995) was 68% (82%).

There are many economic literature on technical change, but little discussed the impact of energy technical change (more energy saving) and the rising price of energy on macroeconomic variables (the growth rate of real wealth, the Harberger - Laursen - Metzler (HLM) effect, etc.), the purpose of this paper is to discuss the impact of more energy saving techniques on the economic performance of a small economy.

The economic literature on technical change could be briefly classified as the microeconomic study of innovation or the macroeconomic study of growth, and

concluded that energy technical change might spur economic growth. From the views of Arrow (1962), Kamien and Schwartz (1982), Reinganum (1989), Clarke and Weyant (2002), more energy-saving technologies would be the endogenous equilibrium of the economic system which could be affected by policies, but they might be underinvested by private sectors. From the discussions of Romer (1986, 1990), Grossman and Helpman (1991), Aghion and Howitt (1992), Clarke and Weyant (2002), Acemoglu (2009), the spillovers of energy-saving technologies might be a source of steady, long run economic growth. IEA(2012) estimated energy efficiency (more energy-saving technologies) could improve energy security and economic growth, and boost cumulative global economic output through 2035 by \$18 trillion.

Many papers have explained why energy price fluctuations would impact on macroeconomic performance. (Jalles (2009)) The energy price (increasing and stochastic) would transmit through the routes of the labor market, aggregation and allocation, interest rate, petroleum product market. (Davis and Haltiwanger (2001), Jones, et al (2004)) Hamilton (1983) extended that increasing energy price might be a factor of U.S. macroeconomic performance. Davis and Haltiwanger (2001) proved the asymmetrical effects of oil price shocks on the creation and destruction of U.S. manufacturing jobs.

For a small economy, energy price fluctuations would be the terms of trade (TOT) shocks, and the literature has proved how validity or invalidity of HLM effect depends upon the economic factors (the specific formulations) of the model, the points of this paper are energy technical change (energy saving), the rising price of energy, and their cross-variance. HLM effect is the specific effect of terms of trade (TOT) shocks on the economy, it means when the levels of investment and income are given, a deterioration in TOT would raise real expenditure, reduce savings, and cause a deterioration in the current account. (Harberger(1950) and Laursen and Metzler(1950))

The time path of energy technology and energy price should be a mix process, so this paper employs a continuous-time, representative household, stochastic optimizing model. Many papers used deterministic models to discuss the economic factors of HLM effect. Obstfeld (1982), Svensson and Razin(1983) used a household's intertemporal utility optimization model to show HLM effect would be contradicting when time preference is increasing in utility. Persson and Mansoorian(1993) constructed a habit persistence and infinite horizon model to show HLM effect holds if the marginal utility of consumption is increasing. Ikeda(2001) used a model with weakly non-separable preferences to find HLM effect occurs if consumers' preference toward imports is wealth enhanced.

This paper is organized as follows. Section 1 is the introduction. Section 2 develops the analytical framework that the impact of energy saving techniques on a small economy. Section 3 analyses macroeconomic variables of a small economy by energy saving techniques' impact. The last section offers conclusions.

2. THE ANALYTICAL FRAMEWORK

This paper assumed a small open economy with exogenous TOT and interest rate, two perfectly competitive and complete market (commodity and financial asset), a representative household (no firm), and no externality and trade barriers (no tariffs and freights), three goods (energy and final good in foreign country which are imported from abroad, final good in home country). A representative household uses energy, labor and capital to produce final good, and consumes final good and imported good.

The path of energy price should be a mix and compiled process with the increasing and stochastic price of oil and other energy products, and the mean and volatility of energy price should impact macroeconomic variables, so the relative price of energy (P_e , the price of the final good in home country is fixed as 1) is exogenous given and generated by the Brownian motion process, and its formula is:

$$dP_e / P_e = \alpha + du_e, \quad (1)$$

where α is the instantaneous expected change rate of the relative energy price and du_e is a random variable with mean zero and variance $\sigma_e^2 dt$. Sadorsky (1999) used a vector autoregression show to show oil prices and oil price volatility could affect economic activity (stock return).

The relative price of the final good in foreign country (P_f) is

$$dP_f / P_f = du_f, \quad (2)$$

where du_f is a random variable with mean zero and variance $\sigma_f^2 dt$. And the relative price of the imported goods (P_i) is $P_i = P_e^{\theta_1} P_f^{1-\theta_1}$ which is aggregated by the prices of energy and final good in foreign country, and θ_1 is the ratio of the imported value of energy by the total imported value, so $0 < \theta_1 < 1$. From (1) and (2), TOT should be $1/P_i$, and followed by the Brownian motion process:

$$dP_i / P_i = \beta dt + \theta_1 du_e + (1 - \theta_1) du_f, \quad (3)$$

where $\beta = \alpha\theta_1 - \theta_1(1 - \theta_1)(\sigma_e^2 + \sigma_f^2)/2$ is the instantaneous expected change rate of TOT.

Final good in home country (Y_h) is produced by the inputs of labor(L), capital (K) and energy (E), let $L=1$, so its production function is:

$$Y_h = T_e f(T_1, K, P_e E) = T_e T_1^{\theta_2} (K + P_e E)^{1-\theta_2}, \quad (4)$$

where T_e is energy technical displacement; T_1 is effective labor force which could be seen as the labor augmenting technical change; $P_e E$ is the inputted (=imported) value of energy; $0 < \theta_2 < 1$.⁹

⁹Labor, capital and energy are the necessity to produce the final good in home country, so

T_e and T_1 are exogenous variables, and energy technical displacement should be a mix process which affected by market experience, government policies, and the development path of technical development, and etc., so $dT_e/T_e = \delta dt + du_{Te}$, $dT_1/T_1 = du_{T1}$ are assumed, where dT_e and du_{T1} are random variables with mean zero and variance $\delta_{Te}^2 dt$ and $\delta_{T1}^2 dt$. IEA (2003) suggest that technical change is a cyclical process which is affected by the feedback between market experience and further technical development. Hooker (2002) showed U.S. energy intensity declined fairly steadily during the period of 1949~2000, with several small rises but more rapid declines. IEA (2000) referred an R&D breakthrough in the production as technology structural change which makes a radical change in technology, like new temperature-resistant materials for gas turbines. Roehrl and Riahi (2000) used the cost assessment of greenhouse gas emissions mitigation to study of technology dynamics, and found that technological change would develop in "path dependent" direction, which cost is increasing to change its course by a particular initial point.

From (1),(4),energy technical displacement, and effective labor force,the stochastic production function is

$$dY_h = Y_h \left[1 + \delta - \theta_2 (1 - \theta_2) (\delta_{T1}^2 + E_R^2 \sigma_e^2) / 2 \right] + du_{Te} + \theta_2 du_{T1} + (1 - \theta_2) E_R du_e \quad (5)$$

where $E_R = P_e E / (K + P_e E)$.

The model is real (no money), and the representative household holds two securities (traded bonds and claims on capital) which are paid by a exogenous interest rate (r), so the return rate of capital (MP_{K+PeE}) should be equal to interest rate for the long-term equilibrium condition of financial asset market, which is

$$MP_{K+PeE} = (1 - \theta_2) Y_h / (K + P_e E) = r \quad (6)$$

The instantaneous utility of the representative household is decided by the consumption of final good in home country and in foreign country ($C_h dt$, $C_f dt$), which form is $U(C_f, C_h) = (C_h^{\theta_3} C_f^{1-\theta_3})^{\rho_1} / \rho_1$, where $-\infty < \rho_1 < 1, 0 < \theta_3 < 1$. And the value function of the representative household at t ($V_t(C_h, C_f)$) is its expected value of discounted utility at t , which form is

$$V_t(C_h, C_f) = \text{Max}_{C_t} \int_t^\infty (C_h^{\theta_3} C_f^{1-\theta_3})^{\rho_1} / \rho_1 e^{-\rho_2 s} ds \quad (7)$$

where ρ_2 is time preference rate of the representative household.

The representative household's real wealth (W) is given by: $W = K + P_e B$, and let $N_{K+PeE} = (K + P_e E) / W$, $N_B = P_e B / W$ are shares of capital and bond in portfolio held of the representative household, so the portfolio shares adding up condition is:

$$N_{K+PeE} + N_B = 1. \quad (8)$$

The instantaneous returns of capital and bond (dR_{K+PeE} , dR_B), and the instantaneous consumption values of final good in home country and in foreign country ($C_h dt$, $P_f C_f dt$) are the sources of wealth accumulation. For (5), (6) and $dR_{K+PeE} = dY_h / (K + P_e E)$, the instantaneous returns of capital is

$$R_{K+PeE} = \frac{r \{ [1 + \delta - \theta_2 (1 - \theta_2) (\sigma_{T1}^2 + E_R^2 \sigma_e^2) / 2] dt + du_{Te} + \theta_2 du_{T1} + (1 - \theta_2) E_R du_e \}}{1 - \theta_2} \quad (9)$$

For (3) and $dR_B = [r P_f B dt + d(P_f B)] / P_f B$, the Ponzi-game condition: $\lim_{t \rightarrow \infty} E[(1 + r)^{-t} P_f B] = 0$, the instantaneous returns of bond is

$$dR_B = (r + \beta) dt + \theta_1 du_e + (1 - \theta_1) du_f. \quad (10)$$

For (9) and (10), the stochastic wealth accumulate equation is

$$\begin{aligned} dW &= W [(N_K + N_{PeE}) dR_{K+PeE} + N_B dR_B - (N_C + N_{PeE}) dt] \\ &= W \{ [\Phi N_{K+PeE} + (r + \beta) N_B - N_C] dt + r N_{K+PeE} (du_{Te} + \theta_2 du_{T1}) / (1 - \theta_2) \\ &\quad + (1 - \theta_1) N_B du_f + (r N_{K+PeE} E_R + \theta_1 N_B) du_e \} \end{aligned} \quad (11)$$

where

$$\varphi = r [1 + \delta - \theta_2 (1 - \theta_2) (\sigma_{T1}^2 + E_R^2 \sigma_e^2) / 2] / (1 - \theta_2),$$

and $N_C = (C_h + P_f C_f) / W$ is share of consumption in portfolio held of the representative household.

The optimization is to choose C_h , C_f , N_{K+PeE} , N_B to maximize (7) subject to (2), (3), (8), (11) and follows the procedure set out in Turnovsky (1998). Defining aggregate consumption (C) by $CC_h + P_f C_f$. The first order optimality conditions:

$$C_h = \theta_3 C, \quad (12a)$$

$$C_f = (1 - \theta_3) C, \quad (12b)$$

$$\begin{aligned} N_C &= \{ \rho_2 - \rho_1 [\Phi N_{K+PeE} + (r + \beta) N_B] + \beta \rho_1 (1 - \theta_3) \} / (1 - \rho_1) \\ &\quad - \{ \rho_1^2 (1 - \theta_3) N_B + \rho_1 (1 - \theta_3) [\rho_1 (1 - \theta_3) - 1] / 2 (1 - \rho_1) \} [\theta_1^2 \sigma_e^2 + (1 - \theta_1)^2 \sigma_f^2] / (1 - \rho_1) \\ &\quad + \{ \rho_1 \{ r_2 N_{K+PeE} (\sigma_{Te}^2 + \theta_2^2 \sigma_{T1}^2) / (1 - \theta_2)^2 + (1 - \theta_1)^2 N_B^2 \sigma_f^2 + (r N_{K+PeE} E_R + \theta_1 N_B)^2 \sigma_e^2 \} / 2 \} \end{aligned} \quad (12c)$$

$$\begin{aligned} \Phi &= (1 - \rho_1) N_{K+PeE} \{ r^2 (\sigma_{Te}^2 + \theta_2^2 \sigma_{T1}^2) / (1 - \theta_2)^2 + r E_R (r N_{K+PeE} E_R + \theta_1 N_B) \sigma_e^2 \} \\ &= r + \beta + (1 - \rho_1) [\theta_1 (r N_{K+PeE} E_R + \theta_1 N_B)^2 \sigma_e^2 + (1 - \theta_1)^2 N_B^2 \sigma_f^2] \\ &\quad - \rho_1^2 (1 - \theta_3) [\theta_1^2 \sigma_e^2 + (1 - \theta_1)^2 \sigma_f^2] \end{aligned} \quad (12d)$$

(12a) and (12b) describe consumptions of final good in home country and in foreign country as fixed fractions of overall consumption expenditure. The mean of (12c) is the aggregate consumption-wealth ratio depends upon portfolio shares and the parameters of preference, permanent and temporary shocks, value

¹⁰ Let .

function and production function of the representative household.

Solving (8) and (12d) for the equilibrium portfolio shares of capital and bond, like N_B is

$$N_B = \{ (r + \beta - \Phi) - \rho_1(1 - \theta_3) [\theta_1^2 \sigma_e^2 + (1 - \theta_1)^2 \sigma_f^2] \} / R_a$$

$$(1 - \rho_1) [r^2 (\sigma_{Te}^2 + \theta_2^2 \sigma_{Ti}^2) / (1 - \theta_2)^2 + r E_R (r E_R - \theta_1) \sigma_e^2] / R_a$$
(13)

Where

$$R_a = (1 - \rho_1) [r^2 (\sigma_{Te}^2 + \theta_2^2 \sigma_{Ti}^2) / (1 - \theta_2)^2 + (r E_R - \theta_1)^2 \sigma_e^2 + (1 - \theta_1)^2 \sigma_f^2]$$

is the risk adjudge factor of N_B . (13) could highlight the speculative and hedging behavior on bond of the representative household. The speculative component depends upon the difference between differential expected real return rates of bond and capital, and the hedging component depends upon the variances of energy price, the final good in foreign country, energy technical displacement, and labor augmenting technical change.

For (8), (12c) and (13), the equilibrium N_{K+PeE} , N_B , N_C should be constancy, then the growth rate of the representative household is:

$$dC/C = d(K + PeE)/(K + PeE) = dP_B/P_B$$

$$= [\Phi N_{K+PeE} (r + \beta) N_B - N_C] dt + (1 - \theta_1) N_B du_f$$

$$+ r N_{K+PeE} (du_{Te} + \theta_2 du_{Ti}) / (1 - \theta_2) + (r N_{K+PeE} E_R + \theta_1 N_B)$$
(14)

which expected value should be equal to the difference between the earning rate and the shares of aggregated consumption of wealth. From (14), the larger is positive permanent shock of energy technical displacement (δ), the faster is the growth rate.¹¹

3. MACROECONOMIC VARIABLES ANALYSIS

Many papers have proved the macroeconomic effects (HLM effect) of permanent, temporary, anticipated, or unanticipated TOT shocks are different. (Svensson and Razin(1983), Svensson (1985), Bean (1986), Sen and Turnovsky (1989), Turnovsky (1993)) Svensson and Razin(1983) show that a temporary (permanent) TOT deterioration implies a deterioration (ambiguous effect) of the trade balance.

4. CONCLUSION

The results of this paper is more complicated than Turnovsky (1998), for this paper consider more permanent and temporary shock of energy price, energy technical displacement, and labor augmenting technical change, than Turnovsky (1998), so the results of this papershould be more real and give more detailed explanations of the impact of energy saving techniques on a small economy.

IEA(2012) estimated electricity prices with significant regional differences as the highest prices of European Union and Japan.

For energy technical change (more energy saving) should be with regional differences, so it could affect the competitive positions and export possibilities of nations or industries through their different energy costs. Narayanan and Sahu (2010) found the energy intensities of the aggregate manufacturing, chemical, diversified, food & beverages, machinery, metals & metal products, miscellaneous manufacturing, non-metallic mineral products, textiles, transport equipment industries from 1990 to 2008 are 4.51, 3.13, 6.26, 2.50, 1.52, 8.52, 9.62, 14.74, 7.14, 1.88.

There might be the different impact of energy technical change and the rising price of energy on macroeconomic variables of a small and large economy. Kilian (2008) proved exogenous oil supply shocks made little difference for the macroeconomic variables, and explained its reason by the endogenous responses of oil producers elsewhere and U.S. policy makers, etc..

The covariance between energy technical change and the rising price of energy ($COV(dP_e, dT_e)$) should be positive, for the increasing energy prices will lead to the induced innovation of capital goods are less energy intensive in use (more energy-saving technologies). Hicks (1932) stated a macroeconomic hypothesis about induced innovation that directed to economizing the use of a factor which has become relatively expensive. There is a considerable literature thought technological change of energy-saving is a global phenomenon which households, firms and governments would all develop or use cars, appliances, or industrial equipment that produce more outputs per unit of energy consumed. From Schumpeter(1939), when energy prices rise, the products with energy-saving items would diffuse faster than it otherwise would.

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¹¹ The absolute value of should be small enough to be consistent the transversality condition.

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THE STRUCTURE OF RURAL HOUSEHOLD ENERGY CONSUMPTION AND RENEWABLE ENERGY POTENTIAL IN LAM NAM NAN SELF SUPPORTING SETTLEMENT, UTTARADIT PROVINCE, THAILAND

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ABSTRACT

According to Thailand energy development plan, Ministry of Energy established the roadmap as strategy to promote renewable and alternative energy development for 25% in 10 years (AEDP: 2012-2021) through a strategic issues of promoting the community to collaborate in broaden producing and consuming renewable energy. Therefore, this study aimed to survey and analyze the structure of energy consumption and assess the potential of renewable energy to encourage suitable renewable energy consumption for the community as a pattern for other communities. Lam Nam Nan Self Supporting Settlement in Uttaradit Province was selected for this research. The samples of 393 households were applied by stratified random sampling. The data was gathered by using survey forms to examine the structure of energy consumption as well as energy cost. The potential of renewable energy was also assessed. The findings indicated that most of the samples or 59.03% were agriculturalists and earned average income of 59,112 Baht/year. The average household income was around 113,958 Baht/year. For the analysis of energy consumption structure, it was found that the most energy consumption were commercial energy such as diesel oil at 30.74% and electricity at 21.24%, followed by renewable energy such as charcoal at 16.29%. In terms of energy end-uses, it was demonstrated that transportation mostly consumed energy at 39.40%, followed by cooking at 34.79%, and other electrical appliances at 14.64%. The average energy consumption per household was 804.92 kgoe/year, accounted for energy cost of 24,862 Baht/year or 21.82% of household income.

Regarding to the potential assessment of renewable energy, analyzed results stated the potentials of solar energy 371,196 toe/year, wind energy 876 toe/year, biomass 19,342 toe/year and biogas 450 toe/year. Renewable energy reform would be an effective way to improve the energy consumption structure, for instance, solar energy for agricultural products drying, solar power generation, and agricultural waste as briquettes as well as biogas for cooking. These findings have important implications for the design and effectiveness of rural household energy consumption designed to promote renewable energy and to eliminate greenhouse effect, regarding to the country's development plan.

Keywords: Energy Consumption Structure, Renewable Energy Potential, Rural Household, Country's Development Plan

1. INTRODUCTION

Thailand is a net importer of crude oil and a net exporter of petroleum products. The country imports over 60 % of its total petroleum needs and almost 85 % of its crude oil consumption. Thus, Ministry of Energy, as assigned by Thai government, provided a renewable energy development plan to promote the use of renewable energy by 25 % or 25,000 toe of total energy consumption in 2021 [1]. This plan aimed to enhance national energy security, especially at community level by emphasizing green technology with renewable

energy in smart community. According to previous surveys and researches, there were some renewable energy resources in Thailand about [2, 3, 4, 5]. Moreover, the targets of energy consumption in 2021 would be accomplished by applying several kinds of potential renewable energy such as solar energy 2,000 MW, wind energy 1,200 MW, and biomass 3,630 MW [1]. Therefore, Thailand has to define suitable plan for energy consumption to encourage the community to participate into energy generation and renewable energy consumption.

NOMENCLATURE

Abbreviation		Symbols	
AEDP	Alternative Energy Development Plan	A_t	Total areas (km ²)
LNNSSS	Lam Nam Nan Self Supporting Settlement	A_f	Forest areas (km ²)
		A_a	Agricultural areas (km ²)
		p	Air density (kg/m ³)
Symbols		V	Average wind speed (m/s)
toe	Tonne of oil equivalent	A	Rotor areas (m ²)
kgoe	Kilograms of oil equivalent	C_p	Efficiency factor
MW	Megawatt	Y	Yield of biomass (kg/year)
MJ	Megajoule	R	Ratio of agricultural waste
m	Meter	H	Heating value of agricultural waste (MJ/kg)
km	Kilometer	N	Number of animals (animal)
hr	Hour	K	Ratio of the average animal weight per livestock
S	Second	W	Volume of wastewater (l/day)
l	liter	D	Retention time of wastewater (day)
P_{solar}	Potential of solar energy (toe/year)	F_g	Ratio of biogas
P_{wind}	Potential of wind energy (toe/year)	M	Amount of manure (kg/animal)
$P_{biomass}$	Potential of biomass (toe/year)	F_m	Ratio of manure
P_{biogas}	Potential of biogas (toe/year)	F_v	Ratio of volatile solids
RI	Average solar radiation intensity (MJ/m ² -hr)		

Based on energy database, it indicates that there are 3 main types of energy consumption in Uttaradit province – electricity, petroleum products, and renewable energy. It is noticed that renewable energy is highly consumed at 33.23 % of total energy consumption [6]. The community has utilized renewable energy and it is accessible to promote renewable energy use. For these reasons, the area in Lam Nam Nan Self Supporting Settlement (LNNSSS) is selected as a case study of this research. Furthermore, the LNNSSS is allocated by the government for the people who have emigrated from flooded area regarding to Sirikit Dam construction. The LNNSSS covers the area of 297.34 km² including 12 sub-districts of Muang Uttaradit District (6 sub-districts), and Ta Pla District (6 sub-districts). More than 80 % of the area is forests and agricultural land [7]. Hence, the aims of research are to investigate and analyze the structure of household energy consumption and assess the potential of renewable energy to encourage suitable renewable energy consumption for the community as a pattern for other communities.

2. METHODOLOGY

This research was to conduct a survey to explore households energy consumption and assesse potential of renewable energy. Questionnaires were used as a tool to examine energy consumption, cost of energy consumption, and activities related to energy consumption as divided below;

Data on energy consumption and cost of energy consumption separated into 2 kinds; commercial energy including electricity, LPG, diesel oil, gasoline 91, gasohol 91, gasohol 95, and renewable energy consisting of charcoal, firewood, and biogas.

Data on energy consumption and cost of energy consumption classified into 6 activities such as cooking,

lighting, electrical appliances, agricultural work, industrial equipment, and transportation.

The samples of this study were local households in chosen community. Firstly, sample size was calculated by applying Taro Yamane's formula [8] to define 393 households as respondents. Then probability theory and stratified random sampling [8] were utilized to randomize the households. An analysis of structure of energy consumption in community was made by computing the quantity of energy consumption and the cost of energy consumption during the data collection.

To the assessment of potential of renewable energy, it was distributed into 4 types – solar energy, wind energy, biomass, and biogas. The data was gathered from primary and secondary sources from concerning government agencies and calculated by the equations as shown below;

Potential of solar energy [9-12] can be calculated by equation (1) as follows:

$$P_{solar} = RI(A_t - A_f - A_a) \quad (1)$$

where P_{solar} is potential of solar energy, RI is average solar radiation intensity, A_t is total areas, A_f is forest areas, and A_a is agricultural areas.

Potential of wind energy [13-15] can be calculated by equation (2) as follows:

$$P_{wind} = 0.5pV^3AC_p \quad (2)$$

Where P_{wind} is Potential of wind energy, p is air density, V is average wind speed, A is rotor areas, and C_p is efficiency factor.

Potential of biomass [12, 16-18] can be calculated by equation (3) as follows:

$$P_{biomass} = YRH \quad (3)$$

Where $P_{biomass}$ is Potential of biomass, Y is yield of biomass, R is ratio of agricultural waste, and H is heating value of agricultural waste.

Potential of biogas [19-20] can be calculated by equation (4) and (5) as follows:

Potential of biogas from the manure of swine

$$P_{\text{biogas}} = NKWDF_g \quad (4)$$

Where P_{biogas} is Potential of biogas, N is number of animals, K is ratio of the average animal weight per livestock, W is volume of wastewater, D is retention time of wastewater and F_g is ratio of biogas.

Potential of biogas from the manure of cow, buffalo, chicken, and duck

$$P_{\text{biogas}} = NMF_m F_v F_g \quad (5)$$

Where P_{biogas} is Potential of biogas, N is number of animals, M is amount of manure, F_m is ratio of manure, F_v is ratio of volatile solids, and F_g is ratio of biogas.

3. RESULTS AND DISCUSSION

3.1 General Information of Respondents

This part covered educational level, occupation, average income per capita per year, average household income per year, and average number of household members. The results found that most local samples or 54.20 % graduated Primary school years 4, followed by Secondary school years 6 or 11.20 % and Primary school years 6 or 10.18 %. The information of respondents' main occupation. It showed that most local people or 35.11 % in the community were agriculturalists and they mostly grew cashew nut, one of Uttaradit Province's popular plants. In addition, some people or 22.39 % were general workers. However, the rest of 14.50 % were unemployed. They were housewives or househusbands because other family members had already earned a living. Furthermore, few people or 10.18 % opened grocery stores selling many stuffs, fresh vegetables, food and drinking water. The important occupational structure of local people here were the agriculturalists or 59.03 % (rice plantation, farming, gardening, general workers, and animal farming). The area in an industrial estate was newly allocated; therefore, there were some forests and agricultural land [7]. For this reason, the agriculturalists were the major occupation for the most local people.

Regarding to the data, the average income per capita per year of the samples was around 59,112 Baht while the average household income per year was 113,958 Baht. This was because most local people devoted themselves to the agriculture as their main career. Thus, their income was not quite high compared to those who owned the grocery stores. For the samples' family member, the average number was about 3-4 persons per household and most of them were single families consisting of only father, mother, and children.

3.2 Structure of Energy Consumption

According to the survey on energy consumption including the quantity of energy usage and cost of energy usage, the average quantity of energy consumption in household was divided by energy type as represented in Table 1. Moreover, there were 6 kinds of commercial energy used in the communities. Electricity was utilized for electrical appliances for

cooking, facility devices, bulbs, agricultural equipments, factory equipments of household industry. LPG was beneficial for cooking whereas diesel oil was used for pick-up, tractor, and agricultural feed pump. gasoline 91 and gasohol 91 were useful for motorcycles meanwhile gasohol 95 was utilized for cars. In Table 1 the highest average household diesel oil consumption was 247.40 kgoe/year. Then the average household electricity consumption was 170.93 kgoe/year. It could be said that the total household commercial energy consumption was 632.40 kgoe/year.

For renewable energy consumption, there were 3 kinds of usages – charcoal, firewood, and biogas. All of them were used for cooking activities. The most renewable energy consumption was charcoal 131.15 kgoe/year, followed by firewood 38.74 kgoe/year. Both charcoal and firewood were still utilized in the community because they were available and produced in local agricultural area. Besides, the quantity of biogas consumption was 2.63 kgoe/year. According to the survey, only a few households in Wang Din Sub-District used biogas while there was not biogas usage in other sub-districts. It was because Wang Din Sub-District was the only area where Sub-District Administrative Organization has promoted to use biogas and people realized renewable energy consumption as seen from the average total household renewable energy consumption was 172.52 kgoe/year whereas the average total household energy consumption was 804.92 kgoe/year.

When energy consumption was classified by 6 activities, the average household energy consumption as shown in Table 1 demonstrated that the transportation mostly consumed energy about 317.17 kgoe/year because the community was large and the houses were far from each other or farming area. Thus, they had to commute very far to their agricultural area. Even though there were some public vehicles, it had to wait for a long time and it was inconvenient as well. Therefore, there was a motorcycle in every households which mostly consumed energy due to poor situation of road and land transportation [21]. Meanwhile the cooking activities consumed energy about 280.03 kgoe/year. The energy consumption of industrial equipment was at least level because there was only one household artificial flower factory.

According to the average cost of energy consumption, it indicated that there were 6 kinds of commercial energy used in the community as following; electricity, LPG, diesel oil, gasoline 91, gasohol 91, and gasohol 95 as presented in Table 2. The average household cost of diesel oil consumption was at the highest or around 8,608 Baht/year, followed by gasoline 91 at 5,961 Baht/year. Hence, the average total household commercial energy consumption was 22,971 Baht/year. In contrast, the cost of renewable energy consumption was involved with the charcoal only or about 1,891 Baht/year. Some households did not have to pay for the charcoal because it was available in their own farming area. The excessive charcoal could be sold for extra income as well.

Table 1 Total and average energy consumption per household classified by activities and energy types (kgoe/year)

Energy types Activities	Electricity	LPG	Diesel oil	Gasoline 91	Gasohol 91	Gasohol 95	Charcoal	Firewood	Biogas	Total energy consumpti on
Cooking	6,119 (15.57)	36,133 (91.94)	-	-	-	-	51,544 (131.15)	15,226 (38.74)	1,032 (2.63)	110,053 (280.03)
Lighting	8,857 (22.54)	-	-	-	-	-	-	-	-	8,857 (22.54)
Electrical appliances	46,325 (117.88)	-	-	-	-	-	-	-	-	46,325 (117.88)
Agricultural work	819 (2.08)	-	20,578 (52.36)	-	-	-	-	-	-	21,396 (54.44)
Industrial equipment	5,056 (12.87)	-	-	-	-	-	-	-	-	5,056 (12.87)
Transportation	-	-	76,651 (195.04)	40,251 (102.42)	5,672 (14.43)	2,075 (5.28)	-	-	-	124,648 (317.17)
Total energy consumption	67,176 (170.93)	36,133 (91.94)	97,229 (247.40)	40,251 (102.42)	5,672 (14.43)	2,075 (5.28)	51,544 (131.15)	15,226 (38.74)	1,032 (2.63)	316,335 (804.92)

To the use of firewood and biogas, there was no cost because the local people could collect firewood from their agricultural land while biogas could be produced from swine manure digestion. The average total cost of energy consumption was 24,862 Baht/year.

When considering the energy consumption activities, Table 2 represented the cost of average household energy consumption stemmed from 6 activities. The transportation mostly consumed the energy cost 13,771 Baht/year due to prices of fuel [21] and wasteful energy consumption of people, especially for the motorcycle.

Table 2 Total and average cost of energy consumption per household classified by activities and energy types(Baht/year)

Energy types Activities	Electricity	LPG	Diesel oil	Gasoline 91	Gasohol 91	Gasohol 95	Charcoal	Firewood	Biogas	Total cost of energy consumpti on
Cooking	199,633 (508)	707,713 (1,801)	-	-	-	-	743,337 (1,891)	-	-	1,650,682 (4,200)
Lighting	288,948 (735)	-	-	-	-	-	-	-	-	288,948 (735)
Electrical appliances	1,511,372 (3,846)	-	-	-	-	-	-	-	-	1,511,372 (3,846)
Agricultural work	26,709 (68)	-	715,939 (1,822)	-	-	-	-	-	-	742,648 (1,890)
Industrial equipment	164,965 (420)	-	-	-	-	-	-	-	-	164,965 (420)
Transportation	-	-	2,666,836 (6,786)	2,325,130 (5,916)	310,697 (791)	109,488 (279)	-	-	-	5,412,150 (13,771)
Total cost of energy consumption	2,191,626 (5,577)	707,713 (1,801)	3,382,775 (8,608)	2,325,130 (5,916)	310,697 (791)	109,488 (279)	743,337 (1,891)	-	-	9,770,765 (24,862)

3.3 Potential of Renewable Energy

The potential assessment of renewable energy was shown in Table 3-6 consisting of solar energy, wind energy, biomass, and biogas. It was found that the potential of solar energy was 371,196 toe/year whereas the potential of wind energy was 876 toe/year. In

addition, the potential of biomass was evaluated from some major popular crops of the community such as rice, corn, sugar cane, cassava, coconut, oil palm, rubber, nut, and cashew nut. Then, the biomass potential was 6,798, 2,667, 8,203, 17, 28, 0.40, 12, 18, and 1,599 toe/year, consecutively. Besides, the potential

of biogas was 1.24, 269, 66, 113, and 0.44 toe/year, respectively, assessed from the manure digestion of swine, cow, buffalo, chicken, and duck. Regarding to the survey, it could be said that the solar energy in the community had the highest potential at 94.73 % because Thailand was located on a tropical zone near the equator and the solar radiation was widely spread. On the contrary, the wind energy was quite low at 0.22 % due to irregular wind in the industrial estate and low wind speed. Hence, it could not be able to generate the electricity. The potential of biomass was at 4.94 % and mostly obtained from sugarcane, rice, and corn plantation. Then, the potential of biogas was quite low

at 0.11 %. Thai government had promoted another 2 types of renewable energy before including the solar energy and the biogas. The first one was applied to power generation to store in a battery for some houses having no power whereas the biogas had ever been produced from the swine manure digestion. Nevertheless, both of them were abandoned later because the local people in community lacked knowledge and understanding to maintain and operate the system. Thus, the equipments to generate solar energy were damaged and the production system of biogas was failed, too.

Table 3 The potential of solar energy in community

Average solar radiation intensity (MJ/m ² -hr)	Total areas (km ²)	Forest areas (km ²)	Agricultural areas (km ²)	Potential of solar energy (toe/year)
1.4404	297.3456	186.6676	108.1926	371,196

Table 4 The potential of wind energy in community

Constant	Air density (kg/m ³)	Averagewind speed (m/s)	Rotor areas (m ²)	Efficiency factor	Potential ofwind energy (toe/year)
0.50	1.15	3.56	23.94	0.55	876

Table 5 The potential of biomass in community classified by type of agricultural waste

Types of plant	Yield of plant (kg/year)	Type of agricultural waste	Ratio of agricultural waste	Heating value of agricultural waste (MJ/kg)	Potential of biomass (toe/year)
Rice	35,539,888	Husk	0.230	14.720	6,798.00
		Straw	0.447	10.240	
Corn	22,874,387	Corn cob	0.273	18.040	2,667.00
		Bagasse	0.291	14.400	
Sugar cane	36,698,781	Sugar cane leaves	0.302	17.390	8,203.00
Cassava	427,100	Cassava trunk	0.088	18.420	17.00
		Coconut shell	0.362	16.230	28.00
		Endocarp	0.160	17.930	
Coconut	122,000	Coconut empty bunch	0.049	15.400	
		Coconut leaves	0.225	16.000	
		Palm empty bunch	0.428	17.860	0.40
Oil palm	1,000	Palm fibers	0.147	17.620	
		Palm shell	0.049	18.460	
		Palm limb	0.604	9.830	
Rubber	133,998	Rubber limb	0.260	14.980	12.00
	19,437	Trunk, leaf and nutshell of soybean	0.663	19.440	18.00
Nut	386,835	Trunk, leaf and nutshell of green beans	0.083	11.420	
	28,890	Trunk, leaf and nutshell of peanut	0.323	12.660	
Cashew nut	3,445,035	Cashew nut shell	0.778	25.208	1,599.00
Total potential of biomass (toe/year)					19,342.00

Table 6 The potential of biogas in community classified by type of manure digestion

Type of animal	Number of animals (animal)	Ratio of the average animal weight per livestock	Volume of wastewater (l/day)	Retention time of wastewater (day)	Ratio of biogas	Potential of biogas (toe/year)
Swine	65,079	0.12	0.255	6.25	0.550	1.24.00
Type of animal	Number of animals (animal)	Amount of manure (kg/animal)	Ratio of manure	Ratio of volatile solids	Ratio of biogas	Potential of biogas (toe/year)
Cow	14,532	5.00	0.500	13.37	0.307	269.00
Buffalo	2,325	8.00	0.500	13.64	0.286	66.00
Chicken	485,024	0.03	0.800	22.34	0.242	113.00
Duck	3,731	0.03	0.400	17.44	0.310	0.44
Potential of biogas (toe/year)						450.00

4. CONCLUSION AND RECOMMENDATIONS

Regarding to the survey on energy consumption structure and potential assessment of renewable energy, it demonstrated that most respondents or 57.75 % graduated lower than basic education level and 59.03 % in the community were agriculturalists, average annual income around 59,112 Baht while the average annual household income was 113,958 Baht. The number of family member was about 3-4 persons. In terms of quantity and cost of energy consumption in the community, it was found that diesel oil and the gasoline 91 were most widely consumed for the transportation activities, followed by electricity for electrical appliances, and charcoal for cooking activities. The potential of renewable energy in the community, solar energy had the highest potential, followed by biomass, respectively.

According to the survey, different kinds of renewable energy had been used in the community such as solar energy, biomass, and biogas. Due to lack of knowledge and understanding on the system maintenance, however, the operating system and equipments were abandoned. Furthermore, the previous studies found some important obstacles to the development of renewable energy consumption as following; 1) fuel supply, 2) technical barrier, 3) financial barrier, and 4) institutional and legislative constrains [3, 22]. Thus, these problems should be solved by supporting renewable energy consumption again, especially solar energy for agricultural products drying [23] to bake cashew nut and generate electricity [24, 25, 26] for some houses having no electricity. In addition, biomass consumption was also beneficial to burn as charcoal and agricultural waste could be produced as fuel or heat to generate electricity [3, 27] or briquettes [28, 29]. Moreover, biogas consumption should be only promoted for the households raising animals. The sufficient biogas production was useful to produce thermal energy and electricity as well [3, 5] whereas the waste of biogas production could be

applied to fertilizer also. For wind energy, it might be difficult to use because of low and irregular wind speed. Hence, wind wheel should be constructed to wind pump [30]. If the community turned to use such renewable energy, it would help reduce the commercial energy consumption, particularly electricity. Nevertheless, the mostly used of commercial energy was diesel oil, therefore biodiesel should be supported to substitute diesel oil [31] and decrease crude oil consumption and import from foreign countries. More importantly, the consciousness of should be encouraged to reduce energy consumption and provide sustainable cost savings as well. This was regarded as a good start of the community to step into low carbon society [32, 33] because the promotion of renewable energy consumption was another way to decrease greenhouse gas and support the renewable energy development plan of Thailand to actually be the green city [3].

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ENERGY CONSUMPTION VS COMFORT: A MULTI-OBJECTIVE OPTIMISATION FRAMEWORK FOR LOW CARBON TOWNS. APPLICATION TO STREET LIGHTING.

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ABSTRACT

A multi-objective methodology is proposed to optimise systems in low carbon town by taking into account not only energetic objectives but also subjective ones such as user preferences. The originality of this method lies in the integration of subjective data obtained from psychophysical tests conducted with panels of end-users into a multi-objective optimisation framework. This method makes it possible to identify all the best tradeoffs and the direct correspondence between power demand and user preferences, and to handle inter-individual differences. The methodology is applied to street lighting. The system to optimise is a LED street lighting luminaire based on a mixture of green and white LEDs. It is shown that 20% savings on power demand can be reached with respect to the white LED reference solution, while satisfying at the same time populations whose preferences with respect to lighting are different.

Keywords: Multi-objective optimization; Energy efficiency; Comfort; Street lighting; Mesopic vision.

1. INTRODUCTION

The current energetic context requires that energy consumption is reduced in a variety of areas. However, new energy efficient solutions have to be developed that also provide acceptable comfort for users. This is the case of a very large number of systems in towns (e.g. street lighting, building heating and cooling, etc.). Thus, the design of a system in a low carbon town can be seen as a Multi-objective Optimisation Problem (MOP) that needs to identify optimal solutions corresponding to the best feasible tradeoffs between energy efficiency and inhabitant preferences [1].

In mono-objective optimisation, the goal is to determine the global minimum of one objective function $f(X)$, with $X=(X_1, X_2, \dots, X_k)$ being a vector of k decision variables that defines each solution. The objective function value of each solution makes it possible to quantify the performance of this solution. The solution that achieves the minimum of the objective function is called optimal solution.

A MOP is characterised by having more than one objective function to minimise. Instead of looking for the global optimum as in mono-objective optimisation, the goal of multi-objective optimisation is to determine the set of best tradeoffs between the various objective functions. Various methods can be employed to solve a MOP. Most of non-Pareto methods consist in transforming the MOP into single-objective optimisation problem(s) so as to enable the use of usual techniques

and achieve one optimal solution (weighted-sum method, ϵ -constraint method, lexicographic method, etc.) [1].

In order to identify the set of the best tradeoffs, Multi-objective Optimisation Methods (MOM) are often based on the Pareto-optimality concept. Pareto defined the optimum in multi-objective optimisation by introducing the Pareto dominance notion: S defined with particular decision variable values dominates another solution S_0 if and only if S is better than S_0 for all objectives i (i.e. $\forall i \in \{1, \dots, m\} \mid f_i(S) \leq f_i(S_0)$ where m is the number of objective functions) and if there exists, at least, one objective j for which S is strictly better than S_0 (i.e. $\exists j \in \{1, \dots, m\} \mid f_j(S) < f_j(S_0)$). The set of non-dominated solutions is denoted Pareto optimal. The image of an optimal solution set in objective space is called the Pareto Front and limits the feasible and the unfeasible tradeoffs in the objective space.

Figure 1 presents a minimisation problem of two objective functions f_1 and f_2 in which solutions are defined with two decision variables X_1 and X_2 . Figure 1(a) presents performances of solutions in objective space (f_1, f_2). C is dominated by A and by B and does not belong to the Pareto Front. A and B are non-dominated and are Pareto-optimal solutions. Each solution is defined with decision variable values in the decision space in Figure 1(b). For example, B is defined with (X_{1B}, X_{2B}) .

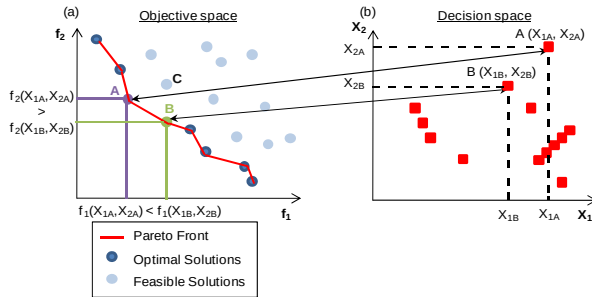


Figure 1 Example of Multi-objective Optimisation Problem (a) Pareto Front in two-objective space (b) Decision space in which solutions are defined

In previous work [2-3], the studies dedicated to the integration of human judgment into a multi objective optimisation process take into account the evaluation of a single user at a time. In this paper we propose a method making it possible to integrate the preferences of a panel of users. For this purpose, a MOM is introduced to optimise street lighting solutions by taking into account not only energy objectives but also subjective ones such as visual preferences. The originality of this method lies in the integration of the subjective data obtained from psychophysical tests, conducted with panels of users, into a multi-objective optimisation framework. In the context of the case study of the paper, the system to optimise is a LED street lighting luminary based on a mixture of green and white LEDs.

The remainder of the paper is organised as follows. In Section 2, an overview of the methodology is presented. In Section 3, the street lighting case study is described. Section 4 is dedicated to the detailed description of the method applied to street lighting optimisation and the obtained results. Finally, conclusions and future work are presented in Section 5.

2. METHOD

The method is aimed at obtaining decision variable values of the system to be optimised to lead to the best feasible tradeoffs between conflicting objectives, *i.e.* physical objectives such as power demand and subjective ones such as user preferences. The method is divided into two steps: (1) psychophysical data acquisition and (2) integration of the psychophysical data into the multi-objective optimisation process. An overview of this method is presented in Figure 2. The first step is dedicated to the acquisition of psychophysical data. In the case of street lighting optimisation, the psychophysical data are related to visual comfort and are denoted as psycho-visual data. A psycho-visual test is conducted for the assessment of lighting solutions sampled over the decision space. Statistical analyses are employed to determine the global judgments of all participants. In the second step, a psycho-visual function is estimated from the psycho-visual data and employed as an objective function in the optimisation process to obtain the Pareto Front, *i.e.* the best feasible tradeoffs between conflicting objectives.

The Pareto Front is estimated through robust evolutionary algorithms, such as NSGA-II [4].

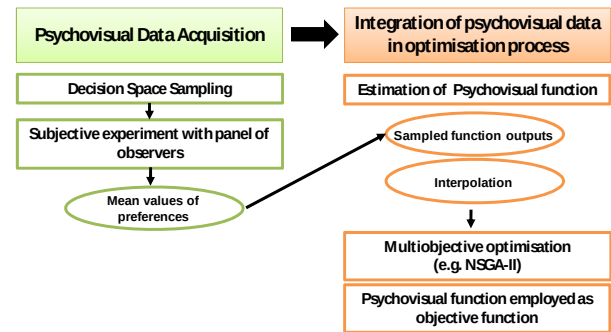


Figure 2 Overview of the proposed methodology

3. STREET LIGHTING CASE STUDY

The method described in section 2 is applied to street lighting. The system to optimise is a LED street lighting luminary installed in Rue Saint Augustin in Lyon, France. Green and white LEDs are mixed up together, in order to obtain a spectrum with a maximum emission close to the maximum of the eye sensibility in the mesopic range.

The decision variables to optimise are the proportion of green LEDs, and the mesopic flux of the luminary. The objectives are the power demand (to minimise), and the visual comfort (to maximize). Indeed as the proportion of green LEDs increases, the power demand needed for the same flux decreases, but the psycho-visual comfort can decrease also because the light becomes greenish, which result in poor color rendering. Thus, the best possible tradeoffs between power demand and visual comfort have to be identified. For illustration purpose, Figure 3 presents the aspect of Rue Saint Augustin lit by various luminaries featuring different proportions of green LEDs and different mesopic flux.

4. STREET LIGHTING OPTIMISATION

This section is dedicated to the application of the optimisation method on the street lighting case study described in section 3. The two steps of the methods are presented and the results are detailed.

4.1 First step: psychovisual data acquisition

Psycho-visual experiment

A psycho-visual experiment was conducted with 30 observers who were asked to express their preferences about the street lighting produced by luminaries obtained for 14 combinations of the decision variables. The proportion of green over white LEDs was between 0% and 90%, and the mesopic flux between 60% and 100% of the maximum flux, arbitrarily set so as to obtain an illuminance of 25 lux on the road surface. Figure 3 presents three examples of the

aspect of the street lit by three luminaries featuring different proportions of green over white LEDs. Figure 4 presents the 14 combinations of the values of the decision variables used corresponding to the luminous environments assessed during the psycho-visual experiment. Table 1 details the photometric features and energy savings reached by each luminary. This experiment was carried out using a virtual reality simulator, presented on Figure 5. The street scene was modeled with the 3DS MAX package.

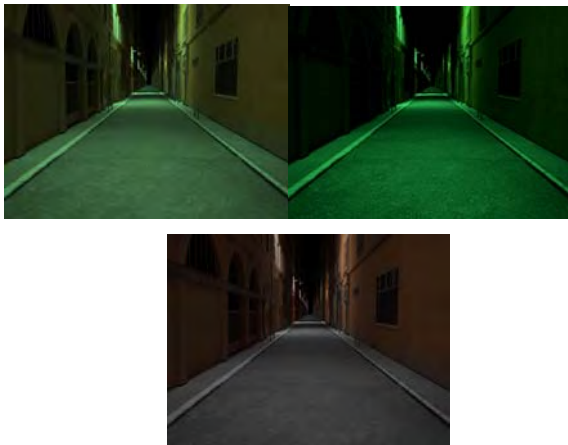


Figure 3 Street (Rue Saint Augustin, Lyon, France) lit by three different LED luminaries. Top left: 90% green – 10% white. Top right: 50% green – 50% white. Bottom: 100% white

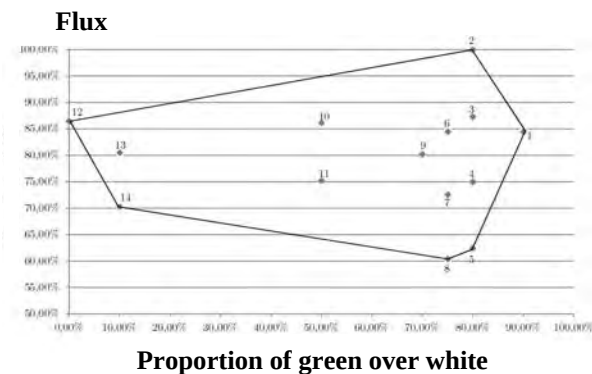


Figure 4 Values of decision variables
DV1: Proportion of green over white LEDs, and DV2: Mesopic flux, for each of the 14 luminaries

Then, 14 renderings corresponding to the 14 assessed luminous environments were computed with the light transport simulation program V-Ray. According to a previous study [5], V-Ray produces images that cause the closest feeling, regarding the luminous environment, to what would be felt in a real-world scene. Observers were sitting at 2.63m in front of the screen. The size of the images was 1.69m x 1.26m and their resolution was 1024 x 768 pixels. Thanks to a grey filter, the luminance dynamic range of the simulator was 0.01 - 5 cd/m², which is in the mesopic area. A spectral characterization of the simulator had

been performed beforehand. The 14 images to be displayed were calibrated based on this characterization: the spectrum output by each pixel of the images was

Table 1 Photometric features and energy savings of each luminary

Luminary Id	% Green LEDs	% Mesopic flux	% Energy savings
1	90	84.5	20
2	80	100.0	20
3	80	87.3	30
4	80	75.0	40
5	80	62.5	50
6	75	84.5	30
7	75	72.6	40
8	75	60.4	50
9	70	80.2	30
10	50	86.1	20
11	50	75.2	30
12	0	86.4	0
13	10	80.5	90
14	10	70.3	30



Figure 5 Virtual reality simulator used to perform the psycho-visual experiment

metameric of the one that would have been reflected by the street material under the light output by the actual green and white LED based luminaries. In addition, before the beginning of the experiment, each observer stayed 30 minutes in the dark (where luminances were lower than 0.1 cd/m²) so as to ensure that his eyes were adapted to mesopic conditions.

During the experiment, a paired-comparison protocol was used. Observers were asked to imagine themselves 'walking in the street'. During the paired-comparison test, observers had to choose between image A and image B by answering the question: 'Which luminous environment do you prefer?'. The participants could observe both images as often as needed. In this test, all the 91 possible comparisons (14 x 13 / 2) were assessed with a different random order for each participant.

Statistical analysis of the psycho-visual data

Data processing was performed according to the following framework. First of all, hierarchical clustering was employed to study the division of the panel opinion and to highlight potential different populations (groups of observers who share the same opinion in their judgment and differ from the other groups). Then, statistical analyses were employed. In the case of two or more populations, statistical analyses were done separately for each population. The results of the hierarchical clustering, employed with the nominal distance and average linkage, show that two populations were identified. To analyze the paired-comparison data, the Thurstone law of comparative judgment case V [6] was employed and gave a scaled value for each sampled solution. The scaled value defines the average preference for each solution: the higher is the scaled value, the more suitable the solution is for the street. For n stimuli, choices are collected in the preference matrix M ($n \times n$) for each participant. If Stimulus i is preferred to Stimulus j , $M_{ij}=1$ and $M_{ji}=0$. By adding each column, the preference score of each stimulus can be deduced for each participant. The sum of the preference matrix of all participants is calculated. Sum of the columns of this total matrix makes it possible to obtain a rank of assessed stimuli. To obtain scale values, a proportion matrix P is calculated from the total matrix. Then, values are deduced from proportions by employing the inverse of the standard normal law. Finally, the scaled value of each stimulus is the arithmetic mean of one matrix line. In addition to preference scores, scaled values are at the interval level and make it possible to highlight the distance between the different assessed stimuli.

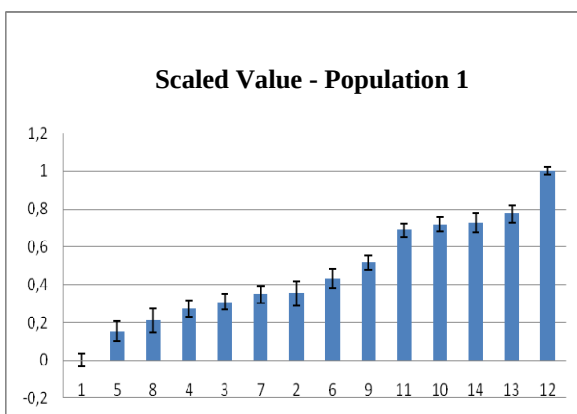


Figure 6 Thurstone case V scaled values for each of the 14 luminaries for population 1 (higher is preferred)

The experiment shown 19 observers (population 1) prefer luminaries featuring high color rendering, even if the flux is low; 9 observers (population 2) prefer luminaries featuring high flux, even if color rendering is low. Figure 6 (resp. 7) shows the scaled values obtained by each luminary, assessed by population 1 (resp. population 2).

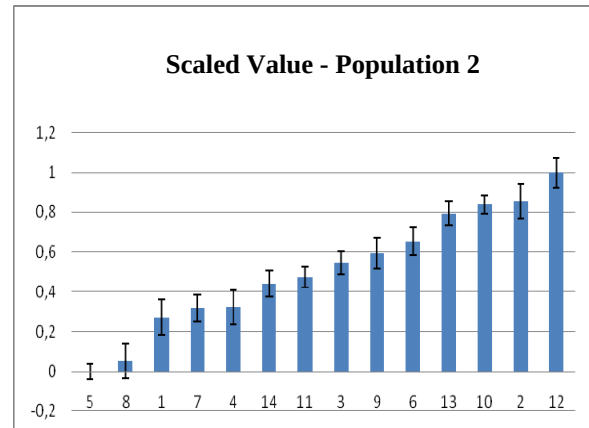


Figure 7 Thurstone case V scaled values for each of the 14 luminaries for population 2 (higher is preferred)

4.2 Second step: integration of psycho-visual data into the multi-objective optimisation process

This step implements the key idea of the method, which consists of integrating the psycho-visual data into the optimisation process by using the psycho-visual functions as objective functions to optimise.

For each assessed solution, the obtained scaled value corresponds to the function value at each sampled point. A bilinear interpolation method was employed to estimate the psycho-visual function throughout the decision space. This assumption of continuity comes from the main concept of psychometrics. The existence of a continuous measure of feeling according to the physical level of the stimulus called the 'psychology continuum'. Linear interpolation has been used in previous work. Lim [7] mentioned the existence of psychophysical or psycho-hedonic function obtained from interpolation. The psychophysical function is related to measures on intensity scales while the psycho-hedonic function is related to preference measures. Linear interpolation is mostly employed in food quality research or in research related to the psychometric concept. Scores of discrete tested values of the attribute are linearly interpolated. In decision making, a utility function is constructed from preferences expressed by a decision maker who attributes weighted values to some selected values of the objective function.

4.3 Results: optimal tradeoffs between preferences of populations 1 and 2 and energy savings

A three-objective optimisation was performed with the three following functions in objective space:

- f_1 : the opposite of the psycho-visual function preference for population 1;
- f_2 : the opposite of the psycho-visual function preference for population 2;
- f_3 : the opposite of energy savings (in %).

$f_2 =$

Each of these functions is first defined at the 14 points of the decision space presented in Figure 4. The value of f_1 (resp. f_2 , resp. f_3) at these points is given by the scaled value presented on Figure 6 (resp. on Figure 7, resp. in the third column of Table 1). The values of these functions elsewhere in the decision space are derived from bilinear interpolation.

The Non-dominated Sorting Genetic Algorithm (NSGA-II [4]) was employed. It was developed by Deb et al. with $O(mN^2)$ computational complexity (N = population size and m = the number of objectives). Individuals are sorted according to dominance rank and crowding distance. The selection is conducted through a binary tournament by first comparing the non-domination rank and then the crowding distance. Simulated Binary Crossover operator and polynomial mutation are employed. NSGA-II Matlab code was employed. Each gene corresponds to one decision variable. The mutation and crossover rates were set to 0.5 and 0.9, respectively. The crossover distribution index c and the mutation distribution index m were each fixed to 20. Population size (set to 500) and number of generations (set to 200) were determined after numerical experimentation to ensure convergence for the optimisation problem.

The result of the optimisation carried out with NSGA-II is a Pareto Front in three dimensional space and is presented in Figure 8. The Pareto Front presents the best tradeoffs that can be achieved between visual preferences of population 1, visual preferences of population 2, and energy savings. This Pareto Front shows that the preferred luminary for both populations 1 and 2 corresponds to maximum energy consumption (0% energy savings). This is consistent with scaled

values presented in Figures 6 and 7, since the preferred luminary for both populations is luminary 12 - the white LED reference luminary, featuring no energy savings. Moreover, the Pareto Front shows that energy savings necessarily imply decreases of the visual satisfaction of both populations and that the relationship between energy savings and preferences is not continuous; thus, decision making should be performed carefully.

A 2D projection of the Pareto Front is presented in Figure 9 (a), and the set of Pareto-optimal solutions in decision space is presented in Figure 9 (b). Each point corresponds to a Pareto-optimal solution, i.e. one couple of proportion of green over white LEDs and % of maximum mesopic flux that makes it possible to achieve one of the best tradeoffs in objective space.

f_2 = -Preferences population 2

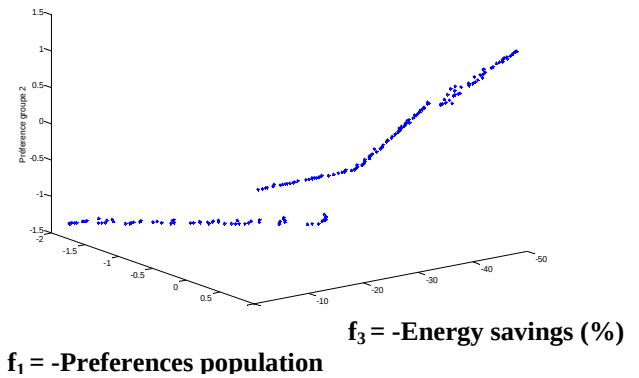
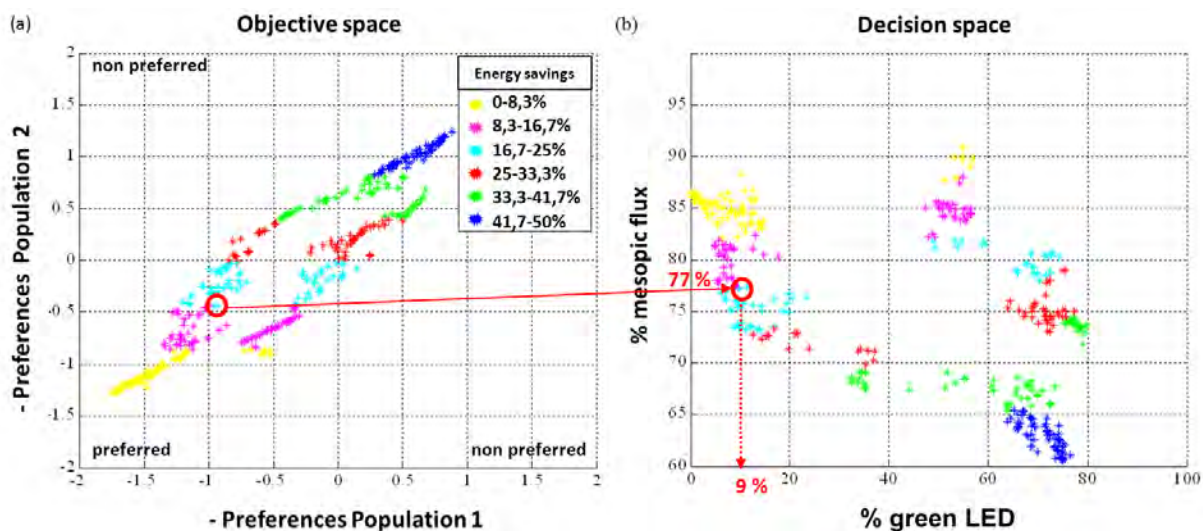


Figure 8 Pareto Front in 3D space (lower is better)



(a) Objective space. (b) Decision Space. 20% energy savings can be achieved while satisfying Population 1 (-1.06 / -2) and Population 2 (-0.51 / -2). Corresponding luminary features 9% of green LEDs and 77% of maximum mesopic flux

Figure 9 Projection of Pareto Front in 2D.

Figure 9 can be used to identify luminaries resulting in energy savings and at the same time

satisfying both populations. For instance, a luminary featuring 9% of green LEDs and 77% of maximum

mesopic flux results in 20% energy savings with respect to the white LED reference solution, while satisfying at the same time Population 1 (-1.06 / -2) and Population 2 (-0.51 / -2).

5. CONCLUSION

A Multi-objective Optimisation Method taking into account user preferences has been introduced to identify the best tradeoffs between energy efficiency and user preferences for systems in low carbon towns. This method, divided into two steps, proposes acquiring psychophysical data as the first step and then to integrate these data in a multi-objective optimisation process as the second step. The originality of this method is the integration, as objective functions, of the subjective data obtained from psychophysical tests conducted on a panel of participants, potentially representative of end users. The obtained Pareto Front provides all optimal feasible tradeoffs. Thus, the proposed framework can help a decision maker by presenting all tradeoffs in the objective space.

The method is illustrated through a street lighting study. Best trade-offs between power demand and preferences of two populations are identified. In particular, 20% of energy consumption savings can be reached with respect to the white LED reference solution. Future work will investigate how to handle psychophysical data uncertainties, so as to identify reliable solutions.

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Factors Influence Electricity Consumption in Residential Sector

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ABSTRACT

The objective of this research was to study factors that influence the energy consumption in the residential sector. The factors in this study were structural factors and behavioral factors : 1) the structural factors were number of story in the house, number of bedrooms, number of members, average of family income, and end use of electricity consumption, 2) the behavioral factors were energy saving lifestyle, energy saving attitude, energy saving knowledge, energy saving responses, and energy saving practice . Research tools were questionnaires and electricity bills. The samples were 400 households in Bangkok and Metropolitan. The stratified random sampling was used in this study. They were 321 completed questionnaires returned or 80.25 percent. Research conducted in 2 phases: 1) 1st phase, study of the use of electricity consumption, 2) 2nd phase, study the relationship of the structural and behavioral factors influence electricity consumption in residential sector. Statistics used in this study were mean, percentage, standard deviation, correlation, covariance, multicollinearity and multiple regression. The findings in 1st Phase found that the average of residential electricity consumption was 336.79 kwh. Most houses have 2 story, 3 bedrooms, 3-4 persons in one house, average family income was between 20,001 - 40,000 baht, 1 refrigerator and most of them don't have air conditioning. Most of them have energy saving lifestyle in moderate level, energy saving attitude in moderate and high level, energy saving knowledge in high level, energy saving responses in high level, and energy saving practice in moderate level. The results of 2nd Phase showed that structural factors are statistically correlated to energy consumption in the residential at the .01 level. Factors that related to energy consumption in the residential sorted from most to least were number of end use of electricity consumption in case of air conditioning, number of residential in the house, number of story in the house, end use of electricity consumption in case of washing machine and computers. The all five variables of structural factors together explain the variance of the electricity consumption in the residential sector with 57.2 percent of the weight of a standardized coefficient .525 .291 .124 .116 and .096, respectively. While behavioral factors are statistically correlated to energy consumption in the residential at the .01 level. Factors that related to energy consumption in the residential sorted from most to least were energy saving lifestyle, energy saving attitude, and energy saving practice . The all three variables of behavioral factors together explain the variance of the electricity consumption in the residential sector with 12.6 percent of the weight of a standardized coefficient were - .281, -.196 and - .117, respectively. The results of the study on factors influence the electricity consumption in the residential sector can be used as a guide in terms of government policies and practices of the residential sector to campaign and give information on the importance of the campaign and information about energy saving.

Keywords: Electricity Consumption; Residential Sector; Structural Factor; Behavioral Factor.

1. INTRODUCTION

The consumption of energy involved the activity in human' life such as traveling, entertainment and occupation, etc. The energy consumption effect on the growth of economic in industrial, transportation, agriculture commercial and residential sector. Thailand is the developing country in economic with rapid of economic growth. Before 1997 the energy demand in the country is increased approximately 8% (1991 – 1996) depend on economic growth approximately 8% also. In 1997 Thailand was poor in economic, economic crisis, the growth rate of economic decrease -0.92% that effect to energy demand decrease -0.33%

Nowadays, department of Alternative Energy Development and Efficiency (DEDE) reported that the final energy consumption of Thailand had increased 8.8% from the previous year. Commercial energy

consumption, which comprises petroleum products, natural gas, coal & its products, and electricity, increased 1.6% from the previous year, and renewable energy, including fuel wood, charcoal, paddy husk, and bagasse, increased 2.9%.

In 2011, the residential sector consumed energy 11,040 ktoe and accounted for 15.6 % of the final energy consumption, up 0.7 % over last year. Energy consumed in this sector comprised renewable energy 56.3 %, electricity 25.4 %, and petroleum products 18.3 %. (Thailand Energy Situation 2011 report,DEDE)

Thailand imports energy from the other country and the trend is increasing every year, because the energy resource are limited. The government support the policy to emphasize development of efficiency for the energy consumption such as development of minimum efficiency standards for household appliances, label No.5 of refrigeration or air

conditioning, setting regulation for control the energy consumption in industrial and building, promulgation about energy conservation for industrial and building.

Because the energy resource in Thailand are limited. The government support the policy to emphasize development of efficiency for the energy consumption such as development of minimum efficiency standards for household appliances, label No.5 of refrigeration or air conditioning, setting regulation for control the energy consumption in industrial and building, promulgation about energy conservation for industrial and building.

In residential sector, energy consumption is electricity, LPG, kerosene, firewood, charcoal, paddyhusk and other renewable. The energy consumption in residential are different from region, in Bangkok and its suburban area are high in electricity and LPG, urban and rural area are high in electricity and renewable energy respectively. The different form of energy consumption in residential have 5 factors :

(a) Site and climate : the climate is important for the energy consumption in warm and humidity country, the temperature, heat transfer and solar radiation,

(b) Building : envelop of building, area, site or location, the age of building and the area management for using,

(c) Application : type of appliance, number, energy type for appliance, efficiency of appliance and time in use,

(d) User and operation : in-house members, age, income, perception, attitude and belief demographic activity,

(e) Marketing and energy supply : cost, market price, interest, government policy, and energy supply performance.

These factors are reflected on the effectiveness of the energy consumption in residential sector. The classification and analytical of the energy consumption in residential sector for these factors and develop of specific energy index in residential sector can use for setting standard of energy management in residential sector and improve energy consumption.

This study is aimed at studying the structural and behavioral factors to influence energy consumption in a residential sector of the single house in Bangkok and its suburban area. The results of this study would reveal that the mentioned factors significantly affected energy consumption and can be use for energy management and to promote energy conservation in Thailand. Each factors indicated that the in-house members behavioral could be adaptable to reduce their residential electricity consumption and also help them realize the importance of the campaign and information about residential energy conservation.

2. SCOPE OF THE STUDY

The factors in this study were structural factors and behavioral factors : 1) the structural factors were number of story in the house, number of bedrooms, number of members, average of family income, and end use of electricity consumption, 2) the behavioral factors

were energy saving lifestyle, energy saving attitude, energy saving knowledge, energy saving responses, and energy saving practice . Research tools were questionnaires and electricity bills. The samples were 400 households in Bangkok and Metropolitan. The stratified random sampling was used in this study. They were 321 completed questionnaires returned or 80.25 percent. Research conducted in 2 phases: 1) 1st phase, study of the use of electricity consumption, 2) 2nd phase, study the relationship of the structural and behavioral factors influence electricity consumption in residential sector. Statistics used in this study were mean, percentage, standard deviation, correlation, covariance, multicollinearity and multiple regression.

3. RESULT

3.1 The study of the use of electricity consumption.

3.1.1 electricity consumption

The household energy surveyed depending on Bangkok and its suburban area. All of household use electricity. Electricity is supplied from the Metropolitan Electricity Authority. Average of Electricity consumption is 336.79 kwh/month ,max. 797.80 kwh/month and min 37.40 kwh/month.(Table 1)

Table 1 Average household of Electricity consumption

(Kwh/Month)					
Rage (Kwh)	Electricity consumption				
	No. house	Min	Max	Mean	SD
<= 200	88	37.40	199.00	127.89	41.61
201-350	94	202.40	342.80	269.99	43.05
351-500	72	350.40	500.00	430.67	46.50
501-650	49	500.40	648.20	566.69	41.90
> 650	18	651.00	797.80	705.59	51.90
Total	321	37.40	797.80	336.79	181.48

3.1.2 structural factors

Most houses have 2 story, 3 bedrooms, 3-4 persons in one house, average family income was between 20,001 - 40,000 baht, 1 refrigerator and most of them don't have air conditioning. (Table 2.)

Table 2 Structural Factors

Structural Factors		Detail	Freq.	%
No. houses story		1 story	74	23.05
		2 story	247	76.95
No. bedrooms		1 bedrooms	16	4.98
		2 bedrooms	120	37.38
		3 bedrooms	131	40.81

Structural Factors	Detail	Freq.	%
	> 3 bedrooms	54	16.82
No. persons	1 – 2 persons	59	18.38
	3 - 4 persons	155	48.29
	5 - 6 persons	89	27.73
	7 - 8 persons	18	5.61
average income	<= 20,000 Baht	45	14.02
	20,001 - 40,000 Baht	114	35.51
	40,001 - 60,000 Baht	82	25.55
	60,001 – 80,000 Baht	45	14.02
	> 80,000 Baht	35	10.90
No. Appliance			
	– A/C		
	no	95	29.60
	1 unit	68	21.18
	2 units	88	27.41
	3 units	45	14.02
	>= 4 units	25	7.79
	– Refrig.		
	1 unit	245	76.32
	2 units	67	20.87
	3 units	9	2.80

3.1.3 behavioral factors

Most of them have energy saving lifestyle in moderate level, energy saving attitude in moderate and high level, energy saving knowledge in high level, energy saving responses in high level, and energy saving practice in moderate level. (Table 3.)

Table 3 Behavioral Factors

Behavioral Factors	Level	Freq.	%
EN.SAV.Lifestyle	Low level	29	9.03
	Moderate level	238	74.14
	High level	54	16.82
EN.SAV. Attitude	Low level	5	1.55
	Moderate level	156	48.60
	High level	156	48.60
	High level	4	1.25
EN.SAV. Knowledge	Lowest level	15	4.67
	Low level	97	30.22
	High level	171	53.27
	Highest level	38	11.84
EN.SAV. Responses	Lowest level	1	0.31
	Low level	9	2.80
	Moderate level	113	35.20

Behavioral Factors	Level	Freq.	%
EN.SAV. Practice	High level	169	52.65
	Highest level	29	9.034
	Non En. saving	52	16.20
	Moderate level	115	35.83
	High level	154	47.98

3.2 The relationship of the factors influence electricity consumption in residential sector.

3.2.1 the relationship of the structural factors influence electricity consumption in residential sector.

The structural factors are statistically correlated to energy consumption in the residential at the .01 level. Factors that related to energy consumption in the residential sorted from most to least were number of end use of electricity consumption in case of air conditioning, number of residential in the house, number of story in the house, end use of electricity consumption in case of washing machine and computers. The all five variables of structural factors together explain the variance of the electricity consumption in the residential sector with 57.2 percent of the weight of a standardized coefficient .525 .291 .124 .116 and .096, respectively.

Table 4 the variance of the electricity consumption

	SS	df	MS	F	Sig.
Reg.	6027929.0	5	1205585.802	84.18	.000
Res.	4511063.9	315	14320.838		
Tot.	10538992.	320			
R	.756(e)	R ²	.572	Adj. R ²	.565

Table 5 the weight of a standardized coefficient

Factors	B	Beta	Sig.	Tol.	VIF
(Constant)	-56.145		.075		
memsize	44.472	.291	.000	.891	1.122
hom_lev	53.317	.124	.002	.878	1.139
ac_no	75.404	.525	.000	.821	1.218
washing	13.585	.116	.003	.895	1.118
comp	8.835	.096	.017	.845	1.183

3.2.2 The relationship of the structural factors influence electricity consumption in residential sector.

The behavioral factors are statistically correlated to energy consumption in the residential at the .01 level. Factors that related to energy consumption in the residential sorted from most to least

were energy saving lifestyle, energy saving attitude, and energy saving practice. The all three variables of behavioral factors together explain the variance of the electricity consumption in the residential sector with 12.6 percent of the weight of a standardized coefficient were -.281, -.196 and -.117, respectively.

Table 6 The variance of the electricity consumption

	SS	df	MS	F	Sig.
Reg.	1325346	3	441782.259	15.20	.000
Res.	9213646	317	29065.130		
Tot.	10538992	320			
	R	R ²	Adj. R ²		
	.355	.126	.117		

Table 7 The weight of a standardized coefficient

Factors	B	Beta	Sig.	Tol.	VIF
(Constant)	875.14		.000		
lifestyle	-101.19	-.281	.000	.989	1.011
behavior	-28.91	-.117	.029	.964	1.038
attitude	-64.06	-.196	.000	.972	1.028

4. CONCLUSION

In this study, there are 2 analysis of energy consumption. The first analysis is the correlation of electricity consumption in household with structural factors. The correlation statistical significant at .01 for structural factors, are in number of end use of electricity consumption in case of air conditioning, number of residential in the house, number of story in the house, end use of electricity consumption in case of washing machine and computers. The second analysis is the correlation of electricity consumption in household with behavioral factors. The correlation statistical significant at .01 for of behavioural factors are energy saving lifestyle, energy saving attitude, and energy saving practice.

In addition, The results of this study revealed that the mentioned factors significantly affected electricity consumption. Each factors indicated that the in-house members behaviour could be adaptable to reduce their residential electricity consumption by considering at energy conservation practice and also help them realize the importance of the campaign and information about residential energy conservation.

ACKNOWLEDGEMENTS

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ANALYSIS OF INDUSTRIAL CARBON EMISSIONS ASSESSMENT AND INFLUENCE FACTORS

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ABSTRACT

Nowadays, as the ecological civilization develops, the low- carbon ecological urban development booms under the background of energy saving and carbon emission reduction in China. In order to put the concept of low-carbon ecological city into practice, Shenzhen City has launched a project to build an international low-carbon city in Pingdi, Longgang district, aim to develop a low-carbon benchmarking demonstration area in the world. Based on this low-carbon city practice, this paper uses the IPCC carbon-emission checklist accounting methods to estimate local industry energy consumption and carbon emission level. Specifically, the paper carried out a statistic study on the carbon emissions and carbon emissions per unit of prime operating revenue of the 21 industries in Pingdi, which is on the basis of the consumption of energy like kerosene, gasoline, oil, electricity and other energy sources. The analysis draws several conclusions, as follow: 1) The currently total industrial carbon emission was 584511.47 tons in this area; 2) Currently, the high pollution and low value-added industries accounted for the main industrial carbon emissions. For example, Plastic Products Industry (business income 9.43%, carbon emission 17.59%), Fabricated Metal Products (business income 7.10%, carbon emissions 9.42%) and furniture manufacturing industry (business income 15.63%; carbon emissions 15.71%). Then, with the help of GIS, the spatial distribution characteristics of the local industrial structure and industrial carbon emissions in Pingdi was analyzed. It's found that the high-emissions industries usually cluster in the area near Dingshan River currently, which makes a significant impact on the local environment. Based on the research above, this study finally brings some suggestions about the optimization of the local industrial development both for the structures adjustment and for the reasonable layout of the spatial distribution in this area, to realize the energy conservation and emissions reduction in the industrial area. Also the suggestion will help to provide the references for the related policy making for the Pingdi low-carbon eco-city.

Keywords: Industrial carbon emissions; Industrial structure; GIS; Spatial distribution.

1.INTRODUCTION

The new urban construction form of low-carbon and ecology is flourishing under the background of ecological civilization and energy conservation nowadays. In order to implement the concept of low-carbon and ecology development, an international low-carbon city has been built in Shenzhen, Longgang district, Pingdi. A global benchmarking of low-carbon development demonstration is looking forward. However, the industry dominants over half of the carbon emission during the urbanization and industrialization process of China. Recent study shows that the three most important issues of energy conservation are: industry (69%), construction (24%) and transportation (7%). Therefore, the verification and assessment of carbon emission for industry are the premise of energy conservation policy system improvement, especially for the district scale

This article is focused on the Shenzhen International low-carbon city. The relationship between

the industrial carbon emission, the industrial structure and the spatial distribution are discussed on the regional scale. The function to support the related policy formulation is expected.

2.THE OBJECT OF ASSESSMENT

Pingdi locates in the northeastern part of Shenzhen. It is an important part of Longgang downtown with strong economy. Its total area is 57.6 square kilometers with 130 thousand residential. The high pollution and carbon emissions are caused by the 29 manufacturing industries here. Specifically: agro-food processing industry, food manufacturing, beverage manufacturing industry, textile industry, textile and garment, shoes, hat manufacturing, leather, fur, feathers (down) and its manufacturing, wood processing and wood, bamboo, rattan, brown, grass products industry, furniture manufacturing, paper and paper products industry, printing and recorded media replication, Educational and Sports Goods, chemical raw materials and chemical products manufacturing, pharmaceutical

manufacturing, chemical fiber manufacturing industry, rubber products industry, plastic products, non-metallic mineral products industry, ferrous metal smelting and rolling processing industry, fabricated metal products, general equipment manufacturing, special equipment manufacturing, transportation equipment manufacturing, electrical machinery and equipment manufacturing, communication equipment, computers and other electronic equipment manufacturing, instrumentation and culture, office machinery manufacturing, handicrafts and other manufacturing, recycling and materials Recycling and processing industry, electricity, heat production and supply industry, as well as water production and supply, sum of 29 industries.

The raw material access and industrial processes are the main energy activity of the industry.

3.DATA SOURCE AND METHODS

The relevant data is supplied by the “Shenzhen Statistical Yearbook 2012” and the enterprise data from the Economy Science Department of Pingdi.

The quantitative research of green-house gas emission is realized by the time series, life cycle and population joint analysis abroad. The local research was started from last 1990s. Since lacking of the directly data of monitor, most of the estimation are based on the energy consumption. The three most popular method of measuring is: estimated by the energy consumption, population and the gross domestic product (GDP); secondly is refer to the IPCC, estimated by the Energy consumption, carbon oxidation factor and heat energy conversion factor; thirdly, followed the department method recommend by IPCC, estimated by the carbon intensity, energy consumption and gross domestic product. Since there is no directly monitoring data, most of the research is based on the estimation of indirect conversion in China right now.

The first method is applied for large scale estimations, large bias would be produced for the city scale. Limited by the relatively more parameters and less statistical energy data of the second method, we applied the third method. The estimation of carbon emission in Pingdi international low-carbon city is done carbon intensity and energy consumption. The factor of carbon emission is shown in table 1 and 2.

Table 1 Emission factor of electricity

Year	2009	2010	2011
Emission factor of electricity (tCO ₂ /MWh)	0.9987	0.9762	0.9489

Note: The data is obtained from the “China’s regional grid baseline emission factor” (EFOM) year of 2009-2011.

Table 2 Emission factor of fossil fuel combustion - stationary combustion source

Fuel	Emission factor
Anthracite	Value Unit

Bituminous coal	1.97	tCO ₂ /t
Lignite	1.86	tCO ₂ /t
Washed coal	2.06	tCO ₂ /t
Middling	2.45	tCO ₂ /t
Slime	0.78	tCO ₂ /t
Coke	2.85	tCO ₂ /t
Crude	3.02	tCO ₂ /t
Fuel oil	3.17	tCO ₂ /t
Petrol	2.92	tCO ₂ /t
Normal Kerosene	3.03	tCO ₂ /t
Diesel fuel	3.10	tCO ₂ /t
LNG	2.90	tCO ₂ /t
LPG	3.10	tCO ₂ /t
Refinery gas	3.04	tCO ₂ /t
Oxide	3.28	tCO ₂ /t
Natural gas	0.0022	tCO ₂ /m ³
Coke oven gas	0.00089	tCO ₂ /m ³
Blast furnace gas	0.00017	tCO ₂ /m ³

Data source: The standardization and guidance documentation of Shenzhen: "The organization of greenhouse gas emissions quantification and reporting of norms and guidelines" (SZDB/Z 69-2012)

4.CALCULATION

Based on the “Shenzhen Statistical Yearbook of 2012” and “Industrial census data” from the Pingdi government, the estimation is done by the following steps:

4.1 Estimation for the strength and amount of carbon emission in Pingdi

4.1.1 estimation for the strength of industrial carbon emission in Pingdi

The carbon emission could be estimated by the equation recommend from IPCC (Equ. 1). The data of energy consumption is obtained from the “Shenzhen Statistical Yearbook of 2012” (Table 1 and 2).

$$\overline{C}_A = \sum_{i=1}^n \overline{E}_i \cdot \overline{\delta}_i \quad (1)$$

$\overline{E}_i$ stands for the energy consumption of i; $\overline{\delta}_i$ stands for the emission factor of CO₂ corresponding to the energy i; i is the different kind of industry; $\overline{C}_A$ is the total amount of carbon emission in Shenzhen.

The strength of carbon emission in Shenzhen would be estimated by the total amount of emission and business income (Equ. 2):

$$CT_A = \frac{\overline{C}_A}{\overline{G}_A} \quad (2)$$

Since there is close relationship between the emission strength and the added value of the industrial chain, the emission strength of Pingdi would be estimated by the coefficient with the strength of Shenzhen, which is $CT_A = \overline{CT}_A \times 1.2$.

4.1.2 estimation for the business income for the industries in Pingdi

The G_A of 29 industries in Pingdi are obtained from the “industrial census of 2011” of Pingdi government.

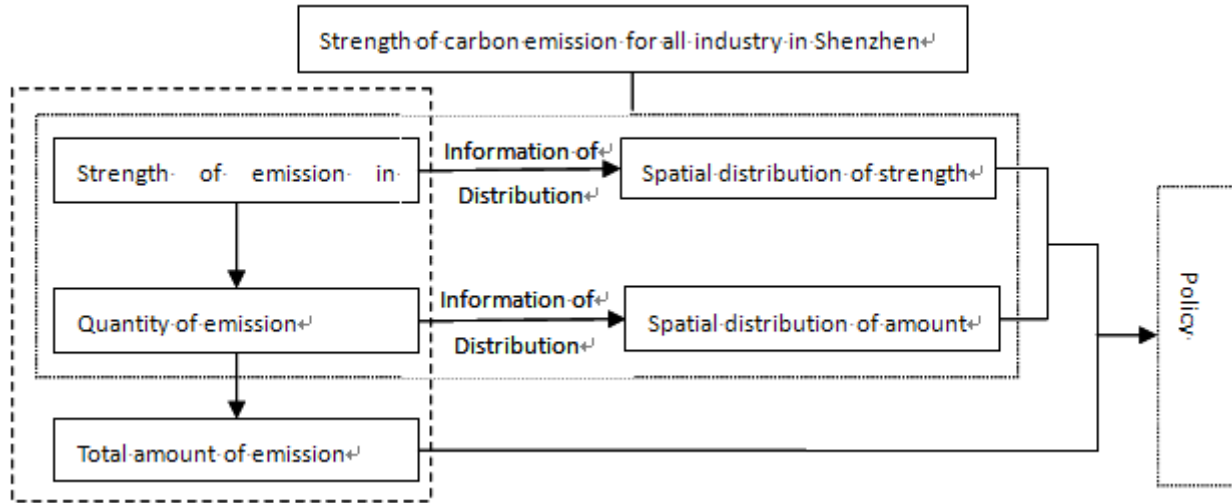


Figure 1 figure for the estimation of carbon emission in Pingdi

4.1.3 estimation for the amount of industrial carbon emission in Pingdi

Followed the equation:

$$C_A = CT_A \times G_A = (\overline{CT_A} \times 1.2) \times G_A \quad (3)$$

C_A is the amount of carbon emission for A; CT_A is the strength of carbon emission for A; G_A is the business income of A.

4.2 Estimation for the spatial distribution of carbon emission in Pingdi

By statistics the amount of carbon emission for all industries (C_i) and the corresponding geography location and the type, the formula is:

$$C_{Di} = \frac{C_i}{S_i}, (i = 1, 2, 3)$$

$i=1,2,3$ stands for the different type of industrial land and S_i is the corresponding acreage. The spatial distribution of carbon emission is estimated by the spatial analysis of Geographic Information Systems (GIS).

5. RESULTS AND ANALYSIS

According to the above evaluation methods, calculation for the carbon emissions from various industries in Pingdi town, Shenzhen, as shown in the following:

As shown in Table 3, the total amount of industrial carbon emissions in Pingdi town is 549,783.72 tons in 2011, and industrial land is 1.07 square kilometers, so average value is 513.86433t / hm². This is a relatively

high level of energy consumption, and the main reason is that existing industrial in Pingdi town mainly are processing and manufacturing industrial with the high energy consumption.

5.1 Analysis of the industrial structure

Pingdi town's industrial structure based on low-end and high energy consumption industries. Communications equipment and electronic equipment manufacturing industry account for 13.32% of the main income of all industries of the district.

Leather, fur, duvet manufacturing industry account for 10.24%; Furniture manufacturing industry accounted for 9.32%; Paper and paper products industry accounted for 9.28%; Electrical machinery and equipment manufacturing industry accounted for 7.86%; Instrumentation and office machinery manufacturing industry accounted for 7.17%; Non-metallic mineral products industry accounting for 6.31%; General equipment manufacturing industry accounted for 5.6%; Plastic products industry accounted for 5.15%; Artwork and other manufacturing industry accounted for 4.38%; Transportation equipment manufacturing industry accounted for 4.31%; Pharmaceutical manufacturing industry accounted for 3.02%; Educational and sporting goods accounted for 2.85%; Clothing, shoes, hat manufacturing industry accounted for 2.66%; Chemical materials and chemical products manufacturing industry accounting for 2.05%; Other industries accounted for 6.48%. This shows Pingdi town's industry type mainly is leather, furniture, paper industries and so on, those are with low-value output and high pollution emissions. This is the main reason

lead to carbon emissions and carbon intensity is too high.

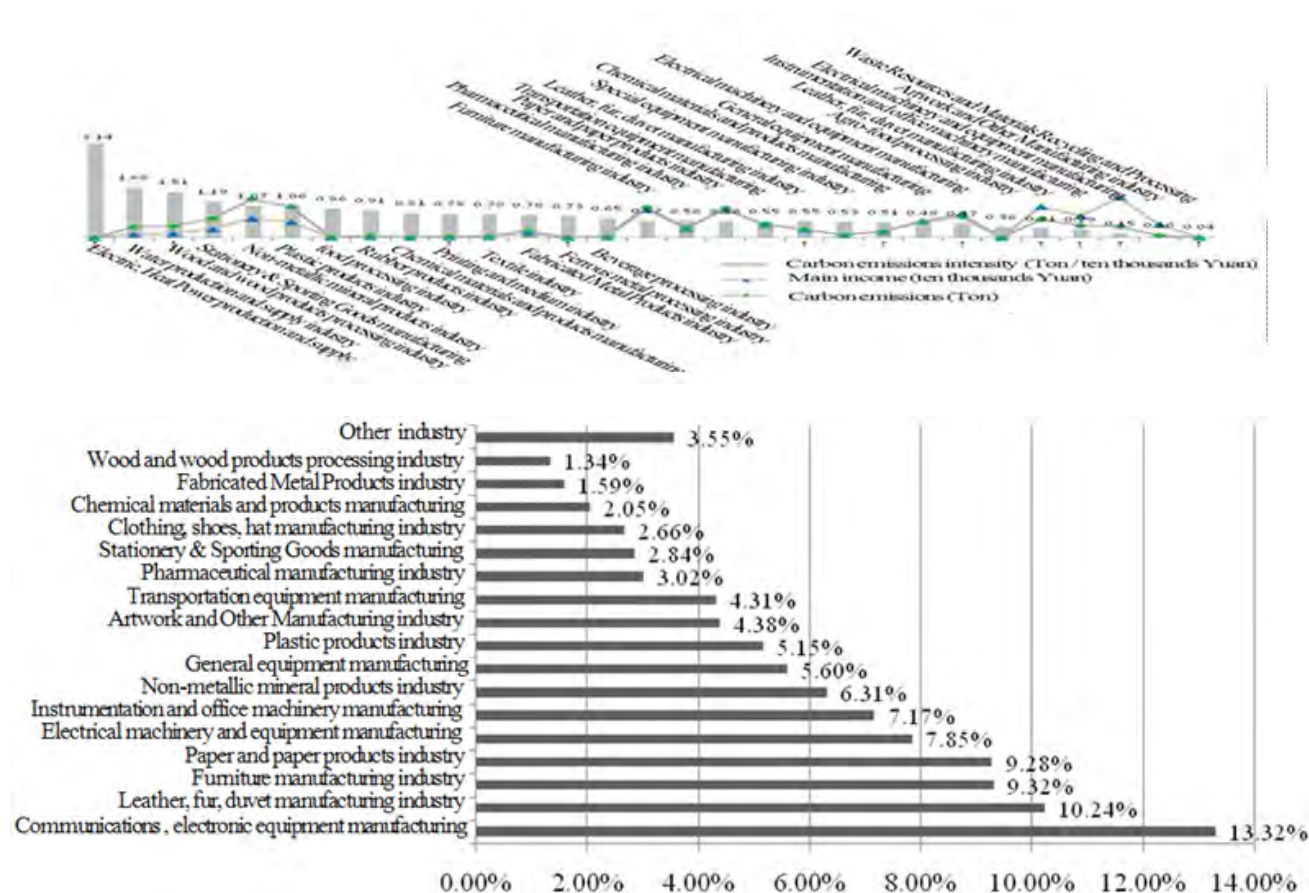


Figure 2 Main business income share in 2011

5.2 Industrial structure and the carbon intensity correlation analysis

As shown in figure 2 and figure 3, Pingdi town's low-pollution industries include communications equipment and electronic equipment manufacturing industry, instrumentation and office machinery manufacturing industry, artwork and other manufacturing and so on. The industry with largest proportion of carbon emissions is communications equipment and electronic equipment manufacturing. Its main business income accounted for 13.32% and carbon intensity is 0.15 tons / ten thousand Yuan.

Figure 3 Carbon emissions intensity, Main income, and Carbon emissions amount comprehensive diagram for various industries at 2011, in Pingdi town, Shenzhen

The vast majority of carbon emissions are from high pollution and low value-added industries. Such as Stationery and Sporting Goods manufacturing industry (operating income account for 2.84%, carbon emissions account for 6.33%, carbon intensity 1.19 tons / ten thousand Yuan), non-metallic mineral products industry

(operating income account for 6.31%, carbon emissions account for 12.61%, carbon intensity 1.07 tons / ten thousand Yuan), plastic products industry (operating income account for 5.15%, carbon emissions account for 10.22%, carbon intensity 1.06 tons / ten thousand Yuan), furniture manufacturing industry (operating income account for 9.32%, carbon emissions account for 10.27 %, carbon intensity 0.57 tons / ten thousand Yuan), paper and paper products industry (operating income account for 9.31 %, carbon intensity 0.56 tons / ten thousand Yuan), leather, fur, duvet manufacturing industry (operating income account for 10.24%, carbon emissions account for 10.27 %, carbon intensity 0.31 tons / ten thousand Yuan).

5.3 Spatial distribution of industrial carbon emissions

On the basis of these mentioned above, depending on GIS spatial analysis, carbon emissions data for more than 700 companies engage with their geographical locations as shown in figure4 and figure 5.

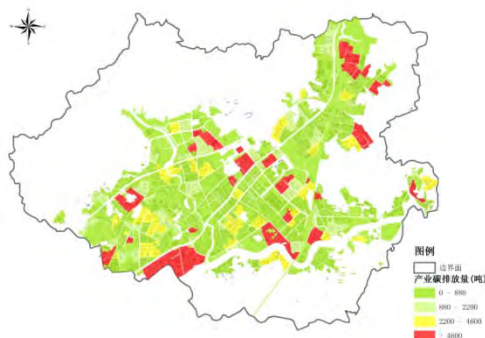


Figure 4 spatial distributions of industrial carbon emissions

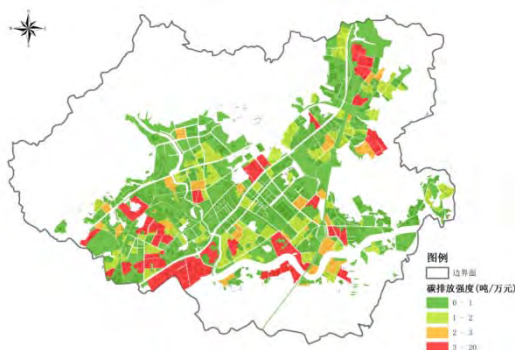


Figure 5 Spatial distribution of industrial carbon intensity

Several Biggest Emission areas are distributed near to Dingshan River and East Incineration plant. Printing and dyeing industry, Mining industry and so on, those high-polluting industries is located close to natural water and mountain region, and resulted in serious impact on the local environment. As shown in figure 5, focusing on the carbon intensity, the biggest emission intensity area mainly is at the northeast and southwest areas of Pingdi. The majority of them is on Dingshan River and caused severe pollution to the river water quality. By actual estimates, Heavy metals, in particular mercury pollution is seriously overweight. It is directly related to the sewage discharge from printing and dyeing industry, Mining industry and so on. This shows that industrial carbon emissions reduction should not only require to change in aspect of industrial structure, also industry spatial distribution should be concerned, since impact on the ecological environment.

6. SUMMARY

Sectoral approach is recommended by IPCC. From top to bottom, Pingdi international low-carbon city's industrial carbon emissions and carbon intensity is calculated. The results showed that, the industrial structure mainly is related to leather, furniture, and paper and so on. Those are high pollution and low added value industries. As a result, industry carbon emission in the region is too high. As energy saving measures for local industry, the dual goals of economic development and ecological protection should be taken

into account. Thus, the key to the construction of ecological city is to optimize the industrial structure and spatial layout planning, and the coordinately develop the industry, living and ecological.

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APPENDICES

Table 3 Carbon emissions data for various industries at 2011, in Pingdi town, Shenzhen

Types of industries	Business income	Carbon emissions intensity	Carbon emissions
Agro-food processing industry	18.70	0.36	6.82
Food manufacturing industry	1288.60	0.96	1238.43
Beverage manufacturing industry	2710.50	0.65	1756.78
Textile industry	2950.40	0.78	2307.018
Clothing, shoes, hat manufacturing industry	27261.00	0.55	14949.348
Leather, fur, duvet manufacturing industry	105011.30	0.31	33053.77
Wood and wood products processing industry	13707.10	1.51	20754.34
Furniture manufacturing industry	95616.30	0.57	54777.39
Paper and paper products industry	95169.10	0.56	53048.98
Printing and medium industry	2718.50	0.79	2148.78
Stationery & Sporting Goods manufacturing industry	29172.60	1.19	34807.42
Chemical materials and chemical products manufacturing industry	21014.70	0.51	10724.77
Pharmaceutical manufacturing industry	30939.90	0.56	17395.81
Chemical fibre manufacturing industry	552.90	0.81	447.61
Rubber products industry	3316.30	0.91	3015.98
Plastic products industry	52858.10	1.06	56211.50
Non-metallic mineral products industry	64753.40	1.07	69339.96
Ferrous metal processing industry	90.00	0.73	65.86
Fabricated Metal Products industry	16298.30	0.76	12315.21
General equipment manufacturing industry	57473.30	0.49	28413.55
Special equipment manufacturing industry	9291.00	0.53	4908.24
Transportation equipment manufacturing industry	44240.00	0.55	24512.46
Electrical machinery and equipment manufacturing industry	80549.30	0.47	37978.23
Communications equipment and electronic equipment manufacturing industry	136548.00	0.15	20311.66
Instrumentation and office machinery manufacturing industry	73534.80	0.29	21639.66
Artwork and other manufacturing industry	44924.90	0.096	4308.29
Waste resources and materials recycling and processing industry	2114.20	0.04	82.84
Electric, Heat Power production and supply industry	110.00	3.14	345.36
Water production and supply industry	11283.30	1.68	18917.69

Data source: According to the 《2011 Shenzhen Statistical Yearbook》, 《2011 industrial census》

UTILIZATION OF MULTIPHASE FLOW TECHNOLOGY FOR BIOGAS INDUSTRY INTEGRATION

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ABSTRACT

Due to congestion, precipitation, scarce liquidity and other key barriers, the traditional biogas technology doesn't work efficiently, which is the reason for limited market share and slow industry growth. This paper introduces two new technologies: The multiphase flow and Air insulated heat preservation. The multiphase flow technology makes the fermentation liquid mixed more adequate through the introduction of fluidization mechanism, which results the reactions more rapidly and properly. Introduction of air insulation technology can reduce the heat loss of fermentation liquid, encourage the anaerobic digestion. The new technology can promote the gas production rate from less than $1.0\text{m}^3/\text{m}^3$ increased to $2.5\text{m}^3/\text{m}^3 \cdot \text{d}$, which brings the comprehensive integration of biogas technology, thereby changing the feature of the industry growth pattern, scale production and systematizing become possible, and the use of biogas is likely to become the routine for rural life energy supply, as well as the treatment for "three wastes". These results a major impact on the direction of biomass energy industry and ecological community, especially construction mode of small rural town.

Keywords: biogas; multiphase flow; energy; treatment of "three wastes"; ecological community.

1. INTRODUCTION

Biomass energy relate directly to people's daily life style and production methods, compared with solar and wind energy, it is more important in the future development especially the future development of low carbon communities. The characteristics of biomass energy has many advantages: 1)biogas is a kind of ecological energy, with no further pollution; 2) fermentation residua is a natural organic fertilizer, which relate closely to ecological agriculture; 3)Biogas raw materials is widely distributed with variety of biomass can be used; as known that biogas have better quality and higher calorific value, is the ideal fuel for daily life and motor power and electric generator etc., which have high economic value. Regards to the advantages, start from 1950s the Chinese government strongly support for the development of biomass energy technology. In recent years, western developed countries have continuous rising interests in biomass energy development; currently the biogas technology is developing rapidly.

2. THEORY AND PATTERN OF LOW CARBON COMMUNITY

Low carbon community is the ideal pattern of the future urbanization infrastructure construction. To be more scientific, it should include three major elements known as ecological life system, high efficient biogas system and distributed energy system.

2.1 Ecological life system

Divide The most essential of ecological life system is people's mode of production and way of life are all base on original cultural environment. According to the dissipative structure theory, both biological system and social system will eventually depend on utilization and conversion issues of energy. Waste management links the use and conversion of biological supply chain and biogas system is the central hub for modern ecological life.

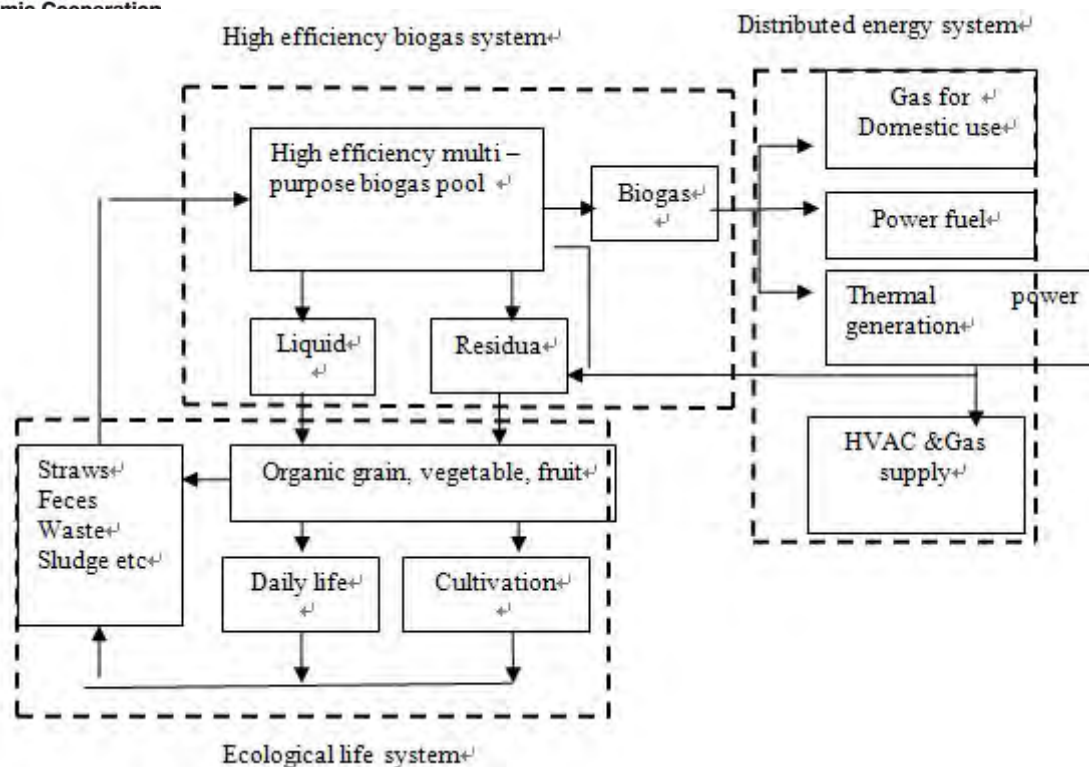


Fig 1 Low carbon community ecological system schematic diagram

2.2 High efficiency biogas system

High efficiency biogas system is combined with the biological technology to processing domestic waste. The high efficient biogas system is based on the existing biogas technology but integrated to a new type of highly efficient, large capacity and high-technology biogas system. Despite the Chinese government vigorously promote biogas industry for nearly 60 years, as for technical development, the traditional characteristics of the static fermentation hasn't changed, the vast majority of rural biogas plants still using natural fermentation. In fact, current existing biogas technology worldwide cannot achieve the requirement of the construction and development for low carbon community.

2.3 Distributed energy system

The low application value and economic effect is one of the major reasons that biogas has yet to form a mature industry. The proposed distributed energy system is based on the existing biogas application technology to raise a new type of biogas utilization system. It is based on community's surrounding social environment, including local residents, climate, the condition of biomass resources, factories and energy consumption and so on carries out the scientific and reasonable production and supply of energy, in order to utilize biogas application value and improves energy efficiency significantly. For example, solely biogas power generation, the thermal efficiency is about 25%, if the distributed energy system is used, its energy efficiency can reach up to 70%, economic benefit can increased by 30 to 50%.

3. HIGH EFFICIENCY BIOGAS SYSTEM IS THE KEY FOR LOW CARBON COMMUNITY DEVELOPMENT

In the above described three systems, green ecological food chain has gradually been recognized, especially in rural China is implement reform of land circulation, intensive farm mode is gradually formed, production of organic food, vegetables and meat have its initial production conditions, the main obstacles is the need of organic fertilizer. If we use agricultural and animal waste to produce of organic fertilizer through anaerobic digestion, the basic resource supply for organic fertilizer production can be solved.

Distributed energy system substantially is part of the traditional energy supply technology. It is crucial to scientifically allocate the energy production and utilization. Therefore, development of biogas system is the key in the low carbon community. The current biogas technology must be under a technical renovation and improvement. Existing process of the fermentation digestion is with basic feature of static and normal temperature fermentation. Static fermentation that refers to the material in the fermentation tank in a stationary state, there is a series of problems of uneven material distribution, slow heat and mass transfer, jams, accumulation, blind zone, poor liquidity, discharging problem, and low level of mechanization, these may cause pool effective volume, lower gas production rate, poor system reliability, short service life and system maintenance difficulties. Although there are some technical improvements in the mainland China previously, such as mixing devices can increase the mass transfer efficiency and accelerate the biogas production process in the biogas project (Liu Y., 2009).

Western developed countries in recent years the development of mechanical stirring technology and so on (Hopfner K., 2006). But those improvement did not improve the essential characteristic of biogas technology, with these defects, the biogas technology industry grow slowly. Each year the burning of straw in rural areas has become a serious social problem, poor manure treatment, industrial and agricultural waste such as sewage sludge cannot get effective treatment, and these not only waste biomass resources, but also pollute the environment

4. TECHNICAL SOLUTION FOR HIGH EFFICIENCY BIOGAS SYSTEM

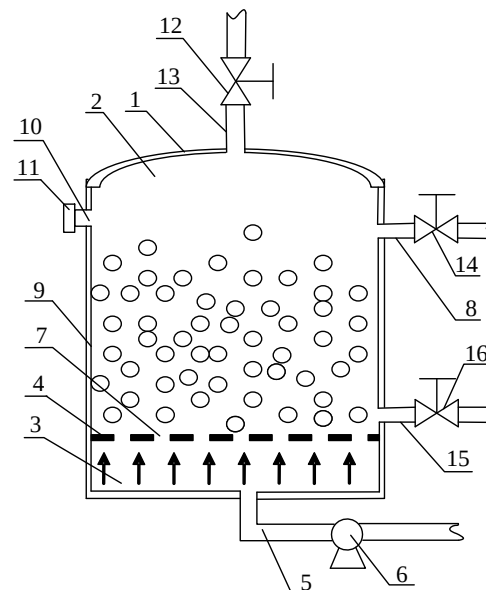
Any references cited in the abstract must be given in full. Unpublished results and personal communications are not recommended in the reference list, but may be mentioned in the text. If these references are included in the reference list they should follow the standard reference style of the journal and should include a substitution of the publication date with either 'Unpublished results' or 'Personal communication' Citation of a reference as 'in press' implies that the item has been accepted for publication.

4.1 Current issues for fermentation temperature

The efficiency of biogas production rate is sensitive to temperature, according to the research of Sun (2008) etc, experiment on the wheat straw materials, with increased fermentation temperature 53 °C and 30 °C, the gas production rate of former is doubled, which also shorten the fermentation time by 37.5%, the cumulative gas production increased by 7.21%, raw materials biogas production rate increased by 8.9%. If fermentation temperature below 10 °C, gas production rate is close to 0 (Bai J., H., 2008). Especially in majority part of mainland China where the winter natural environment temperature is below 10 °C, natural fermentation pool almost impossible to operate. China has approximately 45 million house biogas digesters, most of which are basically using natural fermentation technology. In large and medium-sized biogas plants mesophilic fermentation is widely used, but thermophilic fermentation is rarely used. Hence study of thermophilic fermentation technology can improve the efficiency of biogas system significantly.

4.2 Current issues of Static fermentation

The problem within static fermentation is the key to optimize biogas technology. Lai (2011) proposed the basic structure of multiphase flow fermentation digester. The invention first introduce multiphase flow technology into biogas industry, the basic structure schematic diagram is shown in fig. 2.



1. digester 2. fermentation area 3. liquid room
4. distributor 5. input pipe 6. pump 7. sieve plate
8. output pipe 9. anticorrosive coating 10. inlet
11. feed machine 12. gas valve 13. gas pipe
14. water valve 15. outlet 16. discharge valve
Fig.2 structure of multiphase flow fermentation digester

The basic principle of this technology is use fermentation liquid as fluidization medium, and pump for power, through water chamber and distribution board fluidize the solid material granular, between material and biogas slurry formed fluid - solid two-phase flow. As it also involve methane, carbon dioxide and other gases in digester, the system is a typical system of multiphase flow, digester is referred to as the pool of multiphase flow. However, main structure is liquid-solid two phase flow, which is commonly referred to as the fluidized bed. Therefore, multiphase flow digester can also be called fluidized bed fermentation digester.

As known that fermentation liquid and material formed the fluidization state, it has the general properties of fluid. In comparison with the traditional static digester, including mechanical stirring digester in recent years, it has essential differences in principle and mechanism.

First of all, the basic characteristics of fluidization state are the nature of fluid, so it is easy to flow and mixing, which has fundamentally solved the traditional digester jams, accumulation, blind zone, poor material distribution, the difficulty of discharging. The solutions of these problems can greatly increase the effective volume of the digester and methane gas rate and increasing reliability. Take straw biogas plants for example, solid content within the digester is about 0.15 ~ 0.18 L/g under the condition of high temperature fermentation, the fermentation cycle is 17 days, according to the Sun (2008) and others in the laboratory test, one kilogram of wheat straw gas production rate is 0.535 m³, it can calculate that the unit volume production gas rate should be 4.72 ~ 5.66 m³/m³·d. It

means that digester unit volume gas production rate should be above $4.7 \text{ m}^3/\text{m}^3\cdot\text{d}$ as make full use of the volume. Multiphase flow digester can make fully mixed of materials, and ideally it can makes the gas production rate reach up to 70% , according to author's experiences, multiphase flow digester unit volume gas production rate is approximate about $3.3 \sim 4.0 \text{ m}^3/\text{m}^3\cdot\text{d}$. Even for urban sludge where with low solid content, multiphase flow pool of unit volume gas production rate is above $2.5 \text{ m}^3/\text{m}^3\cdot\text{d}$. For animal waste treatment, the multiphase flow gas pool can produce up to $3.0 \text{ m}^3/\text{m}^3\cdot\text{d}$. Furthermore, it is the process of industrialized device control. For traditional digester, normally use static fermentation process. Temperature, concentration and fermentation degree is uneven distributed. Among them, the unevenly temperature distribution leads to the pool partially overheating or too cold. In this case, the traditional digester will not be able to make to satisfy the overall high temperature fermentation process, more commonly used fermentation process usually adopts to medium temperature fermentation. Therefore, implementation of high temperature fermentation in large biogas project, the issue of temperature control must be solved. Multiphase flow technology effectively solved the problems of the digester temperature and process control, as these will be the solid technical milestone for the large-scale, high temperature fermentation, highly efficient biogas engineering system.

Temperature control has potential to achieving major technological breakthrough in the ideal temperature for anaerobe. As for anaerobic fermentation, it is a bacterial ecological process, involving dozens of bacterial interactions and effects. Various bacteria's ideal fermentation temperature is different, vary from 15°C to 100°C . Especially a bacteria names *M.jannaschii* is the fastest bacteria that generate methane, generation time is only 26 minutes, the best fermentation temperature is as high as 85°C . If the problem of temperature control can be solved, it may adjust the different bacteria optimal fermentation temperature in industrialized system, the realization of this technology will greatly improve the digester gas rate and the related technical indicators.

5. THE PROSPECT OF MULTIPHASE BIOGAS TECHNOLOGY

The advantages of multiphase flow technology can solve many problems brought by the static biogas technology, but may also have new findings in the biogas fermentation process , thereby brings significant influence and development to the entire biogas industry. This theory is sound because it is based on the fact that the multiphase flow has excellent mass transfer properties.

Biogas fermentation process includes stages of hydrolysis fermentation, acid production, hydrogen generation, and methane production. There are dozens of different bacteria which take part in the process, which forming an ecosystem with coaction in between.

The energy and mass transfer in the system greatly affect bacteria's' growth and reproduction. Firstly, mass transfer directly affects the nutriments and excrements in the biogas digester. Obviously, uneven distribution of mass causes uneven bacteria distribution, results in decrement of effective volume to the digester. On the other hand, mass transfer exists in between biological individual and the surrounding environment. As known that biological process need to consume nutrients and discharge waste. Therefore, a well performed mass transfer system has direct influence on the growth of bacteria, reproduction and bacterial density

For static fermentation technology, the mass transfer is mainly relying on gathered diffusion mechanism. In colloidal solution, the mass diffusion coefficient is very big, especially with high concentrations of particulate matter in the pool; mass diffusion will be significantly affected. Therefore, poor mass transfer characteristic is one of the most important defects existing in current biogas technology.

In fluidized state, mass transfer is depends on the fluid motion in the form of emulsion transferred. Normally, the motion of flow field caused by external force is far stronger than the one resulted from gathered diffusion of density gradient, is the matter of differences magnitude order. Therefore, within the fluidized bed the mass distribution is usually uniform, ideally it can be regarded as uniform distribution. Thus infer that multiphase flow distribution of bacteria in the pool is evenly distributed, it will improve the effective volume of the biogas pool significantly. For individual bacteria, the consumption of nutrients and waste discharge needs mass to transfer it. In the static biogas pool, these substances transfer still relies on gathered diffusion mechanism. In the flow field of multiphase flow state, the bacteria gather on excremental source mass and nutrients. The mass exchange occurs in between of the bacteria and the surrounding environment. Which can demonstrate that even in the case of slow flow stream, the mass transfer around the bacteria account as the turbulent diffusion, mass diffusion is mainly depends on the flow field; the transmission speed is far faster than gathered diffusion. As a result, multiphase flow have better living environment for bacteria compared with the static state biogas pool.

6. CONCLUSION

Biogas technology plays a vital important role in low carbon community construction, ecological energy and ecological environment, ecological agriculture and ecological farming. Future ecological low carbon community will include ecological life system, highly efficient biogas system and distributed energy system. Among them, the biogas system is the key to the pattern, it must break through the existing static fermentation process, thorough major technological breakthrough from static fermentation to dynamic fermentation. Multiphase flow biogas pool presented in this paper not only addressed in principle and mechanism of the static fermentation technology, also

from the engineering process where it can solve the problem of existing comprehensive biogas technology, and is likely to be have major breakthrough in biogas biotechnology due to its excellent heat transfer and mass transfer performance.

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ENERGY-EFFICIENT APPROACH BY VARIABLE FLOW AND DUAL TEMPERATURE CHILLED WATER SYSTEM SERVING IN A HIGH-TECH INDUSTRIAL BUILDING

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ABSTRACT

The HVAC systems for high-tech industrial building are energy-intensive and sophisticated in that they operate 24 hours per day year-round and need large amount of super clean air to serve cleanroom requirement for manufacturing. Since the traditional chilled water system with constant flow water pump operates continuously, it is important and significant to consider energy efficient strategies for cost down concern in the operation of the facility. The objective of this study is to present the energy-saving approach on performance evaluation of the chilled and cooling water system retrofitted by a variable speed driven (VSD) chilled and cooling water system. Field measurements of a chilled and cooling water system with 12 chillers (17020 refrigeration tons totally) have been carried out at a high-tech industrial building in the Science Park of Chinese Taipei. The on-site real-time data from supervisory control and data acquisition system for one year has been applied to evaluate the energy-saving while conducting the VSD approach under different ambient air condition for the HVAC system. The results revealed that it is feasible to achieve satisfactory energy-saving of 5-17 % for higher temperature (12 °C) chilled water system and 5-30 % lower temperature (7 °C) chilled water system depending on ambient weather condition. It will stimulate a more robust investigation for energy-efficient approach HVAC system specific for high-tech industrial building.

Keywords: Variable speed driven, Chilled water pump, Cooling water pump, Energy-efficient.

1. INTRODUCTION

The energy consumption of HVAC system in high-tech building is tremendous due to year-round operation for serving cleanroom requirement. It is essential to conduct energy-efficient approach of HVAC system for cost down concern. Variable speed driven (VSD) chillers and pumps can operate much more efficiently at partial load conditions. The most common way employed in high-tech manufacturing building is a primary/secondary chilled water system with constant-speed for primary pumps and variable-speed for secondary pumps respectively. Yu et al. [1] investigated the environmental performance and economic analysis of all-variable speed chiller systems which included primary chilled water pumps, condenser water pumps and cooling tower fans. It decreased the annual electricity use by 20% and make chiller systems more sustainable. Besides, variable primary flow application for chillers could reduce energy consumption by 5% [2]. Economic benefits of a water-cooled chilled system with VSD approach for condenser water pumps and cooling tower fans were studied extensively. It also represented the reduction of energy consumption by 8%

with application of load-based variable speed control approach [3].

Modeling and energy simulation of the variable refrigerant flow system with water-cooled condenser was investigated by Li et al. [4] through energy simulation code Energy Plus. Besides, comparison of energy efficiency between variable refrigerant flow systems and ground source heat pump systems was also studied comprehensively [5]. Furthermore, Hydeman et al. [6] developed a modified chiller model which can be applied to evaluate the VSD chiller under different condenser water flow operation. A parametric analysis for optimizing the control sequence of VSD chiller was presented extensively. Furthermore, Hartman [7] also highlighted the all variable speed chiller plants which chiller, condenser pumps and cooling tower fans were driven by VSD to achieve the highest energy performance especially at partial load conditions.

Although many research has been done on the theoretical study and optimizing methodology for variable speed driven approach of both primary chiller water pumps and cooling water pumps in HVAC system, little quantitative information is available for implementation of VSD approach in a high-tech building especially for a long term monitoring and

analysis. In this study, the power consumption of a fabrication building for high-tech manufacturing in Science Park of Chinese Taipei will be investigated. The long term acquisition of power consumption data for more than one year will be analyzed extensively after retrofitting project by conducting VSD approach at different inverter frequency under different ambient air condition. Energy-saving percentage can be evaluated and presented based on real-time and on-site data year round.

2. SYSTEM DESCRIPTION

The investigated high-tech building is located in the science park of Chinese Taipei. With products spanning the full range of TFT-LCD panel modules and touch panels, its floorage is approximately 263,938 m² with total fabrication cleanroom (ISO class 4, 5, 6) area of 40,184 m². The schematic diagram of chillers plant with dual supply temperature of the HAVC system in this building is shown in Figure 1. There are seven centrifugal chillers (one for back up unit) which supply water temperature is 6°C (CH6) with each capacity of 1360 refrigeration tons to serve the demand from make-up air handling unit (MAU) and HVAC system. Besides, there are five centrifugal chillers (one for back

up unit) with each capacity of 1500 refrigeration tons which serves the demand of process cooling water (PCW), dry cooling coil (DCC) and compressed dry air (CDA) systems. There are eight cooling towers with the specification of flow rate 23,846 liter/min (6300 gallon per minute, GPM) and power consumption of 94 kW (125 HP) for fan.

The primary /secondary chilled water system with constant speed for primary chilled water pumps (CHPs) and variable speed for secondary chilled water pumps respectively serves the HVAC cooling load demands. There are five chilled water pumps with flow rate of 20,595 liter per minute (LPM), head of 526 m and power consumption of 225 kW in the secondary CH6 chiller water system. Four CHPs with specification of flow rate 21,005 LPM, head 496 m and power 188 kW are equipped in the secondary CH12 system. There are 7 primary CHPs (11,442 LPM x 154 m x 37.5 kW) for CH6 system and 5 primary CHPs (12,619 LPM x 154 m x 37.5 kW) for CH12 system respectively. As for cooling water pumps (CWPs), 7 CWPs (13,577 LPM x 275 m x 75 kW) are equipped for CH6 system and 5 CWPs (14,512 LPM x 275 m x 94 kW) are equipped for CH12 system to pump cooling water to cooling towers respectively..

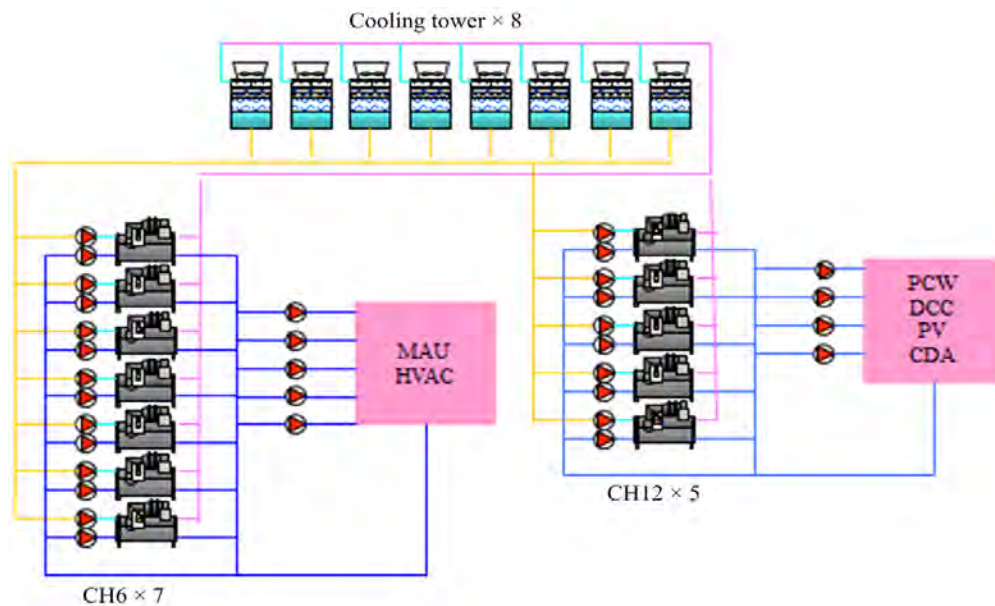
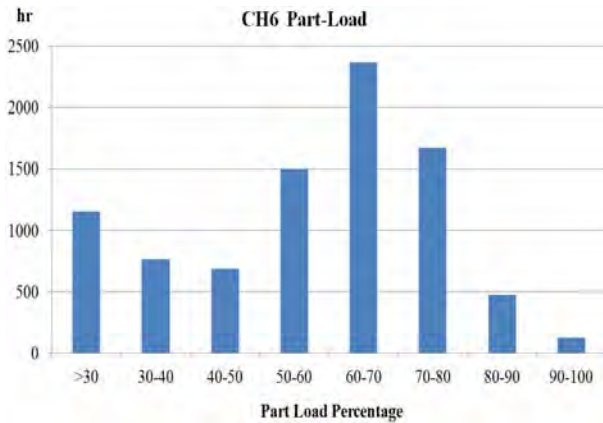


Figure 1 Schematic diagram of chillers plant with dual supply temperature

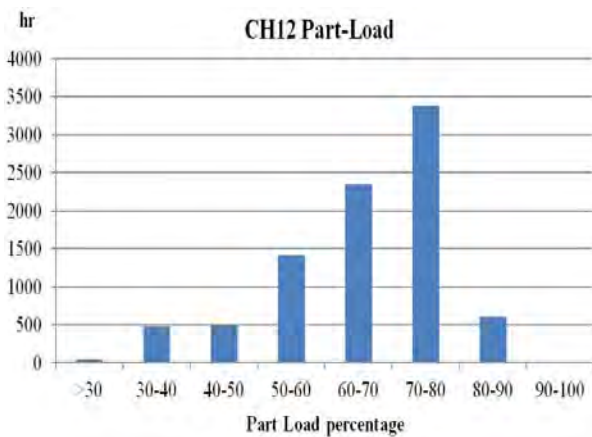
3. VARIABLE SPEED DRIVEN APPROACH AND MEASUREMENT

Figure 2 depicts the operation hours of partial load percentage for the chillers plant with CH6 and CH12 systems in one year. Most of chillers operate at 70 to 80% of full load and the majority of operation hours for the CH12 system is 60 to 70% of full load for CH6 system respectively. It reveals energy-saving potential to conduct the variable speed driven (VSD) approach because the energy performance of partial load chiller is

lower than that of full load condition. The power consumption can be reduced substantially by VSD pumping of primary chiller water pumps (CHPs) and cooling water pumps (CWPs) as well.



(a) lower supply temperature (CH6)



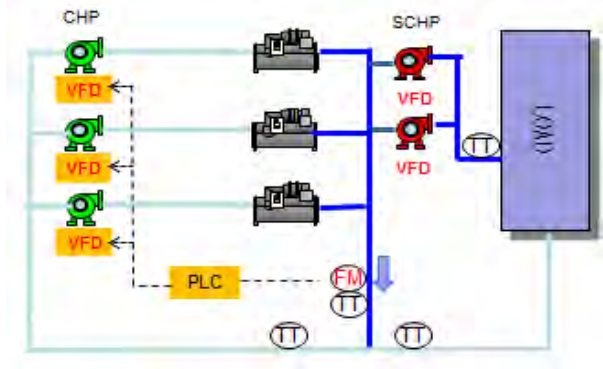
(b) higher supply temperature (CH12)

Figure 2 Operation hours of partial load percentage for dual temperature chilled water system

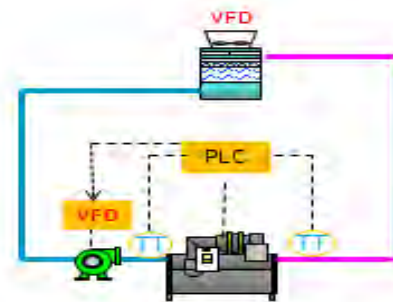
The schematic diagrams of the VSD approach for CHPs and CWP are shown in Figure 3. Inverters were installed to each primary CHP along with a flow meter and a programmable logic controller (PLC). The optimization function is to minimize the chiller water flow rate from common line (decoupling pipe) between primary and secondary chilled water system as shown in Figure 3(a). As the cooling load demand decreases, the flow rate through the common pipe increases accordingly. It will result in lower flow rate than the set point and affect the temperature difference of chillers to modify the inlet guide vane for unloading operation of chillers along with primary pumping systems. Furthermore, the energy-saving approach can be achieved by implementing VSD approach of cooling water pumps (CWPs) as presented in Figure 3(b). The temperature difference of chiller will decrease due to lower cooling load demand. Optimal control through PLC programming is to adjust cooling water temperature of CWPs and partial load of chillers will realize the energy-saving approach through VSD of CWPs especially during partial load operation of HVAC system.

To obtain the reliable measurement data for energy-saving analysis, a very sophisticated system of supervisory control and data acquisition (SCADA) system was employed to conduct the data acquisition

from the entire HVAC system. The operation data of power consumption and frequency of VSD pumps, temperature of chilled water system and cooling water system and wet bulb temperature along with enthalpy of ambient air can be recorded and analyzed extensively. The SCADA systems historically distinguish HVAC control system parameters at all time interval. Snapshot of the SCADA for the investigated HVAC system is presented in Figure 4.



(a) Primary chilled water pumps



(b) Cooling water pumps

Figure 3 Variable speed driven approach

4. RESULTS AND DISCUSSION

After installation of VSD approach for CHP and CWP for the chiller system (VSD mode), data collection from SCADA system for one year was conducted to evaluate the energy-saving potential. Experiments were employed for original operation with constant speed drive at inverter frequency of 60 Hz (base-case mode) for one hour and then shift to the VSD mode for another hour. Figure 5 depicts the typical power consumption for experiment shifting from base-case mode to VSD mode during 2.5 hours including transition stage along with recording ambient temperature variation. The power consumption decreases substantially from 2,600 kW to 2,300 kW at VSD mode under the ambient temperature of 24 °C. It reveals that the implementation of VSD approach is feasible and effective in terms of energy-saving concern.

Power consumption of CHP and CWP might vary due to cooling load variation and different ambient condition. Data acquisition from SCADA system was analyzed extensively under different ambient condition year round. All of the data were collected and compared

based on the same enthalpy of ambient air under base-mode operation and VSD approach. As shown in Figure 6, the power consumption of CHPs of CH12 system decreases while conducting the VSD approach. However, the temperature difference of chiller might increase from 4 °C to 6 °C. The control algorithm of flow rate at the common pipe should adjust automatically to meet the VSD approach to avoid increase of power consumption due to low temperature difference of chillers.

The percentage of energy-saving for VSD approach of CH12 and CH6 systems under different ambient condition is shown in Figure 7. Energy-saving percentage decreases with higher enthalpy air for both of CH12 and CH6 system due to higher cooling load demand under hot weather condition. Limited operation hours will be available to conduct VSD approach during summer time. Furthermore, the energy-saving percentage of CH6 (varying from 5% to 30%) is higher in Figure 7(b) than that of CH12 (varying from 5% to 17%) in Figure 7(a). These results can be explained by less sensitive effect for higher chilled temperature (CH12) system which meets the demand of process cooling water and dry cooling water system for manufacturing inside the cleanrooms. It also reveals that energy-saving potential can be achieved significantly for lower chilled temperature system (CH6) which related to make-air handling units and HVAC system.

Figure 8 presents the power consumption of primary chiller pumps and cooling water pumps while conducting the VSD approach under different inverter frequency (Hz). The power consumption of cooling water pumps decreases due to unloading operation while the inverter frequency decreases. However, as the frequency decreases, power consumption of the primary chiller pumps declines at the beginning but increases after the inverter frequency of 40 Hz. It reveals that there exists limitation for variable primary flow while conducts VSD approach. Further study for optimizing the integration of CHPs and CWP by optimization program is still undergoing to obtain more concrete energy-saving approach.

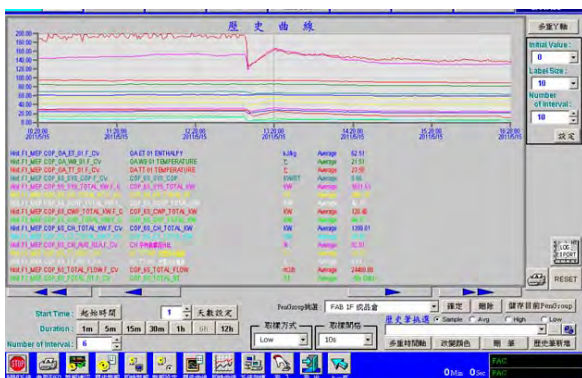


Figure 4 Snapshot of the SCADA system

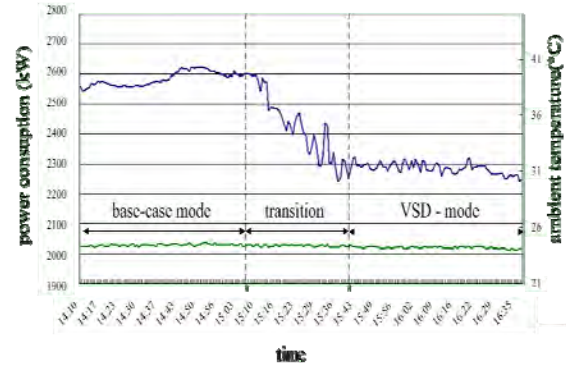


Figure 5 Power consumption of base-case mode and VSD mode

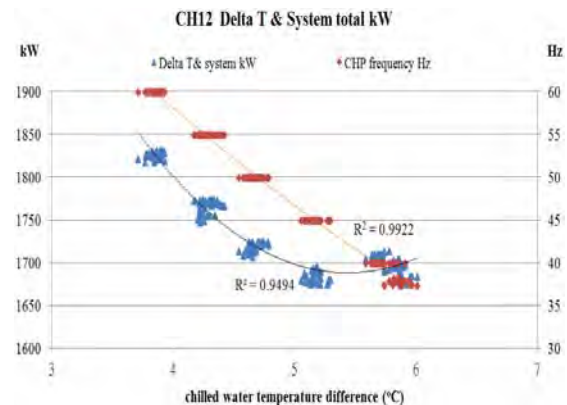
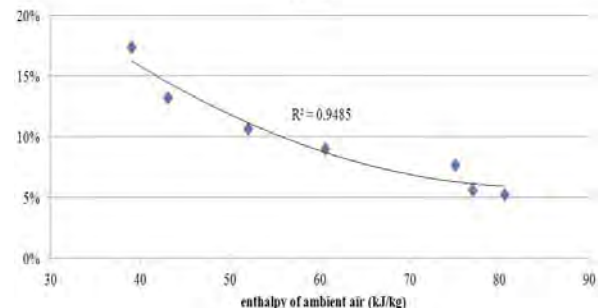
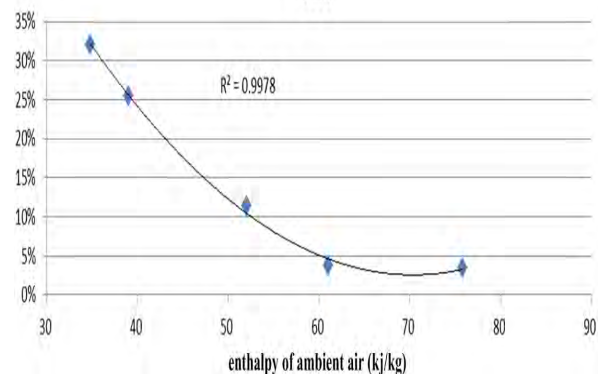


Figure 6 Power consumption variation of chilled water pumps at different frequency CH12



(a) higher temperature supply CH6



(b) lower temperature supply

Figure 7 Energy saving of variable speed driven pumps for dual chilled water system

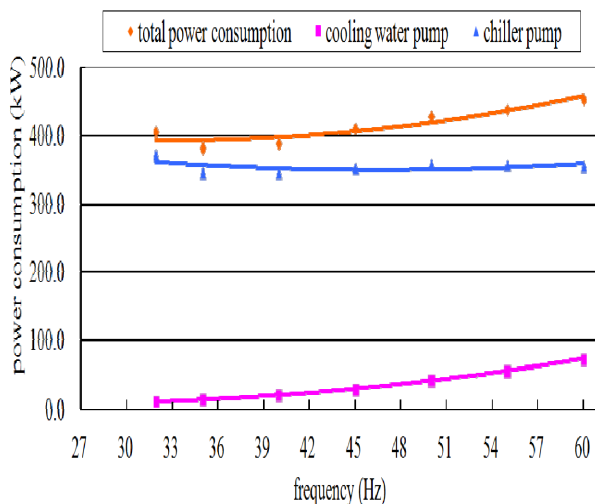


Figure 8 Power consumption of primary chilled water pumps and cooling water pumps

5. CONCLUSION

This study investigates the energy-efficient approach through conducting the variable speed driven pumping for primary chiller pumps and cooling water pumps in the HVAC system of a high-tech building in the Science Park of Chinese Taipei. After retrofitting by the VSD approach, the data acquisition from SCADA system have been analyzed and compared comprehensively to evaluate the energy-saving potential under different ambient air condition for one year. The percentage of energy-saving ranging from 5- 17% for higher temperature chiller system (CH12) and 5-30% energy-saving percentage for lower temperature chiller system (CH6) respectively depending on the ambient air conditions. Substantial energy-saving results in this study by implementing the VSD approach should provide valuable information for the facility engineers facing the energy-saving and cost-down concern in high-tech building

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RESEARCH ON BUILDING ENERGY CONSERVATION COUNTERMEASURE BASED ON ENERGY MANAGEMENT CONTRACTING MECHANISM

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ABSTRACT

Based on energy management contracting mechanism and analysis of China building energy consumption situation and development trend, this paper probes into problems and countermeasures of building energy efficiency in China, which is of important theory value and realistic significance to promoting building energy efficiency in China. Firstly, this paper analyses both the problems in the work of China building energy efficiency and the features of building energy efficiency market, thus analyses major causes of building energy efficiency lagging behind. Secondly, this paper constructs theoretical model of building energy efficiency under market mechanism, and expounds its defining mechanism under market mechanism, and function mechanism of mandatory policy, encouraging policy and technology advance theoretically. Thirdly, by combining the above theoretical study and particular practice of China, and using the experience of other countries for reference, this paper puts forward both the overall train of thought and implementation tactics of China building energy efficiency, and also puts forwards some policy suggestions on its implementation. Furthermore, this paper indicates that the stagnation and backward of China building energy efficiency work is due to imperfect market mechanism in building energy efficiency field, including both market ineffectiveness caused by external economy and information asymmetry and market power deficiency as the result of irrational price mechanism. The function of market mechanism is further restricted because of China government relying more on administrative means to promote building energy efficiency than taking measures to solve the above problems. Therefore, we should put emphasis on stimulating market demand of building energy efficiency, and cultivate and improve building energy efficiency market by means of setting up building energy efficiency sign system and implementing effective encouraging policy.

Key words: Market mechanism; building energy conservation; Energy management contracting; Energy policy.

1. INTRODUCTION

For quite a long time, China government has laid particular stress on industry energy efficiency, yet not put enough emphasis on building energy efficiency[1]. As a result, building energy efficiency has fallen behind over a long period of time and has become the weakest link among the work of energy efficiency in China.

Building energy consumption in China has three characteristics, i.e. sustained rapid growth, significantly increase in rural areas and low building energy efficiency. Building energy consumption continue to grow for three factors: The first one is to speed up the process of urbanization. The second one is improvement of people's living level. With all kinds of household electrical appliances increased, the energy consumption of unit building area is growing faster[2]. The third factor is comfort requirements increase of the urban and rural residents.

At present, China generally implement energy-saving 50% of the standard. Even if implementing energy-saving 65% of the standard, the energy

consumption index is still 50% higher than the average level of the same climatic conditions in developed countries. Therefore, compared with western developed nations, building energy saving in China is backward seriously and much work has to be done in the area. One of effective means to improve building energy efficiency is to introduce energy management contracting mechanism into building sectors[3].

Based on energy management contracting mechanism and analysis of China building energy consumption situation and development trend, this paper probes into problems and countermeasures of building energy efficiency in China, which is of important theory value and realistic significance to promoting building energy efficiency in China. Firstly, this paper analyses both the problems in the work of China building energy efficiency and the features of building energy efficiency market, thus analyses major causes of building energy efficiency lagging behind. Secondly, this paper constructs theoretical model of building energy efficiency under market mechanism, and expounds its defining mechanism under market

mechanism, and function mechanism of mandatory policy, encouraging policy and technology advance theoretically. Thirdly, by combining the above theoretical study and particular practice of China, and using the experience of other countries for reference, this paper puts forward both the overall train of thought and implementation tactics of China building energy efficiency, and also puts forwards some policy suggestions on its implementation.

2. ANALYSIS OF ENERGY MANAGEMENT CONTRACTING MECHANISM

Energy Management Contracting (EMC) describes business ventures that use future energy savings to finance energy conservation measures. Energy service companies (ESCOs) engaged in EMC conduct an “investment-grade audit” of a facility to assess the potential for profitable energy-saving investments[4]. Facility owners sign energy performance contracts for various reasons, such as: Improving building control, decreasing occupant complaints, increasing employee productivity, reducing energy costs, repairing aging equipment, mitigating indoor-air-quality problems, paying for capital improvements with energy savings.

The most critical element in an EMC project’s success is developing a mutually beneficial contract for both the owner and the ESCO. Contract terms usually last five years or more, and call for charting energy consumption over a “base year”—one year of energy consumption before improvements are made. The difference between base-year consumption and post-project consumption defines the savings. The amount each party is compensated depends on measurement and verification of these energy savings [5]. The savings are intended to pay for the improvements and hopefully reward both the ESCO and facility owner with any excess.

EMC involves technical, financial, and legal expertise. Enlisting the right in-house staff together with outside contractors is essential to a successful EMC business venture. Large companies often undertake energy-efficiency projects using their own staff and financial resources. This in-house approach works even on a small scale when a technically oriented manager assembles the staff, contractors, and finances needed to design and manage the project. This approach also avoids a potential for hiring an ESCO that only chooses projects that assure high-return on their investment.

Retro-commissioning is a custom quality-assurance plan for existing buildings and is often an integral part of EMC. It is an excellent way to evaluate an energy-performance project. The advantage of beginning with retro-commissioning is that you can make the inexpensive improvements, reap the savings, and identify the best major investments before signing an energy-performance contract. Retro-commissioning commences with project design, continues through startup, and extends for some months after project completion.

In a typical Shared Savings Contract the ESCO provides the capital. Once completed, the customer has lower energy costs and will share an agreed portion of these savings with the ESCO over the term of the contract. The ESCO uses these savings to repay the loan from the financier. Typically the term of the contract and the loan will match, and the ESCOs share of the savings will exceed the loan repayment costs. Importantly, the financier is taking the risk that the ESCO may be unable to repay the loan; if the ESCO is a small or medium enterprise, the cost of credit may be quite high.

Although this contract arrangement can specify that the ESCO will guarantee, rather than share, the savings, the ESCO is likely to prefer to share the savings. From the ESCO’s perspective, sharing savings incentivizes the customer to minimize energy use and reduces the ESCO’s energy performance risk. As the ESCO is relieving the customer of the need to provide finance, the ESCO is likely to be able to dictate the use of shared savings. The ESCO typically retains ownership of the asset. Interaction of the EMC elements is depicted as Figure 1.

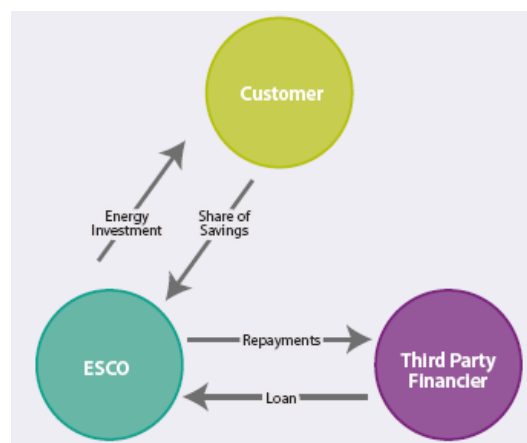


Figure 1 Interaction of the EMC elements

In some cases the energy savings may be valued based on prevailing energy prices, which means the ESCO also takes the energy price risk. These considerations generally mean the ESCO is a large enterprise with strong balance sheet and access to capital markets[6].

3. BUILDING ENERGY CONSERVATION COUNTERMEASURE BASED ON MARKET MECHANISM

Contract energy management mechanism was formally introduced to China in the early 1990s. At that time, the Chinese government and the World Bank experts had many discussions, and reach a consensus with demonstration and promotion of contract energy management mechanism in China. Under supporting of the State Economic and Trade Commission and the World Bank, China in 1997 set up three demonstration of energy service companies for customers to implement energy-saving projects. China Energy

Conservation Association statistics show that the current national energy services company has more than 200. However, energy service companies did not develop as rapid as expected, and the situation is very passive, which greatly hindered the development of the mechanism of the energy management contract in China. A concrete analysis of main obstacles and countermeasures impacting contract energy management mechanism goes as follows in order to provide theoretical references for this energy-saving management mechanism in China.

3.1 Analysis of obstacles

3.1.1 Financial barriers

The building owners in need of energy conservation lack of adequate funding, while energy service companies could provide abundant capital, equipment and technology to help businesses implement energy saving projects, which is the most critical factor for foreign energy service companies survive. However, the vast majority of our energy services company are formed with technical superiority, and tend to have good energy-saving technologies and energy-saving renovation experience while facing with financing problems. As a result, a lot of good energy-saving technologies and products can not find markets due to financing, and energy service companies fall into a very passive situation.

3.1.2. Policy barriers

Contract energy management requires strong policy support and preferential tax policies of comprehensive utilization of resources. Although energy service companies aim at profits, they bear a lot of the public nature conducive to the construction of resource-saving and environment-friendly society. The government may implement preferential policies such as tax relief for energy-saving service company, and reduce their operating costs and risk by improving and perfecting laws and regulations. Contract energy management is a market-based new energy-saving mechanism, but the majority of energy users, potential energy investors, tax authorities lack of full understanding of this.

3.1.3 Technical barriers

Providing new energy-saving products and technologies for those building owners in need of energy-saving is the core of energy management contracting mechanism. But research on a number of large buildings in Changsha show that although these building owners believe that energy service companies can provide them with technology and services with "zero investment", it is difficult for the two sides to reach a consensus on measure of the energy-saving effect and sharing of energy efficiency. This is mainly because there is not a scientific, objective measurement methods and models currently. In addition, many established energy service companies do not really have

the technology and capability of implementation of energy projects due to market access issues.

3.2 Countermeasures and suggestions

3.2.1 Improve the threshold for energy efficiency services to achieve diversification of energy-saving services companies

Many energy-saving services companies have been set up since the establishment of three exemplary energy service companies in China 1997. Because energy services industry in China is still in its infancy and has a relatively low threshold, energy service companies differs greatly in the technical level, the financial strength and scale, even with some false companies entering the field of energy efficiency services. For this reason, the credit level of energy service companies also differs greatly. In order to solve this problem as soon as possible, it is important to establish a rating system for energy service companies.

Those companies applying to join energy services industry should be strictly qualified from technical level, financing capacity, size, and other aspects to improve the energy services industry threshold. For those in operation a credit file has to be established. These companies should be real-time tracked with the effect of the energy-saving projects, profitability and ability to repay the loan. Those companies with technological backwardness, poor profitability and, low level repayment should be revoked their operating qualifications.

Besides, it is important to achieve diversification of the energy-saving service entries. In addition to the Government-set-up non-profit career Promotion type energy service companies, coexistence of multiple types of energy services companies is needed to prosper the market.

3.2.2 Intensify policy support and reduce the financial pressure of the energy services company

Aiming at the issue of single financing channel in contract energy management system, it is recommended that the government take measures to broaden the financing channels of energy service companies, to increase their financing capacity and ensure their ability to undertake large-scale energy-saving projects. Specifically, the following three ways could be considered.

First, the current use of special funds of the World Bank must be replaced with implementation of the loan guarantee mechanism. Second, the state and the provinces can arrange the corresponding energy-saving special funds to support energy efficiency energy-saving projects, and include energy-saving effect in the government evaluation system. Third, energy service companies should be given tax concessions. Energy service companies implementing energy-saving projects not only benefit economically energy service companies

and building owners, but also bring great social benefits.

3.2.3 Establish energy-saving effect evaluation system and improve the benefit allocation mechanism of energy-saving effect

The issue of measure and allocation of energy-saving effect is considered seriously by the building owners when they decide whether or not committing energy-saving projects to services companies. One of the direct cause for energy service companies to bear bad debt is that differences exist in the amount of energy savings and distribution of energy-savings benefits. Building owners often refused to pay in that energy saving does not reach the expected standard.

Although many metric methods are in use currently, there are many impact factors of energy-saving effect from a practical point of view. These factors will cause the predicted improvement effect far away from the actual situation. If you simply change recording instruments or systems of control energy consumption, a lot of changes in the state can not faithfully reflect and record. In addition, the effectiveness of the energy-saving size of the energy-saving projects often depends on the ability to continue with the process of enterprises in the energy service companies to implement energy saving projects. Therefore, many factors need to be considered with the measure of the energy-saving effect and energy efficiency allocation. Contract energy management mechanism thoroughly promotes the urgent need to establish a correct measure of energy-saving effect and reasonable benefits distribution system. This system should be able to consider various factors affecting the energy-saving effect, such as energy service companies and the enterprise's own characteristics, the characteristics of energy-saving projects and the conditions of operation.

4. CONCLUSIONS

Obstacles of contract energy management mechanism are inevitable in the process of implementation as a new energy-saving mode of operation. The key is how to combine practice, policies and market to identify the root of the problem.

On the basis of resource management theory, industry economy theory, sustainable development theory and micro-economy theory, and also based on the analysis of China building energy consumption situation and development trend, this paper probes into problems and countermeasures of building energy efficiency in China, which is of important theory value and realistic significance to promoting building energy efficiency in China. Firstly, this paper analyses both the problems in the work of China building energy efficiency and the features of building energy efficiency market, thus analyses major causes of building energy efficiency lagging behind. Secondly, this paper constructs theoretical model of building energy

efficiency under market mechanism, and expounds its defining mechanism under market mechanism, and function mechanism of mandatory policy, encouraging policy and technology advance theoretically. Thirdly, by combining the above theoretical study and particular practice of China, and using the experience of other countries for reference, this paper puts forward both the overall train of thought and implementation tactics of China building energy efficiency, and also puts forwards some policy suggestions on its implementation.

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LONG TERM ENERGY DEMAND PROJECTION OF SAMUT SONGKHRAM PROVINCE -A TOURISM TOWN IN THAILAND

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ABSTRACT

Samut Songkhram is an ancient province with a long history, located at the mouth of the Mae Klong river to the Gulf of Thailand. It is a principal city with the large total tourism economic output in Thailand. Owing to Samut Songkhram as a tourism town, it leads to a significant change of energy demand structure. This paper presents an energy demand analysis and projection with mathematical model for energy plan in the future. The analysis provided is underlying the actual energy data during the past 10 years in three economic sectors namely residential, commercial and transport. Through this paper, long term energy demand projections were studied with emphasis on future growth. Multiple Linear Regression method was applied to obtain energy demand prediction. The results show that the growth of population, Gross Provincial Product (GPP) and fuel prices are socio-economic indicators that influence future energy demand. Transport sector is the largest amount of energy consumption, following by commercial and residential sector. Diesel oil is used as the major energy in transport sector while commercial sector mainly uses diesel oil for fishery and electricity for electrical appliances. Residential sector consumes both of electricity and LPG, which are increased continuously. The results provide scientific basis for the planned development of energy demand and supply balancing.

Key words: Energy Demand Projection; Samut Songkhram Province; Tourism Town.

1. INTRODUCTION

Thailand has limited domestic oil production and reserves, and imports are a significant portion of the country's oil consumption. Thailand holds large proven reserves of natural gas, and natural gas production has increased substantially over the last few years. However, the country still remains dependent on imports of natural gas to meet growing domestic demand for the fuel. The primary energy consumption is mostly from fossil fuels, accounting for over 80 percent of the country's total energy consumption. Oil was 39 percent of total energy consumption in 2010. As the economy expanded and industrialized, Thailand consumed more oil for transportation and industrial uses. Natural gas has replaced some oil demand and is the next largest fuel, growing to nearly a third of total consumption mix [1].

This research aims to study trend of energy demand of a tourism town and to predict energy demand in the future using energy statistics to analyze the structure of energy consumption including the variables in affected to energy consumption trends such as population, Gross Province Product (GPP) and fuel prices. A mathematical model using simple linear

regression and multiple linear regression are used for forecasting processes. Samut Songkhram province is demonstrated for this study. The results obtained will be applied for a community energy planning in this province and as a guide line for other provinces.

2. METHODOLOGY

2.1. HISTORICAL OF ENERGY DATA

The annual energy data, spanning from 2001 to 2009 used in this study, have been collected from various issues of Energy Statistical Yearbook to analyze energy demand characteristics, energy consumption structure and trends of energy demand.

2.2. Survey data

In this study, we also conducted a survey for gathering some energy data based on the cluster sampling method which the sampling size was derived from Taro Yamane equation as following [2].

$$n = \frac{N}{1 + Ne^2} \quad (1)$$

NOMENCLATURE			
Symbols		Symbols	
n	Sampling number	e_{ij}	Actual energy demand on each sector per year (toe/yr)
N	Statistic population	a_{i0}, a_{i1}, a_{i2}	Coefficient of energy demand in mathematic model
e	Allowable error ratio	Pop_i	Number of population forecast (persons)
p	Percentage (%)	GPP_i	Gross province product forecast (millions of baht)
f	Frequency of number	$Diesel_j$	Diesel price forecast (baht/litre)
N_f	Total number	Pop_{aj}	Actual number population (persons)
E_{Res}	Energy consumption of residential sector (toe/yr)	GPP_{aj}	Actual gross province product (millions of baht)
E_{Resi}	Energy consumption on each type in residential sector (toe/yr)	$Diesel_{aj}$	Actual diesel price (baht/litre)
E_{Com}	Energy consumption of commercial & industrial sector (toe/yr)	F_{ij}	Factor forecast
E_{Comi}	Energy consumption on each type in commercial & industrial sector (toe/yr)	b_{i0}, b_{i1}	Coefficient of factor forecast
E_{Tran}	Energy consumption of transport sector (toe/yr)	N_i	Number of year for factor forecast start in 2001 = 1
E_{Trani}	Energy consumption on each type in transport sector (toe/yr)	A_{ij}	Actual data of factor
E_{Total}	Total energy consumption (toe/yr)	n_a	Total number of actual data (year)
E_{ij}	Energy demand forecast on each sector per year (toe/yr)		

2.3. Energy consumption flowchart

The data obtained from the statistic reports and survey were used to formulate energy consumption flowchart which was classified to three economic sectors namely residential, commercial and industry, and transport based on the principle of the Long-range Energy Alternatives Planning Model (LEAP Model) [3] for specified proportion of energy use.

2.4. Mathematic model of energy consumption

Mathematical model was applied by multiple linear regression. The variables affected to energy use in each category were calculated to determine the coefficients of the equations in relation to population, GPP and diesel price. For energy demand projection, forecasting of population, GPP and diesel price were applied by the simple regression equation [4] using the time (years) as independent variables. Then the predictive variables was used to calculated energy consumption by sectors. Finally, The total energy consumption of the province was predicted in the period of 2010 - 2030.

3. RESULTS AND DISCUSSIONS

3.1. Background of Samut Songkhram

Samut Songkhram is an ancient province with a long history, located at the mouth of the Mae Klong river to the gulf of Thailand with a geographical area of 416.7 square kilometers. It is a principal city with the large total tourism economic output in Thailand. Basic information of Samut Songkhram province is shown in

Table 1. Population statistics and the GPP [6] of Samut Songkhram is shown in Table 2.

Table 1 Basic information of Samut Songkhram

Total Population	211,481 persons
No. Household	53,083 households
Geology	13.42° N , 100° E
Area	416.7 km ²
Adjacent Area	North -Ratchaburi , Samutsakhon South -Gulf of Thailand , Phetchaburi East -Samutsakhon West -Ratchaburi , Phetchaburi
Administration Division	3 districts, 36 communes, 284 villages

Source : Provincial Operation Center [5], Office of the National Economic and Social Development Board [6]

Although population growth rate was slightly increased but the amount of households were increased as well because of the separate families [7]. Growth of GPP were increased steadily.

Table 2 Statistics of population and GPP

Year	Population (Persons)	GPP at Current Price (Million Baht)	GPP at Constant 1988 Price (Million Baht)
		(Million Baht)	(Million Baht)
2001	209,061	9,856	5,566
2002	209,444	9,700	5,543
2003	209,543	10,219	5,764
2004	209,407	11,199	6,065
2005	209,160	13,172	7,086
2006	209,449	14,658	7,286
2007	209,985	15,433	7,455
2008	210,672	16,841	7,598

Source : Office of the National Economic and Social Development Board [6]

3.2 Energy consumption

Energy consumption statistics during the period of 2001 - 2009 are shown in Table 3. Petroleum product is the biggest proportion consumed by transportation and fisheries, followed by charcoal, firewood, LPG, NGV and fuel oil [8], [9].

3.3 Energy consumption situation

A sampling survey was conducted in the three sectors namely residential, commercial & industry and transport. The survey data collection was used to formulate a flowchart based on LEAP Model principle as shown in table 4. The calculation of the total energy consumption of the economic sectors were analyzed and converted to the calorific value in term of ton of oil equivalent (toe). The proportion of energy consumption were used as the forecasting factors in the further study.

3.3.1. Energy consumption in residential sector

In residential sector, the household users were classified into two groups namely the municipal and rural area. The major energy consumption of households in the municipality were electricity and LPG. Electricity was used for electrical appliances especially in air conditioning unit. LPG was the main

energy for cooking while some household was still used fuel wood and charcoal.

3.3.2. Energy consumption in commercial & industry sector

The energy consumption in commercial & industry were classified into 9 groups namely agriculture, fishery, construction, goods and service, food & restaurant, government official, education, hospital and hotel and rental. The survey result was found that the most energy consumption was diesel oil which was 85,677.36 toe per year [10], and the rest used for water pumping in agriculture and small industry. Electricity was mainly used for electrical appliances and LPG for cooking and heating process.

3.3.3. Energy consumption in transport sector

The transportation sector was classified to road way, water way, and railway. The energy consumption for road way was consumed by small trucks accounted for 71,189.35 toe per year, heavy trucks of 47,598.47 toe per year and 28,712.96 toe per year of gasoline for a large amount of motorcycle. LPG and natural gas was consumed by both private and public vehicles tremendously from year 2008 effected from an increase of fuel price. The water way transportation was more consumption because the province areas were surrounded by many canals, river and sea.

Table 3 Energy consumption data on the year 2001 – 2009

Energy Consumption (ktoe/yr)	Year								
	2001	2002	2003	2004	2005	2006	2007	2008	2009
Electricity	15.79	15.72	16.02	17.46	18.97	19.78	20.66	20.97	22.42
Diesel B2	143.22	156.65	168.74	181.23	170.17	157.34	158.26	113.66	81.86
Diesel B5						1.06	8.42	38.82	69.45
Coast Diesel	62.58	71.97	78.22	82.92	81.35	78.22	68.84	65.71	84.62
Gasoline E20								0.10	0.59
Gasoline 91	22.94	24.67	26.08	25.52	24.41	25.39	25.55	20.50	17.86
Gasoline 95	15.01	16.23	16.77	15.24	10.99	7.64	5.23	1.08	0.31
Gasoline 91 (E10)						0.20	1.29	5.52	7.86
Gasoline 95 (E10)				0.01	2.92	6.09	8.09	12.39	14.69
LPG	13.52	13.44	13.9	14.9	16.24	17.63	16.81	19.35	20.00
NGV							0.20	0.77	1.23
Fuel Oil	0.96	1.00	1.42	1.78	1.37	1.70	2.40	1.26	0.79
Charcoal & Fuel Wood	0.31	0.29	0.22	1.00	1.02	0.79	0.82	0.42	0.61

Source : Department of Energy Business: Ministry of Energy [8], PTT Public Company Limited [9]

Travelling by motor boat along the Mae Klong river connected to many canals around the province was well known because of commercial and tourism promotion. Energy consumption of different types of boats which could be divided into three types, personal 5-horsepower motor boat consumed gasoline, medium-sized boat for tourists and ferry boat consumed diesel. The total energy consumption of the water way

transport was 696.51 toe per year of gasoline and 250.94 toe per year of diesel oil. Energy consumption of railway transportation was diesel oil for daily travelling with 30 kilometers of 8 trips per day and total fuel consumption was 126.85 toe per year.

Table 4 LEAP Model for energy consumption of Samut Songkhram province

Sector	Energy	unit/year	toe/year
Residential	Electricity	90,381 MWh	7,701
	LPG	6,510 ton	7,597
	Charcoal	380 ton	260
	Fuel Wood	218 ton	83
Commercial & Industrial	Electricity	170,942 MWh	14,566
	Diesel	16,893 kilolitre	14,561
	LPG	5,975 ton	6,973
	Charcoal	244 ton	167
	Coast Diesel	99,396 kilolitre	85,677
	Fuel Oil	1,507 kilolitre	1,419
	Gasoline	55,953 kilolitre	41,689
Transport	Diesel	159,802 kilolitre	137,746
	LPG	4,790 ton	5,590
	NGV	2,465 ton	1,740

3.4. The proportion of energy consumption

Survey energy consumption data was used to calculate the proportion of energy consumption by type of energy using the equation as Eq.2.

$$p = \frac{f}{N_f} \times 100 \quad (2)$$

The percentage derived from the survey data was divided into the proportion of energy consumption in the past to classify energy consumption by type of energy.

3.5 Mathematical modeling

The variables used for mathematical model of energy consumption were population, GPP and diesel oil price. The regression equation application was based on the multiple linear regression method. Mathematical model for the residential, commercial & industry and transport sectors were shown as the Eq.3 and Eq.5 respectively. Coefficients of the equation modeling in residential sector was derived in equation Eq.4.

$$E_{ij} = a_{i0} + a_{i1}Pop_j + a_{i2}GPP_j \quad (3)$$

$$\begin{bmatrix} n_a & \sum_{j=1}^n Pop_{aj} & \sum_{j=1}^n GPP_{aj} \\ \sum_{j=1}^n Pop_{aj} & \sum_{j=1}^n Pop_{aj}^2 & \sum_{j=1}^n Pop_{aj}GPP_{aj} \\ \sum_{j=1}^n GPP_{aj} & \sum_{j=1}^n Pop_{aj}GPP_{aj} & \sum_{j=1}^n GPP_{aj}^2 \end{bmatrix} \begin{bmatrix} a_{i0} \\ a_{i1} \\ a_{i2} \end{bmatrix} = \begin{bmatrix} \sum_{j=1}^n e_{ij} \\ \sum_{j=1}^n Pop_{aj}e_{ij} \\ \sum_{j=1}^n GPP_{aj}e_{ij} \end{bmatrix}$$

$$E_{ij} = a_{i0} + a_{i1}Pop_j + a_{i2}GPP_j + a_{i3}Diesel_j \quad (4)$$

$$\begin{bmatrix} n_a & \sum_{j=1}^n Pop_{aj} & \sum_{j=1}^n GPP_{aj} & \sum_{j=1}^n Diesel_{aj} \\ \sum_{j=1}^n Pop_{aj} & \sum_{j=1}^n Pop_{aj}^2 & \sum_{j=1}^n Pop_{aj}GPP_{aj} & \sum_{j=1}^n Pop_{aj}Diesel_{aj} \\ \sum_{j=1}^n GPP_{aj} & \sum_{j=1}^n Pop_{aj}GPP_{aj} & \sum_{j=1}^n GPP_{aj}^2 & \sum_{j=1}^n GPP_{aj}Diesel_{aj} \\ \sum_{j=1}^n Diesel_{aj} & \sum_{j=1}^n Pop_{aj}Diesel_{aj} & \sum_{j=1}^n GPP_{aj}Diesel_{aj} & \sum_{j=1}^n Diesel_{aj}^2 \end{bmatrix} \begin{bmatrix} a_{i0} \\ a_{i1} \\ a_{i2} \\ a_{i3} \end{bmatrix} = \begin{bmatrix} \sum_{j=1}^n e_{ij} \\ \sum_{j=1}^n Pop_{aj}e_{ij} \\ \sum_{j=1}^n GPP_{aj}e_{ij} \\ \sum_{j=1}^n Diesel_{aj}e_{ij} \end{bmatrix} \quad (5)$$

Coefficients of the equation modeling in transport, commercial & industry were derived in equation Eq.6.

$$\begin{bmatrix} n_a & \sum_{j=1}^n Pop_{aj} & \sum_{j=1}^n GPP_{aj} & \sum_{j=1}^n Diesel_{aj} \\ \sum_{j=1}^n Pop_{aj} & \sum_{j=1}^n Pop_{aj}^2 & \sum_{j=1}^n Pop_{aj}GPP_{aj} & \sum_{j=1}^n Pop_{aj}Diesel_{aj} \\ \sum_{j=1}^n GPP_{aj} & \sum_{j=1}^n Pop_{aj}GPP_{aj} & \sum_{j=1}^n GPP_{aj}^2 & \sum_{j=1}^n GPP_{aj}Diesel_{aj} \\ \sum_{j=1}^n Diesel_{aj} & \sum_{j=1}^n Pop_{aj}Diesel_{aj} & \sum_{j=1}^n GPP_{aj}Diesel_{aj} & \sum_{j=1}^n Diesel_{aj}^2 \end{bmatrix} \begin{bmatrix} a_{i0} \\ a_{i1} \\ a_{i2} \\ a_{i3} \end{bmatrix} = \begin{bmatrix} \sum_{j=1}^n e_{ij} \\ \sum_{j=1}^n Pop_{aj}e_{ij} \\ \sum_{j=1}^n GPP_{aj}e_{ij} \\ \sum_{j=1}^n Diesel_{aj}e_{ij} \end{bmatrix} \quad (6)$$

Amount of energy used in the calculation was illustrated in equation Eq.7 to Eq.9. Total energy consumption was calculated from the equation Eq.10.

$$E_{Res} = \sum_{i=1}^n E_{Resi} \quad (7)$$

$$E_{Com} = \sum_{i=1}^n E_{Comi} \quad (8)$$

$$E_{Tran} = \sum_{i=1}^n E_{Trani} \quad (9)$$

$$E_{Total} = E_{Res} + E_{Com} + E_{Tran} \quad (10)$$

For energy demand forecasting, the equation Eq.11 was used for calculation of variables which was affected energy consumption. Coefficients of the equation parameters are calculated from equation Eq.12 and Eq.13.

$$F_{ij} = b_{i0} + b_{i1}N_j \quad (11)$$

$$b_{i0} = \frac{(\sum_{j=1}^n A_j)(\sum_{j=1}^n N_j^2) - (\sum_{j=1}^n N_j A_j)(\sum_{j=1}^n N_j)}{n(\sum_{j=1}^n N_j^2) - (\sum_{j=1}^n N_j)^2} \quad (12)$$

$$b_{i1} = \frac{n(\sum_{j=1}^n N_j A_j) - (\sum_{j=1}^n N_j)(\sum_{j=1}^n A_j)}{n(\sum_{j=1}^n N_j^2) - (\sum_{j=1}^n N_j)^2} \quad (13)$$

Comparison of energy consumption by mathematical model and actual data were illustrated in Figure 2.

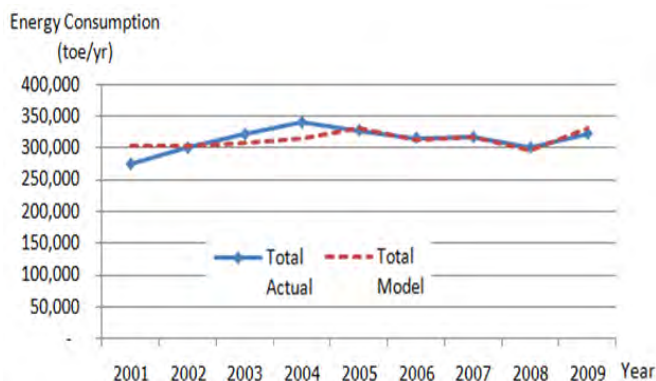


Figure 2 Comparison of actual values and model values for total energy consumption

3.6 Energy demand projection for the future

Energy demand projection obtained from the predictive value of variables for forecasting model were population, GPP and price of diesel oil as shown in Table 5. The results of energy demand projection from year 2010 to year 2030 were shown in Table 6.

The results of variables forecasting showed that population was increased from 210,991 to 215,754 persons or about 2.25%, GPP values increased from 8324.6 to 14,884.9 million Baht or 78.81% and price of diesel oil increased from 31.37 to 76.06 Baht

per liter accounted for 142.46%. The predicted variables were used to forecast energy demand. Model outputs illustrated the total energy demand during the year of 2010 to 2030 accounted for 322,302 toe per year to 359,387 toe per year or increased about 11.51%. Transport sector dominated the use of energy followed by commercial & industry and residential sector respectively.

The energy consumption of residential sector came mainly from the use of electrical energy which was increased from 7,948 toe per year to 14,142 toe per year, followed by LPG from 8,005 toe per year to 12,748 toe per year while charcoal and firewood were slightly increased. The main energy consumption for commercial & industry sector was diesel for fishery. Diesel for fishery boats were increased from 77,621 toe per year to 88,756 toe per year, followed by electricity from 15,086 toe per year to 26,788 toe per year, diesel for other was gradually slow down from 15,173 toe per year to 13,953 toe per year and LPG was increased from 7,283 toe per year to 11,733 toe per year. For diesel in transport sector, it was decreased from 142,757 toe per year to 130,948 toe per year while gasoline consumption was slightly changed from 40,116 toe per year to 40,087 toe per year. Natural gas was increased from 836 toe per year to 3,458 toe per year due to alternative energy implementation and pricing policy by government.

Table 5 Prediction of variables for energy demand forecasting model on the years 2010-2030

Variables	2010	2015	Year 2020	2025	2030
Population (persons)	210,991	212,182	213,373	214,802	215,754
GPP (millions of baht)	8,324.6	9,964.7	11,604.7	13,244.8	14,884.9
Diesel Price (baht / liter)	31.37	42.55	53.72	64.89	76.06

Table 6 Energy demand forecasting of Samut Songkhram province on the years 2010-2030

Energy Demand Forecasting	2010	2015	Year 2020	2025	2030
Energy Demand (toe / year)	322,302	331,573	340,844	350,115	359,387
Residential sector					
Electricity	7,948	9,497	11,045	12,593	14,142
LPG	8,005	9,191	10,376	11,562	12,748
Charcoal	401	495	589	683	777
Fuel Wood	126	155	185	215	245
Commercial & Industrial sector					
Electricity	15,086	18,011	20,937	23,862	26,788
Coast Diesel	77,621	80,405	83,189	85,972	88,756
Diesel	15,173	14,868	14,563	14,258	13,953
LPG	7,283	8,396	9,508	10,621	11,733
Fuel Oil	1,539	1,625	1,710	1,795	1,881
Charcoal	257	316	375	435	494
Transportation Sector					
Diesel	142,757	139,805	136,852	133,900	130,948
Gasoline	40,116	40,109	40,101	40,094	40,087
LPG	5,155	7,210	9,266	11,322	13,378
NGV	836	1,492	2,147	2,803	3,458

4. CONCLUSION

Samut Songkhram province is a small town and has been promoted to a tourism place. Energy demand projection is essential for energy planning in the future. Trend analysis of the variables for forecasting model were population which showed a slightly increase, GPP and diesel price were impacted on demand of energy in the future. The predictive model showed that trend of total energy demand was constantly increased. In year 2030, the total energy demand was about 359,387 toe per year, transport sector consumed about 52.28% of the total energy consumption, followed by commercial & industrial sector of 39.96% and residential sector of 7.77%. Diesel oil and gasoline demand were decreased due to NGV and LPG as an alternative energy replacement particularly for transport sector. For commercial & industrial sector, energy demand was about 88,756 toe per year for diesel oil and 26,788 toe per year of electricity for electrical appliances in the businesses area. For residential sector demand was increased about 14,142 toe per year for electricity and about 12,748 toe per year for LPG. Mathematical model for energy demand forecasting showed a predictable amount of future energy demand in the community. The results provided scientific basis for the planned development of energy demand and supply balancing for the local energy planning.

ACKNOWLEDGEMENTS

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ELECTRICITY CONSUMPTION IN THE COMMERCIAL KITCHEN

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ABSTRACT

Academic and industrial literature concerning the energy consumption of commercial kitchens is scarce. Electricity consumption data were collected from distribution board current transformers in a sample of fourteen UK public house-restaurants. This was set up to identify patterns of appliance use as well as to assess the total energy consumption of these establishments. The electricity consumption in the selected commercial kitchens was significantly higher than current literature estimates. On average, 63% of the premises electricity consumption was attributed to the catering activity. Key appliances that contributed to the samples average electricity consumption were identified as refrigeration (70 kWh, 41%), fryers (11 kWh, 13%), combi-ovens (35 kWh, 12%) bain maries (27 kWh, 9%) and grills (37 kWh, 12%). Behavioral factors and poor maintenance were identified as major contributors to excessive electricity usage with potential savings of 70% and 45% respectively. Initiatives are required to influence operator behavior, such as the expansion of mandatory energy labeling, improved feedback information and the use of behavior change campaigns. Strict maintenance protocols and more appropriate sizing of refrigeration would be of great benefit to energy reduction.

Keywords: Electricity Use, Commercial Kitchen, Public House, Energy Conservation

1. INTRODUCTION

Commercial kitchens are some of the most profligate users of gas, water and electricity in the UK. As a result, they can leave a large carbon footprint, often using ten times the energy of the average commercial building (The Chartered Institution of Building Services Engineers (CIBSE) 2012; S. Mudie 2013). Catering businesses feature in virtually every town and city on a global scale and are vital buildings to be considered in any low carbon town. The UK is committed to an 80% reduction in greenhouse gas emissions by 2050, compared with 1990 levels (HM Government Department for Energy and Climate Change (UK) 2011).

The UK Carbon Trust and the Chartered Institute for Building Services (CIBSE) estimate that the total energy consumption of Britain's catering industry is in excess of 21,600 million kWh per year. CIBSE and the UK Carbon Trust estimate that 50% of this consumption originates from non-commercial catering businesses (hospitals, ministry of defense, schools etc.), with between 30-50% originating from activities of commercial kitchens (restaurants, public houses, cafes etc.) (Carbon Trust 2008; CIBSE 2009). However, these estimations are based on consumption in the USA. Despite their excessive usage, very little industrial or academic research exists pertaining to the current consumption, and energy reduction strategies of commercial kitchens (Carbon Trust 2011). There has

been substantial research relating to energy use and reduction in the building fabric and envelope which may be applied to these premises. However, beyond the procurement of more energy efficient cooking appliances, very little innovation has been achieved. Indeed, the design of cooking technologies have remained virtually stagnant since their creation (Newborough. 1987). Hot and cold meals are being prepared increasingly in public houses due to the increasing number of "off licenses" and shops selling alcohol and the tax levied on alcohol sales (Newborough. M and Probert. S. D 1988; Probert 1996; The Independent 2010). This has resulted in very few practical differences in the operation, licensing and management between restaurants and public houses ("pubs") in the UK, giving rise to the colloquial phrase "gastro-pub".

In order to fill the vast gap in the lack of published knowledge concerning commercial kitchens, the objectives of this study are to provide a comprehensive analysis of the electricity use of commercial catering establishments with a specific focus on the food preparation. The reduction of energy use from food preparation and cooking is seen as the largest challenge for these businesses if they are to continue to reduce consumption.

Consumption patterns of key appliances are explored and preliminary energy reduction strategies are suggested.

2. MATERIALS AND METHODS

2.1 Description of the Sample

Fourteen sites in England were selected to take part in this study. The relatively small sample size reflects the practical and financial constraints of extensive monitoring of such appliances; the type of intensive analysis makes it difficult to target a large sample size (D. Silverman 2011) e.g. 400 individual appliances were monitored in this study.

Each site belongs to a large chain of UK gastro-pubs which comprises 191 premises, all offering the same food menu. The fourteen sites were selected due to their “typical” range of appliances. Typical food items cooked include burgers, pies, sausages, hot sandwiches and casseroles. The equipment contained within the study kitchens is surprisingly varied given that they are part of a large chain. While there is a general template set-up for pubs, the layouts, makes, models and capacities and volumes of the appliances differ greatly.

2.2 Monitoring of Electricity Consumption

Electricity supply was monitored using a system produced by a company - NoWatt Limited. The system consists of many purpose-built transformers which were fitted to breakers in the distribution boards of the incoming electricity supplies (figure 1). Included in this were sensors. The sensors take sub-cycle readings at a rate of 12,800 times a second, enabling the differentiation of devices on a single circuit breaker by an advanced disaggregating software.

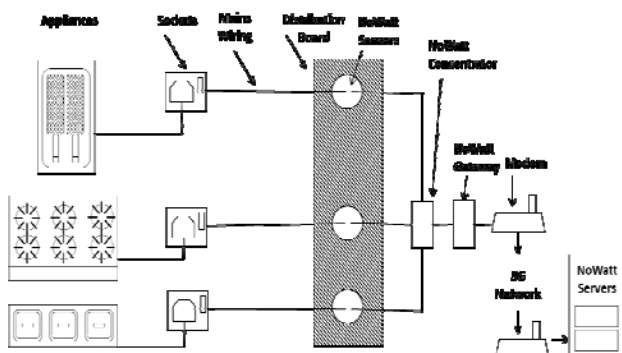


Figure 1 Diagram of appliance monitoring system

The sensors are connected to concentrators which send the high resolution readings (in real-time) to a signal gateway which then transmit the information to a central database via a Global System for Mobile Communications (GSM) network (3G). The raw data is stored in a standard SQL database (MySQL). There is no local display (or other features) on the sensors/concentrator; all information is retrieved from the central database (over the Internet). The advantages of such a system are that no additional wiring is required to commence monitoring, the system is relatively visually and practically unobtrusive and the

data can be accessed in real-time in virtually unlimited resolution.

To ensure that this study was not subject to the Hawthorne effect (i.e. people who are aware they are being observed tend to alter their behavior (G. Wood and Newborough 2003)) kitchen operating staff were not informed of the monitoring.

2.3 Data Analysis

Whole building electricity and gas consumption was monitored for one year at one site to represent a case study of the gastro-pubs annual energy usage. This takes account of seasonal patterns of appliance use (such as heating and air conditioning) and consumer behavior (such as increased “eating out” in the holiday seasons). This information is analyzed by business-area and then by appliance end-use as described in Tables 1 and 2.

In all subsequent analysis, electricity use data has been gathered exclusively from all cooking and food storage appliances, omitting any kitchen ventilation, warewashing or gas consuming appliances. In these kitchens, gas is only consumed by a large chargrill (charbroiler) and a gas fired oven with 6 hobs/burners.

For a daily analysis of the kitchens’ electricity expenditure, consumption was studied in the fourteen monitored kitchens over the period of one week (November 10th to November 16th 2012) as this represents the consumer cycle (more meals were served on Saturdays and Sundays). This is then averaged to form average daily consumption. It must be recognized that data pertaining to the week of study may be subject to seasonal and occupancy variations e.g. school holidays, unusual weather events etc. However, measurements were taken within experimental error to limit these effects. The period of study represents a “typical” weekly consumer cycle that does not fall during a holiday season.

To identify patterns of catering appliance usage, consumption was attributed to broad categories of an appliance e.g. “Fridge” or “Fryer”. Even within the same chain of businesses, the kitchens contain different volumes, makes and models of appliances each with varying capacities, conditions and locations within the kitchens. It is particularly challenging to compare “like for like” appliances. Additionally, the kitchens vary in the volume of meals cooked. Therefore the consumption of each kitchen has been taken and averaged per day to indicate the average usage which should be treated with caution.

Data for each site and appliance were analyzed with Excel spreadsheets, to calculate the key values of consumption.

Table 1 Summary of Business (Physical) Area

Area	Description
Kitchen	Air handling, kitchen lighting, cooking equipment, refrigeration, hot storage, warewashing.
Restaurant	Air handling, restaurant, lighting, central heating, sundry sockets (vacuum etc.)
Cellar	Cellar cooling, beverage cooling, beverage pumps.
Bar	Glass washing, bar lighting, bottle fridges, ice machine, coffee machine.

Table 2 Summary of End Functions

Area	Description
Warewashing	Dish and glass washers.
Freezers and Fridges	All walk-in, stand alone and under counter refrigeration and freezing units, ice machines and cellar beverage chillers.
Cooking	All appliances utilized to prepare meals and store hot food, coffee machine.
Lighting	All external and internal lighting.
Air Handling	All air handling including ventilation, central heating, air conditioning and cellar cooling.
Other	Sundry sockets used ad-hoc (vacuum etc.), building alarms, hand driers, beverage pump, and office computer.

3. RESULTS

3.0 Whole Building Energy Use

Figures 2 and 3 displays the annual profile of energy consumption for the case study site (including gas). The results should be considered within caution due to the singular sample case-study. Considering the business areas, figure 2 clearly demonstrates that the kitchen is by far the largest consumer of energy. In terms of end function, energy consumed in the cooking of food, followed by the storage of food represent the largest expenses; almost 75% of the buildings annual consumption.

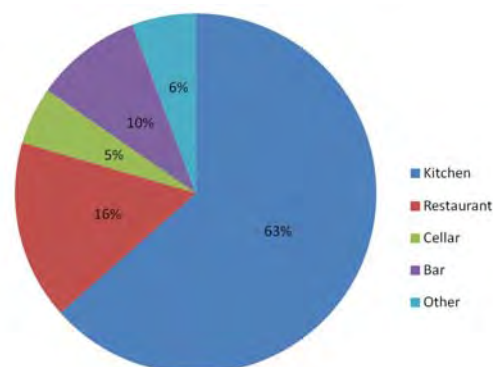


Figure 1 Monitored Energy Consumption by Business (Physical) Area

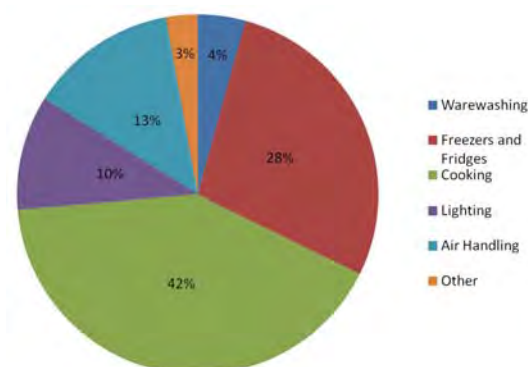


Figure 2 Monitored Energy Consumption by End Function

Annual consumption figures by area and function differ significantly from the previous estimates in published literature (23% and 6% used for cooking and refrigerated food storage respectively) [3]. The consumption in the case study site represented a total expenditure of £37,872 (582,694 kWh); the second largest expenditure to the business after labor costs.

3.1. Kitchen Electricity Consumption

The average daily electricity consumption for the 14 kitchens is illustrated in figure 4, together with the average number of meals served. This clearly demonstrates that consumption is only marginally affected by trade volume. As expected, there is increased consumption over the weekend. However, given a 152% reduction in meal output on a Monday compared with a Saturday (figure 4), consumption only decreases by 20%. This is largely due to operator behavior. Between the sites, there was significant variation in the kWh per meal ratio; 1.52 kWh/meal per week efficiency at Kitchen 3, compared with 3.32 kWh/meal per week at Kitchen 7.

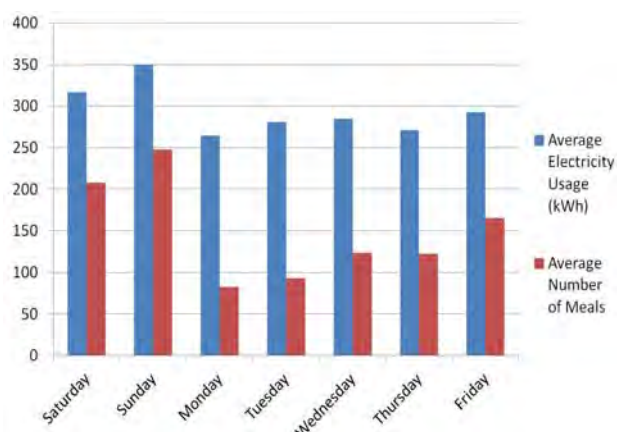


Figure 3 Kitchen Monitored Electricity Consumption and Number of Meals Served

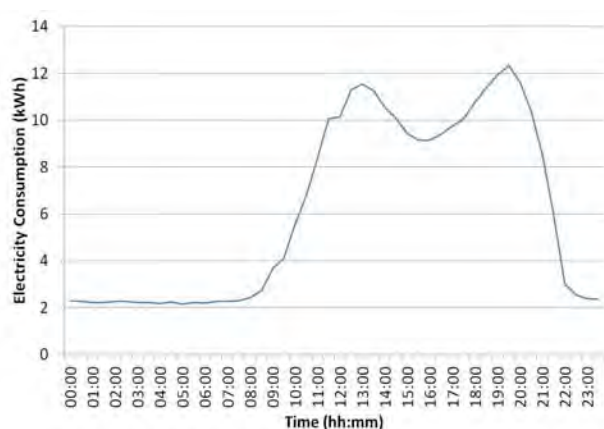


Figure 4 Average 24 Hour Monitored Electricity Consumption (kWh)

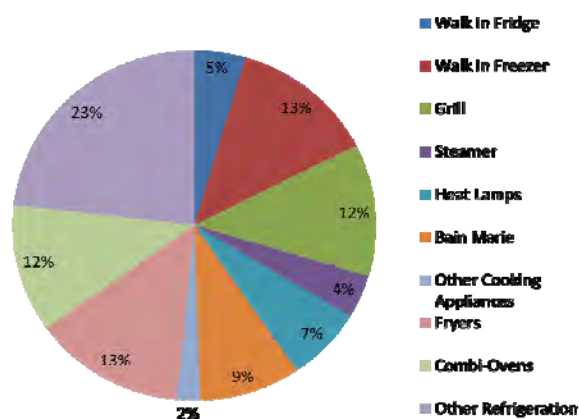


Figure 5 Average Monitored Kitchen Electricity Use Composition (%)

The 24 hour profile of kitchen electricity usage has been averaged across all days of measurement and all kitchens (figure 5). This demonstrates that the kitchens rarely reduce the consumption of appliances during the mid-afternoon lull in cooking volume, but should be interpreted with caution due to the small sample size and for the reasons discussed previously. The average daily usage for the sites was 294.4 kWh.

Figure 5 also indicates that base load consumption (between 00:00-05:00) is 9% of the total load.

Figure 6 allocates the kitchens' electricity consumption into the main broad appliance categories, averaged across all days of measurement and all kitchens. The total average refrigeration consumption is approximately 41%, with the remaining 59% attributed to cooking appliances. The variation in appliance electricity consumption can be viewed in more detail in figure 7 and table 3.

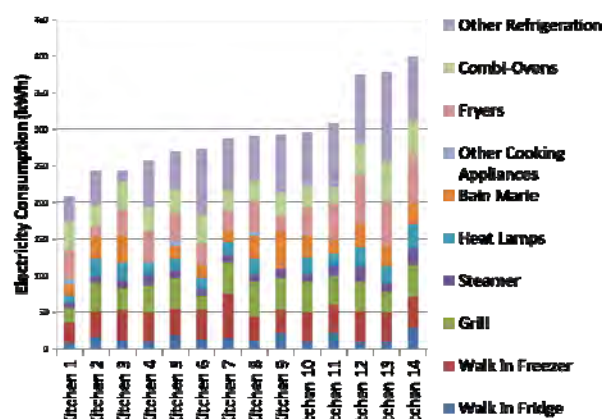


Figure 6 Average Monitored Kitchen Consumption Composition (kWh)

Table 3 Relevant Statistics of the Average Kitchen

Appliance Category	Ave. total daily (kWh)	Relative Standard Deviation (%)	Ave. Daily Carbon (kg CO ₂)	Ave. Daily Cost (£)
Walk in Fridge	13.81	45.89	7.14	1.52
Walk in Freezer	39.17	23.25	20.25	4.31
Grill	36.89	27.89	19.07	4.06
Steamer	11.99	47.35	6.20	1.32
Heat Lamps	20.70	35.42	10.70	2.28
Bain Marie	27.19	44.49	14.06	2.99
Other cooking appliances	6.08	37.35	3.14	0.67
Fryers	40.82	43.54	21.11	4.49
Combi Ovens	35.71	34.16	18.46	3.93
Other Refrigeration	70.13	38.86	36.26	7.71
Total kitchen consumption	294.37	22.99	152.19	32.38

3.2. Key Appliances

3.2.1. Refrigeration

Refrigeration represents the largest consumption in each of the study kitchens. In addition to one large walk-in fridge and walk-in freezer unit at each site, the

average number of additional refrigerating appliances was 9 units; 6 fridges and 3 freezers per site. Overall, the conditions of the refrigerating appliances were poor. Ad-hoc observations indicated that approximately 50% of the units (including walk-ins) had some kind of fault or developing problem.

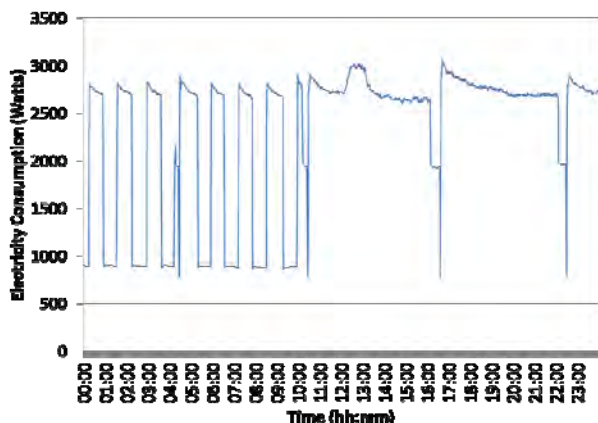


Figure 7 Walk-in Freezer Monitored Electricity Consumption (Watts) at Kitchen 12 on 10th November

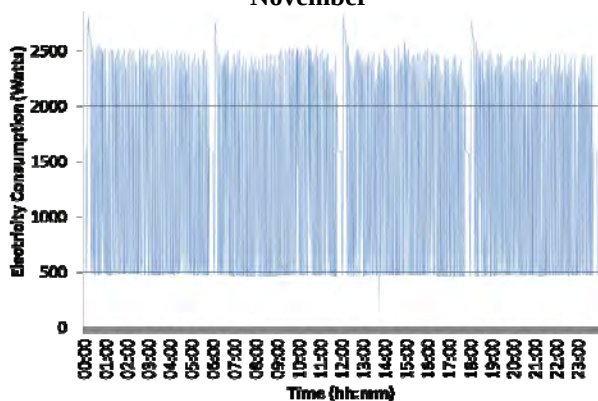


Figure 9 Walk-in Freezer Monitored Electricity Consumption (Watts) at Kitchen 12 on 16th November

While all other appliances in the study group remained in their respective states of repair throughout the monitoring period, maintenance was conducted on the Walk-in Freezer at Kitchen 12 during the week of study. Comparing the pre-maintenance daily consumption (figure 8) with the post-maintenance usage (figure 9), this represented a 45% electricity reduction. A common fault observed was that over 50% of units displayed incorrect temperatures; the digitally displayed temperature is not always the actual monitored temperature of the refrigerator and this will correspond to electricity wastage (and has implications for food safety). Appropriate maintenance is vital to energy reduction in these premises.

The layouts of the kitchens did not lend themselves to energy conservation. Refrigerating appliances in close proximity to a heat source, (grill, etc.) were observed to have raised consumptions by approximately 30% in some instances. However the causality of this

relationship cannot be determined in this study due to the highly variable makes, models, ages and conditions of the appliances. An increased frequency of deliveries, lowering food storage capacity requirements is a consideration for energy reduction.

3.2.2. Grills

Grills contribute a relatively constant load to the kitchens electricity use (12%). The grill is significantly influenced by operator behavior. figure 10 compares two grills of the same make and model on the same day. The grill in Kitchen 10 suffers from poor, (but typical) operator behavior. The appliance is switched to maximum where it remained until the end of service; it consumed 49 kWh. The grill in Kitchen 6 was managed much more appropriately from an energy conservation perspective; the temperature was reduced when not in operation. This grill consumed 14 kWh; 71% saving due to appropriate behavioral management alone.

It was observed during study that the grill was often used merely to brown items (such as sausages or cheese toppings) after oven cooking, which lasted only a few minutes. Options for reducing energy use include extending the oven-cooking time to include browning. The warm-up time of approximately 20 minutes may be an obstacle to turning it down or off.

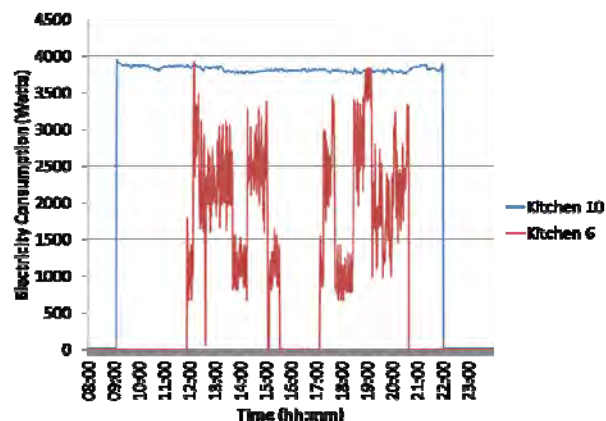


Figure 8 Grill Monitored Electricity Consumption (Watts) at Kitchens 10 and 6

3.2.3. Further Appliance Observations

The vast majority of appliances monitored had the option to be thermostatically controlled with varying levels of heat set by the operator but this was poorly utilised. Only the consumption profiles of microwaves and to some extent fryers and combination ovens were found to relate to usage/food output; consumption increased during cooking and automatically decreases after the pre-programmed cooking times. The appliances are not usually insulated; manufacturers claim this is due to associated fire risk (FSTC 2002).

Bain Maries are regularly employed in commercial kitchens to maintain food at the desired temperature while further food items are waiting to be served. Interestingly, research suggests that this practice reduces the nutrient content and palatability of the items

(S. Probert and M. Newborough 1985). In the sample kitchens, Bain Mariers were only used to keep sauces at serving temperatures and were correspondingly oversized for this function. Sometimes plates were held at high temperature within the cabinet. Hot-holding appeared to be an extremely wasteful and potentially unnecessary activity; with better management of the cooking operations elimination of hot-holding may save an average of 48 kWh per day.

4. CONCLUSIONS

Consumption is vastly greater than literature estimates suggest with the kitchen and food related activities greatly outweighing other energy consuming activities.

Consumption is highly variable between kitchens and appliances.

Catering appliance design does not lend itself to energy-thrift behavior. One clear approach to energy reduction is through better product design; this has been the focus of recent UK and EU policy via the Eco-design of Energy using Products Directive (2005/32/EC), which was recast and enlarged to include catering appliance in 2011 (Eco-design of Energy-related Products Directive – 2009/125/EC).

Operator behavior is a major factor in the excessive use of electricity. A strict “turn-on, turn-down, turn-off” schedule could be provided, however it is likely that in the busy working environment this may be ignored given that controls are available but not used by the operators. Turning appliances down or off when not required would save significant amounts of energy, carbon and cost. More research is required in order to motivate and influence staff in this area.

Poor levels of maintenance significantly contribute to excessive energy use in refrigerators.

There is much potential from further research into the choice of cooking method and corresponding appliances. Many different methods of cooking the same item were observed during this study. Better management of these choices should focus on reducing the need for hot-holding.

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LAKE WATER SURVEY AND A NEW COMPOUND WSHP SCHEME

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ABSTRACT

After detailed investigating in a lake in Hunan province of China on June to July 2012, some field data including water temperature and climate are presented in this paper. Several cold and heat sources schemes for adding new central air-conditioning system were provided. A new compound WSHP Scheme is discussed in detail and shows some potential of feasibility, but it is difficult for cost and complexity on constructing. Survey data and schemes in this research could be used as reference for other similar project.

Keywords: Lake water source heat pump; add capacity for water heat pump; temperature rise of lake water; cooling by evaporating.

1. INTRODUCTION

There exist a water source heat pump (WSHP) air-condition (AC) system in a lake located in middle of Hunan province in hot-summer and cold-winter climate zone. This system produces 8000 kW max cooling capacity in summer for four mansions with total 122600 m². Now, there need to develop another new 40000 m² commercial building around this lake, which contains restaurant, cinema, fitness center, swimming pool, market and so on. Another 8000 kW cooling load will add to new system providing 200 W/m². Finding suitable heat and cold sources for this project is our aim.

This lake consists of three ponds which individual named as up-pond, middle-pond and down-pond. Down-pond covering water area of 42000 m² and deep of 3 m average affords loads alone for existing AC system. This system has been operating in 8 years, but lake water temperature is a little high in summer and electrical heating should be used in coldest days of winter, which means it is impossible to add another water pump system for new buildings. New heat and cold sources should be found here. Layout map and satellite picture from Google-earth shows in Figure 1.

New system loads are approximately 8000 kW in summer and 5000 kW in winter separately. Because middle-pond has been occupied by another WSHP system, only up-pond, hill and ambient soil can be used as cold and heat sources. Besides these surrounding conditions, 600 ton/hour (t/h) water from nearby river could carry off parts of loads. How to choose heat and cool sources should be discussed.

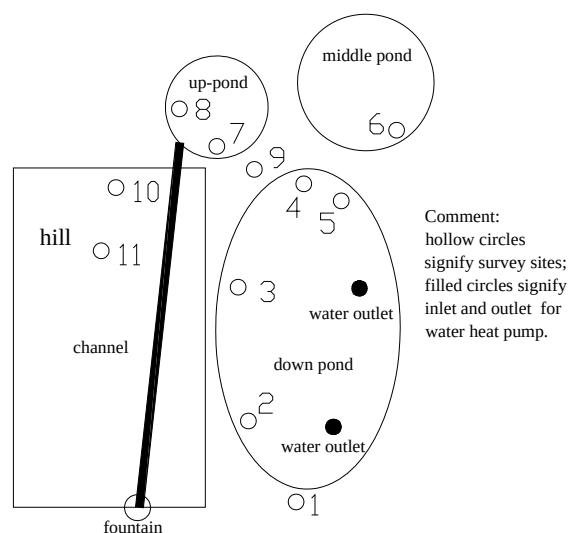


Figure 1 Satellite image and Survey sites

2. SURVEY

2.1 Data from Survey

During June 26 to July 6 in 2012, investigation had been carried three times and data are displayed in Table 1 to Table 3. Temperature measuring sites' depths were in 10 cm below water surface, or 3 cm under soil surface.

Table 1 Survey data on 4:30 pm, June 29

Site	Water/soil Temperature (°C)	Wind Speed (m/s)	Air Temperature (°C)	Relative humidity (%)
1	—	1	36.3	51
2	34.8	—	34	45
3	34.7	—	33.8	54
4	35.0	—	33.9	58
5	34.4	—	33.5	62
6	34.8	—	33.7	58
7	34	—	34.3	58.2
8	33.5	—	34.4	60
9	—	0.8		
10	29.2	—	33.7	55
11	32.2	—	33.1	60

Table 2 Survey data on 4:00 pm, July 6

Site	Water/soil Temperature (°C)	Wind Speed (m/s)	Air Temperature (°C)	Relative humidity (%)
2	36	-	36.5	-
3	35.5	-	-	-
4	35.5	-	-	-
5	35.3	-	-	-
7	34.7	-	34.3	58.2
8	34	-	34.4	60

Table 3 Survey data on 9:20 am, July 9

Site	Water/soil Temperature (°C)	Wind Speed (m/s)	Air Temperature (°C)	Relative Humidity (%)
1	-	2.8	36	50
2	32.8	-	33.4	48
3	34	-	36	45
4	32.5	-	35.8	48
5	33	-	35.2	51.8
6	33.7	-	35.4	52
7	32.4	-	36.8	58.2
8	31.8	-	33.8	41.9
9	-	2.1	-	-
10	28.8	-	35.3	47.6
11	42.5	-	36.4	46

In Table 1, water flux from lake to refrigerator was about 600 t/h. Incoming water temperature in morning

5:00 was 31.2 °C and afternoon 4:30 was 32.3 °C respectively. Meanwhile, chilled water supplying to fan coil was at temperature of 11.5 °C, and return water was 15.7 °C.

In Table 2, supply water was at 33.5 °C, and return water was 38.3 °C

In Table 3, cooling water exceeded 1000 t/h. Imported lake water were at temperature of 32.7 °C on 5:00 am, then 33.8 °C at 9:00 am. At the same time, supply and return chilled water were separately at 14.8 °C and 17.7 °C. On 5:00 pm, imported lake water was at 34 °C.

2.2 Analysis

There are some conclusions based on field survey.

(1) In Table 1 of June 29, lake water flux to refrigerator was 600 t/h and temperature rise of lake water was 0.9°C, which got close to limit of China National Standard for surface water quality.

(2) In Table 2 of July 6, water temperature was 1°C higher than Table 1, which means if hot weather continued lake water temperature would rise too.

(3) In July 6 and 9, flux exceeded 1000 t/h while temperature rise was still more than 1°C, which meant increasing cooling water volume had effect on water but not obviously.

(4) The water temperature under aquatic vegetation was lower than other exposed place inconspicuous.

(5) Temperature of soil in woods was much lower than water, but soil exposed under solar was obviously higher than water, which showed that trees had apparent shaded effect.

(6) After 10 days, lake water temperature rise reached 1.5°C, which caused by successive hot weather.

(7) Compared Table 1 with 3, former water temperature was distinctly higher than latter in similar air temperature, which was caused by higher wind speed of Table 3 than 1 evaporating more water.

(8) All of above draw a conclusion that the existing water heat pump could well operate, but directly add new one is impossible, so new heat and cold source need to be produced.

3. A NEW COMPOUND SYSTEM

There are at least three schemes: cooling tower + gas boiler; ground-source heat pump; compound source. First two methods are common, and the third one will be explained in detail. The third scheme consists of up-pond, 600 t/h river water, fountain effect, water channel, cooling tower, and gas boiler.

3.1 Cooling and heating method

Half cool load is provided by lake water, and another half is provided by cool tower. Shown in figure 2, water is used as cool source when load is lower than 4000 kW. When load is higher than 4000 kW, cooling tower will be added to supply more cold.

Similar as summer, heating load is part from water both lake and river, and others from boiler. If water temperature is below 3.6 °C, boiler will afford all heat loads.

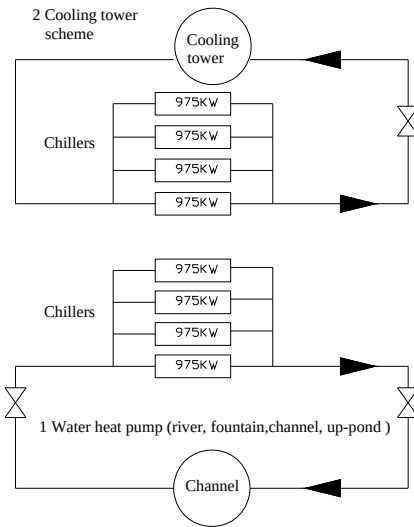


Figure 2 Scheme of cooling sources

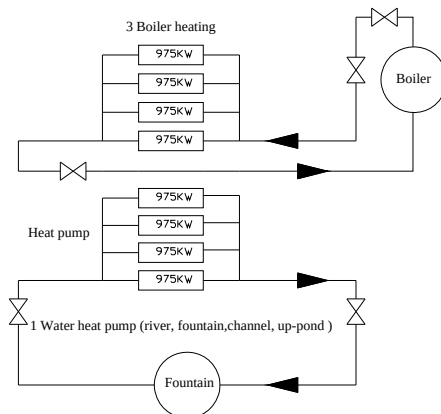


Figure 3 Scheme of heating sources

3.2 Cooling and heating capacity

3.2.1 In summer

(1) Fountain effect

According to former research from literature [5] in which one example was given to display calculated process, then cooling capacity here is about 300 kW.

(2) Up-pond

Area of up-pond is 7500 m² which is one-sixth of down-pond and water depth of former is one-third of latter, so cooling capacity of up-pond is 225 kW about one-eighteenth of down-pond.

(3) Water channel

Heat dissipates from surface water to adjacent atmosphere and channel wall surrounding with soil. Assume that channel cross profile dimensions are shown as Figure 4 and water flux is 600 t/h. The water

surface cooling capacity is approximately 200 W/m² based on data from literature [8-10] and spots investigation. Channel wall dissipates heat about 200 W/m² according to experiment of soil source heat pump system from literature [5]. Length of channel is 400 m and total cooling capacity is about 132.8 kW.

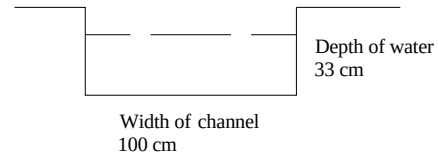


Figure 4 Profile map of channel

(4) River water

Existing lake water system's capacity supplies max 8000 kW with 1500 t/h, so river water gives estimated 3200 kW with 600 t/h. Local hydrologic data show that this the river's highest temperature in summer is blow 32 °C which is obviously lower lake water temperature in this survey, so it is possible to provide 3200 kW cooling power with 600 t/h.

3.2.2 In winter

(1) Up-pond

According to literature [1], heating capacity of lake is average 40 W/m², total power is 300 kW summed from 7500 m².

(2) Water channel

Considering heat balance between winter and summer, it will provide 200 kW of heat by water channel.

(3) River water

Based on hydrologic data, the lowest water temperature of adjacent river is about 7 °C, which is apparently higher than lake. The heat capacity of 600 t/h river water might as well produce hypothetical 3200 kW.

(4) Control strategies

In summer of July to September, both water and cooling tower are used to provide cooling power to buildings, and each cool source occupies approximately 50%. In winter of December to next February, new system's heat load depends on water and boiler too. In spring and autumn, because heat and cool loads are small, it can provide enough heat and cold to buildings only by water, which means saving some electric energy for high-efficiency heat exchange.

4. CONCLUSION

In this paper, it is providing a new compound heat and cold source, which includes fountain, water channel, river, a small pond, cooling tower, and boiler. Fountain makes use of water evaporation to dissipate heat. Water channel makes use of soil by heat conduction. Water from river and pond contribute a lot of cold power. Cooling tower and boiler treat as complementary cold and heat source. In most time of

the year, water is able to supply enough power for cooling and heating, or else cooling tower and boiler launched during a few hottest and coldest days. But complexity and cost are the trouble for this added water system, so this scheme is still under consideration. This paper's scheme might as well regard as reference for other similar project.

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'NON AIR-CONDITIONING' AND ENERGY CONSERVATION OF SEASIDE BUILDING OF XIAMEN IN SUMMER

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ABSTRACT

Xiamen is seaside city. It has favorable thermal environment resources. The natural thermal environment should be utilized for making the satisfying thermal comfort in urban residential building and energy saving should be taken into account by using non-air conditioning approach. Some representative residential building are chosen to test the thermal environment to Xiamen. Some test list are made to note the thermal environment, some thermal environmental climate parameters of typical room, including indoor air temperature, humidity, air flow speed, etc. are tested repeatedly; Meanwhile, make resident's thermal feeling investigation forms, write down residents' thermal feeling with questionnaire investigation and thermal comfortable index of 7 grades of ASHRAE. Result indicates room temperature of summer of Xiamen change range 25-33 degrees Centigrade, relative humidity changes the range as 55- 85%, the velocity of air changes the range as 0-1.0m/s; Residents feel comfortable under the thermal environment of some period, can accept to the thermal environment in most period, but feeling hot under the thermal environment extremely in a few days of summer, can't accept, must run a air-condition. Suggest that, for seaside city in hot days, the natural ventilation and interval natural ventilation should be the main fundamental methods of improving the building thermal environment.

Keywords: Thermal environment; Thermal comfort; Air-conditioning energy saving; Natural ventilation; Seaside resident building.

1. INTRODUCTION

The good and comfortable environment is beneficial to the health, and can improve working efficiency, make people imbued with more creativity. This means strengthening the competitiveness, increase economic efficiency. So, creating the good building environment not merely has a meaning on technology, and has social effect and economic meaning. In order to understand the situation of indoor thermal environment of the seaside residential building generally, a field measurement on the condition of summer was made in Xiamen. The curves of indoor air temperature and relative humidity reflect the character of high temperature and high relative humidity in summer in Xiamen residential building. Discuss the methods of building energy saving. Suggest that, for seaside city in hot days, the natural ventilation and interval natural ventilation should be the main fundamental methods of improving the building thermal environment.

Today, people pay attention to one's own life living environment more and more, indoor climate and human body's comfortable problem cause people's attention day by day. But pursue the increase that the comfortable cost is energy consumption excessively, with the fast development of economy, the proportion of taking total energy consumption of the building of energy consumption of the air conditioner increases day

by day, is up to 30-40%. So how base on the premise that it is satisfied for the thermal comfort, utilize natural environment fully, it should be our basic point of consider in the course of such decision as planning, designing, etc. to realize air conditioning energy-conservation in way that 'it is the no-air-conditioning' ..

2. LOCAL CLIMATE STATUS

The regional climate is different widely in China, and the thermal environment of different regions is very different. In the course of humankind and natural coordination, locals have formed custom and peculiar response of thermal comfort. That is important to know the thermal environment of residential building and people's thermal comfort by test and questionnaire, to study the influence for main environmental parameters (temperature, wind speed, relative humidity, average radiation temperature, etc) to people's thermal comfort.

Xiamen, lying in the southeast of China, has the latitudes of northern 24°27', longitude of east 118°04' with an elevation of 63.2 m., which is showing the features of south shore cities climate; it has the characteristic of high humidity and temperature indoors and high wind speed outdoors in summer, which induce two extremely discomfort climate, one is sunny day with high temperature and large difference of temperature in day-time and night-time, the other is

stuffy day with little difference of temperature in day-time and night-time, but the relative humidity can reach to 83%, which is in favor of the reproduction of bacteria, and the outdoor is more comfortable than indoor. The atmospheric pressure in summer is about 999.1hPa, the extremely highest temperature is 38.5°C, and the average of extremely highest temperature is 36.4°C. Accordingly, it belongs to the region with hot in summer and warm in winter

Annual different levels dry bulb temperature frequency is shown in Figure 1, and day mean relative humidity change is shown in Figure 2. In addition, locality summer sunshine is strong relatively; July total radiation is 529MJ/m², moon for August total radiation 499.5 MJ/m². The average wind speed is 2.5m/s outdoors in summer, the southeaster wind is in the majority[1].

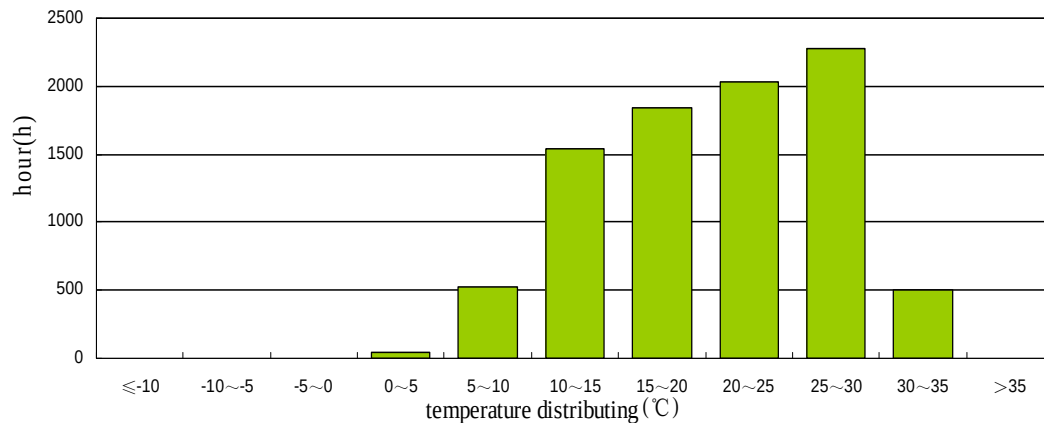


Figure 1 Annual different levels dry bulb temperature frequency

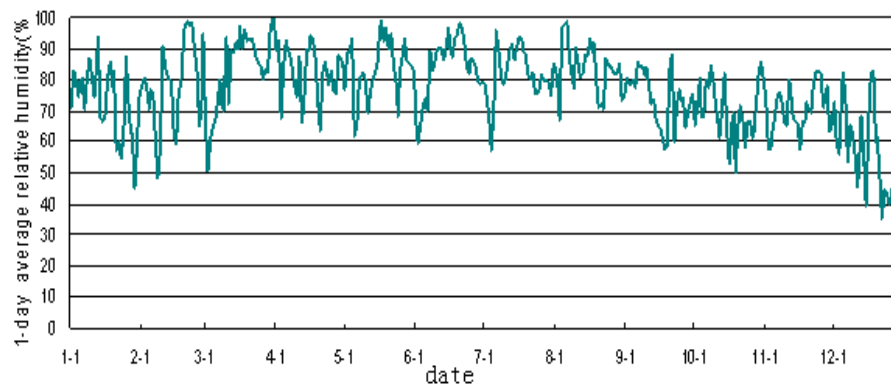


Figure 2 The change of 1-day mean relative humidity

3. THE TEST OF THERMAL ENVIRONMENT AND THE INVESTIGATION OF RESIDENT'S THERMAL SENSE

3.1 The first sample

The residential thermal environment testing is introduced in the nether content. Base on the weather datum, in Xiamen, highest temperature and humidity was occurred on the last ten days of July and the first ten day of Aug in a year, so that, a day was selected on the last ten days of July, 24-July.

The measurement was put in practice. A dwelling house with four rooms was selected for this study. Hereinto, the living room was about 6m×4m×3m (length×width×height) with only an external wall southward, the building material was composed of air bricks, thick was 200mm, and wall tiles; The size of the veranda-window was 2.5m×2.5m (width×height).

The test meter is CENTER 313, digital data-logger Humidity /Temperature meter. The meter measurement range is Humidity; 0%-100%, Temperature: -20°C- +60°C, and accuracy is Humidity warps 2.5%, Temperature warps 0.7°C. The test meter was placed in the distance floor one meter.

The test result of Temperature / Humidity is showed in the following Figure 3 and Figure 4.

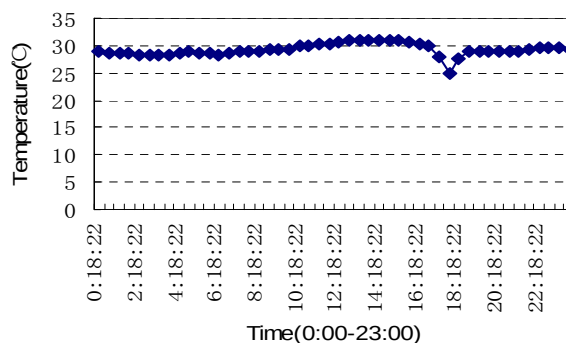


Figure 3 1-hour temperature on 24-July

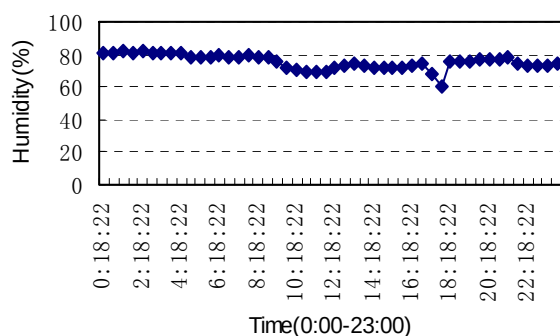


Figure 4 1-hour relative humidity on 24-July

In a majority of time of one day, the indoor Humidity /Temperature were tested by natural Ventilation condition. But during fifth-hour to sixth-hour,an air conditioner was entered start-up and running.

We choose the representative several groups' data of temperature and humidity to analyze from Fig. 3 and Fig. 4. and will the occupant' sensation list in Table 1.

Table 1 The occupant' thermal sensation

The air temperature, °C	The air Humidity, %	thermal sensation
29.5	75	Slight hot
28.6	82.1	warm
31.1	71.5	hot
25	61	Slight cool

When air temperature, relative humidity are 29.5 degrees Centigrade/75% (wet bulb temperature is 25.7 degrees Centigrade), it is representative air state in most period in the room.

When air temperature, relative humidity are 28.6 degrees Centigrade / 82.1% (wet bulb temperature is 26 degrees Centigrade), it has represented the moment indoor air state, at 1:18:22, has the largest humidity at this moment during one day.

When air temperature, relative humidity are 31.1 degrees Centigrade / 71.5% (wet bulb temperature is 26.6

degrees Centigrade), it has represented the moment indoor air state, at 14:48:22, has the highest temperature at this moment during one day. The occupant feels hot at this moment; need to open the air conditioner.

When air temperature, relative humidity are 25 degrees Centigrade / 61% (wet bulb temperature is 19.5 degrees Centigrade), it represent indoor air state on the 17:48:22 moment, when it at this moment, running air conditioner (air conditioner running time as from 17:30:00 to 18:30:00).

3.2 The second sample

Base on the Fig.1, the highest temperature was occurred on the last ten days of Jul and the first ten day of Aug in a year, so that, the measurement was conducted to characterize the impact of different meteorological parameters on thermal environment during 13-Jul to 31-Jul. A dwelling house with four rooms was selected for this study. One of the bedrooms was about 6m×4m×3m (length×width×height) with only an external wall southward, the building material was composed of air bricks, thick was 200mm, and wall tiles.

The size of the window was 1.5m×2m (width×height). Fig.5 and fig.6 was part of the testing results presented, with an average temperature 31-32°C and average relative humidity 60-80% of one day during 13-Jul to 31-Jul which made people feel slightly hot. The sampling was based on an interval natural ventilation condition which means no air-conditioning system.

Base on the test result of the temperature and humidity in the fig.5 and fig.6, at the same time, when air flow speed is different, the occupants' thermal sensation was investigated, as Table 2.

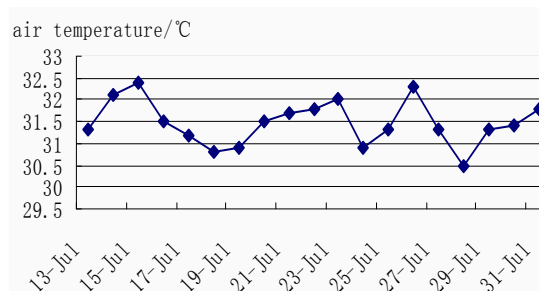


Figure 5 1-day average temperature

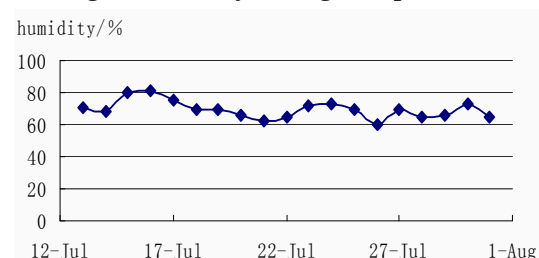


Figure 6 1-day average relative humidity

Table 2 The occupants' thermal sensation in different air flow speeds

Item Date	The air temperature °C	The mean radiant temperature °C	The air humidity %	Thermal sensation		
				Air velocity, m/s		
				0.2	0.5	1
15-Jul	32.4	33.0	79	Hot	Hot	Slightly hot
18-Jul	30.7	31.0	70	Slightly hot	Slightly hot	Slightly warm
21-Jul	31.6	32.1	61	Slightly hot	Slightly hot	Slightly warm
24-Jul	30.8	31.2	72	Slightly hot	Warm	Slightly warm
27-Jul	31.3	31.8	69	Slightly hot	Warm	Warm
30-Jul	31.3	31.8	72	Slightly hot	Warm	Slightly warm

Air velocity is one of the most important factor that impact the human thermal comfort, it is directly related with the heat exchange of convection and evaporation, also the air exchange rate[2][3]. High quality of airflow can not only satisfy the request of occupants with comfortable environment, but also save energy. When air temperature is lower than the surface temperature of human skin, the increase of air velocity should bring on the heat diffuse intensified, and make for human cool, and human being can feel that the air temperature fall down 2 to 3°C when air velocity increase 1m/s[4].

3.3 The third sample

The house chosen to test is 20-401, Centre garden residence region, Jimei, Xiamen; it is in the westernmost end of building. The building faces to south, amount to seven layers, built up at 2003; it is the multi-layer frame structure. It is a set of two drawing rooms and three bedrooms house for 401.

Among them, the test results to the living room are shown as Table 3 and the inhabitant' response to thermal environment in questionnaire investigation is in the Table 4.

Table 3 Thermal state of the living room

Time	Temperature, °C	Humidity, %	Wind speed, m/s
6:00	30.0	67	0.3
8:00	30.3	71	0.3
10:00	30.4	70	0.3
12:00	30.8	65	0.5
14:00	31.3	58	0.6
16:00	30.1	57	0.2
18:00	30.7	57	0.3
20:00	30.1	65	0.5
22:00	30.7	68	0.6
24:00	30.4	68	0.6

Remarks: there is only correspondingly strong radiation of balcony door in the living room, the others are the interior wall, so that, consider the indoor average radiation temperature is $t_r = t_g + 0.5$.

In the test day, the description of inhabitant's living state is in the following tabulation (Table 5).

From table 5, can find out that occupiers like sense of wind, get used to open the door and window; compared with air conditioner, have priority to use the electric fan at the time of heat regulation. This kind of habit has some reason for Xiamen near sea and therefore has a better outdoor wind Environment.

Table 4 The inhabitant' thermal sensation

Time	Numbers of peoples	Occupant' thermal sensation
6:00	one	Slightly warm, accept
8:00	Two	Warm, acceptable
10:00	two	Warm, acceptable
12:00	Two	Slightly warm, accept
14:00	three	Slightly warm, accept
16:00	three	Two man, warm, One women, Slightly warm
18:00	two	warm, acceptable
20:00	four	Slightly warm, accept
22:00	four	Slightly warm, accept
24:00	three	Slightly warm, accept

Table 3 and Table 5 shows that indoor temperature is up to 31.3 degrees Centigrade at 14:00, but the relative humidity is relatively lower at this moment. Occupiers utilized the electric fan to bring airflow, enable themselves to accept the hot environment. The test results indicate, during the most of the time in summer of Xiamen, the occupants can improve the hot environment through regulating their clothing, sunshade and ventilating measure (nature ventilation and running fan), enabling themselves to meet and accept it. Only run the air conditioning system in a hotter period.

Table 5 The description of inhabitant's living state

Time	Indoor ventilation: 1 without wind 2 breathing 3 distinct windiness	Open the door: 1 yes 2 no	Window: 1 open 2 shut	cover curtain: 1 yes 2 no	Running fan: 1 yes 2 no	Running air condition: 1 yes 2 no	Running heat equipments: 1 TV 2 lamp 3 nothing
6:00	1	1	1	2	2	2	3
8:00	1	1	1	2	2	2	3
10:00	1	1	1	2	2	2	3
12:00	2	1	1	2	1	2	1
14:00	2	1	1	2	1	2	3
16:00	1	1	1	2	2	2	3
18:00	1	1	1	2	2	2	3
20:00	2	1	1	2	1	2	1, 2
22:00	2	1	1	2	1	2	1, 2
24:00	2	1	1	2	1	2	3

4. RESULT OF THE TEST AND INVESTIGATION

Result indicate room temperature of summer of Xiamen changes range 25-33 degrees Centigrade, relative humidity changes the range as 55- 85%, the velocity of air changes the range as 0-1.0m/s; Residents feel comfortable under the thermal environment of some period, can accept to the thermal environment in most period, but feeling hot under the thermal environment extremely in a few days of summer, can't accept, must run a air-condition.

During the most of the time in summer of Xiamen, the occupants can improve the hot environment through regulating their clothing, sunshade and ventilating measure (nature ventilation and running fan).

Xiamen is seaside city. It has favorable thermal environment resources. The natural thermal environment should be utilized for making the satisfying thermal comfort in urban residential building. Energy saving should be taken into account by using non-air conditioning approach.

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FEASIBILITY OF AEROGEL GLAZING SYSTEMS FOR BUILDING ENERGY EFFICIENCY IN CHINA

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ABSTRACT

Thermal characteristics including the heat transfer coefficient (U) and the shading coefficient (SC) of the exterior building glass have important effects on the building heating and cooling loads owing to its poor thermal insulation compared to the building walling system. Two newly developed aerogel glazing systems (viz. #4 and #5) and three common types of glass (viz. clear glass #1 and insulating glass #2 and #3) are selected in a model building so as to evaluate their feasibility in different climate zones in China. Both the component and the total air conditioning loads of the model building for five types of glass are calculated in five typical climate zones by using eQuest code. The results show that the aerogel glazing system #5 has the highest energy efficiency based on the clear glass #1, which could be up to 20% in Harbin and 11% and 9% in Beijing and Shanghai, respectively. Therefore, the aerogel glazing system #5 could be extremely feasible in the Severe Cold Region and significantly feasible in the Cold and Hot-Summer Warm-Winter Regions. The findings could be helpful to further optimize the thermal characteristics of glass aiming to develop new high-performance glass for building energy efficiency.

Keywords: Energy conservation in buildings; glass; aerogel; climate zone; China

1. INTRODUCTION

China has become the second largest energy producer and consumer, behind only the USA [1]. In 2010, China's building sector accounted for 20.9% of China's total national energy consumption [2]. Of the annual energy use of public buildings, approximately 50~60% is for HVAC (Heating, Ventilating, and Air-Conditioning), and 20~30% is for lighting [3].

The poor thermal performance of common building glass often leads to a high heating and cooling energy demand. Therefore, the exterior building glass has attracted increasing interests aiming to building energy efficiency. Gasparella et al. [4] evaluated the effect of various glazing systems (e.g. two double and two triple glazing) on the building energy consumption in winter and summer of a well insulated residential building in Southern Europe. Hassouneh et al. [5] studied the effects of the types and orientation of the glass window on the building energy in Arman. They found that increasing the glass area facing south, east and west or decreasing the glass facing north can decrease heating energy in winter. Jaber and Ajib [6] have investigated the feasibility of the four types of glass (viz., single glazed, double glazed L, double glazed H and triple glazed) in the three climate zones (viz., Amman, Aqaba and Berlin). The results showed that the heating load was highly sensitive to windows size and type as compared with cooling load. In China,

Long et al. [7] calculated the building energy consumption of two residential buildings located in 6 different cities for various climate zones in China using the DOE-2 code and the Characteristic Temperature Method (CTM), respectively. They found that reducing the shading coefficient can significantly decreased the annual cooling energy.

Very recently, aerogel has become a promising material for application in highly energy efficient window glass owing to its extremely low thermal conductivity and high visual transmittance. Schultz et al. [8] developed a newly evacuated aerogel glazing of with the area of 55×55 cm² and the aerogel thickness of 15 mm. The aerogel glazing had the heat transfer coefficient of 0.66 W/(m²·K) and the solar transmittance of 76%. The few literatures [8-10] have shown that the aerogel glazing is the ideal material that has the excellent combination of the high solar and light transmittance and the low thermal conductivity.

There are very few literatures focused on the application of the newly developed aerogel glazing systems in building energy efficiency in China. Moreover, owing to the climatic diversity in China, the feasibility and the energy efficiency potential may greatly vary for heating and cooling loads in different climate zones. Therefore, the newly developed aerogel glazing systems and several types of common used glass are selected in this paper so as to evaluate their

feasibility to different climates for building energy efficiency in China.

2. MODEL BUILDING AND VARIOUS GLASS FOR FIVE TYPICAL CLIMATE ZONES

China is divided into the five typical climate zones based on climate characteristics which are Severe Cold Region, Cold Region, Hot-Summer Cold-Winter Region, Hot-Summer Warm-Winter Region, and Temperate (Mild) Region. Five typical cities (e.g. Harbin, Beijing, Shanghai, Guangzhou, and Kunming) are selected to calculate the building air conditioning load in the five climate zones as listed in Table 1.

Table 1 Heat transfer coefficients of exterior walls and roofs

City	Climate zone	Building element	Heat transfer coefficient U ($W/(m^2 \cdot K)$)
Harbin	Severe Cold	Wall	0.44
		Roof	0.35
Beijing	Cold	Wall	0.60
		Roof	0.55
Shanghai	Hot-Summer	Wall	1.00
	Cold-Winter	Roof	0.70
Guangzhou	Hot-Summer	Wall	1.47
	Warm-Winter	Roof	0.89
Kunming	Temperate	Wall	1.47
	(Mild)	Roof	0.89

A 35 m×35 m, 40-storey office building with curtain walling design and a ratio 0.4 of window to wall is selected as a model building in this study. The azimuth of the model building is south. The total gross floor area is 49,000 m² including 41,160 m² air conditioned and 7400 m² non-air conditioned floor areas [11]. The air conditioned space is divided into five zones - four at the perimeter and one in the interior. Details of the office building model can be found in Ref. [12]. For various climate zones and cities, the internal loads and indoor design conditions are assumed to be the same. The design parameters include the occupancy density of 8 m²/person, the lighting power density of 18 W/m² and the electric facility power density of 13 W/m². The indoor design cooling temperature and the relative humidity are set at 26 °C and 40~65% in the summer, respectively. The indoor heating temperature is set at 20 °C during the winter. The infiltration rate is designed according to the number of ventilator of 0.45 per hour throughout the year. The service period of the HVAC and the lighting system in the building is 7:00am to 6:00pm from Monday to Friday all year round. For various climate zones and cities, the designed thermal insulation parameters of the building exterior envelopes are different. For example, the walls and roofs have greater thermal insulation in Harbin than that in Guangzhou. Table 1 lists the designed heat transfer coefficients of the exterior walls and the roofs of the model building located in the five typical cities respectively [13].

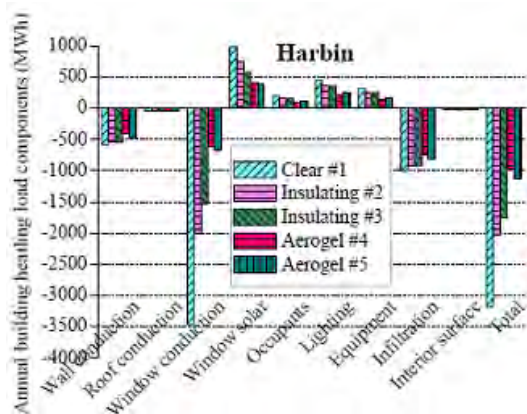
Five types of exterior building glass are chosen to determine the hourly building heating and cooling loads in five typical cities so as to evaluate the feasibility of different glass in different climate zones. The glass #1 is common clear glass, the glass #2 and #3 are insulating glass. The glass #1~3 are selected according to the conventional exterior building glass providing in the Chinese industry standard JGJ/T151-2008 [14]. The aerogel glazing systems (e.g. #4 and #5) with super-thermal insulation performance [15, 16] are newly developed. The aerogel glazing systems #4 and #5 have the same heat transfer coefficient (U) but different shading coefficient (SC). The detailed structure and thermal characteristics (e.g. heat transfer coefficient (U) and shading coefficient (SC) of the five types of exterior building glass are shown in Table 2.

Table 2 Structure and thermal characteristics of exterior building glass

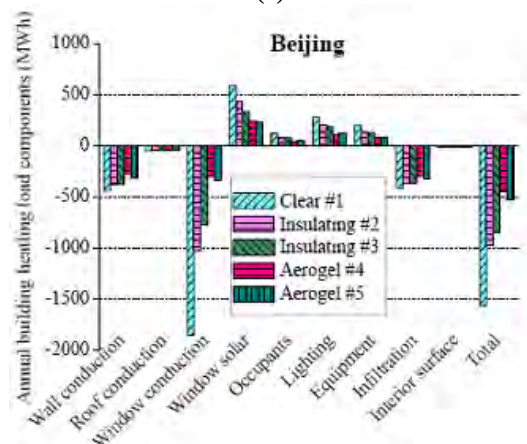
Glass type	Structure	Heat transfer coefficient U ($W/(m^2 \cdot K)$)	Shading coefficient SC
Clear glass #1	12 mm clear glass	5.5	0.84
Insulating glass #2	6 mm clear glass outer / 12 mm air interlayer / 6 mm clear glass inner	2.8	0.86
Insulating glass #3	6 mm low-emissivity glass outer / 12 mm air interlayer / 6 mm clear glass inner	1.9	0.62
Aerogel glass #4	4 mm low-iron glass outer / 15 mm aerogel interlayer / 4 mm low-iron glass inner	0.72	0.85
Aerogel glass #5	4 mm clear glass outer / 15 mm aerogel interlayer / 4 mm clear glass inner	0.72	0.59

3. RESULTS AND DISCUSSION

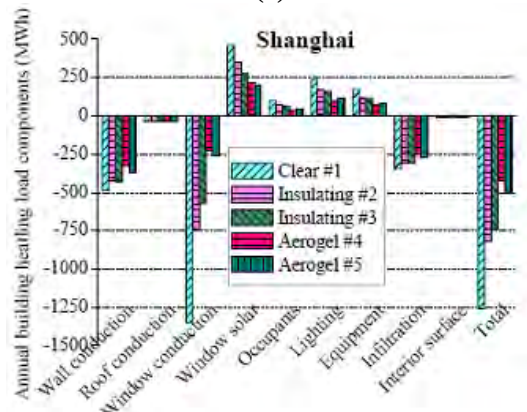
In the five typical cities (viz. Harbin, Beijing, Shanghai, Guangzhou and Kunming), the hourly and annual heating and cooling loads of the model building are calculated by using eQuest based on DOE-2 code [17] for the glass #1~5. It is worth pointing out that the computed building loads from DOE-2 is based on the heat gain/loss that occurred to the space at a single point temperature of 23 °C (mean of the summer and winter set points of 26 °C and 20 °C, respectively).



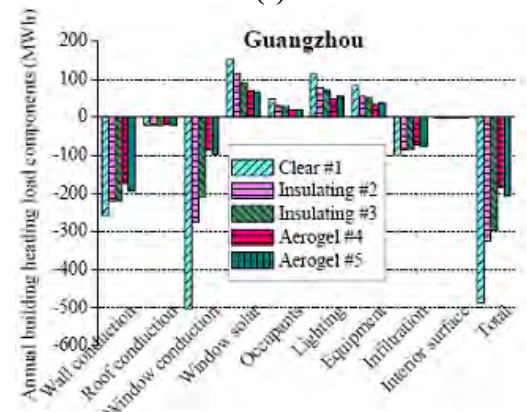
(a)



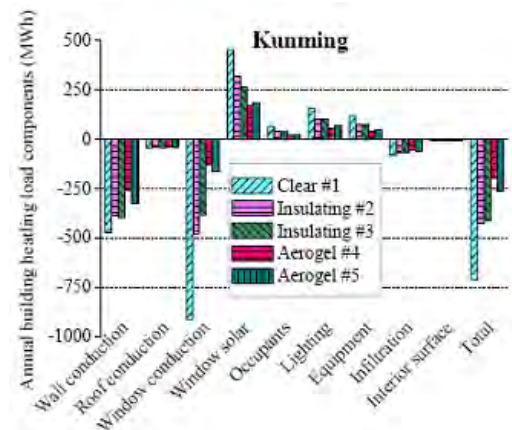
(b)



(c)



(d)



(e)

Figure 1 Annual building heating load components for five types of glass in five typical cities

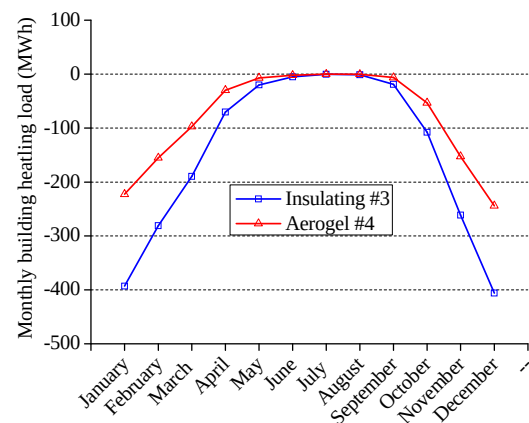


Figure 2 Monthly building heating loads for double glass #3 and aerogel glazing system #4 in Harbin

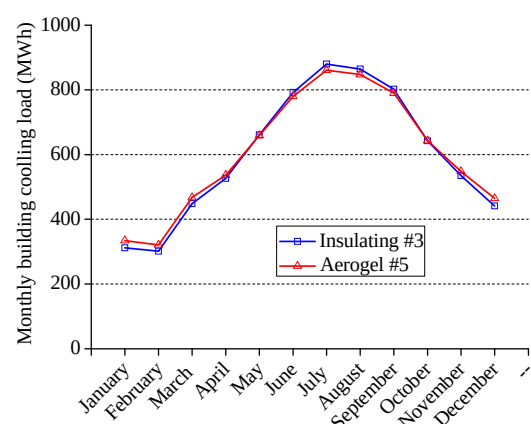


Figure 4 Monthly building cooling loads for double glass #3 and aerogel glazing system #5 in Guangzhou.

3.1 Building heating loads for five types of glass in five typical cities

The building heating load of the model building includes the heat gains/losses through the building

envelopes and the interior heat loads and infiltration so as to maintain the designed indoor thermal comfort environment. The annual building heating load comprises of nine parts. Four parts of the annual building heating load are owing to the heat loss/gain through building envelopes (e.g. wall conduction, roof conduction, window conduction and window solar). Three parts of the annual heating load are owing to the interior heat loads (e.g. occupants, electric lighting and office equipment). The other two parts are owing to the heat through infiltration and the heat conduction of the interior surfaces [18].

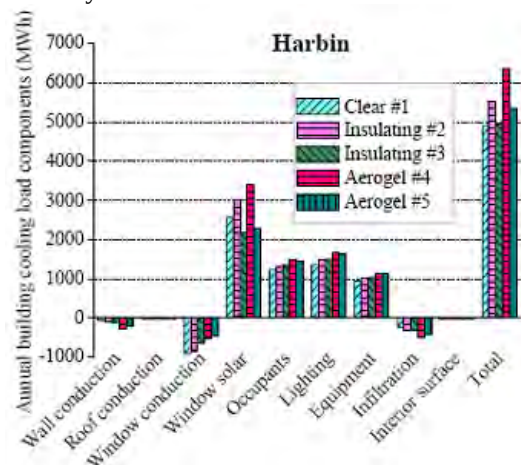
Fig. 1 shows the component annual heating loads of the model building for five types of glass #1~5 in the five typical cities (viz. Harbin, Beijing, Shanghai, Guangzhou and Kunming), respectively. It can be observed from the composition of the annual heating load of the model building in Harbin as a typical city of Severe Cold Region that for the glass #1~3 the largest component of the annual building heating load in Harbin is owing to the heat conduction through the window. Since the window conduction component is dependent on the heat transfer coefficient (U) of the glass, the U value significantly affect the heating load of the building. For example, the glass #1 with the greatest U value of $5.5 \text{ W}/(\text{m}^2\cdot\text{K})$ leads to the largest window conduction component of -3457 MWh while the glass #3 with the smallest U value of $1.9 \text{ W}/(\text{m}^2\cdot\text{K})$ leads to the largest window conduction component of -1533 MWh. Herein the minus sign of the window conduction component means heat losses.

For the aerogel glazing systems #4 and #5, their U values are both $0.72 \text{ W}/(\text{m}^2\cdot\text{K})$, which is significantly lower than those of the glass #1~3. It can be observed that the window conduction components of the glass #4 and #5 in Harbin are significantly decreased to -599 and -678 MWh, respectively. As the U values are the same, the glass #4 with the greater shading coefficient of 0.85 leads to a lower heating load compared to the glass #5 with the SC value of 0.59. It is because that a higher SC value is favorable to the solar heat gain and the reduction of the heating load, which can be also observed in Harbin.

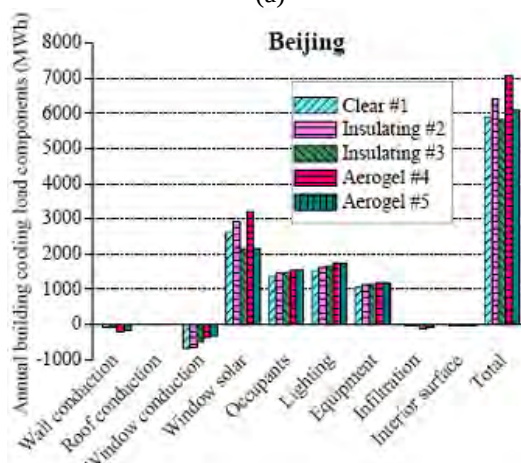
In Beijing as a typical city of Cold Region, for all the five types of glass #1~5, the heat conduction through the window is the largest component of the building heating load. Moreover, the solar heat gain through the window is positive and favorable to reduce the total building heating load. Therefore, it can be concluded that decreasing the U value and increasing the SC value of glass can reduce the building heating load in Cold Region. The same conclusion could be obtained from the simulation results in Shanghai, Guangzhou and Kunming.

Fig. 2 shows the monthly building heating loads for the insulating glass #3 and the aerogel glazing system #4 in Harbin as a typical city of Severe Cold Region. It can be observed that the monthly heating loads for the aerogel glazing system #4 are lower than that for the insulating glass #3. For example, the greatest monthly heating loads for the aerogel glazing system #4 and insulating glass #3 both occur in December and are -244

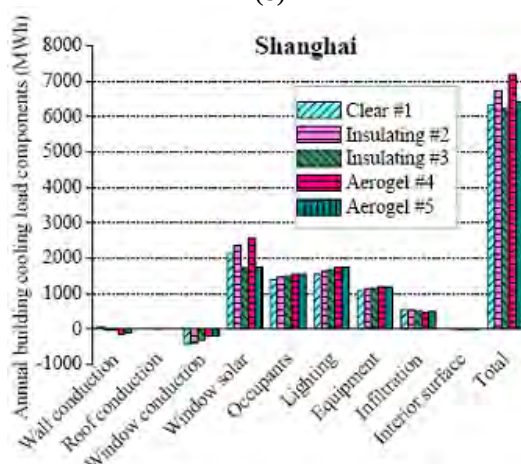
and -406 MWh, respectively. The monthly heating load in December for the aerogel glazing system #4 is just about 61% of that of the insulating glass #3. It is because that the aerogel glazing system #4 has a higher shading coefficient and a lower heat transfer coefficient those of the insulating glass #3. Therefore, the newly developed aerogel glazing systems are of high energy efficiency for heating owing to its excellent combination of a high shading coefficient and extreme low thermal conductivity.



(a)



(b)



(c)

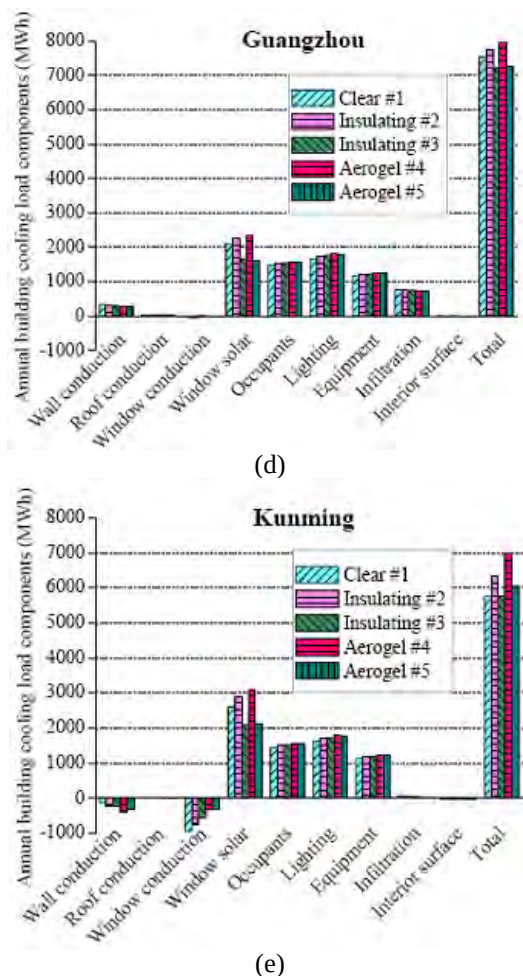


Figure 3 Annual building cooling load components for five types of glass in five typical cities

3.2 Building cooling loads for five types of glass in five typical cities

Fig. 3 shows the composition of the annual component cooling loads for five types of exterior building glass in the five typical cities, respectively. Taking the typical city of Hot-Summer Warm-Winter Region (viz. Guangzhou) as an example, it can be observed that for the clear glass #1, insulating glass #2 and aerogel glazing system #4 the largest cooling load component is the window solar component with 2111, 2265 and 2349 MWh, respectively. It is because that the three types of glass (viz. #1, #2 and #4) have higher shading coefficients of 0.84, 0.86 and 0.85, respectively, which means greater solar heat gain and accordingly higher cooling load. For the other two types of glass (viz. #3 and #5) with lower shading coefficients of 0.62 and 0.59, respectively, the window solar components are decreased and become the second largest cooling load components slightly smaller than the lighting components. On the other hand, the window conduction components owing to the U values are extremely small for all the five types of glass compared to the window solar component owing to SC values. It indicates that lowering the SC value of glass would be an efficient way to reduce the building cooling load. Similar

conclusions can be made through the cooling load components in the other four typical cities (viz. Harbin, Beijing, Shanghai and Kunming).

Fig. 4 shows the monthly building cooling loads for the insulating glass #3 and aerogel glazing system #5 in Guangzhou as a typical city of Hot-Summer Warm-Winter Region. It can be observed that the monthly cooling loads for the glass #3 and glass #5 are very close. For example, the glass #3 and glass #5 have the greatest monthly cooling load of approximately 870 MWh in July and the lowest one of approximately 310 MWh in February. The two types of glass have very close SC value (e.g. 0.62 for the glass #3 and 0.59 for the glass #5) although their U values have a big difference (e.g. 1.9 W/(m²·K) for the glass #3 and 0.72 W/(m²·K) for the glass #5). It can be easily deduced that the cooling load is significantly dependent on the SC value but slightly dependent on the U value of glass.

3.3. Feasibility of five types of glass in five climate zones in China

Fig. 5(a) shows the total heating and cooling loads of the model building for the five types of glass (viz. glass #1~5) in the five climate zones in China. It can be observed that the glass #1 with the highest U value of 5.5 W/(m² K) leads to the greatest total heating and cooling loads in Harbin (Severe Cold Region), Beijing (Cold Region) and Shanghai (Hot-summer Cold-winter Region). It is because that in the three climate zones the buildings are heating dominated and a higher U value will lead to a higher window conduction component of building heating load. In the other two cities (viz. Guangzhou for Hot-summer Warm-winter Region, Kunming for Temperate Region), the glass #4 with higher SC value of 0.85 and lower U value of 0.72 W/(m² K) leads to the greatest total heating and cooling loads. It is because that in the two climate zones the buildings are cooling dominated where a higher SC value will lead to a higher window solar co

Fig. 5(b) shows the energy efficient coefficients of the model building for the four types of glass (viz. glass #2~5) in the five typical cities in China, respectively. It can be observed that the aerogel glass #5 leads to the highest energy efficiency in the four typical cities (viz. Harbin, Beijing, Shanghai and Guangzhou) and a close highest energy efficiency in the fifth city (viz. Kunming). In Harbin for Severe Cold Region, the energy efficiency of the aerogel glass #5 could be up to 20%. In Beijing for Cold Region and Shanghai for Hot-summer and Cold-winter Region, the energy efficiency of the aerogel glass #5 is 11% and 9%, respectively. In Guangzhou for Hot-summer and Warm-winter Region and Kunming for Temperate Region, the energy efficiency of the aerogel glass #5 is 7% and 2.4%, respectively. Therefore, the aerogel glass #5 could be extremely feasible in severe Cold Region and significantly feasible in the Cold and Hot-summer Cold-winter Regions in China for the reduction of building energy consumption. In the other two climate zones (viz. Hot-summer Warm-winter Region, Temperate Region) the aerogel glass #5 could be conditionally selected

through a systemically economic evaluation of the building energy efficiency and additional aerogel cost.

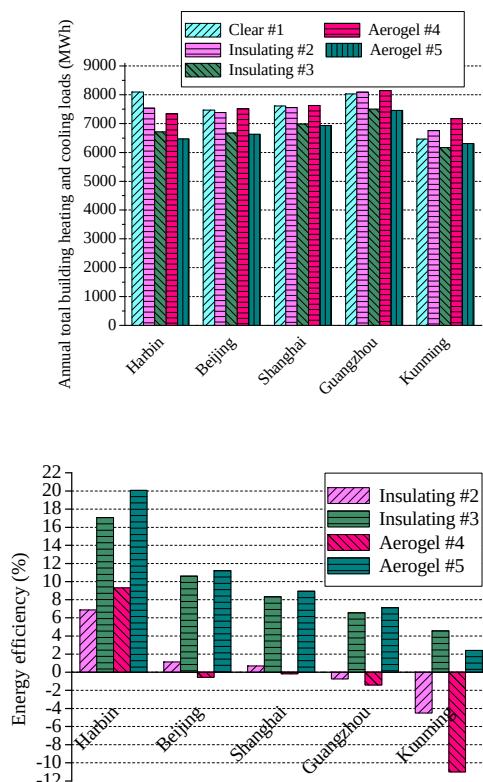


Figure 5 (top) Annual total heating and cooling loads and (bottom) annual energy efficiency for four types of glass (viz. glass #2~5) in five typical cities.

4. CONCLUSIONS

The component heating and cooling load of a model building for five types of glass (e.g. clear glass #1, insulating glass #2 and #3, and aerogel glazing systems #4 and #5) are calculated in five typical cities. The results indicate that the window conduction component relevant to the U value of glass and the window solar component relevant to the SC value of glass are the greatest one in the nine heating and cooling load components, respectively. Through integrating the nine load components, the aerogel glazing systems #5 leads to the lowest total heating and cooling loads in the four cities (viz. Harbin, Beijing, Shanghai and Guangzhou) and close lowest total heating and cooling loads in the fifth city (viz. Kunming).

ACKNOWLEDGEMENTS

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STUDY ON DEWING AND INTERNAL INSULATION BOARD THICKNESS OF EXTERIOR WALL WITH CAPILLARY RADIATION HEATING

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ABSTRACT

In winter, the interface of the internal insulation board and the wall surface is prone to condense, dew and go moldy. For this reason, the renovation exterior wall of the old building in Shanghai with internal thermal insulation is taken as the research object and the one-dimension dynamic heat transfer mathematical model of the exterior wall of the capillary radiation heating room is established calculating the interface temperature by using theoretical and experimental methods. Then, the dewing effect of different centrifugal glass wool thicknesses is analyzed in different indoor air states through MATLAB software simulation. The good consistency of the experimental and theoretical values verifies the applicability of the mathematical model and the numerical simulation results under dynamic and low outdoor temperature can provide some references for the internal thermal insulation renovation of old buildings.

Keywords: Capillary radiation heating; Old building; Internal thermal insulation; Centrifugal glass wool; Dewing.

1. INTRODUCTION

With the increasing anxiety of the energy crisis and environment pollution of the international society, energy conservation becomes one of the most important issues, in which building sector accounts for about one third of the total social energy consumption. Building insulation plays a significant role in the building energy efficiency^[1-3]. At present, there are three forms of thermal insulation, the sandwiched insulation, the internal and external insulation. Among them, the internal insulation board doesn't expose in harsh environment and has smaller technical difficulties. It is also more flexible to be constructed within every family individually. In recent years, the capillary radiation heating is increasingly used especially in Shanghai China due to its comfort and energy efficiency compared to conventional air conditioning^[4-5]. The advantages of internal insulation make it suitable for the exterior wall renovation of old buildings with capillary radiation heating. However, the interface of the internal insulation board and the wall surface is prone to condense which will reduce the performance of the insulation materials and even make the wall moldy and do harm to people. Thus, many experts have studied the heat and moisture transfer of internal insulation wall^[6-7].

In Shanghai, many envelope structures of old buildings are brick and the indoor water vapor partial pressure in winter is usually higher than the outdoor, causing the moisture passing from the indoor to outdoor. If the temperature inside the wall is lower than

the corresponding dew point temperature, dewing will occur. The exterior wall of the old building has reached the wet stability with the years passing by^[8]. The water vapor partial pressure of the indoor and the interface of the insulation board and the wall surface is close due to the good moisture absorption rate of the gypsum board^[9], good air permeability of the centrifugal glass wool^[10] with their thickness being very small and large water vapor permeability resistance of the brick wall. The solar radiation being not rich^[11] in Shanghai can increase the temperature of the exterior wall, which will reduce the condensation risk. Thus, in order to simplify the model and consider the extreme condition, the moisture transfer and the effect of the solar radiation are ignored. In this article, the heat transfer mathematical model of the exterior wall of the capillary radiation heating room is established with centrifugal glass wool as insulation material and the applicability of the mathematical model is verified by comparing the experimental and theoretical values. Then, the dewing effect of different insulation board thickness changing between 1cm and 3cm is analyzed under different indoor and outdoor air states by comparing the calculated temperatures with the dew point temperatures of the indoor air, and the results under dynamic and low outdoor temperature can provide some references for the internal thermal insulation renovation of the old buildings.

2. HEAT TRANSFER MATHEMATICAL MODEL OF EXTERIOR WALL

In this model, some assumptions are made: (1) The heat transfer is one-dimension without taking the moisture transfer into account; (2) There is no heat source inside the wall and the effect of the moisture to the physical parameters and the temperature of the wall is ignored; (3) Ignore the effect of the solar and scattering radiation on the outdoor integrated temperature.

The internal insulation wall is assumed to be a compound structure made up of 5 kinds of materials with different thicknesses and properties. The wall is separated into m parts with the spacing being Δx and the time spacing is $\Delta \tau$ with the calculation beginning at $\tau=0$. The nodes in the finite difference equations are composed of four categories, the internal nodes, the combination nodes of two different materials, the inner and outer surface nodes. From the inside surface of the wall to the outside, i representing the position of the node changes between 1 and $m+1$ with k standing for the time of $k \cdot \Delta \tau$. Based on the explicit difference method, the internal node discrete equations are established by using the Taylor coefficient method with other nodes built up by heat balance method^[12].

The one-dimension unsteady heat conduction differential equation is written as:

$$\lambda \frac{\partial^2 t}{\partial x^2} = \rho c \frac{\partial t}{\partial \tau} \quad (1)$$

The outside boundary condition ignoring the solar radiation is obtained as follows:

$$-\lambda \frac{\partial \tau}{\partial x} = h_0(t - t_0) \quad (2)$$

The inside boundary condition with radiation and convection heat transfer is obtained as follows:

$$-\lambda \frac{\partial \tau}{\partial x} = h_e(t - t_{in}) \quad (3)$$

Where x is the space coordinate and τ is the time coordinate, t the node temperature ($^{\circ}\text{C}$), ρ the density of nodes (kg/m^3), c the specific heat capacity of nodes ($\text{J}/(\text{kg} \cdot \text{K})$), λ the thermal conductivity of nodes ($\text{W}/(\text{m} \cdot \text{K})$), t_0 the outdoor air temperature ($^{\circ}\text{C}$), t_{in} the indoor air temperature ($^{\circ}\text{C}$), h_e the integrated radiation and convection heat transfer coefficient ($\text{W}/(\text{m}^2 \cdot \text{K})$), h_0 the outside convection coefficient taken as $23.26 \text{ W}/(\text{m}^2 \cdot \text{K})$ ^[13].

2.1 Internal node discrete equation

The internal node discrete equations:

$$t_i^{k+1} = F_0(t_{i-1}^k + t_{i+1}^k) + (1 - 2F_0)t_i^k \quad (4)$$

Where $F_0 = \frac{\lambda \Delta \tau}{\rho c \Delta x^2}$, and $F_0 \leq 0.5$ for the steady results.

2.2 Outside boundary discrete equation

The outside boundary discrete equation:

$$t_{m+1}^{k+1} = 2F_0(t_m^k + B_i t_0^k) + (1 - 2B_i F_0 - 2F_0)t_{m+1}^k \quad (5)$$

Where $B_i = \frac{h_0 \Delta x}{\lambda}$, and the condition of $F_0 \leq \frac{1}{2B_i + 2}$

should be satisfied.

2.3 Inside boundary discrete equation

There is a side-wall capillary radiation surface near the external wall. As the radiation heat between the inside surface of the external wall and radiation surface accounts for a large proportion, the radiation heat between the inside surface of the external wall and other internal surfaces is ignored^[14].

$$q_{r1} = \sigma \varepsilon_f \varepsilon_i \psi_{1f} (T_f^4 - T_1^4) \quad (6)$$

Where q_{r1} is the radiation heat between the inside surface of the external wall and the radiation surface (W/m^2), σ the blackbody radiation constant being $5.67 \times 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$, ε_f the emissivity of the radiation surface, ε_i the emissivity of the inside surface of the external wall, ψ_{1f} the angular coefficient of the inside surface of the external wall to the radiation surface, T_f the temperature of the radiation surface (K), T_1 the temperature of the inside surface of the external wall (K).

The inside boundary discrete equation is written as:

$$t_1^{k+1} = 2F_0(t_2^k + B_i t_{in}^k) + (1 - 2B_i F_0 - 2F_0)t_1^k \quad (7)$$

Where $B_i = \frac{h_0 \Delta x}{\lambda}$, $h_e = h_{in} + h_r$, h_{in} is the indoor convection heat transfer coefficient taken as $8.72 \text{ W}/(\text{m}^2 \cdot \text{K})$ ^[13], $h_r = \frac{\sigma \varepsilon_i \varepsilon_f \psi_{1f} (T_f^4 - T_i^4)}{t_{in} - t_1}$ the radiation heat transfer coefficient ($\text{W}/(\text{m}^2 \cdot \text{K})$), and the following relation should be met: $F_0 \leq \frac{1}{2B_i + 2}$.

2.4 Discrete equation of combination node

Ignoring the thermal resistance between the layers, the corresponding node discrete equation is written as:

$$(1 + D)t_n^{k+1} = 2F_0 t_{n-1}^k + 2F_{0,i+1} + (1 - 2F_{0,i} + D - 2F_{0,i+1} D)t_n^k \quad (8)$$

Where $F_{0,i} = \frac{\lambda_i \Delta \tau}{\rho_i C_i \Delta x^2}$, $F_{0,i+1} = \frac{\lambda_{i+1} \Delta \tau}{\rho_{i+1} C_{i+1} \Delta x^2}$, $D = \frac{\rho_{i+1} C_{i+1}}{\rho_i C_i}$,

λ_i , ρ_i and c_i are respectively the thermal conductivity, density and specific heat capacity of the material closing to the inside; $\lambda_{(i+1)}$, $\rho_{(i+1)}$ and $c_{(i+1)}$ are corresponding parameters closing to the outside.

2.5 Outdoor dynamic temperature

The outdoor temperature in the heating period has large fluctuations with a certain regularity. For analyzing the extreme condition, take out the highest and lowest temperatures of the day with the minimum average temperature in the typical meteorological year^[15] of Shanghai and simplify it into a sine function:

$$t_w = -2 + 2.6 \sin \left[\frac{\pi}{12} (t - 8) \right] \quad (9)$$

Where t_w is the outdoor temperature(°C), t the time(h).

3. EXPERIMENT AND DATA ANALYSIS

3.1 Subject of the experiment

In this paper, a middle-tier room of an office building in Shanghai (3.9mx3.8mx3.8m) is taken as a test object with one exterior wall owing one window facing the North, and the adjacent rooms of other internal surfaces are conditioned. There is a radiation heating surface next to the exterior wall with centrifugal glass wool as the insulation material. The compound wall structure is shown in Figure 1.

As the centrifugal glass wool with low thermal insulation and the thermal conductivity unknown is not purchased from a formal manufacturer, a heat flow meter with its accuracy within $\pm 5\%$ of LSI-LASTEM is

used to measure the thermal conductivity by using the enthalpy difference lab to create the temperature difference of the two sides of the insulation board. The result is gotten after conversion and the layout of the measured points is shown in Figure 2.

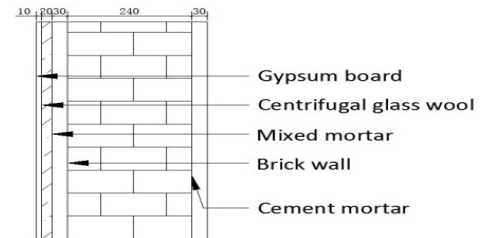


Figure 1 the schematic diagram of the exterior wall

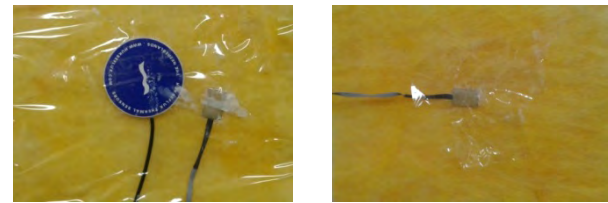


Figure2 the layout on the different temperature sides

The physical parameters of the wall (from indoor to outdoor) are shown in table 1.

Table 1 the physical parameters of the exterior wall^[16]

The envelope structure	Thickness (m)	Density (kg/m ³)	Thermal conductivity (W/m·K)	Specific heat capacity (J/kg·K)	Outdoor convective heat transfer coefficient(W/m ² ·K)	Indoor convective heat transfer coefficient(W/m ² ·K)
Gypsum board	0.01	950	0.33	840		
Centrifugal glass wool	0.02	100	0.072	750		
Mixed mortar	0.03	1700	0.87	840	23.26	8.72
Brick wall	0.24	1620	0.62 ^[8]	1050		
Cement mortar	0.03	1800	0.93	840		

3.2 The process of experiment

The experiment was done in a period of time (3/2/2013, 9:43 ~ 3/3, 9:32) with low outdoor temperature. Nine T-type thermocouples with the accuracy $\pm 1.1^\circ\text{C}$ are mounted on the radiant Panel, the inside and outside surfaces of the exterior wall, the interface of the insulation board and the wall surface and the indoor and outdoor space. Among them, four (b1, b2, b3, b4) are mounted on the interface and two (a1, a2) are on the internal surface of the wall along the window from top to bottom. The data are collected by Agilent 34970A data acquisition instrument with the time interval being 5min. The graphs are shown in Figure 3.

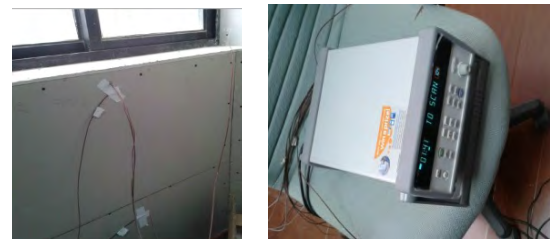


Figure 3 the corresponding graphs of experiment

3.2. The analysis of experimental data

In Figure 4, the temperature changing trend of indoor and the internal surface is the same to the radiant panel. The measuring points of the internal surface and the interface are influenced by the outdoor temperature with the a2, b3 and b4 having the least influence.

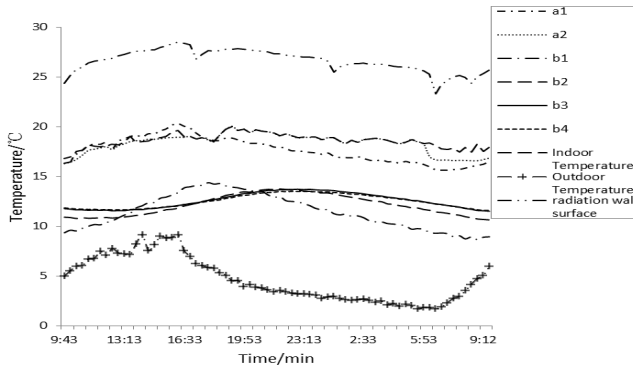


Figure 4 the temperature curves of measuring points

4. CALCULATION RESULTS AND COMPARISON

The node temperatures of the exterior wall calculated by the steady mathematical model are taken as the initial conditions with the radiant panel, indoor and outdoor temperatures being known conditions. The emissivity of surfaces is 0.92 and the angular coefficient is 0.18. Figure 5 shows the temperature comparison of the experimental values a1, b4 and the corresponding theoretical values a2', b4'.

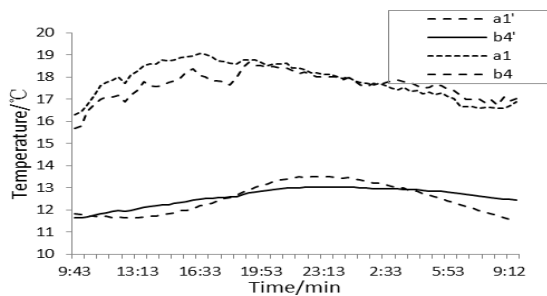


Figure 5 the theoretical and experimental values

In Figure 5, the values of the internal surface temperature are very close with the maximum deviation being 1.1°C maybe due to the poor sealing of the experiment measuring point and the average deviation being 0.24°C. The maximum temperature deviation of the interface is 0.87°C with the average deviation being 0.1°C. In General, the good consistency of the experimental and theoretical values verifies the applicability of the mathematical model.

5. ANALYSIS OF THE INSULATION BOARD THICKNESS

The indoor calculation parameters of comfort air conditioning in winter are defined with the temperature between 18°C and 24°C and the relative humidity between 30% and 60%^[17]. The interface temperature is calculated with the indoor air being 18°C, 20°C, 22°C, the relative humidity 30%, 40%, 50%, 60% and the insulation board thickness between 1cm and 3 cm under the standard centrifugal glass wool properties of its density 48kg/m³ and the thermal conductivity 0.0377W/(m·k). Then the dewing effect is analyzed by

comparing the interface temperature and dew point temperature of the indoor air. The calculation results are shown in Figure 6, Figure 7 and Figure 8.

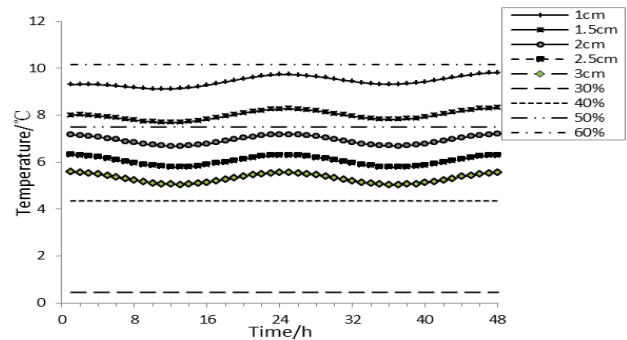


Figure 6 the interface temperature with indoor air 18°C

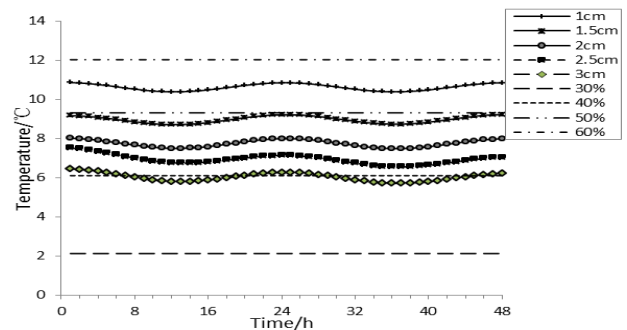


Figure 7 the interface temperature with indoor air 20°C

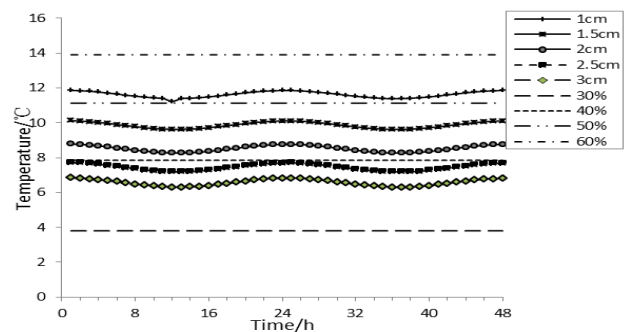


Figure 8 the interface temperature with indoor air 22°C

In the above figures, with the increasing of insulation board thickness, the interface temperature decreases, and the dewing will always occur in the case of relative humidity being 60% and never occur in the case of 30%. When the indoor temperature is 18 °c, the relative humidity of 40% is safe with the no dewing thickness for 50% below 1.5cm. When the indoor temperature is 20°C, the safe insulation board thickness for 40% is less than 3cm, while the thickness for 50% is less than 1.5cm. When the indoor temperature is 22°C, the safe insulation board thickness for 40% is below 2.5cm, while the thickness for 50% is not more than 1cm.

6. CONCLUSION

(1) The heat transfer mathematical model of the external wall of the old building with internal insulation is established and an experiment is done in a cold day in Shanghai. The applicability of the mathematical model is verified by comparing the experimental and theoretical values.

(2) Based on the mathematic model, the interface temperature is calculated with the extreme outdoor temperature, the indoor temperature being 18°C, 20°C, 22°C, the relative humidity 30%, 40%, 50%, 60% and the insulation board thickness between 1cm and 3 cm by using the MATLAB software. Then the effect of insulation board thickness on the dewing is analyzed by comparing the interface temperature and dew point temperature of the indoor air, which will provide some useful references to the internal insulation system design of old buildings.

(3) Because of the ignoring of the moisture transport and factors influencing the dewing being too many, the calculation results might have some diversion. In addition to the rainy days and the solar radiation, the external window also has influence on the dewing. Thus, the findings can only be applied to the assumptions in this article and other factors for the dewing of the internal insulation will be discussed in further study.

7. ACKNOWLEDGEMENTS

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EXPERIMENTAL STUDY ON THE INFLUENCE OF ENERGY USE INFORMATION ON OCCUPANT BEHAVIOR

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ABSTRACT

The energy consumption of air-conditioning system and lighting systems is mainly decided by the room occupants' behavior. Part-space and part-time operations of air-conditioning and lighting systems will consume obviously less energy than that of full-space and 24-hour operations. Therefore, changing the occupants' behaviors on using air-conditioning and lighting systems will potentially save building energy. This paper studies the influence tendency of energy use awareness to the occupants' behaviors of using air-conditioning and lighting systems through experiments. Furthermore, the impact of energy use awareness on building energy saving is summarized. Experimental test bed was constructed in an ordinary office room with the floor area of 20 m². Indoor air temperature, humidity, illuminance and CO₂ concentration were controlled by air-conditioning system, lighting system and ventilation system. A touch panel human-machine-interface (HMI) was set to let experimental subjects adjust the indoor environment by inputting their perceptions when they felt uncomfortable. Totally, 24 subjects took part in the experiments with each subject conducting the experiments for 3 days. About 25% of subjects cancelled the perception inputs when they were prompted with the energy use information. Among all the perception inputs that caused increases of energy use, 11% of them were cancelled when the subjects were informed with energy use information. Meanwhile, the reason why the subjects did not cancel the perception inputs was investigated through questionnaires. The investigation showed that two thirds of the subjects did not cancel their perception inputs because they cared more for comfort rather than energy use. The experimental results showed that the energy use information awareness could contribute to building energy saving by triggering the room occupants to reduce their energy inefficient behaviors.

Keywords: Building energy saving; Energy using behavior; Energy use awareness; Experimental study; Indoor environment control.

1. INTRODUCTION

The energy consumption of air-conditioning system and lighting system accounts for about two thirds of the total building energy consumption. Therefore, reducing the energy consumption of the air conditioning system and lighting system is of great significance to the building energy efficiency. The energy consumption of air-conditioning system and lighting systems is mainly decided by the room occupants' operational modes. Therefore, changing the occupants' behaviors on using air-conditioning and lighting systems will potentially save building energy. This paper studies the influence tendency of energy use awareness to the occupants' behaviors of using air-conditioning and lighting systems through experiments.

A lot of research was conducted to study impact factors of room occupants' energy use behaviors. Peffer et al. [1] reviewed the situation of homeowners using thermostats and it showed that 89% of homeowners list energy efficiency as very important. The review also showed that energy consumption feedback could change energy consumption behavior and help users to

understand the connection between HVAC use, cost and the environment. Rishee et al. [2] studied the relation between eco-feedback interface usage and energy use changes in a residential building. The study revealed that historical comparison and incentives were the key components to drive higher engagement and reduction in energy consumption. Liu et al. [3] found that natural adaptive behaviors such as adding/reducing clothes could reduce the use of energy, but when environmental parameters could not fully satisfy occupants' comfort requirements, negative behaviors could conflict with energy saving. How to use energy consumption feedback to change energy consumption behavior has been widely studied as well [4-12]. However, most research was focused on the influence of energy feedback on the household electricity consumption. It was still unclear how much the energy use information could influence the energy using behaviors of office room occupants.

Therefore, a series of experiments were conducted to investigate the influence of energy use awareness to the office room occupants' behaviors of using air-conditioning and lighting systems. And the impact of

energy use awareness on the building energy was summarized.

2. EXPERIMENTS

The experiment methodology [13] is to let a subject sit in an ordinary office room and do his/her daily work. When the subject feels any discomfort in the indoor thermal, light, acoustic environment, or air quality, such as hot, cold, humid, dry, glare, dark, noisy, stuffy and etc., the subject can express his/her perceptions through a touch panel human-machine-interface (HMI). The control system will tune air-conditioning system or lighting system accordingly and meanwhile the HMI will inform the subject the estimated energy increase for the purpose of eliminating the discomfort perceptions. The subject can decide to cancel the discomfort input to prevent the energy use increase or confirm the discomfort input even if the energy use will increase. This is the method to check how much the energy use information awareness will influence the energy use behaviors of office room occupants.

2.1 Experimental test bed

An experimental test bed was constructed for conducting the experiments. As shown in Figure 1, the experimental test bed included an ordinary office room with the floor area of 20 m². The room was equipped with an air-conditioning system to control temperature, humidity, and indoor air quality, a dimmable lighting system to control artificial lighting, as well as a blind with position and blade angle tunable to control day lighting. A set of sensors were set to measure indoor environmental parameters including dry-bulb temperature, radiant temperature, humidity, air speed, illuminance, CO₂ concentration, and sound level. Outdoor air temperature and humidity were measured as well.

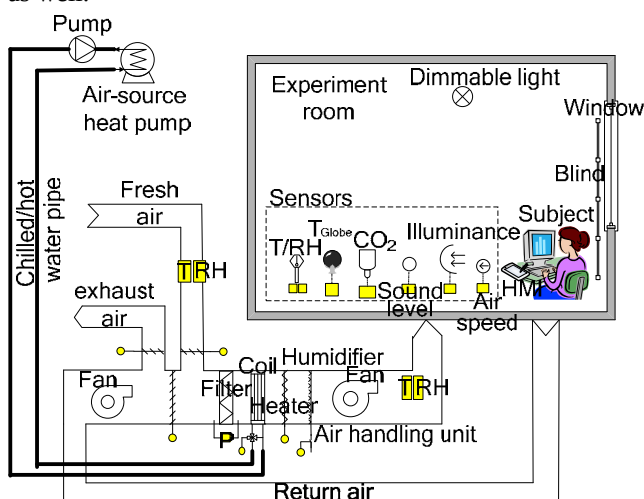


Figure 1. Experimental facility: an office room, environment control devices and sensors

2.2 Human machine interface

An HMI, as shown in Figure 2, was prepared for subjects to input their discomfort perceptions, including nine perceptions of “hot”, “cold”, “dry”, “humid”, “glare”, “dark”, “noisy”, “draft”, and “stuffy”.



Figure 2. Human machine interface

2.3 Experiment subject recruitment

Cattell's 16 personality factors questionnaire was used to select the subjects for this experiment [14]. In total, 26 subjects were selected to participate in the experiment. Twenty-two of the subjects were college students. Four of the subjects were clerks of a research institute. The length of an experimental period was four consecutive hours, either in the morning, i.e., 8:00~12:00 or in the afternoon, 13:00~17:00. A 4-hour experimental period was called an experiment unit. And each subject was planned to conduct the experiments for 6 experiment units.

2.4 Experiment process

When subjects entered the room, they worked or studied as usual. They inputted their perceptions via the HMI when they felt uncomfortable. The automatic control system tuned related devices to change the environment and estimates energy consumption changes. If the energy consumption increased because of occupants' inputs, the HMI would pop-up a dialog box, as shown in Figure 3, to inform the subjects the energy increase and let subjects decide to cancel or confirm the discomfort inputs.

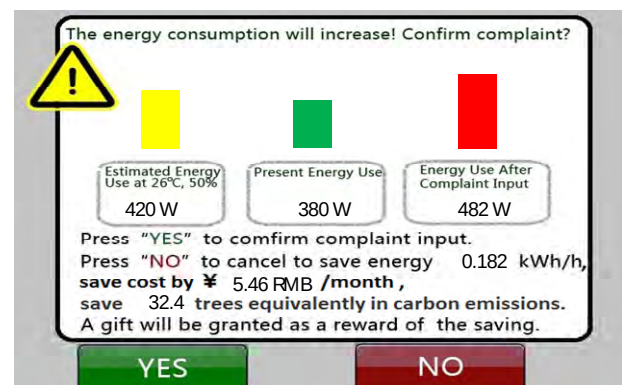


Figure 3. Energy information awareness interface

All the environmental and discomfort input data were recorded to a database in a control computer. During the experiment, subjects were required to record their detailed perceptions when they inputted perceptions through the HMI. After they finished an experiment unit, the subjects were required to fill out a questionnaire to explain the reasons why they canceled or confirmed the discomfort inputs.

3. EXPERIMENTAL RESULTS

Two stages of experiments were conducted. During the first stage, the subjects were informed only with the energy consumption information when their input led to the increase of energy consumption. And besides of the energy consumption information, the subjects were offered a pencil as a reward if their input improved the energy consumption in the second stage of experiment. The purpose of this reward was to encourage more cancellation. The value of the reward was decided according to the average energy cost saved by the cancellation.

3.1 Summary of experiment results

Table 1 summarizes the experiment results. Total discomfort inputs include all the nine discomfort perceptions. Among the nine discomfort inputs, only some of them may cause energy use increase. For example, for cooling operation, “hot” input may increase energy use while “cold” input may not. Only the discomfort inputs that cause energy consumption increases will trigger a pop up dialog box on the HMI to let subjects choose whether to cancel the input or not. The cancellation rate is calculated by dividing the number of cancelled inputs by the total inputs that will cause energy consumption increases. The cancellation rates of with and without reward are similar, roughly 10%, which implies that 10% of room occupants behaviors of tuning indoor environment can be changed by the energy information awareness. Meanwhile, the similarity between the cancellation rates of with and without reward shows that the reward of a pencil cannot induce more cancellations.

Table 1 Summary of the experiment results

	Energy info. prompt only	Energy info. prompt + reward for cancellation
Experimental period	2012.06.04 ~ 2012.08.06	2012.08.17 ~ 2012.10.15
Experiment units	64	68
Number of subjects	15	16
Total discomfort inputs	225	180
Discomfort inputs causing energy increase	94	54
Cancelled inputs	11	5
Cancellation rate	11.70%	9.26%

3.2 Input cancellation vs. discomfort type and energy increase

The cancellation number and cancellation rate of discomfort inputs vs. discomfort type and energy increases are shown in Figure 4. From the figure it can be found that the cancellation rate does not increase along with the amount of energy use increase. However, the cancellation rate correlates to discomfort type. The discomfort inputs of dark and hot show higher cancellation rate than the other types of discomfort inputs. This implies that the dark and thermal discomfort inputs tend to be easily canceled, which shows that people can tolerate the dark and thermal discomfort to some extent and there is energy saving potential.

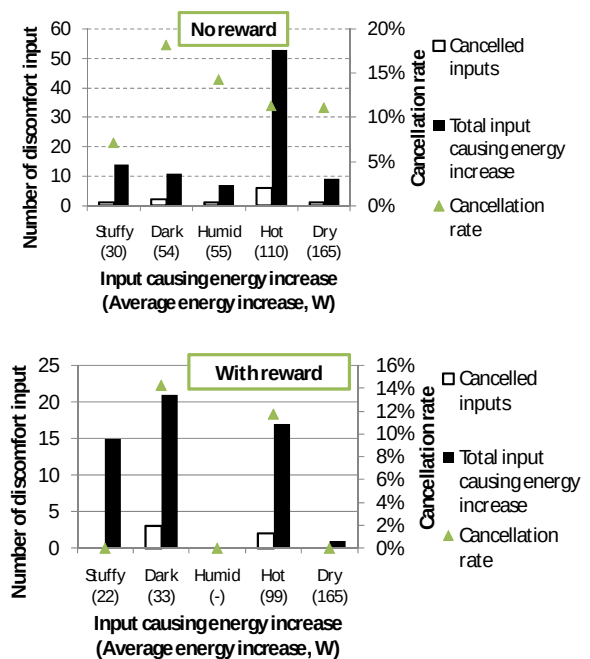


Figure 4. Input cancellations vs. discomfort type and energy increase

3.3 Thermal discomfort input cancellation vs. indoor air temperature

Considering that the thermal perceptions closely relate to indoor air temperature, the correlations between the thermal discomfort inputs and the indoor air temperature are investigated. As shown in Figure 5, for the air temperature ranging from 26-28°C, which is the common comfort range for most of Chinese people, the cancellation rate is relatively large. Regarding the high cancellation rate of 31-32°C, it is the data from one specific subject. The interview after the experiment shows that the subject does feel acceptable for the temperature of 31°C. While for the temperatures far away from thermal comfort range, the cancellation rates tend to be small or even zero. This tendency implies that when the temperature is not far away from thermal comfort range, it is acceptable for most people to tolerate the discomfort thermal environment for the

sake of energy saving. This shows a potential for using energy information awareness to achieve energy saving.

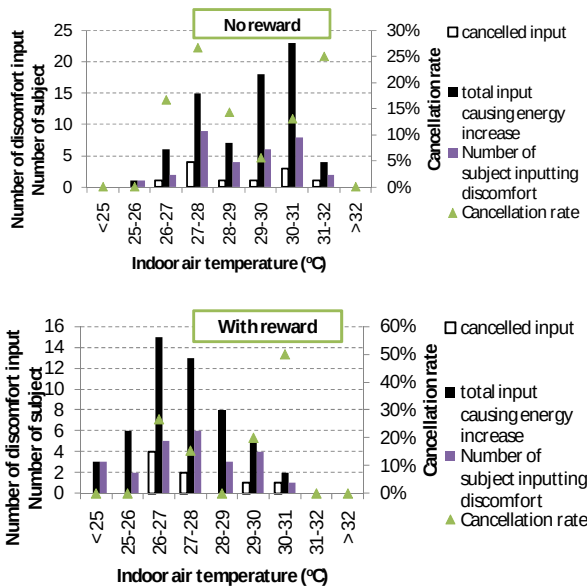


Figure 5. Thermal discomfort input and cancellation vs. indoor air temperature

3.4 Individual diversity of discomfort input and cancellation

Figure 6 summarizes the number of total discomfort inputs and cancelled inputs of every individual subject. It can be seen from the figure that both discomfort input times and cancellation times diverge largely.

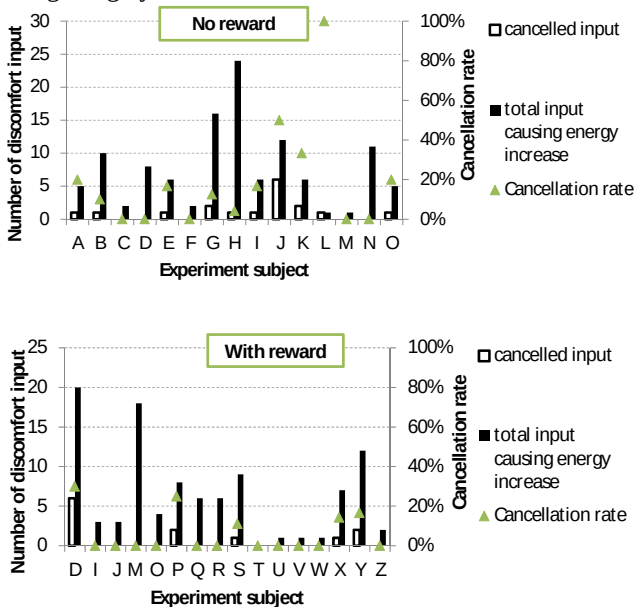


Figure 6. Individual diversity of cancellation

If take a look at subject D, I, J, M, and O who participated the experiments of both with and without reward, we can see that cancellations with reward are not more than that without reward. This implies for the same individual, a reward with the value of energy cost

saving in one-day magnitude are not attractable enough to make office room occupants to change their energy use behaviors. This trend can be seen as well statistically by comparing the two-stage experiment results.

4. POST-EXPERIMENT QUESTIONNAIRE

For the purpose of clarifying the reasons why a subject did not cancel his/her discomfort input and how much energy cost saving can trigger a cancellation, post-experiment questionnaire was conducted. Figure 7 and Figure 8 show the answers for the two questions, respectively.

Regarding the reasons why subjects did not cancel a discomfort input, 63.6% of the subjects selected “care more about comfort”. This result can be reflected by the correlations between cancellation rate and indoor air temperature as well, i.e., the farther away from comfort range, the less cancellation. About 27% of the subjects selected “not understand the amount of energy increase”, which means that the energy information displayed on the HMI needs to be expressed using an easy-to-understand way for the people without any ideas of energy consumption magnitude. About one tenth of subjects did not cancel because they did not pay for the energy cost. This is common thinking for office room occupants. This issue has to be considered when inducing the energy saving behaviors for office room occupants.

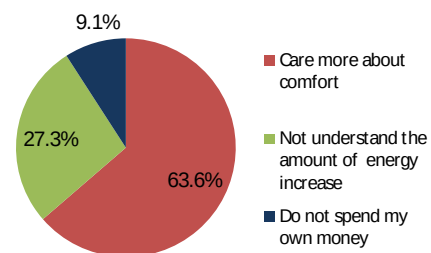


Figure 7. Reasons why subjects did not cancel a discomfort input

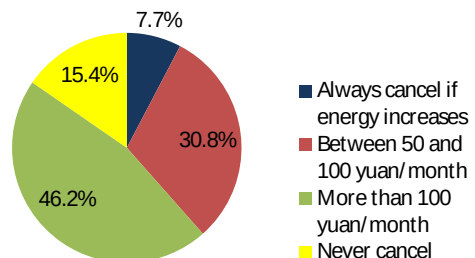


Figure 8. Energy cost saving that can trigger a cancellation of the discomfort input

Regarding how much energy cost saving can trigger a cancellation, most subjects (46.2%) selected “more than 100 yuan/month”, which is more than the monthly total air-conditioning energy cost of an office room with the floor area of 20 m². This shows that most

people do not know the magnitude of energy cost for indoor environment control. It should be noted that about 15% of occupants never canceled a discomfort input no matter how much energy was consumed. This shows that education is needed to induce energy efficient behavior.

5. CONCLUSION

This paper describes the experimental study on the energy saving effect achieved by prompting room occupants with energy consumption information. The results show that: 1) about 10% of the occupants will cancel a perception input when they are informed that the input will cause energy use increase; 2) people tend to cancel a perception input when the indoor environment is not far away from comfort range; 3) the cancellation shows a large diversity among the experiment subjects. The post-experiment questionnaire shows that 2/3 of the subjects did not cancel the perception inputs because they cared more about comfort than energy use.

ACKNOWLEDGEMENTS

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THE DISCUSSION OF EVAPORATIVE COOLING WATER CHILLER AND WATER STORAGE COMBINATION AIR CONDITIONING SYSTEM

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ABSTRACT

Brief overview the energy saving effect of evaporative cooling water chillers operation in certain weather conditions, and the energy saving significance of water storage technology application in the field of air conditioning. For evaporative cooling chiller combined with chilled water thermal storage air conditioning system forms and operation mode are introduced, In XinJiang region, From every night at 23 o'clock to eight o'clock in the morning, Statistical and analysis the evaporative cooling of water chiller's water temperature hours, It is concluded the feasibility that application of evaporative cooling water chiller and water storage technology combined of air conditioning system, Analysis of the energy saving effect. To provide certain theoretical reference for related design personnel using evaporative cooling water chillers and chilled water thermal storage's air conditioning system combined in the building.

Keywords: Evaporative cooling; chiller; Water storage; Energy saving

1. INTRODUCTION

Along with our country residents living standard has increased, people's requirement of indoor comfortable environment become more and more high, which makes the air conditioning has been widely used in urban construction to create a good indoor thermal comfort environment^[1-2]. But the quality of the indoor environment, make the air conditioning system energy consumption is higher and higher. According to statistics, China's building energy consumption accounts for 30%-40% of total energy consumption, the air conditioning energy consumption accounts for about 70% of the total energy consumption, air conditioning system is not only reflected on the energy consumption, at the same time, residents of power consumption is due to the operation of the air conditioning system gradually rise, especially in the peak and trough ratio imbalance. In recent years, under the call of the national energy conservation and emission reduction policy, at the end of 2009, the national total installed electricity capacity of 874 million kW, Basic power supply and demand are balance, energy conservation and emissions reduction work has achieved some results^[3]. Therefore, To deal with the problem of air conditioning system energy consumption and utilization have certain necessity and urgency. As evaporative cooling technology and storage technology of mature gradually, There are some relevant scholars put forward the concept of evaporative cooling chiller, Using evaporative cooling technology,

as a cold source of air conditioning system, It can provide air conditioning system with 16-18 °C high temperature cold, Wet and dry bulb temperature difference greater at night, wet bulb temperature is relatively low under the environment, So the evaporative cooling chiller water temperature will be lower, combined with cool storage technology, taking advantage of the peak cut across and difference of peak valley price policy, to alleviate the contradiction of electricity utilization, as well as balance and decrease the air conditioning system energy consumption has played an important role, It can't be ignored. This article will combined with evaporative cooling chiller and chilled water thermal storage technology, design of evaporative cooling chiller combined with chilled water thermal storage air conditioning system, providing some reference for related design research..

2. EVAPORATIVE TECHNOLOGY AND COLD STORAGE TECHNOLOGY

2.1 The combination advantage of Evaporative cooling water chiller and chilled water thermal storage technology

Evaporative cooling air conditioning technology is a kind air conditioning technology of energy conservation, environmental protection and sustainable development, But its application still has some limitations at present^[1], the main reason is limited by

outdoor weather conditions, outdoor wet and dry bulb temperature difference is large, cooling effect is better, Conversely, cooling effect is limited. therefore, in the night, the wet and dry bulb temperature difference is big, and the wet bulb temperature is low, low electricity prices and electricity trough of national policy, For evaporative cooling chiller night cool storage provides a good condition, we can use evaporative cooling chiller for low water temperature during the night, with reservoir storage, at the same time can take advantage of the evaporation cooling air handling units for building internal ventilation cooling at night, take away the heat inside the room, reduce the heating load, cold storage with the building itself, and can play a certain energy-saving effect^[4]. From this analysis, the evaporative cooling technology and chilled water thermal storage technology combined, It can play a role in energy saving and consumption reduction of air conditioning system.

The local source use Evaporative cooling chiller and application mechanical refrigeration composite water chillers , during the night, evaporative cooling chiller paly a dominant role associated with mechanical refrigeration , and through evaporative cooling air handling unit ventilation storage at night, using cold water tank and cooling during the day. Below is a brief overview of preparing high temperature cold water principle and composite water chiller profile structure..

2.1.1 Evaporative cooling chiller for making high temperature cold water

As shown in figure 1 is evaporative cooling chiller for making cold water of the principle of enthalpy wet figure. Through indirect evaporation cooling outdoor air (precooling process $W_x - W_{x1}$), then through the direct evaporative cooling air and heat exchange with rain water($W_{x1} - O$), Finally it will get the high temperature of sub-wet bulb temperature (O)..

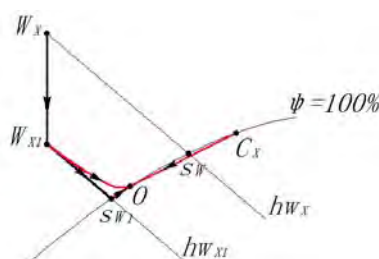


Figure 1 evaporative cooling chiller for making cold water

According the sub-wet bulb efficiency calculation formula estimated indirect and direct evaporative cooling chiller water temperature T_o ^[5-6] .

$$\eta = \frac{T_{w_x} - T_o}{T_{w_x} - T_{sw}} \times 100\% \Rightarrow T_o = (1 - \eta)T_{w_x} + \eta \cdot T_{sw}$$

2.1.2 The introduction of Evaporative cooling and mechanical refrigeration composite high temperature water chiller

As shown in figure 2, For evaporative cooling and mechanical refrigeration compound high-temperature chiller (patent number: CN 101788174 A), the unit for evaporative cooling chiller composite of evaporative cooling and mechanical refrigeration chiller, adopting two stage air cooler outdoor air precooling , precooling air happen heat and humidity exchange by water spray with extender, End up with high temperature cold water. Mechanical refrigeration part adopts air-cooled heat pump, using evaporative cooling chiller ventilation to cool the condenser, improving the efficiency of the condensing air cooled heat pump, and further improve the efficiency of air-cooled heat pump ^[7-9].

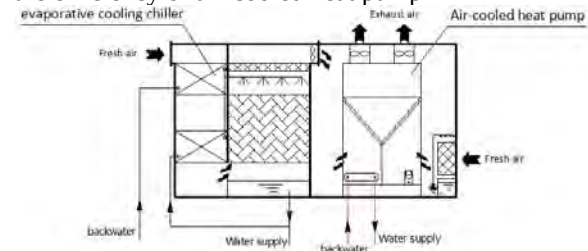


Figure 2 evaporative cooling and mechanical refrigeration compound high-temperature chiller

2.2 The introduction of chilled water thermal storage technology

Chilled water thermal storage technology is the use of electricity at night , trough Run air conditioning cold source, the production of cold water storage tank and fire pool space, during the day, by making use of the water supply pump in the water tank supply cold water to the air terminal, undertake indoor load^[10]. The technology has the advantages of low initial investment, the system is simple, reliable, easy maintenance, low technical level request ^[11], and the capacity of the refrigeration system can be choose according to the daily average load, and used in existing project of energy-saving renovation project, the use of fire pool as a cool storage pool for storage, reducing initial investment. Studies show that when need cold quantity greater than 7000 kW · h (6.03 million kcal), or cool storage volume is more than 760 m³ after the most economic and reasonable. At present, there are 113 of water storage air conditioning engineering, with a dozen independent patent technology, work load cooling temperature from 5 °C to 10 °C, made the storage density of 5.8 kW/m³ (5000 kcal/m³)increased to 11.6 kW/m³(10000 kcal/m³) or more, It has a very

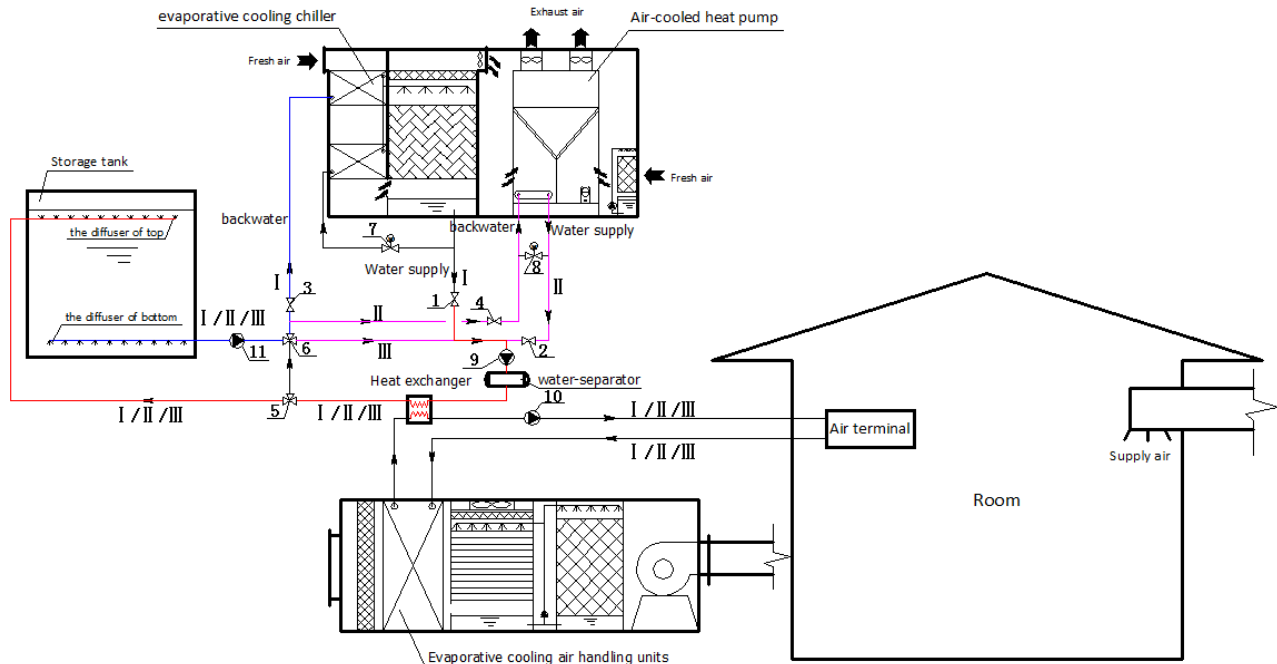
good promotional value^[3]. At night the temperature is reduced, the refrigeration efficiency will be accordingly increased by 6% to 8%, greatly increasing system at run time, It make the air conditioning system total power saving rate of 10% to 22%.

3. EVAPORATIVE COOLING CHILLER AND CHILLED WATER THERMAL

STORAGE COMBINATION AIR CONDITIONING SYSTEM

3.1 Principle of air conditioning system

As shown in figure 3, evaporative cooling of water chiller and chilled water thermal storage technology combined air conditioning system schematic diagram.



1, the valve; 2, valve; 3, valve; 4, valves; 5, the 3-way valve; 6, the 3-way valve; 7, electric valve; 8, electric valve; 9, cold water pumps; 10, cold water pumps; 11, storage pumps

Figure 3 evaporative cooling chiller and chilled water thermal storage combination air conditioning system schematic diagram

The evaporative air conditioning system consists of evaporative cooling and air cooling heat pump water chiller, evaporative cooling air handling units, storage tank, heat exchanger, water segregator, indoor sensible heat, air supply pipe and room.

Among them, the cold source for evaporative cooling and air-cooled heat pump compound patent (patent number: CN 101788174 A), through high temperature air cooler and low temperature air cooler for outdoor air precooling and Heat exchange with filling water in extender, reduce the water temperature; Mechanical refrigeration section uses the air-cooled heat pump units, condensing end into the side of the wind using evaporative cooling ventilation to cool the air cooled heat pump condensing coil pipe, on the other side into the style with a wet curtain to precooling of outdoor air, improve the efficiency of air-cooled heat pump cooling, thus enhancing the COP^[7]; Evaporative cooling air unit consists of the filter section, air cooler, tube type indirect evaporative cooling, direct evaporative cooling section and Supply section; Cool storage tank according to construction characteristics using fire pool or custom storage tank,

the water is used in the change of the density under different temperature, in the water storage tank of natural stratification, in the water storage tank at the top and bottom with diffuser, reduce flow disturbance and damage to the water layer^[11].

As shown in figure 3, the valve 1 and 3 control evaporation cooling chiller for backwater, the valve 2 and 4 control air-cooled heat pump for water, the three-way valve 6 control backwater of evaporative cooling chiller and air-cooled heat pump and ice storage tank of the cooling waterway, the three-way valve 5 control night cool storage air conditioning system and the cooling function of the switch during the day, the valve 7 and 8 respectively control evaporation cooling chiller and air-cooled heat pump of water supply water supply the by-pass pipe; Of cool storage air conditioning system/cooling process by 9 and 11, cold water pumps and cooling through the cooling water pumps at the end of 10. Waterway I, II, III represent the evaporative cooling chiller/cooling, air cooling heat storage cold storage and cooling, storage tank cool storage/cooling three waterways. Combined air conditioning system

schematic diagram analyse the running of the air conditioning system.

3.2 The operation mode of air conditioning system

The air conditioning system using evaporative cooling chiller and air-cooled heat pump for joint operation of cool storage/cooling, it can be divided into evaporation cooling chiller and cold storage at the night, evaporative cooling chiller and air-cooled heat pump units in combination with the night cool storage two modes; Cooling way according to load characteristics can choose cold storage water tank to provide cooling capacity, evaporation cooling chiller provide cooling capacity and evaporation cooling chiller with mechanical refrigeration chiller provide cooling capacity operation three models.

3.2.1 Storage process

1) Evaporative cooling chiller cold storage at night

As shown in figure 3, evaporative cooling water chiller and water storage air conditioning air conditioning system schematic diagram shown in waterway I, at night, when electricity trough, open evaporative cooling of water chiller refrigeration, close the valve 2 and 4, at this time by evaporative cooling cold water unit of cold water through the waterway I through valve 1 by the water pump to enter the water segregator, through manifold into the storage tank at the top of the diffuser in the; The storage of higher temperature at the bottom of the water flow through the powder at the bottom of the device, by storage pumps 11 after a three-way valve 3 back to evaporative cooling chiller.

Because night outdoor wet and dry bulb temperature difference is big, and wet bulb temperature is low, the evaporation cooling air handling units of tube type indirect evaporative cooling and direct evaporative cooling, indoor air supply for building, take away the heat of indoor room, the use of building itself cool storage capacity for ventilation cool storage.

2) Evaporative cooling chiller with mechanical refrigeration storage at night

As shown in figure 3, evaporative cooling water chiller and water storage air conditioning air conditioning system, as shown in the principle of waterway II, electricity trough during the night, open air cooling heat pump refrigeration, close the valve 1 and 3, at this time by air cooling heat pump from the cold water through the waterway II through valve 2 from cold water pump to the water separator, after manifold into the diffuser at the top of the storage tank; Storage department at higher temperature at the bottom of water flow through the powder at the bottom of the device, by storage pumps 11 after a three-way valve 3 back to air cooling heat pump evaporator side. Run the

evaporative cooling air handling units for building ventilation storage.

3.2.2 The cooling process

1) Storage tank supply cold energy alone process

As shown in figure 3, evaporative cooling water chiller and water storage air conditioning air conditioning system schematic diagram shown in waterway III, after night electricity trough cold source of cold storage, cold storage tank have accumulate a certain quantity, when close the valve 1, 2, 3 and 4, storage water tank in the cold storage of low temperature in the cold water through the waterway III entered the water segregator, by the water pump 9 into heat exchangers, and air conditioning water system for heat transfer, heat transfer temperature rise after the return to storage tank water through the three-way valve 5; Indoor water supply by end of the cold water pump 10 to indoor sensible heat and water through evaporation cooling air handling units after the outdoor air precooling, back to the heat exchanger in a heat cold water, complete the system cycle.

2) Evaporative cooling chiller supply cold energy process

Storage tank cold quantity release, building indoor still need cooling, at this time, In outdoor weather conditions allow conditions, open evaporative cooling for cooling water chillers.

As shown in figure 3, evaporative cooling water chiller and water storage air conditioning air conditioning system schematic diagram shown in, close the valve 2 and 4, Evaporative cooling cold water unit of cold water through the waterway I via cold water pump 9 into the water-separator, by manifold into to the heat exchanger, water and the air conditioning system for heat transfer, heat transfer temperature rise after the water back in evaporative cooling water chillers; Indoor water supply by Cooling water pump 10 to end indoor sensible heat, backwater through the evaporative cooling air handling units precooling the outdoor air, back to the heat exchanger in a heat cold water, complete the system cycle.

3) Evaporative cooling chiller and air-cooled heat pump combined supply cold energy process

Storage tank cold quantity finished release, local outdoor weather conditions don't meet the evaporative cooling chiller cooling separately, we can enable air-cooled heat pump and evaporation cooling chiller combined operation form for air conditioning system.

As shown in figure 3, evaporative cooling water chiller and water storage air conditioning air conditioning system schematic diagram, as shown in evaporative cooling after cold water chiller and air-cooled heat pump manufacture cold water, After cold water pump 9 mixed in water segregator for high temperature and cold water, the high temperature and cold water through the segregator to the heat exchanger,

with the backwater of the air conditioning system transfer sensible heat, the water of heat transfer temperature rise back in evaporative cooling chiller and air-cooled heat pump respectively; through the cooling water pump 10 supply the water to the indoor terminal, the backwater return to the evaporative cooling air handling units precooling the outdoor air, then Back to the heat exchanger in heat with cold water, complete the system cycle.

4. AIR CONDITIONING SYSTEM OPERATION ANALYSIS OF URUMQI IN XINJIANG REGION

4.1 Analysis of evaporative cooling chiller water temperature

Air conditioning system design in the Xinjiang Urumqi area, according to the evaporative cooling cold water unit of cold source form, two levels of air cooler and direct evaporative cooling section, the way it deals with air as shown in figure 1. Take the indirect - direct evaporative cooling chiller efficiency of sub-wet ball can choose 120%^[5], according to the literature [12], The outdoor meteorological conditions in this region from May to September 23:00 to 08:00 in the morning ,and in the time interval computing evaporative cooling water temperature of the chiller T_o , add up the hours.

Chilled water thermal storage of cold source of cold source generally uses mechanical refrigeration, cold storage temperature generally uses 6 to 10 °C, cold storage temperature is usually 4 to 6 °C^[13], but according to the evaporation cooling air conditioning system of 16 to 18 °C high temperature cold water of the air terminal and consideration of heat exchanger in heat efficiency, we can relax appropriately cold storage temperature, in this paper, cold storage temperature setting is 0 to 12 °C, expanding the scope of the cold storage temperature will cause a certain volume of chilled water thermal storage equipment to reduce cooling capacity, but cold source using evaporative cooling chiller at night big wet and dry bulb

temperature, wet bulb temperature low, under the condition of less energy consumption of evaporative cooling chiller parts, at the same time, low energy consumption, it also can meet the requirements of cold storage, compared with mechanical cold source has a certain meaning of energy-saving.

The division of evaporative cooling chiller water temperature: the $T_o \leq 0$ °C, 0 °C $< T_o \leq 12$ °C and 12 °C $< T_o \leq 18$ °C, $T_o > 18$ °C, which considering the $T_o \leq 0$ °C range part of the actual water temperature won't be negative, so will the $T_o \leq 0$ °C hours interval calculation out of the water temperature in the whole calculation out. At May to September, night 23:00 to 8:00 in the morning out of the water temperature as shown in figure 4, according the water temperature on regional water temperature hours statistics, statistical results as shown in figure 5 and table 1, 0 °C $< T_o \leq 12$ °C hours within the temperature range of the percentage and the total time is shown in figure 6 and table 2.

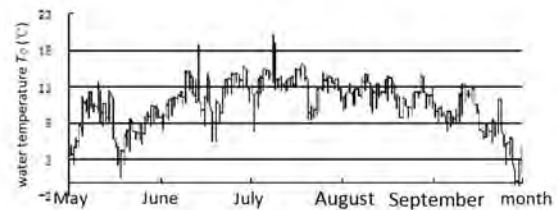


Figure 4 evaporative cooling chiller 5-9 months out of the water temperature

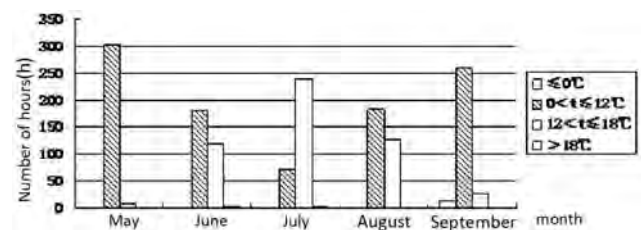


Figure 5 Evaporative cooling chiller May to September, 23:00 to 8:00 o'clock different water temperature range statistics.

Table 1 Evaporative cooling chiller May to September, 23:00 to 8:00 o'clock different water temperature range statistics

The temperature T_o (°C)	May	June	July	August	September
$T_o \leq 0$	0	0	0	0	15
$0 < T_o \leq 12$	301	179	71	182	258
$12 < T_o \leq 18$	8	119	238	128	27
$T_o > 18$	0	2	1	0	0

Table 2 Proportion of water temperature $0^\circ\text{C} < T \leq 12^\circ\text{C}$ range to the total number of hours per month

	May	June	July	August	September	summary
The total number of hours (h)	310	300	310	310	300	1530
$0^\circ\text{C} < T_o \leq 12^\circ\text{C}$	301	179	71	182	258	992
The percentage (%)	97	60	23	59	86	65

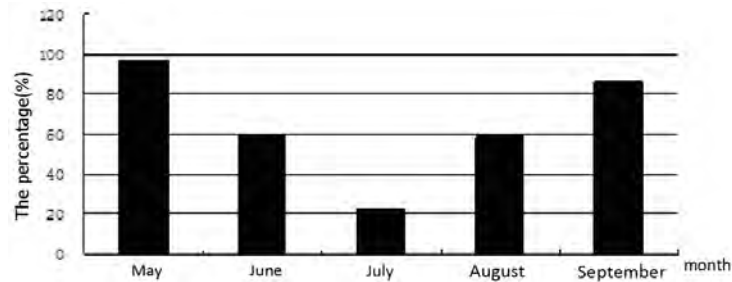


Figure 6 Proportion of water temperature $0^{\circ}\text{C} < T \leq 12^{\circ}\text{C}$ range to the total number of hours per month

4.2 The analysis of system running status

According to the water temperature analysis, uses the form of cold source of evaporative cooling chiller water storage, in the May to September, at night 23:00 to 8:00 in the morning, water temperatures can reach 0 to 12°C range for 301 h in May, June is 179 h, July is 71 h and August is 182 h, 258 h in September, the total number of hours of May to September 5 period is 65%, the total number of hours that accounts for the largest percentage of May, It can reach 97%.

According to July 27, 2012, Urumqi peak valley time sharing sales price list, according to the business of electricity price level for 10 KV, peak price is 1.089 RMB/kW·h, low electricity prices is 0.279 RMB/kW·h, difference of peak valley price is 0.81 RMB/kW·h.

By peak valley price difference, the night cold storage technology of energy saving effect is obvious, integrated evaporative cooling chiller water temperature calculation and the number of hour statistics, using evaporative cooling chiller to night chilled water thermal storage scheme has certain feasibility. Evaporative cooling chiller operation of low energy consumption, and It can be better achieve satisfaction of building air conditioning system energy saving effect

5. CONCLUSION

Evaporative cooling technology with water as refrigerant, the use of outdoor air in the wet and dry bulb temperature do driving power, and with air heat and humidity exchange to prepare in cold water and cold wind, It is more economic and environmental protection. At night environment, outdoor air wet bulb temperature difference greater, wet bulb temperature is low, It has a certain improvement for the efficiency of evaporative cooling, combined chilled water thermal storage technology, and using evaporative cooling chiller to cool storage at night. Statistics obtained in Urumqi region May to September night, cool storage using evaporative cooling chiller hours can be reach 65%, difference of peak valley price in 0.81 RMB/kW·h, at the same time, using evaporative cooling air handling unit in the night, building their own cold storage capacity of ventilated cool storage has certain feasibility in terms of energy saving. therefore, this paper Put forward the application of evaporative cooling chiller air conditioning system combined with chilled water thermal storage technology, for the

building of environmental protection and energy saving air conditioning system play an important role, it also for evaporative cooling unit combination with cool storage technology provide a reference design.

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THE APPLICATION PROSPECT OF EVAPORATIVE AIR CONDITIONING IN TOWN RESIDENTIAL BUILDING

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ABSTRACT

Application of evaporative air conditioning in town residential building is classified. Characteristics of evaporative air cooler, window evaporative air conditioner, cabinet evaporative air conditioner and spit type evaporative air conditioner are analyzed. The application of the four forms of evaporative air conditioning in the different forms of town residential building is given. Split evaporative air conditioner can be achieved multi-split. Storage, semiconductor refrigeration and solar power are combined, it respectively can be used to peak clipping fill valley, improve the evaporative air conditioning working stability and resolve the winter heating. From energy conservation, indoor air quality and environmental protection etc, the application prospect of evaporative air conditioning in town residential building is stated.

Keywords: Evaporative cooling; Evaporative air conditioning; Town residential; Energy storage; semiconductor refrigeration; solar energy

1. INTRODUCTION

The development of our national economy is developed stably and rapidly, according to the national bureau of statistic announced that in 2012 China's GDP grew by 7.8% year on year, quarter year-on-year growth of 7.9%. Under the rapid economic development, people's living standard is also to be increased constantly. Speed up the pace of urbanization, town was turned into large construction industry. The average housing area of town residents in China has reached the 28 m². People enjoy a rich life at the same time, environmental and health problems had caused the attention of people.

Recently, the "atmospheric haze" weather has occurred in major cities of China. The dust, sulfuric acid, nitric acid and organic hydrocarbon compounds such as a large number of tiny particles of dust uniformly floating

in the air, resulting in rise of the index of PM¹⁰ and

PM^{2.5}. And entering into the state of "power cuts" has happen ahead of time in many provinces this summer. Behind these problems, low carbon, environmental protection and energy saving concept of life begin to enter people's life. Countries vigorously promote energy conservation and emissions reduction, to require the use of energy security and health. At the same time the energy ladder should be used.

Building energy consumption including heating, ventilation, air conditioning, lighting, household appliances, transportation, power, cooking, feed water, hot water supply and so on. According to statistics,

building energy consumption accounts for about 30% of the total energy consumption, of which 70% of building energy consumption (about 20%) building energy consumption for heating, ventilation and air conditioning, with the development of urban construction, building air conditioning is required to be more energy-saving, environmental, also to take a sustainable development road. Evaporative air conditioning is different from traditional air conditioning, natural refrigerants and no four parts consumption are used, through the air and the water heat and moisture exchange, it have an effect of cooling and air purification, in dry area or transition season, also can be played the role of humidifier.

2. APPLICATION OF PRIMARY EXPLORATION

2.1 Town construction development

According to using properties in our country, Construction can be divided into productive construction and civil buildings. Residential building is included in the civil building, according to <Code for design of civil building> (GB50325-2005), residential building is used to offer short term or long term for people living, because of the town population growth and economic development, the civil building share into a rising trend. The "three highs" residential had turned up in big cities: high-level, high density and high capacity rate of housing. Urban residents generally live in the purchase of residential building room, which of the area is small, the space is little, low storey height and so on.

In 1962 the American writer Rechal Carson published a book called "Silent Spring" book, under the influence of people's behavior, once cheerful, lively village became dark and gloomy was be described in the book. This is the first time when people begin to pay close attention to living environment. Under the light of lead and after more than half a century of development, people put forward the "green buildings", It contains two aspects that the concept of indoor environment and energy requirements was put forward. Occupant comfort, health, and indoor air quality was accentuated in indoor environment. The EER of the residential air conditioning and the utilization rate of energy will be improved.

Residential construction development speed is gradually increased; annual growth rate reached

3.4%.Residential building energy consumption and carbon dioxide emissions also is raised year by year.

2.2 Classification and characteristics of evaporative air conditioning

Evaporative air conditioning can be divided into direct evaporative cooling and indirect evaporative cooling according to the cooling technology. Appeared on the market at present products mainly is direct evaporative cooling technology. It also can be divided into an evaporative air cooler and an evaporative air conditioner on the basis of the form of air processing equipment. Appeared in the market at present, the product was most of the evaporative air cooler, in Fig. 1 the physical picture of an evaporative air cooler and evaporative air conditioner are shown.

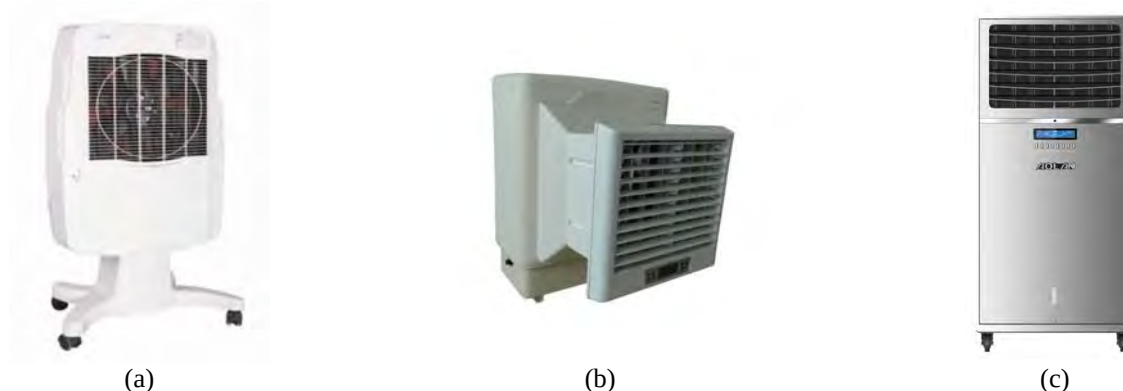


Figure 1 (a) Evaporative air cooler; b Window evaporative air conditioner; b Cabinet evaporative air conditioner

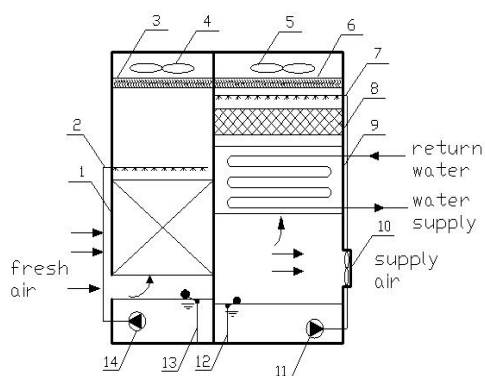


Figure 2 The structure diagram of Split evaporative air conditioner

Different type of evaporative air conditioners is needed to apply in different residential building area and room load. Table 1 is showed the main types of characteristics of evaporative air conditioner.

3. THE SIGNIFICANCE OF APPLICATION

3.1 Comfort and energy saving of urban residential building

Residential building indoor thermal comfort is not only required to provide thermal comfort, also provide a good indoor air quality. At the same time aim at guaranteeing people can absorb the fresh air, oxygen enrichment and inspire vitality of health in air-conditioned comfort. Studies had shown that indoor pollutants concentration is as high as outdoor pollution 5 ~ 10 times, living without fresh air or less indoor fresh air volume long time can appear giddy, lack of power, such as tinnitus syndrome "air-conditioning symptom". Just decorating good house, the density of indoor formaldehyde (usually is a kind of flammable, colorless and stimulating gas) is higher, its effect on human health is mainly embodied in the abnormal smell, stimulation, allergies, lung function abnormalities, abnormal liver function and abnormal immune function etc. residential construction should be timely added fresh air. Indoor air environment is included Indoor Thermal Environment (ITE) and Indoor Air Quality (IAQ) [2].AS shown in the table2

Table 1 the main types of evaporative air conditioning

Form	Applications	Characteristic	Professional Standard
Evaporative air cooler	Proper cooling and humidifying home are demand for open type or well ventilated room. It can be dropped 5-13 °C in the northern dry areas and 5-8 °C in southern moderate humidity area.	Air volume from 350~1800m ³ /h, It is Simple construction, convenient using, cheap, easy to move and more flexible.	National standard GB/t23333-2009 《Evaporative air cooler》
Window evaporative air conditioner	Window evaporative air conditioner use area is greater than evaporative air cooler, it is satisfied with the requirement of the room ventilation and cooling, as well as can be reached 5-8 °C temperature drops. It is Suitable for dry areas or medium humidity area where room was not needed strict temperature and humidity.	Air volume from 2500 ~6000 m ³ /h, It is Simple structure, compact structure, small volume, Low noise, large wind Angle and range. It can be installed on the Windows, also against the wall installation.	National standard 20081657-T-604 < Evaporative air conditioner>
Cabinet evaporative air conditioner	Cabinet evaporative air conditioner is satisfied with the requirement of room ventilation and cooling, also can achieve 2-13°Ctemperature drop. It can be applied to the room of large range of human activities and air conditioner convenient mobile room. Post air supply can be realized.	air volume from 4000 ~ 6000 m ³ /h.It is compact structure, small volume, Low noise, large wind angle. It can be placed in any location of the room, beautiful and easy, and interior design can be integrated.	
Spit evaporative air conditioner	It is given high temperature cold water and fresh air to the room, Indoor area is increased, also installation height can be greatly increased less than or equal to 15 m. It can be used in the bedroom, such as more families, as well as the villa building.	It was formed the close cold water system to avoid contamination of circulating water and reducing the pump power consumption; it can offer cooling and purification of air, in order to improve indoor air quality, as well as it can be achieved "multi-split".	At present products are studied

Table 2 the current standard requirements of Indoor thermal environment parameters

Parameters	Unit	Standard values			Notes
Temperature	°C	22~28	22~28	24~28	summer air-conditioning
		18~24	18~24	18~22	Winter heating
Relative Humidity	%	40~65	40~80	≤70	summer air-conditioning
		30~60	30~60	≥30	Winter heating
Air Velocity	m/s	≤0.3	≤0.3	≤0.3	summer air-conditioning
		≤0.2	≤0.2	≤0.2	Winter heating
Fresh Air Volume	m ³ /h·per	30			
Sulfur Dioxide	mg/ m ³	0.50			One hour average value
Nitrogen Dioxide	mg/ m ³	0.24			One hour average value
Carbon Monoxide	mg/ m ³	10			One hour average value
Carbon Dioxide	%	0.10			daily average value

Hydrogen Nitride	mg/ m ³	0.20	One hour average value
Ozone	mg/ m ³	0.16	One hour average value
Formaldehyde	mg/ m ³	0.10	One hour average value
Benzene	mg/ m ³	0.11	One hour average value
Toluene	mg/ m ³	0.20	One hour average value
Xylol	mg/ m ³	0.20	One hour average value
[a]B(a)P	mg/ m ³	1.0	daily average value
Inhalable Particle	mg/ m ³	0.15	daily average value
Total Volatile Organic compounds	mg/ m ³	0.60	eight hours average value
222Rn	mg/ m ³	2500	Setting by the instrument
Aerobic Bacterial Count	Bq/ m ³	400	annual mean

Wen - whai Li for the E1 Paso of west Texas region 10 inside the bedroom air particulate matter were determined in the summer of 2001. The experimental results were showed that after evaporative cooler indoor PM₁₀ particulate matter had reduced 50%, PM_{2.5} particulate matter had decreased by 10% and 40%[3-4]. A lot of outdoor air and indoor return air can be used to or for clean air after an evaporative air conditioning. Residential construction is used for heating, hot water, fridge and air conditioning, this part of the energy consumption accounts for 67% of total energy consumption. The development of town house and rising of per capita GDP, residents of household air conditioning had also increased year by year, Towns across the country per household air-conditioner ownership is 30.8 units. Every 100 household of air-conditioner ownership is increased to 112 in 2010 in national town. All power supply basically is relied on thermal power in coastal city of faster economic development, And household air conditioning power consumption is continued to increase, causing an explosion of air conditioning load during the day, at the same time bringing tension of city power system. So the residential building energy saving is of great significance.

3.2 Our energy consumption and environmental problem

The energy consumption is affected directly and indirectly by the life of people, due to the energy consumption of summer to open air conditioning, the demand for electricity and the pressure of power supply are increased. In the low efficiency of thermal power generation (China's thermal power efficiency is only 37.72% in 2009), emissions of carbon dioxide with the rise in. Fig.3(Gehlin 2002) is showed china's per capita carbon emissions the years from 1988 to 2006, per capita carbon dioxide emissions are 4.7 tons in 2006.

Due to the use of chlorofluorocarbon, it has appeared that reduction in atmosphere Ozone

concentrations and hole in the ozone. layer. China's regional atmospheric ozone concentration change curve is measured by satellite, Fig 4 is showed a falling concentration of ozone from 1980 to 2000, and keeps a dropping trend.

Causing the "greenhouse effect" is caused by rise in atmospheric carbon dioxide concentration and the hole in the ozone layer, in which carbon dioxide emissions is the dominant factor. Nearly 100 years of global average surface temperature rose by nearly 2 °C, some scholars has been predicted a future the average surface temperature will be increased of 2.7 ~ 2.9 °C in the next 100 years of china through simulation. To alleviate the electric energy of consumption, per unit of GDP carbon emissions by 2020 than in 2005 will be decreased by 40% ~ 50%(Gehlin 2002).

For representing an objective and scientific urban resident's use of the traditional household air conditioning influence on resources and environment, the method of life cycle analysis is used. Life cycle assessment (LCA) is the evaluation of a product, process or activity, from raw material acquisition to the product of the production, transportation, sales, use and final disposal and recycling use of the entire life cycle system diagram about the environmental load of process. Through the software of SimaPro simulating, results are showed that the traditional household air conditioning a 1.5 P in its life period, emit carbon dioxide of 1.89 x 10 Kg, 3.033 Kg of carbon monoxide, sulfur oxide of 78.017 Kg and sulfuric gases of 1.277 Kg, 90% of the pollutants from the power system. And the harmful material of evaporative air conditioning in the use phase. But output of evaporative air conditioning is only equivalent to 1/6 of the mechanical refrigeration(Fig 5).

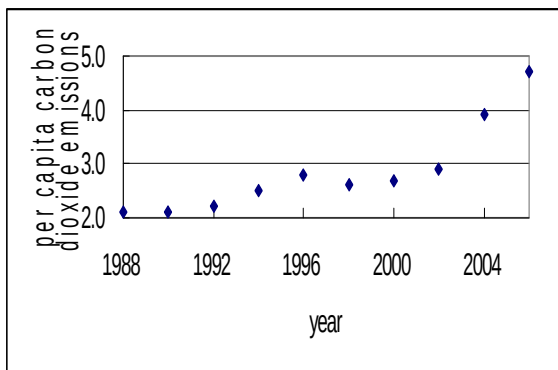


Figure 3 China's per capita carbon dioxide from 1988 to 2006

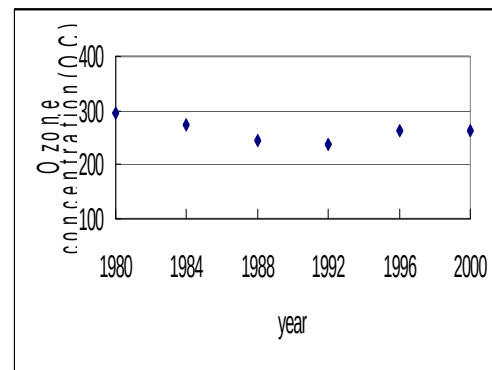


Figure4 Ozone concentration emissions from 1980 to 2000

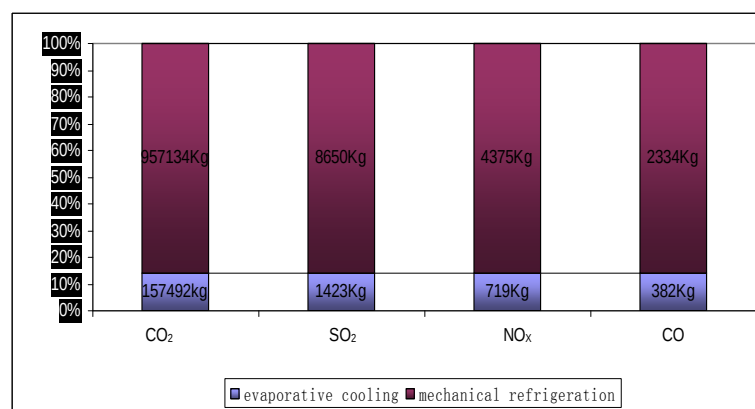


Figure 5 Emissions of harmful substances of air conditioning unit of evaporative cooling and mechanical refrigeration during using phase

Building energy consumption including heating, ventilation, air conditioning, lighting, household appliances, cooking, water supply and hot water supply, such as energy consumption about building. energy consumption accounts for 70% of building energy consumption (20%) used for heating, ventilation and air conditioning. The use of natural energy sources should be promoted under development of the china urban residential construction vigorously. The evaporative air conditioning takes full advantage of the dry air which can be used in cooling by water vapor partial pressure. Using this kind of air-conditioning system, certain economic and environmental benefits can be brought.

3.3 The forms of application

Difference between urban economic conditions and living standards, the form of residential construction is change as well. Generally in towns are less developed area, there is still a six floors of residential buildings in the following. The houses had built for a long time, neat appearance and family pattern similar to form a unified whole. In xi 'an area, for example, a room area is 30 ~ 50 m², a model AZLO4 - LC12D window evaporative air conditioner can be

chosen, the maximum air volume is 4000 m³/h, which can be meet user needs the summer air conditioning. The evaporative air conditioning are set as shown in figure 6 to hang outside the window, outdoor hot air through the air conditioner, the temperature of is reduced, air can be provided people with comfortable environment through the duct to indoor.



Figure 6 Application of schematic diagram about evaporative air conditioner in the residential building

Under the rapid economic development and population growth in towns, high-rise residential has been arisen at the historic moment. Community building over six floors of people's life, and is equipped with

elevators. As shown in figure 7, an evaporative air conditioner can be in sitting room or bedroom place to absorb the heat from indoor return air, and then the cool wind through tuyere circulation back to the room, indoor temperature will be reduced. But has been circulating room return air, the air quality is reduced, people can't absorb the fresh air in a timely manner, after evaporative air conditioning of air humidity will be increased, When using an evaporative air conditioner, open the window should be opened and the outdoor air can be entered so that the people's demand for fresh air is ensured and the indoor humidity field balance is guaranteed. Xi 'an region as an example, a sitting room area is 36 m^2 in residential area, the bedroom area is 15 m^2 , a AZLO4 - LC13G window evaporative air conditioner which of air volume is $4000\text{ m}^3/\text{h}$ can be chosen for a sitting room, Air is supplied for the sitting room to help people get through the hot summer months. Choose AZLO35 - LY13C cabinet evaporative air conditioner which of air volume is $3500\text{ m}^3/\text{h}$ can be chosen for a bedroom, and the adornment of the bedroom be in harmony are an organic whole, beautiful and generous.



Figure 7 Application of schematic diagram about Cabinet and Window evaporative air Conditioner in the residential building

Villa construction area is larger and the greening rate around the residential is higher, the pattern of room is different, the area of air conditioning is larger than residential buildings. Due to split evaporative air conditioner output the cold water and cold air at the same time, can be applied in the villa. General Villa construction area in $150 \sim 300\text{ m}^2$, if use the traditional air conditioning, severe energy consumption can be made. Through indirect evaporative cooling air conditioning technology, fresh air can be cooled with constant humidity. Split evaporative air conditioner as

outdoor, grille is set indoor to ensure uniform air distribution, coil as the indoor machine. As shown in figure 8, split evaporative air conditioner is used in the residence villa.



Figure 8 Application of schematic diagram about split evaporative air conditioner in the residence villa.

4. ROSPECT

4.1 The combination of cool storage technology

Cool storage technology according to the medium is classified into water storage, ice storage and eutectic salt storage. If the energy saving and cost saving consideration are considered, water storage has the certain superiority. Water temperature is required to reach $3 \sim 7\text{ }^{\circ}\text{C}$ in traditional water cool storage. In the xinjiang Altay, for example, data year by year is extracted from the Taylor meteorological parameters from June to August, electricity valley value period: 07:00 $\sim$ 23:00 dry bulb temperature and wet bulb temperature are taken. It is calculated that the average dry bulb temperature is $18.3\text{ }^{\circ}\text{C}$, the wet bulb temperature is $12.1\text{ }^{\circ}\text{C}$ and the dew point temperature is $7.8\text{ }^{\circ}\text{C}$. Through the indirect /direct evaporative cooling air conditioning technology, $7 \sim 9\text{ }^{\circ}\text{C}$ cold water will be made, If heat temperature difference for $4 \sim 6\text{ }^{\circ}\text{C}$, $11 \sim 15\text{ }^{\circ}\text{C}$ high temperature cold water will be provided in the peak period of electricity during the day. It will have the effect of "peak load shift". Ice storage also can be achieved, and nowadays development of phase change materials technology, cold quantity will be stored by phase change of special materials. This is a kind of physical energy storage process, the process of physical energy -cold energy - physical energy conversion can be realized.

4.2 The combination of Semiconductor refrigeration technology

Evaporative cooling and semiconductor is combined skillfully to achieve complementary, and the advantages of both are united in terms of air conditioning integrated innovation .After the 1850 s, the

rapid development of semiconductor materials, semiconductor refrigeration are entered in to some areas of people's life. It has merit of no moving parts, simple structure, fast refrigeration and don't need refrigerant, etc. The coefficient of water-cooled thermal radiation heat transfer is 100 ~ 1000 times larger than air natural convection cooling owing to the combination evaporative cooling and semiconductor. Cold water temperature is also reduced under Semiconductor refrigeration technology so that improving the stability of evaporative cooling air conditioning system.

The outdoor climatic parameter in xi 'an of a moderate humidity region is that dry bulb temperature is 35.1 °C and wet bulb temperature is 25.8 °C, after experimental verification, the cold water is prepared from evaporative cooling firstly, then the temperature of cold water is dropped at 3.7 °C through two groups of semiconductor refrigeration device of 250L/h flow[10]

4.3 The combination of solar energy technology

In the late 1970 s, passive solar house has began to be studied and used in china. This is the first step using solar energy in residential architecture, Late of preparing solar hot water technology is became relatively mature, a solar collector was used to solve living hot water in many families. Used in the construction of residential solar technology, already can be the integration of refrigeration, heating and domestic water. Solar cooling application of two ways: one is prepared for hot water to drive equipment refrigeration; The other is a refrigeration equipment is powered by solar power(Gehlin 2002).In order to solve the problem of residential buildings in winter heating, Solar collector to produce hot water can be used and sent into terminal for heating. Solar energy can also be used to generate electricity, evaporative air conditioning just is needed to offer energy consumption of water pump and blower, compared with traditional air conditioning, the part of the energy consumption is small. Solar energy is converted into electrical directly and stored in the battery trough the solar array.

For the future integration of urban residential building of air conditioning will be realized the integration of refrigeration, heating and domestic water. During cooling season, split air conditioner can be prepared for cold water and cold air at the same time. Cold water can be used to meet indoor air conditioning demand, cold air can be sent into indoor to fulfill the need of fresh air. Solar energy can be absorbed into solar collector and convert into hot water. Summer can be supplied people's living and living hot water. Winter not only can be contented with living and living hot water, but also can be satisfied with the water demand of terminal.

5. CONCLUSION

1)According to the form of equipment, evaporative air conditioning can be divided into an evaporative air cooler and an evaporative air conditioner.

2)Urban residential construction application of evaporative air conditioning can be applied in different forms according to different form of housing. It can be equipped with different sets of evaporative air conditioning according to the size of the area. In one residential building form, it can mix the two and two or more of the air conditioners.

3) Evaporative air conditioning can clean indoor air; the problem of indoor dry will be solved in dry area, also the indoor humidity balance will be maintained through opening window.

4) Primary exploration of union with cool storage technology. In the xinjiang Altay, for example, the data year by year is extracted from the Taylor meteorological parameters from June to August, electricity valley value period: 07:00 ~ 23:00 dry bulb temperature and wet bulb temperature are taken. It is calculated that the average dry bulb temperature is 18.3 °C, the wet bulb temperature is 12.1 °C and the dew point temperature is 7.8 °C.

5) The combination of new energy will be promoted the development of the evaporative air conditioning. Equipped with two sets of 250 l/h semiconductor refrigeration device, moderate humidity area of evaporative cooling water temperature can be reduced 3.7 °C . Combined with solar energy technology, it can be realized the integration of refrigeration, heating and living hot water

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DEVELOPMENT OF A DYNAMIC SIMPLIFIED THERMAL MODEL OF ACTIVE PIPE-EMBEDDED BUILDING ENVELOPE

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ABSTRACT

This paper presents a dynamic simplified thermal model of the active pipe-embedded building envelope with a thermal network structure of lumped thermal mass. These lumped resistances and capacitances are identified in frequency domain by using generic algorithm by comparing its frequency characteristics with the theoretical frequency characteristics of this structure obtained with frequency-domain finite difference method. In this method, the heat transfer equation is represented in frequency domain, and this structure is discretized at the meantime. A case study is presented to demonstrate the development of the simplified model of this structure.

Keywords: Active pipe-embedded building envelope; Frequency characteristic; FDFD method; Simplified model; Genetic algorithm.

1. INTRODUCTION

Active Pipe-embedded Building Envelope is a pipe-embedded building external wall/roof. Pipes are embedded in the wall slab or roof slab to allow water circulated in these pipes for heat transfer [1, 2]. This structure is similar to the pipe-embedded floors or ceilings for radiant cooling/heating, which may significantly enlarge the surface between the structure mass and the water in the pipe for allowing low temperature difference heat transfer [3-5] and use various low-grade energy sources such as groundwater, the water produced by cooling towers, and geothermal energy produced by the ground coupled heat exchanger system, etc. [6, 7]. Similarly, the active pipe-embedded building envelope has the same advantages.

Model development and performance prediction of the active pipe-embedded building envelope is of importance for system design and indoor environment control. In the past, there are many researchers to develop thermal models of pipe-embedded structure as floors/ceiling for analyzing the heat transfer. Antonopoulos et al. [8] developed one-dimensional steady-state analytical model and a two-dimensional steady-state numerical model for temperature field analysis in the panel. Kilis et al. [9, 10] proposed a steady-state fin model for modeling the heat transfer of the pipe-embedded structure as floor/ceiling. This model was validated using the numerical solution of the structure by using a finite element package [11]. Simplified models are also presented by many researchers for performance evaluation for this structure. Koschenz and Dorer [12] presented a simplified steady-state model of the active pipe-

embedded structure for air-conditioning. Weber and Jóhannesson [13] presented a dynamic simplified RC-network model for pipe-embedded structure (called "thermally activated building component system (TABS) in this article).

Active pipe-embedded building envelope is similar to floors/ceiling on structure. However, the heat transfer of the active pipe-embedded building envelopes is affected additionally and significantly by the outdoor environment. Zhu and Xu [2] presented the frequency-domain finite difference (FDFD) model of the active pipe-embedded building envelope for the frequency response analysis of the active embedded-pipe building envelope under various disturbances, while frequency characteristics may represent exactly the dynamical thermal behaviors of this structure. However, this model cannot be solved easily for convenient integration with energy simulation software.

Undoubtedly, dynamic simplified RC model is the best candidate for modeling active pipe-embedded structure since it can simulate the dynamic thermal process in the structure and can be easily integrated with the traditional energy simulation software. However, the model structure and parameter determination of the RC model have a significant effect on the thermal performance of this structure.

This paper presented a dynamic simplified thermal model (i.e., RC model) of the active pipe-embedded building envelope, and its parameters are identified.

Frequency-domain finite difference (FDFD) model is used to calculate the theoretical frequency thermal performance of the active pipe-embedded building envelope under various disturbances with different frequency (i.e., theoretical model). A Genetic algorithm

NOMENCLATURE

Abbreviation		Subscripts	
FDFD	Frequency-domain finite difference	j	Number of the cells or boundary nodes
GA	Genetic algorithm	X	heat flux frequency characteristics on the external wall surface
Symbols	C	Y	heat flux frequency characteristics on the internal wall surface
	R	Z	heat flux frequency characteristics on the internal pipe surface of the embedded pipe
T	Temperature in frequency domain (K)	m	Disturbance
u	Real part of the complex quantity (K)	Greek Letters	Frequency ($\text{rad}\cdot\text{s}^{-1}$)
v	Image part of the complex quantity (K)		
$\bar{T}$	Temperature in time domain (K)		
PA	Phase angle ($^{\circ}$)		
AM	Amplitude (K)		
J	Objective function		
$G_m(j\omega_n)$	Predicted frequency characteristic by using the simplified thermal model		
$G_m'(j\omega_n)$	Predicted frequency characteristic by using the theoretical model		
W	Weighting factor		

(GA)estimator is developed to identify the parameters of the simplified model by matching the frequency characteristics of the simplified model with the theoretical frequency responses of the theoretical model. A case study is presented.

2. DESCRIPTION OF THE ACTIVE PIPE-EMBEDDED BUILDING ENVELOPE PAPER STRUCTURE

An active pipe-embedded building envelope can be coupled with low-grade energy sources, and the coupled system schematic is shown as Figure 1. This system has five parts, i.e., heat exchange system of the building envelope, thermal energy storage system, auxiliary cooling source system, auxiliary heating source system, and power transport system. External walls or roofs with embedded pipes are acted as the heat exchange system of the active pipe-embedded building envelope, and the fluid (usually water is used) is circulating in the pipe to extract heat/coolth inside the structure directly and can intercept the heat or coolth from the ambient air to the indoor space resulting reduced cooling load or heating load. Ground soil or aquifer may be the thermal energy storage system, and can be used as the thermal energy storage medium for storing heating/cooling extracted from building envelope or supplied by the auxiliary heating/cooling system. Cooling tower may be used as the auxiliary cooling source system, and it can utilize the low wet-bulb air temperature to produce cold water for storing coolth in ground soil or aquifer. Solar water heater may be used for auxiliary heating source system to produce hot water for storing heat in ground soil or aquifer. For this system, pump is needed to power the circulating fluid in the pipe. Through the switch of control valves and water loops, different control modes can be achieved for heat/coolth extraction, auxiliary cooling, and auxiliary heating etc.

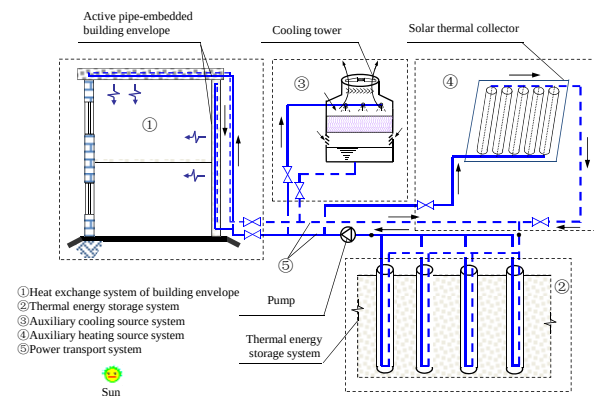


Figure 1 System schematic of an active pipe-embedded building envelope coupled with low-grade energy sources

In the system, only fan for cooling tower and pump in the water system consume high grade energy (such as electricity). When photo voltaic system is available, they can be powered by solar energy indirectly. Therefore, this system with active pipe-embedded building envelope consumes much less high-grade energy than conventional air-conditioning systems.

3. THEORETICAL FREQUENCY CHARACTERISTICS OF THIS STRUCTURE BY FDFD MODEL

Figure 2 shows the typical cross section of the active embedded-pipe building envelope. The heat transfer along the length direction of the pipe is neglected, and the heat transfer can be simplified as a two-dimensional heat transfer problem. In this study, typical cross section *abcd* of the embedded-pipe building envelope is used for model development.

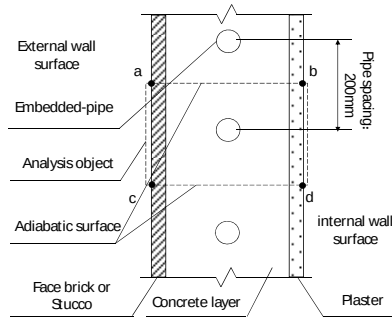


Figure 2 Schematic of the cross section of the active embedded-pipe building envelope

Finite-difference method is used for discretizing the typical section $abcd$ as a series of cells as shown in Figure 3 for modeling the thermal performance of the structure by using FDFD method. Each cell in this model is represented by using four resistances and one capacitance, and the temperature of the capacitance represents the temperature of the cell. The temperature of each cell with frequency ω may be represented in complex domain, and can be expanded as real component and image component. When the thermal equilibrium in matrix form of each cells or boundary nodes are described, linear simultaneous equations can be obtained to present the FDFD model of the active pipe-embedded building envelope structure. A FORTRAN code by using generalized minimal residual method (usually abbreviated GMRES) [14] is developed and used for problem solution. Detailed deduction can be found in Literature [2].

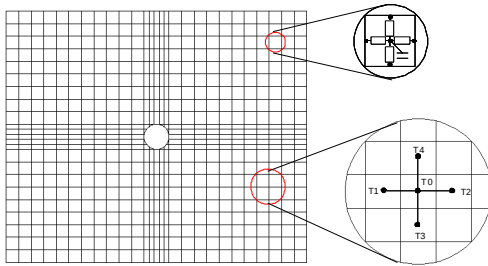


Figure 3 Discrete grid model of the active-embedded building envelope

4. DEVELOPMENT OF THE DYNAMICAL SIMPLIFIED THERMAL MODEL OF THIS STRUCTUREACTIVE PIPE-EMBEDDED BUILDING ENVELOPES

The typical section $abcd$ as shown in Figure 2 is simplified as the 5R2C (five resistance and two capacitance) thermal model as shown in Figure 4. In Figure 4, T_{out} is the temperature of the external wall surface. T_{in} is the temperature of the internal wall surface, T_w is the temperature of the internal surface of the embedded-pipe. To simplify the model development, the temperature of the internal surface of the embedded-pipe is assumed to be the same as the circular temperature of the fluid in the pipe since the heat transfer resistance of the internal surface of the

embedded-pipe and circular fluid is small and can be neglected, T_1 and T_2 is the node temperature of the simplified model. To calculate the frequency characteristics of the simplified model of the active pipe-embedded building envelope in the concerned frequency range, the transfer functions of nodes of simplified models are deduced as follows.

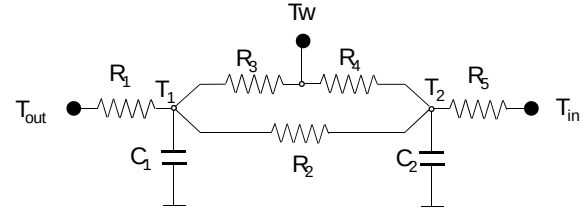


Figure 4 Schematic of the simplified thermal model of the active-embedded building envelope

The simplified thermal model contains two thermal nodes C_1 and C_2 . The thermal equilibrium equation of the node C_1 and C_2 can be written as the Equation (1) and (2) respectively, and the temperature is expressed in complex form. The linear simultaneous matrix equations as Equation (3) can be obtained by simple transform and operation to present the simplified thermal model of the active pipe-embedded building envelope structure.

$$C_1 \frac{dT_1}{dt} = \frac{T_{out} - T_1}{R_1} + \frac{T_2 - T_1}{R_2} + \frac{T_w - T_1}{R_3} \quad (1)$$

$$C_2 \frac{dT_2}{dt} = \frac{T_1 - T_2}{R_2} + \frac{T_w - T_2}{R_4} + \frac{T_{in} - T_2}{R_5} \quad (2)$$

$$\begin{bmatrix} \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} & -C_1\omega & -\frac{1}{R_2} & 0 \\ C_1\omega & \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} & 0 & -\frac{1}{R_2} \\ -\frac{1}{R_2} & 0 & \frac{1}{R_2} + \frac{1}{R_4} + \frac{1}{R_5} & -C_2\omega \\ 0 & -\frac{1}{R_2} & -C_2\omega & \frac{1}{R_2} + \frac{1}{R_4} + \frac{1}{R_5} \end{bmatrix} \begin{bmatrix} u_1 \\ v_1 \\ u_2 \\ v_2 \end{bmatrix} = \begin{bmatrix} \frac{u_{out}}{R_1} + \frac{u_w}{R_3} \\ \frac{v_{out}}{R_1} + \frac{v_w}{R_3} \\ \frac{u_w}{R_4} + \frac{u_{in}}{R_5} \\ \frac{v_w}{R_4} + \frac{v_{in}}{R_5} \end{bmatrix} \quad (3)$$

Where: C is the capacitance ($J/(m^2 \cdot K)$), R is the resistances ($m^2 \cdot K/W$). T is the temperature in frequency domain (K), ω is the frequency ($rad \cdot s^{-1}$), u is the real part of the complex quantity (K), v is the image part of the complex quantity (K),

The variables u and v can be solved directly through (3) when the boundary conditions (i.e., the surface temperatures) are given. The variables u and v can be described in terms of $\bar{T}$ and PA as Equation (4). Then, the amplitude and phase angle of the frequency characteristics can be easily obtained as Equation (5) based on the solved u , and v . Therefore, the temperature and heat flux numerical solutions on the external wall surface, internal wall surface and internal surface of embedded-pipe of simplified model in frequency domain can be obtained easily, and could be compared with the corresponding solutions of the FDFD model for optimizing their parameters.

$$u = \bar{T} \sin(PA), v = \bar{T} \cos(PA) \quad (4)$$

$$AM = \sqrt{u_j^2 + v_j^2}, \quad PA_j = \arctan \frac{v_j}{u_j} \quad (5)$$

Where: $\bar{T}$ is the temperature in time domain (K), PA is the phase angle ($^\circ$), AM is the amplitude (K) subscripts j is the number of the cells or boundary nodes.

For parameter identification of the simplified 5R2C model in frequency domain, the frequency characteristics of FDFD model and simplified model should be calculated first as described in the earlier sections. With the FDFD model, the frequency characteristic of this structure can be calculated easily. With the assigned parameter values of the simplified models, the frequency characteristics can also be calculated conveniently. The frequency characteristics with FDFD model are used as objective for the frequency characteristics of the simplified 5R2C model for identification. The parameters to be optimized are the resistances and capacitances of the 5R2C model which give the best fitting between the frequency characteristics of the simplified model and the frequency characteristics by using FDFD model. The objective function of such optimization is expressed in Equation (6).

$$J(R_1, R_2, R_3, R_4, R_5, C_1, C_2) = \sum_{n=1}^N \sum_{m=1,2,3} \left\{ \left| \frac{W_{m,X}^{AM}}{W_{m,X}^{AM}} \left[AM(G_{m,X}(j\omega_n)) - AM(G'_{m,X}(j\omega_n)) \right] \right| + \left| \frac{W_{m,Y}^{AM}}{W_{m,Y}^{AM}} \left[AM(G_{m,Y}(j\omega_n)) - AM(G'_{m,Y}(j\omega_n)) \right] \right| + \left| \frac{W_{m,Z}^{AM}}{W_{m,Z}^{AM}} \left[AM(G_{m,Z}(j\omega_n)) - AM(G'_{m,Z}(j\omega_n)) \right] \right| \right\} + \left\{ \left| \frac{W_{m,X}^{PA}}{W_{m,X}^{PA}} \left[PA(G_{m,X}(j\omega_n)) - PA(G'_{m,X}(j\omega_n)) \right] \right| + \left| \frac{W_{m,Y}^{PA}}{W_{m,Y}^{PA}} \left[PA(G_{m,Y}(j\omega_n)) - PA(G'_{m,Y}(j\omega_n)) \right] \right| + \left| \frac{W_{m,Z}^{PA}}{W_{m,Z}^{PA}} \left[PA(G_{m,Z}(j\omega_n)) - PA(G'_{m,Z}(j\omega_n)) \right] \right| \right\} \quad (6)$$

where J is the objective function, AM is the amplitude, PA is phase angle, $G_m(j\omega_n)$ is the predicted frequency characteristic by using the simplified thermal model, $G'_m(j\omega_n)$ is the predicted frequency characteristic by using the theoretical model. The subscript X indicates the heat flux frequency characteristics on the external wall surface, Y indicates the frequency characteristics on the internal wall surface, Z indicates the frequency characteristics on the internal pipe surface of the embedded pipe. W is the weighting factor. The subscript m indicates the disturbance. These seven parameters are constrained by the total resistance and capacitance of concerned cross section of the active pipe-embedded building envelope. A GA driver developed by Carroll [15] was revised in this study.

5. CASE STUDY

A case study is presented to demonstrate the development of the simplified model by using FDFD method and genetic algorithm (GA). This case study is a light weighted wall for embedding pipes, which is selected from ASHRAE Handbook of Fundamentals [16]. It is mainly composed of stucco with 20mm, light density concrete with 100mm and plaster with 20mm. The detailed properties of the wall in SI unit are given in Table 1.

Table 1 Details of the light weighted wall

Description	Thickness and thermal properties				
	L(mm)	$\lambda(W \cdot m^{-1} \cdot K^{-1})$	$\rho(kg \cdot m^{-3})$	$C_p(J \cdot kg^{-1} \cdot K^{-1})$	$R(m^2 \cdot K \cdot W^{-1})$
Stucco	20	0.692	1858	840	0.029
Light density concrete	100	0.571	609	840	0.175
Plaster	20	0.727	1602	840	0.028

In this study, the amplitude and phase angle of the predicted heat flux on the external wall surface, internal wall surface and internal surface of the embedded-pipe are used for performance prediction comparison between simplified thermal model (i.e 5R2C model) and FDFD model (i.e, theoretical model). The parameters of the simplified model are identified based on the frequency characteristic comparison by using GA. For the development of this simplified 5R2C model, parameters are identified considering the external wall surface disturbance, the internal wall surface disturbance and the internal pipe surface disturbance evenly. This means that the weights of external wall surface disturbance, internal wall surface disturbance and internal pipe surface disturbance are 1.

To analyze the frequency characteristics of the active pipe-embedded building envelope, usually three disturbances (i.e., the external wall surface temperature disturbance, internal wall surface temperature disturbance, and internal pipe surface temperature disturbance) are imposed on these surfaces respectively. The frequency characteristics of this structure under various disturbances can be calculated by using the theoretical model and the 5R2C model. For conciseness and space saving, only the dynamic thermal performance of this structure imposed by the external wall surface temperature disturbance within the frequency range of normal concern (10^{-8} to 10^{-3} rad $\cdot$ s $^{-1}$) [17] by using the theoretical model and the 5R2C model is presented and analyzed.

Based on these identified parameters, the frequency characteristics of the 5R2C model can be plotted and compared with that of the theoretical model.

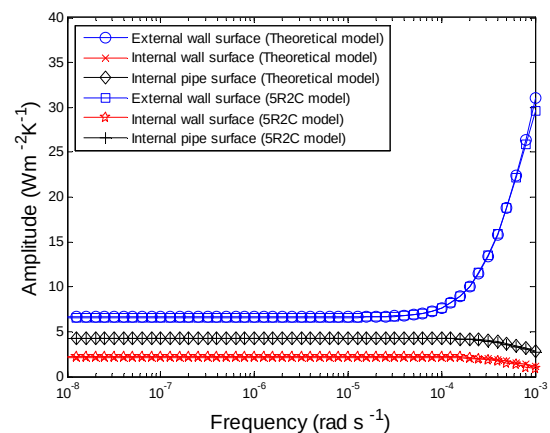


Figure 5 Comparison of heat flux amplitudes on these three surfaces by using the theoretical model and the optimal model

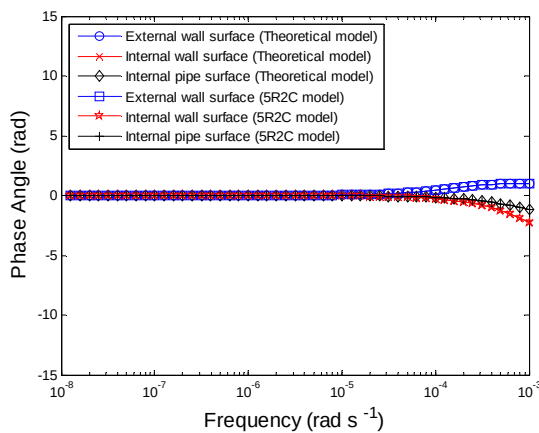


Figure 6 Comparison of heat flux phase angles on these three surfaces by using the theoretical model and the optimal model

Figure 5 and Figure 6 show the amplitude and phase angle of the heat flux on the external wall surface, internal wall surface and the internal pipe surface by using the theoretical model and the 5R2C model respectively. The comparison results show that the amplitude of heat flux on these three surfaces predicted by using the 5R2C model can well match that by using the theoretical model in the frequency range of normal concern as shown in Figure 5. Figure 6 indicates that the phase angle of heat flux on these three surfaces predicted by using the 5R2C model also matches that by using the theoretical model in the frequency range of normal concern. It means that the 5R2C model can present the good agreements with the theoretical performance by using the theoretical model. Based on the above analysis, it shows that the dynamic simplified model (i.e., 5R2C model) can predict the frequency response characteristic of the active pipe-embedded building envelope under the various disturbances for matching well with frequency response characteristics by FDFD model (i.e., theoretical model).

6. CONCLUSION

This paper presents a simplified thermal model (5R2C) of active pipe-embedded building envelopes since a simplified thermal model is the best candidate for modeling the dynamic thermal behaviors and integration in conventional energy and indoor environment simulation packages. For the parameter determination of the simplified model, Frequency-Domain-Finite-Difference (FDFD) model (i.e., theoretical model) is used to calculate the theoretical frequency characteristic of this structure for reference. GA estimator is used to identify the parameters of the simplified model by comparing the simplified model prediction with the theoretical model prediction. A case study is presented demonstrating the model development of this 5R2C model and the parameter identification. The result shows this simplified model can predict the frequency response characteristic of the active pipe-embedded building envelope under the various disturbances for matching well with frequency

response characteristics by FDFD model. The simplified model (RC model) can be solved very easily and can be integrated with conventional building simulation package for building performance evaluation.

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STUDY ON UNSTEADY CHARACTERISTICS AND FIELD MEASUREMENT OF NATURAL WIND

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ABSTRACT

Nature wind characteristics consist of averaged and fluctuating wind characteristics. Taking natural wind as constant flow may cause a larger error or even utterly wrong conclusions. With the change of time and space, natural wind mainly studied by field measurement and statistical analysis appeared to have a strong nonlinear random feature. Since field measurement is one of the important ways to analyze natural wind characteristics, the working principle and application scope of general instruments for measuring wind velocity, such as hot-wire anemometer, impeller anemometer, Pitot tube anemometer, laser Doppler anemometer, ultrasonic anemometer, were introduced in this paper. Also, research on wind measurement and analysis at home and abroad is summarized, and the following conclusions are obtained: Ultrasonic anemometer and small weather station are widely used for collecting data of nature wind characteristics. Database of wind characteristics, wind speed and direction model, empirical formula are obtained according to the actual characteristics of turbulent wind based on statistical analysis.

Key words: natural wind, fluctuating characteristics, experiment methods, experiment instruments, field measurements

1. INTRODUCTION

In the field of architecture construction, natural ventilation is one of key techniques to improve the indoor thermal condition and decrease the energy consumption of air conditioning system by utilizing natural energy sources. So far, studies on natural ventilation at home and abroad have improved measurably and entered the fields of unsteady state natural ventilation. Natural wind is a consequence of atmospheric movement, and wind direction and speed are changing from minute to minute^[1]. Basic data and variety characteristics of wind direction and speed are crucial for the study of natural ventilation indoors and outdoors^[1].

However, researches on natural ventilation, no matter CFD numerical simulation or wind tunnel test, often taking wind direction and average wind speed provided by weather station as constant outer field boundary conditions. It may lead to deviation of the results from the truth^[1], and it is difficult to evaluate the effects of natural ventilation. Thus researchers in this area should fully understand the characteristics of natural wind in theory. It is generally considered that the wind is consisted of average wind speed component and turbulent components. The average wind speed is represented by observational data in a certain period of time. Atmospheric turbulence shows the characteristics of nonlinear random fluctuation strongly in time and space, and it couldn't be described theoretically by now, so measurement and statistical analysis are the main

way to study turbulent wind characteristics^[2]. In this situation, since the 1990s many researchers at home and abroad have devoted to experimental study of turbulent wind. Research results, such as Databases of wind characteristics of the region, models of wind speed and direction, general empirical formula, etc. are obtained. Although marked results of natural wind test are achieved in our country up to now, we should keep exploiting and surpassing ourselves, and make more contributions.

2. Natural wind characteristics

Atmospheric boundary layer wind characteristics comprise the average wind and turbulence properties, and mean wind is the statistical laws of observation data in a certain period, including average wind speed, wind direction distribution and wind profiles; atmosphere turbulent components shows strong nonlinear stochastic pulsation characteristics in time and space, generally expressed by fluctuating wind speed, turbulence intensity, turbulence relevance, gust factor, integral scale and fluctuating wind power spectrum. In addition, some researchers have been using fractal chaos theory to study turbulence properties of natural wind and propose some characteristic parameters of natural wind.

The mean wind and turbulent wind characteristics of natural wind are introduced in detail in literature^[3] and will not be repeated here. Some supplements of the turbulent wind characteristics are as follows.

2.1. Fluctuating wind speed

The fluctuating wind speed is analyzed in two ways, vector method and scalar method. Vector method decomposes horizontal wind vector of atmospheric turbulence into longitudinal horizontal component $u(t)$, level horizontal component $v(t)$ and vertical component $w(t)$; Scalar law defines horizontal wind speed with level absolute wind speed and level wind direction.

2.2. Power spectral density function

Integration of turbulence power spectrum density function in frequency domain is equal to turbulent kinetic energy in corresponding direction, whose distribution in frequency domain can describe the proportion of the turbulent kinetic energy on different scales^[4]. Turbulent power spectrum density can be obtained by the time correlation function $R(\tau)$ of fluctuating velocity after the Fourier transform and the power spectrum curve also can be directly measured by anemometers.

2.3. Gust factor

The gust factor G is a ratio of gust wind speed to average wind speed. The gust factor is relevant to turbulence intensity. The greater turbulence intensity, the larger gust factor is. Besides, it also depends on duration of gust, the longer duration, the smaller gust coefficient.

2.4. Fractal and Chaotic Characteristic Analysis

Natural wind is random and irregular macroscopically while it contains different scales swirls with self-similar structure microscopically, reflecting Chaos feature of turbulence. Fractal and chaos are inseparable and it means that a seemingly chaotic and disorganized graph has a fine structure, which is called the self-similar structure, mainly expressing an intrinsic geometric regularity, that is, the self-similarity of proportion. In 2000 Zhu Yingqiu^[7] defined a ratio(δ) of width to length of the phase space reconstruction graph to describe shape of the graph and considered δ as an important criterion distinguish natural wind from mechanical wind. In addition, she also proposed two indices, information entropy S_i and information dimension D_i , which describe how the structure and form of the system disordered^[8]. In 2010, based on the phase space reconstruction theory, Chen Jun analyzed three groups natural wind speed time series samples and indicated that the two-dimensional phase space reconstruction graph of natural wind is spindle and natural wind is fractal. In 2011, Guo Hao, etc. calculated the natural wind velocity samples using G-P algorithm, educed fractal dimension of natural wind, and shown that fractal dimension was related to surface roughness.

3. Test methods of wind

Wind measurements include wind direction tests and wind speed tests. The vane is the most common device for measuring wind direction. A balanced position pointing to the coming flow obtained under dynamic pressure is the wind direction. The wind directions are denoted with 16 azimuths. Speed tests contain the measurements of mean wind speed and that of turbulence components. Test methods of wind speed conclude mechanical method, heat dissipation rate method, dynamic pressure method, Laser Doppler velocimeter and ultrasonic measurement. Some explanations of these methods are listed below.

3.1. Heat dissipation rate method

The way tests wind speed is based on the relationship between the heat dissipation of metal wire and air velocity. It can be divided into: (1) Constant temperature type: Temperature of hot wire remains constant when the current through hot wire changes. Hot-wire anemometer is commonly used under this condition, whose sensitivity is quite high in low speed; (2) Constant current type: Current of hot wire remains invariable when the temperature changes. Hot-bulb anemometer is commonly used in this condition, whose sensitivity is quite high in high speed. Sometimes a metal film replaces the metal wire, known as the hot-film anemometer to enhance effect.

Hot-wire anemometer is fit to low wind speed measurement, and measure both the mean speed and fluctuating wind velocity. Hot-bulb anemometer applied to high wind speed measurement, and only to the mean speed test. It has same strengths such as high precision, good dynamic performance, small interference on flow field and an advantage of temperature compensation, but easy to damage.

3.2. Mechanical method

The wind cup or vane anemometer is common used in this way. Its working principle is that the air drives the blades to rotate, whose rotating speed proportional to the wind speed and then wind speed is tested by measuring rotating speed.

The range of the wind cup/impeller anemometer is from 0.6m/s to 40m/s and the measurement precision is 1%. It has simple framework and poor dynamic performance, and is suitable for testing the low average wind speed and unfit to instantaneous wind speed.

3.3. Dynamic pressure method

The principle of this method is the Bernoulli equation. Commonly used instrument in this method is the Pitot tube. When operating, measuring section must be in the steady flow region, total pressure hole toward the flow direction, static pressure holes vertical to the flow direction. Then the wind speed is calculated by measuring the difference between static pressure and dynamic pressure.

Pitot tube ranges in wind speed from 2.5m/s to 80m/s, accurate to 3%. It has simple structure, easy to carry, whose dynamic performance is poor, and it only

fit to steady flow and average value test, incapable of fluctuating value test. Generally it used to measure wind speed in wind tunnel and pipeline.

3.4. Laser Doppler velocimeter

The principle of Laser Doppler anemometer: When a laser beam penetrates fluid and irradiates particles moving with fluid together, scattering light frequency of particle is detected and according to the optical Doppler Effect, and then the velocity of particles or fluid is determined.

Laser Doppler anemometer testing velocity range is between $0.05\mu\text{m/s}$ and 10^6m/s , and testing precision is from 0.1% to 1%. It is a non-contact measurement, which doesn't disturb flow field, doesn't require regular calibration, and distinguish the flow direction. Besides, there are some other advantages of it, such as good dynamic response, high precision and wide measuring range. Particles in turbulent flow can't follow the fluid well, so this way usually is unable to measure fluctuating wind velocity.

3.5. Ultrasonic measurement

The method utilizes speed difference when ultrasonic pulse propagates in downstream and upstream to reflect the fluid velocity. Ultrasonic anemometer measures wind speed based on the principle that the sound speed in the air stacks with the airflow velocity toward wind direction. If the ultrasonic propagation direction is the same as wind direction, airflow will speed up; conversely, it will speed down.

Measuring range of ultrasonic anemometer is large, and measuring precision of it is high, up to 1%. There is no particular demand of measurement environment for it. So it is applied widely.

3.6. Summery

Among these five testing technologies, hot wire is suitable for low speed measurement, which can test both the average velocity and fluctuating wind speed. Hot bulb only fit to high average speed measurement, unsuitable for measurement of fluctuating wind velocity. Mechanical method suits for testing the low average wind speed and can't test pulsing wind velocity. Pitot tube ranges in testing mean wind speed from 2.5m/s to 80m/s , but incapable of fluctuating value test. Laser Doppler anemometer has the maximum measurement range and generally is not fitted to measure fluctuating air velocity. Since measurement ranges and application scope of ultrasonic anemometer is all extensive, ultrasonic method can test both the mean wind speed and the fluctuating component.

4. Studies of wind experiment

The most effective method in order to accurately describe the wind characteristics of a particular area is that having an analysis for a large number of wind observation data in the region and obtaining appropriate empirical models and statistical parameters^[13]. At the end of the last century, some countries advanced in the

wind engineering research have established wind characteristics database of their region, such as Froya database of Norway^[14], offshore wind observations database of Canada and the UK. Relatively complete analysis based on a great deal of large-scale observations in time or space, similar to the U.S. Sparks^[15,16], Japan Kato^[17] and Ohukuma^[18], was obtained. Chinese researches field observational studies of wind characteristics have lagged behind, but considerable progress has been made in the recent 10 years. Case analysis of filed measurement of wind measured in China and abroad over the past decade is then given.

4.1. Typhoon measurement

Lin Zhixing et al. measured the wind characteristics of Typhoon Prapiroon and Typhoon Jelawat with ultrasonic anemometers in 2000. The anemometer probe mounted on the roof of 5m, at a height of about 20 m above ground. Taking the easterly direction as the main wind direction, the 3D strong winds sample is collected at a rate of 10 kHz for 20h. Analysis results show that turbulence integral length is about 80m, horizontal turbulent power spectrum density functions is consistent with Simiu spectrum, but the vertical turbulent power spectrum is different from Panofsky spectrum^[12].

In 2003, the wind characteristics of Typhoon Maemi were recorded at an experimental site for electric cables of the Furukawa Electric Co. Ltd. by Cao et al. The wind speeds were measured simultaneously by nine vane and seven sonic anemometers at a height of about 15 m. Instantaneous wind speeds were recorded by the sonic anemometers in three orthogonal directions. The sampling frequency is 10Hz and sampling Interval is 10 minutes. Turbulence intensity and their variations with wind speed were obtained. The analysis showed: turbulence intensity decreases with wind speed and remain almost constant when the wind speed becomes high. The averaged values of gust factor and peak factor are 1.6 and 3.3, respectively. The spatial cross correlation and decay factor of the coherence function increase slightly with wind speed. The probability density function of fluctuating wind speed of a strong typhoon follows a Gaussian distribution, and the power spectrum of strong wind can be expressed by a Karman-type spectrum at the low frequency. The wind characteristics of this strong typhoon are shown to be very similar to those of non-typhoon winds^[18].

In 2004, a field wind measurement was conducted by Pang Jiabin at Shenzhen suburb, Shanghai Pudong suburb and the middle span of the Second Minjiang River Bridge in Fuzhou when they were affected by typhoon. The test sites also include Sutong Bridge. Continuous wind speed samples were recorded by ultrasonic anemometers at a rate of 10Hz for 3h near the peak of wind speed. According to the analysis of measured data, the results showed that average wind speed calculated by scalar method is larger than that of vector method, but turbulence intensity and gust factor are reverse; turbulence power spectrum density

functions curves obtained with both methods are similar^[4].

In 2007, time series data of wind speed and direction of Typhoon Rosa were recorded at a rate of 20Hz with sonic anemometer placed at the height of 436m on Shanghai World Financial Center by Wu Zhanke. Based on the statistical studies, high-altitude wind characteristics, such as typhoon turbulence intensity, gust factor and turbulence integral length were obtained. According to Kolmogorov theory, he also fit the turbulence power spectrum density function of horizontal and vertical direction^[19].

The wind measurements taken by Tse et al., during the passages of several tropical cyclones or typhoons in Hong Kong by the wind-profilers and the Doppler Sonic Detection And Ranging (SODAR) have been utilized to calculate both the hourly and 10-min mean wind profiles between 2007 and 2009. The results showed that the low-level jet, which was commonly observed for the mean wind profile in the tropical cyclone boundary layer over the sea, was not found. In addition, the surface roughness derived by fitting the observed profile to the log-law was unusually high ($z_0 > 1.5\text{m}$)^[20].

4.2. Normal natural wind measurement

Li Jie measured wind speed on a bridge in China with the two three-axis ultrasonic anemometers at the monsoon season in 2007, and obtained wind speed data for January, February and March. The sampling frequency is 4Hz and the total sampling time was 2046h. The function expression of random Fourier spectrum was established based on experimental data of March, then verified its reasonability and commonality with data of January and February^[21].

Between 2008 and 2010, Tan Hongwei set up two small weather stations and an ultrasonic anemometer to measure the wind characteristics every minute at two teaching building of Tongji University different in height and shading rate and outskirts of Shanghai Xinzhuang. According to the data of wind direction changes of each minute and each hour, the following conclusions were reached: the hourly main wind direction was stable within the time span of a few hours to a few dozen hours; but the main wind direction of each minute changed all the times and showed significant unsteady-state characteristics, wind direction of each minute was changing from -45° to 45° around the main wind direction and the fluctuation was not significant^[22].

In 2010, Chen Jun measured wind speed at some sites near a university campus, such as open area around buildings, lakeside, building roof and forest park. The probe of hot-wire anemometer was 1.6 meter above ground, the sampling interval is 10min and the sampling frequency is 10Hz, sample data collected at each location is composed of 3 groups including 4096 samples. The $1/f$ turbulence characteristics of natural wind were determined by analyzing the power spectrum exponent calculated by the distribution characteristics of power spectrum density of the sample^[6].

From 2009 to 2010, Ji Liang et al. continuously monitored the wind characteristics at 1 min intervals by using a small weather station at the top of a tall building in the central of Shanghai. Based on wind-direction-dependent analysis and Markov Chain, wind direction change model has been structured^[1].

A long-term field measurement of wind speed and directional fluctuation on wind-driven cross ventilation in a mock-up building was conducted by J.S.Park between 2010 and 2011. The wind speed and direction was monitored at 1 min intervals using a weather station, and the static pressure difference measured by Pitot tube and a data logger at 10 s intervals. According to the data analysis, the pressure coefficient at the openings was significantly influenced by the fluctuating components of the wind speed and direction, and the unsteady airflow rates were also sensitive to the fluctuating components^[23].

4.3. Natural wind in forest measurement

In 2003 natural wind was measured in a mixed forest at Jiufeng experimental forest farm of Beijing Forestry University by Zhu Shoulin et al. Hot-film wind speed sensor is used for collecting signal samples of two-dimension(X,Y). Sample data collected was composed of 9 groups including 3072 samples. Sampling time interval was 200ms and sampling frequency is 5 Hz^[24].

Studies showed that the mean and variance of wind speed of the main wind direction are high, but the average turbulence intensity was less than that of side wind; natural wind direction was pronounced changed; two-dimensional velocity of main wind direction and side wind direction showed statistically normal distribution; main wind direction and side wind direction of natural wind near the ground showed a high degree of randomness; the correlation between the two-dimensional wind speed signal of main wind direction and side wind direction was very low^[24].

4.4. Natural wind indoor measurement

In 2004, wind Sampling experiments were made by Ou Yangqin et al. at roof terrace, space around buildings, open lawn far away from buildings, the beach and the interior of natural ventilation building. The measuring sites outside were arranged in a 1.6m in height from the ground or the roof. Several samples were sampled in the same work condition with the environmental temperature of 15°C to 39.0°C . Based on power spectrum exponent and turbulence intensity averages, natural wind spectrum characteristics of all conditions were similar, but β value were different, the built environment and average wind speed had a certain impact on natural wind spectrum characteristics^[25].

In 2012, Gu Xinyu continuously measured pulse wind with digital anemometer located on the 10th floor of an office building in Shenzhen. Wind speed samples of 15 min were recorded for three times. Analyzing the statistical measure of pulsation period and pulsation strength, change law of pulse wind velocity was attained. Effect of wind pulsation characteristics on

building ventilation in urban areas was discussed; the volatile of urban wind should be considered in the design of natural ventilation^[26].

4.5. Summary

In summary, persons conducted wind experiment are mostly engaged in structure engineering, bridge wind engineering and other field of architecture. Ultrasonic anemometer, hotwire anemometer, small weather station, Laser Doppler velocimeter are their major tools. Wind characteristics databases were established based on wind pulse characteristics, empirical formula, and some empirical formula have been proposed on the basis of measured results and theoretical formula. Lacking of measurement of the characteristic of wind indoor under natural ventilation now, researchers should explore and thoroughly study in this field.

5. CONCLUSIONS

In this paper, three conclusions were obtained as followed.

1. Nature wind characteristics consist of averaged and fluctuating wind characteristics. Average wind characteristics such as wind speed, distribution of wind direction and wind profiles are represented by statistical law of observational data in a certain period of time. Atmospheric turbulence shows the characteristics of nonlinear random fluctuation strongly in time and space, generally expressed by fluctuating wind speed, turbulence intensity, turbulence relevance, gust factor, integral scale and fluctuating wind power spectrum.

2. The vane is often used for measuring wind direction, and hot-wire/ball anemometer, vane/rotor meter, anemometer, Pitot tube anemometer, Laser Doppler anemometer, ultrasonic anemometer for wind speed. Hot wire is suitable for low speed measurement, which can test both the average velocity and fluctuating wind speed. Hot bulb only fit to high average speed measurement, unsuitable for fluctuating wind velocity. Mechanical method suits for testing the low average wind speed and can't test pulsing wind velocity. Pitot tube ranges in testing mean wind speed from 2.5m/s to 80m/s, but incapable of fluctuating value test. Laser Doppler anemometer has the maximum measurement range and generally is not fitted to measure fluctuating air velocity. Since measurement ranges and application scope of ultrasonic anemometer is all extensive, ultrasonic method can test both the mean wind speed and the fluctuating component.

3. Ultrasonic anemometer, hot-wire anemometer and small weather stations are used for measurement of natural wind at home and abroad. Fluctuating characteristics of natural wind, database of wind characteristics and empirical formulas are obtained on the basis of field measurement of nature wind. Researchers in the field of natural ventilation mostly are involved in the following areas: outdoor natural wind under normal condition, the turbulence characteristics of indoor natural wind in natural ventilation environment, the way to create dynamic air with the

properties of natural wind indoor. Although some achievements are achieved, many researches remain to be done. At present, it is of significance to pay more attention in the research of the field measurement of natural ventilation by using predecessors' experiences.

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HEAT AND HUMIDITY ENVIRONMENT TEST AND ANALYSIS OF AIR-CONDITIONING SYSTEM OF MUSEUM

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ABSTRACT

Museum, as a cultural treasure of human cultural and natural heritage, is the window to show the progress of human civilization. The artifacts provided a suitable preservation of the environment is the most basic and the most effective means of protection, and adverse temperature and humidity, not only bring relics of a physical deformation, but also become a breeding ground for harmful microorganisms to erode heritage. Therefore, Monitoring of temperature and relative humidity in indoor visiting environment and the cultural relic's conservation environment are the most basic parameters. This paper is mainly based on the different geographical climate environment, do comparative analysis, and with the field investigation and one years' observation of heat and wet parameters from indoor visiting environment and the cultural relic's conservation environment in Liaoning Province Museum which is in a cold region, a full year of the regional temperature and humidity conditions are tested, the change rules of indoor air temperature and relative humidity were obtained in each visiting environment and artifacts stored showcase. Getting a comprehensive understanding of present situation of the cultural relic's conservation environment in Liaoning Province Museum and at the same time it provides a basic reference for staff to regulate the air-conditioning system.

Key words: Museum; Air-conditioning system; Temperature; Relative humidity.

1. INTRODUCTION

Museum is a place of cultural relics preservation and display. The effective protection of cultural relics, in addition to consummate the management system, its preservation environment control also directly affects the "life" of cultural relics. Based on the concept of preventive conservation, museum of cultural relics protection should be fully aware of the cultural relics storage and display environment quality problem. Chen Yuan sheng in museum of cultural relics preservation environment quality standards research pointed out that: temperature, relative humidity, gaseous pollutants, particle pollution and lighting are the main environmental factors which caused cultural quality degradation^[1]. Among them, the temperature and humidity is the most basic and is also very important to monitor and control indicators. Bad temperature and humidity environment will directly influent even determine cultural relics all physical, chemical, biological function, accelerate the cultural relics of the aging^[2]. Therefore, taking various measures to master and regulate museum of cultural relics preservation environment temperature and humidity balance is the current international consensus in the field of science and technology in the protection of cultural relics, and

has important practical significance to the proper preservation of the cultural relics collection.

2. TEMPERATURE AND HUMIDITY REGULATION

2.1 Regional effects on temperature and humidity

Museum set up throughout the various provinces and regions in our country, because of the difference of the local climate environment characteristics, climate environment of the museum in various regions will present different geographical climate environment such as wet and dry, hot, cold, etc. According to the data in 《The civil thermal design specifications》^[3] and 《2003 China Statistical Yearbook》^[4], Summed up the basic characteristics of each region partition, which is divided into different distribution range of the average temperature of the hottest month, the average temperature of the coldest month the hottest month average humidity and the coldest month average humidity, as shown in Table 1

Table 1 Each region partition environment characteristics

Climate zone	The hottest month average temperature	The coldest month average temperature	The hottest Month average humidity	The coldest month Average humidity
Frigid region	20°C~25°C	-20°C~-10°C	50%~75%	60%~75%
Cold region	23°C~28°C	-5°C~0°C	60%~70%	40%~65%
Hot summer and cold winter area	25°C~30°C	0°C~10°C	75%~80%	75%~80%
Hot summer and warm winter area	27°C~28°C	10°C~14°C	80%~80%	65%~68%
Moderate area	18°C~20°C	8°C~10°C	70%~75%	50%~65%

From the above data which can get the hottest month average temperature difference in the geographical district is small, at 20°C ~ 30°C. The difference in the coldest month average temperature, from 14 °C to -20 °C, and the temperature difference in division of the district is very obvious, fully reveal the basic climate characteristics of the divided areas; In terms of average relative humidity, the most thermidor of hot summer and warm winter area is in high humidity circumstance, followed by hot summer and cold winter region, its value is less than 40% in cold region. Therefore, as to the control of the different district museum in our country to the indoor thermal environment, paying close attention to regional climate characteristics. Regulation should think about local conditions and the actual conditions of temperature and humidity. Consider to each texture artifact which needs appropriate range of temperature and humidity, control target value cannot exceed the limits of the safety of cultural relics preservation.

2.2 Control range of temperature and humidity parameters in liaoning provincial museum

Heritage preservation of the environment in museum cannot use a single save benchmarking environment which should be determined to a different basis on the different geographical characteristics of the environment. Because our country has not set rules for museum collection preservation standards according to the different geographical environment, so Control range of temperature and humidity parameters in Liaoning Provincial Museum is determined to the existing specification and some scholars recommended values. According to different circumstances, providing for reference temperature and humidity of historical relic's exhibition hall library, as shown in Table 2. In addition, Zheng Ai ping ^[5] suggest that the diurnal variation of the type of building indoor air temperature should not exceed 5° C, relative humidity should not exceed 5%. 《Museum architectural design specification》 (JGJ66-91) ^[6]; more sensitive to cultural relics collection chamber should be set air-conditioning equipment to ensure that the heritage preservation environment temperature and humidity constant. Literature [7] proposed indoor air temperature control range of 20 to 22°C, relative humidity of 40% to

55%, and 24 h per day must be continuous control, is not allowed cyclical fluctuations in temperature and humidity. Refer to the above temperature and humidity control range, visiting environment in exhibition hall should be as far as possible to control the temperature of 18 ~ 24°C, relative humidity control in (50 + 10) % is appropriate; painting area showcase should control the temperature at 23±2°C, relative humidity control in(60±5)%.

Table 2 Reference temperature and humidity of Historical relic's exhibition hall library

Heritage collection category	Indoor temperature /°C	Relative humidity /%
Metal	18~24	40~50
Bamboo	16~20	50~60
Ceramic	18~24	50~60
Paper	14~18	50~60
Leather	16~20	60~65
Silk cotton wool	16~20	50~65

3. TEST SCHEME

3.1 Test Object

Liaoning Provincial Museum is a modern large-scale comprehensive museum, covers an area of 16,000 square meters, total construction area of 31,095 square meters. Building main body: a layer underground, three construction main body for 1 to 3 layers, the cultural relics for the underground layer storage warehouses, building height is 24 meters. The interior of the building is a ring, the middle is the hall, the cultural relics exhibition hall on both sides. Heritage exhibition hall did not have windows, 3rd floor roof is light wells. Exhibition Hall of the museum with a glass case Heritage closed and isolated from the audience. Showroom uses central air conditioning system to control the indoor thermal environment, air-conditioning running time is generally from 8:00 to 16:00. Painting area showcase uses constant temperature and humidity air conditioning control

system to control the temperature and humidity of the shelves inside, running 24 hours a day, the other enclosed heritage glass case without air-conditioning equipment.

3.2 Test equipment

The main content of the field test: temperature and relative humidity in visiting environment and heritage preservation environment. Use test instruments: China Academy of Building Research Institute developed the air-conditioning temperature and humidity recorder RR002, temperature range of -40 to 85°C, accuracy to $\pm 0.5^{\circ}\text{C}$, humidity measurement range of 0 to 100% RH, accuracy of $\pm 3\%$ (10% ~95% RH, 25°C), the instrument automatically sampling, interval is set to 15 minutes.

3.3 Arrangement of measuring points

Placing a total of 12 measuring points in the exhibition hall of the museum, each exhibition hall places 1 to 2 points. The arrangement of the three measuring points in Paintings District Showcase. These points respectively test the temperature and humidity in visiting hall as well as temperature and humidity in the painting area showcase. Due to the special circumstances of the 1st on the first floor and second floor of No. 3, No. 4 Hall, winter parameters cannot be tested. The same time, due to the special nature of the artifacts in the Museum Showcase cannot test for all showcase heat and wet environment.

4 TEST RESULTS AND ANALYSIS

4.1 Outdoor meteorological parameters

Shenyang is located in the middle latitude area, which belongs to cold region city climate partition, B area in the 《public building energy efficiency design standard》^[8] (GB50189-2005). The annual average temperature to 8.1°C, cold winter, the average temperature of -9.1°C, the minimum temperature at minus date average of 51 days, the cold long periods of time, winter extreme minimum temperature to -30.6°C. Summer average temperature of 22.3°C, extreme maximum temperature of 38.3°C.

The summer testing time for the August 2011 to October 2011 and August 2012 to October 2012, the winter test time from December 2011 to March 2012.

4.2 Thermal environment test results and analysis in visiting environmental

Arrangement and analysis of test results of measuring point in visiting environmental of Liaoning province museum related to the results:

(1) The summer test results

Interception from August 3 to September 2, 2012 test data analysis the hot and wet environment in every hall, temperature and humidity curve shown in

Figure 4.1 and 4.2. The minimum outdoor air temperature in the test period to 11.89°C and the maximum air temperature to 32.38°C. Reference to the determined range of temperature and humidity, combined with energy-saving ideas, the museum exhibition hall design summer temperature is 26°C, relative humidity of 60% or less. From the graph of temperature and humidity, temperature and relative humidity trend of each measuring point is basically the same, the same measuring point temperature change of less than 5°C. Air conditioning running stage, the first floor of the temperature below 24°C, the temperature of the second floor is generally controlled at 22~27°C, and most of the third floor of the exhibition hall temperature at 24°C to 28°C, humidity of all measuring points in the museum is generally higher than 60% and the humidity fluctuation is bigger, especially from 8:00 am to 16:00 pm humidity varies widely. After investigation and analysis there are two reasons for it, one is air conditioning running at 8:00, shut down at 16:00 pm, humidity in the museum is controlled; The other is during that time more visitors visit the museum, making humidity of the museum unstable.

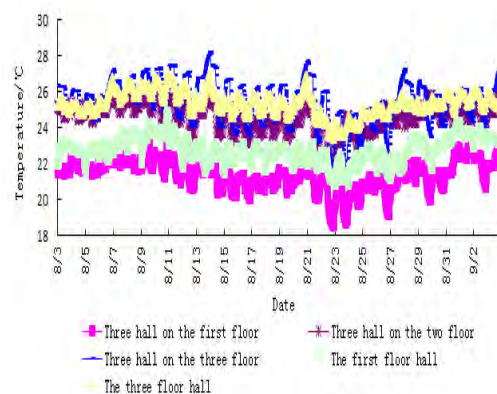


Figure. 4.1 Temperature graph of each measurement point in summer

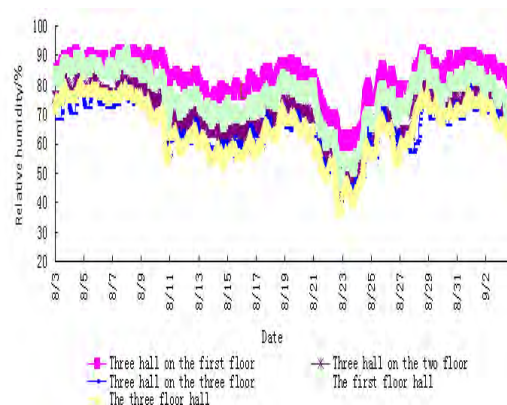


Figure. 4.2 Relative humidity graph of each measurement point in summer

(2) The winter test results

Interception from 13 January to 12 February 2012 test data analysis the hot and wet environment in every hall, temperature and humidity curve shown in Figure

4.3 and 4.4. The minimum outdoor air temperature of the test period to -24.75°C and the maximum air temperature to 2.32°C . Reference to the determined range of temperature and humidity, combined with energy-saving ideas, The museum exhibition hall design temperature of 18°C in winter, the hall design temperature is 20°C , relative humidity of 40% or higher. Air conditioning operation stage, the first floor in 19°C temperature above, most exhibition hall temperature of the second floor and third floor control in $16 \sim 18^{\circ}\text{C}$, and part of the exhibition hall on the third floor temperature is below 16°C . Where the temperature range of the exhibition hall 3 on the first floor reach 6°C , is beyond the state of the museum interior daily temperature range; smooth changes in relative humidity within 5%. In the museum the temperature range from 12.5 to 26.47°C and a relative humidity range from 7% to 19%.

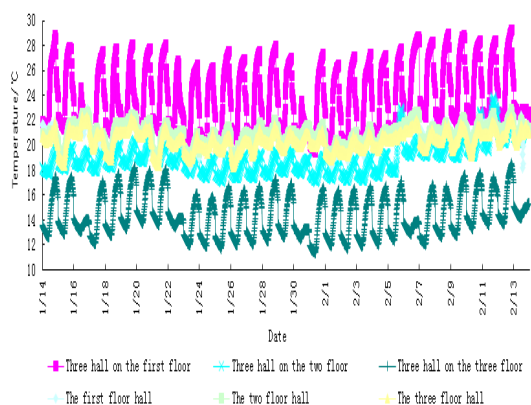


Figure. 4.3 Temperature graph of each measurement point in winter

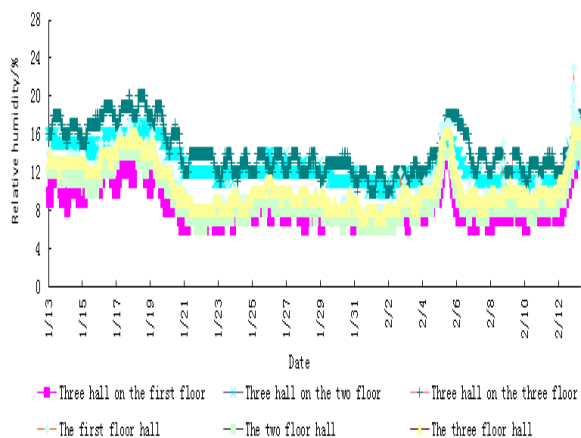


Figure. 4.4 Relative humidity graph of each measurement point in winter

4.3 Thermal environment test results and analysis in constant temperature and humidity showcase

Test data of thermal and wet environment in constant temperature and humidity showcase selected

12 hours of continuous collection data for analysis, the temperature and humidity graph shown in Figure 4.5. Through the observation of temperature and humidity graph, temperature change is small in showcase within 12 hours, the range 1°C or less, the relative humidity controlled at less than 5%, which achieves the design standards.

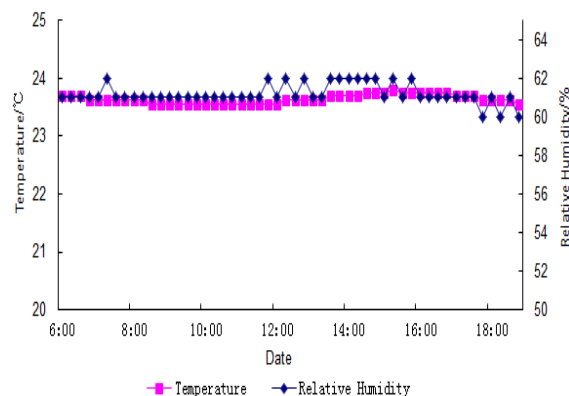


Figure. 4.5 Graph of temperature and humidity regulation in showcase within one day

Figure 4.6 is a graph of changes in temperature and humidity from October 1, 2011 to November 1. Continuous acquisition test data within one month shows the temperature in the showcase generally controlled $23 \pm 2^{\circ}\text{C}$, relative humidity controlled within $60 \pm 5\%$, which achieve the goal of heritage preservation temperature and humidity control. Temperature and relative humidity are to meet the design standards, which shows thermal environment is very stable and air-conditioning systems is in better regulation, provides good quality of the preservation of the environment for the painting class relics. But on October 25 temperature and relative humidity of showcase within the larger fluctuation, which is due to exhibits in painting and calligraphy showcase which is constant temperature and humidity are replaced in that day, showcase is opened, heat and wet environment in Showcase is affected by the hall, so there has been greater volatility.

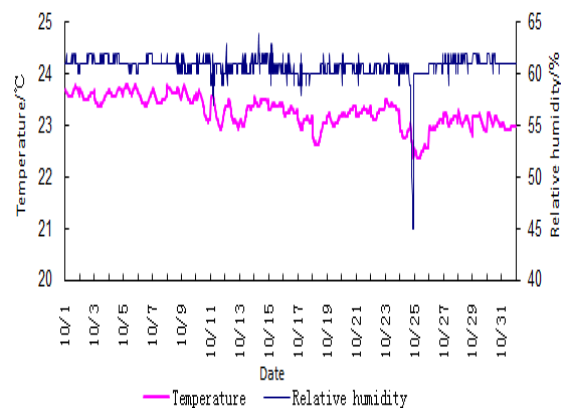


Figure.4.6 Graph of temperature and humidity regulation in showcase within one month

5. CONCLUSIONS

Comparative analysis of climate and environment in different geographical Museum, regulation of indoor environment in the different geographical Museum should be adapted to local conditions, the regulation of temperature and humidity is in accordance with local climatic conditions. Based on one years' tracking test of thermal environment in Liaoning province museum, the results show that the temperature of tour environment in summer range 22~26°C, humidity is 65% ~ 90%, the humidity is high; Winter temperature 16 ~ 22°C and humidity range from 7% to 19%, the humidity is low. As can be seen from the test data, the museum environment temperature basically accords with human body thermal comfort requirements; the humidity control is still lack. Air-conditioning system operation management personnel should regulate to ensure that the requirements of humidity in visiting environment with the changes in the number of visitors in the summer. Air of cold areas in winter is relatively dry, should raise the relative humidity of the air supply parameters and enhance the effect of humidification section of the air-conditioning system. Ambient temperature in showcase is controlled within $23 \pm 2^{\circ}\text{C}$, relative humidity is $60 \pm 5\%$. Temperature and humidity are within its design limits and hot and humid environment are better meet the requirements of painting and calligraphy heritage preservation specifications, which provides a safe and stable thermal environment for museum painting class cultural relics.

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SIMULATION AND ANALYSIS OF ENERGY SAVING STRATEGIES OF AIR-CONDITIONING SYSTEM IN OFFICE BUILDING

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ABSTRACT

Per unit area of energy consumption of office building is not the highest in public buildings, but because of its huge construction area, the total energy consumption is one of the largest of all sorts of public buildings. Office buildings are general installation of central air-conditioning system, and with a very regular use time, air-conditioning system energy saving is the key of the office building energy conservation. In this paper, the office building as an example, analysis of the use of energy, and characteristics of energy consumption, to simulate set temperature of air conditioning system, cold and heat source selection, and the system operating mode, etc, and analyze its effect on the air-conditioning system. The results shown that improve the summer air conditioning set temperature and lower winter indoor heating temperature can obviously reduce building energy consumption of air conditioning, which can produce a very good effect of energy saving. Choose renewable energy as a cooling and heat sources of air-conditioning system, such as geothermal energy and solar energy systems, has a great potential for energy saving. Optimization of system operation mode, such as setting up reasonable intermittent operation of air-conditioning system and take standardized management; it can reduce the system energy consumption.

Key words: Air-conditioning system; Simulation and analysis; Energy saving strategies.

1. INTRODUCTION

In today's information age, the office buildings are increasingly becoming the main building of the social type. About half of the staff in most government agencies and institutions is engaged in the work of the office. In addition to residential, office building is most closely associated with people's life of building types. In the building construction industry in full swing in China, the construction of office buildings accounted for a large proportion. Combined with other various reasons, office building becomes a major energy consumer. Therefore, the office building energy consumption analysis is necessary, in order to prescribe the right medicine in the energy-saving work.

In office building annual energy consumption, consumption about 50% ~ 60% in air conditioning refrigeration and heating system. So the office building energy consumption characteristics of air conditioning has important significance. In this paper, the office buildings in Shenyang, for example, to identify the place of air-conditioning systems with energy-save potential.

2. BUILDING BASIC INFORMATION

2.1 Construction general situation

This building belongs to the normal building, located at No.9 of Hunnan East Road, Shenyang. Shenyang is located in the south of northeast China, due to solar radiation, atmospheric circulation and the influence of geographical environment, belong to the temperate zone monsoon climate, annual average temperature of 6.2 ~ 9.7. The Shenyang city climatic conditions, such as shown in Table 1.1, Annual average temperature in month change is bigger, belongs to cold region climate conditions.

Fig.2.1 for office buildings of the south facade rendering, a total construction area of 10997.38m², total building height of 19.8m. The building has a layer underground, there are five layers, and central has a five-story atrium space. The underground are the air conditioning room and distribution rooms; five layers is office space on the ground. Nature of this office building is relatively single, office hours is more unified, centralized management is equipped with central air conditioning, this is a good choice to analysis of air conditioning energy consumption.

Table 1.1 Shenyang regional climate conditions

Location	Shenyang
Latitude/longitude	41.8°N /123.4°E
Summer design wet and dry bulb temperature	31.4°C/25.4°C
Winter design dry bulb temperature	-22°C
Average wind speed	Summer: 2.9m/s Winter: 3.1m/s
Predominant wind direction	Summer: S Winter: N



Figure.2.1 office building south facade renderings

3. SIMULATION OF ENERGY SAVING TECHNOLOGY OF AIR CONDITIONING

3.1. Simulations software

DesignBuilder is a user-friendly modeling environment where you can work (and play) with virtual building models. It provides a range of environmental performance data such as: energy consumption, carbon emissions, comfort conditions, daylight luminance, maximum summertime temperatures and HVAC component sizes.

3.2. Set temperature of air conditioner

According to the regulations of the "public building energy efficiency design standard", the summer air conditioning set temperature shall be not less than 25°C, shall not be higher than 20°C in winter, that air conditioning set temperature effect on the system energy consumption is very big. According to hot feeling from people in the door, when indoor temperature below 28°C, people will feel cool and not feel hot; When indoor temperature between 28 ~30°C, people feel hot but it doesn't affect life. But now, many of the owners, designers, choose high standards when select indoor design parameters and the value of high and low temperature will direct impact on building energy consumption. In order to further study the

indoor setting temperature on the impact of building energy consumption. For a certain office building in Shenyang, humidity of the indoor design remains the same. Building model, building envelope and indoor thermal perturbation are use the construction under the condition of original parameters, then selecting for summer air conditioning of the set temperature of 23°C , 24°C , 25°C, 26°C and 27°C, the simulation results as shown in Fig. 3.1.

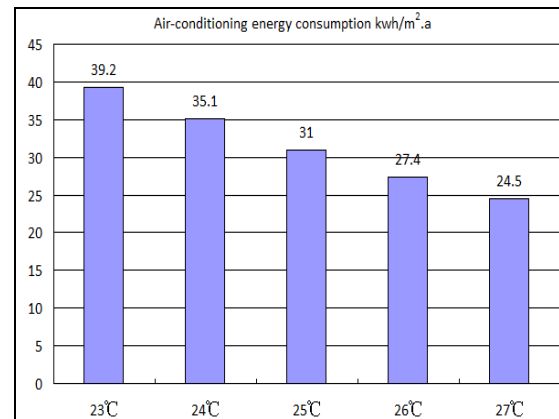


Figure.3.1 Refrigeration energy consumption with different set temperature in summer

From Fig.3.1, when the summer air conditioning set temperature increased from 23°C to 27°C, the cooling energy consumption decreases, cooling energy consumption fell 14.7 kwh/m², and with 37.5% energy-saving rate.

As can be seen from the Fig.3.2, when winter air-conditioning set temperature decreased from 22°C to 18°C, building heating energy consumption reduced, and heating energy consumption of 16.45 kwh/m², with 41% energy-saving rate.

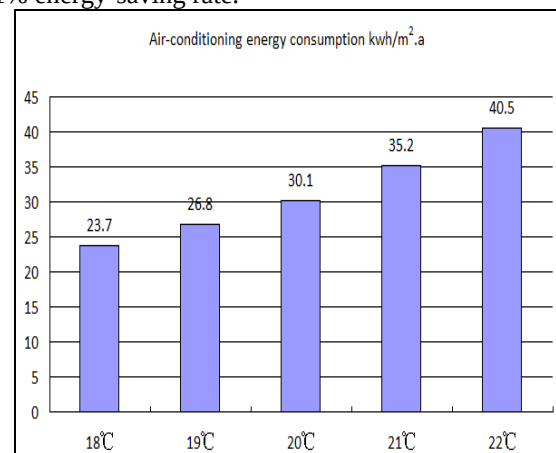


Figure.3.2 Heating energy consumption with different setting temperature in winter

By calculation and analysis above, you can see that improve the summer air conditioning set temperature and lower winter indoor heating temperature can obviously reduce energy consumption of building air

conditioning; it can have good energy saving effect. But now the office room temperature between 23°C to 24°C, the temperature of air conditioner less than the national office temperature of 25°C. Use of personnel of the public buildings, a general lack of awareness of energy conservation, the indoor temperature in summer are set very low, which is obviously unreasonable, resulting in a huge waste of energy. From the Angle of human body comfort, room temperature is less than 28°C can meet the requirements, if you can take steps to set the air terminal temperature not less than 25°C, it can produce very good energy saving effect.

3.3. Choice of hot and cold source

3.3.1. Geothermal energy

The office building is characterized by high daytime usage rate, the rest periods utilization is very low or even zero, under normal circumstances, night and holiday usage to zero. This characteristic of architectural determines the load persistent of weak. Geothermal energy is now strongly encourage low-grade energy, soil as a huge accumulator body, reservoir of cold winter for summer access, and the summer heat reservoir for winter access. Ground-source heat pump belongs to the more energy-efficient way of cooling and heat sources. The main factor about the impact of a ground source heat pump system is heat exchange performance between ground heat exchanger with the surrounding soil. For air conditioning systems with intermittent operation mode, the recovery of soil around the buried tube heat exchanger has the certain time; it enables the heat exchanger can always efficient heat exchange, and can make the air conditioner for a long time and stable operation.

3.3.2. Solar energy

The annual sunshine time generally of more than 2000 h, even as high as 2800h to 3300 h in Shenyang, while the solar energy utilization is advantageous. Solar energy can be used as auxiliary heat source, and used in combination with other hot and cold source. In addition, office building of air conditioning system that not running at night, especially the local office building with time-sharing electricity price policy, considering ice (water) energy storage is an important technology of energy saving.

3.4. Optimize the system mode of operation

3.4.1. Intermittent operation of air conditioning system

According to the office staff work and rest time, the working hours is 9:00 to 12:00 and 14:00 to 17:00, then lunch break is 12:00 to 14:00, while person is relatively small in the room. So we can consider adopting the intermittent operation mode, shut off the cooling unit at noon until the afternoon to work, so that we can save energy consumption.

From the comparison we can see in Fig.3.3, we adopt the intermittent operation decreased energy consumption is 4 kwh/m² than continuous run, with

10.5% energy-saving rate. At the same time, considering the effect that intermittent operation for indoor temperature, we select a typical office room on 3rd floor, and analyze the hourly temperature on July 15.

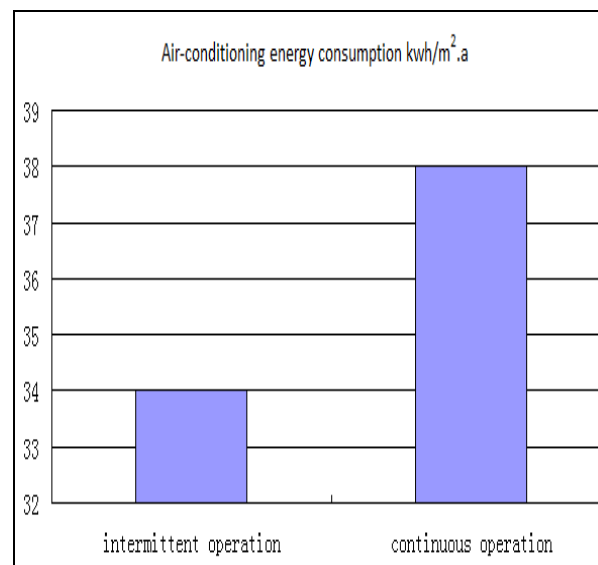


Figure.3.3 Comparison of energy consumption of different operation modes

As can be seen from Fig.3.4, shutdown the cooling unit at noon, the indoor temperature rose from 25°C up to 28.2°C, increased by 3.2°C. Although beyond the comfort of human body temperature slightly, but because the indoor personnel little at the moment, and time duration can be within the scope of the bear. In actual operation, host and the cooling water pump can be taken off in the noon break time, and chilled water pump and fan coil can continue to run.

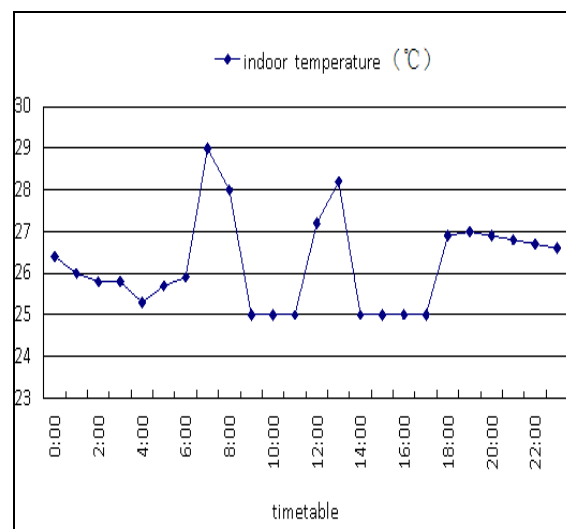


Figure.3.4 Indoor temperature of intermittent operation

3.4.2. Standardized management

Building energy management requires a legal basis; the status of management of building energy is

not high in office buildings, especially in government office. Energy saving strategy will be affected by the intervention of leadership, or building energy management personnel quality is not high, that don't know how to do to achieve the purpose of energy saving. It needs to establish a formal energy conservation laws and regulations to solve it. At the same time, it should be introduced a building energy efficiency incentives. Incentives can't instead of building energy efficiency standard, but can play a powerful role. We should reward those who exceed energy saving standard units, and to encourage the new energy conservation technology, and promote the development of energy-saving technologies.

4. CONCLUSIONS

In this paper, three conclusions were obtained as followed.

1. Reasonable setting of indoor environment parameters, then simulate the energy consumption respectively of air conditioning ,the summer set temperature are 23°C , 24°C, 25°C, 26°C and 27°C,the winter set temperature are 18°C,19°C,20°C,21°C and 22°C. Analysis to improve the summer air conditioning set temperature and lower winter indoor heating temperature can obviously reduce energy consumption of building air conditioning; it can make a good energy saving effect. Therefore, according to the regulations of the "public building energy efficiency design standard" then air conditioning set temperature is 25°C in the summer, and heating temperature is 20°C in winter, to reduce energy consumption demand of system.

2. The design of energy-saving in air-conditioning system, select renewable energy, such as geothermal energy, solar energy as cold and heat source, have great potential for energy saving.

3. Optimization of system operation mode, such as setting up reasonable intermittent operation of air conditioning system, and standardized management, reduce system energy consumption.

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STUDY ON ADAPABILITY OF DIFFERENT AUXILIARY HEAT SOURCES IN RESIDENTIAL SOLAR WATER HEATING SYSTEM IN CHINA

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ABSTRACT

The solar water heating system (SWH) has the potential to reduce household energy usage substantially and also provide external benefits of reducing emissions of CO₂ and other pollutants. But SWH is unstable which needs auxiliary heat source to meet continuous hot water demand. There are various types of auxiliary heat sources applied in SWH, such as electricity heater, town gas heater, heat pump and oil-fired heater. Despite previous studies have proposed the suitable solar fraction in different regions of China, the auxiliary heat source form in different regions has less discussion, which should be somewhat different due to geographic variations and local conditions. This paper analyzes the adaptability of the auxiliary heat source in different regions, which helps to guide the applications of SHW in China.

In this paper, the optimized solar fraction and regional applicability of auxiliary heat source from aspects of energy efficiency, economic performance and carbon emissions during the life cycle in the typical cities in China are discussed by using Geographical Information System (GIS) mapping techniques and RETScreen software. The residential solar water heating system base model is developed, provincial capital cities with geographic variations in the solar resource, supply water temperature, local energy price, hot water usage, and available incentives are selected for analysis.

The results show that the suitable auxiliary heat source form and optimized solar fraction vary in different regions. For example, town gas boosted SWH is appropriate for most regions especially in the west region of China. Meanwhile, it is better to choose electricity boosted SWH in the east part including Shanxi, Shandong, Hebei and Tianjin province. In addition, heat pump boosted SWH is a good choice in the southern coast region. This study may be useful in guiding government decisions and providing suggestions on the design for the solar thermal system.

Keywords: Solar water heating system; Auxiliary heat source; Geographical Information System; RETScreen.

1. INTRODUCTION

Solar water heating has the potentials to reduce household water heating electricity consumption, which also provide benefits of reducing emissions of CO₂ and other pollutants. However, the solar water heating system is unstable, and needs auxiliary heat source to overcome the shortcomings. Hannah Cassard et al. [1] examined the regional, technical, and economic performance of residential rooftop solar water heating technology in the U.S.A. N.Aste et al. [2] proposed a quick and easy-to-use graphical tool to size and assess the performance of solar thermal system. Hong Li et al. [3] analyzed the evaluation results of conventional solar water heater systems and solar heat pump system for hot water production in Hong Kong. Though previous studies have put forward the suitable range of solar fraction, the auxiliary heat source form in different regions is not taken into consideration.

There are a variety of various types of auxiliary heat sources. In this paper, electricity, town gas, heat pump and oil are selected as auxiliary heat source which

are called electricity boosted SWH, town gas boosted SWH, heat pump boosted SWH and oil boosted SWH.

This paper analyzes the optimized solar fraction and regional applicability from the aspects of energy efficiency, economic performance and carbon emissions estimation during the life cycle in the typical cities in China by using Geographical Information System (GIS) mapping techniques and RETScreen software. Based on the results, suggestions are given on the selection and design for the solar water system.

2. TECHNICAL AND ECONOMIC EVALUATION

Energy demand for hot water heating is a function of climate conditions, inlet water temperature, fuel price, house size, and usage patterns of the residents. In order to determine the annual amount of energy required for water heating, we used the RETScreen, an Excel-based tool dedicated to the pre-design of renewable energy system developed by the Natural Resources Canada. Detailed discussion of the solar water heating system is provided below.

NOMENCLATURE

F	solar annual load fraction	Q _i	solar thermal energy per year (MJ)
Q _Z	heat load for DHW production (MJ)	L	total cost per year (RMB)
L ₁	plant specific cost per unit of installed surface (RMB/m ²)	L ₂	tax fee (RMB)
L _{aux}	auxiliary plant cost (RMB)	L ₃	fuel cost per year (RMB)
N	lifespan of the solar system(year)	C	the rate of CO ₂ reduction
S	area of the surface of the panels (m ²)	E _b	CO ₂ emission of electric water heater
E	CO ₂ emission of solar water heating system (t)	PBT	simple payback time(year)
DHW	domestic hot water	C _i	initial cost (RMB)
		C _s	cost saving per year (RMB)
		SWH	solar water heating system

2.2 Evaluation methods

In this section, details of the evaluation procedure are described as follows:

Firstly, the solar fraction is defined by the following equation:

$$F = \frac{Q_j}{Q_z} \times 100\% \quad (1)$$

Secondly, the total annual cost within life cycle is given by:

$$L = (L_1 \times S + L_2 + L_{aux} + L_3 \times N) / N \quad (2)$$

Thirdly, the CO₂ reduction rate calculation is based on the savings compared to the use of an electric water heater, which is estimated by:

$$C = \frac{(E_b - E)}{E} \times 100\% \quad (3)$$

Finally, the simple payback time is then evaluated:

$$PBT = \frac{C_i}{C_s} \quad (4)$$

2.3 Base case model

A base case for a domestic hot water system (DHW) is defined for the following analysis. The DHW is mainly composed of a SWH and an auxiliary system. The hot water demand for a typical household is 100L/day, the hot water temperature is set as 55°C. Collectors are assumed to be evacuated tube collector with gross area 2.38m² per solar collector, aperture area 1.34 m² per solar collector. The technical characteristic F_r is 0.42. For the base case, we assumed the collectors are south-facing with a tilt angle of local latitude in order to receive the maximum solar radiation, the storage tank capacity is 80 L/m² per collection area, and the heat loss coefficient is 16.19W/(m²K). The power consumption of the circulation pump for the plant designed is assumed to be 5W/m² per collection area. The lifespan of solar system is assumed to be 10 years.

For the economic analysis, the initial cost of each auxiliary is determined by market survey. In this study, it is assumed that the solar collector specific cost per unit of installed surface is about 750RMB/m², the initial

investment of electric auxiliary water heater is 1000RMB/two-person family, the town gas auxiliary water heater is 2645 RMB/family, oil-fired auxiliary water heater is 2138 RMB/family, heat pump auxiliary water heater is 3150 RMB/family.

2.4 Regional variation in climate conditions

It is well known that China has abundant solar energy resource, but the solar radiation intensities in different regions vary from the minimum of 3058 MJ/(m²·a) in Chongqing to the maximum of 7331 MJ/(m²·a) in Lhasa, as shown in Fig.1.

The regions among China are divided into four classes according to local solar radiation intensities. (see Table1).

Table1. Annual solar radiation distribution

Division	Solar radiation/[MJ/(m ² ·a)]
I	≥6700
II	5400-6700
III	4200-5400
IV	<4200

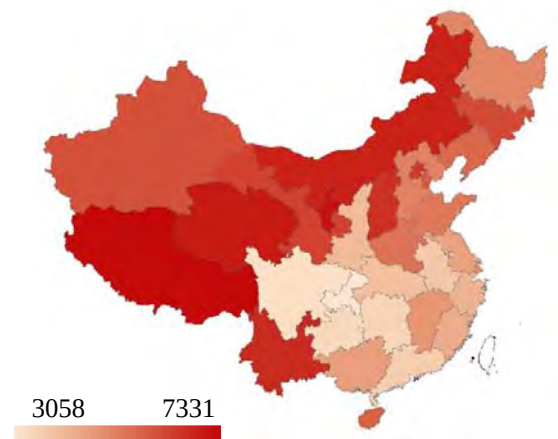


Figure.1. Regional variation in solar radiation [MJ/(m²·a)]

2.5 Regional variation in fuel price

The fuel prices of electric, towngas and oil used in the auxiliary heat sources is different, which will play an important role on the operation cost for SWH.

The regional distributions of fuel price including electricity, towngas and oil in each provincial capital city of China are illustrated in Fig.2~4.

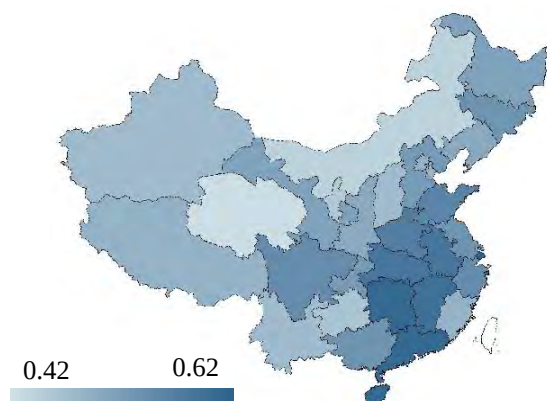


Figure.2. Regional variation in electricity price (kWh/RMB)

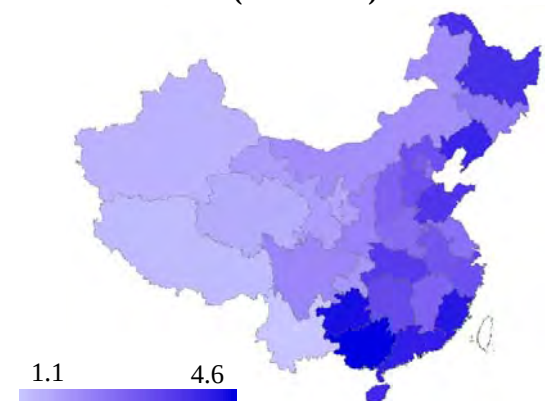


Figure.3. Regional variation in towngas price (m³/RMB)

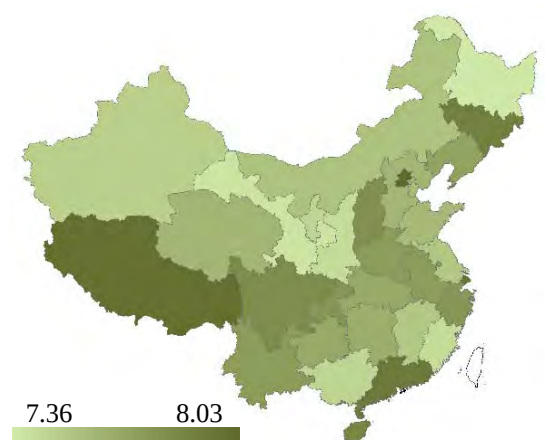


Figure.4. Regional variation in oil-fired price (L/RMB)

3 CASE STUDY

A case study is carried out for the technical and economic performance analysis of a residential solar water heating systems with different auxiliary heat sources. The total cost in lifecycle, optimal solar fraction, CO₂ reduction rate, simple payback time and suitable auxiliary heat source are discussed.

The total cost in life cycle of DHW includes the initial cost of the plant and fuel charge during the operation. Assuming a lifespan of 10 years of the solar system, the total annual cost has been calculated for the solar thermal system described below.

3.2 Total cost in life cycle

Take Shanghai as an example, in order to determine the economic performances of a solar thermal system for DHW production, the total cost in life cycle is discussed below.

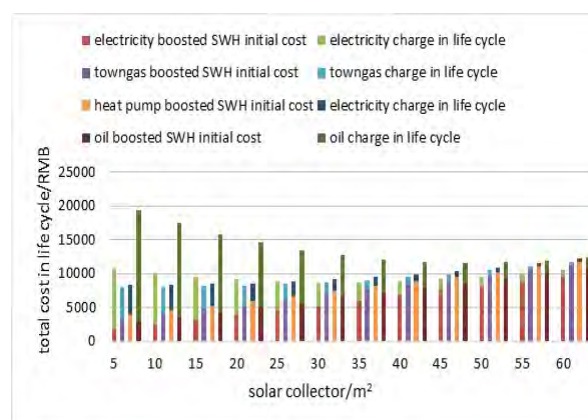


Figure.5. Effect of total cost in lifecycle on solar collector

Fig.5 presents the total cost in life cycle for the base DHW with different auxiliary heat sources. It is clearly shown that the total initial cost increases with the increasing solar collector area which plays a key role on the total cost. The results also show that with the increment of the solar collector, the total cost decrease first and then increase which result in the minimum cost in the life cycle.

3.3 Optimal solar fraction

Since the optimal solar fraction is affected by the total annual cost within life cycle to a large extent, the optimal solar fraction corresponds to the one that minimizes the total annual cost of the system as a whole described above.

Fig.6 presents that the solar fraction is obviously increased with the increment of the solar collector area. The results also show that the solar thermal system with different auxiliary heat source have different optimal solar fraction, oil boosted SWH has the highest annual cost because of higher fuel price, towngas boosted SWH has the lowest annual cost owing to the lower fuel

price. So towngas water heater is appropriate auxiliary energy source for Shanghai. However, for each auxiliary heat source, the optimal solar fraction is different, which are 68%, 40%, 28% and 84% for electricity boosted SHW, towngas boosted SHW, heat pump boosted SHW and oil boosted SHW. The analyses of other cities in China are similar to Shanghai.

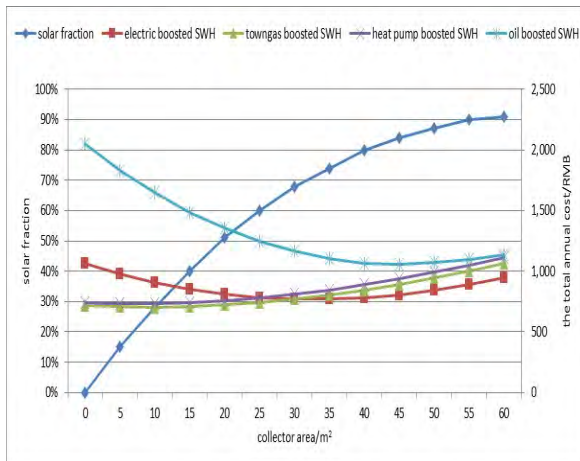


Figure 6. Effect of solar fraction on the solar collector and total annual cost

3.4 The CO₂ reduction rate

Since the environmental pollution depends on the amount of energy consumption and the process of energy conversion, it is reasonable to assess different hot water systems' global warming impacts through evaluating the amount of CO₂ produced by these hot water systems during their lifecycle. The CO₂ reduction rate is highly sensitive to the fuel consumption and the carbon emission factor of the fuel type. The lower the fuel consumption and carbon emission factor is, the higher the CO₂ reduction rate is. In China, the average CO₂ emission factor of electricity is 0.859, the average CO₂ emission factor of towngas is 0.4483, and the average CO₂ emission factor of oil is 0.5921 from IPCC2006 data.

As illustrated in Fig.7, based on the savings compared to the use of an electric water heater, the CO₂ reduction rate is increased with the increment of the solar collector area. Despite electricity boosted SHW has lowest CO₂ reduction rate, CO₂ decrease evidently as the solar collector increase. Towngas boosted SHW has the highest CO₂ reduction rate.

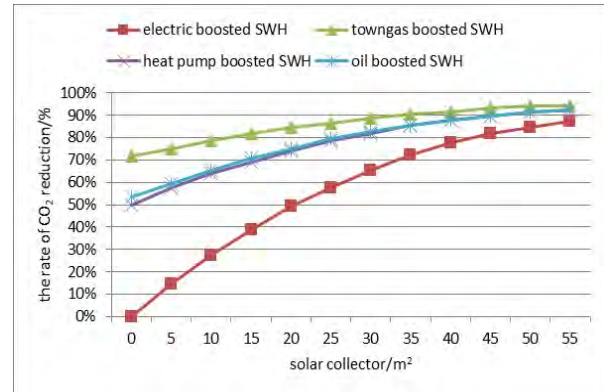


Figure 7. Effect of solar collector on the rate of CO₂ reduction

3.5 The simple payback time

The simple payback time is defined as the length of time required to recover the initial investment of the solar thermal system. The shorter the length, the more economically attractive to investors the solar thermal system is.

Fig.8 presents the effect of solar collector on the simple payback time. The PBT of four different water heating systems are all around 5-10 years due to high initial investment.

It is clear that with the increment of the solar collector, the PBT of electric boosted SHW decrease sharply first and then increase as the solar collector greater than 15m² which result in the minimum PBT of 5.7 years. However, the PBT of heat pump boosted SHW and towngas boosted SHW always increase since the initial costs grow faster than the annual cost savings. It also can be noticed that oil boosted SHW have the longest payback time due to high investment cost and fuel price.

As can be observed, considering relatively long payback time, the government should take some measures to help making the price of the SHW systems more affordable to the customers.

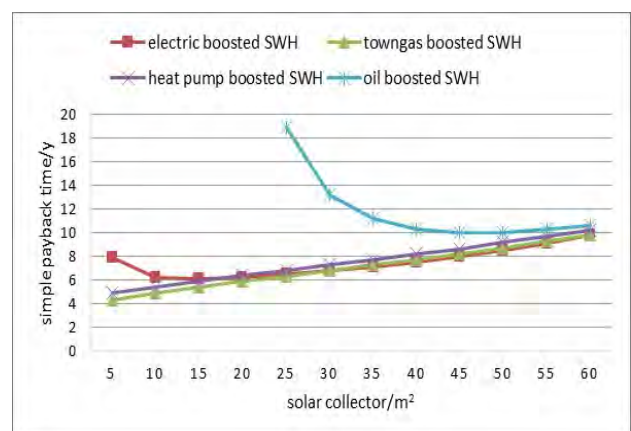


Figure 8. Effect of solar collector on the simple payback time

3.6 Optimal auxiliary heat source

For a country like China with significant spatial variation in climate conditions, fuel price, inlet cold water temperature, the optimal auxiliary heat source may be different for different geographic locations. Using the simulation data from RETScreen tool and Geographical Information System (GIS) mapping techniques, we obtain the optimal configuration solar thermal system with minimum total annual cost in each typical city, Fig.9 illustrates the regional distribution of auxiliary heat source for thirty-one provincial capital cities in China.

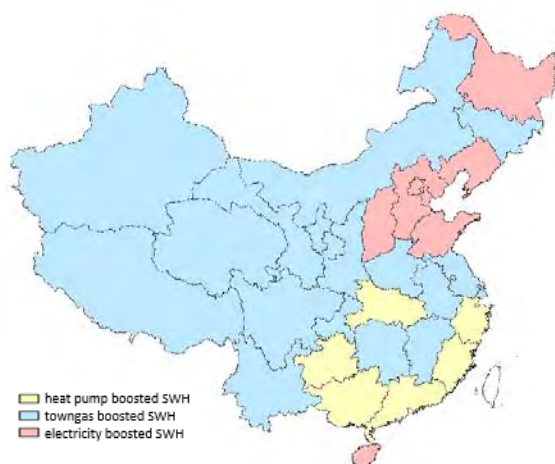


Figure.9. Regional variation in auxiliary heat source

Towngas boosted SWH is appropriate for most regions especially in the west region such as Xizang, Xinxiang, and Sichuan province due to lower towngas price which result in the lower total annual cost.

Meanwhile, due to low solar radiation, it mainly depends on the auxiliary heat source to meet the demand for hot water usage in the southern coast region such as Guangzhou, Fujian and Zhejiang province. With high towngas price and low electricity price, heat pump boosted SWH with high efficiency is a good choice.

In addition, it is better to choose electricity boosted SWH in the northeast part including Shanxi, Shandong, Hebei and Tianjin province because of the lower initial cost compared to heat pump system.

4. CONCLUSIONS

In an attempt to select suitable auxiliary heat source in typical cities of China, the evaluation of the technical and economic performance are tested using three indicators: total annual cost within the lifecycle, simple payback time and CO₂ reduction rate.

Since the simple payback time of the solar thermal system is all around 5-10 years described above, it is recommended that the government should adopt supports policies that motivate customers to use solar thermal system. From the analysis on the CO₂ reduction rate of different solar thermal systems, it is concluded

that the towngas boosted SWH system has the highest CO₂ reduction rate.

This paper also demonstrates that the suitable auxiliary heat source form and optimized solar fraction vary in different regions. For example, towngas water heater is appropriate auxiliary heat source for most regions especially in the west region, meanwhile, it is better to choose electricity heater in the northeast part including Shanxi, Shandong, Hebei and Tianjin province. In addition, heat pump is a good choice in the southern coast region.

ANOWLEDGEMENTS

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A STUDY ON APPLYING VENTILATED DOUBLE-SKIN FACADE TO A BUILDING DURING TRANSITION SEASONS IN COLD ZONE OF CHINA

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ABSTRACT

It is crucial for ventilated double-skin facade (DSF) compared with single-skin facade to avoid high energy consumption, optic pollution and inadequate ventilation problems. In this paper, the performance of ventilated double-skin facade applied to a building located in a university of cold zone (Tianjin) of China is tested, in order to investigate the temperature distributions of both the ventilated air cavity of DSF and indoor space with different experimental modes in transition seasons. The results show that the temperature of the ventilated air cavity of DSF mainly depends on the intensity of solar radiation; It's more reasonable that the bottom air-vent of external facade and the air-vent of internal facade should be opened during the daytime; The higher the height of façade is, the higher the air temperature of the ventilated air cavity of DSF and the indoor space will be, the users' needs can be better satisfied.

Key words: Ventilated double-skin façade; Temperature; Transition seasons.

1. INTRODUCTION

Ventilated double-skin facade is a new type of building envelope, and it was very popular in the construction industry because of its thermal insulation in winter and summer since the 1970s. The ventilated double-skin facade began to be used as building envelope since entering the 21st century in our country. Especially in recently years, building energy efficiency has no time to delay in this special period, the use of ventilated double-skin facade plays a vital role.

Although we have adopted the relative mature experience from foreign technology, the advantages of ventilated double-skin façade may be not shown because of the differences of climate. Furthermore, the unreasonable use-pattern can cause the great waste of energy. H. Manz [1] mentioned that not taking reasonable ventilating-pattern about the ventilated double-skin facade causes the temperature of the ventilated air cavity and indoor space high, which can't reduce the energy of conditions finally. Many domestic and abroad scholars made a lot of studies about the ventilated double-skin facade. Mingotti Nicola [2] makes it possible that preheating the room in the winter to the utmost and preventing the temperature of the room and ventilated double-skin façade too hot in the summer through adjusting the air-vents size and the height of the facade. Tanaka Hiroaki [3] received that the effect of natural ventilation and the position of shading devices has a great influence on the distribution of temperature in the ventilated air cavity of DSF in summer through experimental tests. The field testing and simulation analysis about the ventilated double-skin

façade are most for the summer and winter [4], and rarely involves the transition seasons. Therefore, the research about the ventilated double-skin facade influencing on indoor thermal environment in transition seasons is of great significance.

Represented by cold area [5], this paper selects an actual building with ventilated double-skin facade as base, sets the experiment system about the ventilated double-skin facade influencing on indoor thermal environment. Also the paper was based on the experiment research about the solar radiation intensity, the use of airflow windows of both external and internal facade and the height of DSF on the indoor thermal environment in transition season, the experimental results were compared and analyzed.

2. EXPERIMENTAL SUBJECT

The room which was tested is located in a laboratory building in Hebei University of Technology of Tianjin. The room tested is situated on second to fourth floor and the south external envelope structure of the room adopts ventilated double-skin facade. The outer facade is single layer coated glass, the inner facade is 12mm thick ordinary double-layer insulating glass. The width of the ventilated air cavity of facade is 800mm, there exists iron perforated strainer between the layers of the air cavity of facade. When conducted the test of the independent room, we used benzene board as partition material between the floors.

3. EXPERIMENTAL SCHEME

3.1. Experimental period

The time of Experimental test is divided into spring and autumn. The time of the test in spring is from March 18, 2012 to April 30, 2012, and the time in autumn is from October 15, 2011 to November 20, 2011, a total of 78 days.

3.2. Experimental contents

Experiment contents include:

1. The influence of solar radiation intensity and outdoor temperature on the ventilated air cavity of DSF;
2. The influence of the use of air-vents on the temperature of the ventilated air cavity of DSF and indoor room;
3. The influence of the facade height on the temperature of the ventilated air cavity of DSF and indoor room.

4. DATA ACQUISITION

4.1. Experimental apparatus

The temperature of the ventilated air cavity of DSF and indoor air measured by Aglient34970A data acquisition instrument, T thermocouple and computer, data acquisition 1 times/min. Outdoor temperature, solar radiation intensity are collected through the TRM - ZS2 automatic meteorological collection device, data acquisition 1 times/min.

4.2. Monitoring point arrangement

The measuring point layout on the ventilated air cavity of DSF: in vertical direction, the distance from the measuring point in the ventilated air cavity of DSF

on second and third floor to the bottom of their facade are 0.8m and 2.8m, the distance from the measuring point in the ventilated air cavity of DSF on

fourth floor to the bottom of facade are 0.8m, 1.8m and 2.8m; On the horizontal direction, the distance from the measuring point in the ventilated air cavity of DSF on second, third and fourth floor to their external facade are 0.2m, 0.4m and 0.6m; The measuring point layout on indoor: the distance from the measuring point in the room on second and fourth floor to their ground are 1.5m; the distance from the measuring point in the room on third floor to their ground are 0.2m and 1.7m. The specific arrangement of the measuring point in the ventilated air cavity of DSF and indoor is show in figure 1.

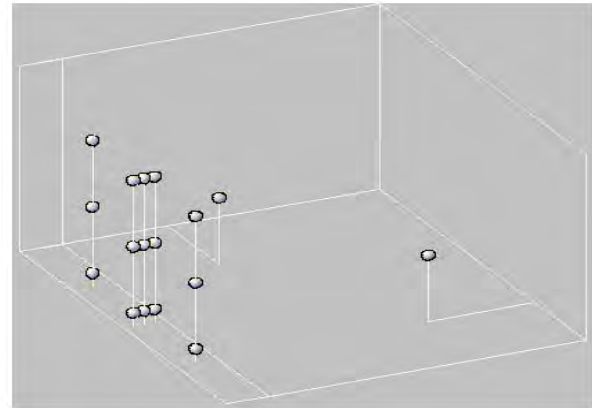


Figure 1 Layout of the measuring point

5. EXPERIMENTAL RESULTS AND ANALYSIS

5.1. The influence of solar radiation intensity and outdoor temperature on the ventilated air cavity of DSF

Figure 2 show that the temperature changes on vertical direction of the ventilated air cavity of DSF under different solar radiation intensity and outside temperature.

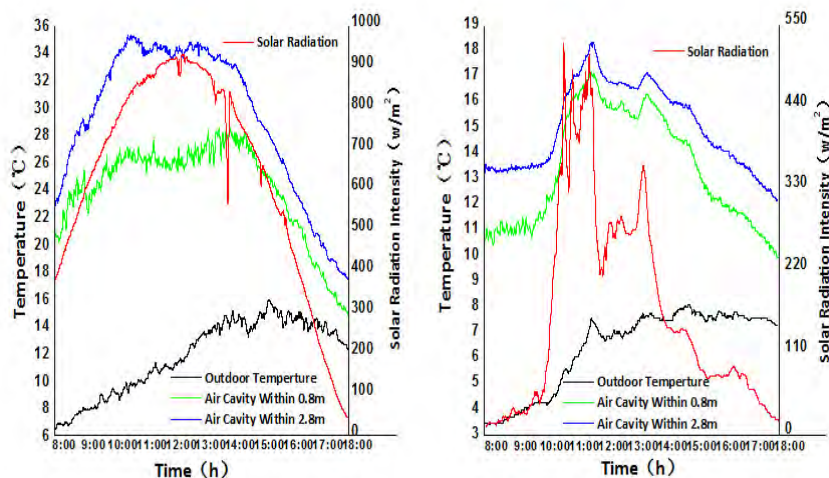


Figure 2 Temperature distribution of cavity with cloudy weather or sunny weather
(Left: sunny weather, Right: cloudy weather)

The figure 2 shows that cloudy and sunny day, the maximum temperature of the ventilated air cavity of DSF occurred at about 12:00, the same as the solar radiation intensity, however, the maximum temperature of outdoor occurred at about 13:00. Before 12:00, the temperature of the ventilated air cavity of DSF rise gradually, the same as the solar radiation intensity and outdoor temperature; Temperature is gradually reduced in the ventilated air cavity of DSF after 12:00, also gradually reduce the intensity of solar radiation, outdoor temperature, however, increased or maintained in maximum again. So the temperature of the ventilated air cavity of DSF change trends and the solar radiation intensity changes in basic consistent. Through an analysis of the experimental measured data processing can be concluded that two days, the ventilated air cavity of DSF within 0.8m temperature difference is 10.67°C, and 2.8 m in difference in temperature is 14.28°C, the sun radiation intensity difference is 478.72w/m², the average outdoor temperature difference value of 5.43°C. Cloudy and sunny day, the average outdoor temperature difference compared with the average solar radiation difference almost can be ignored, but it led to the temperature difference of the ventilated air cavity of DSF up to 10°C. So the influence of the temperature of solar radiation intensity within the air cavity effect is greater than the outdoor temperature on the air cavity of temperature effects.

5.2. The influence of the use of air-vents on the temperature of the ventilated air cavity of DSF and indoor room

5.2.1 The influence of the internal facade air-vent when it's open or not on the indoor temperature

Figure 3 show that the indoor temperature changes in two rooms (one of which internal facade air-vent open and the other closed).

The figure 3 shows that regardless of the internal facade air-vent open or closed, both room indoor temperature changing trends is relatively flat, but you can clearly see that the indoor temperature of room which internal facade air-vent open throughout the day were higher than the indoor temperature of the room which internal facade air-vent closed. Through an analysis of the experimental measured data processing can be concluded that the internal facade air-vent open room indoor average temperature is 22.03°C, the internal facade air-vent closed room indoor average temperature is 19.83°C, the temperature difference is 2.2°C. So we should open the internal facade air-vent, so that we can make the high temperature gas heated by solar radiation inside the ventilated air cavity of DSF

and indoor low temperature gas for heat exchange, finally achieve the goal of improve indoor temperature.

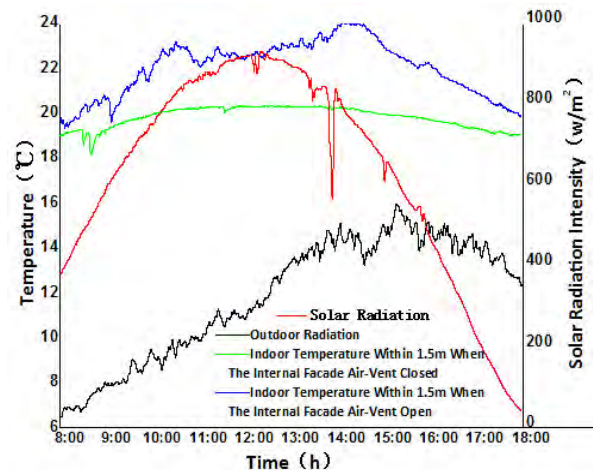


Figure 3 Indoor temperature when air-vent is open or not

5.2.2. The influence of the external facade air-vent when it's open or not on the air cavity temperature and indoor temperature

Figure 4 show that the indoor temperature and temperature in air cavity when air-vent is opened with different style

The figure 4 shows that opening air-vent at the top of the room, the air cavity within 0.8 m above 2.8 m in temperature, this is because the position of the top air-vent basically the same as the measuring point within 2.8 m in height position, so the air cavity temperature within 2.8m keep the same as outdoor temperature. The rest of the two trends of temperature in air cavity keep the same, their temperature are increased with the increase of height. What's more, the temperature of the air cavity and indoor in the room which air-vent at the bottom of the façade open is significantly higher than the other two rooms. Through an analysis of the experimental measured data processing can be concluded that: the average temperature of the air cavity and indoor in the room which air-vent at the top of the façade open is 21.77°C and 20.16°C, the average temperature of the air cavity and indoor in the room which air-vents at the top and bottom of the façade open is 22.32°C and 20.59°C, two room temperature are almost the same. however, the average temperature of the air cavity and indoor in the room which air-vent at the bottom of the façade open is 25.63°C and 23.02°C, significantly higher than the other two rooms. Therefore, the way opening the vent at the bottom of the façade is more reasonable.

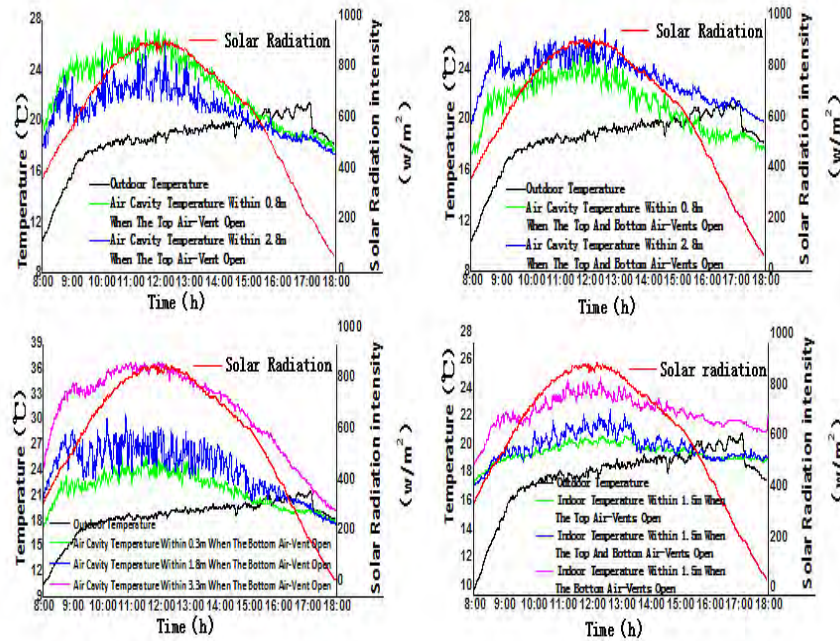


Figure 4 Indoor temperature and air temperature in cavity when air-vents is opened with different style

5.3. The influence of the facade height on the temperature of the ventilated air cavity of DSF and indoor room

Figure 5 shows that air cavity temperature and indoor temperature in two rooms (one of which facade height is 3.6 m and the other is 7.2 m).

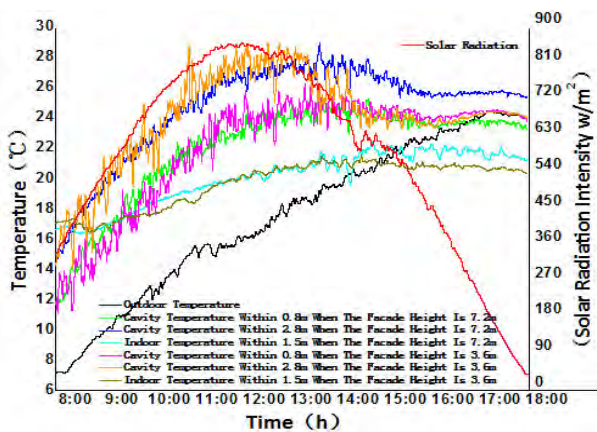


Figure 5 Air temperature in the cavity and room when the height of cavity is 3.6m or 7.2m

The figure 5 shows that before 13:00, the temperature changes of air cavity and indoor in two rooms are basically consistent. However, after 13:00, the temperature of air cavity and indoor in the room which facade height is 7.2m is significantly higher than that of the room which facade height is 3.6 m. Therefore, on the whole, the room which facade height is 7.2m is higher than the room which facade height is 3.6m on air cavity temperature and indoor temperature.

Through an analysis of the experimental measured data processing can be concluded that the room which facade height is 3.6m air cavity average temperature and indoor average temperature is 23.15 °C and 19.69 °C, the room which facade height is 7.2m air cavity average temperature and indoor average temperature is 23.36 °C and 20.06 °C. Therefore, the higher the height of the facade is, the higher the temperature of the air cavity and indoor room.

6. CONCLUSIONS

In transition seasons, the use of the ventilated double-skin facade has a great influence on the indoor thermal environment, so reasonable use the ventilated double-skin facade to improve the indoor thermal environment plays a vital role. This paper, taking a building with the ventilated double-skin facade in cold area as the research object, through the experiment method to study the influence of the intensity of solar radiation, the use of air-vents of both external and internal facade and the facade height on indoor thermal environment, receive the following conclusions:

1. The temperature of the ventilated air cavity of DSF mainly depends on the intensity of solar radiation
2. It is more reasonable that the bottom air-vent of external facade and the air-vent of internal facade should be opened during the daytime. So that both can improve the indoor thermal environment and meet the demand of users on fresh air.
3. The higher the height of facade is, the higher the air temperature of the ventilated air cavity of DSF and the indoor space will be, the users' needs can be better satisfied.

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THE RESEARCH OF THE EXPERIMENTS OF THE INFLUENCING FACTORS OF OPENING TIME RATIO FOR THE HEAT USERS

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ABSTRACT

Based on the on-off time and area method of on-off heating mode for the heat allocation, the opening time ratio of on-off valve is the main basis. In order to research the influences of opening ratio, this paper make the experiments to find the influences by using the on-off time and area method. Through the heating system we got the quantitative data of the solar radiation, the set indoor temperature, the temperature difference between indoor and outdoor and the radiator areas which all can affect the opening ratio. Therefore, through these to enhance the fairness of heat allocation and it is of great significance to the development of on-off heating mode applications.

Keywords: on-off valve regulation; opening time ratio; heat allocation; experiment.

1. INTRODUCTION

The on-off type of heat metering is a promising metering mode for its suitability to our country. The on-off time and area method not only solves the problems of the fairness of heat metering and the device block, but also can realize the control, the adjustment and the metering of households^[1]. Nowadays, this method has become one of the four methods of household heat metering in <<Heating metering and technical regulations>> (JGJ 173-2009), and it already came into the stage of practical applications^[2].

The working principle of on-off system is firstly to set the indoor set temperature, and then the thermostat changes the temperature signal into the electric signal and sends to the heat distribution meter. Finally, by comparing the set temperature and the actual temperature, the meter controls the valves of the temperature controller. When the actual temp is below the set temp, valves open; when the set temp is lower, valves closed. The distribution meter records the times of valve actions, and the opening time ratio of heat users is calculated.

Nowadays a new way to the heat metering is the on-off time and area method which suits our country very much^[3]. This method can be used intuitively and realize the functions of controlling the indoor temperature, so it can be used in the indoor resistance unchanged heating system which has separated ring pipelines for each house. However, this measurement is not for the heat given by the heating system but the sharing heat through the on-off time, so the difference makes the result inaccurately^[4]. This method is proved

to be a good temperature controller after practice, but there comes problems too, such as the accuracy and the fairness of heat allocation and the unfairness of different locations in the buildings^[5]. The most important basis of heat allocation is the opening time ratio which directly related to the fairness of users^[6]. In this paper, experiments are carried out to test some influences of the opening time ratio and the experimental data are analyzed and summarized.

2. EXPERIMENTAL SYSTEM AND TEST

We pick two next rooms which have the same areas in an office on the second floor of Hebei University of Technology as the experimental rooms. They are used to simulate the users who have different load because of the building envelope. Room 1 is located in the top of the building, and the external wall and window are facing south. The other three sides are interior walls. Room 2 is located in the end of the building, and one more westward external wall than room 1. Specific parameters are shown in Tab.2.1. The on-off experimental simulated system is shown in Fig.2.1

In the two circulatory systems each has three groups of radiators which are numbered one two three. Both of the No.1 and No.2 have 8 pieces, and the distance of supply and return pipe center is 600mm. The No.3 has 5 pieces and the distance is 500mm. Between the adjacent radiators there equipped with vales which control the circulation path of the heat medium. Each group of radiators can be connected or not.

Table 2.1 The basic parameters of the experimental rooms

Experimental rooms	Geometric dimensions (length×width×height)(m)	Size of southward external walls(m)	Size of westward external walls(m)	Size of southward external windows (width×height) (m)	The thickness of external walls (mm)
Room 1	9.08×3.1×3.4	3.1×3.4	—	2.33×2	370
Room 2	9.08×3.1×3.4	3.1×3.4	9.08×3.4	2.33×2	370

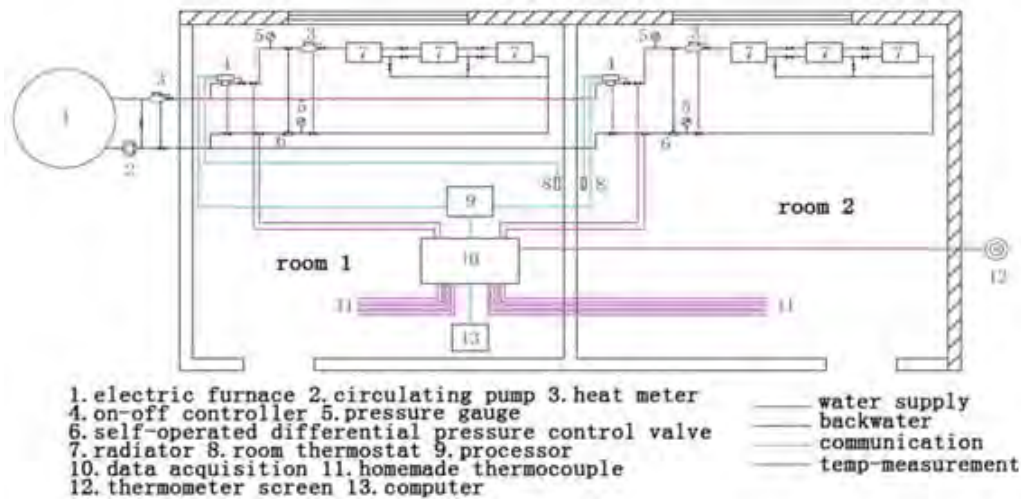


Figure 2.1 Experimental system

The experiments get the intensity of solar radiation in heating period through the ecological environmental monitoring system of automatic meteorological instrument in Hebei University of Technology. The system records 481 data in a day every 3 minutes. The level of total radiation means all of the received radiation in a fixed horizontal plane and the average daily level of total radiation data represent the daily solar radiation intensity. For residential customers, the indoor thermostat should be arranged in the area which controls easily, and the measured temperature represents the average temperature of the heating room.

Use the control variables method to compare the experiments. Change the different factors separately to find the influences of opening time ratio. The weather conditions, the changes of room temperature and the valve actions are recorded, and finally the influences are analyzed.

3. EXPERIMENTAL RESULTS AND ANALYSIS

Firstly select the stable conditions, and then set the indoor temperature. After that we measure and record the difference temperature of indoor and outdoor, and

connect the radiators according to the experimental contents. At last, change different conditions to see the differences.

Experiments were done from November, 2012 to May in 2013 and the data were chosen which had stable conditions. The indoor set temperature were ranging from 18~26. Whether the radiators matched the load or not, they were also the experimental conditions, so was the group of accessed radiators.

3.1 The opening ratio changes in relations with the solar radiation

The conditions which have the same set temperature, the same accessed radiators and the same temperature difference are carried out. Different solar radiation corresponding to the different opening ratio is recorded in Tab.3.1.

According to the Tab.3.1, each condition of each room has a group of comparative experiments, and the four conditions reflected the impact of solar radiation on the opening ratio. Comparing different radiation to different ratio in each group we get the Fig.3.1.

Table 3.1 The opening ratio in different solar radiation

Rooms	Conditions	Date of test	Radiator connection	Solar radiation(W/m ²)	Opening ratio(%)
Room 1	1	Jan 10	Group 1(match)	51.72	91.81
		Jan 11	Group 1(match)	113.02	74.38
	2	Mar 21	Group 1,2,3	171.67	86.94
		Mar 22	Group 1,2,3	129.90	95.56
Room 2	3	Feb 15	Group 1,2(match)	97.39	62.43
		Feb 18	Group 1,2(match)	183.87	24.17

4	Feb 23	Group 1,2,3	158.87	9.58
	Feb 26	Group 1,2,3	196.11	2.36

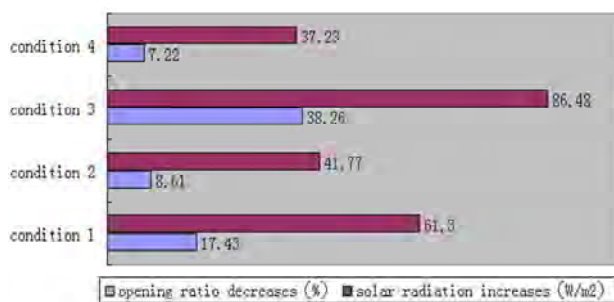


Figure 3.1 Influence of solar ratio to the opening ratio

As can be seen from Fig.3.1, the opening ratio of users who have the same heating state will be reduced when the solar radiation is enhanced. Because the solar radiation meets part of the demand for heating, so it is in line with the law of actual heating.

3.2 The opening ratio changes in relations with the indoor set temperature

Choose the same conditions except the indoor set temperature. The data of the two-room conditions are shown in Tab.3.2

Table 3.2 The opening ratio in different set temperature

Rooms	Conditions	Date of test	Set temperature(°C)	Radiator connection	Opening ratio (%)
Room 1	1	Feb 26	19.5	Group 1,3	1.04
		Mar 13	25.5	Group 1,3	48.75
	2	Jan 12	24	Group 1(match)	81.25
		Jan 8	25	Group 1(match)	96.46
Room 2	1	Feb 26	19	Group 1,2,3	2.36
		Mar 13	25	Group 1,2,3	71.32
	2	Jan 12	24	Group 1,2(match)	93.75
		Jan 8	25	Group 1,2(match)	100.00

As shown in Tab.3.2, four conditions are performed in accordance with the different set temperature which is the basis of measuring different opening ratio respectively. The difference of set temperature and the difference of opening ratio are compared in Fig.3.2.

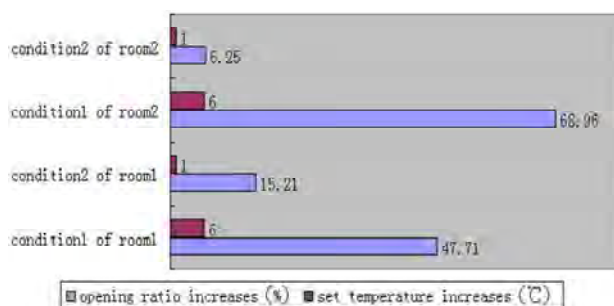


Figure 3.2 The influence of set temperature to the opening ratio

According to the Fig.3.2, the opening ratio is increased with the increase of indoor set temperature while the other factors remain unchanged. Due to the disadvantage location of room2, the coefficient of thermal reserve is lower and the heat lost easily, so the increase of opening ratio is larger than room1 with the increasingly set temperature. When the set temperature is higher than the maximum of heating capacity, the opening ratio reach 100% and is no longer increasing.

3.3 The opening ratio changes in relations with the temperature difference between indoor and outdoor

The temperature difference between indoor and outdoor is an important factor to affect the residential heating. Choose different conditions to make experiments for the temperature difference and record the data in Tab.3.3.

Table 3.3 The opening ratio in different temperature difference

Room	Conditions	Date of test	Radiator connection	Temperature difference(°C)	Opening ratio (%)
Room 1	1	Feb 10	Group 1(match)	19.61	43.47
		Feb 8	Group 1(match)	21.70	72.50
Room 2	2	Feb 10	Group 1,2(match)	19.61	73.75
		Feb 8	Group 1,2(match)	21.70	94.86

The data of Tab.3.3 shows there are two conditions, and each condition is divided into two

groups depending on the temperature difference. The influence to the opening ratio is shown in Fig.3.3.

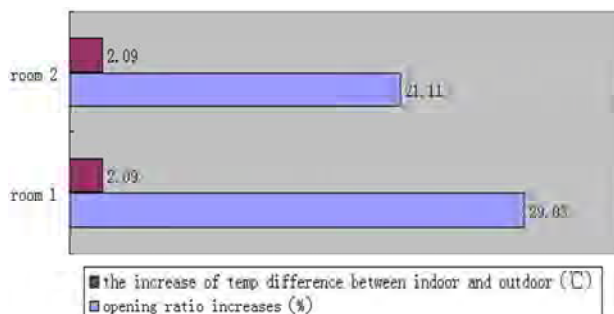


Figure 3.3 The influence of temperature difference to the opening ratio

When the solar radiation, the set temperature and the radiator connection are all basically the same, the greater temperature difference is, the higher opening ratio becomes. Even though the difference is small, there has a big influence to the opening ratio.

3.4 The opening ratio changes in relations with the radiator areas

Radiator area is an important factor to the opening ratio, and it is one of the main factors of practical engineering problems. Make a testing cycle which has the same set temperature, and adjust the radiator connections that one is match with the loads of room, the other is not. Record the different opening ratio in Tab.3.4.

Table 3.4 The opening ratio in different radiator areas

Room	Conditions	Date of test	Radiator connection	Opening ratio (%)
Room 1	1	Feb 16	Group 1(match)	35.97
		Feb 25	Group 1,3	15.29
	2	Mar 10	Group 1(match)	67.51
		Mar 12	Group 1,3	48.06
Room 2	3	Mar 7	Group 1,2(match)	70.83
		Mar 14	Group 1,2,3	65.83
	4	Mar 10	Group 1,2(match)	85.28
		Mar 12	Group 1,2,3	68.13

Tab.3.4 is the screening out data to reflect the conditions of changing radiator areas. The areas of radiator in room 1 are increased by 52.08%, and room 2 are 26.04%. Calculate the average of the opening ratio as a final result in order to make the calculation accurately. Comparative results are shown in Fig.3.4.

We can make the conclusion in Fig.3.4 that the increasing radiator areas cause the opening ratio reduced, and which lower the heat allocation so that decrease the heat cost. As areas increase by, the rate of decreasing will slow down gradually. When the radiator areas increased by 26.04%, the opening ratio reduced by 11.08%; When the radiator areas increased by 52.08%, the opening ratio reduced by 20.07%.

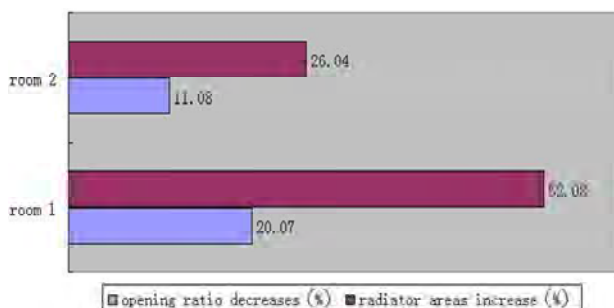


Figure 3.4 The influence of radiator areas to the opening ratio

4. CONCLUSIONS

In order to study the influencing factors of the opening time ratio, this paper builds the experimental system and makes the conclusions as follows:

1. Through the analysis of experimental data, we can find that the enhancement of solar radiation decreases the opening ratio;
2. When the indoor set temperature is increasing, the opening ratio will significantly increase;
3. When the temperature difference between indoor and outdoor is higher, the results show that the opening ratio also becomes higher;
4. When increasing the radiator areas, the opening ratio will decrease, and the rate of decreasing will slow down gradually.

ACKNOWLEDGEMENTS

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THERMAL CONTROL BY NATURAL AIR CONVECTION IN A DOUBLE-SIDED PV FAÇADE CAVITY BEHIND THIN-FILM AMORPHOUS SILICON SOLAR CELLS

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ABSTRACT

This paper studies the thermal performance of a double-sided PV (DSPV) façade with integrated thin-film solar cells by investigating the air flow characteristics in the air cavity. For a PV façade, undesirably high temperatures experienced by solar cells, due to the absorbed heat from solar radiation, will decrease their conversion efficiency. In addition, overheating in the closed glazing cavity, of a double-sided PV (DSPV) façade due to the inadequate air circulation may cause thermal discomfort to indoor occupants especially in severe summer conditions. Thermal regulation of the temperature rise, therefore, by fitting a naturally ventilated open channel in the back of the PV panel, can not only provide an effective means of heat release thereby enabling maximum solar to electricity conversion efficiency, but also help to reduce the heat gain of the building external envelope. To model fluid flow, natural convection flow in a 2D tall glazing cavity heated along one vertical side was investigated numerically. The flow in the façade cavity was considered to be either laminar or turbulent. Different inlet conditions for laminar flow were considered and various $k-\varepsilon$ turbulence models (RNG $k-\varepsilon$ model, Realizable $k-\varepsilon$ model, and Stand $k-\varepsilon$ model) were compared and analyzed in detail. Comparison indicates that the Realizable $k-\varepsilon$ model and Stand $k-\varepsilon$ model give similar velocity profiles at mid-height of the cavity. The Stand $k-\varepsilon$ model provides a more reliable simulation of the air convection in the cavity at the back of the PV panel. In addition, the thermal performance of the DSPV façade system was analyzed by experimentally comparing the reference façade with clear float glass. It was found that the maximum difference between the cavity air temperatures in the DSPV facade and the reference facade could reach 11°C. Consequently, a heat gain reduction for the DSPV façade can be achieved and thermal control of the DSPV façade for practical application should be considered.

Keywords: Air flow; double-sided PV (DSPV) façade; semi-transparent PV; amorphous silicon (a-Si) cell; thermal performance

1. INTRODUCTION

Development of green energy resources as alternatives to traditional fossil fuels has become important for the sustainable development of society. Photovoltaic (PV) technology is one of the most promising energy technologies among the wide variety of renewable or green energy contenders, i.e. wind energy, biomass energy, et al. throughout the world. A major barrier, however, which obstructs the development of the application of PV system, is the high price of the silicon cell. Development of the thin-film (TF) solar cell, however, provides a lower cost solution. TF is one of the newest PV technologies, whereby the TF modules are made by depositing a very

thin film of semiconductor onto a substrate, for example, of glass or plastic. The price of these modules is currently about half that of mono- or multi-crystalline (MC) modules [1]. In addition, individual crystals are not necessarily needed in this technology. TF or amorphous modules, therefore, should help to reduce manufacturing costs as well as materials costs for future products.

Over the past few years, building-integrated photovoltaic (BIPV) systems have been developed rapidly. A substantial number of BIPV projects can be found all over the world. Much research has been conducted examining the effect of a PV façade on the overall building envelope thermal performance as well as in the indoor environment, such as in relation to wall heat gain [2-4], heating and cooling air-conditioning

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loads for A/C system [5-6], thermal and visual comfort [7], Various investigations on ventilated PV double façades have been conducted. Measured results for active ventilation at conventional curtain wall PV façades and cooling by water flow was reported [8]. Thermal performance of the ventilated PV façade of a public library at Mataro, Spain was investigated [9].

During this study a comparative analysis was conducted to examine the thermal performance of both a DSPV facade using TF solar cells and a conventional glazing system both in conjunction with an internal curtain. Both laminar and turbulence flows in the DSPV facade cavity were studied and analyzed. Accordingly, the potential for energy savings for various operational modes of a double-glazed system integrated with thin film solar cells was explored. A better design of a semi-transparent TF PV configuration for optimum heat transmission control, based on the simulation results, is proposed and recommended.

2. PROBLEM DESCRIPTION AND PHYSICAL MODEL

The effect of a ventilated PV glazing system on the heat transmission through a building envelope was investigated to find possible heat control strategies. An indoor testing facility in the Hong Kong Polytechnic University was used. The experimental facility comprises two identical test cells, each one with the dimensions $L \times W \times D = 1.22 \text{ m} \times 0.82 \text{ m} \times 0.99 \text{ m}$. Detailed parameters of the test facility are listed in Table 1.

Table 1 Basic parameters of components in the test rig

Components	Specification
PV glazing	
Panel dimensions ($H \times W \times D$)	$0.95 \text{ m} \times 0.98 \text{ m} \times 0.012 \text{ m}$
Solar cell type	Amorphous silicon
PV area	0.466 m^2
PV efficiency	6%
Max power voltage (V)	69 V
Max power current (A)	0.28 A
Open circuit voltage (V)	89 V
Short circuit current (A)	0.38 A
Clear float glass	
Dimensions ($H \times W \times D$)	$1.18 \text{ m} \times 0.98 \text{ m} \times 0.005 \text{ m}$
Effective transmittance	0.86
Air channel behind PV	
Thickness of the air layer	16 cm
Dimensions of the air inlet opening ($H \times W$)	$0.1 \text{ m} \times 0.98 \text{ m}$
Dimensions of the air outlet opening ($H \times W$)	$0.1 \text{ m} \times 0.98 \text{ m}$
Height of the air channel	1 m
Dimensions of the hot-box ($H \times W \times D$)	$1.22 \text{ m} \times 0.82 \text{ m} \times 0.99 \text{ m}$

The test rig allows changes to be made to the solar radiation level, and easy comparisons can be made between different operational conditions with the same solar irradiation level. The solar radiation is artificial, and produced by a large scale solar simulator. This solar simulator can provide steady state irradiation capable of irradiating the tested façade surface. It comprises 363 dichroic tungsten halogen lamps of 75 W each covering a test area, $2 \text{ m} \times 2 \text{ m}$. The solar irradiance of the solar simulator is adjustable from zero to approximately 1300 W/m^2 . The performance of the solar simulator is able to meet Class C requirements specified in the International Standard IEC 6904-9 [6]. Such a study is hardly possible under outdoor testing conditions as weather parameters i.e. solar radiation and ambient air temperature, are continually changing. Previous researchers have attempted to find similar weather conditions among all test data collected for comparison purposes. This is likely to introduce some comparison errors. Fig. 1 shows a schematic view of the experimental apparatus, and Table 1 gives the specification of the a-Si photovoltaic module.

The geometry of the 2D tall cavity of width L and height H is shown in the schematic diagram of Fig. 2. Outside air is drawn into the bottom inlet of the cavity due to the buoyancy force. The airflow inside the glazing cavity is considered as steady state, laminar and turbulent. The air layer within the rectangular cavity is heated and cooled by isothermal vertical walls, hot T_h and cold T_c . The two horizontal walls are under adiabatic conditions by assuming zero heat flux boundary conditions at the horizontal surfaces. The air properties were assumed to be constant, except for the density, which is treated with the Boussinesq approximation.

3. MATHEMATICAL FORMULATION

In most practical engineering problems, the natural convection flows in the glazing cavity are turbulent, which becomes a highly complex phenomenon. Fully developed turbulent flow is featured by entangled eddies of different sizes. Prediction of the effects of each eddy in the flow is complicated, which is not favorable for engineering applications.

Theoretically, it is possible to directly apply the conservation equations to the entire flow field considered. It is, however, unreasonably difficult to do it practically. It is therefore, important to describe turbulent motion in terms of average quantities in order to create a usable numerical model of a turbulent flow field as well as of the fluctuating parts. For the steady 2D flow in a vertical channel created by two parallel walls, the time averaged Navier Stokes equations of motion can be written according to the study in the literature [10]. Fluid flow is assumed to be incompressible and the Boussinesq approximation is considered.

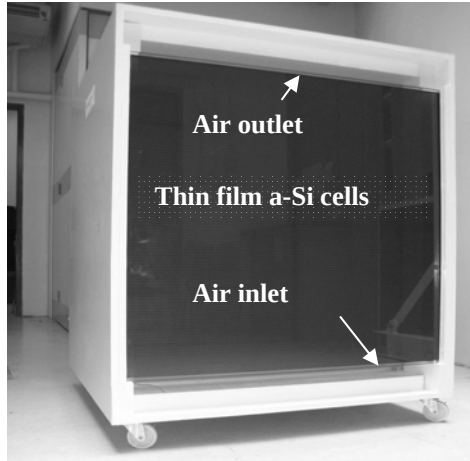


Figure 1. View of hot-box with naturally ventilated DSPV façade using a-Si thin film solar panel as external skin

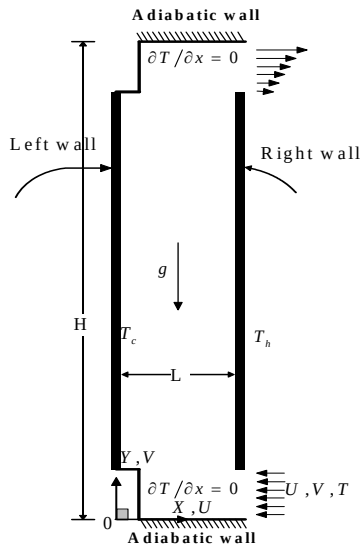


Figure 2. Schematics of the problem for natural convective heat transfer in air cavity created by two vertical wall and two openings are located on top and bottom, respectively.

The continuity equation for the conservation of mass is described as follows:

$$\frac{\partial(\rho \bar{u})}{\partial x} + \frac{\partial(\rho \bar{v})}{\partial y} = 0 \quad (1)$$

The equations for the conservation of momentum in x, y directions are as follows:

$$\frac{\partial}{\partial x}(\rho \bar{u} \bar{u}) + \frac{\partial}{\partial y}(\rho \bar{u} \bar{v}) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left[(\mu + \mu_t) \frac{\partial \bar{u}}{\partial x} \right] + \frac{\partial}{\partial y} \left[(\mu + \mu_t) \frac{\partial \bar{u}}{\partial y} \right] - \frac{2}{3} \rho \frac{\partial k}{\partial x} \quad (2)$$

$$\frac{\partial}{\partial x}(\rho \bar{u} \bar{v}) + \frac{\partial}{\partial y}(\rho \bar{v} \bar{v}) = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left[(\mu + \mu_t) \frac{\partial \bar{v}}{\partial x} \right] + \frac{\partial}{\partial y} \left[(\mu + \mu_t) \frac{\partial \bar{v}}{\partial y} \right] - \frac{2}{3} \rho \frac{\partial k}{\partial y} + (\rho - \rho_o)g \quad (3)$$

The equation for conservation of energy is described as follows:

$$\frac{\partial}{\partial x}(\rho \bar{u} T) + \frac{\partial}{\partial y}(\rho \bar{v} T) = \frac{\partial}{\partial x} \left[\left(\frac{\kappa}{C_p} + \frac{\mu_t}{Pr_t} \right) \frac{\partial T}{\partial x} \right] + \frac{\partial}{\partial y} \left[\left(\frac{\kappa}{C_p} + \frac{\mu_t}{Pr_t} \right) \frac{\partial T}{\partial y} \right] \quad (4)$$

In the above equations, μ_t is the turbulent dynamic viscosity, which is to be calculated from the kinetic energy of turbulence k and turbulent kinetic energy dissipation rate ε . Various turbulence correlations were employed in order to constitute a closed set of equations for solution. Laminar flows can be modeled by setting $\mu_t = 0$ and neglecting the equations for k and ε .

The $k-\varepsilon$ model was applied in this study for turbulence modeling. Turbulence kinetic energy, k , and its rate of dissipation, ε , are obtained from the following transport equations:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k \quad (5)$$

and

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - \rho \varepsilon - C_{2\varepsilon} \rho \frac{\varepsilon}{k} + S_\varepsilon \quad (6)$$

For the turbulent viscosity

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (7)$$

with $C_\mu = 0.09$ for the standard $k-\varepsilon$ model.

The boundary conditions are vital of importance for the numerical solution of the current problems. Specific boundary conditions are needed to solve the partial difference equations described above. The boundary conditions are listed below:

- **Boundary Conditions at inlet and outlet**

$$P_i = P_a, \quad T_i = T_a \quad \text{at } y=0 \quad (8)$$

$$P_o = P_a, \quad P_o = P_a \quad \text{at } y=H \quad (9)$$

- **Boundary Condition at the walls**

No slip boundary condition on the walls except at inlet and outlet are considered.

$$\bar{u}_i = 0, \text{ for } 0 \leq y \leq H, \quad x=0 \text{ and } x=L \quad (10)$$

$$k = 0, \text{ for } 0 \leq y \leq H, \quad x=0 \text{ and } x=L \quad (11)$$

$$T = T_l, \text{ for } 0 \leq y \leq H, \quad x=0 \quad (12)$$

$$T = T_h, \text{ for } 0 \leq y \leq H, \quad x=L \quad (13)$$

$$\frac{\partial T}{\partial x} = 0, \text{ for } y=0 \text{ and } y=H \quad (14)$$

4. NUMERICAL PROCEDURE

4.1 Grid independent study

The numerical procedure used to solve the governing equations of this project is based on the finite volume technique [11]. The computational domain has been divided into non-overlapping control volumes. Fine grids were chosen near the walls, inlet and outlet to capture the characteristics of the boundary layer. A grid independence test was carried out to make sure that the grid size would not affect the computed results. Fig. 3 shows temperature profiles at mid the cavity height $y/h = 0.5$ with various grid sizes (for example 90×26 , 90×100 , 90×110 grid points in the $x - y$ direction). Fig. 4 demonstrates the velocity profiles along height of the cavity, at $x/L = 0.5$ with various grid sizes. As can be seen from Fig. 3, the difference is very small. The medium mesh was employed in order to save computational time and obtain reasonable numerical solutions. It was found that the difference in results obtained using 90×100 and 90×110 grid sizes is small. Results obtained with finer grids did not show any noticeable changes in the predicted temperature field. As a compromise between computational time and results accuracy the computations presented were performed on a 90×100 grid.

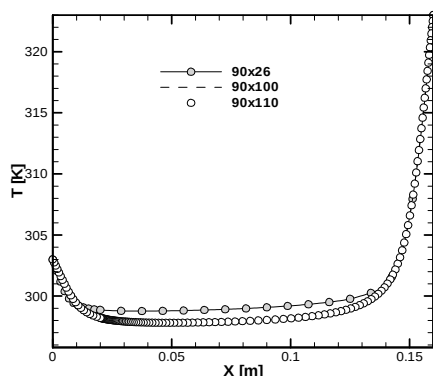


Figure. 3. Temperature profiles at mid height of the cavity $y/h = 0.5$ with various grid sizes

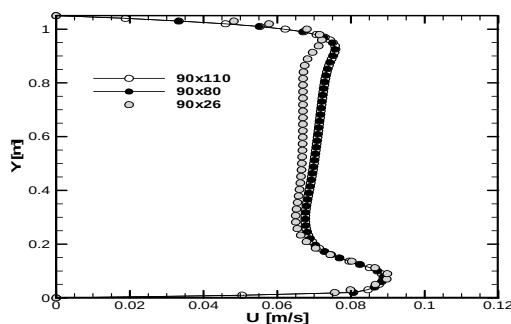


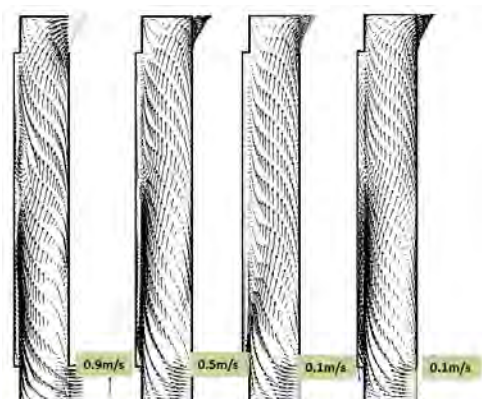
Fig Figure.. 4. Velocity profiles along the height of the cavity at $x/L = 0.5$ with various grid sizes

4.2 Results and discussion

The fluid flow characteristics of natural convection in a vertical channel created by two glass panes were studied for both laminar and turbulent regimes. A non-uniform or condensed grid with high refinement near the walls was used to increase the accuracy of the computational results in the proximity of walls. The details of the flow and flow patterns are obtained from the solution of the mathematical model together with various turbulence models. Results from different turbulence models were analyzed to predict the complex flow structure of natural convective heat transfer through the glazing cavity. Different inlet conditions for laminar flow were considered and analyzed. The simulation results are discussed in this section.

Many previous researches have indicated that inlet condition is of vital importance in the flow simulations in double skin façade systems. In these modeling studies, the ambient temperature is often used as the inlet temperature. The inlet modeling is important to the whole BIPV system energy analysis [12]. Inlet numerical modeling coupled with many other physics, see large turbulence, such as inflow and backflow, due to many factors affected. Fig. 5 shows the effect on the flow structure of various inlet velocities for the laminar flow velocities at 0.9, 0.5, 0.1 m/s and 0.1 m/s for turbulent flow.

Fig. 6 presents the velocity profiles for various average inlet air velocities. The driving force for natural ventilation is the reduction of density due to increase in air temperature. This increase is greater near the heat sources. The laminar profile, shown in Fig. 6, is non-symmetric, as different magnitudes of the heat sources from the walls. This agrees with the results [13]. Comparisons of vertical velocity profiles for various inlet velocities at mid height of the cavity, along the channel width are given for the computational domain. The velocity profiles develop as the average air speed increases.



a) $V_{in}=0.9m/s$ b) $V_{in}=0.5m/s$
c) $V_{in}=0.1m/s$ d) $V_{in}=0.1m/s$

Figure. 5. Effect of various inlet velocities on the flow structure for velocities at 0.9, 0.5, 0.1 m/s for laminar flow (a, b, c) and turbulent flow (d)

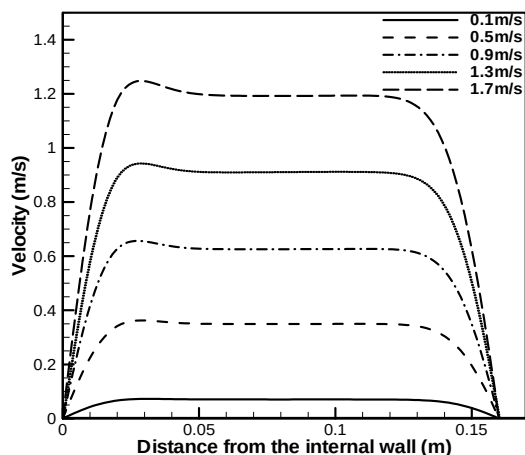


Figure 6. Velocity profiles along the center line of the cavity height for various inlet velocities

Turbulence flow both in an enclosed cavity and in an open cavity provides a model for many flows. The flow in the cavity is driven by buoyancy forces. The force may affect both the general fluid movement and the turbulent mixing processes [14]. In this study, three different turbulence models were used to predict the low Reynolds number turbulence existing in the current natural convective heat transfer and fluid flow field. To examine and validate the turbulent flow model, velocity profiles at the mid-height of the glazing cavity $y/h=0.5$ were compared and are presented in Figs. 7 and 8. In Fig. 8, two velocity peaks near the two vertical walls are observed for the standard $k-\varepsilon$ model. The velocity profile for laminar is symmetric with the highest velocity at the center. Fig. 9 shows interior surface temperature variations for the ventilated PV façade and conventional façade with internal curtain. The measured data indicate that the interior surface has a lower temperature than that of the conventional clear glass façade with internal curtain. This is mainly because of the occurrence of lower heat transmission in the PV façade glazing cavity. Data were collected in the laboratory of Hong Kong Polytechnic University for three days. The maximum difference in the temperatures for the first day reached at 8°C, 11°C for the second day, and 10°C for the last day. This could indicate that it is possible for a lower indoor temperature to be achieved, in summer, in a building with a PV double skin façade, thus creating a comfortable indoor environment for occupants during the summer season.

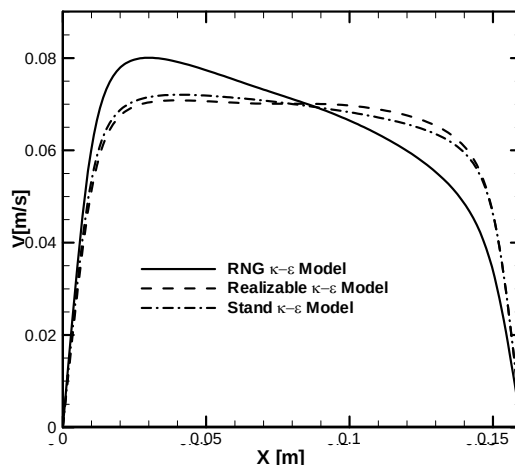


Figure 7. Velocity profiles at mid-height of the cavity $y/h = 0.5$ with various turbulence models

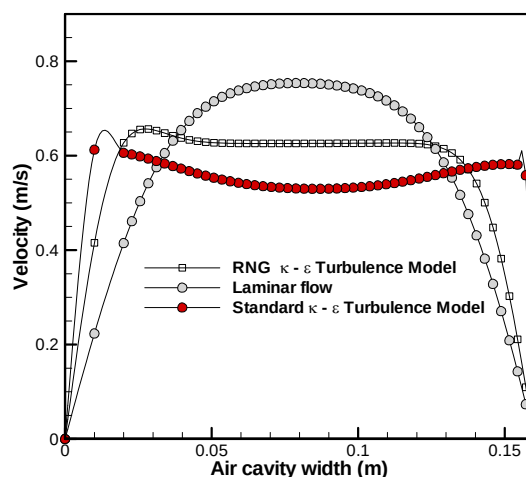


Figure 8. Comparison of the velocity distribution at mid-height of the air cavity

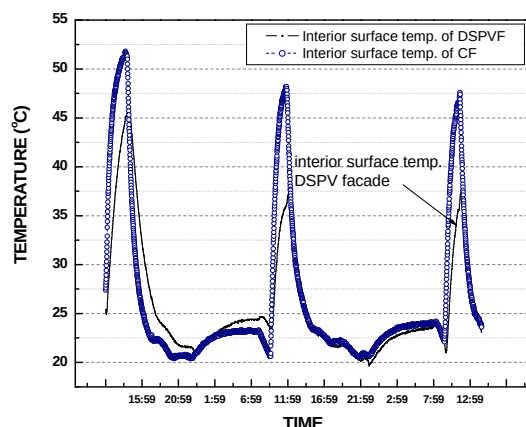


Figure 9. Interior surface temperature variations for ventilated PV façade and reference façade with internal curtain

5. CONCLUSIONS

This study investigated the naturally convective air flow in 2D glazing cavities between two parallel vertical walls which were integrated with thin film solar cells. Various inlet conditions for laminar flow were evaluated and various $k-\varepsilon$ turbulence models (RNG $k-\varepsilon$ model, Realizable $k-\varepsilon$ model, Standard $k-\varepsilon$ model) were compared. A comparison of the internal surface temperature of the PV façade and the conventional façade was made by experimental measurement. It was found that the maximum temperature difference could reach 11°C. Thus it is possible that lower indoor temperatures could be achieved in buildings with a PV double skin façade in summer. The cooling load could be expected to decrease as heat gain through the PV façade is considerably reduced. The effect of the DSPV façade on the space cooling load will be evaluated in the future work. This study will help to gain more information on the design of a DSPV facade with semi-transparent thin-film solar cells for the thermal control of PV temperature.

ACKNOWLEDGEMENTS

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HEAT TRANSFER PERFORMANCE OF GROUND HEAT EXCHANGERS BASED ON HEAT STORAGE MODE OF DOMESTIC HOT WATER

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ABSTRACT

The vertical buried ground-source heat pump system provides cold and heat for the controlling of indoor environment. Besides cooling and heating, the recovery ground-source heat pump unit also can provide domestic hot water. In summer, the supply of hot water can reduce the load undertaken by the ground heat exchangers and is beneficial to the recovery of underground temperature. However, in winter, the increased load of the ground heat exchanger has bad effects on the operating efficiency. Through the project design of ground-source heat pump system, which is for the reception center of a scenic spot in Sichuan province in China, the influence of the domestic hot water heat storage mode on the performance of the ground heat exchangers in summer and winter was analyzed. From the analysis on the ability of the domestic hot water heat storage under different operating modes, the dynamic performance of heat exchanger under different working conditions was obtained, and adjusting measures based on the load characteristics in this project were present.

Keywords: ground-source heat pump, domestic hot water, heat storage, the performance of ground heat exchangers.

1. INTRODUCTION

Utilizing underground soil as low-level heat source or heat sink, the ground-source heat pump system is generally used for space heating and cooling. Besides that, the ground-source heat pump unit with heat recovery also can provide domestic hot water (DHW) for the building. Comparing with the gas-fired or oil-fired hot water boiler, using ground-source heat pump system with heat recovery to supply DHW is more energy-saving and beneficial to environment.

Nowadays, the researches about DHW supplying by ground-source heat pump system mostly focus on the hybrid system [1-6]. Hepbasli did an exergetic analysis of a solar assisted DHW tank integrated ground-source heat pump system [3-5]. Trillat Berdal carried out an experiment study on a ground-coupled heat pump combined with solar collectors providing heat, cold and DHW for a residence [6]. In China, most of the researches emphasize the advantages of energy conservation and economic evaluation [7-11]. In reference [9], a ground-source heat pump system supplying DHW for a residence was presented. From

the economical comparison with air-source heat pump system, it obtained that the operating costs of supplying DHW by ground-source heat pump system was about 47% of that by air-source heat pump system, and the recover investment period was 3.7 years. Ping Cui carried out a modelling study on a hybrid ground-coupled heat pump (HGCHP) with DHW supplying system. The results obtained from this study show that the HGCHP system can effectively alleviate the imbalanced load of the ground heat exchangers and can provide almost 95% DHW. The energy saving for DHW heating is about 70% compared with an electric heater [10-11].

Using ground-source heat pump system to supply DHW has several ways. Different supply ways as well as different working modes of system will accordingly influence the heat transfer performance of GHEs, which is the prerequisite for stable operation of ground-source heat pump system. The main objective of this study is to analyze the influence of supplying DHW based on heat storage mode on the performance of GHEs through a project design of ground-coupled heat pump system in China.

NOMENCLATURE

Abbreviation		Q_{re}	Recovery heat(KW)
DHW	Domestic hot water	Q_{grou}	Load undertaken by GHEs(KW)
HGCHP	Hybrid ground-coupled heat pump	W_{comp}	Compressor work of heat pump unit(KW)
GHE	Ground heat exchangers	Q_{Cond}	Condensing heat of heat pump unit(KW)
Symbols		y	heat exchange amount per unit borehole depth(W/m)
Q_{Cl}	Cooling load(KW)	x	inlet water temperature($^{\circ}C$)

2. MATERIALS AND METHOD

2.1 Thermal balance analysis under heat storage mode

In ground-source heat pump system with DHW supplying by heat recovery, the unit has two condensers, one is for air-conditioning, the other is for heat recovery. In summer, the unit under refrigeration condition generates a mass of condensing heat, a part of which can be recovered to heat domestic water, then, the extra released to the underground soil by GHEs. In contrast, the unit under heating condition needs to extract more heat for both space heating and domestic water heating in winter.

From the analysis of thermal balance, along with the using status of air-conditioning system and DHW in building, dynamic change will be occurred between cooling load or heating load, condensing heat, recovery heat, and load undertaken by GHEs.

(1) In summer condition, the thermal balance formulas,

$$Q_{Cl} + W_{comp} = Q_{Cond} \quad (1)$$

$$Q_{Cond} = Q_{re} + Q_{grou} \quad (2)$$

In formula (1) (2), Q_{Cl} —Cooling load, W_{comp} —Compressor work of heat pump unit, Q_{Cond} —Condensing heat of heat pump unit, Q_{re} —Recovery heat, Q_{grou} —Load undertaken by GHEs.

In Conventional ground-source heat pump air-conditioning system, $Q_{Cond} = Q_{grou}$, while using heat recovery to supply DHW, due to a part of condensing heat is recovered, the load undertaken by ground heat exchangers is reduced effectively. From formula (1), (2), When Q_{Cl} keeps constant, if Q_{re} increases, Q_{grou} reduces inversely.

(2) In winter condition, the thermal balance formulas,

$$Q_{grou} + W_{comp} = Q_{Cond} \quad (3)$$

$$Q_{Cond} = Q_{re} + Q_{HI} \quad (4)$$

In formula (4), Q_{HI} —Heating load

In Conventional ground-source heat pump air-conditioning system, $Q_{Cond} = Q_{HI}$, while using heat recovery to supply DHW, the condensing heat is used to undertaken both the heating load of air-conditioning and the heating load of DHW. From formula (3)(4), When Q_{HI} keeps constant, if Q_{re} increases, Q_{Cond} and Q_{grou} increases accordingly.

From analysis above, we can see that in ground-source heat pump system with DHW supplying, Q_{grou} will be different under different conditions, and it has different influence on the heat transfer performance of GHEs. The paper takes a project for example to analyze this issue.

2.1 Project description

The project is a villa-type hotel in the reception centre of a scenic spot, with area of 832.5 square metre and 15 rooms, which is located in Qing Chuan County, Sichuan province, China. Qing Chuan County belongs to subtropical moist climate area, the average annual temperature is $13.7^{\circ}C$.

From the calculating results, space cooling load in summer is 50.24 kw, space heating load in winter is 16.76 kw, and the DHW's design heat consumption of maximum hour in winter is 49.4 kw. Comparing with the space cooling load and space heating load, the large heating load of DHW contributes to the increase of total load undertaken by the GHEs and the quantity of buried exchangers in winter. Consequently, the initial investment of heat exchangers greatly increased, and the heat transfer performance of the GHEs severely declined. Considering the actual condition of this project, the most effective adjusting measure is using heat storage tank. The guarantee time of heat storage tank is taken as three hours, therefore, the volume of the heat storage tank is calculated as 2.4 cubic meter according to the reference [12].

2.2 Design of GHEs

2.3 Thermal response test

In order to know the geologic structure, initial temperature distribution and the heat transfer characteristics of GHEs in the project location, the rock-soil thermal response test (TRT) has been done before implementation. The diameter of test borehole is 130millimetre, the diameter of test exchangers is 32millimetre, and the depth of double U-tube is 100metre. Flow of single U-tube is 0.3 cubic meter per hour, heat quantity of single U-tube is 4.5 kilowatt which input by an electric heater, and the heating time continues for twelve hours. The rock in this area is mainly sandstone, and mixture of expansive soil and sand can be used as backfill material. From the test results, the initial temperature of soil blow depth of 10meter underground is almost unchanged and the average value is about $14^{\circ}C$.

In the early heating period, due to the voltage fluctuation, heat exchange amount per unit borehole depth heat appears unstable, and then gradually becomes steady (Fig.1)

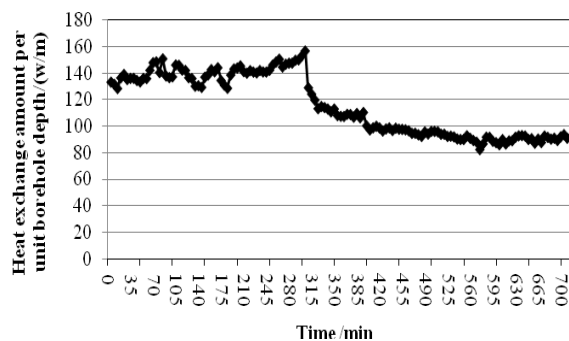


Figure.1. Heat exchange amount per unit borehole depth under twelve hours' heating condition

According to the results analysis, under twelve hours' heating condition, the linear relationship between heat exchange amount per unit borehole depth in corresponding working condition and inlet water temperature is obtained(formula (5)).

$$y = 8.2993x - 104.76 \quad (5)$$

In formula (5), y—heat exchange amount per unit borehole depth, W/m; x—inlet water temperature, °C

The average inlet water temperature under TRT is 26 °C, which fits the value of design condition. According to the results analysis, the average heat exchange amount per unit borehole depth under refrigeration condition and heating condition is 90 watt per meter and 48 watt per meter respectively.

2.4 Preliminary calculate of GHEs

Based on the geological exploration results and construction process of the testing exchanger, the larger of the depth, the construction difficulty will be greater. Combined with the heat transfer effect and drilling difficulty level, the suggesting depth is 70 metre. If we calculate using the space cooling load in summer, the number of GHEs is eight, if using the space heating load in winter, the number is five, however, if using the sum of space heating load and heating load of DHW in winter, the number is greatly increased to twenty. Because of the bad geological condition in this area, the construction cost of vertical U-tube exchangers achieves two hundred RMB per meter, while the cost of heat storage tank is much lower. Take both the technical analysis and economic evaluation into consideration, using the sum of space heating load and DHW heating load in winter to calculate the number of heat GHEs is not appropriate, so that adopting heat storage mode with heat storage tank can reduce the investment by a large margin. Comprehensively consideration of the initial investment and performance of GHEs, the number of GHEs is designed as fifteen.

2.5 Operation adjustment methods under heat storage mode

Although the supply of DHW in hotel buildings lasts for 24 hours usually, using time is relatively concentrated. The reference [14] pointed out that there are two using peaks of DHW in hotels, the first peak

lasts from 6:00 am to 8:00 am and the second is from 6:00 pm to 22:00 pm. Air-conditioning peak load often appears at noon and evening after tourists back to hotel.

2.6 Summer condition

In summer condition, we should pay attention to the dynamic change of air-conditioning load and DHW heating load. Because demand of DHW in summer is relatively less, when the air-conditioning load is high, for example, at noon or evening, heat recovered by heat recovery condenser can completely satisfy DHW heating demand, so that we can use heat storage tank to store hot water and heat, then release the extra condensing heat to the underground soil by GHEs. When the space cooling load is lower and DHW heating load is opposite higher, for example when the first DHW peak comes in the morning, with the lower outdoor temperature, air conditioning load also is relatively lower but DHW load is higher, heat recovery probably will fail to meet user's demand. The solution is to use the stored DHW in heat storage tank. Heat storage tank not only contributes to peak shaving, but also decreases the undertaken load of GHEs that is helpful to improve the heat transfer performance of ground heat exchangers.

2.7 Winter condition

In winter condition, unit converts between running condition and shutdown condition under heat storage mode of DHW, correspondingly, underground heat exchangers also include continuous condition and intermittent condition. In this project, DHW supplying includes the following two stages:

(1) In the daytime, most of the tourists go outside, therefore with few air-conditioning usage or low air-conditioning load and low DHW heating demand in hotel, unit can be under intermittent operation condition. During the shutdown process, the hot water can be supplied by the heat storage tank. GHEs under long time continuous operation condition in the previous night caused underground soil temperature drastically declined and the heat transfer performance also declined. During the energy storage time, underground soil temperature can get a certain degree to recover.

(2) In night, due to tourists are back to the hotel in succession, the peak load of space heating and DHW heating will appear at the same time, that makes unit must supply heat both for space heating and DHW. For the purpose of guaranteeing the high heat transfer performance of GHEs, we can start unit for producing DHW in advance before tourists back and stored in the hot water storage tank. When they come back, DHW in storage tank is firstly used, thus it can avoid the superposition of space heating load and DHW heating load and reduce heat extracted from underground soil. When the energy storage period passing by, along with the decreased demand of DHW, the adverse effect of unit's mixed operation condition to GHEs also can be alleviated.

2.8 Simulating analysis on the performance of GHEs under heat storage mode

In order to analyze the influence of supplying DHW under heat storage mode on the heat transfer performance of the GHEs, this paper used CFD software Phoenix to simulate, and respectively calculated the performance parameters of GHEs under continuous operation condition and after the energy storage period (shutdown process).

2.9 Establishment of the Model

In accordance with design conditions in the project, physical model was established, which is shown in Fig.2. The detailed parameters are listed in Table1.

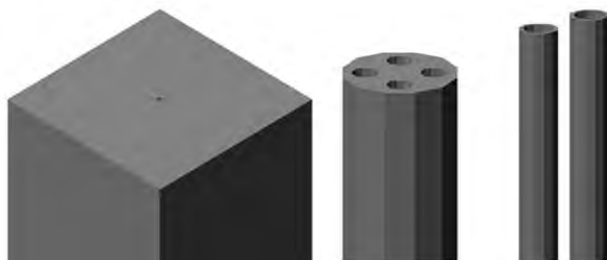


Figure.2. The physical model of soil, borehole and

Table 1 The parameters of the physical model

Parameter name	Value
Depth of U-tube /m	70
Diameter of U-tube / mm (PE)	25
Diameter of borehole/mm	110
Radius of calculated region/m	4
Specific heat of soil (J/kg·°C)	921.1
Thermal conductivity of soil (W/ m·°C)	2.1
Density of soil (kg/m ³)	2400
Thermal conductivity of backfill material (W/ m·°C)	2.41

2.9.1 Assumptions

- (1) The physical composition and the thermal physical parameters of soil remain constant.
- (2) The velocity of the liquid in the U-tube is uniform.
- (3) Neglect the water migration in the soil.
- (4) Ignore the thermal resistance of buried tubes and the contact thermal resistance between buried tubes and backfill material.
- (5) The initial temperature on the whole tube remains unchanged.

2.9.2 Boundary and initial conditions

- (1) Inlet boundary condition

In summer condition, due to different inlet water temperature (IWT), there are two kinds of working conditions. IWT is 27°C and 30°C under first condition and second condition respectively. Inlet flow is 0.8 cubic meter per hour for each U-tube and the running time lasts for fourteen hours (from 6:00 pm to the 08:00 am) for both two conditions.

In winter condition, IWT is 10°C, inlet flow is 0.6 cubic meter per hour for each U-tube, working condition can be divided into two modes: firstly, running time lasts for fourteen hours, then, unit shutdown process lasts for three hours.

- (2) Surface boundary condition

Conduit surface is a fixed wall. In the calculation, wall function is adopted without sliding and inner heat resource. The flowing of water in U-tube is coupled with heat transfer of soil. The top air-soil surface boundary is assumed to be of the convective type.

- (3) Initial condition

The initial average temperature of rock-soil is 14°C.

3. RESULTS AND DISCUSSION

3.1 Calculation results

3.1.1 The summer condition

Supply of DHW will decrease the undertaken load of GHEs and IWT of U-tube is decreased accordingly, so that temperature distribution around the borehole and the performance of GHEs will be different under different modes after fourteen-hour's continuous operating condition.

Fig.3 presents the water temperature distribution in U-tube under two conditions after fourteen-hour's continuous operating. It indicates that after fourteen-hour's continuous running, comparing the high inlet temperature condition, the outlet water temperature (OWT) under lower inlet temperature condition is obviously lower, outlet water temperature difference in two working conditions of No.1 U-tube and No.2 U-tube are respectively 1.0°C and 0.84°C. Take No.1 U-tube and No.2 U-tube's axis of symmetry for centre line, the soil temperature distribution in calculating region at different radius in summer is shown in Fig.4. Temperature rise of the shallow layer soil around the borehole is relatively larger, and the soil temperature under higher inlet temperature condition is obviously higher, the maximum difference is 2.21°C. It illustrates that temperature rise of soil and heat accumulation is serious along the radius direction. In addition, the longer running time is, the larger the heat accumulation form, and the greater temperature rise.

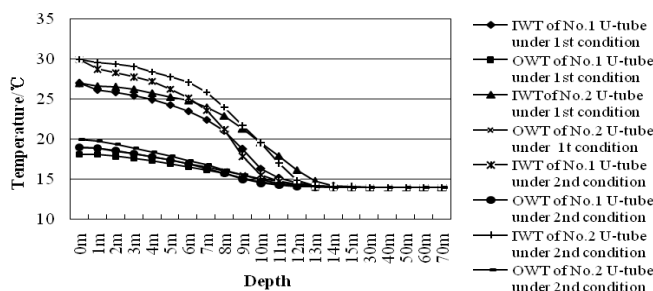


Figure.3. The water temperature distribution in U-tube after fourteen-hour's running in summer

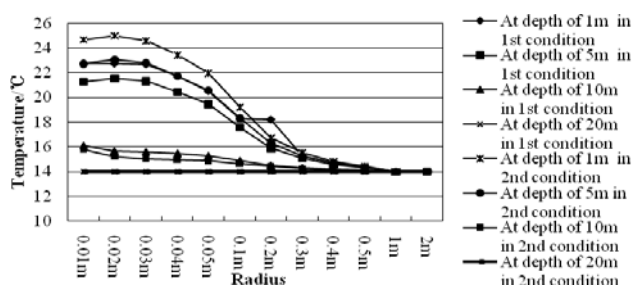


Figure.4. The soil temperature distribution in calculating region at different radius in summer

From Fig.3 and Fig.4, it shows that the temperature distribution trend along depth direction and radius direction keeps corresponding consistent under the two conditions and the influencing degree along the two directions is both weakened. For shallow layer soil, which is the main heat exchange layer, the high inlet temperature will make the temperature greatly rise and easy form heat accumulation around the borehole under long-term running condition.

3.1.2 The winter condition

From the calculating results in winter condition, the soil temperature distribution in calculating region at different radius under continuous condition and shutdown condition is respectively shown in Fig.5 and Fig.6. As is shown in the two figures, after fourteen-hour's continuous running, the temperature of underground soil declined in different degree along depth direction and radius direction, while after the energy storage period, soil temperature get a different degree of recovery under DHW heat storage mode, and the maximum value of temperature recovery reached to 1.8°C. Consequently, heat storage mode of DHW is beneficial to the recovery of underground soil temperature and heat transfer performance.

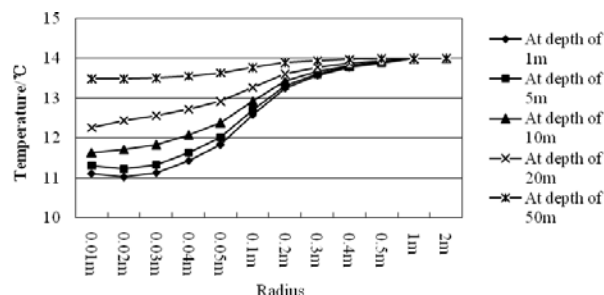


Figure.5. The soil temperature distribution at different radius after continuous condition in winter

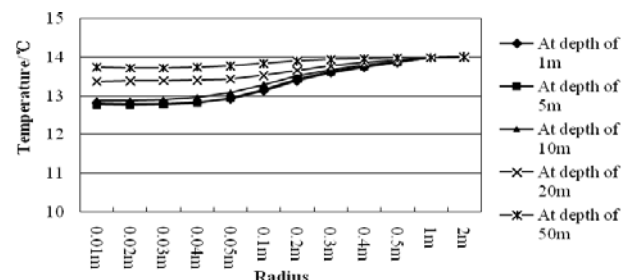


Figure.6. The soil temperature distribution at different radius under shutdown condition in winter

3.2 Experimental verification

The author used the testing dates in an actual project for verification. In the project, the ground-source heat pump unit provides DHW by heat recovery. The first condition is refrigeration condition, and the second condition is both refrigeration and producing DHW by heat recovery simultaneously. Because of the almost same outdoor meteorological condition and the small difference of air-conditioning load in the building during the two conditions section, we can take the two conditions for comparison. From Fig.7, we can see that in the second condition, due to recovering a part of condensing heat for heating DHW, the IWT and OWT both fell than that in the first condition. The maximum inlet water temperature difference of two conditions is 2.5°C, which is slightly less than theoretical calculation value, but the maximum outlet water temperature difference is 1.0 °C, which is close to calculating results. The continuous running time is eight hours in the experimental condition, slightly smaller than that of the calculation condition. The experimental test results can still validate the correctness of theoretical analysis.

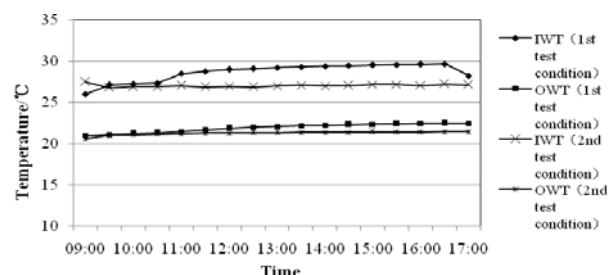


Figure.7. The water temperature distribution in U-tube under two test condition

4. CONCLUSION

(1) Producing DHW by recovering condensing heat in summer decrease the heat released to the underground soil by GHEs, so that the inlet water temperature of U-tube dropped accordingly. Through theoretical calculation results in summer condition, the outlet water temperature under inlet water temperature of 27°C condition is obviously lower than that under inlet water temperature of 30°C condition after fourteen-hour's continuous running. the temperature difference of OWT is almost 1.0 °C, furthermore, the heat accumulation of underground soil is more serious, the maximum soil temperature difference of two IWT condition is 2.21 °C. It illustrates that temperature rise of soil and heat accumulation caused by high inlet water temperature is more serious. Using heat storage mode of DHW in summer reduce the heat released to the underground soil, relieve the heat accumulation of soil, and is helpful for the long-term and stable operation of GHEs.

(2) For the buildings with high heating load of DHW in winter, it suggested to use heat storage mode for DHW supplying. According to the characteristics of DHW supplying, using heat storage tank can reduce the number of GHEs and reduce initial investment of ground-source heat pump system. After fourteen hours' continuous running, the temperature of soil around borehole fell severely, the maximum value reached to 2.89 °C. While during the three-hour's energy storage period, underground soil temperature get a different degree of recovery, and the maximum value of temperature recovery reached to 1.8 °C because of the heat storage mode of DHW supplying. The heat storage mode of DHW slow down the drastic decline of heat transfer performance caused by long-term and large load of continuous operation.

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ANALYSES ON ENERGY CONSUMPTION CHARACTERISTICS AND SAVING MEASURES OF AIRPORT BUILDINGS IN HOT SUMMER AND COLD WINTER ZONES

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ABSTRACT

Through researching the current situation and actual energy consumption of one airport in hot summer and cold winter zones, the energy consumption of airport buildings in hot summer and cold winter zones is analyzed and the statistics of airports in other countries and zones is compared with. Some energy saving measures for the equipment such as lighting, elevators and the water cooling storage system in the airport are analyzed in this paper. Through analyzing the different energy saving measures, the energy saving rate of the lighting and elevators is 30 percent and 33 percent respectively. The water cool storage system is applied, and the cost saving rate is 49 percent. The total energy saving rate of the three measures above is 38 percent, and the benefit per unit area is 17 yuan/m². The energy consumption of airport buildings in hot summer and cold winter zones is very huge, so proper energy saving measures will bring a considerable benefit.

Keywords: Hot summer and cold winter zones, Airport buildings, Energy consumption, Energy saving.

1. INTRODUCTION

There are many airport buildings in hot summer and cold winter zones where the transfer personnel quantity is huge. Generally, the airports are very large and the requirement for environment is high, which is the one of the largest energy consumption buildings in their city. According to the data published by Civil Aviation of China, the energy consumption of airport buildings reaches about 3 percent of the total consumption in the civil aviation industry in China[1]. Although the proportion is not high, the value of the energy consumption runs to about several hundred thousand tons of standard coal and the potential of the energy saving can't be neglected. According to the research and practice in recent 30 years in the energy field, the energy saving for buildings is accepted as the most potential measure and the most useful way, which is one of the most effective measures to relieve the energy crisis and solve the contradictions between the economic social development and the shortage for energy[2]. Recently, we are in the airport buildings construction boom. According to the statistics from Civil Aviation of China, 14 airports are being built in 2012 and 12 airports will be built in 2013. The total number of newly-built airport and extended airports run to 53 [3]. Therefore, the study to the characteristics and energy saving of airport buildings not only is significant but also can provide reference to the newly-built airports.

The energy saving measures to airport buildings which are adopted in China mainly include rebuilding

the building envelope, using the new systems and enhancing energy management. From the view of the energy structure, the main energy to airports is electricity and coal in northern China and the consumption of coal is 44 percent which is mainly used to supply heating in winter. In southern China, the main energy of airports is electricity which is 68 percent of the total energy consumption and is used to not only supply heating and cold but supply power[4]. However, the majority of airports don't fully take advantage of the new energy such as solar energy and wind energy because of the limit to the technology.

In US, there are 40 percent of airports using electricity as their main energy, so they pay more attention to using new energy and improving the energy efficiency while discarding the unnecessary equipment[5]. In the San Francisco International Airport, the total consumption of electricity in 2010 is 322927MWh which equal to demand of 48000 residents for the electricity in California. Solar cell panels installed in the roof of terminal produce 747MWh per annum[6]. The Denver international airport use solar lighting instead of lighting during the day making the energy saving rate run to 40%~60%[7].

With the global developing, different energy saving technology in each country is learned from each other. However, the effect is different even if the same energy saving measure because of the difference of climate and energy structure. So far, researches for energy consumption of airports at home and abroad is not much and the number of relevant papers is less than 10. Because the climate of each region is different, energy consumption of airports in different region and

especially air conditioning system is different. In this paper, the research object is one airport in hot summer and cold winter zones, meanwhile, the characteristics of energy consumption and energy saving measures will be analyzed.

2. ENERGY CONSUMPTION SITUATION OF AIRPORTS IN HOT SUMMER AND COLD WINTER ZONES

The research object is one airport in hot summer and cold winter zones. The area of this terminal is 802000m² with two energy center supplying energy to the terminal. From Fig.1, the energy consumption per unit area in 2009 is 136.3kgce/m²[8], which is the largest in the recent three years. Because of some effective measures, the total energy consumption of this airport is declining year by year. The energy consumption in 2010 is declining 18.7% compared with 2009, and the energy consumption in 2011 is declining 3.3% compared with 2010.

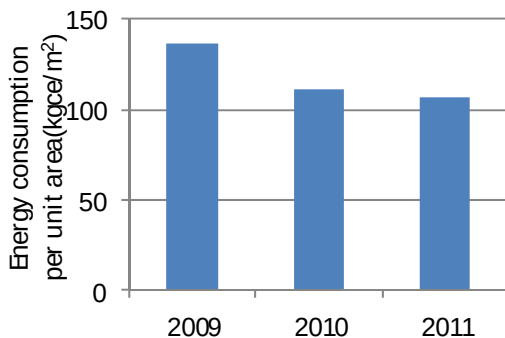


Figure.1 Energy consumption per unit area in 2009~2011

From Fig.2, the main energy consumption is electricity which is used to supply cold, lighting and power equipment in this airport run to 78 percent of the total energy consumption. Then consumption of gas which is used to supply for heating is 20 percent of the total energy consumption. The consumption of diesel is just two percent of the total energy consumption.

According to the data from the research, we get the energy consumption proportion of the main equipment in the terminal. The bridge equipment works 7.5 months per year and 10.8 hours one day from the relative standard[9].

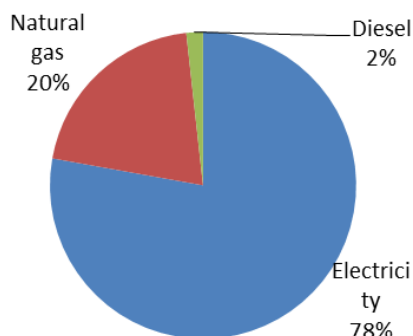


Figure.2 The proportion of various energy in 2011

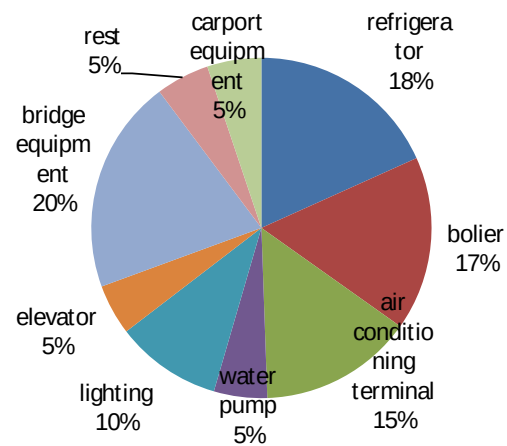


Figure.3 The energy consumption proportion of the main equipment

From Fig.3, the largest energy consumption equipment is bridge equipment which is 20 percent of the total energy consumption. Then, the boiler for supplying heating is 17 percent of the total energy consumption; the lighting is 10 percent of the total energy consumption; either of the elevator, the water pump of the air conditioning terminal and equipment in the carport is 5 percent of the total energy consumption.

The energy consumption per unit area each month in 2011 is shown in Fig.4. The total energy consumption of airports in hot summer and cold winter zones is very large. This kind of buildings has large energy saving potential, and it is necessary to adopt energy saving measures.

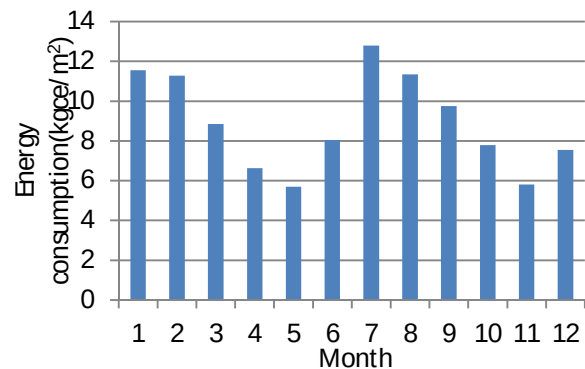


Figure.4 The energy consumption per unit area each month in 2011

3. COMPARISON OF AIRPORTS IN DIFFERENT REGIONS AND COUNTRIES

According to the research, the energy consumption per unit area in the terminal is 184.4kWh/m², which compared with the data of the Beijing Capital International Airport, is little larger. The average energy consumption per unit area of 29 airports in Greece is 234kWh/m²[10]. The energy consumption per unit area of the San Francisco International Airport is

207kWh/m². These differences mainly result from the differences of the region and the energy structure and the scale of the airports.

4. ANALYSES ON ENERGY SAVING MEASURES

4.1. The energy saving measures for lighting

A large number of lighting in this airport is transformed by LED and energy saving lamps. The intensity of the illumination is detected in the terminal so that the lights without being used will be turned off timely. A large amount of energy is saved by this measure.

According to the data, the total power of the lighting in the terminal is 7152.4kW and the opening rate is 70 percent. The lights will be opened about 16 hours each day. The total energy consumption of the lighting in one year is:

$$5006.7 \times 70\% \times 365 \times 16 = 20467389.6 \text{ kWh}$$

Before the transformation, the total energy consumption of the lighting is about 7152.4kW and the opening rate is also 70 percent. The lights will be opened about 16 hours each day. The total energy consumption of the lighting in one year before the transformation is:

$$7152.4 \times 70\% \times 365 \times 16 = 29239011.2 \text{ kWh}$$

According to Table.1, the average electricity charge in this region is 0.64 yuan/kWh, so the charge will be saved:

$$(29239011.2 - 20467389.6) \times 0.64 = 5614000 \text{ yuan}$$

Compared the two charge above, the amount of the saving electricity is 8.772 million kWh which equal to 2631.6 tons standard coal. The amount of the saving standard coal per unit area is 3.28kgce/m². The energy saving rate is 30 percent and 5.614 million yuan is saved by this measure. The benefit per unit area is 7 yuan/m².

4.2. The energy saving measure for water cool storage system

There are four water cool storage tanks in this airport of which the total volume of four water cool storage tanks runs to 116000RTh. The water cool storage system takes advantage of cheap off-peak power during nights to produce cooling water stored in tanks. The water cool storage system is used in a low electric valley period, and in a high electric period releasing its energy, and this make the water storage system more effective.

From Fig.5, the volume of the total cool storage in 2011 is 13.18MRTh and the total cooling capacity is 201.03MRTh. The proportion of the total cooling storage and the total cooling supply is 62.7%.

As Fig.6 shown, this airport start to supply cooling from May in 2011, and the electricity consumption of the cooling system in July is 6.023 million kWh which runs to the largest. In July, the electricity consumption in a low electric valley period is 3.923 million kWh,

which is 65.1 percent of the total electricity consumption in this month. The electricity consumption in a high electric period is 0.474 million kWh which is 7.9 percent of the total electricity consumption in this month. The total cooling capacity and the total power consumption in each period are shown in Table.1.

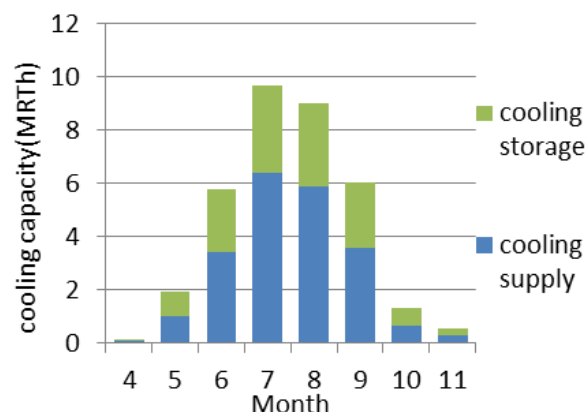


Figure.5 The statistics to the cooling storage and cooling supply in 2011

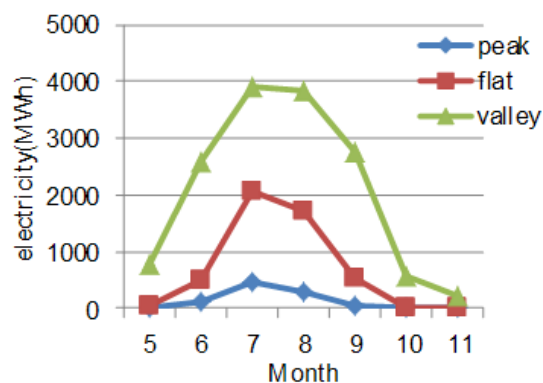


Figure.6 The electricity consumption in each period of the water cool storage system monthly

According to the rule for the electricity charge in region, the price in a low electric valley period is 0.239yuan/kWh, in an ordinary electric flat period is 0.646 yuan/kWh and in a high electric peak period is 1.049 yuan/kWh. Combined with the actual electricity consumption during each period, the charge for the water cool storage system in 2011 is:

$$14594947 \times 0.239 + 4741340 \times 0.646 + 854931 \times 1.049 = 7447920.6 \text{ yuan}$$

If the conventional refrigerating unit is adopted in this airport, we can calculate the annual charge for running the system.

$$20191218 \times (20.25\% \times 0.239 + 39.26\% \times 0.646 + 40.49\% \times 1.049) = 14674113 \text{ yuan}$$

From here we see that the water cool storage system saves 722.6 million yuan in 2011 and the energy saving rate run to 49 percent. The benefit per unit area is 9 yuan/m² reflecting the huge economic efficiency and the energy saving efficiency.

Table.1 The statistics of the water cooling storage system and the electricity charge

Period	Valley	Flat	Peak	Total
Cooling supply(RTh)	4258153	8255074	8515023	21028250
Cooling storage(RTh)	11530916	1647274	0	13178190
Electricity consumption(kWh)	14594947	4741340	854931	20191218
Cooling supply(%)	20.25%	39.26%	40.49%	—
Electricity consumption(%)	72.28%	23.48%	4.24%	—
Price(yuan/kWh)	0.239	0.646	1.049	—

4.3. The energy saving measure for the elevator

Automatic pedestrian escalators with the infrared sensing device are adopted in this terminal so that the automatic pedestrian escalators will detect the sign sent by the infrared sensing device to identify whether someone passing by. When are in light condition, the automatic pedestrian escalators will be turn to low speed.

According to the research, there are 105 automatic pedestrian escalators in the terminal and the total power is 811kW. The opening rate is 70 percent. Neglect the time that automatic pedestrian escalators run in low speed. The automatic pedestrian escalator will work 18 hours every day. The annual electricity consumption is:

$$811 \times 70\% \times 365 \times 12 = 2486526 \text{ kWh}$$

Before the transformation, the opening rate is also 70 percent but automatic pedestrian escalators will work about 18 hours each day. The total electricity consumption of the automatic pedestrian escalators in one year before the transformation is:

$$811 \times 70\% \times 365 \times 18 = 3729789 \text{ kWh}$$

From Table.1, the average electricity charge is 0.64yuan/kWh, so the charge will be saved:

$$(3729789 - 2486526) \times 0.64 = 796000 \text{ yuan}$$

Compared the two date above, the amount of the saving electricity is 1.234 million kWh which equal to 372.98 tons standard coal. The amount of the saving standard coal per unit area is 0.47kgce/m². The energy saving rate is 33 percent and 0.796 million yuan is saved by this measure. The benefit per unit area is 0.99 yuan/m².

4.4. Other energy saving measures

Frequency conversion air conditioning boxes with CO₂ concentration sensors are adopted in this terminal. Fresh air volume will be reduced when the CO₂ concentration is decreasing so that the fresh air load will decrease. With the guarantee that the indoor air quality remains same, this measure has a potential energy saving effect and the maximum can run to more than 50 percent[11].

The flight linkage technology is adopted in this airport and very effective. With linking the airport flight information system, the air conditioning and lighting will be turned on and turn off in advance automatically.

5.CONCLUSIONS

A series of energy saving measures are adopted in this terminal and very effective. A large number of

lighting in this airport is transformed by LED and energy saving lamps. The energy saving rate is 30 percent and the quantity of energy saving run to 2631.6 tons standard coal. Automatic pedestrian escalators with the infrared sensing device are adopted in this terminal, which saves 372.98 tons standard coal, and the energy saving rate run to 33 percent. There are four water cool storage tanks in this airport of which the total volume of four water cool storage tanks runs to 116000RTh. The water cool storage system is used in a low electric valley period, and in a high electric period releasing its energy, and this make the water storage system more effective. The water cool storage system saves 722.6 million yuan in 2011 and the energy saving rate run to 49 percent. The total energy saving rate of the three measures above is 38 percent, and the benefit per unit area is 17 yuan/m². The energy consumption of airport buildings in hot summer and cold winter zones is very huge, so proper energy saving measures will bring a considerable benefit.

ACKNOWLEDGEMENT

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ENERGY EFFICIENT MANAGEMENT STRATEGY FOR AC SYSTEM WITH LOCAL COOLING UNITS AGAINST HIGH LOAD ZONE

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ABSTRACT:

The essential of BEMS for energy saving of AC is to establish an optimization strategy on the base of the energy consumption model. However, AC system is a typical complex multivariable nonlinear time-varying system and its energy consumption involves various factors and varies with the different running schemes, therefore, energy consumption is difficult to model. The aim of the present paper is to establish an operation energy consumption model of centralized all-air AC systems equipped with local cooling unit against some areas with extremely high cooling load in underground engineering, and draws up an energy saving strategy. In the strategy, solve the optimal supply air state based on load prediction of the air conditioned zones, optimize the allocation of cooling load among fresh air, air handling unit (AHU) and dry fan-coil unit (FCU) for zones and get the operation scheme with the lowest energy consumption and meeting the underground engineering requirements.

Keywords: High cooling load, AC, Local cooling units, Energy efficient management, Energy consumption model.

1. INTRODUCTION

The energy consumption of the ventilating and AC system in public building makes up to more than 50% of the total. Its influencing factors are various and with a broad changing range, in order to carry out the energy saving operation of AC system, the energy consumption model and the corresponding energy saving operation strategy should be built. Because the influencing factors are strongly related to each other, it is very difficult to build the energy saving algorithm. Most of the studies in the past were limited in the influence of just one or some simple factors[1,2], such as energy saving operation by using the fresh air, the set value of the indoor temperature, the demand control ventilation and so on. D. Kolokotsa[3] established the state equation of the indoor parameters (the indoor temperature, the corresponding humidity, the CO_2 density and the lighting intensity), built the prediction model of the energy consumption based on the state equation, which takes opening the window, blocking the sunlight, managing the AC and controlling the lightening those four items as the import, and estimated the coefficients of the state equation by system recognition. The management strategy is more comprehensive, but the complex process makes the calculation load quite heavy at the same time. The author of this article aimed at the centralized AC system of underground engineering,

putting forward an AC energy optimization management algorithm based on the supply air state[4], which considers all the factors that influence the operation of the AC system, and takes solution of supply air state as the basis of the algorithm, just as Figure 1 shows. On one side, as for the demand-side, the factors influencing the supply air state S in the underground engineering include the room excess heat Q , excess humidity W , the Enthalpy settings (i_{set}), the relative humidity settings (ϕ_{set}), and the CO_2 density settings (C_{set}). On the other side, as for the supply-side, the influencing factors include the outdoor air enthalpy (i_o), the humidity value (d_o) and the AC scheme (y) of the air handling system. There are many choices of the AC schemes which meet the supply air state S , and the energy consumption of each AC scheme is different. Therefore, studying all the possible AC schemes and calculating their energy consumption help us to find an optimal AC scheme. But this algorithm is only adaptable to all-air AC system in the underground engineering. It cannot be used for the underground engineering which contain some zones with extremely high load (such as the communication workshop). At present, usually equip the AC system with local cooling units to against the high load, which requires targeted research.

NOMENCLATURE

Abbreviation		Subscripts	
AC	air conditioning	-set	set value
AHU	air handling unit	-s	supply air state
FCU	fan coil unit	-A	zone A
		-B	zone B
		-o	outdoor
		-fresh	fresh air
		-return	return air
		-B1	about AHU stage in zone B
		-B2	about FCU stage in zone B
		-cfan	fan for conveying
		-pump	pump
		-fresh	fresh air
		-fresh-p	positive fresh air
		-fresh-n	negative fresh air
		-CT	cooling tower
		-CHW	chiller water
		-demand	minimum demand
Symbols			
G	air volume		
P	power		
Q	excess heat		
W	excess humidity		
i	enthalpy		
C	CO ₂		
d	humidity ratio		
P	energy consumption to provide per unit load		
Greek Letters			
ϕ	relative humidity		

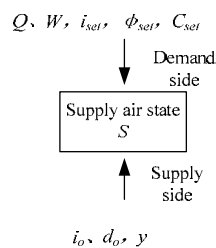


Figure 1 Operation and management algorithm based on supply air state

2. DESCRIPTION OF THE SYSTEM

The underground engineering generally adopts the centralized all-air AC system. As shown in Figure 2, the cold source of air cooler is generally chilled water of about 7°C which is made by the low temperature chiller (as chiller1 in Figure2).

Figure 3 shows the centralized AC system with local cooling unit, where zone A is the ordinary air conditioned area and zone B is the air conditioned area with high load. Because of its perennial need of cooling capacity, Zone B is equipped with a local cooling unit, which can be a radiation plate or dry FCU. Considering the dry FCU is more convenient for the transformation of existing underground engineering than the radiation plate, this paper takes the dry FCU as research object.

The cold source of FCU can be various, this paper selects the high temperature chiller (as chiller2 in Figure 3) to supply high-temperature cooling water.

Compared with low temperature chiller of AHU, the coil temperature for cooling can be considerably increased, e.g. from current 7°C to 17°C, providing enough potential of energy saving for the optimal control of the system.

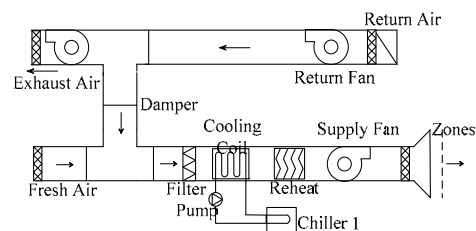


Figure 2 Common centralized AC System

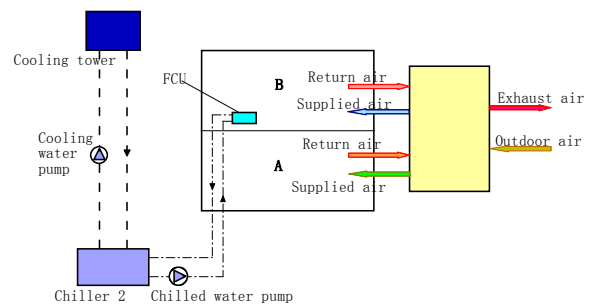


Figure 3 Centralized AC system with local cooling units

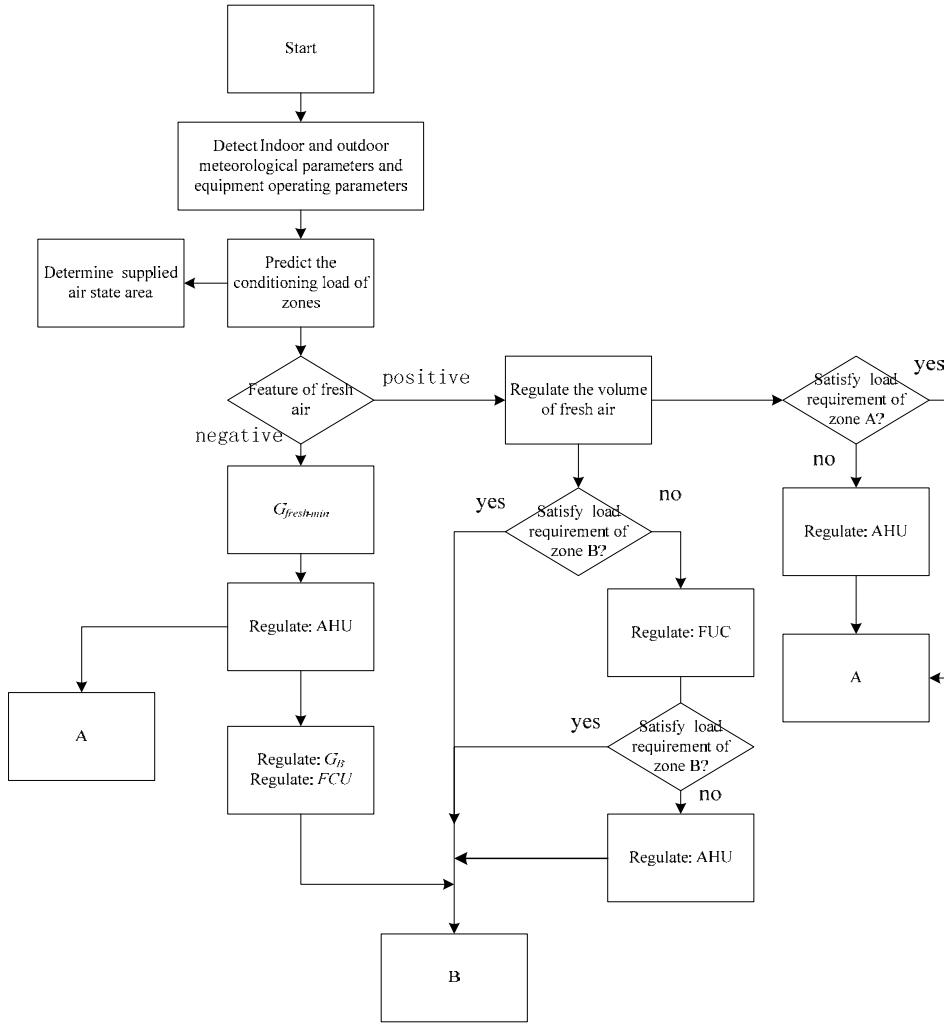


Figure 4 Flow diagram of Energy efficient management strategy for AC system

3. ENERGY EFFICIENT ANALYSIS

3.3 Energy consumption model

Supplied air enters zone A and zone B. In the zone A, the AC load is all drain away by the entering air, while all of the latent load and part of sensible load named air-cooled load is treated by supplied air and dry FCUs are in charge of the remaining sensible load named water-cooled load in the zone B.

Composition of the entering airflow:

$$G = G_A + G_B = G_{fresh} + G_{return}$$

Composition of the system sensible load, latent load and total energy consumption:

$$Q = Q_B + Q_A + Q_{fresh} = Q_{B1} + Q_{B2} + Q_A + Q_{fresh}$$

$$W = W_B + W_A + W_{fresh} = W_{B1} + 0 + W_A + W_{fresh}$$

$$P = P_B + P_A + P_{fresh} = P_{B1} + P_{B2} + P_A + P_{fresh}$$

In the zone A, the energy consumption was mainly composed of handling and conveying energy consumption for the supplied air of G_A , including the

energy consumption of low temperature chillers, chilled water pumps, fans, cooling towers, etc.

Energy consumption composition of zone A:

$$P_A = P_{chiller1A} + P_{CHWpump1A} + P_{cfan1A} + P_{CWpump1A}$$

The energy consumption of air-cooled in zone B was mainly composed of handling and conveying energy consumption for the supplied air of G_B , including: energy consumption of low-temperature chillers in the AHU, the energy consumption of chilled water pumps, cooling tower energy consumption and transport fan energy consumption; the energy consumption of water-cooled load in the zone B, which is mainly from FCU, including: high-temperature chiller energy consumption, chilled water pump energy consumption, cooling tower energy consumption and the fan-coil's fan energy consumption.

Composition and distribution of the energy consumption of zone B:

$$P_B = P_{B1} + P_{B2}$$

$$P_{B1} = P_{chiller1B} + P_{CHWpump1B} + P_{cfan1B} + P_{CT1B}$$

$$P_{B2} = P_{chiller2} + P_{CHW2pump} + P_{cfan2} + P_{CT2}$$

3.4 Energy efficiency analysis

Based on the previous system description, we can see that the impact of the air-conditioned area load change mainly contains three aspects: the introduction of outside air, handling the hybrid wind in the AHU and local cooling in FCU. The organic combination of the three basic stages compose different AC scheme. Analyze the energy consumed in the three basic stages to provide unit load analysis for the system, and then compare the energy consumption of different AC scheme, finally draw up an optimal energy-saving management strategy.

The energy consumption of the introduction of fresh air is only the conveying consumption.

The energy consumption of dehumidifying the hybrid wind in the AHU is including the consumption of low temperature chillers, chilled water pumps, fans, cooling towers of the AHU.

$$P_{AHU} = P_{chiller1} + P_{CHWpump1} + P_{fan1} + P_{CT1} \quad (1)$$

The energy consumption of FCU is including: high temperature chillers, chilled water pumps, fans, cooling towers of the FCU.

$$P_{FCU} = P_{chiller2} + P_{CHW2pump} + P_{fan2} + P_{CT2} \quad (2)$$

Comparing the three stages, when the load of introduced fresh air is positive for load of the AC system, the efficiency is the highest due to its low consumption; on the contrary, when the fresh load is passive to the system, the efficiency is negative and lowest.

The energy consumption of FCU and AHU is compared for zone B. Comparing the formula (1) and formula (2), when water pumps, fans and cooling towers in the FCU and AHU operate to provide same load for the zone B on a similar condition, difference of their consumption is in a small limited range. However, Energy consumption of chillers makes up a large percent of total consumption, and the COP of high temperature chillers in FCU is much higher than low temperature chillers in FCU. In the result, energy consumption of FCU to provide per unit load for system is lower than AHU's.

It comes to the conclusion:

$$P_{fresh-p} < P_{FCU} < P_{AHU} < P_{fresh-n} \quad (3)$$

p is energy consumption to provide per unit load for system.

3.5 Energy efficient management strategy

Since the operation scheme of AC system is organically combined of the three stages, as showed in Figure 4, energy efficient management strategy can be draw up on the basis of formula(3).

4. MANAGEMENT TECHNOLOGY

4.1. Load prediction

The accurate prediction of the hourly load is the premise of modern control of the AC system. Most used is based on the BP algorithm which is a static feedforward network. Static neural networks to identify the dynamic system, in fact, equal to solve the dynamic modeling problem as a static modeling. Some investigations have been carried out to show that the dynamic regression neural network can be more vivid, more directly reflect the characteristics of dynamic system [5,6]. This paper discusses the Elman regression neural network to predict the load of the system.

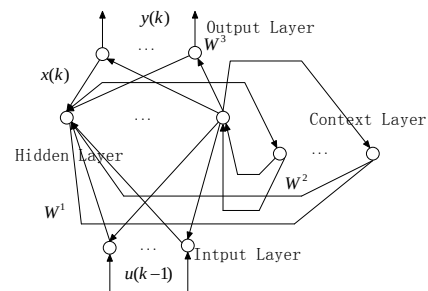


Figure 5 Load prediction model based on Elman networks

As shown in Figure 5, set outside input vector of the network $u(k-1) \in R^r$, output vector of the network $y(k) \in R^m$, and input vector of hidden layer $x(k) \in R^r$, we can establish the following nonlinear state space expression.

$$x(k) = f(W^1 x_c(k) + W^2 u(k-1))$$

$$x_c(k) = x(k-1)$$

$$y(k) = g(W^3 x(k))$$

$$x_c(k) = x(k-1) = f \left[(w_{k-1}^1 x_c(k-1) + w_{k-1}^2 u(k-2)) \right]$$

$f(\bullet)$ is the activation function of hidden layer nodes. $g(\bullet)$ is the activation function of output layer. W^3 , W^2 , W^1 separately represent connection weight matrix from hidden layer to output layer, input layer to hidden layer and context layer to hidden layer. Considering $x_c(k-1) = x(k-2)$, further expansion formula can be solved, which explains that $x_c(k)$ varies with the connection weights of the different times in the past and shows that it is a dynamic recursive process.

AC system hourly load is a set of numbers sequence in chronological order, which have certain statistical and random relationship between them, so it is difficult to accurately function with some equations. Time series analysis method can be used for load prediction. Time series prediction using Elman neural network is to train the network with sample data and predict future data. In the model, use the previous N moment data of indoor and outdoor meteorology

parameters to predict the next moment load of zone A and zone B, and predict in a rolling way.

4.2. Determination of the supply air state

In the actual project, the temperature and relative humidity of zones does not require to an accurate point, while are allowed fluctuating within a certain range. Zone A and zone B share the same supply air state in the centralized all-air AC system. Solve a tolerate area of the supply air state is a basis for energy efficient management. In this paper, firstly solve a tolerate area of supply air state for zone A using the load prediction of zone A, and then find the optimal point of supply air state in the tolerate area though the proposed energy efficient management strategy to guarantee the zone B in a tolerate area too, and realize energy conservation.

Based on the heat balance and wet balance equation and load prediction result we can figure out a tolerate area of the supply air state for air-conditioned zone A [4]. As shown in Figure 6, N is the object area with a temperature range from t_{nL} to t_{nH} and a relative humidity range from ϕ_L to ϕ_H , and S is the tolerate area of the supply air state which is figure out for air-conditioned zone A.

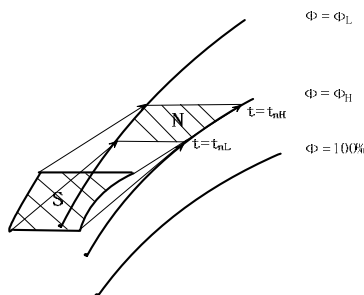


Figure 6 Psychrometric chart for supply air state area

4.3. Load allocation for high load zone

Since zone B is equipped with a dry FCU, there are certain requirements on the humidity of the zone in order to avoid the generation of condensation. Therefore, control of the dry fan coil system should first ensure that the indoor humidity to meet the minimum dehumidification requirements which is set to avoid condensation, and ensure its performance on cooling. When the cooling capacity is insufficient for the zone load, the AHU should take care of remaining cooling load.

Therefore, the minimum value of the entering air volume G_{B-min} should meet the three conditions at the

same time: minimum air volume of fresh air demand for zone B, the supplementary cooling load when maximum load from FCU is still less than the required load and minimum dehumidification requirements against condensation. So:

$$G_{B-min} = \max(G_{demand}, \frac{Q_B - Q_{B2-max}}{h_n - h_s}, \frac{1000W_{B-min}}{d_n - d_s})$$

The load of zone B is divided by the fresh air load, the air-cooled load of AHU and the water-cooled load of FCU. Since the zone B and zone A shared the same state, when the state is fixed we can only adjust the volume of the supply air of zone B to choose the air-cooled load. Therefore, choose of the value of G_B is the essential for the load allocation in the zone B.

The expression (3) shows that the energy efficiency drops gradually in the order of positive load of fresh air, cooling load from FUC, cooling load from AHU and negative load of fresh air. So, when the load of fresh air is positive, we should increase the volume of supply air according to actual required load, if the passive load is not enough against the excess load, utilize FCU to provide the complementary load, if the load is still not enough, finally adopt AHU to provide the remaining load. When the load of fresh air is negative, adopt G_{B-min} , increase the load rate provided by FCU, and if the load provided is still not enough, adopt AHU to provide the remaining load.

5. APPLICATION ANALYSIS

Energy-saving operation and management of the AC system runs in seasonal weathers. Based on the energy efficient management strategy for AC system of underground engineering, analyze the specific application under different climatic conditions, and obtain the corresponding operation schemes.

In Figure 7, point R_w and point R_s are typical samples of outdoor air state in winter and summer, and the state point 1 to 5 are the samples of outdoor air state point in transition season. The scheme at the point 1 and point 2 is combined to the table1.

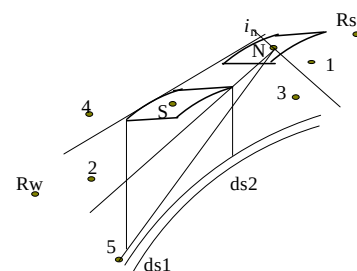


Figure 7 Samples of seasonal climate

Table 1 Management strategy under typical winter and summer condition

	Summer and Point-1		Winter and Point-2	
	Zone_A	Zone_B	Zone_A	Zone_B
Feature of fresh air load	Negative (wet and hot)	Negative (wet and hot)	Positive (dry and cold)	Positive (dry and cold)
Volume of Fresh air	$G_{A-fresh-min}$	$G_{B-fresh-min}$	Regulating: $Q_{A-fresh}=Q_A$	If $Q_B < Q_{B-fresh-max}$ Regulating: $Q_{B-fresh}=Q_B$ Else: $G_{B-fresh-max}$
G_B	0	G_{B-min}	0	If $Q_B < Q_{B-fresh-max}$ Regulating: $Q_{B-fresh}=Q_B$ Or: G_{B-max}
Load of FCU	0	Q_{B2-max}	0	If $Q_B < Q_{B-fresh-max}$: OFF Else Regulating: Q_B-Q_{B-max}
Load of AHU	$Q_A+Q_{A-fresh-min}$	$Q_B+Q_{B-fresh-min}-Q_{B2-max}$	0	0
Operating mode of AHU	Cooling dehumidity		OFF	

Table 2 Management strategy under some transition season condition

	Transition season					
	Point-3		Point-4		Point-5	
	A	B	A	B	A	B
Feature of fresh air load zone						
Volume of Fresh air	$G_{A-fresh-max}$	$G_{B-fresh-max}$	$G_{A-fresh-min}$	$G_{B-fresh-min}$	$G_{A-fresh-min}$	$G_{B-fresh-min}$
G_B	0	G_{B-max}	0	G_{B-min}	0	G_{B-min}
Load of FCU	0	Q_{B2-max}	0	Q_{B2-max}	0	Q_{B2-max}
Load of AHU	Q_A- $Q_{A-fresh-max}$	$Q_B-Q_{B2-max}-$ $Q_{B-fresh-max}$	Q_A+ $Q_{A-fresh-min}$	Q_B-Q_{B2-max} $+Q_{B-fresh-min}$	Q_A+ $Q_{A-fresh-min}$	Q_B-Q_{B2-max} $+Q_{B-fresh-min}$
Operating mode of AHU	Cooling dehumidity		Cooling on dry condition		Heating	

6. CONCLUSION

Based on the energy consumption model, energy efficient management strategy is established for AC system with local cooling units against high load zone in the underground engineering. The management strategy functions with three AC technologies: the prediction of AC load, the determination of supply air state and the load allocation for high load zone. The strategy avoids too much calculation load and can help to find an optimal AC scheme which meets the requirements of environment and allows the AC system running in a low energy consumption way in underground engineering.

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A STUDY OF COOLING LOAD CALCULATION OF STRATIFIED AIR CONDITIONING SYSTEM BASED ON THE SIMULTANEOUSLY SOLVING MODEL

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ABSTRACT

A calculation method for cooling load of stratified air conditioning system based on the simultaneously solving model is presented. Thermal environment experiment under summer condition is carried out in a large space building, indoor air temperature, inner wall surface temperature and cooling capacity were obtained; the stratified air conditioning cooling load calculated by the simultaneously solving model is compared with air conditioning cooling capacity in experiment, and it shows a good agreement, the relative error is 14%, taking the loss of cooling capacity and leakage loss, it can be thought that the simultaneously solving model can be used to calculate cooling load of stratified air conditioning system.

Keywords: Simultaneously solving model, Wall temperature, Cooling load, Vertical temperature distribution

1. INTRODUCTION

The air distribution of stratified air conditioning system is usually adopted in large space buildings, which has a different cooling load calculation method. Currently, the stratified air conditioning cooling load is calculated with the method proposed in "Practical Heating and Air Conditioning Design Manual" compiled by Professor Yaoqing Lu. The method recommends the results of the research conducted by "Stratified Air Conditioning Technology Research Group" in the charge of the Institute of Air Conditioning at China Academy of Building Research and the former Design Institute at First Machine-Building Ministry in the 1980s. The main idea of the method is to analyze the effect of the non-air-conditioned zone on the air-conditioned zone taking the heat of radiant transfer and heat of convective transfer into account. Other parts of load calculation are almost the same with those of the typical air conditioning systems. Chunhua Yang has also designed calculation software for this method. However, a uniform stratified air conditioning system cooling load calculation method for air distribution with low-side air supply system hasn't been proposed yet. Besides, most of the methods are restricted in the air distribution with air supply at low levels and air return at the ceiling level of the buildings with typical heights. Guorong Yang et al have comprehensively analyzed and summarized the typical cooling load calculation methods of this kind of whole space air conditioning systems with air supply at low levels and air return at high levels: 1)the cooling load calculation method for displacement ventilation

systems; 2)the cooling load method for the low-level air supply based on the high and low zones of the stratified theory; 3)the cooling load calculation method using a correction factor for low-level air supply systems; 4)the cooling load calculation method for typical high-level air supply systems with the vague stationary front. But few researches of the stratified air conditioning system with low-side air supply system have been conducted.

Aiming at the circumstance, this paper proposes a cooling load calculation method for stratified air conditioning system based on a simultaneously solving model, and discusses the possibility and accuracy of this model.

2. THE FOUNDATION OF THE MATHEMATICAL MODEL

Details of the simultaneously solving model is given before^[8], here is the simple introduction. The basic components of the simultaneously solving model for the air temperature and the inner surface temperature are shown in Fig. 1, in which $t(i, k)$ is the air temperature of the i^{th} body flow layer, $m_c(i+1)$ is the air mass flow rate from the $(i+1)^{\text{th}}$ body flow layer to the i^{th} body flow layer, m_{out} and m_{in} are the air mass exchanges between the wall flow and the body flow, m_d is the air mass exchange between two certain wall flow layers, t_m is the mean temperature of a wall flow layer, m_s , m_j and m_h are the supply air mass flow rate, return air mass flow rate and entrainment volume, $q(i-(i-1))$ is the heat exchange caused by temperature difference between the i^{th} and the $(i-1)^{\text{th}}$ body flow layer. In this model, the energy equations of the body flow model and the radiant model are composed as the functions of

$t(i, k)$ and $t_b(i, k)$. The energy equation of the radiant model can be expressed by Eq. (1), while the energy equation of body flow is Eq. (2)^[8].

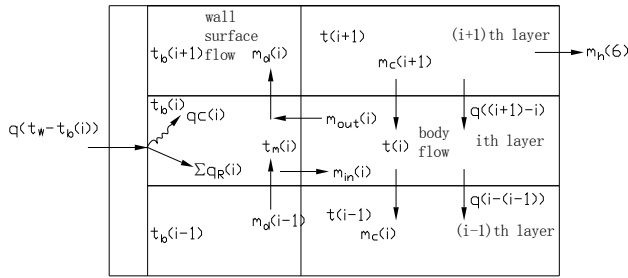


Figure.1 Basic components of the simultaneously solving model

$$\begin{aligned}
 & \alpha_d \cdot T(I) - [\alpha_d + 4T_{bm}^3 \cdot \varepsilon_i \cdot \sigma \cdot (\sum_{j=1}^{I-1} F_{ij} + \sum_{j=I+1}^N F_{ij} + F_{if} + F_{ic} + \alpha_w)] \cdot T_{bi} \\
 & + 4T_{bm}^3 \cdot \varepsilon_i \cdot \sigma \cdot [\sum_{j=1}^{I-1} (F_{ij} \cdot T_{bj}) + \sum_{j=I+1}^N (F_{ij} \cdot T_{bj}) + F_{if} \cdot T_f + F_{ic} \cdot T_c] + \alpha_w \cdot T_a = 0 \\
 & 0 = \sum_{K=1}^m C \cdot m_{IN}(I, K) \cdot [T_M(I, K) - T(I)] \\
 & + C_B(I) \cdot A_B(I) \cdot [T(I-1) - T(I)] + C_B(I+1) \cdot A_B(I) \cdot [T(I+1) - T(I)] \\
 & + C \cdot m_c(I+1) \cdot [T(I+1) - T(I)] - C \cdot m_c(I) \cdot [T(I-1) - T(I)] + C \cdot m'_s \cdot [T'_s - T(J)] \\
 & q_{cl} = \alpha_f A_f [t_f - t(1)] + \alpha_c A_c [t_c - t(n)] \\
 & + \sum_{i=1}^n \alpha_{di} A_i [\theta_{wi} - t(i)]
 \end{aligned} \tag{1}$$

The simultaneously solving model can be used to calculate the vertical indoor air temperature distribution, the vertical inner surface temperature distribution and the convective heat transfer coefficient between the air and inner wall surface. By analyzing the simultaneously solving model, the cooling loads of the air-conditioned zone under summer conditions is found including the cooling loads caused by 1) the heat source of the air-conditioned zone; 2) convective heat transfer between the air and the floor; 3) the air transfer from the non-air-conditioned zone to the air-conditioned zone; 4) the thermal energy of the return flow from the body flow into the wall flow; 5) the temperature difference between the non-air-conditioned zone and the air-conditioned zone. Besides, all the cooling loads mentioned above each include the heat of convective transfer and the heat of radiant transfer referred in the traditional calculation method. Thus, the cooling load of the stratified air conditioning system can be calculated by Eq. (3) below:

Where, α_f is the convective heat transfer coefficient ($W \cdot m^{-2} \cdot ^\circ C^{-1}$) between the floor and the first air body flow layer, A_f is the area(m^2) of the floor, t_f is the temperature($^\circ C$) of the floor, $t(1)$ is the temperature($^\circ C$) of the first body flow layer, α_c is the convective heat transfer coefficient($W \cdot m^{-2} \cdot ^\circ C^{-1}$) between the ceiling and the n th body flow layer, A_c is the area(m^2) of the ceiling, t_c is the temperature($^\circ C$) of the ceiling, $t(n)$ is the temperature($^\circ C$) of the n th body

flow layer, α_{di} is the convective heat transfer coefficient($W \cdot m^{-2} \cdot ^\circ C^{-1}$) between the i th inner surface layer and the i th body flow layer, A_i is the area(m^2) of the i th inner surface layer, θ_{wi} is the temperature($^\circ C$) of the i th inner surface, $t(i)$ is the temperature($^\circ C$) of the i th body flow layer.

3. EXPERIMENTAL STUDY

3.1 The experimental testing program

Indoor thermal experiment under the summer condition has been taken in the large space experimental base of University of Shanghai for Science and Technology in order to get the actual vertical air temperature gradient and the cooling capacity of the air conditioning system. The experimental base has dimensions of 20m×14.8m×6m, and a sloping roof, which has a highest point of 8.75m. Modular air cooling heat pump chiller is adopt as the cold source, which cooling capacity is 57kW, heating capacity is 73kW, COP is 3.4. The AHU is primary return air system; rated air volume is 20000m³/h, the fresh air is shut down when test. The experiment adopts nozzles as the air outlets. The locations of the nozzles are shown in Fig.2. 10 nozzles are installed at 4m on the west wall, each having a diameter of 174 mm. 4 air return grilles with dimensions of 600 mm×900 mm are installed below nozzles.

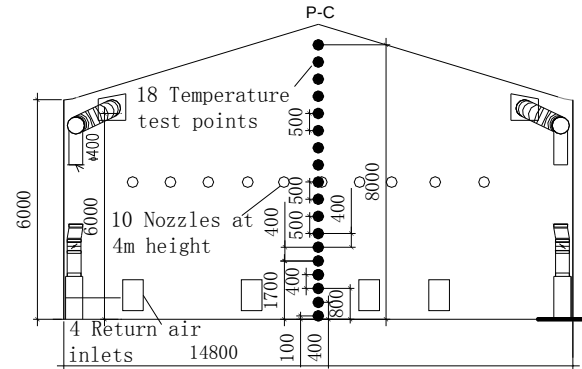


Figure.2 The locations of the nozzles and the measuring points

Considering the characteristics of the air distribution of the air jets, a vertical temperature measuring line (P-C) is positioned at the center of the room. The first measure point is at 0.1m height from the floor. Other measurements are performed at different distances from the floor, from 0.1m to 2.5m with step of 0.4m, from 2.5m to 8.75m with step of 0.5m. The temperatures of the supply air, return air and indoor air are all measured with K-type thermocouples, and are automatically recorded by Agilent data acquisition machine. The temperatures of the inner surfaces are measured by a thermal infrared imager, and are handled by ThermaCAM Research Pro2.7 software. The velocity of the supply air and return air are tested by a

Testo 435 anemometer. All the instruments have been calibrated by a water bath or an authority.

3.2 Testing results and the analysis

The supply air volume is 7600m³/h. In order to obtain the cooling capacity under steady condition, Fig.2 shows the trends over time of the indoor temperature of 1.7m height along the P-C, supply air temperature, return air temperature. It can be seen from the graph, the indoor temperature of 1.7m height and return air temperature is consistent and stable after 11:30; the analyzed time quantum can be after 11:30; According to supply air temperature, return air temperature and supply air volume, the AHU cooling capacity can be calculated for each time, which also shows in Fig.2, after 11:30, it also can be considered steady. For cooling load calculation below, the data at 12:00 is chose and listed in Table 1.

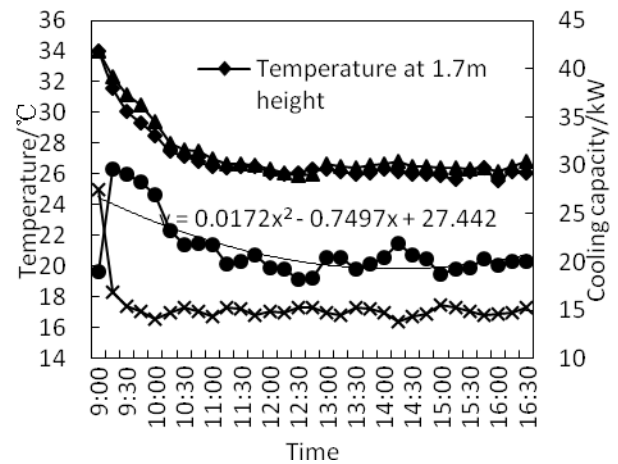


Figure.3 Test data trends over time

Table 1 Testing results at 12:00

Outdoor air temperature/ °C	Horizontal solar radiation / W/m ²	Vertical solar radiation / W/m ²	Supply air temperature/ °C	Supply air volume/ m ³ /h	Return air temperature /°C	Floor temperature e/°C	Ceiling temperature/ °C	Cooling capacity /W
35	445	300	17.1	7600	26.3	30.3	36.6	19422

4. CALCULATION BY MODEL AND ANALYSIS

According to the model, the testing building is divided into 8 layers along the height, the construction parameters are showed in Table2. The indoor air temperature and inner wall surface temperature, heat transfer coefficient and cooling load of each layer are calculated by simultaneously solving model (Eq. (1), Eq. (2)) and cooling load model (Eq. (3)) of the stratified air conditioning system.

Fig.4 shows both the calculated results and the experimental data of vertical inner surface temperature distribution and vertical air temperature distribution. In this figure, when the nozzles are at 4m (the 5th layer), there is an apparent stationary front at 5m around, above which is the non-air-conditioned zone, and below is the air-conditioned zone. The vertical temperature distribution of the air-conditioned zone is relatively uniform. Because of the effect of the heat transfer between the floor and the air, there is a gradient of the air temperature near the floor. But the human thermal comfort can be satisfied in general because the difference between the air temperature at 0.1m and that at 1.1m is less than 3°C. It can be seen from the figure that the indoor air temperature distribution and the inner surface temperature distribution of both calculation and experiment are in good agreement. The max relative error between calculation and testing air temperature is 13.5%, which is 17% between calculation and testing inner wall surface temperature. It can be concluded that,

the calculation results can be used for cooling load calculation.

Table2 Input data of construction for model

Number of layers	Supply air layer	Return air layer	Inner wall surface area of each layer/ m ²	Area of ceiling / m ²	Area of floor / m ²
8	5	1	109	316	296

Table3 shows the cooling load for each layer, the sum cooling load of heat transfer between ceilings and the 8th layer, floor and 1th layer is about 30% of total value. In the remaining (about 70%), because of the nozzle air supply, heat transfer cooling load is maximum in air supply layer, and the variation trend of cooling load agree with the real situation; The total cooling load at 12:00 calculated by the model is 17028W, compared with 19422W listed in Fig.1, the relative error is 14%. Taking the loss of cooling capacity and leakage loss, the relative error is reasonable, it can be thought that the simultaneously solving model can be used to calculate cooling load of stratified air conditioning system.

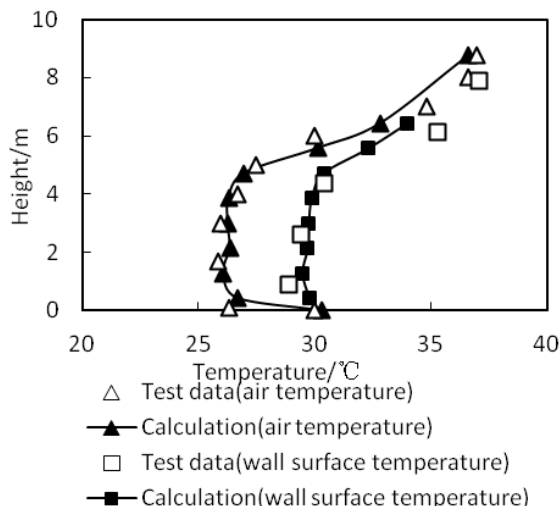


Figure.4 Validation of the vertical indoor air temperature distribution and vertical inner surface temperature distribution

Table3 cooling load calculation result for each layer

Lay ers	Air Temp eratur e/°C	Inner wall temp eratur e/°C	Heat transfer coeffici ent/W/ m ² °C	Coolin g load/ W	Sum/W
Ceiling	/	36.6	3	2496	17028
8	32.8	34.0	3.12	387	
7	30.1	32.3	4.45	1052	
6	27.0	30.4	4.62	1719	
5	26.3	29.9	4.96	1929	
4	26.3	29.8	4.92	1865	
3	26.4	29.7	4.8	1751	
2	26.1	29.5	4.82	1783	
1	26.7	29.8	4.65	1559	
Floor	/	30.3	2.33	2486	

5. CONCLUSION

In this paper, two conclusions were obtained as followed.

1. According to the results of calculation and test for air and inner surface wall temperature, the max relative error is 17%, the calculation results can be used for cooling load calculation.

2. Taking the loss of cooling capacity and leakage loss, the simultaneously solving model can be used to calculate cooling load of stratified air conditioning system.

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RESEARCH ON THE ENERGY-SAVING AND HEAT TRANSFER PERFORMANCE OF WASTEWATER SOURCE HEAT PUMP

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Abstract

There is a large potential in the heat loss from the urban wastewater. In all kinds of low-grade heat sources, urban wastewater is gradually widely used because of its advantages such as huge quantities, small variation of temperature, warm in winter and cool in summer, etc. By integrating a heat pump to utilize this heat, the wastewater source heat pump (WWSHP), which is a new water reuse and treatment strategy to tackle the water problems of China and to realize the sustainable development, recovers the heat of wastewater. The WWSHP air conditioning system as residential application can reduce the need of energy systems based on fossil fuel for cooling, heating, and hot water's supply purposes. The system's working principle, classification and the wastewater's characteristic parameters were discussed in this paper. And the scheme of using WWSHP system for central air-conditioning compared with other refrigerating and heating equipments was analyzed in detail. Taking a building in middle-and-lower reaches of Yangtze River in China as an example, the energy-saving and cost-effective analysis on using urban wastewater source heat pump technology for heating and cooling were discussed in detail. The investigation into COP (Coefficient of Performance) and PER (Primary Energy Ratio) was shown in this paper. It was discovered that the WWSHP system's PER in heating condition reaches to 1.509, the PER in cooling condition reaches to 1.317, and the PER in annual operating condition reaches to 1.386. Among the four kinds of common air conditioning units, when the loads are same respectively, the WWSHP has the most significant energy-saving effect. It is the advanced energy-saving cooling and heating equipment. Meanwhile, based on the cost-effective analysis, the WWSHP system's operating cost is derived to be lowest, which is 47.1 RMB Yuan/m². The WWSHP air conditioning system will be feasible as residential application. Furthermore, mechanism of heat transfer enhancement for falling-film evaporation and condensation surface of tubes packed with wire-mesh is also studied in detail. And through analyzing the effect of tube bundle arrangements with this kind of falling-film surface to the heat transfer performance, the optimal tube bundle arrangement will be obtained. Theoretical simulation will be carried out for the heat transfer process and the flow of falling-film outside horizontal tubes. Based on these results, improvement of controlling parameters influencing the evaporation and condensation performance could be made, and then led to an increase in the refrigerating quantity and the COP. Consequently, this WWSHP system has its theoretical significance and promising engineering application potentials for the residential refrigerating and air-conditioning system using low-temperature waste heat source.

Keywords: Wastewater source heat pump, wastewater reuse, energy-saving, heat transfer

1. INTRODUCTION

China is facing water crisis. More than 400 cities are lacking enough water resources and more than half rivers are polluted. To realize sustainable development in the 21st century, we should first provide enough water resources to every person and keep a friendly, leisurely water environment. Under the idea of sustainability, concept for a sustainable urban sewerage system is put forward [1]. An urban sewerage system cannot only be a basic facility for draining rainwater and wastewater to protect the urban environment and public water bodies, but must also contribute to the restoration of the water environment in order to maintain a healthy social water cycle. Wastewater source heat pump is an application apparatus hosted in the urban sewage heat recovery and the effective

utilization. Environmental improvements require waste minimization, reduced air pollution from vehicles, distributed energy generation, reduction of greenhouse gas emissions [2]. To monitor emission of these greenhouse emissions, an agreement was made with the overall pollution prevention targets, the objectives of the Kyoto Protocol agreement [3]. Heat pumps can either be fossil fired or driven by heat from combined heating and power (CHP) systems or other sources. In addition, it has outstanding energy saving effect since it is operated at high coefficient of performance (COP) without air pollution. Utilizing solar energy, geothermal energy, waste heat and exhaust heat as the heat source to complete the refrigerating and air conditioning process is an important way to develop more efficient utilization of energy and launch the technological transformation taking energy-saving as its key part.

NOMENCLATURE

Abbreviation		Subscripts	
WWSHP	Waste Water Source Heat Pump	-r	heating condition
COP	Coefficient of Performance	-l	cooling condition
PER	Primary Energy Ratio	-o	annual operating condition
Symbols		Greek Letters	
Q	Energy quantity (W)	η	Efficiency (-)
W	Electricity consumption (W)		

The wastewater source heat pump (WWSHP), which is a new water use and treatment strategy to tackle the water problems of China and to realize the sustainable development, recovers the heat of wastewater. The WWSHP air conditioning system as residential applications is feasible and can reduce the need of energy systems based on fossil fuel for cooling, heating, and hot water's supply purposes.

Due to the temperature of wastewater from wastewater treatment plant is relatively stable in the four seasons. The temperature is 10 to 16 °C in winter, which is higher than air temperature. The temperature is 20 to 26 °C in summer, which is lower than air temperature. According to the characteristic parameters of the wastewater, ground surface water, and groundwater, WWSHP exchanges heat with wastewater through the heat exchanger. In winter, the treated wastewater flows into pipes of evaporator and the heat of wastewater is absorbed by the refrigerant in evaporator, and then the heat from the condenser is transmitted to buildings through the air conditioning system. At the time, the wastewater heat is regarded as the "heat source". In summer, the treated wastewater flows into pipes of condenser and absorbs heat from condenser, and then the heat from the evaporator after the refrigerating cycle is transmitted to buildings through the air conditioning system. At the time, the wastewater heat is regarded as the "cold source".

Recently, researchers have carried out much research about urban WWSHP systems, including original sewage source heat pump system [4]. In Japan and Korea, Baek et al. [5] did a simulation study of district cooling and heating systems using sewage water as an energy source. Results show that compared with conventional air-source heat pumps, WWSHP could help reducing energy consumption by 34%, lowering the emission of carbon dioxide (CO₂) by 68% and controlling the generation of nitrogen oxides (NO_x) by 75%. Sun et al. [6-9] studied the soft-dirt characteristic of the heat-exchanging pipe and its effects on the performance of an urban sewage source heat pump (USSHP) system, and studied the technical and economic analysis of the temperature rise of heat pump in the sewage disposal process.

In the past studies of our research group, the wastewater characteristic parameters in Xi'an, China, and the feasibility analysis on WWSHP system using urban treated or untreated wastewater have been already evaluated [10].

The purposes of this study are to use the WWSHP system through wastewater heat recovery and to evaluate the system's classification, characteristics, and

the economic analysis. Then the scheme of using WWSHP system for central air-conditioning compared with other refrigerating and heating equipments was analyzed. Taking a building located in Changsha city in middle-and-lower reaches of Yangtze River in China as an example, the energy-saving analysis on using urban wastewater source heat pump technology for heating and cooling in annual operating condition was discussed in detail. Furthermore, mechanism of heat transfer enhancement for falling-film evaporation and condensation surface of tubes packed with wire-mesh is also studied in detail. And through analyzing the effect of tube bundle arrangements with this kind of falling-film surface to the heat transfer performance, the optimal tube bundle arrangement will be obtained. Theoretical simulation will be carried out for the heat transfer process and the flow of falling-film outside horizontal tubes. Based on these results, improvement of controlling parameters influencing the evaporation and condensation performance could be made, and then led to an increase in the refrigerating quantity and the COP. Consequently, this WWSHP system has its theoretical significance and promising engineering application potentials for the residential refrigerating and air-conditioning system using low-temperature waste heat source.

2. WORKING PRINCIPLE

According to whether there is the direct heat exchange between the wastewater and the heat exchanger in heat pump or not, that is to say, whether the wastewater directly flows into the evaporator or the condenser of the WWSHP or not, WWSHP system can be divided into two kinds of system. One is the system of direct using wastewater source, the other is indirectly using wastewater source.

Direct-type system has higher requirements for the source and quality of the water flowing into the condenser or the evaporator, which also should be designed to adapt to poor water quality. It needs reliable anti-clogging, anti-pollution and anticorrosion ability. The secondary effluent of municipal sewage treatment, some industrial wastewater, groundwater, and water from rivers, seas or lakes, are usually restricted to use in direct-type system due these water qualities are relatively better.

Indirect-type system has lower requirements for water quality. But this kind of system needs to add a waste water heat exchanger and then intermediate water adopted as reuse water flows into the evaporator or the condenser. By the anti-blocking and anti-fouling

technology, even the original urban sewage can be used indirectly in the whole system.

For indirect-type system, when the quantity of original urban sewage is shortage, the WWSHP unit should be in variable condition operation. Antifreeze liquid should be added into the intermediate water in winter thereafter the evaporation temperature can be reduced below 0°C, whilst the condensing temperature should be increased in summer, so as to increase the water temperature difference to improve the heat exchange.

Compared with the indirect-type system, direct-type system has higher heat transfer efficiency and simpler system. Based on thermodynamics analysis, heat resistance increases and the using interval of sewage temperature difference is reduced owing to the existence of the intermediate water as an intermediary medium. Thus the system efficiency falls subsequently. Consequently, direct-type system will become the mainstream of future system.

The designed indirect-type and direct-type WWSHP system is shown in Figure 1.

In the winter, valves 1–4 are open, and valves 5–8 are closed. The wastewater heat exchanger takes the heat from the second effluent municipal wastewater treatment plant or the original urban sewage filtered through filth block device, and sends it to the evaporator side of the heat pump as a low grade heat source. The compression heat pump system releases the heat at the condenser side, and sends the heat to the heat-supplying system. That is the heating cycle of the WWSHP system.

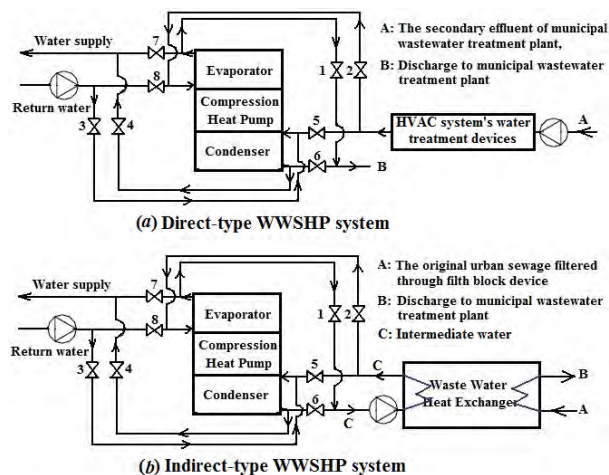


Figure 1 Operation principle of WWSHP system

In the summer, valves 1–4 are closed, and valves 5–8 are open. The condensing heat is released to the treated wastewater or filtered original sewage through the wastewater heat exchanger. The compression heat pump system absorbs the ambient heat at the evaporator side. Thus the temperature of the coolant water decreases to cool down the ambient temperature of the end users. That is the refrigerating cycle of the WWSHP system.

As shown in Figure 1, the urban original sewage source heat pump system is also working as this

principle. But it needs to add filtrating equipments in the system, such as filth block device. Before the urban original sewage flows into the wastewater heat exchanger, it must be filtered through filth block device to reduce the corrosion and extend the service life of the wastewater heat exchanger. Meantime, spray heat exchanger should be used as the wastewater heat exchanger because spray heat exchanger has higher anti-blocking and anti-fouling performance than shell and tube heat exchanger or immersion heat exchanger. Also it has higher heat transfer coefficient.

The combination of the wastewater heat exchanger and filth block device is equivalent to the cooling tower in traditional air conditioning system.

3. ENERGY-SAVING ANALYSIS

COP is commonly used to show a system's performance and energy efficiency in refrigeration unit, which is the ratio of energy output and energy input. The greater the COP is, the higher the energy utilization efficiency is.

This article also adopted primary energy ratio (PER) to analyze the ratio of the air conditioning system's energy gain (Q_{gain}) and the primary energy consumption (Q_p), and to compare the WWSHP unit with other air conditioning units applied widely such as air source heat pump, electric boiler + water chilling unit, direct-fire lithium bromide absorption refrigeration unit, etc.

3.1 PER in Heating Condition

PER in heating condition (PER_r) can be expressed as equation (1).

$$PER_r = \frac{Q_r}{Q_p} = \frac{Q_r}{W} \frac{W}{Q_p} = COP_r \eta_f (1 - \eta_s)(1 - \eta_e) \quad (1)$$

Where Q_r , W , COP_r , η_f , η_s , and η_e are the heating capacity, the electricity consumption, COP in heating condition, the power generation efficiency of the primary energy, the power grid line loss rate, and the equipment loss rate, respectively. Here, in China, following the Annual Report on the Development of China's Electric Power Industry in 2012, η_f is 37.34% and η_s is 6.52%. And in common, we adopt η_e as 8%. For direct-fire lithium bromide absorption refrigeration unit, it is equivalent to a gas fired boiler and is not driven by electricity when it is in heating condition. Thus its heating efficiency, that is COP_r , is equal to that of a gas fired boiler, which is 93%. According to the regular COP in heating condition of common air conditioning units, COP_r data are shown in Table 1. Therefore, PER_r data are derived by equation (1) and also shown in Table 1.

As shown in Table 1, through comparisons of COP_r and PER_r between WWSHP and other air conditioning units, the WWSHP system's PER_r is highest and it reaches to 1.509. As indicated in Table 1, WWSHP is the advanced energy-saving heating equipment.

Table 1 Comparisons of COP and PER in heating condition between WWSHP and other air conditioning units

Air Conditioning Units	COP _r	PER _r
Air source heat pump	3.5	1.124
Electric boiler + water chilling unit	0.96	0.308
Direct-fire LiBr absorption refrigeration unit	0.93	0.856
WWSHP	4.7	1.509

3.2 PER in Cooling Condition

PER in cooling condition (PER_i) can be expressed as equation (2).

$$PER_i = \frac{Q_l}{Q_p} = \frac{Q_l}{W} \cdot \frac{W}{Q_p} = COP_i \eta_f (1 - \eta_s)(1 - \eta_e) \quad (2)$$

Where η_f , η_s , and η_e are the same values as those above-mentioned. Q_l and COP_i are the refrigerating capacity and COP in cooling condition. According to the regular COP in cooling condition of common air conditioning units, COP_i data are shown in Table 2. For direct-fire lithium bromide absorption refrigeration unit, due it is not driven by electricity, its PER_i can be derived by that the cooling efficiency, that is COP_i , times $(1 - \eta_e)$, which is 92%. Other PER_i data are derived by equation (2).

Table 2 Comparisons of COP and PER in cooling condition between WWSHP and other air conditioning units

Air Conditioning Units	COP _i	PER _i
Air source heat pump	3.2	1.028
Electric boiler + water chilling unit	3.9	1.252
Direct-fire LiBr absorption refrigeration unit	1.23	1.132
WWSHP	4.1	1.317

As shown in Table 2, through comparisons of COP_i and PER_i between WWSHP and other air conditioning units, the WWSHP system's PER_i is highest and it reaches to 1.317. As indicated in Table 2, WWSHP is the advanced energy-saving cooling equipment.

3.3 PER in Annual Operating Condition

In cold areas, for example, in middle-and-lower reaches of Yangtze River in China, air conditioning system will run throughout the year as a result of without district central heating in winter. PER in annual operating condition (PER_o) is defined as equation (3).

$$PER_o = \frac{Q_r + Q_l}{\frac{Q_r}{PER_r} + \frac{Q_l}{PER_l}} = \frac{1 + \frac{Q_l}{Q_r}}{\frac{1}{PER_r} + \frac{Q_l}{Q_r} \cdot \frac{1}{PER_l}} \quad (3)$$

As shown in equation (3), besides the PER_r and PER_i, PER in annual operating condition is also related to the ratio of refrigeration capacity and heating capacity. A building located in Changsha city in China, which has 10,000 m² for central air-conditioning, is adopted as an example case. The air-conditioning heat

index is 90 W/m². And air-conditioning cold index is 70 W/m². The heating period is counted by 60 days, while the cooling period is counted by 120 days. Therefore, based on Table 1 and Table 2, PER_o data are derived by equation (3) and shown in Table 3. As indicated in Table 3, the WWSHP system's PER_o is highest and it reaches to 1.386. Among the four kinds of air conditioning units, when the loads are same respectively, the WWSHP has the most significant energy-saving effect. It is the advanced energy-saving cooling and heating equipment.

Table 3 Comparisons of PER in annual operating condition between WWSHP and other air conditioning units

Air Conditioning Units	PER _o
Air source heat pump	1.064
Electric boiler + water chilling unit	0.569
Direct-fire LiBr absorption refrigeration unit	1.005
WWSHP	1.386

4. ECONOMIC ANALYSIS

After comparing with the coal fired boiler + water chilling unit, district central heating + water chilling unit, and direct-fire lithium bromide absorption refrigeration unit, the WWSHP system's first cost is highest, which is 262 RMB Yuan/m², while that of direct-fire LiBr absorption refrigeration unit is lowest. But the operating cost of the direct-fire unit is highest, which is 104.2 RMB Yuan/m². The WWSHP system's operating cost is lowest, which is 47.1 RMB Yuan/m². As shown as Table 4.

Table 4 Comparisons of first cost and operating cost for different air conditioning units

System Type	First Cost (RMB Yuan/m ²)	Operating Cost (RMB Yuan/m ²)		
		Heating	Cooling	Annual Operating
coal fired boiler + water chilling unit	258.5	33.0	23.5	56.5
district central heating + water chilling unit	248.5	44.0	24.5	68.5
direct-fire LiBr absorption refrigeration unit	236	73.6	30.6	104.2
WWSHP	262	27.5	19.6	47.1

5. HEAT TRANSFER PERFORMANCE

In a cross-flow horizontal falling-film heat exchanger, the staggered coil is wrapped with a stainless-steel wire-mesh screen of five, ten, fifteen or twenty meshes/inch to hold up more fluid film. The partial enlarged three-dimensional cutaway view of the coil with enhanced heat and mass transfer surface is shown in Figure 2. The performance of the heat and mass transfer is improved due to an increase in the film uniformity [11]. Consequently, the principal objective

of this study is to evaluate the performance of the refrigeration system with high falling-film performance.

When the inlet velocity of the drip tube in the evaporator is 2.2 m/s, the temperature field for the coil with smooth surface is shown in Figure 3(a). And the temperature field for the coil with enhanced heat and mass transfer surface, which is the surface wrapped with the metal wire mesh, is shown in Figure 3(b). It is shown that the fluid disturbance intensity in the flow field of enhanced heat and mass transfer tube bundle is strengthened and higher than that of the smooth tube. Thus the kinetic energy of the liquid film increases. And the wetting ratio of the heat- or mass-transfer surface also increases due to the mesh-screen surface tension. Therefore, this is benefit to the evaporation of the refrigerant vapor. Consequently, the design of the wire mesh surface contributes to the enhancement of heat and mass transfer.

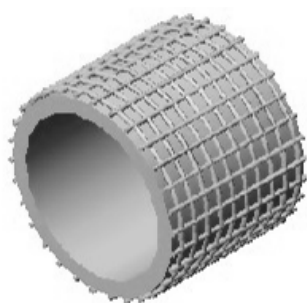
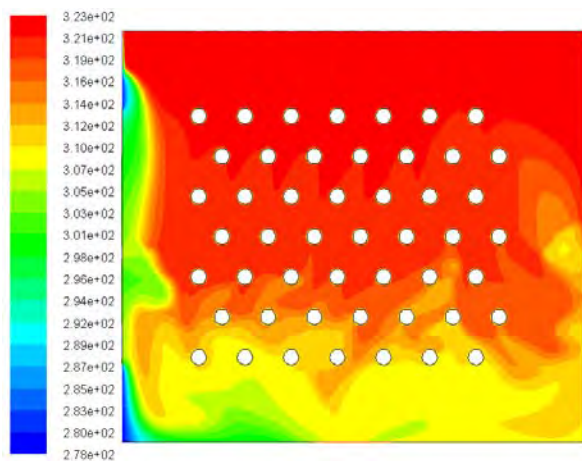
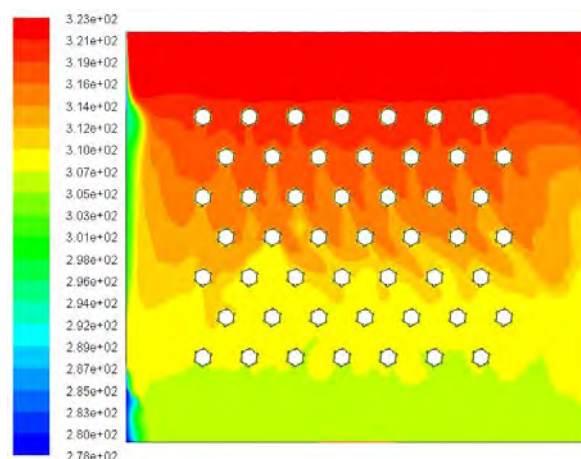


Figure 2 The partial enlarged three-dimensional cutaway view of the coil with enhanced heat and mass transfer surface



(a) The coil with smooth surface.



(b) The coil with enhanced heat and mass transfer surface.

Figure 3 Diagram of the temperature field

6. CONCLUSION

The potential of integrating a heat pump directly into the heat recovery from wastewater was studied in this paper. The wastewater source heat pump (WWSHP) system can recover the heat of wastewater and has the most significant energy-saving efficiency. Its operating cost is lower than other conventional air-conditioning units. It is a new water use and treatment strategy to tackle the water problems of China and to realize the sustainable development. And improvement of controlling parameters influencing the evaporation and condensation performance could lead to an increase in the refrigerating quantity and the COP of the WWSHP system. Consequently, this WWSHP system has its theoretical significance and promising engineering application potentials for the residential refrigerating and air-conditioning system using low-temperature waste heat source.

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NUMERICAL AND EXPERIMENTAL ANALYSIS OF OPTIMAL MATCH OF CAPILLARY TUBE LENGTH AND REFRIGERANT CHARGE OF SMALL REFRIGERATING CAPACITY AIR CONDITIONER

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ABSTRACT

Small refrigerating capacity air conditioner with dry condition is the air conditioner whose cooling capacity is 400w and evaporating temperature is 11 °C. Different matching between capillary tube length and the refrigerant charge may have a great influence on the performance of the air conditioner. The best match of both can make the optimal performance of the air conditioner. Because this kind of match is often obtained through the system operating repeatedly, the process is blindness and the results are not accurate. Therefore, the aim of this report is to set up the distribution parameter model of capillary tube to analyze the best matching point of an air conditioner. Furthermore, numerical calculation methods and experimental research are utilized in the analysis of the capillary tube's length. The test was completed in the enthalpy potential laboratory. The results find out that the optimal capillary length is 200mm, the optimal refrigerant charge is 80g and the maximum COP of air conditioning system is 2.7. According to the theoretical and experimental analysis in this article for matching of capillary length and refrigerant charge, it can benefit the design of small refrigerating capacity air conditioner with dry condition.

Keywords: capillary tube length; refrigerant charge; the optimum match.

1. INTRODUCTION

Nowadays, the precision air conditioner is always applied to the control of the temperature and humidity in a computer equipment room. Traditional computer room air conditioners are used to cool the whole computer room to meet the requirements of battery temperature, but area of the room is large and requirements of the equipment for temperature and humidity are different. If we ensure the whole room the same temperature and humidity, it certainly will cause energy waste. Small refrigerating capacity air conditioner with dry condition is applied to the battery cabinet, making the temperature and humidity control separately. The design temperature varies from 20 to 25 °C in general. Other electric equipment inside the room can work in the environment whose temperature varies from 25 to 35, so the design temperature of the precision air conditioner can be improved to reduce energy consumption. This air conditioner is a small refrigeration device installed in a storage battery incase with simple structure and high evaporating temperature. As the supply air temperature is above the dew point temperature, it can keep the surrounding of storage

battery dry and cool, also saving energy at the same time.

Small refrigerating capacity air conditioner uses the capillary tube with simple structure and low cost as the throttling equipment. The capillary length and the refrigerant charge have a significant influence on the performance of air conditioner, so finding out the optimal length of the capillary and the optimal refrigerant charge becomes a key problem. Many scholars such as Guo Xianmin^[1], Long Jianyou^[2], Shao Shuangquan^[3], Dai Guomin^[4] made theoretical analysis and experimental studies of air conditioning units whose cooling capacity were above 1000W and evaporating temperatures were 5-7 °C. Researches found that it existed the best matching between the capillary length and the refrigerant charge. Dai Guomin^[2] and Long Jianyou^[4] only found out the best matching point by test rather than theoretical calculation. However Shao Shuangquan^[3] just used the straight capillary of the simulation calculation model rather than the spiral capillary calculation model. As a result, the charge and spiral capillary calculation remain to be further studied.

NOMENCLATURE

Symbols		Subscripts	
p	Pressure (pa)	f	Liquid phase
T	Temperature (°C)	g	Gas phase
u	Velocity (m/s)	i	Capillary inlet condition
h	Specific enthalpy (J/kg)	i+1	Capillary outlet condition
s	Specific entropy (J/kg·k)	tp	Two-phase
x	Quality	m	Mean
k	Entrance loss coefficient	ck	Choked flow
f	Friction factor	sc	Sub-cooling
d	Capillary internal diameter (m)	Greek Letters	
D	Coil diameter (m)	ρ	Density (kg/m ³)
L	Capillary length (m)	μ	dynamic viscosity (pa·s)
G	Mass flow rate per unit area (kg/s·m ²)	ν	specific volume (m ³ /kg)

The object of this research is a small refrigeration unit using a spiral capillary tube whose cooling capacity is only 400 W and evaporation temperature is 11°C. As the air conditioning refrigerating capacity is very small and the evaporating temperature is higher than traditional air conditioners, little research is carried out on small refrigerating capacity air conditioner with dry condition in China. Therefore, in this paper, we use a numerical method to calculate the spiral capillary length and then the optimal capillary length and refrigerant charge are gained by test. This test verifies the algorithm correctness. This paper provides a reliable experimental foundation for the design of small refrigerating capacity air conditioner with dry condition.

2. NUMERICAL CALCULATION FOR THE ADIABATIC CAPILLARY LENGTH

2.1 Mathematical model

The whole process of Refrigerant flow in capillary tube can be divided into three zones: period of single-phase zone, metastable two-phase zone and thermal equilibrium two-phase zone. Metastable flow phenomenon is a kind of unstable phenomenon, there is no common accepted conclusion. Metastable flow does not affect the universality of the model, so the metastable flow is ignored. This paper uses the distributed parameter model, and assumes that the refrigerant flow is one-dimensional adiabatic flow and the influence of the gravity on the refrigerant flow is ignored. At the same time, it assumes that two phase flow in a capillary is homogeneous flow, and any flows of interface attains to the thermodynamic equilibrium state. On this basis, the control differential equation of the fluid is as follows:

Mass conservation equation:

$$\frac{d(\rho u)}{dx} = 0 \quad (1)$$

Momentum conservation equation:

$$\frac{d(\rho u^2)}{dx} + \frac{dp}{dx} = -\frac{f}{d} \cdot \frac{1}{2} \rho u \quad (2)$$

Energy conservation equation:

$$\rho u \frac{dh}{dx} + \frac{\rho u d \left(\frac{1}{2} u^2 \right)}{dx} = \quad (3)$$

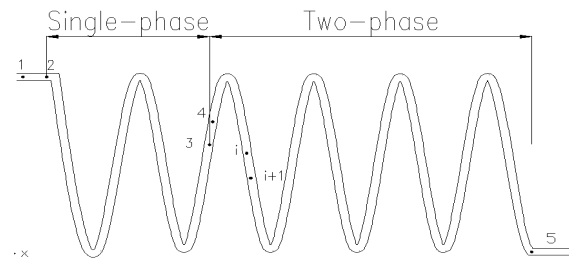


Figure 1 Schematic diagram of an adiabatic helical capillary tube

The equations can be discretized as follows:

$$\rho_i u_i = \rho_{i+1} u_{i+1} \quad (4)$$

$$\rho_i u_i (u_{i+1} - u_i) + p_{i+1} - p_i = -4 f_m \Delta L \rho_i u_i (u_i + u_{i+1}) / D_i \quad (5)$$

$$\rho_i u_i (h_{i+1} - h_i) + \rho_i u_i (u_{i+1}^2 - u_i^2) / 2 = 0 \quad (6)$$

$$f_m = \frac{f_i + f_{i+1}}{2} \quad (7)$$

$$u_m = \frac{u_i + u_{i+1}}{2} \quad (8)$$

The discussions of different capillary calculating zones are these:

(1) Entrance to the sudden shrinkage

$$p_1 - p_2 = (1 + k) * \rho u^2 / 2 \quad (9)$$

For the sharp edge entrance, k=0.5.

(2) Single-phase flow region

$$L_{sc} = \frac{2d(p_2 - p_3)}{G^2 f_{sc} \nu_{sc}} \quad (10)$$

The friction coefficient of single-phase region uses Ito 1959 equation:

$$f = (d/D)^{0.5} \{0.0725 + 0.076[\text{Re}(d/D)^2]^{-0.25}\} \quad (11)$$

$$\text{Re} = \frac{ud}{\mu} \quad (12)$$

(3) two-phase region

$$h_{tp} = h_f + x(h_g - h_f) \quad (13)$$

$$v_{tp} = v_f + x(v_g - v_f) \quad (14)$$

$$h_{f4} + x_4(h_{g4} - h_{f4}) + \frac{G^2}{2}[v_{f4} + (v_{g4} - v_{f4})x_4]^2 = h_3 + \frac{G^2 v_3}{2} \quad (15)$$

$$ax_4^2 + bx_4 + c = 0 \quad (16)$$

where

$$a = \frac{1}{2} G^2 (v_{g4} - v_{f4})^2$$

$$b = G^2 v_{f4} (v_{g4} - v_{f4}) + h_{g4} - h_{f4}$$

$$c = \frac{1}{2} G^2 (v_{f4}^2 - v_3^2) + h_{f4} - h_3$$

The dry degree is between 0 and 1.

$$x_4 = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \quad (17)$$

$$\Delta L_{tp} = \frac{2d}{G^2 v_{tp} f_{tp}} [p_1 - p_2 - G^2 (v_2 - v_1)] \quad (18)$$

By summing all the control volumes of two-phase region, we can get the whole length.

$$L_{tp} = \sum \Delta L_{tp} \quad (19)$$

The friction coefficient of two phase region also uses Ito 1959 equation, but Dukler equation is adopted to calculate the viscosity:

$$\mu = \frac{(1-x_g)v_f\mu_f + x_g v_g \mu_g}{(1-x_g)v_f + x_g v_g} \quad (20)$$

(4) Choked flow

As the refrigerant pressure drops, the flow rate increases. When the velocity of the refrigerant reach the local velocity of sound, the flow rate is no longer increasing and this flow is choked flow.

Choked flow equation:

$$G_{ch} = \left[\frac{x}{G_{cg}^2} + \frac{1-x}{G_{cf}^2} \right]^{-\frac{1}{2}} \quad (21)$$

$$\frac{1}{G_{cg}^2} = \frac{v_g - v_f}{s_g - s_f} \frac{ds_g}{dp} - \frac{dv_g}{dp} \quad (22)$$

$$\frac{1}{G_{cf}^2} = \frac{v_g - v_f}{s_g - s_f} \frac{ds_f}{dp} - \frac{dv_f}{dp} \quad (23)$$

2.2 Algorithm design and Solution method

For single-phase region, due to the constant temperature, specific volume, and specific enthalpy, according to formulation(9), we can directly calculate single-phase region length. In the two-phase region, 2000 control volumes are divided, and the grid number above 2000 has little influence on the results of the numerical calculation. For each infinitesimal section, export dryness can be obtained by formulation(17). In turn, the outlet specific volume, enthalpy, viscosity and coefficient of friction can also be obtained. By formulation(18) infinitesimal length is obtained. Capillary infinitesimal length is calculated by iteration. An iterative termination condition for capillary is that outlet pressure equals to evaporation pressure or refrigerant flow reaches choked flow.

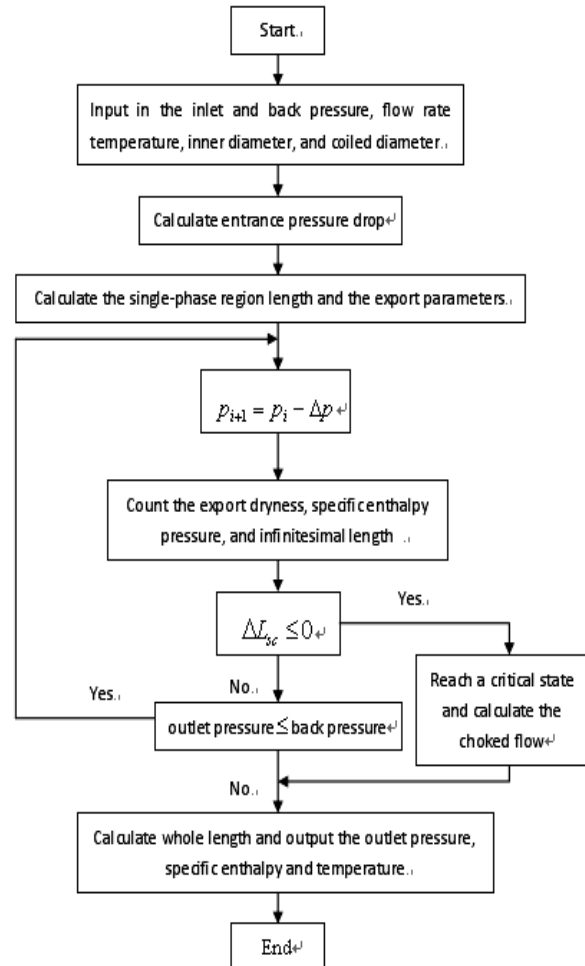


Figure 2 Capillary tube model flow chart

3. EXPERIMENT

3.1 Test unit and the design parameters

The size of air conditioning unit is 600 mm *300 mm. The inner structure of it is shown in Figure3.1. This air conditioner refrigerating capacity is 400W and evaporating temperature is 11℃, so the process of dealing with indoor air is sensible cooling. The supply of indoor air is completed by little enthalpy difference and large air volumes, so the supply air temperature is above the dew point temperature, and it can satisfy the requirements of battery capboard with no water.

Table 1 Design Parameters

Design parameters	value
Indoor dry bulb temperature (℃)	27
Indoor wet bulb temperature (℃)	19
Outdoor dry bulb temperature (℃)	35
Outdoor wet bulb temperature (℃)	24
Evaporating temperature (℃)	11
refrigerating capacity (W)	400
The compressor exhaust (ml/rev)	418
Capillary inner diameter (mm)	2.0
Type of refrigerating fluid	R134a

3.2 The measured parameters

There are seven measuring points, and parameters and position of each measuring point are shown in table 2. Test instruments are thermocouple, pressure gauge, the scale, and vacuum pump.

Table 2 measured Parameters

Measuring point	position	measured parameters
1	inlet of compressor	temperature, pressure
2	outlet of compressor	temperature, pressure
3	outlet of capillary	temperature
4	inlet of capillary	temperature
5	outlet of blower	temperature
6	return air grille	temperature
7	outlet of fan	temperature

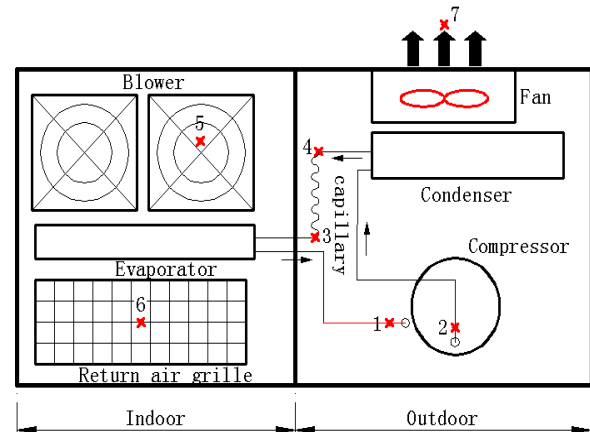


Figure 3 the distribution of test point of small refrigerating capacity air conditioner

3.3 Discussions of the experimental results

3.3.1. Evaporating temperature

The experiment shows that when the refrigerant charge is certain evaporating temperature rises with the increase of capillary length. When the length is certain, evaporating temperature rises with the increase of refrigerant charge. This result fits to the result of Shao Shuangquan^[3]. When the refrigerant charge is 70 g, and capillary length is 400 mm, the evaporating temperature is 5 ℃. It appears frost phenomenon in the outlet of capillary tube, the reason is that the capillary length is too long, which causes that throttling pressure drop is large, and the capillary outlet pressure is low. Because saturated temperature corresponds to saturation pressure, evaporating pressure is low.

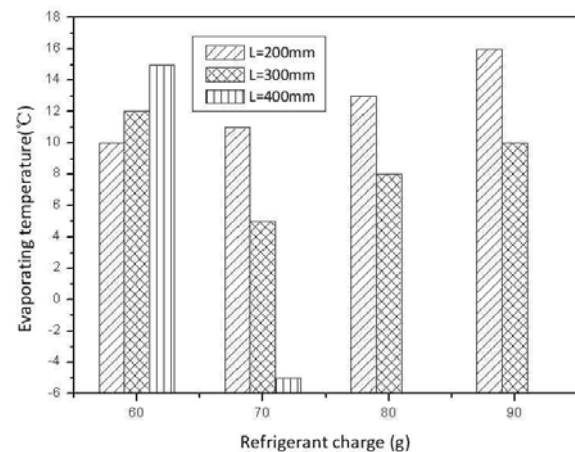


Figure 4 the effect of capillary length on evaporating temperature

3.3.2. Condensing temperature

Experimental data shows that the condensing temperature declines with the increase of the capillary length slightly, and increases with the increase of the refrigerant charge. There is no doubt that the refrigerant charge shouldn't be too much. Although the refrigerant charge increases can improve the evaporating

temperature, it also improves the condensing temperature.

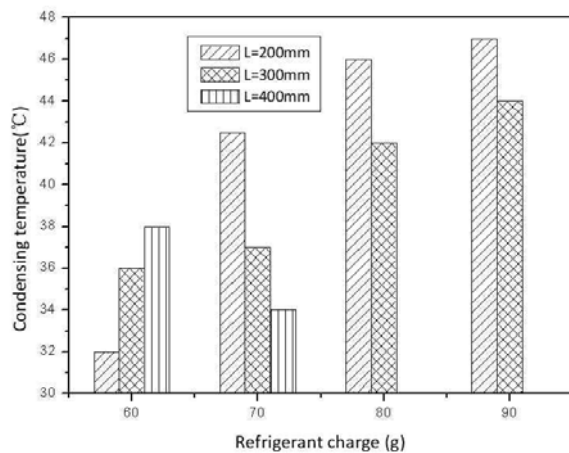


Figure 5 the effect of capillary length on condensing temperature

3.3.3. Refrigerating capacity

The test results show that the refrigerating capacity increases with the increase of the capillary length, and decrease after the peak. It indicates that it exists a optimal length to maximize the refrigerating capacity. When capillary length is constant, with the refrigerant charge increasing, the flow rate increases, and the refrigerating capacity increase. But when the refrigerant charge is over the peak, it appears liquid seal in the capillary tube and parts of liquid refrigerant leaves in condenser. Actual flow of refrigerant into the evaporator is reduced, the refrigerating capacity declines.

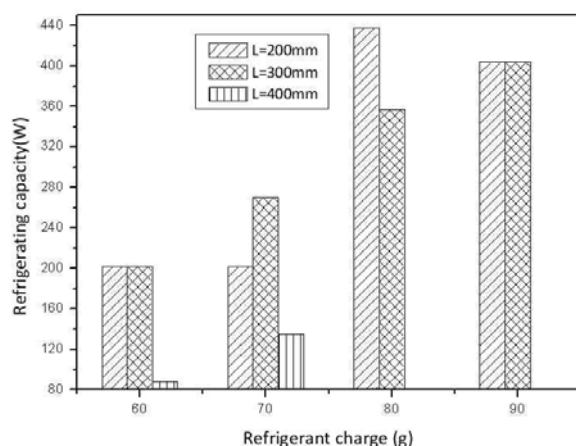


Figure 6 the effect of capillary length on refrigerating capacity

3.3.4. COP of air conditioner

The trend of COP value varying with the change of capillary length and filling quantity is agreed with the trend of refrigerating capacity. The main course is that the compressor power consumption changes little than the refrigerating capacity.

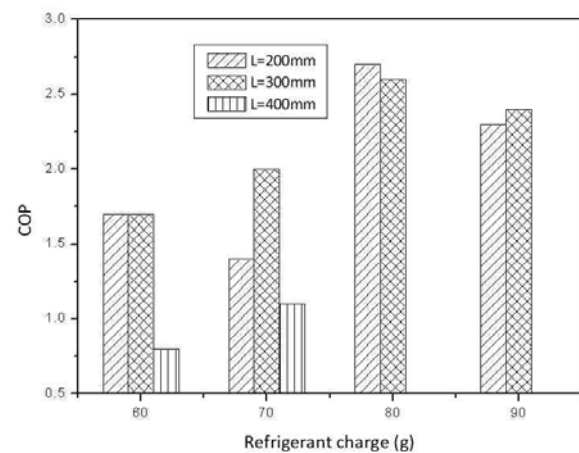


Figure 7 the effect of capillary length on refrigerating capacity

4. CONCLUSIONS

We can conclude the following conclusions:

1. By the test, it is confirmed that the air conditioner performance is affected by capillary length and the refrigerant charge. When the refrigerant charge is not enough, the COP of air conditioner is very low. When the refrigerant charge is more than 90 g, the COP is also reduced.

2. The optimal capillary length under different refrigerant charge is different. It indeed exists the best matching value of both. The best matching value is (80 g, 200 mm), and the COP value is 2.7.

3. This paper analyses the influence of capillary length and the refrigerant charge on the evaporating temperature, condensing temperature, refrigerating capacity and COP value. The results reflected by figure 4 to figure 7 fit to the research results of Guo Xianmin^[1] and Shao Shuangquan^[3]. These indicators reflect the variation of air conditioning performance indirectly. This paper provides reference for design of air conditioner products, and has a certain guiding significance.

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EFFECT OF CuSO_4 AND H_2O_2 ON CONTROLLING ALGAL BIOMASS IN TWO LANDSCAPE LAKES

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ABSTRACT

The urban landscape water is an important part of the modern city, which are mostly closed slowly flowing water with poor mobility, low self-purification capacity and easy to pollution. Closed slow flow lead to water nutrients surplus, when temperature and light are in suitable conditions, blooms of algal cause imbalance in the aquatic environment, while algae mass extinction will cause water anaerobic, even black and stink, which seriously affect the landscape water in sightseeing, leisure, entertainment effect.

The copper sulfate is widely used as algicide, but hydrogen peroxide as algicide in domestic research is much less, the combined addition of H_2O_2 and Cu^{2+} has not yet been reported. In this study, the effect of hydrogen peroxide (H_2O_2) and copper sulfate (Cu^{2+}) on controlling algal biomass of the water samples taken from two landscape lakes were investigated in Shenzhen. For lake A, with Chlorophyll *a* (Chl *a*) concentration of 10 $\mu\text{g/L}$ algal biomass, which is on behalf of the algal bloom alert level. For lake B, with Chl *a* concentration of 100 $\mu\text{g/L}$ algal biomass, which represents the serious level of algal blooms.

Sampling barrel was used to get a certain amount of surface water (below the surface 25 cm or less) from both Lake A and B, respectively in 500 mL conical flask with 300 mL water. Then adding hydrogen peroxide (H_2O_2) and copper sulfate (Cu^{2+}) according to certain concentration gradient. After adding algicide, the samples tested were trained in light incubator, with a 12 h:12 h Light-dark proportion and a constant temperature of 25°C. Low algae biomass group water samples tested were trained for 13 days while high algae biomass group water samples tested were trained for 9 days, both regularly sampling work were done during the training period. The Chl *a* concentration was measured in the modulation fluorescence analysis to stand for algae biomass by measurement analysis instrument PHYTO-PAM phytoplankton classification fluorescence spectrometer (WALZ company, German). For the lake A, the algal biomass were suppressed to less than 5 $\mu\text{g/L}$ on the 13th day after adding 0.2-2 mg/L Cu^{2+} and the suppression effects were reinforced by increasing Cu^{2+} concentration within 0.2-0.6 mg/L. Adding 20 mg/L H_2O_2 suppressed the algal biomass to 17 $\mu\text{g/L}$ within 13 days. For the lake B, the algal biomass were suppressed to 12-41 $\mu\text{g/L}$ on the 9th day after adding 0.4-2 mg/L Cu^{2+} and the suppression effects were reinforced by increasing Cu^{2+} concentration within 0-0.8 mg/L. Adding H_2O_2 of 20-40 mg/L induced suppression effect on the algal biomass that were 30% and 6% of the control level within 9 days. The combined addition of H_2O_2 (0-40 mg/L) and Cu^{2+} (0-2 mg/L) significantly suppressed algal biomass to less than 14 $\mu\text{g/L}$ on the 9th day after chemicals addition. Combined addition of 10 mg/L H_2O_2 and 2 mg/L Cu^{2+} induced a reduction of Chl *a* concentration to 53 $\mu\text{g/L}$ on the 1th day, much lower than the samples only exposed to 2 mg/L Cu^{2+} with Chl *a* concentration of 88 $\mu\text{g/L}$, suggesting that adding H_2O_2 marked enhanced the suppression effect of Cu^{2+} on algal biomass and may reduce the Cu^{2+} dose required.

Key words: Landscape water; Algal biomass; Hydrogen peroxide; Copper sulfate.

1. INTRODUCTION

Closely related to people's life, the urban landscape water is an important part of the modern city, which are mostly closed slowly flowing water, with poor mobility, low self-purification capacity, easy to pollution^[1]. Closed slow flow lead to water nutrients surplus, when temperature and light are in suitable conditions, blooms of algal cause imbalance in the aquatic environment, while algae mass extinction will cause water anaerobic,

even black and stink^[2,3], which seriously affect the landscape water in sightseeing, leisure, entertainment effects. In recent years, the governance of the city eutrophication and landscape water bloom has been reported^[4], but it is still using sewage treatment ideas, has not yet formed a perfect technology system.

At present, the copper sulfate is widely used as algicide in domestic, but hydrogen peroxide as algicide in domestic research is much less, the hydrogen peroxide and the copper sulfate combination kill algae

has not yet been reported. In this study, two landscape lakes in Shenzhen were taken for example. For lake A, with Chl *a* concentration of 10 µg/L algal biomass, which is on behalf of the algal bloom alert level^[5]. For lake B, with Chl *a* concentration of 100 µg·L⁻¹ algal biomass, which represents the serious level of algal blooms. In this study, the effects of hydrogen peroxide (H₂O₂) and copper sulfate (Cu²⁺) on controlling algal biomass were investigated.

2. MATERIALS AND METHODS

2.1 The raw water

Low algae biomass group test samples were taken from landscape lake A, whose area is about 6500 m², with the mean depth of about 2.3 m, water capacity of approximately 15000 m³ and an initial experimental Chl *a* concentration of 10 µg/L. The group of high algae biomass samples tested were taken from landscape lake B, whose area is about 5000 m², with the depth of about 2.0 m and about 10000 m³ water, while with an initial Chl *a* concentration of 100 µg/L. Sampling barrel was used to get a certain amount of surface water (below the surface 25 cm or less) from both Lake A and B, respectively in 500 mL conical flask with a sample water volume 300 mL. Before the start of the experiment, the group of low algae biomass samples' initial Chl *a* concentration was 7.8 - 8.2 µg/L, pH value was 8.0 - 8.5, while the group of high algae biomass samples' initial Chl *a* concentration of 88 - 95 µg/L, pH value was 9.6 - 9.9.

2.2 Additive agent

Copper sulfate (CuSO₄·5H₂O, analytically pure, Tianjin fuyu fine chemical co., LTD.), hydrogen peroxide (H₂O₂, analytically pure, mass fraction is 30%, Tianjin damao chemical reagent factory) were involved as experimental algicide, those two algicides concentrations are shown in table 1. Keep shaking when adding algicide.

Table 1 Algicide concentration

The Group of Low Algae Biomass(mg/L)		The Group of High Algae Biomass(mg/L)				
Cu ²⁺	H ₂ O ₂	H ₂ O ₂	+	Cu ²⁺	Cu ²⁺	H ₂ O ₂
0	0	0	+	0	0	0
0.04	0.5	20	+	0.4	0.4	5
0.2	1	40	+	0.8	0.8	10
0.4	2.5	10	+	2	2	20
0.6	5					40
1	10					
2	20					

2.3 Sampling and analysis method

After adding chemicals, the samples tested were trained in light incubator, with a 12 h:12 h Light-dark proportion and a constant temperature of 25 °C. The group of low algae biomass water samples tested were

trained for 13 days while The group of high algae biomass water samples tested were trained for 9 days, both regularly sampling work were done during the training period. According to the standard experimental method, samples of 3 mL were picked up for algae analysis. The Chl *a* concentration was measured in the modulation fluorescence analysis to stand for algae biomass by measurement analysis instrument PHYTO-PAM phytoplankton classification fluorescence spectrometer (WALZ company, German) in this study. The growth rate percent inhibition (*Ir*) also be introduced to evaluate the effects of algicides for the algae growth inhibition^[6], all the computation formulas are as follows:

$$Ir(\%) = \frac{\mu_c - \mu_T}{\mu_c} \times 100$$

Among them, the *Ir* is for growth rate percent inhibition as well as μ_c for average growth rate of the control group (d⁻¹) and μ_T for average growth rate of the agent treatment group (d⁻¹). When *Ir* value was greater than zero, the algal cell growth slew down. While *Ir* value was 100% the algal cell growth had been completely inhibited. *Ir* value more than 100% shown that cell density was less than the initial value and *Ir* value less than zero shown that algae biomass of the treatment group was higher than those in the control group, algal cell growth was accelerated. Values represent the mean of 3 replicates and drawing with the Origin of 8.0 software (Originlab corporation, the United States).

3. THE EXPERIMENTAL RESULTS

3.1 The group of low algae biomass

3.1.1 The algae suppression effects of adding copper sulfate to the group of low algae biomass

Adding copper sulfate (Cu²⁺) to the group of low algae biomass water samples according to certain concentration gradient, the influence of the algal biomass is shown in figure 1-a. During the period of training, the control group was in normal growth, with Chl *a* concentration rose from 8 µg/L to 52 µg/L. The trend was consistent with the control group and the treatment group which add 0.04 mg/L Cu²⁺, but the Chl *a* concentration was significantly lower than the control group. Chl *a* concentration was 43 µg/L on the 13th day, took 83% of the control group. Chl *a* concentration significantly reduced to 1-4 µg/L on the 1th day after adding 0.2, 0.4, 0.6, 1 and 2 mg/L Cu²⁺, then keep stable, Chl *a* concentration reached to 5.4, 2.1, 1.2, 0.4 and 0.2 µg/L on the 13th day, which were taken the control group 10%, 4%, 2%, 0.8% and 0.4%. The results showed that Chl *a* concentration less than 5 µg/L level on the 13th day after adding 0.2-2 mg/L Cu²⁺. The *Ir* values with copper sulfate doses of change are shown in figure 1-b. *Ir* increased significantly by increasing Cu²⁺ concentration within 0-0.2 mg/L. It was shown that the suppression effects were reinforced by increasing Cu²⁺ concentration. Taking samples on the

5th day, *Ir* value rose from 25% to 124% by increasing Cu^{2+} concentration within 0.04-0.2 mg/L. *Ir* rose slowly by increasing Cu^{2+} concentration within 0.2-0.6 mg/L, indicating that further improve Cu^{2+}

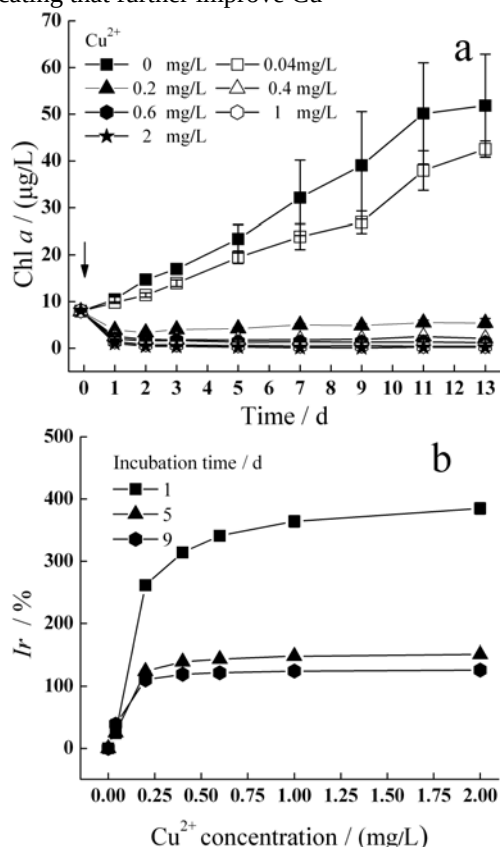


Figure 1 The algal suppression effects of adding copper sulfate to the group of low algae biomass
(The arrows indicate the adding time)

concentration helps to reinforce suppression effects. Taking samples again on the 5th day, *Ir* value increased from 124% to 139%-143% by increasing Cu^{2+} concentration from 0.2 mg/L to 0.4-0.6 mg/L. *Ir* value is stable without significant difference by increasing Cu^{2+} concentration within 0.6-2 mg/L, which is indicated that algal suppression effects tend to be stable in Cu^{2+} concentration more than 0.6 $\text{mg}\cdot\text{L}^{-1}$, even improving Cu^{2+} concentration largely, and it also can not achieve a better controlling effect of algae. It is worth noting that the *Ir* value all reached the maximum with different Cu^{2+} concentration on the first day, presumably Cu^{2+} get into the algal cells quickly and produces toxic, algal suppression process characterized by acute toxicity effect.

3.1.2 The algae suppression effects of adding H₂O₂ to low algae biomass group

Adding H_2O_2 to the group of low algae biomass water samples according to certain concentration gradient, the influence of the algal biomass is shown in figure 2-a. During the period of training in 13 days, the control group was in normal growth, with Chl *a* concentration rose from 8 μg/L to 99 μg/L. The 0.5, 1, 2.5, 5 mg/L H_2O_2 treatment group kept a rising trend in

Chl *a* concentration, respectively reached 93, 64, 74 and 105 μg/L on the 13th day, which took 94%, 65%, 75% and 106% of the control group. The 10 μg/L treatment group's Chl *a* concentration moved down slowly at the first 2 days, then rose quickly to 112 μg/L on the 13th day, took 113% of the control group. The 20 mg/L treatment group's Chl *a* concentration gradually reduced at the first 4 days, then increased slowly, then reached to

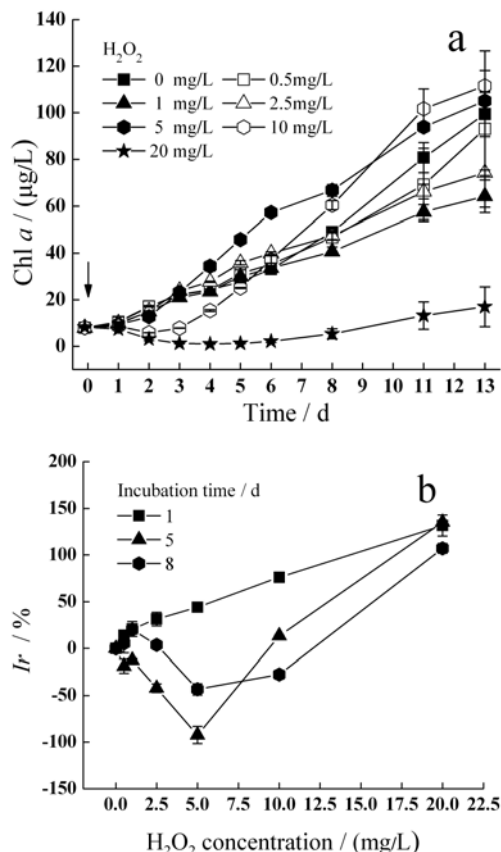


Figure 2 The algal suppression effects of adding H₂O₂ to the group of low algae biomass
(The arrows indicate the adding time)

17 μg/L on the 13th day, took 17% of the control group. It was shown that adding 20 $\text{mg}\cdot\text{L}^{-1}$ H_2O_2 in the water alone would make Chl *a* concentration less than 17 μg/L level in 13 days. The *Ir* values with H_2O_2 doses of change are shown in figure 2-b. In the process of training, in addition to *Ir* value increased significantly by increasing H_2O_2 concentration on the 1th day, *Ir* value reduced gradually by increasing H_2O_2 concentration within 0.5-5 mg/L, while *Ir* value rose gradually by increasing H_2O_2 concentration within 5-15 mg/L, and the *Ir* value was negative by increasing H_2O_2 concentration within 0.5-10 mg/L.

3.2 The group of high algae biomass

3.2.1 The algae suppression effects of adding copper sulfate to the group of high algae biomass

Adding copper sulfate (Cu^{2+}) to the group of high algae biomass water samples according to certain concentration gradient, the influence of the algal

biomass is shown in figure 3-a. During the period of training in 9 days, Chl *a* concentration of the control group rose to 117 $\mu\text{g/L}$ gradually at the first 3 days, then stayed stable in 107-117 $\mu\text{g/L}$. The trend of the 0.4 mg/L Cu^{2+} treatment group was consistent with the control group at the first 3 days, thereafter reduced significantly and Chl *a* concentration reached 41 $\mu\text{g/L}$ on the 9th day, took 38% of the control group. The trend of the 0.8 mg/L

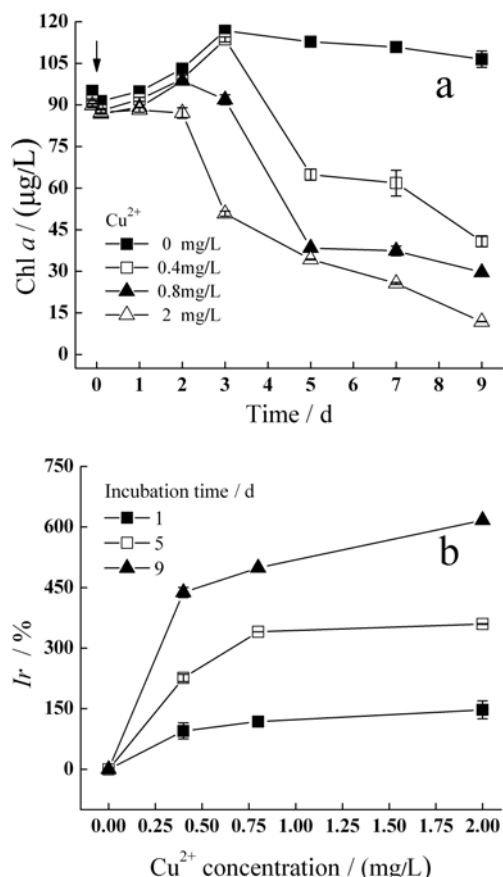


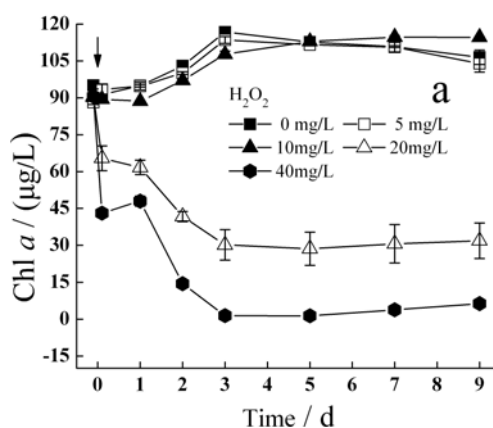
Figure 3 The algae suppression effects of adding copper sulfate to the group of high algae biomass
(The arrows indicate the adding time)

treatment group was consistent with the control group at the first 2 days, thereafter reduced significantly and down to 30 $\mu\text{g/L}$ on the 9th day, took 28% of the control group. The 2 mg/L treatment group kept stable at 88 $\mu\text{g/L}$ at the first 2 days, then reduced significantly and reached 12 $\mu\text{g/L}$ on the 9th day, took 11% of the control group. It was shown that the additive alone 0.4-2 mg/L Cu^{2+} in the water would make Chl *a* concentration at 12-41 $\mu\text{g/L}$ level in 9 days. The *Ir* values with copper sulfate doses of change are shown in figure 3-b. The *Ir* value lift quickly by increasing Cu^{2+} concentration within 0-0.8 mg/L, which showed that the suppression effects were reinforced significantly by increasing Cu^{2+} concentration. The *Ir* value tend to be stable by increasing Cu^{2+} concentration within 0.8-2 mg/L, which showed that algal suppression effects tend to be stable in $\text{Cu}^{2+} > 0.8$ mg/L, even improving Cu^{2+} concentration largely, it also can not achieve better

suppression effects of algae. With Cu^{2+} exposure time grown, the *Ir* value rose significantly by increasing Cu^{2+} concentration, which was speculated that Cu^{2+} toxicity effects were enhanced by Cu^{2+} accumulating within the algal cells while exposure time prolonged, algal suppression process characterized by cumulative toxicity effect.

3.2.2 The algae suppression effects of adding H₂O₂ to the group of high algae biomass

Adding H_2O_2 to the group of high algae biomass water samples according to certain concentration gradient, the influence of the algal biomass is shown in figure 4-a. During the period of training in 9 days, Chl *a* concentration in the control group rose gradually at the first 3 days, then moved down to 107 $\mu\text{g/L}$ slowly. The trend of the 5-10 mg/L treatment group is consistent with the control group. The Chl *a* concentration reduced significantly by adding 20 and 40 mg/L H_2O_2 at the first 3 days, then tends to be stable, thereafter reached 32 $\mu\text{g/L}$ and 6 $\mu\text{g/L}$ on the 9th day, respectively took 30% and 6% of the control group, which were shown that adding 20-40 mg/L H_2O_2 in the water alone would make Chl *a* concentration at 6-32 $\mu\text{g/L}$ level in 9 days. The *Ir* values with H_2O_2 doses of change are shown in figure 4-b. *Ir* value decreased by increasing H_2O_2 concentration within 0-5 mg/L at the first day. *Ir* value rose quickly by increasing H_2O_2 concentration within 5-20 mg/L, however *Ir* value grew slowly by increasing H_2O_2 concentration further to 40 mg/L. The *Ir* value was in the same change rule on the 5th day and 9th day, *Ir* value was slightly lower by adding 0-10 mg/L H_2O_2 , *Ir* value increased significantly by adding 10-20 mg/L H_2O_2 , which showed that the suppression effects were reinforced significantly by increasing H_2O_2 concentration, and *Ir* value tend to be stable by adding 20-40 mg/L H_2O_2 , which indicated that algal suppression effects tend to be stable by adding more than 20 mg/L H_2O_2 , even improving H_2O_2 concentration largely, it also can not achieve better suppression effects of algae.



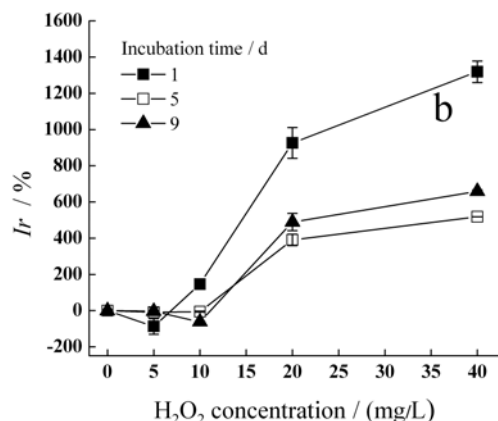


Figure 4. The algae suppression effects of adding H₂O₂ to the group of high algae biomass (The arrows indicate the adding time)

3.2.3 The algal suppression effects of combined addition of hydrogen peroxide and copper sulfate to the group of high algae biomass

The combined addition of H₂O₂ and Cu²⁺ to the group of high algae biomass, the influence of the algal biomass is shown in figure 5-a, Including combined addition group A (20 mg/L H₂O₂+0.4 mg/L Cu²⁺), group B (40 mg/L H₂O₂+ 0.8 mg/L Cu²⁺) and group C (10 mg/L H₂O₂+2 mg/L Cu²⁺). During the period of training in 9 days, Chl *a* concentration of the control group rose gradually at the beginning of 3 days, then moved down slowly to 107 µg/L. The Chl *a* concentration of group A and B reduced significantly on the 2nd day after chemicals addition, then tend to be stable, reached 14 and 0.3 µg/L on the 9th day, took 13% and 0.3% of the control group respectively. The Chl *a* concentration of group C continued to reduce to 8 µg/L on the 9th day, took 7% of the control group. It was indicated that the combined addition of H₂O₂ and Cu²⁺ in the water would make Chl *a* concentration less than 14 µg/L level on the 9th day after group A, group C is the minimum, which speculated that high concentrations of H₂O₂ makes large contribution to algal suppression effects in a short time after adding addition of chemicals is shown in figure 5-b. The *Ir* value of group B is the highest at the first day, followed by group A, group C is the minimum, which speculated that high concentrations of H₂O₂ makes large contribution to algal suppression effects in a short time after adding chemicals. The *Ir* value of group A, B and C has no significant difference on the 5th and 9th day, which speculated that the control effect of algae mainly comes from Cu²⁺ toxicological effects.

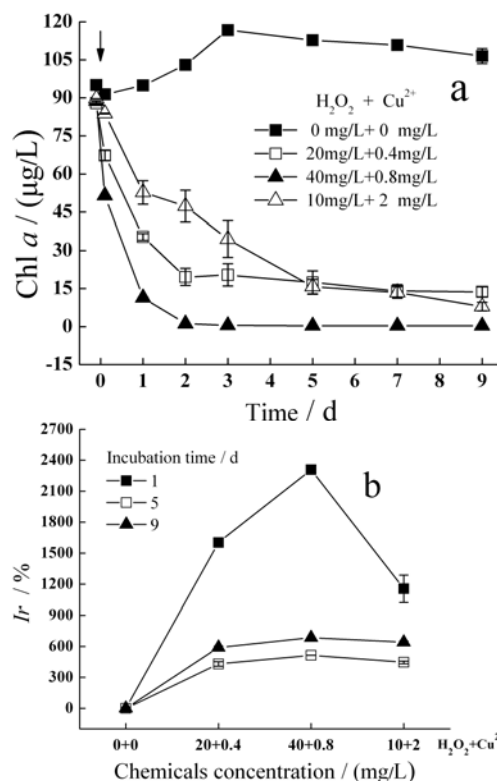


Figure 5. The algal suppression effects of combined addition of hydrogen peroxide and copper sulfate to the group of high algae biomass (The arrows indicate the adding time)

4. RESULTS AND DISCUSSIONS

It was indicated that adding 0.2-2 mg/L Cu²⁺ has achieved significant algal suppression effect for the initial Chl *a* concentration respectively about 10 µg/L and 100 µg/L in two landscape lake. The algal suppression effects rose gradually by increasing Cu²⁺ concentration less than 0.6-0.8 mg/L, while the effects tend to be stable by increasing Cu²⁺ concentration exceed 0.6-0.8 mg/L. QiJun Kuang's^[7] research showed that it has significant suppression effect to Synuraceae urelin and Dinobryon by adding 0.75 mg/L Cu²⁺. At the same time, It was indicated that algal suppression process characterized by acute toxicity effect via adding Cu²⁺ when the initial Chl *a* concentration is 10 µg/L, and it characterized by cumulative toxicity effects via adding Cu²⁺ when the initial Chl *a* concentration is 100 µg/L in the landscape lake.

It was indicated that the combined addition of hydrogen peroxide and copper sulfate can effectively improve the algal suppression effect of copper sulfate. Taking combined addition group C (10 mg/L H₂O₂ + 2 mg/L Cu²⁺) as an example, it made Chl *a* concentration drop to 53 µg/L in short time (1 d), which was significantly lower than the group by adding 2 mg/L Cu²⁺ alone (88 µg/L). Meanwhile, it is found that combined addition of hydrogen peroxide and copper sulfate may lower Cu²⁺ concentration in this study. Taking combined addition group A (20 mg/L H₂O₂ + 0.4 mg/L Cu²⁺) as an example, the algal suppression effect

on the 9th day is better than that of the group by adding 0.8 mg/L Cu²⁺ alone.

It was indicated that it has significant suppression effect only by adding H₂O₂ alone 20 mg/L above. The Chl *a* concentration of 20 mg/L treatment group is 17% of the control group on the 13th day in the experiment of low algae biomass, and the Chl *a* concentration of 20 and 40 mg/L treatment group is respectively 30% and 6% of the control group on the 9th day in the experiment of high algae biomass. It was also found that adding 5-10 mg/L H₂O₂ played a certain role in promoting algae growth. Zhengzhen Dong's^[8] research shows that 5-10 mg/L H₂O₂ play a weak role in promoting *Platymonas helgolandica* Var. growth, and it has suppression effect only by adding H₂O₂ 15 mg/L above, which is consistent with the research conclusion.

5. CONCLUSIONS

For the low algae biomass landscape lake with initial Chl *a* concentration of 10 µg/L, adding 0.2-2 mg/L Cu²⁺ in the water alone will make Chl *a* concentration less than 5 µg/L in 13 days, algal suppression process characterized by acute toxicity effect, and the algal suppression effects were reinforced gradually by increasing Cu²⁺ concentration among 0.2-0.6 mg/L. There was suppression effect by adding 20 mg/L H₂O₂ alone, but adding 5-10 mg/L H₂O₂ would lead to further raise algae biomass.

For the high algae biomass landscape lake with initial Chl *a* concentration of 100 µg/L, adding 0.4-2 mg/L Cu²⁺ alone can effectively reduce Chl *a* concentration in 9 days, and control it in the level of 12-41 µg/L, algal suppression process characterized by cumulative toxic effect, and the algal suppression effects were reinforced by increasing Cu²⁺ concentration among 0-0.8 mg/L. There was suppression effect by adding 20 mg/L H₂O₂ alone, but adding 5-10 mg/L H₂O₂ would lead to further raise algae biomass. combined addition of hydrogen peroxide and copper sulfate will make Chl *a* concentration less than 14 µg/L in 9 days, and combined addition chemicals can effectively improve the algae suppression effects of copper sulfate.

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STUDY ON MODIFIED ZEOLITE IN REMOVING PHOSPHORUS AND ITS INFLUENCE FACTORS

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ABSTRACT

The research material is selected from Zhengzhou, Henan Province, the test wastewater containing phosphorus is the water from Wastewater Treatment Plant. The effects of modified zeolite in removing phosphorus is discussed on dosage of modified zeolite, adsorption time, pH and initial concentration. The results show that when the dosage is 6g, the highest phosphorus removal rate is 83.29%; About 70 minutes after adsorption, the average of phosphorus removal rate is 85.1%; In between pH=5~6, the phosphorus removal rate is gradually rising, when the pH=6, the highest phosphorus removal rate is 88.7%, when the pH>6, the phosphorus removal rate is gradually declining. When the initial concentration is 1mg/L, the phosphorus removal rate is 84.6%, as the concentration is increasing, the phosphorus removal rate is gradually declining.

Keywords: Modified zeolite; Modified methods; Phosphorus removal rate; Influence factors.

1. INTRODUCTION

At present, the main method of sewage treatment is biological treatment in China, this method has little effect in removing nitrogen and phosphorus, most of nitrogen and phosphorus are discharged into river eventually, the results of these substances cause eutrophication. To solve this problem, countries around the world had carried out extensive research[1,2]. The most common methods are precipitation, crystallization, biological removal, adsorption and ion exchange[3] at home and abroad, at the same time, the adsorption is concerned by more and more scholars for its simple process and easy operation[4]. The zeolite is a new adsorbent material in removing phosphorus. However, the natural zeolite is not good at removing phosphorus[5]. In this paper, we use natural zeolite as raw material. In addition, these natural zeolite are modified by using some methods, and then we will do research by use of modified zeolite to explore the effect of modified zeolite in removing phosphorus and its influence factors.

2. TEST INSTRUMENTS AND MATERIALS

2.1 Test instruments and equipment

721 Spectrophotometer; PHS-3C precision pH meter; HZ-9811K temperature two-speed oscillator; electric dryer blast; YXQG02 portable electric pressure steam boiler; box-type resistance furnace; electronic balance. The determination of TP is Molybdate Spectrophotometry (GB11893-89).

2.2 Test materials

Zeolite: The research material is selected from Zhengzhou, Henan Province, Chemical composition and the mass fraction: SiO₂ (67.34%), Al₂O₃ (11.61%), CaO (1.07%), K₂O (3.63%), Fe₂O₃ (0.78%), Na₂O (2.45%), K₂O (0.67%). Chemical reagents: H₂SO₄, 1+1; K₂S₂O₈, 50g/L; Ascorbic acid, 100 g/L.

Chemical reagents: H₂SO₄, 1+1; K₂S₂O₈, 50g/L; Ascorbic acid, 100 g/L; Molybdate solution; phosphorus standard solution: 2mg/L; the mass fraction of 1% NaOH; the mass fraction of 1% HCl; Chemical drugs: AlCl₃, MgCl₂.

3. Study on Modified Zeolite in Removing Phosphorus and its Influence Factors

3.1 The best method of modified

Aluminum and Magnesium were respectively added to zeolite of a certain quality and make them fully dissolved, pH=7, standing for 48h, and then put them in drying oven at 100°C for sparing. 25ml phosphorus solution is added to a series of 250mL flasks, the solution concentration is 2mg/L, and then the four different zeolite are respectively added to flasks. They will continue constant shaking in a 150r·min⁻¹ at 25°C. After 70 min, we will take supernatant from flasks. At last, we will measure the concentration of phosphorus in the filtrate. The comparative of natural zeolite and modified zeolite in removing phosphorus is showed in Table 1.

Table 1 The comparative of natural zeolite and modified zeolite in removing phosphorus

Modified methods	Natural zeolite	Mg-Al modified zeolite	Mg modified zeolite	Al modified zeolite
Phosphorus removal rate(%)	43.6	62.86	49.4	52.3

The experiments show that zeolite of adsorption to phosphorus can be greatly improved after Mg-Al modification. The modified zeolite by a certain percentage of adsorption performance is better than pure reagent or reagent combination on phosphorus. And it is on phosphorus adsorption capacity is not the same as the original on phosphorus adsorption capacity of zeolite and reagent adduct, but its essence has changed, the zeolite 、 reagents are produced a synergistic effect, which will markedly enhance its capability of adsorption of phosphorus.

Table 1 shows that Mg-Al modified zeolite has the highest phosphorus removal rate. In addition, quality score, time and temperature is also important influence factors of phosphorus removal. According to table 2, get a certain amount of zeolite, AlCl_3 , MgCl_2 and add water to mixture ratio. Under the $\text{pH}=7$, standing for 48h, and then put them in drying oven at 100°C for sparing, take the above zeolite and put them in the box-type resistance furnace, and then change the heating time and temperature (table 3). At last, we will weigh 2g modified zeolite and add them into 25ml phosphorus solution, the solution concentration is 2mg/L. After 70 min, we will take supernatant from flasks. At last, we will measure the concentration of phosphorus in the filtrate.

Table 2 Types of modified zeolite

Types	The ratio of quality
The modified zeolite A	Zeolite: AlCl_3 : MgCl_2 = 6:2:1
The modified zeolite B	Zeolite: AlCl_3 : MgCl_2 = 6:1:1
The modified zeolite C	Zeolite: AlCl_3 : MgCl_2 = 6:2:0.5

Table 3 Factor levels list

Factors	Factor		
	The modified zeolite A	The modified zeolite B	The modified zeolite C
Quality ratio			
Heating time(h)	2	4	6
The heating temperature($^\circ\text{C}$)	320	450	500

Table 4 The orthogonal experimental analysis

Factors	Quality ratio	Heating time	The heating temperature	Removal rate(%)
Experiment 1	1	1	1	62.3
Experiment 2	1	2	2	78.4
Experiment 3	1	3	3	56.2
Experiment 4	2	1	2	56.2
Experiment 5	2	2	3	62.9
Experiment 6	2	3	1	17.6
Experiment 7	3	1	3	5.4
Experiment 8	3	2	1	2
Experiment 9	3	3	2	0.68
Average 1	62.30	41.30	27.30	
Average 2	45.56	44.43	41.76	
Average 3	2.69	24.82	41.50	
Range	59.60	19.60	14.46	

Table 3 and 4 shows that the orthogonal experimental analysis. In addition, experiments show that the phosphorus removal rate is the highest when the mass fraction of zeolite and Mg^{2+} 、 Al^{3+} is 6:2:1 and the heating time is 4 h, the temperature is 450°C .

3.2 The effect of modified zeolite dosage in removing phosphorus

25ml phosphorus solution is added to a series of 250mL flasks, the solution concentration is 2mg/L, the modified zeolite that particle size is 30-100 is ready, and then respectively put 0.1g、0.3g、0.5g、0.7g、1g、2g、4g、6g、8g、10g of the above modified zeolite into 250mL flasks. They will continue constant shocking in a $150\text{r}\cdot\text{min}^{-1}$ at 25°C . After 70 min, we will take supernatant from flasks. At last, we will measure the concentration of phosphorus in the filtrate. The removal of phosphorus of different dosage modified zeolite is showed in Table 5 and 6.

Table 5 The removal of phosphorus of different dosage modified zeolite

Dosage(g)	0.1	0.3	0.5	0.7	1.0
Residual concentration (mg/L)	1.59	1.13	1.01	0.91	0.67
Removal rate(%)	20.5	43.5	49.53	54.4	66.3
	1	9		6	6

Table 6 The removal of phosphorus of different dosage modified zeolite

Dosage(g)	2	4	6	8	10
Residual concentration (mg/L)	0.48	0.36	0.33	0.35	0.34
Removal rate(%)	75.84	81.94	83.29	82.62	82.79

Table 7 The removal of phosphorus of different adsorption

Adsorption (min)	10	30	50	70	130	180
Residual concentration (mg/L)	0.94	0.74	0.68	0.48	0.27	0.27
Removal rate(%)	52.8	63.0	65.9	81.7	86.7	86.9

Table 5、Table 6 and Figure 1 show that with the increasing of modified zeolite ,the removal rate of phosphorus is gradually increasing. When the dosage of modified zeolite is 6g ,the highest phosphorus removal rate is 83.29%.From this point,with the increasing of modified zeolite, the removal rate of phosphorus is gradually achieving a balance. This is because the pore of modified zeolite is opened,Tt enables the rapid increase in the initial adsorption process to phosphorus. At the same time ,in the initial adsorption ,phosphate molecules are adsorbed by the large pores, this time the spread resistance is relatively small ,the adsorption rate is faster than other time. But with the increasing of modified zeolite, the concentration of phosphorus is gradually declining from the effluent,it makes the adsorption slower.After adsorption equilibrium, the average phosphorus removal rate is about 82%.

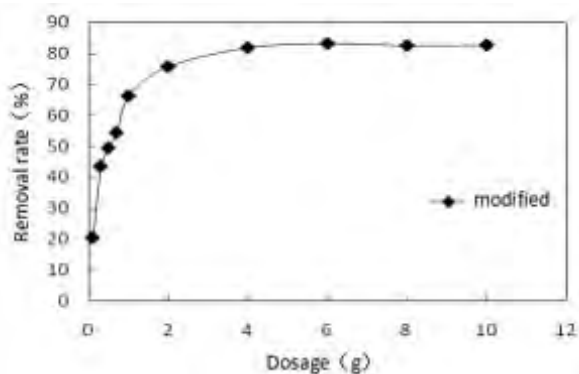


Figure 1 The removal of phosphorus of different dosage modified zeolite

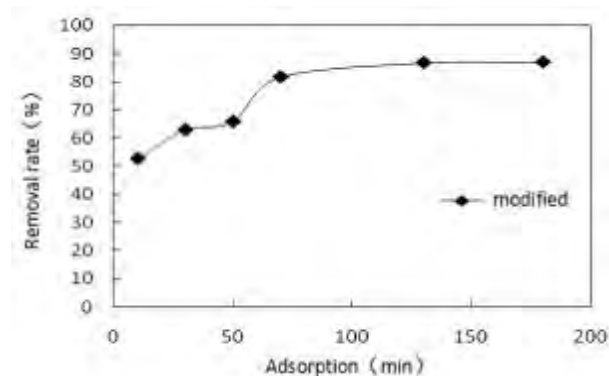


Figure 2 The removal of phosphorus of different adsorption influence curve

3.2 The effect of adsorption time in removing phosphorus

25ml phosphorus solution is added to a series of 250mL flasks, the solution concentration is 2mg/L, and then put 4g modified zeolite in flasks and continue constant shaking in a same temperature. Under $150\text{r}\cdot\text{min}^{-1}$, after different adsorption time, we will measure the concentration of phosphorus in the filtrate. The different adsorption time are respective 10 min、 30 min、 50 min、 70 min、 130min、 180 min .The removal of phosphorus of different adsorption is showed in Table 7.

Table 7 and Figure 2 show that with the increasing of adsorption time, the removal rate of phosphorus is gradually increasing. In particular, about 70 minutes after adsorption, the average of phosphorus removal rate is 85.1%. before 50min, the removal of phosphorus is obvious, after 70min the adsorption is gradually to balance. This is because that the modified zeolite have larger pores, and the more adsorption will occur on this surface of this larger aperture easily.

3.4 The Some Common Mistakes

25ml phosphorus solution is added to a series of 250mL flasks, the solution concentration is 2mg/L, respectively adjust the pH of every beaker, the pH is 5、 6、 7、 8、 9、 10. The modified zeolite that particle size is 30-100 is ready, and then the above 4g modified zeolite is respectively added to flasks. they will continue constant shaking in a $150\text{r}\cdot\text{min}^{-1}$ at 25°C .After 70 min, we will take supernatant from flasks. At last ,we will measure the concentration of phosphorus in the filtrate. The removal of phosphorus of different pH is showed in Table 8.

Table 8 The removal of phosphorus of different pH

pH	5	6	7	8	9
Residual concentration (mg/L)	0.27	0.23	0.29	0.33	0.47
Removal Rate(%)	86.70	88.7	81.3	80.6	76.50

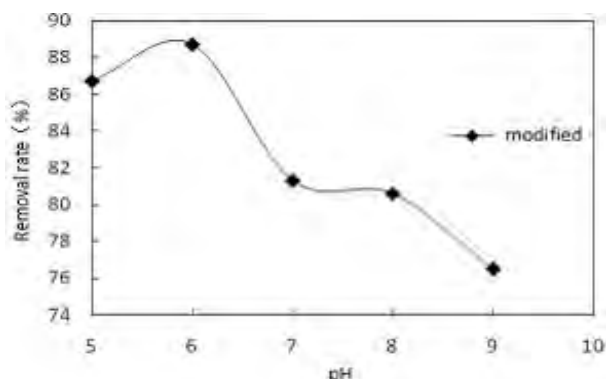


Figure 3 The removal of phosphorus of different pH influence curve

Table 8 and Figure 3 show that the modified zeolite has higher removal rate of phosphorus in acidic condition than the neutral or alkaline environment. The removal rate of phosphorus is gradually declining with the increasing of pH. When the pH=6, the highest phosphorus removal rate is 88.7%. This is because that ligand adsorption of replace hydroxyl is destroyed after aluminum, magnesium and phosphate interaction. In this case, only the ion exchange is carried out between ions. In others words, adsorption rate will be in a stable phase in some swap space.

3.5 The Effect of Initial Concentration in Removing Phosphorus

The modified zeolite that particle size is 30-100 is ready, and then the above 4g modified zeolite is respectively added to a series of 250mL flasks, we will put different initial concentration of phosphorus into 250mL flasks, the different initial concentration is respective 0.5, 1, 2, 4, 8, 10 mg/L they will continue constant shocking in a $150 \text{r} \cdot \text{min}^{-1}$ we will take supernatant from flasks. At last, we will measure the concentration of phosphorus in the filtrate. The removal of phosphorus of different initial concentration is showed in Table 9.

Table 9 show that modified zeolite has higher removal rate of phosphorus in appropriate concentration. This is because that the phosphorus will increase with the increasing of initial concentration within the certain range. At the same time, the concentration difference will be greater between fluid body and the film of the outer surface of zeolite. It will make the migration dynamics increase that the phosphorus move to zeolite surface. As the concentration increase, the removal rate shows a downward trend. When the initial concentration is 1mg/L, the phosphorus removal rate is 84.6%. That explains that the higher concentration of phosphorus will hinder adsorption in some extent.

Table 9 The removal of phosphorus of different initial concentration

Initial concentration (mg/L)	0.5	1	2	4	8	10
Residual concentration (mg/L)	0.40	0.31	0.58	1.55	3.97	5.47
Removal Rate(%)	79.9	84.6	81.9	61.2	50.4	45.3

5. CONCLUSION

When the modified zeolite dosage is 6g, the highest removal rate of phosphorus is 83.29%. Adsorption is an influence factor of the removal rate of phosphorus. About 70 minutes after adsorption, the average of phosphorus removal rate is 85.1%. pH is an important factor in the process of adsorbing phosphorus, under acidic conditions, the adsorption is obvious. In between pH=5~6, the phosphorus removal rate is gradually rising, when the pH=6, the highest phosphorus removal rate is 88.7%, when the pH>6, the phosphorus removal rate is gradually declining. When the initial concentration is 1mg/L, the phosphorus removal rate is 84.6%, as the concentration is increasing, the phosphorus removal rate is gradually declining.

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HOW FAR CAN "BAFFLE SHUTTERS" ATTENUATE OUTDOOR NOISE WHILE MAINTAINING ACCEPTABLE NATURAL VENTILATION RATES?

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ABSTRACT

Natural ventilation is an attractive solution for cooling. However its implementation on building is often limited due to privacy, security and external noise. The use of "baffle shutters" could reduce the noise exposition but it also could decrease the air flow rate, particularly in the case of single-sided natural ventilation.

This paper assessed the impact of different types of shutter on the acoustic attenuation and on the air change rate. The methodology is based on numerical simulation using Computational Fluid Dynamics for ventilation study and finite element method for the acoustic study. Two kind of shutters have been studied, one type with chicanes and the other one with louvers. A parametric study on the geometry of both shutter types has been undertaken accounting for the spacing between slats, their width, their inclination angle and their position (vertical or horizontal).

The results are compared to the same simulations without any shutter. The best reduction of noise reached is 13dB while the air change rate is reduced by 80%. The reduction of ventilation rate is unavoidable since the apparent opening surface is reduced by 50% due to the shutter. However, another configuration of shutter shows a reduction of air change rate of 50% only but 3dB on noise. The choice of the best design of shutter is a trade-off between noise attenuation and air change rate.

Key words: Natural ventilation; noise attenuation; numerical simulation; shutter; baffle; louver.

1. INTRODUCTION

Natural ventilation is an attractive cooling solution in the design of sustainable building environments. In the past years, many research works have been dedicated to the evaluation and improvement of natural ventilation performance [1-3].

Among the different natural ventilation strategies, the single-sided ventilation appears to be the easiest to integrate in buildings since this strategy limits the problems of privacy and noise between rooms. However the air change rate is lower compared to cross or stack ventilation. The single-sided ventilation is rather well accepted by the occupants but its implementation is generally limited by the outdoor noise.

The introduction of "baffle shutters" including air channels and noise obstacles could be a solution to reduce noise exposure and still allow indoor environment to "breathe". Since the shutters are obstacles reducing the air change rate compared to an opening surface, their design should be a trade-off between an adequate air change rate and significant noise attenuation.

This paper provides an analysis of noise attenuation and air change rate for 2 types of baffle

shutters based on CFD and FEM numerical simulations. One type of shutters is with two or three rows of baffles and the other one with inclined louvers. A parametric study on the geometry of both shutter types is then undertaken, varying the width, the inclination angle and the spacing of slats as well as the vertical or horizontal position of slats. The objective of this study is to identify the best concepts which will be assessed in later experiments.

In the single-sided natural ventilation, both analytical and numerical methods have been developed in the last decades for the assessments of the air change rates [4-5]. Analytical models have been successfully developed for top-hung window [6-7] and have been validated by experiments [8]. However, in the case of opening with shutters, the geometry is more complex, so that the numerical simulation turns to be the most appropriate tool which can reveal the impacts of various shutter design parameters, especially on the stack and wind effects which are the two driven forces in single-sided natural ventilation.

2. NUMERICAL SIMULATION METHODS

2.1. General description

In this paper, two types of shutters have been studied. The shutter is installed on a window with dimension of 1m*1m. The first shutter type consists in multi-layer slats: each slat is 20cm high (H1) with a thickness (e) of 1 cm or 2 cm; the slats are disposed on the windowsill; the slats are side by side from a view in normal direction; the shutter width (E) is of 10 cm or 20 cm. Two examples are shown in Fig.1(a&b). The second type consists in parallel louvers: the thickness of louvers 5 cm or 10 cm; the height for each louver is 2 cm or 4 cm and two inclination angles (11° and 22°) are studied. Two examples are shown in Fig.1(c&d). Both the slats and louvers could be set horizontally or vertically.

This work employs Finite Volume Method for ventilation study and Finite Element Method for acoustic study, respectively within the commercial calculation environment Fluent 14 and Comsol Multiphysics 4.3. In the CFD part, a RSM model is introduced and several conditions concerning the indoor and outdoor temperature as well as the wind speed and its direction are examined. For the FEM study, an external noise source is set up facing the facade and representing a plane incident sound wave in the normal direction. Sound Reduce Indices are obtained by evaluating the weakened sound transmission through

the aperture. The details of the phenomena of absorption and diffraction on noise frequency are presented according to different shutter geometries and spacing between baffles.

2.2. CFD simulations methodology

In the 3 main turbulence models existing (Direct Numerical Simulation, Large Eddy Simulation and Reynolds-Averaged Navier Stokes), the RANS can capture well the turbulent characteristics of air flows while consuming less calculation resource. Caciolo et al. [8] confirms that in most situations the RANS provides acceptable results with 20% of inaccuracy in about 30 times less computation time than the LES. We apply the Reynolds Stress Model (RSM) which is expected to increase accuracy compared to eddy-viscosity models. As an anisotropic turbulence model, additional transport equations are solved for each term of the Reynolds stresses tensor and the dissipation of the turbulent kinetic energy [9] so that the individual effects of the fluctuating quantities on the flow are taken into account.

The equations are solved by the commercial software Fluent. The pressure is interpolated by a "PRESTO!" scheme.

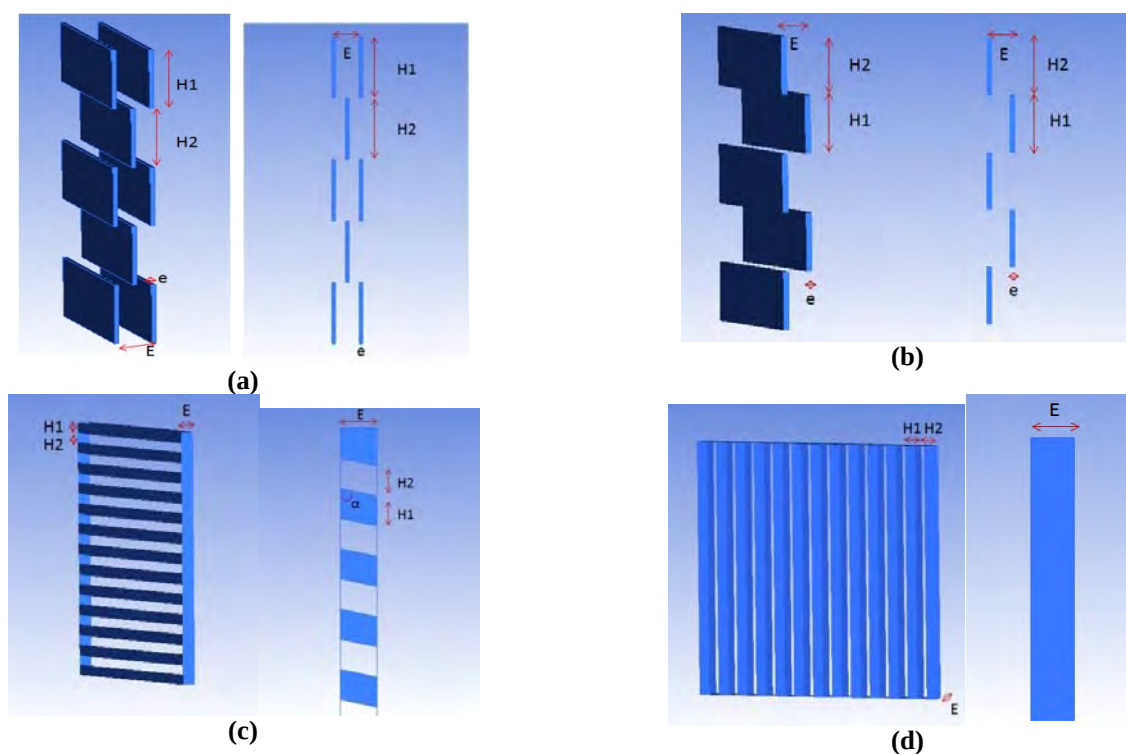


Figure 1 Shutter typology:

(a) 3 layers slats

(b) 2 layers slats, 3 slats outside and 2 slats inside (c)horizontal louvers (d)vertical louvers

The convective term, the diffusive term, the pressure term and the energy term use a second order upwind scheme. Pressure and velocity are coupled in the SIMPLE algorithm. Solution is considered to be converged while all residuals are confined in at least 10-3 of the initial calculation and remain stable for around 50 iterations. The convergence is also verified

by checking the mass and energy balances between indoor and outdoor environment with user-defined functions (UDF).

The domain is discretized in two structured meshes, separating as indoor and outdoor domain, linked by conformal interfaces. In order to restrict the border effects, the inlet, outlet and upper boundaries are

apart from the indoor domain ($L \times L \times H$) with a distance of $5L$, $10L$ and $5H$. The resulting mesh consists of around 600 000 cells.

The inlet boundary condition is based on the incident wind profile of imposed reference speed and turbulence intensity. The inlet speed uses a logarithmic profile neglecting the thermal effects which makes the advantage of consistency with the terrain roughness model [10]. The outlet boundary uses a pressure profile following the Boussinesq approximation, which presents a hydrostatic stress. In the room, a heat source, representing a computer and a printer, is set as convective surface with a heat flux of 100W/m^2 . The indoor walls are hold to a constant temperature (299K). The outdoor walls are smooth and adiabatic. The outdoor temperature is 295K .

2.3. FEM parameters

The fluid model for pressure acoustics used in this study is a linear elastic fluid by defining the density ρ and the speed of sound c . The pressure acoustics application solves for the acoustic pressure, p [11-14].

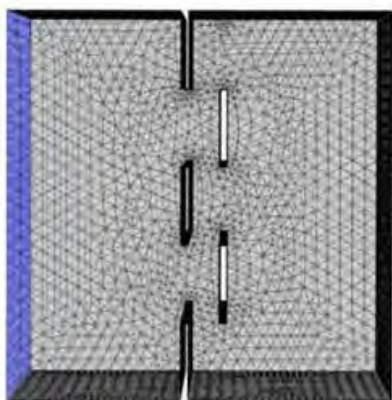


Figure 2 Example of FEM model (geometry and mesh) for pressure acoustics

Moreover, losses are accounted by introducing corrections related to the distortion of flow and increment of the mass participating to the total motion of air. The FEM modeling of the acoustical domains corresponding to holes, slats and louvers is a crucial point. In fact, from previous studies it has been demonstrated that using real valued density and sound speed for the air, the FEM model does not account correctly losses. The formulation used here is based on the equivalent dissipative fluid approach. According to this theory the holes, slats, louvers and porous materials can be considered as an equivalent fluid completely described by means of “equivalent” sound speed and density, complex valued, that are able to justify internal energy losses (visco-inertial and thermal effects) and phase shifts between pressure and particle velocity. In the widely used equivalent fluid model of Johnson-Champoux-Allard [15] (mainly utilized for porous materials but also for holes or perforated plates [16]), expressions for “equivalent” sound speed and density

are proposed. An example of the domains is depicted in Fig. 2.

A brief description of the domains and boundary conditions is given as follows:

-Air: $\rho=1.21\text{ kg/m}^3$ and $c=343\text{ m/s}$

Boundaries:

-Inlet (source): radiation condition ($p=1\text{ Pa}$)

-Outlet: radiation condition ($p=0\text{ Pa}$)

The radiation boundary conditions allow an outgoing wave to leave the modeling domain with minimal reflections.

-Remaining: hard wall (particle velocity is equal to zero on those boundaries).

The Sound Hard Boundary (Wall) feature node creates a boundary condition for a sound hard boundary or wall at which the normal component of the acceleration is zero.

Mesh (free tetrahedral) has been created according to the rule of 10 finite elements per wavelength.

The transmission loss is defined as:

$$TL = 10 \log \frac{P_{in}}{P_{out}}$$

where P_{in} and P_{out} denote the total acoustic power at the inlet and the outlet, respectively. The transmission loss is thus defined as a function of sound frequency.

P_{in} is defined by the surface integral:

$$P_{in} = 2 \int_{\text{inlet}} \frac{p_0^2}{2\rho c}$$

The power through the outlet is defined in an analogous surface integral over the outlet boundary, but with the computed pressure instead of the applied one:

$$P_{out} = 2 \int_{\text{outlet}} \frac{|p|^2}{2\rho c}$$

3. RESULTS AND DISCUSSION

CFD and FEM simulations have been for the different configurations of shutter s as described in § 2.1. For each configuration, CFD simulations are carried out with wind speeds of 0 m/s , 1.5 m/s , 3.5 m/s and 5.5 m/s ; in windward direction (30°) and leeward direction (150°). All these shutters reduce the apparent opening surface by 50%. The results of a CFD simulation are given in terms of average pressure, velocity and temperature fields. As shown in Tables 1 and 2, the air change rate results in column 5 are the range of all wind speeds and are compared to the reference simulations without shutter in column 6. The noise diminution in column 7 is the arithmetic average on 500, 1000, 1500, 2000, 2500 Hz. The 3-layers cases have the layer of 3 slats outside. The two layers cases can be distinguished by having the layer of 2 slats inside (2i) or outside (2o).

Table 1 Slats shutter ventilation and acoustical performance

slats layers	Horizontal / vertical	E (cm)	e (cm)	Air change rate (vol/h)	Comparison (%)	Noise diminution (dB)
3	H	20	2	1.0-2.0	15%	13
3	V	20	2	2.0-2.8	22%	13
3	H	10	2	1.1-2.0	15%	5
3	V	10	2	1.6-2.6	19%	5
2i	H	10	2	2.4-3.2	25%	10
2i	V	10	2	2.5-4.0	30%	10
2o	H	10	2	2.2-2.6	22%	11
2o	v	10	2	2.5-3.6	29%	11

Despite of the reduction on apparent opening surface of 50%, the air change rate (ACR) reaches between 15% and 60% of the ACR without shutter. The best noise reduction is 13 dB with the 3-layer shutter while the ACR is reduced by about 80%. The air recirculation is largely stopped by the shutters so that this configuration should not appear as favorable.

Table 2 Louvers shutter ventilation and acoustical performance

Horizontal / vertical	H1(cm)	E(cm)		Air change rate (vol/h)	Comparison (%)	Noise diminution (dB)
H	2	5	22	1.3-2.6	18%	3
V	2	5	22	2.9-4.8	50%	3
H	4	10	11	2.0-3.5	25%	2
V	4	10	11	4.4-6.6	60%	2

For louvers shutters, the ACR could reach even more than 50% of the ACR without shutter in the vertical position, which means that both the stack and wind effects are well preserved, but the noise attenuation is limited.

3.1. Ventilation performance

In the following part, a focus on 4 shutters is undertaken by analyzing the flow fields referring to the vertical section in the middle of window (view A) and the opening plan section in the center of shutter from an indoor view (view B). With the simulation results, the air change rate is calculated as the sum of convection and diffusion flows over the opening plane.

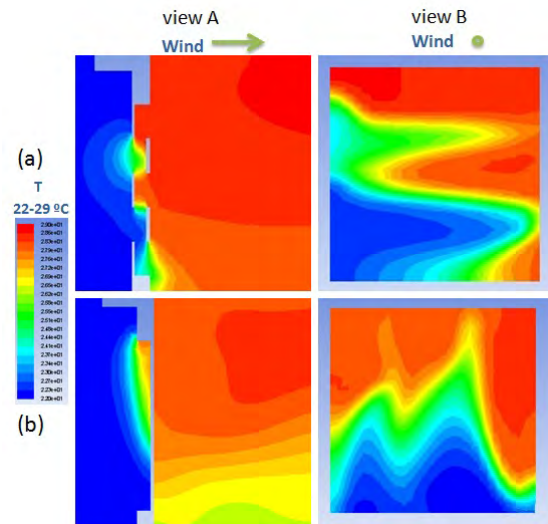


Figure 3 Steady-state 2D temperature fields: (a) horizontal slats (b) vertical slats

In the two cases presented, the horizontal and the vertical shutters have exactly the same geometry characteristics from an acoustic point of view with the noise source stemming from normal direction so that the acoustical performance is the same.

First cases are horizontal and vertical 2-layer slats shutters (2i). Fig. 3 shows the computed temperature fields.

For both horizontal and vertical cases, the air enters in the room by the bottom apertures and goes out through the top apertures. As the stack height is truncated by horizontal slats and is kept with vertical slats, the horizontal ones tend to reduce the buoyancy effect contrary to the vertical ones. The wind effect is strongly reduced in both cases. Finally the horizontal slats have lower air change rate than the vertical slats.

For horizontal and vertical louvers, the temperature field (Fig. 4) clearly shows that stack effect is well preserved for both shutters. In vertical louver case, the vortex in front of the shutters is amplified and forms a larger mixing zone.

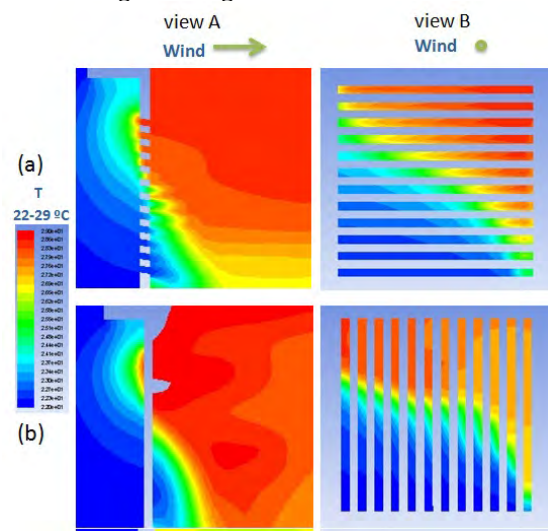


Figure 4 Steady-state 2D temperature fields : (a) horizontal louvers (b) vertical louvers

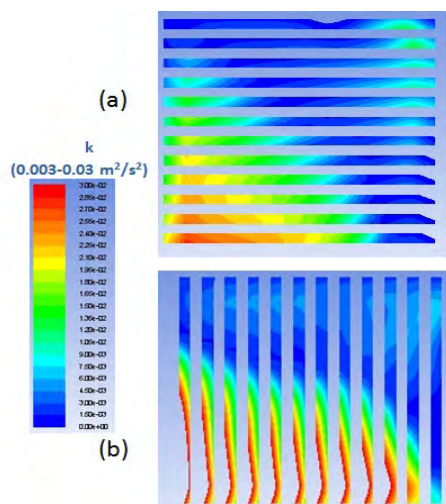
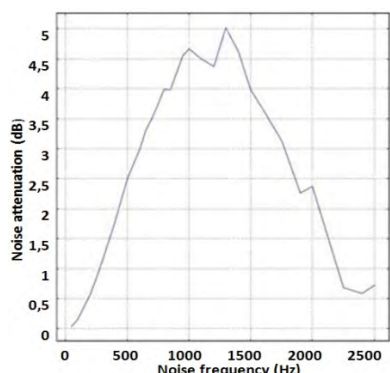


Figure 5 Steady-state 2D turbulent kinetic energy fields : (a) horizontal louvers (b) vertical louvers

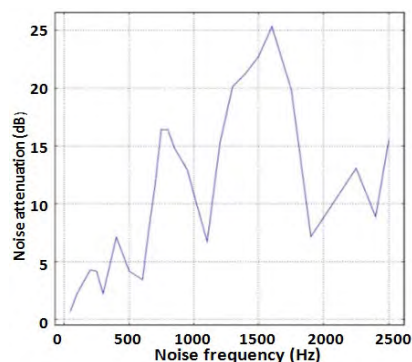
The calculated turbulent kinetic energy is higher over the opening plane (Fig. 5). By this contribution of wind effect, the vertical shutters reach a better air change rate.

3.2. Acoustical performance

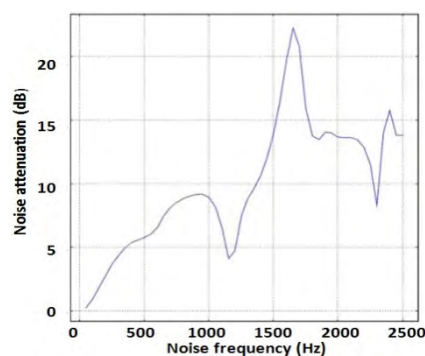
In the following part, different shutters tested are picked out as examples to show the main differences between slats or louvers. Indeed, as shown in Table 1 and in Fig. 6, there is a large difference in attenuation between louvers (around 3dB) and slats (more than 10dB).



(a) horizontal louvers



(b) horizontal 3-layer slats (E=20cm)



(c) horizontal 2-layer slats (20)

Figure 6 Transmission Loss (dB) in function of Frequency (Hz)

For louvers, even if there is a peak in Transmission Loss around 1000Hz, this peak is very small (around 5dB) so the attenuation is very low. Outside the frequency range where the peak is localized, the sound is transmitted so the isolation is very bad (Fig.7).

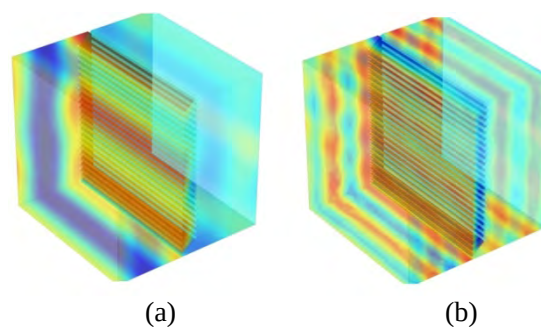


Figure 7 FEM model: pressure distribution for louvers : (a) around 1000Hz and (b) around 2500Hz

For slats louvers, the Transmission Loss is more interesting, where the pressure distribution shows the efficiency of slats (there is no direct transmission through slits). The acoustic pressure can be very low behind the slats (Fig. 8).

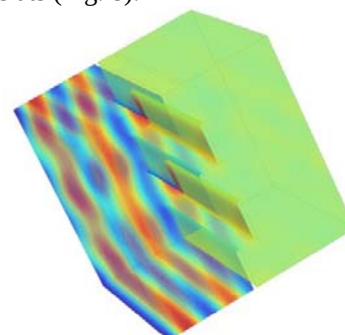


Figure 8 FEM model: pressure distribution for 2-layer slats around 2500Hz

A good compromise can be found with 2-layers slats where the performance is quite good (around 10dB).

4. CONCLUSION

In order to design new shutters reducing external noise while maintaining acceptable natural ventilation, this study has been carried out using CFD for ventilation aspect and the FEM for acoustical aspect.

The louver shutters limit the reduction of the air change rate compared to the reference case without shutters. However the acoustical diminution is of 3dB on average. The slat shutters, on the contrary, reducing significantly the noise by 10 dB, lowers strongly the ventilation rate. Two designs with the best trade-offs, the 2-layers vertical shutters and the vertical louvers, are selected to be studied experimentally in future works.

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STUDY ON THE HEAT REJECTION AND COOLING EFFECT OF AIR DISTRIBUTION ON EXCAVATION ROADWAY

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ABSTRACT

In this article, the cooling technology of thermal insulation and air diffuence in excavation roadway is analyzed. By using the FLUENT software, the air flow distributions in the roadway with the air velocity changing between 1m/s and 5m/s, and the distance between the insulation board and the surrounding rock of excavation roadway B being 0.2m, 0.25m, 0.3m, and angle of diversion plate α changing between 15° and 75° are simulated. The results show that, in addition to v , the values of B and α also have impact on the cooling effect of the thermal insulation and air diffuence technology, and the best results are obtained at the distance between the heat insulation plate and surrounding rock of excavation roadway B being 0.2m, and the angle of diversion plate changing between 30° and 45°.

Keywords: Air distribution, Excavation roadway, Insulation plate, Cooling effect, FLUENT software simulation

1. INTRODUCTION

With the mining depth increasing, the underground temperature rises and the heat harm of high temperature mines becomes worse. Solving this problem is an urgent and challenging task. In general, the mine cooling technologies can be divided into non-artificial and artificial refrigeration^[1-4]. The non-artificial refrigeration methods include ventilation^[5-7], controlling heat sources, filling and the pre-cooling of the entering air^[8]. The artificial refrigeration methods mainly include water cooling technique, artificial ice cooling technology^[9] and compressed air refrigeration. But the efficiency of artificial refrigeration is low and high energy consumption. Given these problem, Zhang Xijun made mechanical refrigeration combine traditional cooling methods for the integrated cooling^[10]. He Manchao, professor of the China University of Mining and Technology in Beijing, put forward the HEMS cooling technology, which uses cold mine gushing water as a cold source^[11]. However, it is very important to develop the non-artificial refrigeration in this energy shortage society. Then, the

professor Zou Shenghua, proposed a non-artificial refrigeration technique: thermal insulation and air diffuence technology in heading face^[12]. This article intends to make a further research for that technology.

A good airtight insulation board made of the material with the thermal conductivity being less than 0.05W/m·K is set a certain distance to the surrounding rock of the roadway in that technology, dividing the airflow from the loaded local fan into two parts. The space within the insulation plate is the mainstream road with the airflow creating the thermal and wet environment in the excavation roadway. The annular channel composed of the insulation plate and the surrounding rock is the secondary flow road and the airflow mainly exhausts harmful gases, dust, heat and humidity in the roadway. A diversion plate with adjustable angle is located at the end of the insulation board. This technology achieves the purpose of cooling by cutting off the heat transfer of the leading airflow and the secondary airflow. Compared to other methods of cooling, it is simpler, easy maintenance and economic. The physical model is shown in Figure 1.

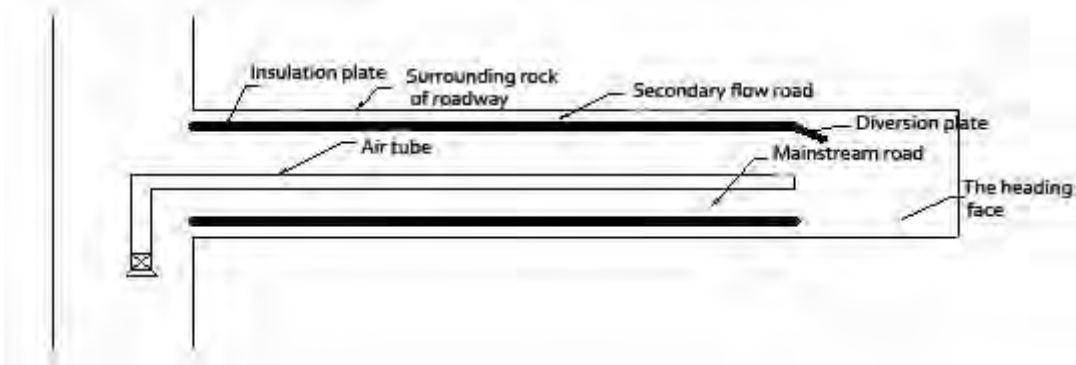


Figure 1 Scheme of thermal insulation and air diffuence technology in excavation roadway

2. NUMERICAL SIMULATION OF AIR DISTRIBUTION

For the thermal insulation and air diffuence technology, there are many factors influencing the cooling effect. In this article, the cooling effect of air velocity, the distance between the insulation board and the surrounding rock, and the angle of the diversion plate on the excavation roadway are simulated and analyzed by using the FLUENT software, based on a specific model.

2.1 Mathematical model

In order to simplify the model, the following assumptions are made: (1) The airflow in the roadway is incompressible and the heat consumption caused by the fluid viscous force is ignored; (2) The physical properties of solids and gases are considered as constants, and the evaporation of the water vapor and the gas desorption are not taken into account; (3) The flow is steady and turbulent with the fluid turbulent viscosity is isotropic; (4) Only the heat caused by the surrounding rock is considered, ignoring the influence of excavating equipment and other heat sources; (5) The surrounding wall is airtight and no chemical reactions are produced in the roadway.

The airflow in the excavation roadway complies with the laws of mass, momentum and energy conservation. The control system of the turbulent heat transfer model using standard $k-\epsilon$ two-equation model includes the continuity equation, the momentum equation, the energy equation, k equation and ϵ equation. These equations may be written as the following general form^[13]:

$$\frac{\partial(u\Phi)}{\partial x} + \frac{\partial(v\Phi)}{\partial y} + \frac{\partial(w\Phi)}{\partial z} = \frac{\partial}{\partial x} \left(\Gamma \frac{\partial \Phi}{\partial x} \right) + \frac{\partial}{\partial y} \left(\Gamma \frac{\partial \Phi}{\partial y} \right) + \frac{\partial}{\partial z} \left(\Gamma \frac{\partial \Phi}{\partial z} \right) + S_\Phi \quad (1)$$

Where u , v , w , are velocity components, Φ the common variables which can represent the solution variables of u , v , w , T , etc., Γ the diffusion coefficient, S_Φ the source item of the general equation.

2.2 Geometric model

The profile of the excavation roadway is semi-circular arch with straight walls and a good airtight insulation board is set a certain distance to the surrounding rock. The simplified model of the excavation roadway has a length of 25m with the width being 3m, the height being 2.8m (the height of straight walls 1.3m and the arch 1.5m). The insulation board with a length of 16.5m is 3.5m and the outlet of the duct with the diameter being 0.3m is 6m away from the heading face. The distance of the insulation board and the surrounding rock is set to 0.2m, 0.25m, 0.3m and the air velocity changes between 1m/s and 5m/s. The dimension of the diversion plate with its angle being 15°, 30°, 45°, 60°, 75° is 0.6m x 2m. The coordinate origin is located at the end of the insulation board being 5m away from the outlet. The three-dimension model is shown in Figure 2.

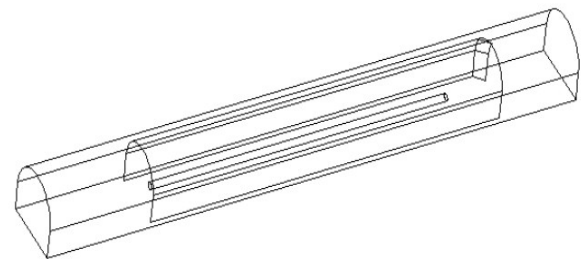


Figure 2 Three-dimension model of the excavation roadway

2.3 Settings for the parameters and boundary conditions

The density of the air is set to 1.225 kg/m³ with the viscous coefficient being 1.79x10⁻⁵ Pa·s; The operation pressure is set to 1.01x10⁵ Pa; The boundary condition of the air duct outlet is set as the velocity inlet with the temperature of the supply air being 293K. According to the size of the roadway model, the velocity of the supply air within 5m/S has been studied. The free section of the roadway is set as the pressure outlet without relative pressure. The Walls in the roadway are without slip and have constant temperatures with the

surrounding rock being 313K, the heading face 314K and the roadway floor 299K. The interface of heat insulation board and the adjacent channels is the solid-fluid coupled heat transfer boundary, selecting the coupling conditions with the thickness of heat insulation plate being 0.01m and thermal conductivity being 0.047W/m·k. The Pressure-Velocity Coupling model is SIMPLE and the discretization of pressure is standard with others being first-order upwind scheme.

The calculation formula of hydraulic diameter is written as:

$$D = \frac{4A}{S} \quad (2)$$

Where A is the flow section area (m^2), S the circumference of the junction of fluid and solid (m),

The calculation formula of turbulence intensity is written as:

$$I = \frac{u'}{u} = 0.16(Re_H)^{-\frac{1}{8}} \quad (3)$$

Where u' is the turbulent speed (m/s), u the average speed (m/s), Re_H the Reynolds number, ν the kinematic viscosity of air, $14.8 \times 10^{-6} m^2/s$.

3. NUMERICAL SIMULATION ANALYSIS OF THE COOLING EFFECT OF AIR DISTRIBUTION

3.1 The influence of different air velocities on the cooling effect

Depending on the model size and the distance between the air duct outlet and the heading face, the velocities of the supply air between 1m/s and 5m/s is studied. The distance between the insulation board and the surrounding rock is 0.25m with the angle of the diversion plate being 45° . The airflow temperature distribution graph of the mainstream road is shown in Figure 3.

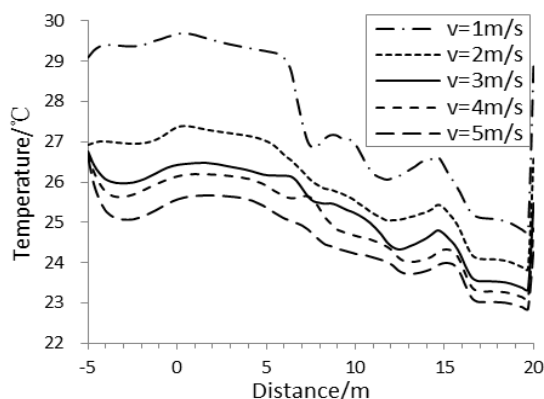


Figure 3 Airflow temperature distribution of the mainstream road under different values of v

In figure 3, with the heat transfer of the supply air and the high temperature air closing to the heading face, and the surrounding rock, the air temperature in the

mainstream road increases from the heading face to free section of the roadway. When the air velocity is 1m/s, the air temperature near the heading face closes to $25^\circ C$ and the highest air temperature in the mainstream road is up to $29.7^\circ C$; When the velocity increases to 2m/s, the temperature closing to the heading face drops by about $1^\circ C$ and the highest temperature in the mainstream road $2.2^\circ C$. The highest temperature in the mainstream road drops by $1.2^\circ C$ with the air velocity being 3m/s. While continue to increase the air velocity, the cooling rate reduces. Thus, it is necessary to choose the suitable supply air velocity by considering both the cooling capacity and economic factors in the actual design.

3.2 The influence of the distance between the insulation board and the surrounding rock on the cooling effect

The supply air velocity is set to 3m/s and the angle of the diversion plate is 45° with the distance between the insulation board and the surrounding rock being 0.2m, 0.25m, 0.3m. The airflow temperature distribution of the mainstream road is shown in Figure 4.

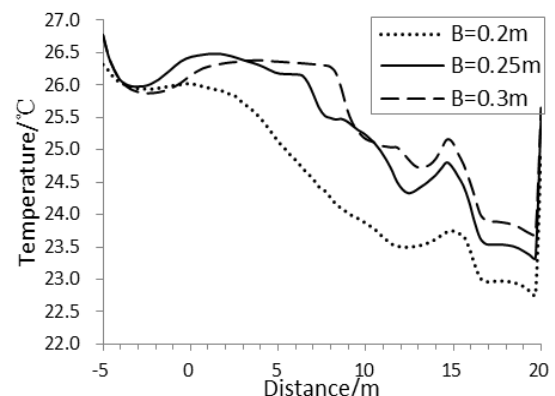


Figure 4 Airflow temperature distribution of the mainstream road under different values of B

As is shown in figure 4, when B is 0.2m, the air temperature in the excavation roadway is the lowest. With the increasing of the distance B , the airflow temperature of the mainstream road reduces, while the air temperatures in the mainstream road and closing to the heading face increase. When B is 0.25m, the air temperature closing to the heading face increases $1^\circ C$ and the highest air temperature in the mainstream road increases $3^\circ C$. While when B is 0.3m, the air temperature closing the heading face increases by $0.5^\circ C$ and the highest air temperature in the mainstream road increases by $1.6^\circ C$. Therefore, the distance should be smaller as possible in case of meeting the design requirements.

3.3 The influence of the angle of the diversion plate on the cooling effect

The angle of the diversion plate changes between 15° and 75° with the supply air velocity being 3m/s and the distance B being 0.2m. The airflow temperature distribution is shown in Figure 5.

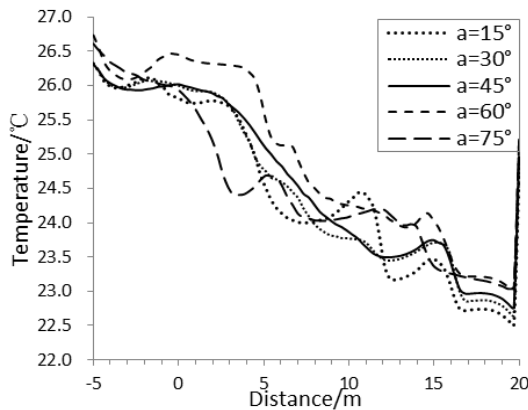


Figure 5 Airflow temperature distribution of the mainstream road under different diversion plate angles

In the excavation roadway, the supply air and the airflow closing to the heading face have a fierce heat transfer, producing the first backflow by dashing on the heading face. The secondary backflow is produced when the first backflow hits against the diversion plate with part of the airflow passing through the secondary flow road and having heat transfer with the surrounding rock and other part of the airflow mixing with the other first backflow maintaining the thermal and wet

environment in the excavation roadway. When the angle of the diversion plate is 15° , the volume and velocity of the airflow in the roadway are small under the effect of the diversion plate, causing the uneven temperature distribution. The temperature distribution in the mainstream road is improved with the increasing of the angle of the diversion plate. However, when the angle is beyond 45° , large fluctuations appear again and the temperature of the air closing to the heading face also increases with the angle being 60° the worst.

3.4 The temperature and velocity distribution of the air in the excavation roadway

The Figure 6 and Figure 7 respectively show the temperature and velocity distribution of contour with v being 3m/s, B being 0.2m and a being 45° . In Figure 6, the supply air and the airflow closing to the heading face have a fierce heat transfer, driving the air temperature closing to the heading face below 23°C , and the backflow is divided into two parts. The air temperatures of the secondary flow road and the increasing gradient are higher than those in the mainstream road with the both of their air temperatures increasing along the air flow direction. There is an air eddy whose temperature is higher than the surrounding at the back of the diversion plate due to its Shielding effect. In general, the cooling effect of the insulation board has achieved good results. As is shown in figure 7, most of the air velocity in the excavation roadway is low and uniform except for the large gradient near the air inlet



Figure 6 the temperature distribution of contour in the excavation roadway



Figure 7 the velocity distribution of contour in the excavation roadway

4. CONCLUSIONS

(1) The insulation board set in the excavation roadway can achieve good cooling effect by cutting off the heat and moisture exchange between the high temperature air near the surrounding rock and the air in the excavation roadway, and hindering the radiation heat transfer between the surrounding rock and the people, equipment, facilities.

(2) In this article, the cooling effect of the supply air velocity, the distance between the insulation board

and the surrounding rock, the angle of the diversion plate are simulated by using the FLUENT software. The results show that the air temperature in the excavation roadway decreases with the increasing of the supply air velocity, while the cooling effect is not obvious by continuing to increasing the air velocity when it reaches a certain value. Therefore, it is necessary to choose the suitable supply air velocity by considering both the cooling capacity and economic factors in the actual design. The distance between the insulation board and the surrounding rock also has effect on the air distribution of the excavation roadway and the distance

should be smaller as possible in case of meeting the design requirements. The air distribution becomes better and then worse with the increasing of the diversion plate angle and the best results are achieved between 30° and 45°.

(3) In addition to the above factors, the distance between the heading face and the closer end of the air duct or the insulation board, the length and material of the insulation board also have impacts on the air distribution of the excavation roadway. There are some limitations in this article with only taking a specific model into consideration. Therefore, a combination of factors should be considered, based on the actual situation with regard to the specific design, so as to obtain the best cooling design of the excavation roadway.

ACKNOWLEDGEMENTS

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STUDY ON RELATION BETWEEN MICRO-STRUCTURE AND MACRO-PERFORMANCE OF DIATOMITE-BASED HUMIDITY CONTROL BUILDING MATERIAL

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ABSTRACT

The paper shows a kind of Diatomite-Based Humidity Control Building Material (DBHCBM) which can automatically modulate indoor air relative humidity. It is found that calcination pretreatment of main raw material-diatomite can effectively impact the micro-structures and the humidity control performances. Therefore, Micro-structures are characterized by means of SEM and BET, macro performances of DBHCBM are tested in the artificial environment chamber in order to compare with the micro-structure. The results show that micro-structure of diatomite including pore structures significantly affect macro performance of diatomite and DBHCBM, that is, a better humidity control performance of diatomite is attribute to a smaller pore diameter, a big number of small pores; the best comprehensive humidity control ability of DBHCBM is when the stirring time is 15 minutes.

Keywords: Humidity control performance; Micro-structure; Pore structure; EM; diatomite; BET.

1. INTRODUCTION

It is well known that the high or low indoor air humidity will cause various problems, such as occupant's comfort, goods storage, durability of building envelope and energy consumption. Therefore, indoor air humidity must be modulated to a certain range. In the past, HVAC (Heating, Ventilating and Air Conditioning) are mainly applied to control indoor air humidity, considering its large energy consumption, Humidity Control Material (HCM) was first proposed by a Japanese scholar Nishifuji Miyano [1], HCM can sense the variations of temperature and relative humidity (RH) of air without any power source or mechanical equipment, and then adsorbs or releases moisture automatically. Thus, HCM is of great importance to energy conservation, indoor air humidity control and sustainable development.

HCM receives much concern both home and abroad for its unique properties, in the 1990s, Tsinghua university carried out the research on HCM [2], Luo Xiyun [3] introduced the humidity control performances and studied the mechanisms of silica gel, inorganic salt, montmorillonite, organic polymer, Ran Maoyu [10] studied the humidity control performance of silica gel and investigated the mechanism of humidity control performance. Hou Guoyan [4] prepared a putty powder with diatomite and sepiolite, the humidity control performance was superior to normal putty powder. Zuoyi Chen [5] invented five kinds of HCM by using starch to modify diatomite / white cement and also did experiments to study the characteristics of HCM. A Japanese company-INAX developed a healthy building material which can

modulate indoor air humidity and reduce harmful chemical components [6]. F.Caturla et al. [7] carried out the research on humidity control performances of sepiolite and active carbon composite. Prabal Talukdar [8] tested the temperature, relative humidity and moisture accumulation distributions in different parts of the humidity control materials (spruce plywood and cellulose insulation) under different airflow Reynolds numbers above the materials, he also analyzed the uncertainty and repeatability of the measured data. The predictions agree well with the above testing results [9]. Dinh-Hieu Vu [10] investigated the impact of sintering time, sintering temperature and portions of the raw materials on the properties of humidity control porous ceramics.

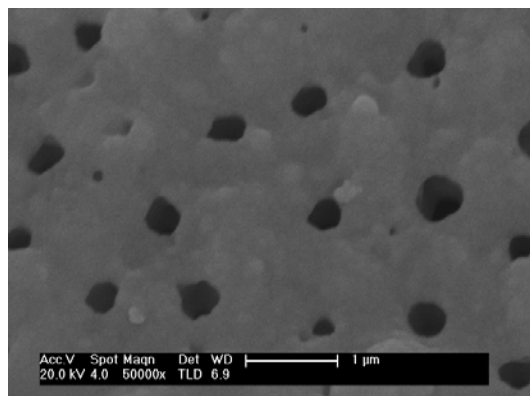
The paper aims at improving the humidity control performance of DBHCBM, diatomite in different regions are studied, calcination pretreatment of diatomite can effectively impact its microstructure and then influence the humidity control performance of DBHCBM, SEM and BET are applied to study the properties of diatomite and analyze the humidity control mechanism of diatomite. Some experiments of diatomite and DBHCBM are carried out to assess the impact of calcination times and calcination temperatures of diatomite, the stirring times of DBHCBM on their humidity control performance.

2. EXPERIMENT

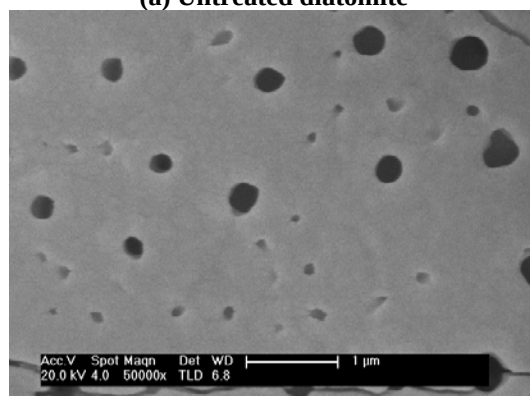
2.1 Pretreatment of diatomite

Diatomite is the main humidity control component of DBHCBM, Figure 1 shows the SEM micrographs of

untreated and calcined diatomite, the pores distribution is even in untreated diatomite, in comparison, calcined diatomite appears fractures, breakages and many smaller pores. Calcination changes the micro-morphology of diatomite which can improve the humidity control performance of diatomite. Actually, the essence of the pretreatment is to change the micro pore structure of diatomite.



(a) Untreated diatomite



(b) 3h calcined treated diatomite at 500°C

Figure 1 Scanning electron micrographs of diatomite

2.2 Experimental apparatus

Saturated solution method is adopted to make different levels of ambient relative humidity which shown in Table 1, three kinds of solutions are used to create different humidity levels to satisfy different environments, testing chamber can create different levels of ambient temperature, we don't concern the impact of ambient temperature on humidity control performance, so 20 °C is chosen as the testing environment temperature. The facilities shown in Figure 2 also include hygrothermograph, high precision electronic balance and sealed plastic box etc., the facility can measure adsorption content and desorption content of diatomite or DBHCBM within a certain time, and then evaluate its humidity control properties.

Table 1 Relative humidity of saturated solution at 20°C

Name	Molecular formula	Relative humidity (%)

magnesium chloride	MgCl ₂	33.07±0.18
sodium bromide	NaBr	59.14±0.44
potassium chloride	KCl	85.11±0.29



Figure 2 Experimental apparatus

2.3 The preparation of DBHCBM

Cement is the main source of structural support; Pulverized fuel ash can be used as a supplement to reduce cement consumption, which can lower the cost and utilize the industrial waste; Diatomite is the main humidity control material while poplar cellulose is the assistant humidity control material; Mildew preventive and antimicrobial are added to DBHCBM for environmentally-friendly effect.

All the components are thoroughly stirred and mixed, pulped, poured into molds, shaped, removed from the molds and maintained at a certain temperature and humidity to produce the final DBHCBM products shown in Figure 3.



Figure 3 Picture of DBHCBM

2.4 Measurement protocol

Before adsorption experiment, the prepared DBHCBM testing blocks or diatomite samples should be put into drying oven at 120°C until the changes of their weights are within 1% which can be regard as thoroughly dried, after that, the testing blocks should be taken out of the drying oven swiftly and put into desiccator until the temperature of the testing blocks or diatomite samples reduce to ambient air temperature, it

should be noted that waterproof and airtight treatment should be done on the five surfaces except adsorption / desorption surface of DBHCBM, after that, adsorption content measurement could begin. When the adsorption process approaches to saturated, diatomite or DBHCBM should be put into low humidity environment swiftly to start desorption content measurement.

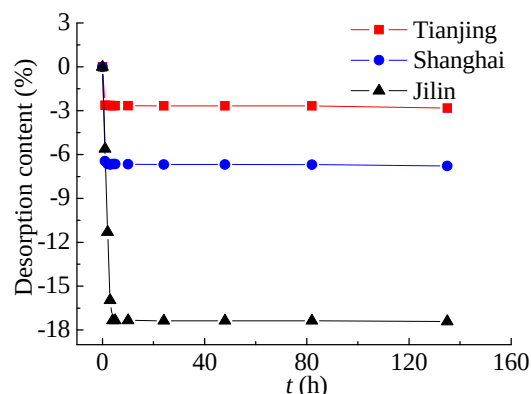
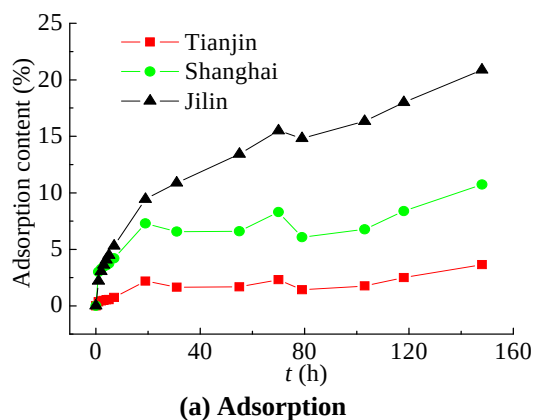
when the artificial environment are stable, the initial weight of testing blocks should be weighed and then put them into the prepared artificial environment box, in the first six hours, testing blocks should be measured every hour, after that, two times a day until the variation of their weights are within 1% which means we can finish the measurement, the measurement protocol could be adjusted according to the measured data.

3. RESULTS AND DISCUSSIONS

The experiments mainly study the humidity control performance of diatomite and DBHCBM, respectively, so the temperature environments are settled as 20°C.

3.1 Humidity control performance of diatomite from different regions

Diatomite samples from Tianjin, Shanghai, and Jilin were chosen as possible materials to serve as the major component of DBHCBM. The results in Figure 4 show that in terms of adsorption and desorption, humidity control performance of the diatomite from Jilin is obviously superior, it is also proved from the BET results shown in Table 2 that the surface area and pore volume of diatomite from Jilin are the smallest, so we can conjecture that humidity control performance of diatomite does not necessarily attribute to large surface area and pore volume, or the surface area and pore volume can't be tested with BET method, which might be attribute to high surface free energy. But the pore size of diatomite from Jilin is the smallest which can produce large capillary forces, therefore, diatomite from Jilin was chosen as the main raw material of DBHCBM.

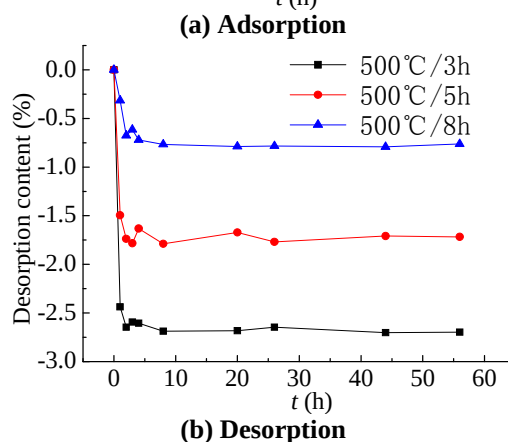
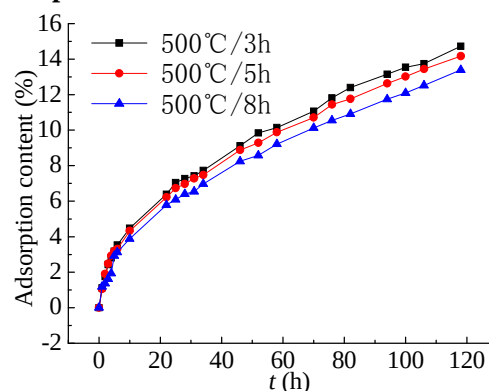


(b) Desorption
Figure 4 Adsorption / desorption contents of DBHCBM at different regions

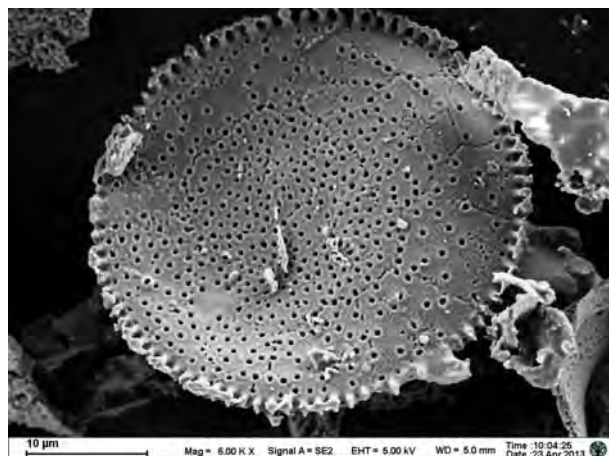
Table 2 BET results

Name	Surface area(cm ² /g)	Pore volume(cm ³ /g)	Pore size (nm)
Tianjin	1.2646	0.001673	5.68
Shanghai	1.2241	0.001264	4.13
Jilin	0.5557	0.000977	2.58

3.2 The impact of calcination times on humidity control performance of diatomite



(a) Adsorption
(b) Desorption
Figure 5 Adsorption / desorption properties of three diatomite samples treated at different calcination times (3h/5h/8h)



(a) 500°C/3h



(b) 500°C/5h



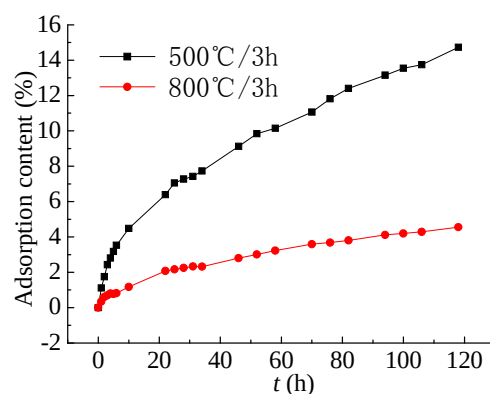
(c) 500°C/8h

Figure 6 Scanning electron micrographs of the three diatomite samples treated at different calcination times (3h/5h/8h)

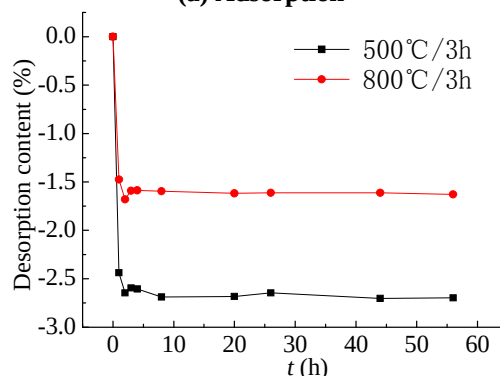
Figure 5 shows the adsorption / desorption properties of three diatomite samples which has been 500 °C calcination treated for different times (3h/5h/8h). It is illustrated that the longer the calcination time, the worse the adsorption and desorption properties of the diatomite. From the scanning electron micrographs of three diatomite samples, we can see clearly that the pores of diatomite

in Figure 6(a) are small and dense, some tiny fractures and breakages appears; the pores of diatomite in Figure 6(b) are bigger than Figure 6(a); the pores of diatomite in Figure 6(c) are blocked by some melt because of the longest period of calcination. Therefore, different pore structures of diatomite samples result in different humidity control performances.

3.3 The impact of calcination temperature on humidity control performance of diatomite

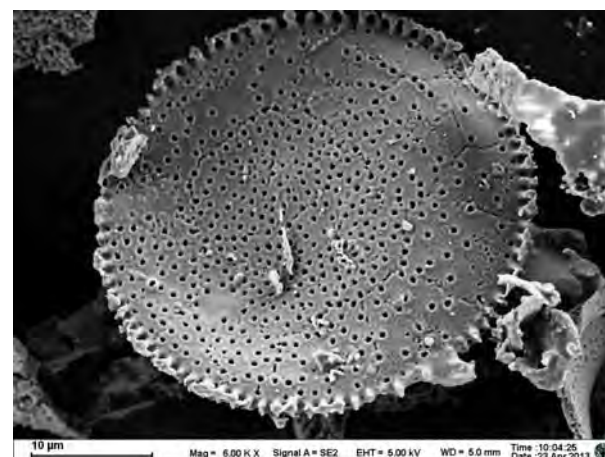


(a) Adsorption



(b) Desorption

Figure 7 Adsorption / desorption properties of three diatomite samples treated at different calcination temperatures (500°C/800°C)



(a) 500°C/3h

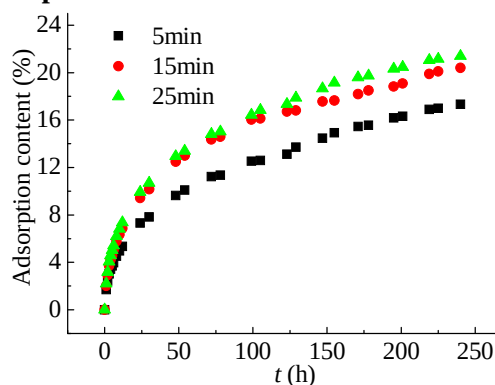


(b) 800°C/3h

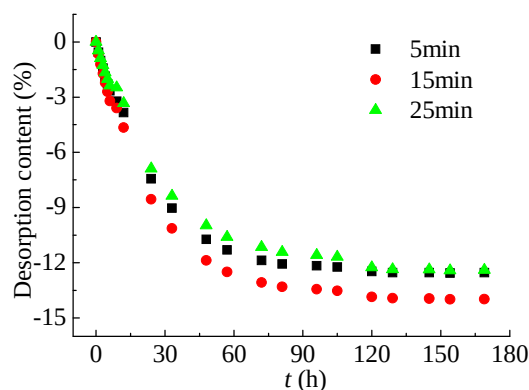
Figure 8 Scanning electron micrographs of the three diatomite samples treated at different calcination temperatures (500°C/800°C)

Figure 7 illustrates the adsorption / desorption properties of three diatomite samples which has been three hours' calcination treated at different calcination temperatures (500°C/800°C). It is indicated that the higher the calcination temperature, the worse the adsorption and desorption properties of the diatomite. Figure 8 shows the scanning electron micrographs of three diatomite samples, the pores in Figure 8(a), the same as Figure 6(a), are small and dense, some tiny fractures and breakages appears, the pore structure is relatively complete; the pore structure in Figure 8(b) is uncompleted and some of the pores are blocked by the melt, so the humidity control performance is not preferred.

3.4 The impact of stirring time on humidity control performance of DBHCBM



(a) Adsorption



(b) Desorption

Figure 9 Adsorption / desorption isotherms of DBHCBM at different levels of stirring time

The experiment investigates the adsorption / desorption isotherms of DBHCBM at different levels of stirring time when preparing and same stirring speed (2000r/min) which are shown in Figure 9. The best comprehensive humidity control ability of DBHCBM is when the stirring time is 15 minutes, the greatest adsorption content in high humidity environment is around 19% which a little bit less than the adsorption content when the stirring time is 25 minutes, but the greatest desorption content in low humidity environment is about 13.5% which is larger than other stirring times. It is clearly seen that the humidity control performance changes nonlinearly with stirring time which exist an optimal value. The drawbacks of short stirring time is the uneven mixture of raw materials and the bigger the diameter of pores of DBHCBM, whereas, the drawbacks of long stirring time is that the pore connectivity is likely to be destroyed.

4. CONCLUSIONS

The paper illustrates the relation between micro-structure and macro-structure of diatomite and studies the humidity control performance of DBHCBM at different levels of stirring time, the results show as follows:

- 1) Better humidity control performance of diatomite does not necessarily demand smaller pore size.
- 2) Humidity control performance of diatomite treated 3h calcination at 500°C is superior to the others (5h/8h), humidity control performance of diatomite treated 500°C calcination is superior to 800°C calcination.
- 3) The best comprehensive humidity control ability of DBHCBM is when the stirring time is 15 minutes, which means that there is an optimal value, too short or too long stirring time will result in unfavorable performances of DBHCBM.

ACKNOWLEDGEMENTS

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STUDY ON INDOOR THERMAL ENVIRONMENT IMPROVEMENT OF ULTRA-THIN BUILDING ENVELOPE WITH PCM

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ABSTRACT

Owing to low cost, easy to assemble and transport, tents play an important role in disaster relief and are used as Herdsman's residence in the west of China, which belong to typical ultra-thin building envelope. However, the tent provide a poor living condition, behaving hot in summer and cold in winter, as a result of low thermal resistance and weak thermal inertia. In this present study, phase change material (PCM) was applied inside of the tent, in order to improve thermal environment. Air temperature of the tents with PCM were tested and compared with that of the ordinary tent. The results show that the fluctuation and rise of air temperature in tent are controlled and reduced by using PCM, which can improve the thermal environment of tent effectively.

Keywords: Ultra-thin building envelope; Phase change material; Indoor thermal environment.

1. INTRODUCTION

Ultra-thin lightweight building envelope generally refers to that the thickness and weight are 1% -10% of those for conventional construction, in which usually lived by special populations. Owing to low thermal resistance and weak thermal inertia, tent is susceptible to solar irradiation, which is a typical ultra-thin envelope building. It leads to be hot in summer and cold in winter. This poor living condition will bring bad impact on people physically and psychologically [1]. Therefore, it is very important to research theoretically as well as practically on the thermal environment for the inner space of tents. In recent years, many researchers have done much work in order to improve the thermal environment of tents. Wang Tao, et al. [2] have tested and simulated their double-root tents which are designed and produced by themselves. It indicates that both indoor temperatures and inner surface temperature of the roof are a bit lower in double-roof tents, compared to the normal ones. The result shows obviously better insulation of double-roof tent. Based on the attached tent with aluminum foil, Hu Shaohua, et al. [3] proposed an improvement on the thermal environment in tents, using passive cooling by evaporation and got good result. According to the concept of integrated design process for green building, Zhao Xiang, et al. [4] designed a new kind of tents that are suitable for long term nomadic life. Then they proposed an important index to evaluate the effect of

natural ventilation, named ratio of clerestories area to envelopment area (RCE).

Meanwhile, phase change material (PCM) can achieve the goal for storing and exerting the energy. It has a great energy storage density and is able to keep an almost constant temperature during heat-transfer process [5-6]. As used as building material, PCM can effectively lessen the fluctuation of indoor temperature.

Hence, taken weight into consideration, it can improve the thermal comfort quality with only a little bit more weight [7]. There are several researchers having studied the characteristics of PCM, applying it into the wall, ceil and floor [8-10]. But their researches all focused on the lightweight envelopes using PCM. We have not found any research on PCM used as part of envelopes of tent. Therefore, we designed a structure by combining PCM with tent and applied it into the double-roof tents. We made a comparative experiment on the new tent model and normal tent model in Chengdu, then gave the analysis and evaluation to the effect and potential of PCM for improving the indoor thermal environment of tents. Such research could provide important experimental guidance for the control of the indoor thermal environment in tents.

2. PHASE CHANGE MATERIAL

A Natural Temperature Control Materials Energy Saver (TCM) is used as PCM in our research. Such material is easy to attach with wall. It is sealed in aluminum composite membrane and forms TCM tube. Generally, a TCM tube is 100-120g, 175mm long,

45mm wide, and 20-25mm thick. The material in the tube is non-toxic and nonflammable. Its latent heat is 130~150kJ/kg, the temperature range for phase changing is 18°C~26°C, its density is 1300~1380kg/m³ and operating temperature is 0~60°C.

3. EXPERIMENT SYSTEM

A For the purpose of effect of TCM in tents, we tested three same tent models, the tent models are the same with in structure, material. it is only scaled according to a corresponding ratio. The detailed scale is shown in Figure 1.

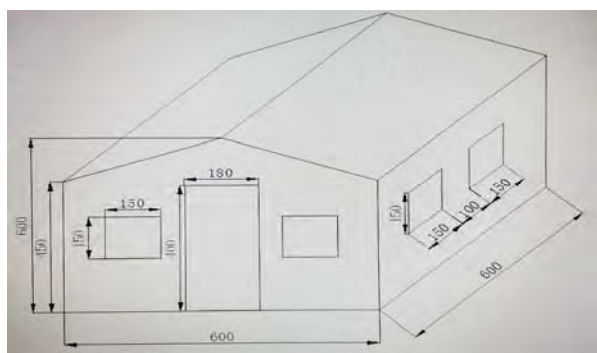


Figure 1 Schematic diagram of the structure of tent model

The test is operated in the unshaded roof of the College of Architecture and Environment building in Sichuan University. A blue double-roof tent model is applied with TCM in the roof. Another one is applied with TCM both in the roof and around envelopes. There is also a white double-roof tent model for comparison. Figure 2 shows the models.



Figure 2 The photo of tent models

The tent models are all posed with their doors facing east. Temperature test points are set at inside and outside surface of the tent models and the center of the models. Climate temperature is tested according to the requirements: no direct solar radiation, good ventilation, 1.5m high to the roof. The equipment for testing temperature is JTRG-II building thermal engineering temp meter. The indoor temperatures and inside surface temperatures are tested by T-thermocouple, its accuracy can be 0.01°C, all thermocouple are calibrated in the water thermostat system. All the statistics are input through RS232port to the computer. The temperatures were tested every 10min.

4. RESULTS AND DISCUSSION

This experiment was conducted under the operation condition of the doors and Windows are closed, the test time is on March 19th to April 19th, 2013, in order to more clearly reflect the test results.

4.1 The result of tent model applied with TCM in the roof

It is discernable from Figure3 that, when the sun radiate directly to the top of models in the morning, the surface temperature exceeds 18°C, and the TCM begins to melt and absorbs heat in order to slow down the heat transfer from the outside to the indoor environment, influencing the indoor temperature.

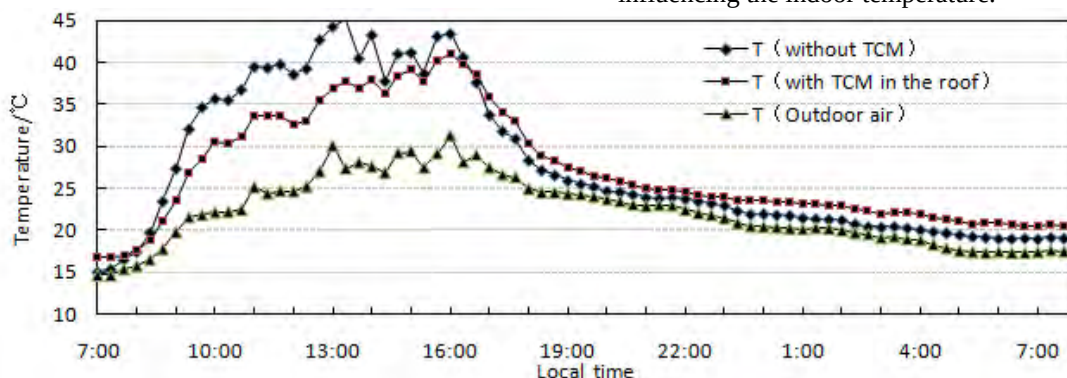


Figure 3 Indoor temperature change curve of tent models

In the noon, the indoor temperature are lower than the climate temperature by 5°C~7°C, the maximum difference is 7.4°C and emerges at 13:20. TCM are melt totally after 15:00, contributing no effect to control

indoor temperature, making the model with TCM be the same with that of normal tent model. However, due to the thermal resistance of TCM itself, the heat transferred from the inner environment are slowed

down. There is a little difference between the model with TCM and normal tent model. At dawn (4:20), the climate temperature is below 18°C and the TCM begins to solidify and release heat.

4.2 The result of tent model applied with TCM all around

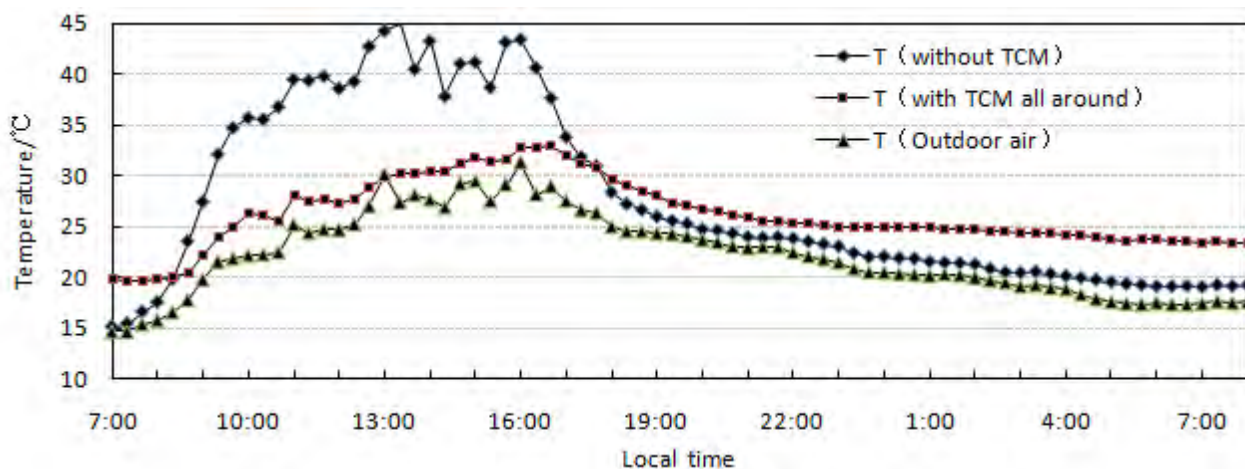


Figure 3 Indoor air temperature change curve of tent models

It is obvious from Figure 4 that the tent model with TCM around has an improved ability in absorbing and storing energy, compared to the tent model which is only set in the roof. The material begins to absorb heat after 8:20. Difference are maximum at 13:20, the maximum is 14.9°C at 4:20 of next day. When the climate temperature is below 18°C, TCM begins to solidify and release heat, inducing g the drop of the indoor temperature. The highest inner temperature is 32.9°C and appears at 16:40, which is lower than normal tent model by 12.1°C. Moreover, the highest temperature is delayed for 3 hours in the model around by TCM, compared with normal one. In addition, when the climate temperature fluctuates dramatically, normal

tent model will fluctuate with climate greatly, but the model with TCM around keeps in an almost constant condition.

4.3 Compare of total performance

Climate temperature difference between day and night in one day is 16.6°C, the max inner temperature difference of normal tent model can be 29.9°C, while that of tent model with TCM around is 13.3°C, which is reduced by 44.5%. The effect of TCM controlling temperature is obvious, seeing Table 1.

Table 1 Compare of the temperature control performance

	Highest temp (appearance time)	Lowest temp (appearance time)	Max temperature difference
Climate temp	31.3°C (16: :00)	14.7°C (7:20)	16.6°C
Normal tent	45°C (13:20)	15.1°C (7:00)	29.9°C
With TCM in top	41°C (16:00)	16.7°C (7:20)	24.3°C
With TCM all around	32.9°C (16:40)	19.6°C (7:20)	13.3°C

5. CONCLUSIONS

We propose a new structure tent, combined both TCM and double-roof. According to the experiment, we conclude that:

1) TCM can be set to produce template tube and is easy to be combined with tent. The material has potential for application.

2) The tent with TCM can reduce the indoor temperature climax, and delay the time it appears. The max temperature difference is reduced by 44.5%, compared to the normal one.

3) TCM is able to reach the phase change point during the experiment with great sunshine in the daytime. The energy it storing in day can be released at night for raising the inner temperature.

4) The experiment is conducted under the condition that all the doors and windows are closed. If the tent has natural ventilation, the convection is improved and high temperature is avoided, which is necessary for the process of solidification.

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THERMAL PERFORMANCE OF CLAY HOUSE IN THAILAND

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ABSTRACT

The extensive environmental problems (climate changes, impoverishment of resources, etc.) cause global concern for sustainable development. Clay house is a low cost house, an eco- friendly, and a low embodied energy. The objective of this research aims to investigate the thermal property of clay house in Thailand. The clay house was made from clay sand mixed with straws. The experiment was set up at Naresuan University in Phitsanulok province, Thailand. Findings from data shown that during the night time, an inside surface temperature of the house was 31.32°C which was 0.67°C higher than an outside surface temperature of the house in average; and an inside air temperature was 29.03°C which was 1.07°C. higher than an outside temperature of the house in average. During the day time, an inside surface temperature was 31.55°C which was 1.27°C lower than an outside temperature in average, and an inside air temperature was 32.42°C which was 0.62°C lower than an outside temperature in average. Data from comparison during the day shown that an inside surface temperature of the house was 8°C in different; whereas, an outside surface temperature of the house was 11.5 °C in different. During the night, a variation of surface temperature either inside or outside the house was 4.7°C in different. This study revealed that the clay house can create the inside temperature to be comfortable for living.

Keywords: Thermal Comfort; Heat Transfer; Conduction.

1. INTRODUCTION

Clay house, vernacular architecture is an ancient wisdom which developed to provide shelter and comfortable living conditions. The clay house has been well known in the world of construction for a thousand years. Thailand in the past it has been built mud house as well the evident is the local wisdom about the barn with bamboo plastered with cow dung mixed with clay. Adobe brick which made from straw and unbaked clay was easily converted back to the ground, different from construction material nowadays which have many processed and used a lot of energy to produced. However, the use of the adobe brick was decline at the end of the 19th century. Due to the abundant supply of industrial construction materials began. When the oil crisis in 1970s was began, the Clay house was returned into the mention because of in climatic regions where they had been used traditionally they provided a basic standard of thermal comfort for low rates of energy consumption. According to the United Nations an estimated 30% of the world's population still lives in homes constructed from unbaked earth [1].

Coffman et al. [2] reported that the mud-house construction have natural air conditioning effect because the rooms are cool during daytime and warm during nighttime. According to Zuni Indians [1] that the high thermal capacity of earth could store much of the heat it absorbed during the day, keeping the interior of a dwelling constructed from earth relatively cool. When

the outside temperature had dropped at night, the walls would emit the heat stored during the day into the building interior. The specific heat capacity of the adobe material is considered to be a key factor in its ability to moderate temperature swings in buildings. The application of mud as wall material was investigated to control room air temperature for buildings by Duffin and Knowles [4]. The most common passive solar building architecture comprises of massive walls to reduce the temperature fluctuations inside a building. This is known as the thermal flywheel effect as mentioned by Duffin and Greg [5].

The objective of this study is to investigate the thermal comfort and energy consumption of the clay house. Therefore, a simulation was conducted to investigate the thermal property of wall. From the derived data, it has been confirmed that clay house contribute energy benefits to the buildings.

The objectives of this study therefore are:

1. Examine the factor which effects the thermal comfort of living in clay house
2. Examine the energy transfer into the building

2. METHODOLOGY

2.1 Description of the model

The model floor plan measured 2 x 3 m. An open plan lay out was adopt to keep the cost of the house low. Adobe brick which made from straw and unbaked clay wall was used. The cement sheet board was used as

roofing material as illustrated in Figure.1. The wall was face south making it possible to direct solar radiation.



Figure 1 Experimental Model

2.2 Description of the measurement

The measurement was done from 20th September 2012 to 25th September 2012 at Naraesuan University Phitsanulok Province Thailand. The experiment was set to collect the temperature and relative humidity in different point of the model as illustrate in Figure 2.

The temperature were measured and recorded at 5 minute interval. The surface temperature measurements were recorded in the interior and exterior environments. The measuring points are illustrated in Figure 2. For all surface temperatures, thermocouple wires were placed in close contact with the relevant positions.

The instruments used for the experiment included:

- Agilent model 34970A data logger for recording temperatures & solar radiation
- Thermocouple type K for measurement of the temperature
- Kipp & Zonen pyranometer range: 1e1400 W/m²; error 5% for measurement of the solar radiation

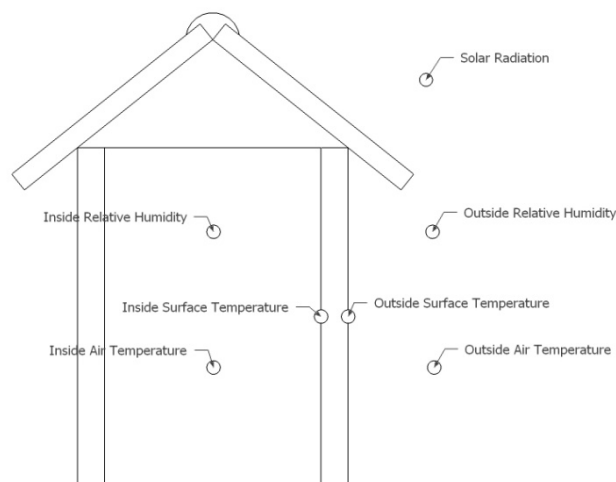


Figure 2 Measurement Point

3. Results

During the daytime the thermal behavior of the model was strongly influence by the solar radiation. During this period the outside air temperature and surface temperature was increase according to the solar radiation while the inside temperature was gradually increase.(Figure 3) From the data found that an inside surface temperature was 31.55°C which was 1.27°C lower than an outside temperature in average, and an inside air temperature was 32.42°C which was 0.62°C lower than an outside temperature in average. the day time, an inside surface temperature was 31.55°C which was 1.27°C lower than an outside temperature in average, and an inside air temperature was 32.42°C which was 0.62°C lower than an outside temperature in average. Data from comparison during the day shown that an inside surface temperature of the house was 8°C in different; whereas, an outside surface temperature of the house was 11.5 °C in different. During the night, a variation of surface temperature either inside or outside the house was 4.7°C in different. This study revealed that the clay house can create the inside temperature to be comfortable for living.

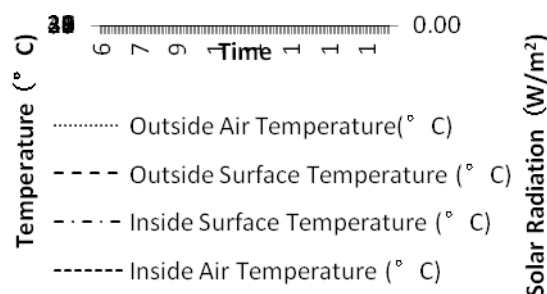


Figure 3 Temperature in different position (Day Time)

At the night time the inside temperature was higher than the outside. (Figure 3) Findings from data shown that during the night time, an inside surface temperature of the house was 31.32°C which was

0.67°C higher than an outside surface temperature of the house in average; and an inside air temperature was 29.03°C which was 1.07°C. higher than an outside temperature of the house in average.

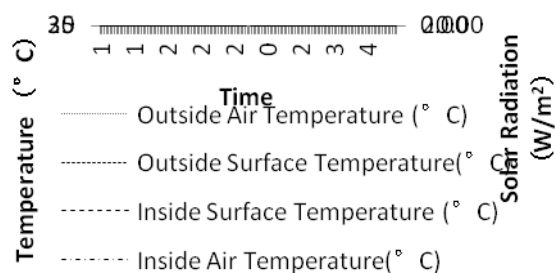


Figure 4 Temperature in different position (Night Time)

4. Discussion

The variation of the indoor and outdoor temperatures and solar radiation it imply that the indoor and outdoor temperature follows the fluctuations of the solar radiation, with the outdoor temperatures responding faster than the indoor temperature.(Figure 3) This implies that the solar intensity can be an indicator for predicting the opening and closing of window to maintain the indoor temperature within the comfort levels. The indoor and outdoor peak temperature was different can be imply that the adobe brick can localized heat transfer.

5. Conclusions

The construction of the low-cost houses can result in reduction in energy consumption. The thermal property of mud such as low thermal conductivity and high thermal heat capacity are responsible for maintaining the inside room air temperature value not much fluctuation. A time lag between the peaks and troughs of the indoor temperature was experienced in relation to the ambient temperature. Management of building component (use of doors and windows) greatly affected the thermal performance of the building. The indoor temperature was influenced by a number of factors including thermal inertia of the building and climate factors. The indoor temperature followed the outdoor temperature implying that the heating or cooling system had minimal influence on the indoor temperature

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INVESTIGATION OF SLEEP ENVIRONMENT FOR AN INFANT BY FIELD EXPERIMENT

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ABSTRACT

In January 2013, a field experiment was conducted in a room regulated by a wall-mounted air conditioner (AC), with the purpose to understand the actual state of the sleep environment of an eight-month old infant. The room was in an ordinary residential building built in 2006. During the experiment, the door and window were closed; furthermore, the window was covered with a curtain to alleviate the impact of outdoor weather on indoor environment. Temperature and velocity were measured at the air supply opening of AC and a number of positions in a vertical section that crossed the middle of the bed by around 0.4 m away from the infant head. Moreover, an infrared camera was used to scan temperature distribution around infant head. Two conditions were considered, one without the infant sleeping in the room and another with the infant inside. According to the experimental results, air was supplied at a speed of 5.4 m/s with a temperature of 49.4 °C while the temperature set of AC was 29 °C; the existence of infant made slight changes on air temperatures and velocities; surface temperature varied from 26.8-33.3 °C at infant head. Both of air infiltration and heat loss through the external wall and windows showed significant influence on indoor thermal conditions. There is a need for developing air-conditioning method or device to protect the microenvironment in the vicinity of an infant in sleeping state in a room with poor thermal insulation.

Key words: Infant; Sleep environment; Thermal comfort; Field experiment.

1. INTRODUCTION

Thermal comfort is a primary issue of indoor environmental study; however, there are few researches conducted with consideration on infants [1-4]. On the other hand, the epidemiological researches reveal that thermal discomfort might trigger the sudden unexpected death of an infant (SUDI) [5-7]. It should be an important contribution factor to prevent SUDI by keeping infants in a safe thermal environment. In the ordinary residential buildings in Central China, wall-mounted air-conditioners (AC) are the most popular and primary tool to achieve thermal comfort for people. And different from North China, usually there is no specialized heat system installed in the buildings. In order to improve the thermal comfort of infants, we firstly need to know the real thermal environment where the infant is living. Therefore, we designed and conducted a field experiment to figure out the reality of thermal status of sleep environment for an infant.

2. EXPERIMENT METHOD

The experiment was conducted on Jan, 10, 2013. It was cloudy, and the temperature was about 8 °C, the relative humidity was about 60 percent and the velocity was about 2.4 m/s.

2.1. ROOM CONDITIONS

The experiment was applied in an apartment at the fifth floor of a six-story ordinary resident building in Wuhan, which was built in 2006. The bedroom for the infant was sized 2.85 m in length, 2.55 m in width, and 2.60 m in height. Its floor plan and layout are provided in Figure 1. During the measurement, the door and window were fully closed and a curtain was used to isolate the window, with the aim of alleviating the impact of outdoor environment on indoor climate. There was a wall-mouthed AC (Midea air conditioner, KFR-26 GW/DY - J (E5)) mounted with its bottom 2.2 m above the ground on the wall right behind the bed. The temperature set of AC was 29 °C during the experiment. The building has no specialized heat system.

2.2. HUMAN SUBJECT

The subject of this experiment was an eight-month old female infant. She was 75 cm in height, and weighted 9 kg, the surface area was about 0.433 m² [8]. As the infant was sleeping in the experiment, her metabolic rate could be assumed to be 0.95 Met [9]. When the infant was sleeping, she was covered with a small and thin quilt, the total thermal insulation for the clothes (including an undergarment, a thin sweater, a

woolen trousers, a thin cotton-padded jacket, trousers) and quilt.

2.3. EXPERIMENTAL SETUP

Temperatures were measured at the air supply opening of AC as shown in Figure 2, and on the locations as shown in Figure 1, which were in a vertical section across the middle of the bed and parallel to the wall with AC. For the measurement of temperature, it was done at the locations A and G at heights of 0.10 m, 0.30 m, 0.50 m, 0.80 m, 0.95 m, 1.15 m, 1.45 m and 1.75 m; however, at the other locations, it was only done at the heights above 0.5 m. Velocities were measured at the air supply opening of AC, and at the locations H and I 0.30 m, 0.80 m, 1.15 m, 1.45 m and 1.75 m above the floor, respectively. Besides, we

measured surface temperature of the walls (①, ② and ③) at the center (the height was 1.3 m) in every rectangle wall plane, because of the window on the wall (④), we measured the temperature at the center (the height was 1.3 m) of the curtain instead of the wall, as shown in Figure 1. In addition, an infrared camera was used to get the surface temperature distribution at infant head. The details of all of the equipments were listed in Table 1.

The experiment was conducted in two conditions, one without the infant in the room and the other with the infant sleeping inside. In the second condition, the cross-section was 0.40 m away from infant head. Moreover, the measurement continued for 120 minutes for the first condition, and for 40 minutes for the second condition.

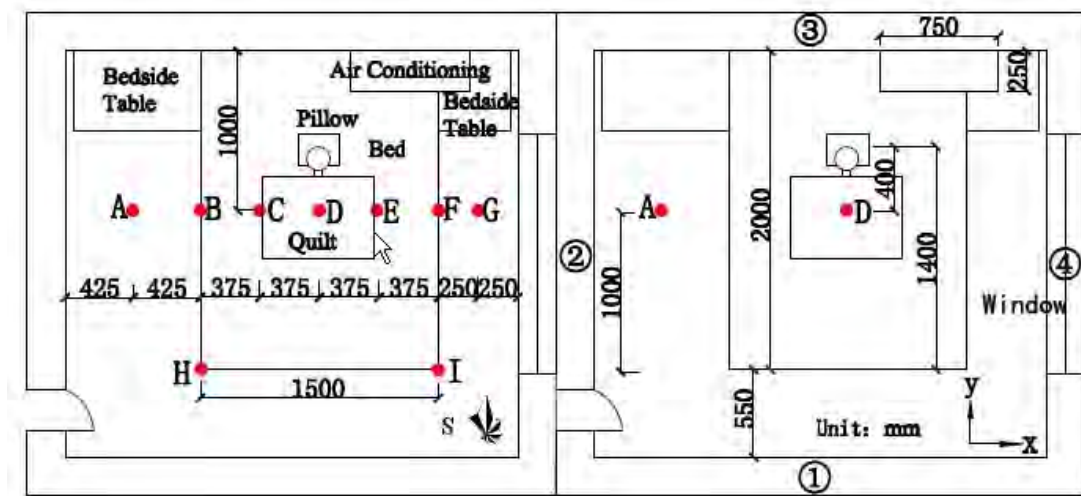


Figure 1 Outline of bedroom and measurement location



Figure 2 Measuring Points at air supply opening

Table 1 Details of experimental equipment and measuring interval

Equipment	Parameter	Accuracy	Interval
Hot wire anemometer (TES-1341)	Velocity	$\pm 3\%$	2 s
Multichannel Microanemometer (PC-2FW)	Velocity	$\pm 2\%$	10 s
iButton(DS1922L& DS1923)	Temperature	$0.065\text{ }^{\circ}\text{C}$	15 s
Infrared camera (F30W)	Temperature	$0.65\text{ }^{\circ}\text{C}$	

3. Results

The temperature and velocity presented here were the average in the experiment period.

3.1. Air supply conditions

The temperature of AC was set to be 29 °C in the experiment, and the air was supplied towards the low part of the room. We did the measurement at three positions (Figure 2), and the results were presented in Table 2. It indicated that the air was supplied at a very high temperature of around 49.4 °C and at a very high average speed of 5.4 m/s, to achieve the set temperature.

Table 2 Temperature and velocity of air supply

Parameter	①	②	③	Ave rage
Temperature/(°C)	49.3	49.5	49.3	49.4
Velocity/(m/s)	4.9	5.4	5.9	5.4

3.2. Wall temperatures

We measured temperature of the four walls at the diagonal intersection in every wall plane, the average temperature of the four walls were shown in Table 3. Temperature was lowest at the external West wall of the room, and only 2.8 °C higher than the outdoor air temperature. As the neighboring rooms were not using AC, the temperatures of the internal walls were also much lower than the indoor air temperature.

Table 3 Average temperature of the four walls

Parameter	①	②	③	④
Temperature (°C)	20.5	19.0	10.8	23.1

3.3. Velocity distribution

The results of velocities at the locations H and I were presented in Table 4. According to Table 4, the wind greatly weakened at the locations 2.00 m away

from the wall with AC. We could also see that the velocity was obviously high on the side of AC, and the existence of infant made slight changes on the velocities

at each measuring point as the locations were 1.40 m away from the infant.

3.4. Air temperature distribution

Figure 3 and Figure 4 compared the horizontal distribution of air temperature at each height level. According to Figure 3 and Figure 4, the existence of infant slightly increased air temperature at each measuring point. At the locations on the opposite side of AC (A, B, C, D), air temperature varied between 22-26 °C, and was higher in the upper room. At the locations on the side of AC (D, E, F, G), The heights of AC (2.20 m) and sill (0.7 m) indicated to be determining factors for the pattern of air temperature distribution. Under the same height, while location A, B, C and D were close to the air supply opening in turn, but the temperature were gradually fallen from A to D, due to when hot air from the air supply opening reached at location E and F, because hot air rose and cold air fell and formed a circulating air flow that in turn through A, B, C and D, besides, at the height levels below 1.15 m, air temperature gradually rose from D and reached its maximum of over 31 °C at F; then dramatically dropped at G where was close to the window, and the temperature decrease was larger at the higher height level, at the height levels above 1.15 m, the temperature were gradually rose from D to G.

Figure 5 showed the vertical distribution of air temperature at each location, which were obtained when the infant was sleeping in the room. According to Figure 5, the bed indicated a barrier for AC to transfer heat to where closed to the floor, and the cold air infiltration from the windows made air temperature as low as around 10 °C there. In the room space where was higher than the bed, vertical air temperature distribution was different on the locations depending on their spatial relationship to AC. At the locations A, B, C and D, air temperature indicated to rise with the height, having its values varied between 23-26 °C; on the locations E and F, air temperature was higher than 30 °C at the lower height levels and decreased to around 26 °C with the increase in height. On the location G, air temperature increased in proportion to the height from 11 °C to 30 °C where under 0.8 m and then jumped to around 32 °C and was kept at the temperature in the higher measuring positions

Table 4 Vertical velocity distribution at location H and I (m/s)

Height	0.3 m		0.8 m		1.15 m		1.45 m		1.75 m	
Location	Without infant	With infant	Without infant	With infant	Without infant	With infant	Without infant	With infant	Without infant	With infant
Location H	0.00	0.00	0.01	0.01	0.02	0.01	0.01	0.01	0.01	0.01
Location I	0.18	0.17	0.09	0.10	0.34	0.35	0.29	0.28	0.08	0.07

Moreover, we used an infrared camera to take some pictures to observe temperature distribution around the infant, and although the temperature distribution of the infant's head was about 26.8-33.3°C,

but the temperature distribution around the infant's head was about 20.4-26.8 °C as showed in Figure 6, Figure 6 indicated a warm thermal condition where the infant was sleeping with air temperature might be around 26 °C

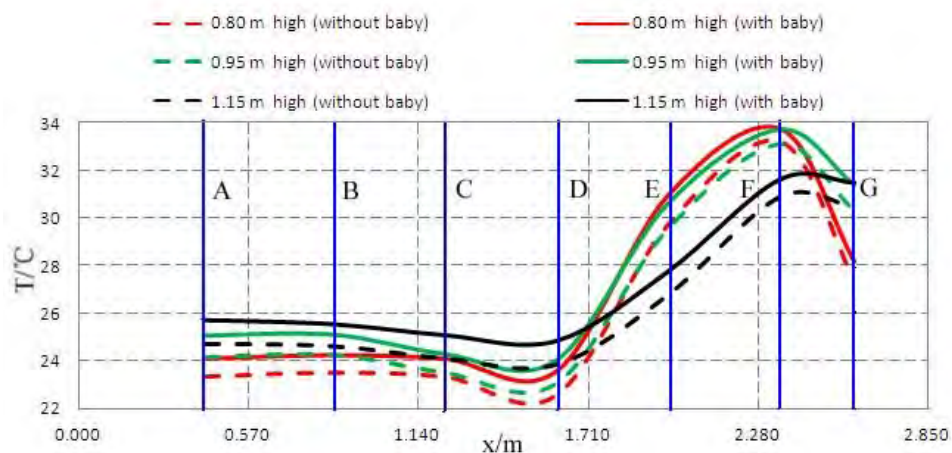


Figure 3 Horizontal distribution of air temperature at heights of 0.80 m, 0.95 m and 1.15 m

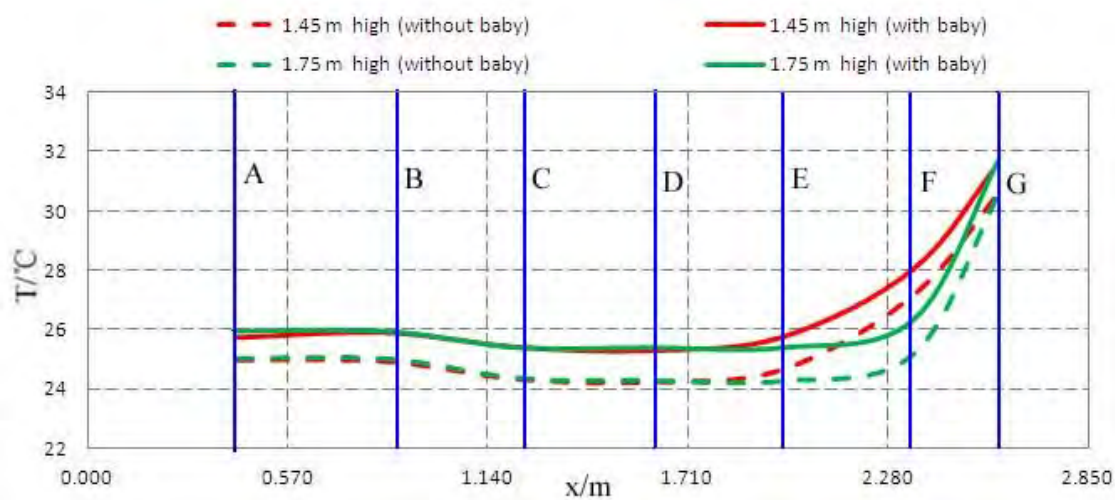


Figure 4 Horizontal distribution of air temperature at heights of 1.45 m and 1.75 m

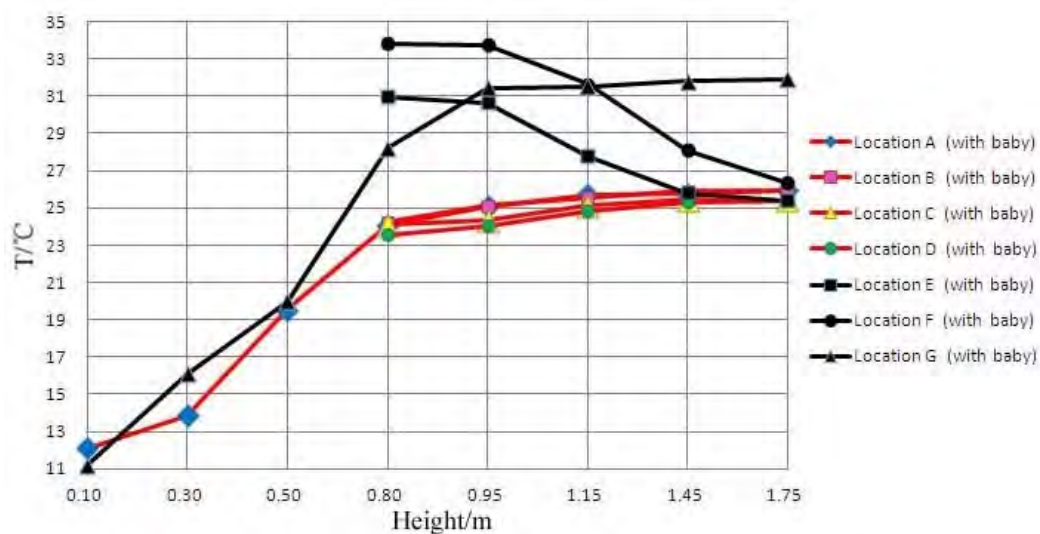


Figure 5 Vertical distribution of air temperature at location from A to G

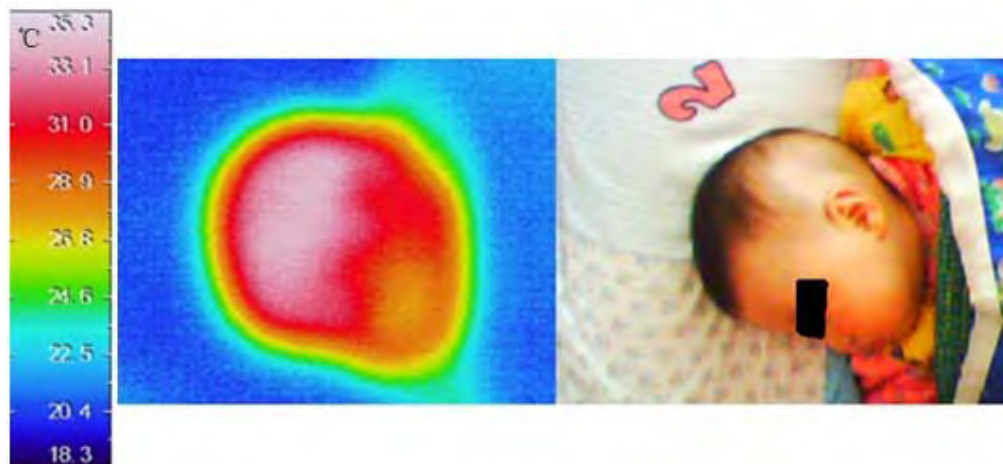


Figure 6 Surface temperature distribution around infant head

4. DISSCUSSION

This study examined thermal conditions of the sleep environment for an infant in terms of air temperature and velocity distributions in a room. Far as we know, most of the studies on thermal environment for an infant are focused on infant incubators [12-14], which is small and easy to be controlled. However, incubators are usually only for the new-born babies. We are lack of the knowledge of the real thermal conditions in the room where the infants spend most of their time. Therefore, the experiment was conducted to provide the real scenes of the sleep environment in an ordinary room environment in Wuhan in Winter. And we can see that in the experiment, AC was failed to well air-condition the room and the infant might be exposed in an environment with a little large air temperature distribution. According to the experiment results, it was mainly resulted from two reasons. First, the complicated spatial construction hindered the route for supplied air to warm up the whole room, especially the area near the floor; second, due to the poor thermal insulation, the surface temperatures of the walls were significantly affected by the outdoor climate or neighboring room temperature. Moreover, although the air filtration was not measured in the experiment, we believe that the air tightness of the room is also very poor in respect to the building envelop. The cold air infiltration would lead to an adverse effect on indoor thermal comfort.

5. CONCLUSIONS

This paper examined the real thermal conditions of the sleep environment of an infant in an ordinary residential building in Wuhan. Due to complicated spatial construction, poor thermal insulation and air tightness of the building, it indicated impossible to successfully control the indoor thermal comfort only with an AC. Therefore, it is necessary to develop the specific measures to regulate the thermal environment around the infant to ensure his/her thermal comfort, it is too difficult to well control indoor thermal environment with a wall-mounted air-conditioner which is the most

popular air-conditioning device in China; therefore, there is a need for developing individual air-conditioning method or device to protect the microenvironment in the vicinity of an infant in sleeping state.

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VOLATILE ORGANIC COMPOUNDS IN NEWLY RENOVATED HOMES AND ASSOCIATED CANCER RISK IN GUANGZHOU, CHINA: A PRELIMINARY STUDY

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ABSTRACT

Associated with rapid urbanization in China, decoration and refurbishment at homes is becoming popular which has led to elevated levels of indoor benzene, toluene, and xylenes (BTX), posing adverse health effects to occupants. In this study, BTX concentration levels were determined at homes with new renovations in urban and suburban areas in Guangzhou, China. All samples were collected using passive sampling for 24 h and analyzed by thermal desorption-GC/MS. The arithmetic average concentrations of benzene, toluene, m/p-xylene and o-xylene were 18.5 $\mu\text{g}/\text{m}^3$, 173.2 $\mu\text{g}/\text{m}^3$, 58.1 $\mu\text{g}/\text{m}^3$ and 40.8 $\mu\text{g}/\text{m}^3$ respectively. The BTX levels in this study were similar or higher than those reported before for new homes or recently renovated homes, and generally higher than those present at old homes. Higher BTX concentrations were observed at urban homes than at suburban homes, though only toluene and m/p-xylene concentrations showed significant difference ($p < 0.05$). Statistical results indicated common indoor sources (new renovations) for elevated BTX levels in these measured houses. The mean daily exposure and inhalation cancer risk of benzene were 1.1 $\mu\text{g}/\text{kg}/\text{day}$ and 6.0×10^{-5} , higher than those occurred at old homes. Monte Carlo simulation and sensitivity analysis were performed to assess uncertainty of cancer risk. The inhalation risk fell in the range from 4.61×10^{-6} to 7.21×10^{-5} . The Oral Slope Factor and the concentration distribution of benzene were the two most influential factors contributing to the variance of risk estimates.

Keywords: Indoor air quality (IAQ); Aromatic hydrocarbons; Exposure; Cancer risk; Monte Carlo simulation.

1. INTRODUCTION

Public concern regarding indoor airborne volatile organic compounds (VOCs) continues to grow worldwide, due to their widely distributed indoor emission sources, their adverse human health effect and the fact that modern people spend around 90 % of their life time in indoor (Klepeis, Nelson et al. 2001). Among these VOCs, benzene, toluene and xylenes (BTX) have been recognized as principal air pollutants in indoor. BTX were ranked the most frequently detected compounds in indoor (Wolkoff and Nielsen 2001; Zhu, Newhook et al. 2005) and their concentrations were generally higher in indoor air than in outdoor air (Kim, Harrad et al. 2001; Ohura, Amagai et al. 2009; Mo, Zhang et al. 2012). Benzene has been reported as the predominant risk compound in indoor air in comparison with formaldehyde, naphthalene etc. (Loh, Levy et al. 2007; Sarigiannis, Karakitsios et al. 2011). In addition, epidemiologic studies and case studies provide clear evidence of a causal association between exposure to indoor BTX and adverse health effects like irritation of the eyes, skin, mucous membranes and respiratory tract (Jones 1999; WHO 2010) and also suggest evidence for chronic asthma and cancer (Hulin, Caillaud et al. 2010; WHO 2010).

Exposure at homes represents a significant proportion of total exposure to BTX for population (Raw, Coward et al. 2004; Loh, Levy et al. 2007). People spend about two-thirds of the day inside their homes (Farrow, Taylor et al. 1997; Levy 1998). Indoor elevated BTX levels can result from widely existed potential indoor sources, such as building materials, decoration and renovation materials, household products, cooking and Environmental Tobacco Smoking (ETS) (Brown, Sim et al. 1994; Jones 1999; Huang, Ho et al. 2011), as well as can be permeated from outdoor air pollution (Guo, Lee et al. 2003). Loh et al. (Loh, Levy et al. 2007) ranked the cancer risks of organic air pollutants in different microenvironments in USA, finding that the exposure and associated risk occurred at home accounted for over 40 % of the total exposure and over 50 % of the total cancer risks.

Due to the economic boom in last three decades, China has experienced rapid and massive urbanization with large growth of urban population and housing area. For example, the permanent population in Guangzhou grew from 9.97 million to 12.75 million from 2006 to 2011 and the per capita housing area in Guangzhou increased from 3.97 m^2 to 21.89 m^2 from 1980 to 2011 (Guangzhou Statistics Bureau 2013). Accompanying the trend in China, decoration and refurbishment in residential buildings is becoming popular (Ohura,

Amagai et al. 2009; WHO 2010; Liu, Zhang et al. 2012). Therefore, the IAQ of newly renovated homes has become an uprising public concern. (Brown 2002) found decoration materials or new furniture in new or renovated dwellings led to much higher indoor VOC concentrations, which could persist above “baseline” levels for several weeks. And other relevant studies show that the elevated BTX levels in new or renovated dwellings were to maintain even for 2 or 3 years (Park and Ikeda 2006; Shin and Jo 2012). In China, Liu et al. (Liu, Liu et al. 2013) compared the VOCs levels at homes with renovations age less than 5 years and renovations age more than 5 years, finding that the former had nearly 2 times higher benzene level than in latter. However, the health risks associated with exposure to VOCs at homes with new renovations in China or other countries have been rarely evaluated. Therefore, the objective of this study was to characterize BTX exposure at homes with new renovations in urban and suburban areas in Guangzhou, China and to identify possible sources for indoor BTX. Also, the daily inhalation exposure and the associated cancer risk were estimated and compared with those in other cities in China and abroad.

2. METHOD

2.1 Sampling site and study

The field study was carried out in urban and suburban districts of Guangzhou, the capital city of Guangdong Province, which is one of the most rapidly developing regions in China over the last three decades. It has a permanent population of 12.7 million distributed over a 7434.4 km², and belongs to the Pearl River Delta.

Nowadays public concern about IAQ in recently renovated homes is rising. In this study, 30 and 13 homes (apartments) distributed in urban and suburban areas in Guangzhou were selected for exposure assessment based on (a) recent renovation and (b) locations with relatively high population density. All were voluntarily selected and checked for recent renovation with the help of a local company (Amway) in Guangzhou. Diffusive THPDS samplers developed in our previous study (Du, Mo et al. 2013) were used for IAQ investigation. Along with field sampling, a questionnaire was completed by the participants to collect information on apartment characteristics and potential sources of the selected analytes during the sampling period. Of the measured apartments, over 60 percent were constructed after 2000 and all have been recently renovated (i.e. decorated and/or refurnished) within two years. To evaluate the contribution of Environmental Tobacco Smoking (ETS) to BTX concentrations, ETS during smoking was recorded and 15 smoking homes were identified.

2.2 Sampling method and analysis

Air samples in selected residences were collected by passive sampling for 24 h in December 2012.

Diffusive samplers (THPDS) were mailed by Built Environment Test Center, Tsinghua University, to householders of these apartments with instruction materials. As the bedroom and living room are the most frequently occupied areas in home, the samplers were placed in the bedroom (preferred) or living room with a height of approximately 1.5 m from the floor and away from local sources of VOCs and ventilation system. After sampling, the samplers were stored in sealed aluminum bags at -4 °C before sent back to the center via mail.

The BTX samples on passive samplers were analyzed by an automated thermal desorber (Series 2 ULTRA, Markes International Ltd) interfaced with a gas chromatograph (Agilent GC-6850) and a mass spectrometer (Agilent MS-5975C). The desorber was operated in the two-stage mode; the cold trap was filled with Tenax TA 60/80 mesh. An analytical column with a structure of 30 m × 0.25 mm, 0.25 μm in thickness (Agilent 19091S-433E) was used for chromatographic separation. More details can be seen in ref (Du, Mo et al. 2013).

Quality control was implemented in the analysis of the target compounds during this study including desorption efficiency, field blanks and method detection limits (MDLs). The accuracies of this method used in this study has already been confirmed using a commonly-used active sampling method as a reference in the field test. Details can be found in the previous literature (Du, Mo et al. 2013). BTX levels that were below the MDLs or were non-detected were replaced with half of the MDLs.

2.3 Data analysis

Correlation analysis, non-parametric test and principal component analysis were performed using the SPSS software package (version 16.0, SPSS Inc., USA). Mann-Whitney-Wilcoxon test was undertaken to confirm the significant difference between two independent samples. Statistical significance was acknowledged using a *p*-value of 0.05.

Daily inhalation exposure (*E* in μg/kg/day) of an individual compound was calculated by the following equation (Feng, Wen et al. 2004; USEPA 2013):

$$E_{ij} = 0.5 \times C_{ij} \times IR \times ED_i / BW \quad (1)$$

where *C* is the measured exposure concentration for chemical *j* in the microenvironment *i*, μg/m³; *IR* the adult inhalation rate, m³/h; *ED* the exposure duration in the microenvironment *i*, h/d; *BW* an adult body weight, kg, and the coefficient of 0.5 represents the absorption factor for inhalation exposure (USEPA 2013). For adults, a mean body weight of 70 kg, a mean inhalation rate of 0.83 m³/h (Wallace 1991; USEPA 2013) and a mean residence time of 10 h were used to obtain the daily dose (Majumdar, Mukherjee et al. 2012; Liu, Liu et al. 2013).

Benzene has been recognized as carcinogen for human. The inhalation cancer risk (*ICR*) of benzene was estimated as a simple multiple of the exposure and the potency proposed by the (Wallace 1991) and (USEPA 2013):

$$ICR_{ij} = E_{ij} \times OSF_j \quad (2)$$

where OSF is the Oral Slope Factor for chemical j , $(\mu\text{g/kg/day})^{-1}$, which is derived from the Inhalation Unit Risk for chemical j . It should be noted that (USEPA 2013) provides two values of OSF for benzene, i.e. 1.5×10^{-5} and 5.5×10^{-5} .

3. RESULTS AND DISCUSSION

3.1 Concentration levels

A statistical summary of residential indoor BTX concentrations is presented in Table 1. The arithmetic average levels of benzene, toluene, m/p-xylene and o-xylene were 18.5 $\mu\text{g/m}^3$, 173.2 $\mu\text{g/m}^3$, 58.1 $\mu\text{g/m}^3$ and 40.8 $\mu\text{g/m}^3$ respectively, higher than their corresponding geometric means. A considerable variation in indoor levels was observed with big standard deviations. Clearly, there were differences between the arithmetic mean, geometric mean and median concentrations of BTX, which indicated that the concentration distributions of the selected compounds were highly skewed. Toluene was the most abundant compound, which composed about 60 % of the total compounds, followed by m/p-xylene, o-xylene and benzene.

Table 1. Descriptive statistics of BTX concentrations ($\mu\text{g/m}^3$) in 43 Guangzhou homes

	AM	GM	Median	Maximum	SD
Benzene (72.1 %)	18.5	14.0	18.8	43.7	11.6
Toluene (90.7 %)	173.2	95.8	181.0	431.0	129.5
m/p-Xylene (86.0 %)	58.1	33.3	46.0	347.7	62.8
o-Xylene (74.4 %)	40.8	25.2	33.9	54.7	39.7

Note: data in the bracket represents the percentage above the MDLs, AM: Arithmetic Mean, GM: Geometric Mean, SD: Standard Deviation

In several developed countries, indoor benzene, toluene and xylenes levels were significantly consistent with outdoor concentrations which were affected by outdoor sources, such as vehicle emission and industrial combustion (Sax, Bennett et al. 2006; Ohura, Amagai et al. 2009; Caselli, de Gennaro et al. 2010). However, in this study, the indoor BTX concentrations were noticeably higher than the ambient BTX concentrations in Guangzhou (benzene: 8.8 $\mu\text{g/m}^3$, toluene: 37.7 $\mu\text{g/m}^3$, m/p-xylene: 12.9 $\mu\text{g/m}^3$, o-xylene: 4.9 $\mu\text{g/m}^3$) reported by Tang et al. (Tang, Chan et al. 2007). It suggested that the indoor elevated BTX levels in the recently renovated residences were more contributed by indoor sources than outdoor vehicle emission and industrial sources.

Currently, the guideline levels for short-term (1 h) exposure to indoor BTX in China are 110 $\mu\text{g/m}^3$ for benzene, 200 $\mu\text{g/m}^3$ for toluene and 200 $\mu\text{g/m}^3$ for xylenes (GB/T 18883 2002). The 24 h average concentrations of BTX were lower than the limits; however, 17 and 4 out of 43 homes had toluene and xylenes concentrations above the limits. Though the exceeded ratios seem to be tolerable and the Chinese limits seem to be adequately protective for acute effects, concentrations under these limits would imply carcinogenic risks 2-3 orders of magnitude higher than the commonly acceptable risk of 10^{-6} for long-term exposure (Sarigiannis, Karakitsios et al. 2011).

Table 2 presents the residential BTX concentrations in cities at home and abroad. For new homes or recently renovated homes, the mean exposure level of benzene (18.5 $\mu\text{g/m}^3$) obtained in Guangzhou is similar to those found in Beijing (Huang, Mo et al. 2013; Liu, Liu et al. 2013). The toluene mean level in Guangzhou falls in the range of 250 to 6.9 $\mu\text{g/m}^3$ reported by (Brown 2002) for a new dwelling within 35 weeks after construction in Melbourne, but higher than that in Beijing (Liu, Liu et al. 2013). However, noticeably greater levels of xylenes were observed in this study than those in other studies. For old homes or homes without recent renovations, the average concentration of benzene occurred in Guangzhou is generally more significant than in other cities, apart from indoor benzene levels in Asan, Seoul and Kolkata, which was either notably influenced by ambient air pollutions (Son, Breyse et al. 2003) or associated with indoor fuel combustion (Majumdar, Mukherjee et al. 2012). Of the toluene and xylenes mean concentrations listed in Table 2, the values present here are the highest, though a concentration of toluene up to 522.5 $\mu\text{g/m}^3$ was detected in an urban family house in Clermont-Ferrand, France (Hulin, Caillaud et al. 2010) and a higher single value of 552.6 $\mu\text{g/m}^3$ for toluene was reported by Liu et al. (Liu, Liu et al. 2013) at a Beijing's home. In brief, the levels of BTX in this study were similar or higher than those reported before for new homes or recently renovated homes, and generally higher than those found at old homes with few exceptions. It is thus clear that indoor BTX pollution at newly renovated homes in China is very serious.

Though reasons for the elevated BTX levels in the residences of Guangzhou can't yet be identified fully, we can still see that along with the rapid urbanization, decoration or renovation materials containing BTX are frequently used in Chinese residences for renovation (Liu, Liang et al. 2009; Ohura, Amagai et al. 2009; Liu, Liu et al. 2013), though the use of BTX in solvents and adhesives was decreasing since 1970s when their adverse health effects were confirmed and thus their levels in indoor were reduced in developed countries (Weschler 2009). Another possible factor may be the different limits on indoor BTX levels set by different countries or regions. For example, the value implemented for benzene in Flanders is 2 $\mu\text{g/m}^3$, whereas the guideline in China is 110 $\mu\text{g/m}^3$. A stricter regulation may make manufacturers to reduce the amount of BTX contained in consumer products.

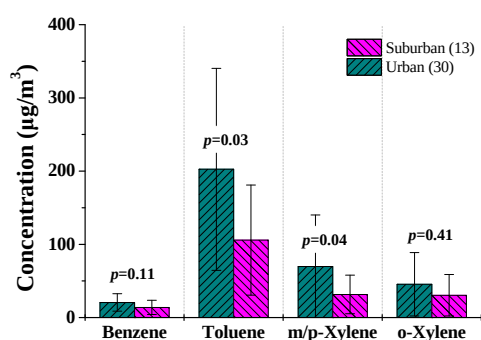


Figure 1. Comparison of BTX concentrations between urban and suburban homes (*p*-value: Mann-Whitney-Wilcoxon test)

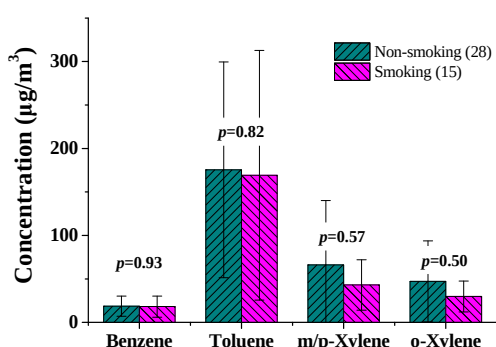


Figure 2. Comparison of BTX concentrations between non-smoking and smoking homes (*p*-value: Mann-Whitney-Wilcoxon test)

Besides, as recorded in Table 2, active and passive sampling are the most frequently methods for indoor IAQ measurements. Active sampling is more suitable for short-term exposure, while passive sampling is more popular in assessing long-term time-weighted averages of 24 h or even several weeks for air pollutants, which is more useful to reflect the picture of people's chronic exposure.

3.2 Comparison of urban and suburban levels

Fig. 1 shows BTX concentrations relationships between urban and suburban homes in Guangzhou. Results of Mann-Whitney-Wilcoxon test reveal that only the concentrations of toluene and m/p-xylene presented significant difference between urban and suburban dwellings with *p*-values lower than 0.05 while benzene and o-xylene concentrations in homes from these two districts were not significantly different as the *p*-values are 0.11 and 0.41 respectively. However, the BTX concentrations were generally higher in urban houses than in suburban houses, with the urban/suburban ratios ranging from 1.5 to 2.2, which may due to socio-economical difference, affecting the indoor consumer product use and the strength of indoor sources. Similar results were reported in previous studies. Up to two times higher BTX concentrations in

urban dwellings than in rural ones were observed by Hulin et al. (Hulin, Caillaud et al. 2010) in France. Also, Son et al. (Son, Breyse et al. 2003) compared the BTX levels in two Korean cities, Asan (a medium city) and Seoul (a metropolitan city), finding that BTX levels in Seoul were significantly higher than those in Asan.

3.3 Source identification

It is known that benzene, toluene and xylenes are associated with ETS. Herein, BTX concentrations in 15 smoking and 28 nonsmoking homes are presented in Fig. 2, with significant level obtained by Mann-Whitney-Wilcoxon test. Interestingly, nonsmoking homes were found with higher mean concentrations of BTX than in smoking homes though the nonparametric test shows no significant difference between these two samples. It clearly indicates that additional, dominant, non-ETS sources of BTX exist in domestic environments.

Correlation analysis was performed to further identify possible emission sources of indoor BTX. Relationships among residential indoor BTX levels in Guangzhou are given in Table 3. The ratio of BTX averages was 1.0/9.5/3.2/2.0 (benzene/Toluene/m/p-xylene/o-xylene). And the target aromatic compounds were significantly correlated, with correlation coefficients (*R*) ranging from 0.48 to 0.90 and *p*-values equal to or less than 0.001. The good inter-correlation of the compounds indicates common sources for indoor BTX.

However, correlation analysis alone is not sufficient to demonstrate that indoor BTX has the same sources. So principal component analysis (PCA) was used. Only one factor (eigenvalues = 3.0) was extracted by PCA analysis and it accounted for 76 % of the total variance. It infers that only one main source exists for indoor BTX, which agrees with the result derived from correlation analysis. According to the pre-mentioned results, it is concluded that new renovation including new decoration material, use of solvents and new furniture is the most possible and most important domestic source of BTX, which is prior to Environmental Tobacco Smoking.

Table 3. Correlation matrix for indoor BTX in Guangzhou homes

Correlation	1	2	3	4
Benzene	1	0.90	0.60	0.48
Toluene	$p < 0.001$	1	0.66	0.56
m/p-Xylene	$p < 0.001$	$p < 0.001$	1	0.86
o-Xylene	0.001	$p < 0.001$	$p < 0.001$	1

Note: Upper triangle area contains the correlation coefficient, while lower triangle area contains *p*-value.

3.4 Assessment of exposure and cancer risk

Here we report the estimates of daily inhalation exposure and inhalation cancer risk (ICR) of the selected compounds, calculated using Eq.1 and Eq. 2.

The higher Oral Slope Factor (OSF) for benzene was used in this study to obtain a maximum estimate of cancer risk from exposure to benzene. The mean exposures to benzene, toluene, m/p-xylene and o-xylene were 1.1 $\mu\text{g/kg/day}$, 7.7 $\mu\text{g/kg/day}$, 3.7 $\mu\text{g/kg/day}$ and 2.4 $\mu\text{g/kg/day}$, respectively. The ICR associated with exposure to benzene in the newly renovated homes in Guangzhou was 60 excess cases per million, higher than the acceptable risk level of 10^{-6} (Sarigiannis, Karakitsios et al. 2011). And the ICR of benzene in newly renovated residences from urban and suburban areas of Guangzhou showed no significant difference ($p > 0.05$).

Comparing to results in previous studies, as presented in Fig.3, it is found that the ICR of benzene observed in Guangzhou is consistent with its counterparts in Beijing for the recently renovated homes. However, much lower cancer risks related to the exposure to benzene were observed at old homes of Tianjin, England, and Ottawa (Liu, Liu et al. 2013), which is only one-third to one-sixth of that in Guangzhou. Noteworthily, homes with extra considerable indoor sources such as indoor fuel combustion could have a much greater risk of benzene (1.4×10^{-4}) (Majumdar, Mukherjee et al. 2012). The comparison highlights the great health risk level at newly renovated homes in China. And recent studies show that elevated levels of BTX in residences could be maintained for several months or even five years (Brown 2002; Park and Ikeda 2006; Liu, Liu et al. 2013), therefore the integrated risks associated to the benzene exposure exerted on inhabitants during this period should be significant and require serious consideration. In fact, estimation of the risk associated with the benzene exposure in the newly renovated dwellings should be acknowledged only as preliminary, because occupants in such dwellings are actually exposed to a mixture of air toxics which could pose much higher health risks.

3.5 Uncertainty and sensitivity analysis

High uncertainty accompanies risk assessment especially when using a single point value as input to estimate the risk for population. Therefore, given the distributions of relevant parameters in Eq.1 and 2, the Monte Carlo simulation and sensitivity analysis were implemented using Crystal Ball software. The probability distribution of risk is presented in Fig. 4. A range of inhalation cancer risk of 4.6×10^{-6} (5th percentile) and 7.2×10^{-5} (95th percentile) with the mean and median values equal to 2.7×10^{-5} and 2.0×10^{-5} respectively was obtained. Clearly, the mean risk obtained by the Monte Carlo simulation is lower than that aforementioned, however, it is still over 20 times higher than the acceptable level, suggesting that benzene pollution in indoor air is a serious threat to human health. Lognormal distribution is the best model fitting the frequency distribution of the estimated cancer risk associated with benzene, which is similar to results obtained by Zhou et al. (Zhou, You et al. 2011).

The result of sensitivity analysis showed that the OSF and the benzene concentration distribution were the two most influential variables which contributed approximately 90 % of the total variance of risk. The uncertainties related to the OSF affects the risk assessment, because it is derived from the cancer potency factor which is obtained by a linear extrapolation from the high-dose animal or human studies to the low doses of environmental exposure using either a maximum likelihood estimate or the upper 95 % confidence limit on the dose response function (Loh, Levy et al. 2007; USEPA 2013). And uncertainties exist in the low-dose exposure scenarios and a clear understanding of the mode of action is not achieved yet. Also, great uncertainties were produced when the benzene concentration distribution was generated using limited field samples. In addition, BTX measurement procedures such as passive sampling, analytes desorption and instrument analysis, may also lead to the final uncertainty.

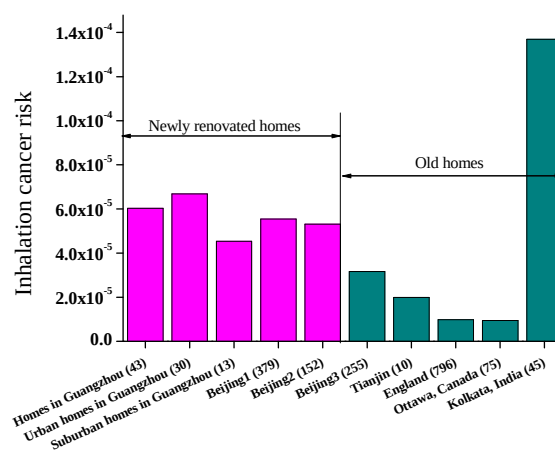


Figure 3. Inhalation cancer risks of benzene at home from different countries

Note: Beijing1 from (Huang, Mo et al. 2013), Beijing 2&3 from (Liu, Liu et al. 2013), Tianjin from (Zhou, You et al. 2011), England from (Raw, Coward et al. 2004), Ottawa, Canada from (Zhu, Newhook et al. 2005), Kolkata, India from (Majumdar, Mukherjee et al. 2012). Data in the bracket means the number of sampling homes or sites.

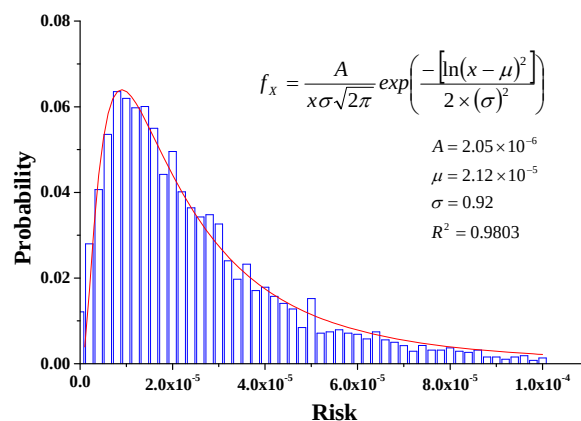


Figure 4. Estimate of inhalation cancer risk of benzene by the Monte Carlo simulation (5000 iterations)

Note: Lognormal distribution for benzene concentration (location: $4.2 \mu\text{g}/\text{m}^3$, mean: $18.5 \mu\text{g}/\text{m}^3$, Std. Dev.: $11.6 \mu\text{g}/\text{m}^3$), Normal distribution for exposure duration (Mean: 10 h/d, Std. Dev.: 2 h/d), inhalation rate per hour ($0.83 \text{ m}^3/\text{h}$, $0.2 \text{ m}^3/\text{h}$), and body weight (70 kg, 10 kg); triangle distribution for Oral Slope Factor (minimum: 0, likeliest: $1.5 \times 10^{-5} (\mu\text{g}/\text{kg}/\text{d})^{-1}$, maximum: $5.5 \times 10^{-5} (\mu\text{g}/\text{kg}/\text{d})^{-1}$).

4.0 CONCLUSION

In this study, we measured the 24 h average concentration levels of prevailing volatile organic compounds (benzene, toluene, and xylenes) at homes in Guangzhou, China. Of the BTX, toluene was the richest compound, in the descending order of m/p-xylene, o-xylene and benzene. The BTX levels in this study were similar or higher than those reported before in Beijing for new homes or recently renovated homes, and generally higher than those observed at old homes with few exceptions. And the BTX concentrations at urban homes were generally higher than those in suburban ones in Guangzhou, though significant difference was only confirmed for toluene and m/p-xylene concentrations. Statistics analysis demonstrated that the selected compounds were significantly correlated and only one factor was extracted by PCA, which suggested new renovations including new decoration material, use of solvents and new furniture were the dominant domestic source of BTX prior to Environmental Tobacco Smoking in these investigate houses.

The daily exposure and the inhalation cancer risk of benzene were estimated based on the methodology proposed by the USEPA. The benzene exposure ($1.1 \mu\text{g}/\text{kg}/\text{day}$) and associated risk (6.0×10^{-5}) occurred in Guangzhou showed no significant difference from those reported in Beijing for new homes or recently renovated homes, however, they were generally higher than those at old homes. Based on results of Monte Carlo simulation, the risk from exposure to benzene was 1–2 orders of magnitude higher than the acceptable risk level of 10^{-6} . Some precautions should be taken to increase the accuracy of the risk assessment such as increasing the sizes of samples and improving of the sampling method.

More efforts are needed to assess the cancer risks from exposure to a mixture of air contaminants and meanwhile urgent measures for reduction of indoor air pollution in newly renovated residences.

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Table 2. Comparison with other studies on BTX measurement in dwellings

Studies	Locations	Sampling methods	Remarks	Concentrations ($\mu\text{g}/\text{m}^3$)			
				Benzene	Toluene	m/p-Xylene	o-Xylene
Current	Guangzhou, China	Passive sampling for 24 h, December 2012	43 homes	18.5 ^a /11.6 ^b	173.2 ^a /129.5 ^b	58.1 ^a /62.8 ^b	40.8 ^a /39.7 ^b
<i>New homes or homes with new renovations</i>							
Huang et al. (Huang, Mo et al. 2013)	Beijing, China	Active sampling for 45 min, July 2008 - September 2012	379 homes newly remodeled within 1 year	17 ^a /16 ^b	-	-	-
Liu et al. (Liu, Liu et al. 2013)	Beijing, China	Active sampling for 30 min, November - December 2009	152 homes recently renovated within 5 years	16.3 ^a /14.7 ^b	45.2 ^a /24.7 ^b	12.3 ^a /5.3 ^b	
(Brown 2002)	Melbourne, Australia	Active sampling for 30 – 50 min	A new dwelling (living room)	~ 30 – 2.2 ^a (2 – 246 days)	250 – 6.9 ^a (2 – 246 days)	30 – 2.8 ^a (2 – 246 days)	-
(Shin and Jo 2012)	Daegu, Korea	Active sampling for 3 h	25 new houses	-	94 – 27 ^a (1 – 24 weeks)	20 – 2.8 ^a (1 – 24 weeks)	8.7 – 2.3 ^a (1 – 24 weeks)
<i>Old homes or homes without new renovation</i>							
Lee et al. (Lee, Li et al. 2002)	Hong Kong, China	Canister sampling for 8 h	Living room of 6 homes	4.7 ^a /0.5 ^b	52.1 ^a /8.4 ^b	3.9 ^a /1.2 ^b	4.5 ^a /0.4 ^b
Guo et al. (Guo, Kwok et al. 2009)	Hong Kong, China	Passive sampling for 24 h, November 2002	Living room of 100 homes	-	15.3 ^a /45.4 ^b	-	3.0 ^a /2.3 ^b
Zhou et al. (Zhou, You et al. 2011)	Tianjin, China	Passive sampling for 5 days, May 2008	Living room of 10 homes	6.1 ^a /7.6 ^b	7.5 ^a /4.2 ^b	1.6 ^a /1.6 ^b	0.5 ^a /0.6 ^b
Liu et al. (Liu, Liu et al. 2013)	Beijing, China	Active sampling for 30 min, November - December 2009	255 homes with renovations > 5 years	9.7 ^a /5.7 ^b	26.4 ^a /13.2 ^b	11.5 ^a /3.4 ^b	
Ohura et al. (Ohura, Amagai et al. 2009)	Hangzhou, China	Active sampling for 30 min, August in 2006, January in 2007	Bedroom of 14 homes	8.5 ^{GM}	26.7 ^{GM}	11.9 ^{GM}	4.6 ^{GM}
	Shizuoka, Japan	Passive sampling for 24 h, August - September in 2006, January - March in 2007	Bedroom of 57 homes	1.0 ^{GM}	7.2 ^{GM}	2.0 ^{GM}	0.8 ^{GM}
Kim et al. (Kim, Harrad et al. 2001)	Birmingham, England	Active sampling for 2 h	12 homes	13.9 ^a /13.8 ^b	38.4 ^a /21.7 ^b	7.2 ^a	1.9 ^a /1.3 ^b
Son et al. (Son, Breysse et al. 2003)	Asan, Korea	Passive sampling for 24 h, August 2001	Living room of 30 homes	20.3 ^a /12.6 ^b	17.3 ^a /13.1 ^b	10.4 ^a /7.5 ^b	9.3 ^a /6.7 ^b
	Seoul, Korea			43.7 ^a /80.1 ^b	170.6 ^a /183.3 ^b	27.5 ^a /65.0 ^b	33.4 ^a /72.2 ^b
Raw et al. (Raw, Coward et al. 2004)	England	Passive sampling for 4 weeks	Bedroom of 796 homes	3.0 ^{GM}	15.1 ^{GM}	3.8 ^{GM}	-
Zhu et al. (Zhu, Newhook et al. 2005)	Ottawa, Canada	Active sampling for 100 min, November 2002 - March 2003	Living room of 75 homes	2.9 ^a	11.5 ^a	7.5 ^a	5.1 ^a
Majumdar et al. (Majumdar, Mukherjee et al. 2012)	Kolkata, India	Active sampling for 3 - 4 h, March - June 2005	Living room of 45 homes	42.0 ^a /33.2 ^b	69.3 ^a /41.4 ^b	52.1 ^a /31.1 ^b	21.6 ^a /10.6 ^b

TRANSPORTATION CARBON EMISSIONS ASSESSMENT OF LOW-CARBON ECO-PARK: A CASE STUDY IN PINGDI, SHENZHEN

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ABSTRACT

Urban carbon emission has sharply increased owing to the rapid urbanization and motorization. Since transportation is a main source of carbon emission, it is urgent to build a low-carbon transport mode with characteristics of environment-friendly, high efficiency, low energy cost and safe. With the IPCC's methods and spatial analysis, transportation carbon emission of Pingdi International Low-carbon Eco-park in Shenzhen was evaluated. Road vectorization was operated on ArcGIS10.0, while traffic distribution based on the dual-constrained gravity model was operated on TransCAD. Furthermore, by studying the structure of residents travel and the carbon emission per unit distance of each motor vehicle, the traffic carbon emission of the study area was evaluated. The total CO₂ emission was 127127.57 tons, and the average emission intensity was 1056 tons/km per year. The transportation emission structure shows that private cars, official cars and taxi account for the largest proportion of the emission, each for 42.9%, 18.2% and 16.5%, while the proportions of bus and motorcycles were less than 10%. The spatial distribution shows that the largest intensity appears in roads of heavy traffic, such as Huiyan Express and Yanlong Avenue. On the contrary, districts of more slip roads have low carbon emission intensity. Influencing factors of transportation carbon emissions, including land use structure, structure of travel, transport vehicle, traffic network and low-carbon consciousness, were deeply analyzed in order to provide some references for developing the transportation mode dominated by public transport and the implementation of energy saving policy.

Keywords: Low-Carbon Eco-park; Transportation; Carbon Emission; Emission Structure; Spatial Analysis.

1. INTRODUCTION

Urban traffic is the third major greenhouse gas emission sector following energy supply and industrial production, accounting for 13.1% of the global emission (IPCC, 2007). In the past dozens of years, along with the development of social and economic activities, urban space was gradually expanded and vehicle constantly increasing, the growth rate of traffic carbon emission is higher than any other sector (MA Jing et al, 2011).

In order to reduce the carbon emission, developed countries have made the emission reduction plans. Special attention is paid to the traffic sector to reduce the emissions, by improving traffic technology, adjusting traffic structure and stressing the public traffic. Britain puts forward the concept of low-carbon economy for the first, whose Traffic Department issued Low-carbon Traffic Innovation Strategies in 2007 to put forward the practical technology of road, air, railway and water transport and the study direction of the future low-carbon traffic technology. Study on Low-carbon Social Mode and Feasibility of Japan proposes to transform the urban traffic structure to the traffic mode oriented by the public traffic by elevating the land use rate and strengthening the urban function (SU

Chengyuan et al, 2012). Singapore released the White Paper of Traffic Development-Build the First-class Land Traffic System in 1996 to determine to keep balance between cars and public traffic and draft strict measures about the traffic demand management (ZHANG Liang et al, 2011). Traffic Planning, the third generation of urban traffic planning developed by France in 1990s, expands the target to slow traffic and focuses on the complementary integration of walking, non-motorized vehicle, car and public traffic, and transits from the pursuit of the technical solution of traffic system to overall organization and the quality of urban mobility service (ZHUO Jian et al, 2010).

Along with the significant growth of traffic carbon emission, studies on the carbon accounting, emission structure and the construction of low-carbon traffic also gradually increase. International studies on the traffic carbon emission mainly focus on two aspects: one analyzes the existing traffic carbon emission reduction plan to discuss the low-carbon traffic mode of the city and the macro regulation strategy of the traffic, while the other quantitatively analyzes the characteristic of the traffic carbon emission structure and spatial distribution. International quantitative studies are quite mature and some institutions have transformed the theoretical study to the practical planning evaluation software. For example, INDEX (U.S.), I-PLACE3S

(U.S.) and Tool for Evaluating Neighborhood Sustainability (Canada) (GAO Yang et al, 2012) evaluates the traffic carbon emission of the study area by establishing the relation model between automobile mileage and urban characteristic (e.g. mixed land utilization extent, accessibility of public service facility, etc.).

Domestic studies on traffic carbon emission mainly focus on the emission structure, influencing factors and quantitative distribution characteristic. Existing studies indicate that private cars account for a quite large proportion of CO₂ emission in the urban traffic (SU Chengyuan et al, 2012). Urban population and vehicle capacity significantly influences the urban traffic carbon emission by influencing the volume of passenger and cargo transport, and the proportion of bus negatively influences the urban traffic carbon emission (SU Taoyong et al, 2011). Influenced by the long-distance travel of the rural residents, the average traffic carbon emission per household gradually increases from downtown to suburbs (GAO Yang et al, 2012).

Currently, most studies of the traffic carbon emission focus on the whole city or the comparison of several cities but few focuses on the park. Eco-park is an experiment of low-carbon eco-city and an important part of the low-carbon urban traffic. Studies on the traffic carbon emission of the park are helpful in decomposing influencing factors of the urban traffic carbon emission in the meso perspective and putting forward corresponding strategies. Therefore, this paper regards Shenzhen Pingdi International Low-carbon Eco-park as the object, constructs the meso-scale carbon emission computation formula, and conducts spatial processing of the park. Taking the structure of residents travel and the carbon emission per unit distance of each motor vehicle into account, this study accounts the total volume, structure and spatial distribution of the traffic carbon emission of the park, and analyzes influencing factors, by which theoretical basis for developing the transit oriented development (TOD) mode and the implementation of energy saving policy was supplied.

2. METHODS AND CALCULATION

2.1 Methods

In traditional studies, the traffic carbon emission is mainly measured by the emission factor method of *National Guidelines for Greenhouse Gas Emission of IPCC 2006*, i.e. it is achieved by the fuel consumption of each vehicle multiplied with the emission factor of each kind of fuel. This method applies to the area with large space (e.g. city) but is quite limited in the data availability in the park. Besides, this method may only obtain the regional carbon emission but may not study the spatial characteristic of the traffic carbon emission.

Therefore, for the purpose of computing the traffic carbon emission of the park/city with small space and analyzing the distribution of the carbon emission space, this study computes the carbon emission per unit distance of each vehicle as the emission factor with reference to the empirical value of existing studies.

2.1.1 Accounting Model

$$E_{CO_2 \cdot day} = \sum_{j=1}^n \sum_{i=bus}^{subway} Q_j \times P_{ij} \times L_j \times EF_i$$

Where:

$E_{CO_2 \cdot day}$ -Traffic carbon emission, unit: kgCO₂/day;

j -Code of road section independently measured in the space;

i -Motorized travel mode (including bus, taxi, private car, official car, motorcycle, truck and subway);

Q_j -Total traffic flow of road j , unit: person/day;

P_{ij} -Share rate of travel mode i on road j , unit: %

(see 2.2);

L_j -Length of road j , unit: km;

EF_i -Emission per unit distance of travel mode i , unit: kgCO₂/ (km·person-time), see Table 1.

Annual carbon emission of the traffic department is computed by the daily traffic carbon emission (unit: ton CO₂/year):

$$E_{CO_2 \cdot a} = E_{CO_2 \cdot day} \times 365 / 1000$$

2.1.2 Flow Computation Model

Traffic distribution by dual-constraint gravity model: determine the travel distance between traffic zones by Multiple Shortest Path of TransCAD, take the distances as the impedance matrix, and determine the inverse power function as the impedance function. Model of the flow between zones is:

$$T_{ij} = P_i \times \frac{A_j \times f(d_{ij})}{\sum_z P_z \times f(d_{iz})} \quad (\text{Constraint of travel flow})$$

$$T_{ij} = A_j \times \frac{P_i \times f(d_{ij})}{\sum_z P_z \times f(d_{iz})} \quad (\text{Constraint of attraction flow})$$

Where:

T_{ij} -Estimated flow attracted to zone j produced by zone i ;

P_i -Estimated travel flow produced by zone i ;

A_j -Estimated travel flow attracted to zone j ;

d_{ij} -Resistance from zone i to zone j ;

$f(d_{ij})$ -Fraction factor between zone i and zone j .

Table 1 Carbon (CO₂) Emission Factor per Unit Distance of each Vehicle

Unit: kgCO ₂ / (km·person-time) or kg CO ₂ / (km·ton)	
Motor Vehicle Type	Emission factors
Bus	0.066
Taxi	0.276
Private Car	0.239
Official Car	0.239
Motorcycle	0.082
Truck	0.141

Source: Heidelberg Research Institute: *Chinese Traffic: Energy Consumption and Emission of Different*

Transportation Modes (with reference to CO₂ emission of per unit distance per capita of Shanghai in actual loading 2005¹).

¹ Carbon emission per unit distance per capita is reduced when the load rate is higher by car sharing.

2.2 Calculation

Currently, there is no subway in the study area. This study mainly considers the contribution of the road traffic within the park to the traffic carbon emission, and motor vehicles in this study mainly refer to the bus, taxi, private car, official car, motorcycle and truck (See Figure 1).

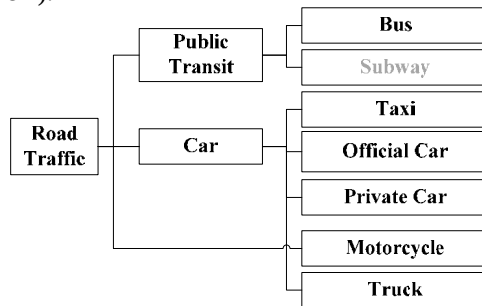


Figure 1 Target of Evaluation of Traffic Carbon Emission

2.2.1 Vectorization of Road Network

Traffic road network is mainly obtained by vectorization in ArcGIS, according to the aerial image of Pingdi and the traffic survey of Pingdi. Traffic data mainly comprising of road network and nodes is used to establish the topological relation of the vectorized traffic data generates the complete road network data.

2.2.2 Travel Demand Computation

Import the road network data established in ArcGIS to TransCAD. The road network of Pingdi is divided into 483 sections and classified by the road level as shown in Figure2. Every road section is numbered and attached with the detailed information.



Figure 2 Topological Map of Road Traffic Network of Pingdi

According to experience, the travel time per capita of a city is quite stable. In view of the survey data of Longgang District and various streets, it is estimated that the current motorized travel rate per capita is 1.4 person time/day. In 2011, the total population of Pingdi is about 175 thousands, so the motorized traffic flow is estimated to be 245,000 person-time/day.

2.2.3 Traffic Zones Division

Traffic zones are mainly divided according to principles followed:

a. Land use, economic characteristic and social characteristic within zones shall be consistent with each other;

b. The administrative area shall be kept complete as much as possible to utilize the existing statistical material of the administrative area;

c. Mass center of the zone is the node in the road network;

d. Quantity of zones shall be appropriate that it shall neither increase the load of planning nor reduce the analysis accuracy;

e. Principle of uniformity and gradual increase outwards. Traffic zones within the study area shall be kept with proper uniformity in acreage, population and traffic attraction. Due to the lowered requirement on accuracy, traffic zones outside the study area shall be gradually enlarged as the distance from the study area is becoming farther.

According to the land use property, 154 traffic zones are divided in Pingdi, as shown in Figure 3, where blue areas are traffic zones.. These traffic zones are basically surrounded by main and subsidiary roads, and some important branch roads.



Figure 3 Traffic Zone Map

2.2.4 Freight Volume

Computation of freight volume: The demand for internal freight volume of Pingdi is 5,000pcu/day and the demand for external freight volume is 14,000 pcu/day, which mainly flows to Pinghu, Henggang, Pingshan, Dongguan and Huizhou. Transit freight volume is 21,000 pcu/day, which is mainly the freight volume between downtown and eastern Pearl River Delta and Yantian Port.

2.2.5 Traffic Flow Distribution

The traffic flow for each road is estimated according to the gravity model, in combination with current traffic survey data. The traffic flow is distributed in TransCAD by the Stochastic User Equilibrium (SUE) method. Freight volume, bus traffic flow and motor vehicle traffic flow are respectively distributed and the traffic flow of each road section is added to obtain the motorized travel distribution of Pingdi.

2.2.6 Motorized Travel Structure

Current travel modes of Pingdi's residents mainly include walking, bicycle, bus, taxi, private car, official car and others (mainly motorcycle). According to the travel situation, roads in Pingdi are not completely covered by the bus routes, thus the motorized travel structure of Pingdi shall be respectively distributed according to the road with bus and without bus (Table 2).

Table 2 Motorized Travel Structure of Residents of Pingdi (Unit: %)

Traffic Mode	Road with bus	Road without bus
Regular Bus	34	0
Taxi	10	16
Private Car	30	48
Official Car	16	26
Motorcycle	10	10

2.2.7 Spatial Display of Data

Traffic carbon emission result of Pingdi computed by the traffic carbon emission model was imported to ArcGIS. Maps of traffic carbon emission and emission intensity of Pingdi were established by map symbol system of ArcGIS.

3. RESULTS

3.1. Traffic Carbon Emission Assessment

3.1.1 Amount and Structure of Traffic Carbon Emission

According to the computation, traffic departments of Pingdi altogether emit **127,127.57** tons of CO₂. Carbon emission structure of traffic departments is shown in Figure 4, from which it can be seen that the private car, official car and taxi account for the largest proportion of emission, occupying 42.9%, 18.2% and 16.5% respectively; the bus and motorcycle account for relatively small proportion of emission, occupying less than 10% of the traffic carbon emission.

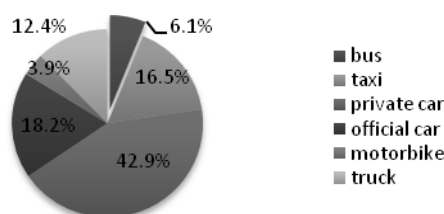


Figure 4 Carbon Emission Structure of Traffic Departments

3.1.2 Comparison of Traffic Carbon Emission Structure

Studies differ from each other in the motor vehicle structure when accounting the traffic carbon emission, as shown in Figure5. It can be seen from the picture that the proportion of the private car's carbon emission of Pingdi is less than that of Shanghai and Xi'an, but the carbon emission of the official car and taxi of Pingdi is 4.6% and 12.8% higher than that of Shanghai and Xi'an respectively. On the contrary, the carbon emission of bus is 9.1% and 6.61% lower than that of Shanghai and Xi'an respectively. It indicates that Pingdi shall take measures as soon as possible to adjust the travel mode of residents to elevate the ridership of the public traffic and optimize the traffic carbon emission structure.

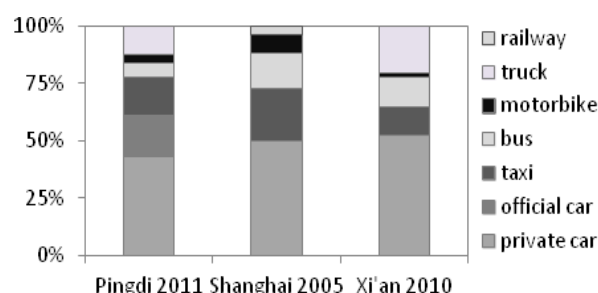


Figure5. Comparison of Pingdi, Shanghai and Xi'an in Carbon Emission Structure

Source: Shanghai-Heidelberg Energy and Environment Research Institute, 2008; Xi'an-Study on Urban Traffic Carbon Emission Basing on Scenario Analysis, 2012.

3.1.3 Spatial Distribution

Spatial distribution of the traffic carbon emission of Pingdi is shown in Figure6. Sections with large-quantity carbon emission include Huiyan Highway, Yanlong Avenue and Shenhui Road while the carbon emission of branch roads is relatively low.

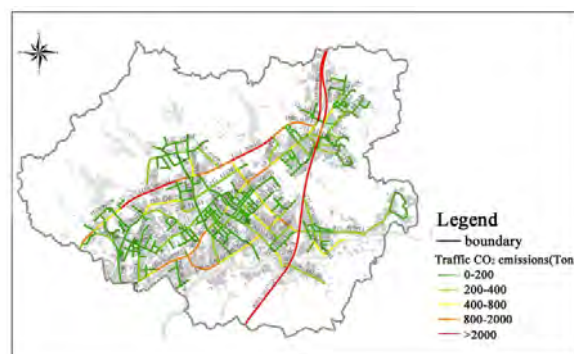


Figure6. Spatial Distribution of Traffic Carbon Emission of Pingdi

Spatial distribution of the traffic carbon emission intensity is shown in Figure 7. Sections with intensified carbon emission include Huiyan Highway followed by Yanlong Avenue and Shenhui Road, while the branch roads' traffic carbon emission intensity is relatively low. It can be seen that carbon emission is positively related with the traffic flow to a certain extent, and the spatial distribution of carbon emission is basically consistent with the spatial distribution of traffic flow.

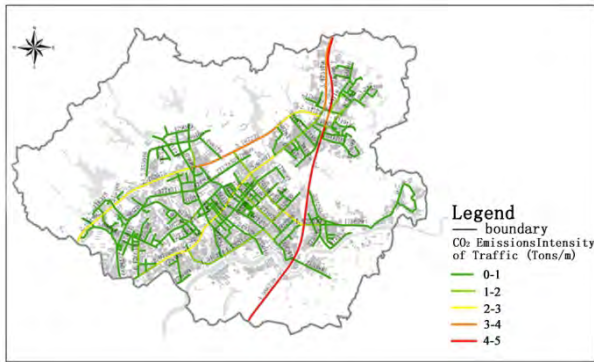


Figure7. Spatial Distribution of Traffic Carbon Emission Intensity of Pingdi

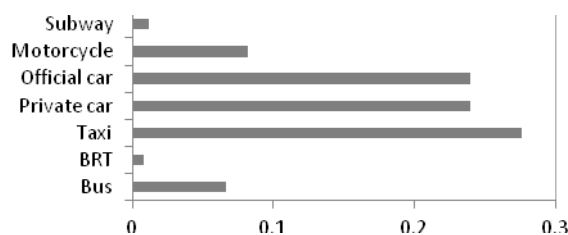
3.2. Influencing Factors

3.2.1 Land Use Structure

Land use structure may influence the traffic demand, spatial distribution and travel distance, among which the traffic demand and travel distance are critical factors for the traffic carbon emission. Reasonable arrangement of urban land, adjusting regional proportion of commerce and residence, mixed development and transformation may meet the residents' demand for various services within the short distance, which can effectively reduce the demand for motorized travel and realize the carbon emission reduction of the traffic department.

3.2.2 Motorized Travel Structure

Emission factor per unit distance of every motor vehicle differs from each other largely (Figure8): carbon emission factor per unit distance of the car is relatively high, while that of the BRT, metro and bus is relatively low. For example, the carbon emission factor per unit distance per capita of BRT is less than 3% of that of the taxi. When the travel flow and travel distance are kept unchanged, adjustment of the travel structure may influence the traffic carbon emission. For example, in Pingdi, the carbon emission per unit distance of the road without bus is 0.227 kg/ (day•km) while that of the road with bus is only 0.1473kg/ (day•km), 64.9% of the former one. It can be seen that it may effective reduce the traffic carbon emission by increasing the proportion of vehicles with low emission factors, such as bus, metro and BRT.



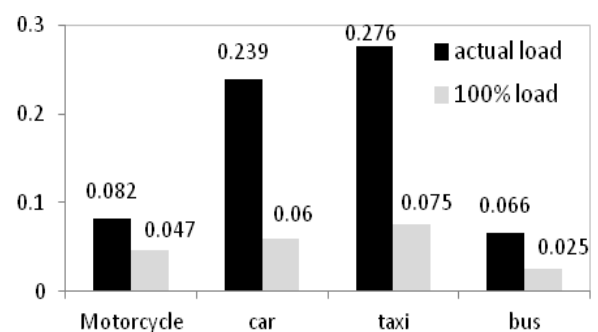
Unit: kg CO₂/ (person•km)

Figure8. Carbon emission factor per unit distance per capita of each motorized travel mode

3.2.3 Traffic Road Network

The carbon emission intensity of the highway and expressway is quite high, for they share the external transportation of Pingdi. The emission intensity of the park mainly comes from the main road, followed by the subsidiary road and branch road. As for the total emissions, the highway, main road and branch road emit much carbon while the expressway and subsidiary road emit less CO₂. The carbon emission intensity of the branch road is relatively low but the total emission is large. It's because the branch road network is quite dense, and it connects the subsidiary road and the residential area. Sometimes the roads take on large traffic flow and the vehicle speed is slow, so that their carbon emission is indirectly increased.

As for the bus network, other than the highway and expressway, number of bus routes in Pingdi is basically positively correlated to the traffic flow of the road, but there is larger improvement space: firstly, only 43.2% roads of Pingdi are covered by the bus route while some roads with large traffic flow has no bus; secondly, some sections covered by the bus route are insufficient in transportation that bus routes shall be increased to share the traffic pressure; On the contrary, some traffic routes are too concentrated. For example, there are 17 bus routes on Shenhui Road, much more than 1.5-2.5² recommended by the *Traffic Engineering Manual*. It can be inferred from Figure 9 that when the load rate of bus is 100%, the carbon emission per unit distance per capita may be reduced to 37.9% of the former carbon emission. As a result, it shall according to the residents' travel demand on different roads to arrange the bus route reasonably to enlarge the coverage and reduce the repetition rate of the bus route, so as to elevate the load rate and utilize the bus to the maximum extent.



Unit: kg CO₂/ (person•km)

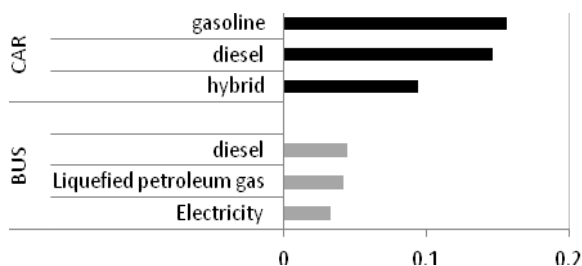
Figure9. Carbon emission per unit distance per capita under different load rates (Heidelberg Energy and Environment Research Institute, 2008)

3.2.4 Energy Consumption Structure

Energies differ from each other in the carbon emission per unit. It can be inferred from Figure 10 that

²Refer to Comprehensive Traffic Improvement Plan of Longcheng, Longgang and Pingdi-Pingdi Subdistrict.

the CO₂ emission factor of the hybrid automobile is 39.7% lower than that of the gasoline automobile, and the CO₂ emission factor of the electric bus is 27% lower than that of the diesel bus. As a result, the promotion of the application of the low-carbon vehicle may effectively reduce the traffic carbon emission.



Unit: kg CO₂/(person•km)

Figure10. Carbon (CO₂) emission factor per unit distance of various motor vehicles (QIU Lixun et al, 2008)

3.2.5 Green Travel Consciousness

Other than improving the land use structure and traffic network and forming green traffic network, the residents' consciousness of green travel is critical for the traffic carbon emission reduction. For example, short-distance travel shall choose zero-carbon ways like walking and bicycle, the travel of 1 to 3 persons shall choose the low-carbon ways like bus and metro, so as to effectively reduce the traffic carbon emission within the region.

4. CONCLUSIONS AND DISCUSSIONS

This paper constructs the park-scale computation model of the carbon emission to analysis the traffic carbon emission of Pingdi low-carbon eco-park of Shenzhen. Traffic flow distribution analysis of road is conducted in ArcGIS and TransCAD, and the traffic carbon emission structure and spatial distribution characteristic of Pingdi eco-park is established. Furthermore, influential factors were analyzed. Result indicates:

(1) The private car accounts for the largest proportion of carbon emission in the park, followed by the official car, while bus and motorcycle account for relatively small proportion.

(2) Spatial distribution of the traffic carbon emission is basically consistent with spatial distribution of the traffic flow. Huiyan Highway and Yanlong Avenue with large traffic flows emit much CO₂.

(3) Important factors influencing the traffic carbon emission include land use structure, travel structure, traffic network, traffic vehicle and residents' travel consciousness.

According to the analysis above and the actual situation of traffic carbon emission of Pingdi, it is suggested to take following measures to reduce the carbon emission of traffic departments:

a. Optimize the land use structure and adjust the road network: Construct the mixed land to avoid

separated commerce and residence to reduce long-distance travel. Optimize road network structure and increase subsidiary roads to connect the main road and branch road to reasonably arrange the traffic flow. Construct the slow traffic system like bicycle lane and green lane for the convenience of the residents' zero-carbon travel instead of motorized travel.

b. Optimize the travel structure: Reasonably arrange the official car, and avoid private use of the official car. Reduce the utilization of the private car and encourage the travel by public transit. Improve the public traffic system to elevate the proportion of bus, metro and BRT in the travel structure to optimize the travel structure and reduce the intensity and flow of carbon emission of each motorized road section.

c. Use the green vehicle: Optimize the automotive power, strengthen the automotive technology and choose the clean-energy vehicle to reduce the carbon emission of the motor vehicle and enlarge the application of the clean-energy bus.

d. Strengthen the green travel consciousness: Encourage and publicize the green travel and reduce the demand for the motor vehicle. Meanwhile, strengthen the green travel consciousness by campus education and publicity to guarantee the green and low-carbon traffic.

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EFFECT OF BUILDING PLAN ON WIND ENVIRONMENT AROUND BUILDINGS OF RESIDENTIAL DISTRICT

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ABSTRACT

Numerical simulation was applied to analysis the characteristic of outdoor airflow of residential district. With the index of air age, ventilation efficiency of street canyons of different building plans was discussed. Research indicates that keeping the buildings height and length constant, the radius of counter-clockwise airflow swirl in street canyon will increase along with the increase of the spacing between buildings, which is helpful to reduce pollution and heat accumulation, then the air quality is improved. Under the circumstance of the same ratio of height and length of the buildings, the area-averaged air age of the windward face and the leeward face of lee buildings increased with the length of buildings. When the height of building is fixed, the reducing of space and length of street canyons is be conducive to removing the pollution and enhancing heat dissipation. Appropriate building plan is helpful to improve the airflow in street canyon, which is contribute to strengthen transpiration of green land, then the temperature of air around will decrease.

Keywords: Numerical simulation; Building plan; Building height; Building length; Building space; Air age; Airflow.

1. INTRODUCTION

It is one of the gigantic challenges of the urbanization that the rapidly expansion of department requirement which is caused by the rapidly increasing of population. Accordingly, building microclimate has attracted great attention as a major branch of the city environment; as the major part of the building microclimate, the wind environment of residential district has close relationship with the comfort of residents, diffusion of pollutant, air quality and so on. There are several major factors that affect the wind environment [1-2], namely building plan, wind angle and blocking effect of underlying surface vegetation. The factors of building plan mainly include the orientation, structure and density of buildings, space between buildings. The wind environment of residential district affects not only air quality and the distribution of pollutant within the residential district but also the thermal environment [3-9]. Most common building plan as street canyons are, airflow movement within them directly influence the convective diffusion of canyon pollutants and quality of heat [12-15]. The research concerning characteristics within street canyon will contribute to understand the distribution of the pollutants systematically.

2. WIND ENVIRONMENT WITHIN STREET CANYON

Setting coming flow air to satisfy the model of wind velocity gradient, namely the air velocity is different on different location:

$$V_h = V_0 \left(\frac{h}{h_0} \right)^n \quad (1)$$

Where, $V_{h=10} = 4.8 \text{ m/s}$, $n = 0.22$.

Following the principle of control variate method, the effects of outdoor fluid field caused by solar radiation and building wall heater transfer is put aside. The schematic of building size of street canyons as follows.

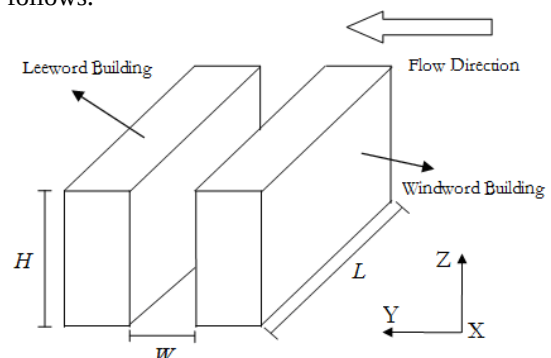


Figure 1 the schematic of street canyons

NOMENCLATURE

Abbreviation		Subscripts	
AWLP	The APEC Workshop on Low-carbon Town and Physical Energy Storage	-0	reference altitude
Symbols		Greek Letters	
V	Air Speed (m/s)	ρ	Density (kg/m ³)
V _h	Air Speed on Height h(m/s)		
V _{h0}	Air Speed on Height h ₀ (m/s)		
h	Height(m)	γ_s	Shortwave reflectivity
h ₀	Reference Height(m)	ε_L	Long wave emissivity
n	Exponent	λ	heat transfer coefficient(W/m • K)
H	Height of Building(m)	C _p	specific heat at constant pressure(kJ/ m ³ • K)
W	Width Between Buildings(m)		
L	length of Building(m)		

2.1 Boundary condition

Entrance boundary condition was set as entrance of gradient wind and the coming flow wind direction is the south. Taking the reference wind velocity as Equ.1. Pressure outlet boundary was set, and the roof and the both sides of the entrance were set as translucent wall. The building height was fixed as H, the building space W, the building length L.

Table 1 The structure size of building plans

Type	Length L	Space W	Height H	The ratio of height and width H/W
plan1	30m	16m	48m	1.5
plan2	30m	53m	48m	0.8
plan3	30m	96m	48m	0.5
plan4	160m	16m	48m	1.5
plan5	160m	53m	48m	0.8
plan6	160m	96m	48m	0.5

2.2 Air flow distribution within street canyon

2.2.1 The air flow distribution within street canyons of different building space

Being affected by blunt body effect [9-11], when flow air dashed the windward face vertically, the air is resisted with the reducing of dynamic pressure and the increasing of static pressure, which main form three parts of air flow. The air flow of part I will accelerate

form jets to go forward on both sides of the corner of the building. The air flow of part II will circle building top and continue to go forward, which form over flow. Because of the change of flow direction caused by impact, the air flow of part III will become two U shaped and flow to the sides of building, then the air flow go forward to the negative pressure area of leeward to increased pressure of leeward. Finally, the three parts of air flow will converge again and form tail swirl on the tail negative pressure area of building. As shown in Fig 2, most of air flow fly past the top of canyons directly and form a steady counter-clockwise swirl which caused by airflow shear force. The airflow around the leeward face of windward building is rising, while the windward face of leeward is falling. Contrast fig. (b) and (c), we can find that the air average velocity in the bottom of street canyons ranges from 1.59m/s to 1.45m/s when the building space ranges from 16m to 53m, and that the whirl center slips to the right, and that the radius of drive diminishes. With the building space range from 16m to 96m, the above phenomenon is more obvious. The radius of whirl within street canyons diminishes obviously, which weakens the ability of rising airflow to remove the pollution and heat accumulation in the canyons bottom. Meanwhile, with the increasing of building space, the radius of tail swirl and the space between swirl center and leeward face will increase. Through the above comparative analysis, keeping the buildings height and length constant, the radius of counter-clockwise airflow swirl within street canyons will increase along with the space between buildings, which is helpful to reduce pollution and heat accumulation, then the air quality is improved.

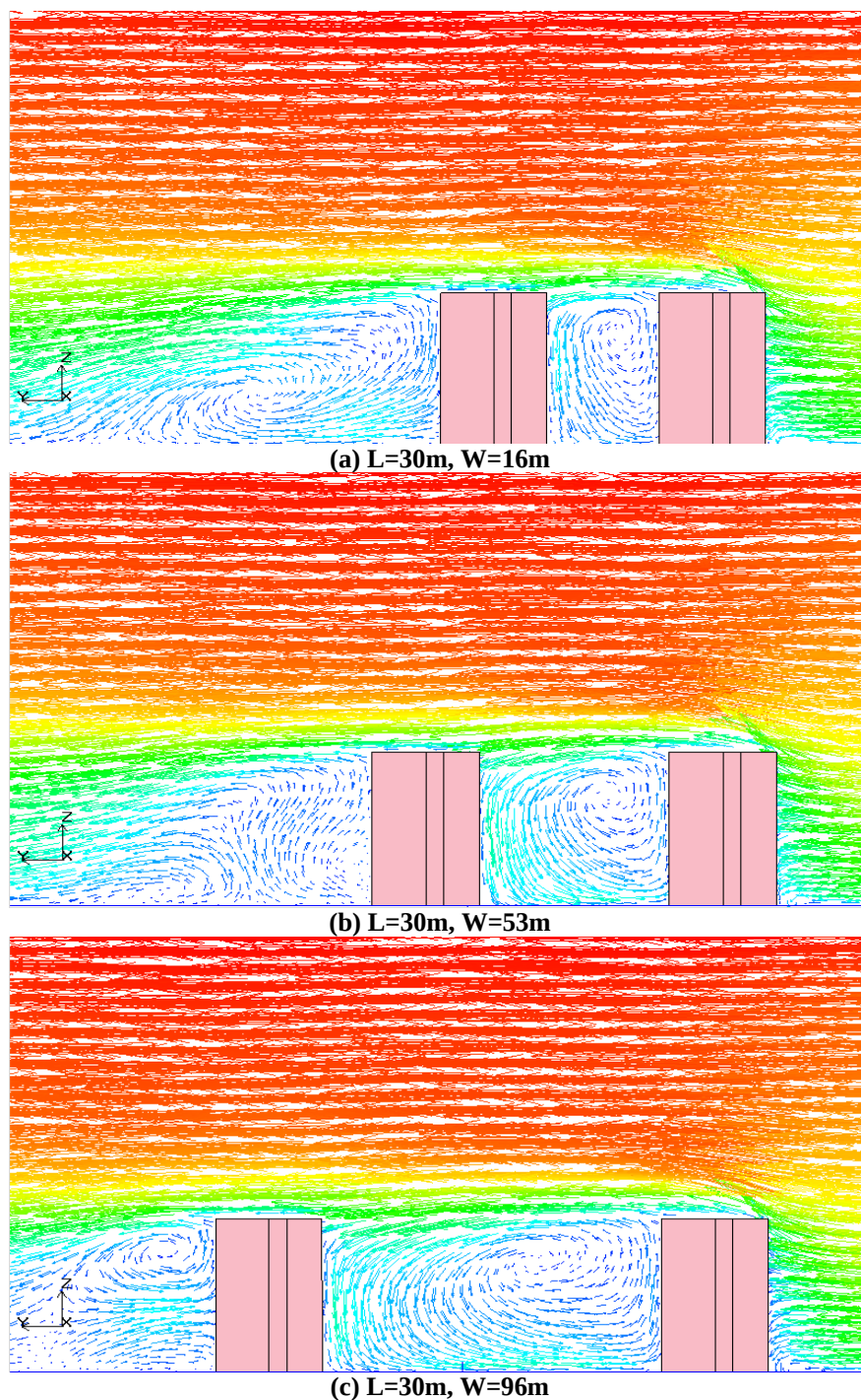
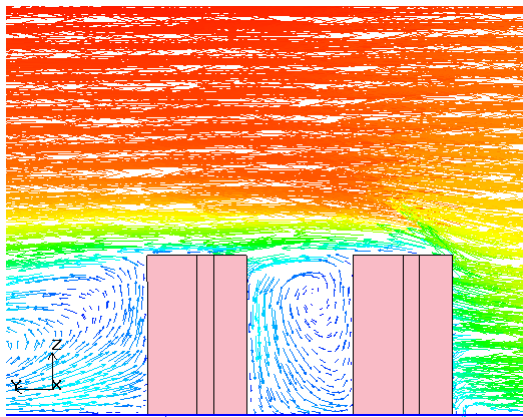


Figure 2 The vertical air flow distribution within street canyons of different space between buildings

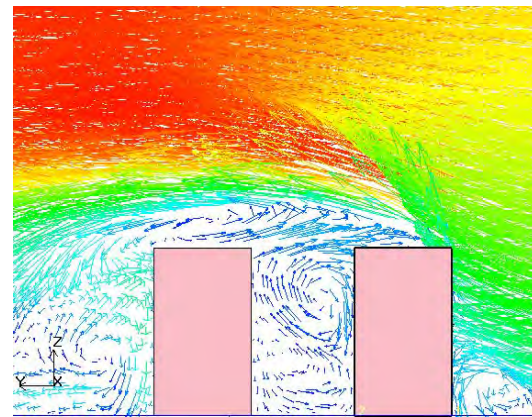
2.2.2 The vertical air flow distribution within street canyons of different building length

As shown in Fig 3, when the height and space of building is fixed and the length of building ranges from 30m to 160m, the single counter-clock swirl transforms the vertical double swirl. Closing to the top of canyons is counter-clock swirl which birth a clock swirl in the bottom of the canyons. The clock swirl in the bottom of

canyons is almost completely resisted by the counter-clock swirl. The air flow velocity of bottom are all blow 0.5m/s and hardly exchange with top air, which is harmful to ejectment pollution and heat accumulation, then the wind environment is deteriorate. Because of the increasing of area of windward face in fig 3 (b), there are more air to hit the wall, and the flow direction is changed, forming a apparent swirl in the bottom of the building, which will affect the ventilate of low-rise homes of windward.



(a) $L=30m$



(b) $L=160m$

Figure3 The vertical air flow distribution within street canyons of different length of buildings, with W being 16m.

In conclusion, when the height H of building is fixed, the reducing of space H and length L of street canyons is be conducive to removing the pollution and enhancing heat dissipation, then the quality of air is improving effectively within street canyons.

needed for air from into the room to leave the room. In this paper, air age is used to evaluate the ventilation efficiency within street canyons of different building plan.

2.3 Air age distribution within street canyon

As one of the evaluation index of quality of indoor air and dissipation of pollutant, air age refers to the time

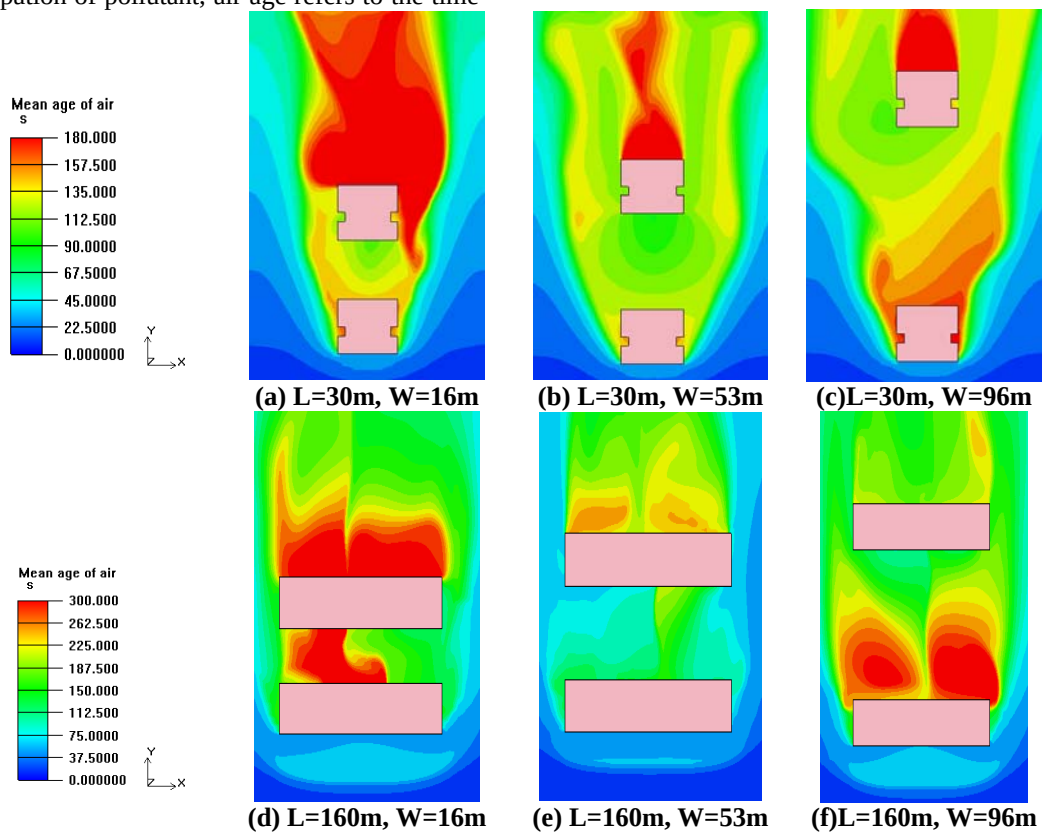


Figure 4 The distribution of air age at $Z=1.5m$ horizontal plan

As shown in fig 4, the distribution of air age is similar to the distribution of swirl. Compared with other

regions; the air age of horizontal double swirl area in the wind shadow area and canyons is relatively high. Under

the circumstance of the same length of the buildings, the tail swirl of the leeward building and air age reduced with the increasing of building space. While under the circumstance of the same space of the buildings, the air age within the street canyon increase with the increasing of the building length and its maximum increases by the 180s to the 300s, the maximum velocity attenuation by 3.25m/s to 2.64m/s. The reason for the above phenomenon is the airflow in the corner of building accelerated when the airflow hit the windward of the building and the intermediate portion forming the graze flow bypassing the top of the construction. When the length of the building increases, the intersection angle between the high-speed jet of the building on both sides is increasing, making the time of the collision of the two air flow in the tail negative pressure reflux area extend, which leads to the swirl area near the leeward face of the building increases. Under the circumstance of the length of building, the distance from the horseshoe formed in the both sides of windward building to the negative pressure reflux area rear the building of leeward is reducing, thereby contributing to the formed of tail canyon, and thus the age of air increase.

When the length of building L is 30m, corresponding to the building space turn to 16m, 53m and 96m, the area-averaged air age within the street canyons turn to 122s, 108s and 142s; When the length of building L is 160m, corresponding to the building space turn to 16m, 53m and 96m, the area-averaged air age within the street canyons turn to 186s, 125s and 258s. The fig 5 shows, the area-averaged air age will increase with the increasing of building length, and the maximum increase is 81.7% when the building space is 16m, while the minimum is 15.7% when space is 53m.

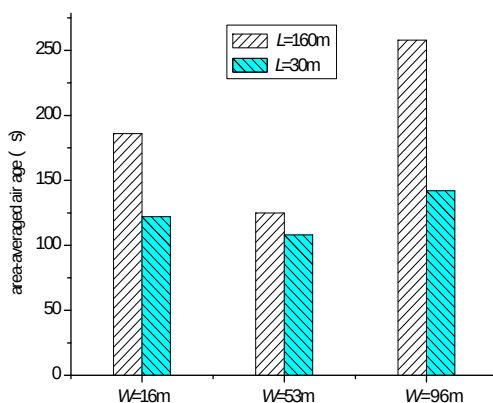


Figure 5 The average value of air age at Z=1.5m horizontal plan

The distribution of air age within street canyons under the different building plan was contrasted, the found indicate that Under the circumstance of the same ratio of height and width of the buildings, the area-averaged air age of the windward face and the leeward face of lee buildings increased with the length of buildings, which is same as the air age of leeward face of leeward buildings. When building width is 53 m, the

building length affect the minimum to air age within the street canyon.

3. OUTDOOR ENVIRONMENT OF RESIDENTIAL DISTRICT

The temperature, and wind speed of representative position in residential district was measured at 9:00 am, 12:00 and 15:00 pm on June 1st, 2012. The number of buildings in residential district and measurement points are shown in Fig 1, namely, the entrance road (Point 1-7), typical semi-enclosed buildings (Point 8-10), the middle of the residential district of the street canyon (Point 11-14), the windward and leeward of buildings and roof (Point 15-17), the shelter of buildings (Point 18), and the green belt (Point 19). Measurement points are 1.5m higher than local ground.

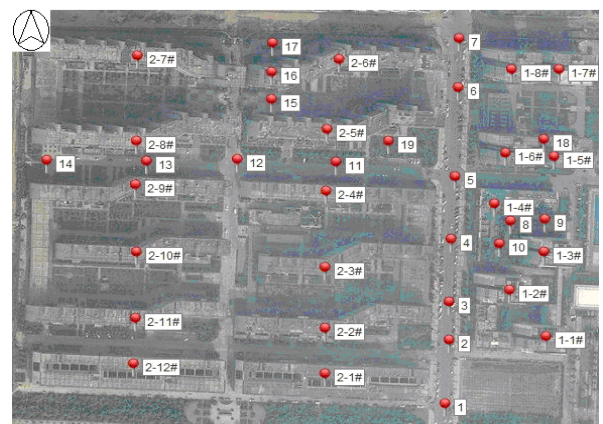


Figure 6 Plan of residential district and test points distribution

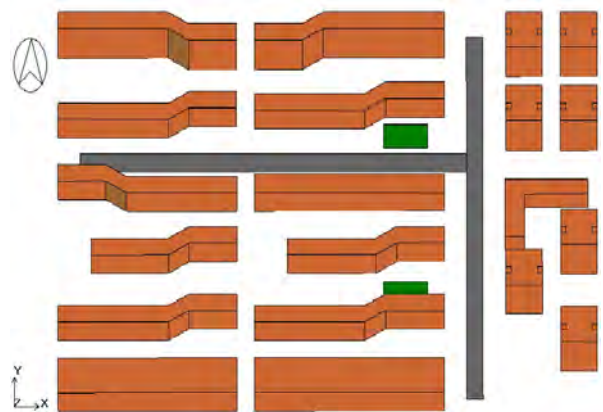


Figure 7 Three-dimensional calculation model of residential building

As shown in fig 6, the physical model of residential district was set in Airpak3.0. The calculation area size of the physical model is X, Y and Z: 1500m, 1200m and 150m, namely 3 times, 5 times and 3times of the relative actual size, which ensures the fully developed requirement of the boundary of the calculation area. The numerical calculation model of residential district is established with the following simplification [16-18]: (1) to simplify the building appearance, the principle of uniform appearance is used and ignored the tiny area of convex and concave ;(2) to simplify the underlying

surface, roads of scatter arrangement and green belt is simplified into two east-west and north-south asphalt road and two typical rectangular green belt. The initial velocity of wind from south is set 2.2m/s at the height of 10m. The top and the both sides of calculation domain is

set as transparent wall with the transparency 0.8. The initial air temperature was set as the actual average temperature of the area that height 1.5m than ground, namely 33.8℃.

Table 2 Physical parameters of underlying surface in residential district [19, 20]

building	material	Short wave reflectivity	Long wave emissivity	Heat transfer coefficient ($W/(m^2 \cdot K)$)	Specific heat at constant pressure ($kJ/(m^3 \cdot K)$)
Wall	Plaster wall	0.20	0.90	7.73	1900
Roof	Concrete roof	0.12	0.90	7.73	2000
Ground	Gravel	0.24	0.95	5.28	2100
Road	Pitch	0.05	0.95	5.28	2000

Table 3 Parameters of green belt [21].

Greening types	Model size (m^3)	Short wave reflectivity	Long wave emissivity	Short wave dissipation coefficient	Long wave dissipation coefficient
Shrub and grass	40×20×0.6	0.25	0.98	0.6	0.8

3.1 Comparison of experimental data and simulation result

As shown in fig 8, the difference of simulation results and measurement results of most point wind speed are below 0.5 m/s, and the absolute error between the measurement wind velocity and simulation velocity of the

point in residential is within the allowable range. By the contrast of simulation results and measurement results of residential outdoor microclimate, CFD simulation can get relatively close to the real distribution of air flow and temperature fields of residential environment thus can contribute to the plan and construct of residential

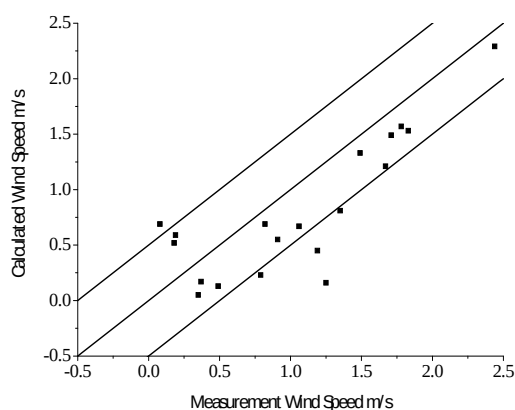


Figure8 Comparison of measured value and simulation value of wind speed

As shown in fig 9, the points 1-5 have a better entrance effect and the overall level of wind velocity is slighter higher than the other points. Because the prevail wind direction is south on that day, the east-west wind is shielded by the building, causing the wind speed within the street canyons is lower. So the measurement value of

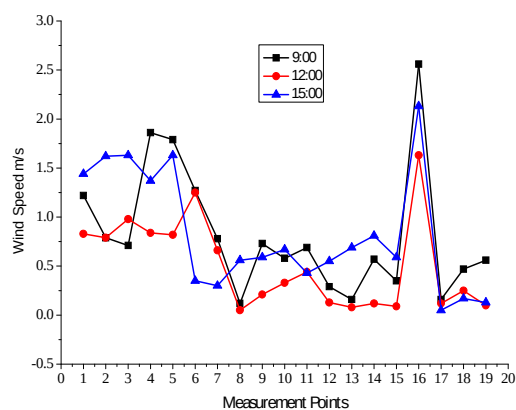


Figure 9 Measured values of wind speed at different time

point 11-14 is lower. The average wind speed of point 16 in roof is highest. The speed reach 2.56m/s in 9:00.

3.2 Analysis of velocity field

For the reason that the wind shadow area is increasing with the length of windward buildings, the turbulent energy of air flow losses with the bluff-body effect [22, 23]. Because of the narrow width between left side buildings, the wind at the corner of buildings along the left road is blocked, resulting in an average velocity of 0.68m/s. Meanwhile, the relatively larger amount of air flow on the entrance road, with a relatively larger

width of 53m, is combined with the jet on the left of building 2-1# and the high-speed flow at the left corner of building 1-2#. Consequently, the mean wind speed on the entrance road reaches 1.42m/s, which has an increment of 52% compared with the air flow on the left road. Since the buildings at the corner of the district block the air flow, most buildings at the downstream of the air flow are in the wind shadow area, leading to dead angle of the flow.

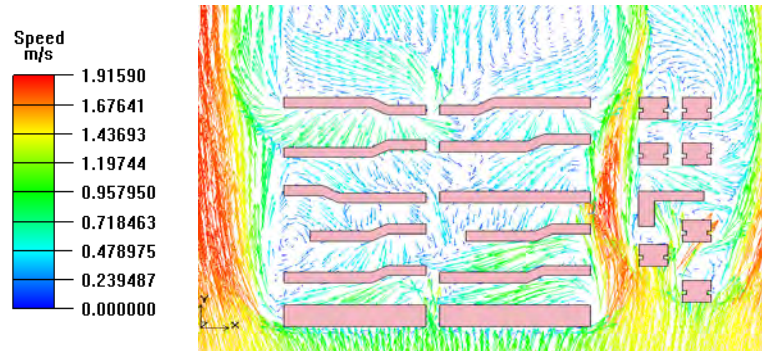
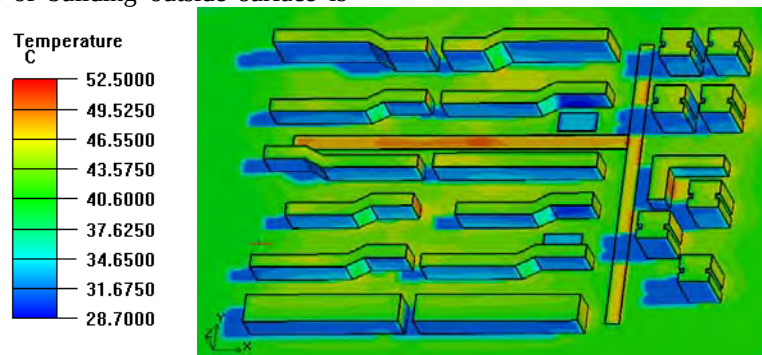


Figure 10 Vector diagram of wind speed at Z=1.5m horizontal plane

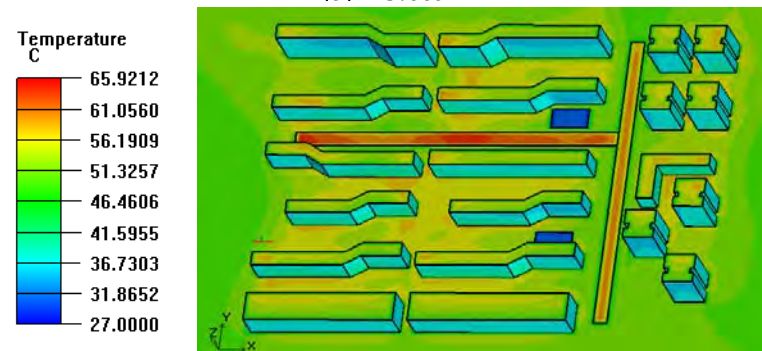
3.3 Analysis of temperature field

Because the results of numerical simulation are greatly influenced by boundary conditions and the temperature distribution of building outside surface is

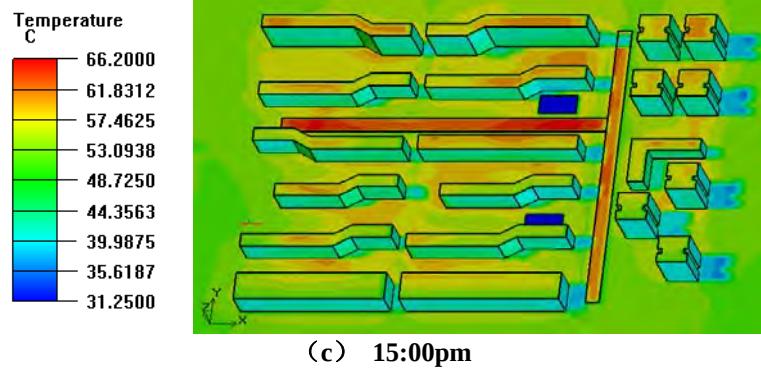
closely related to the summer air conditioning energy consumption, studying the distribution of surface temperature of residential at 9:00 am, 12:00 and 15:00 pm will contribute to the comprehensive understanding of the outdoor thermal environment to the residential.



(a) 9:00am



(b) 12:00am



(c) 15:00pm
Figure 11 Temperature field of surfaces of residential district at 9:00, 12:00, 15:00

As shown in fig 11, the outdoor temperature distribution on the surface of the ground and building is represented respectively. Results indicate that the surface temperature distribution inside the residential district has an obvious difference in each time. From 9:00 to 15:00, with the change of the sun from east to west, because building blocks each other, the shadow area formed on the surface changes obviously. By blocking effect between solar direct radiation and building, the influence of ground mainly in three types of temperature distribution, the empty area temperature, sunlight shadow area temperature, the wind shadow area temperature. From the surface temperature distribution, the temperature of street canyon ground inside the residential district is significantly higher than the residential outside open area, and the biggest difference in temperature is 5.9 °C. The average temperature range of the building surface is: temperature on the surface of the road, temperature of roof surface, temperature of wind shadow area ground, temperature of sunshine area

ground, temperature of sunshine area wall, temperature of shadow area wall, temperature of shadow area ground, temperature of green surface, and higher temperature area mainly concentrated in the roof and road surface.

With the coming flow direction, the air flow is better and the surface convective heat transfer is stronger, so to the two surface of the same material of pavement in residential, the north-south road surface temperature is lower than the east-west road surface temperature. 1-1 # building located at the upstream of the windward side and its average wind speed 1.36 m/s, while 2-7 # building is located at downstream in residential district, because the flow is blocked obviously by in front of the building, its roof wind speed under 0.4 m/s, so 2-7 # roof surface convective heat transfer function relatively weak in 1-1 # building, leading to the former temperature is 4.7 °C higher than the latter.

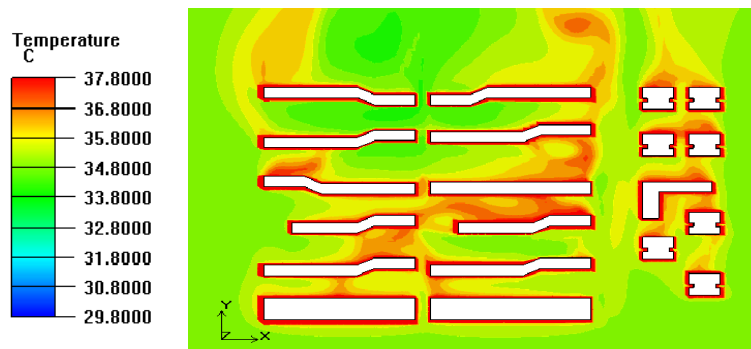


Figure 12 Temperature field at Z=1.5m horizontal plane at 15:00

As shown in fig 12, the simulated result indicates the high-temperature area where the air temperature more than 37°C at 1.5m height mainly concentrated in between 2-3# buildings and 2-10 #, and 1.8 °C higher than the average air temperature inside the residential district. The reason is the 2-3 # building and 2-10 # building located at the center of the wind shadow, coupled with high temperature radiation from the ground, the hot air near the ground is blocked by the counter-clockwise swirl of cooling air flow, so that the heat

accumulation at the bottom of the canyon, the formation of local high temperature regions. Compared to the green belts of same size and configuration, the cooling effect near the main entrance of residential district is better than the effect of the downstream canyons. Because the former near the main entrance, where wind speed of green belts is larger than the latter, so the former transpiration effect is greater than the latter, to reduce the effect of the ambient air temperature is more significant.

4. CONCLUSION

1. Under the same buildings height H and length L , the air average velocity in street canyon will increase as the spacing between buildings increase. This is helpful to reduce pollution and heat accumulation, and to improve the air quality. (simple sentences can be used below, too.)

2. Under the circumstance of the same ratio of height and length of the buildings, the area-averaged air age of the leeward face of lee buildings increased with the increasing of building length. While the area-averaged air age of the leeward face of lee buildings reduced with the increasing of building space.

3. When the height of building is fixed, the reducing of space and length of street canyons is conducive to removing the pollution and enhancing heat dissipation. Then the quality of air within street canyons is enhanced signally.

4. Compared to the green belts of same size and configuration, the cooling effect near the main entrance of residential district better than the effect the downstream canyons. Because the former near the main entrance, where wind speed of green belts is larger than the latter, so the former transpiration effect is greater than the latter, to reduce the effect of the ambient air temperature is more significant.

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REVIEW OF EXPERIMENTAL RESEARCH AND ANALYSIS METHODS ABOUT IN-CAR AIR QUALITY

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ABSTRACT

The main purpose of this paper is to review the experimental research and analysis methods about indoor air quality in vehicles. It also summarizes the sources and hazards of the car indoor organic pollutants, as well as the measures for interior environmental issues which have been taken by the government in some countries. At present, benzene, toluene, xylene, ethylbenzene, styrene, formaldehyde, acetaldehyde and acrolein have been given concentration limit in China, which are the recommended standards of the evaluation of air quality in car. A large number of studies have shown that interior temperature and days lapsed from the delivery of the car are the main factors to affect the concentration of volatile organic compound in the car. Selecting an appropriate mode of ventilation when drive in different road conditions, can effectively improve the air quality inside the vehicle. In addition, this article points out the problems existing in the current study and prospects the future research directions.

Keywords: In-car air quality; Experimental research; Analysis methods

1. INTRODUCTION

The relationship between humans and cars can be traced back to the invention of wheels BC 3,000 years, until the 18th century, the emergence of steam power, the wheels and the engine have a handover at the first time; at the end of the 19th century, the invention of internal combustion engine contributes the appearance of the first car in the world[1]. Since then, automobile not only promoted the development of human society, but also consumed a lot of the earth's limited resources and released a large amount of environmental pollutants at the same time, which was a big threat to the sustainable development of human society.

With the rapid development of China's economic, cars are playing an increasingly important role in people's lives and have become an indispensable "buddy" in our everyday life as they are growing in number and being more than just a means of transportation. However, while offering convenience to people, they impose potential risks such as auto pollution. The World Health Organization estimated the number of people died from air pollution each year reaches 2.4 million[2]. In addition to the air pollution caused by automobile exhaust has been well-known by the public, the car interior air quality problems are more secretive and grim.

More than 20 global environmental monitoring agency study shows: the vehicle air pollution is 10 times higher than the surrounding area in a crowded street. With the popularity of automobile, it is getting longer and longer to stay in the car for people, the data

show that: the time spent by the Americans in a closed car on average is more than 60 minutes a day[3]. Narrow interior space, dissemination of VOCs (Volatile Organic Compounds) from materials and other pollutants entered from outside the vehicle can easily form a serious air pollution, and long-term exposure to which will severely damage human health. The interior ambient syndrome resulting, as early as in 1987, Ritchie and Lehen[4] put forward "indoor air Syndrome" and "composite chemicals allergies", later Molhave and Norback[5,6] expanded this research successively.

China's first case of interior environment pollution problems took place in 2002. Investigation found that the reason for the accident was the high concentration of harmful gases in the new car. After the incident, China National Interior Decoration Association Indoor Environmental Monitoring Center monitored one hundred new vehicles in the Beijing area, the results showed the proportion of new car with pollution problems was up to 90% or more. This finding aroused widespread concern in society at once, experts and scholars began to put eyes on the car interior environmental pollution problems.

2. THE SOURCES AND HAZARDS OF THE CAR INDOOR ORGANIC POLLUTANTS

2.1 The sources of the car indoor organic pollutants

Interior air pollutants include respirable particulate matter, a variety of harmful gases and microorganisms that can be divided into physical pollution, chemical

pollution and biological pollution. Main sources of pollution inside the car, including the release of harmful substances contained in automotive materials and devices, the human body and its activities, auto parts and interior decoration materials, paint, glue, adhesives, car seat cushion, steering wheel sets and so on. Pollutants released by the auto parts and interior decoration materials are mainly benzene, toluene, formaldehyde, olefins, aromatics, etc.; harmful substances contaminants entering from outside the vehicle include mainly hydrocarbons, carbon monoxide, sulfur dioxide, nitrogen oxides, particulate matter and so on.

2.2 The hazards of the car indoor organic pollutants

"Guideline for air quality assessment of passenger car"(the"Guide") issued jointly by the Ministry of Environmental Protection and AQSIQ (Administration of Quality Supervision, Inspection and Quarantine) was implemented formally in March1,2012[7]. The" Guide" defined clearly pollutant concentration limit of benzene, toluene, xylene, ethylbenzene, styrene, formaldehyde, acetaldehyde, acrolein inside the vehicle, so we describes the dangers of the eight substances in detail.

2.2.1 Formaldehyde

Formaldehyde[8] is the most simple aldehydes, it is a flammable, colorless and irritating gas under normal circumstances. When formaldehyde's concentration exceeds $0.1\text{mg}/\text{m}^3$ in air, it can cause damage to the eyes and the cells of mucous membranes. Formaldehyde is classified as a possible carcinogen by EPA (U.S. Environmental Protection Agency), and as a human carcinogen by IARC (International Agency for Research on Cancer).

Formaldehyde is an important organic raw material, mainly used in synthetic adhesives, and formaldehyde resin will emit formaldehyde slowly and continuously, therefore formaldehyde becomes one of common indoor air pollution. The major hazards of formaldehyde is stimulation to the skin and mucous membrane, formaldehyde is protoplasmic poison material, can be combined with protein. Inhalation the high concentration of formaldehyde can result in severe respiratory tract irritation, edema, eye irritation and headache.

2.2.2 Acetaldehyde

Acetaldehyde[9] is liquid with colorless, flammable, pungent odor under room temperature. Low concentration of acetaldehyde can cause irritation and bronchitis of eye, nose and upper respiratory tract. Inhalation of high concentration is narcotic effect. Manifestations include headache, lethargy, confusion, bronchitis, pulmonary edema, diarrhea and myocardial fatty change, which even can cause death.

2.2.3 Benzene

Due to the large volatility of benzene, it is easily spread when exposed to the air. Skin contact or inhalation of a large number of benzene into the body,

can cause acute and chronic benzene poisoning. Research report that, part of reason of benzene poisoning is benzene phenol generated in vivo. International Cancer Research Center (IARC) has confirmed that benzene is carcinogen [10,11].

Benzene, producing paralysis of the central nervous system, causes acute poisoning. There will be severe headache, nausea, vomiting, blurred consciousness, loss of consciousness, coma, convulsions. Small amounts of benzene can make people sleepy, dizziness, rapid heart rate, headaches, trembling, confusion, unconsciousness phenomenon. Long-term exposure to benzene will make blood damaged, cause chronic poisoning and neurasthenia syndrome.

2.2.4 Toluene and Xylene

Toluene[12] is the most simple and important one of the aromatic compound. Many of its nature are very similar with benzene, and often replace benzene as an organic solvent in practical application. Xylene[13] is the compound obtained in the two methyl groups substituted on the phenyl ring, including 45% to 70% of the meta-xylene, 15% to 25% of a mixture consisting of xylene and 10% to 15% o-xylene.

Toluene has only minor damage to the human body. It is absorbed into the body mainly by inhalation, ingestion and percutaneous. Xylene has a moderately toxic, absorbed through the skin. The influence of xylene on health is far smaller than benzene. Xylene and toluene have a stimulating effect to eye and upper respiratory tract, anesthetic effect of them on the central nervous system occurs at their high concentrations.

2.2.5 Ethylbenzene and Styrene

The main application of Ethylbenzene is produced in petrochemical industry as the intermediates of styrene production. Styrene, which is mainly used for the production of styrene series resin and styrene-butadiene rubber, is also one of the raw materials to produce ion exchange resins and pharmaceuticals. The United States Environmental Protection Agency described styrene as suspected carcinogens and suspected toxins harmful to the gastrointestinal tract, kidneys and respiratory system. Ethylbenzene and styrene is classified as a potential carcinogen by International Agency for Research on Cancer[14].

2.2.6 Acrolein

Acrolein is the simplest of the unsaturated aldehydes. It is a very important synthetic intermediate in the chemical industry, and is widely used in the resin production and organic synthesis.

Acrolein has strongly irritating and highly toxic. Inhalation of vapors can damage the respiratory tract, contact with the skin can cause burns; inhalation of large quantities can cause pneumonia, pulmonary edema, heart failure and even death. Acrolein is the third type of material in IARC classification, whether have a carcinogenic effect on humans hasn't been confirmed at present[15]. Acrolein is one of the metabolites of cyclophosphamide, can cause

hemorrhagic cystitis. A recent study also found a link between the presence of acrolein in tobacco smoke and lung cancer[16].

3.THE MEASURES FOR CAR INTERIOR ENVIRONMENTAL ISSUES

The aftermath of the Los Angeles photochemical smog makes the environmental pollution of car gradually get attention. But it was not until 1960, the first limit the car environmental pollutants emissions regulations was promulgated in California[17]. In 1966, Japan enacted the national automobile environmental pollutant emissions regulations[18]. In the 1960s to 1970s, the advanced industrial countries began to develop vehicle emission standards and experimental methods. Until 1983 due to the late start of China's automobile industry, China enacted restrictions on vehicle emissions environmental pollution regulations-car idling emissions standard[19].

Germany: Germany environmental protection department in conjunction with the Automobile Manufacturers Association, the Consumers' Association and other related departments to develop the automotive interior environmental protection regulations, refine the indicators. Such as the car formaldehyde content must be no more than 0.08PPM. This not only makes carmakers have reference, but also protects consumers.

Germany encourages to use skylights of car. Polluted air inflows from car tuyere, then outflow from the rear vent, which formed a circulation in the vehicle and trace amounts of contaminated air will be stranded in the car in this process, when the pollution inside is 10 times the outside of the vehicle. Therefore, more than half of the car design skylight in Germany, rely on the rapid moving of the air flow at the top of skylight to form a negative pressure inside the car, to decompose pollution in the air vents, and ventilation is carried out in air supply way. German car attach great importance to air purifier, air purifier using cellular reflux-effect is more pronounced. In order to thoroughly antibacterial, Germany also uses the popular automobile new antibacterial purifying technology - the photocatalytic technology.

United States: as early as in the eighties and nineties, America had appeared too much hurt caused by in-car air pollution. In view of this, the U.S. put indoor and vehicle pollution as one of the top five hazards to human health.

The EPA requires that the materials automakers use must be declared and examined to ensure that the extent of injury of materials on the environment and the human body reaches the lowest point. Declarer once in violation of regulations, will afford huge fines and recall products to clean up pollution, the main person in charge and even sentenced.

Japan: Japan Motor Vehicle Association (JAMA) released a "sedan car air pollution control Guide". This guide was released in the form of a voluntary action plan, required major Japanese car company to reduce car air pollution after 2005. The guide also released JAMA recommended test method to measure

formaldehyde emissions in the state of 40°C, and measure VOCs emissions when turned on the air conditioning with control target was 25°C. Japanese car manufacturers for vehicle air pollution specifically developed a series of ancillary products, such as new automotive air conditioning with a sterilization function. Toyota also introduced "sterilization ion generator" which was used to eliminate the car bacteria, floats and odor. In addition, Japan also clean air inside the car through a technique called "light catalyst", which can turn harmful gas into harmless water and carbon dioxide by means of oxidation reduction reaction carried out on the harmful gas.

China: March 1, 2012, "Guideline for air quality assessment of passenger car" issued by the Ministry of Environmental Protection and AQSIQ is formally implemented in China, which clearly defined the limits of the concentration of some volatile organic compounds in car, such as benzene, toluene, xylene, ethylbenzene, styrene, formaldehyde, acetaldehyde, acrolein. Since then, the air quality inside the automobile has a recommended standard. Although this is only a "recommended" standard, this will make the enterprise pay more attention to the procurement, production manufacturing, and it also will strengthen consumer attention to car environmental indicators. This invisible driving force is bound to promote the healthy development of the whole car industry.

4.EXPERIMENTAL RESEARCH

Dor et al.[20] found that the single aromatic hydrocarbon exposure level within private cars was 2~3 times than that of the other traffic tools. They also pointed out certain pollutants in the car comes from the adjacent vehicle emissions, which permeate through natural or ventilation equipment into the body. Chan[21] surveyed the VOCs inside public transport vehicles in 2003. Studies found that the choice of passengers on the bus was the main factors influencing the VOCs exposure levels. Taxis, for example, average benzene exposure levels ($33.6\mu\text{g}/\text{m}^3$) was the highest, air-conditioned buses followed by ($13.5\mu\text{g}/\text{m}^3$). Chien[22] investigated the VOC levels of different brand, scale and type of new car in 2007, also considered the in-car temperature and decoration. Results showed that the pollutant concentration was different in different brands and models of cars. Single VOC concentration range was from below the detection limit to several thousand.

2005, Toshiaki Yoshida and Ichiro Matsunaga[23] conducted an experimental research about air of a new car for more than three years, identified 162 compounds in the car, and kept track of the concentration of time-varying. Experimental results showed that most compounds concentrations decreased with the increase of time, and increased with the rising of temperature. The interior temperature and days lapsed after delivery were the main factors affecting the interior concentrations of most compounds according to multiple linear regression analysis. Report pointed out that concentration of most aliphatic hydrocarbon in the car and its concentration outside of the vehicle was in

the same horizontal line in winter. But in the summer, the car's aliphatic hydrocarbon concentration was higher than its concentration outside of the vehicle. Their research provided useful information for the investigation of in-car air pollution caused by automobile interior materials. The Geiss O, Tirendi S et al.[24] surveyed VOCs of 23 private cars in winter and summer, including aromatic, aliphatic compounds, low molecular weight aldehydes and ketones, and a set of the phthalates. Indoor VOC concentration often exceeded the standard value, such as the concentration of benzene can be achieved $149.1\mu\text{g}/\text{m}^3$. In summer, when temperature inside the car reached 70°C , all compounds concentration would be 40% higher than in winter. Bauhof and Wensing[25] detected the change of the concentration of TVOC inside the six new car within 40 days. Detected for 8 hours a day at the same time the temperature was raised to 65°C . In 40 days, the car's TVOC concentration was from 7 to $24\text{mg}/\text{m}^3$ down to 1 to $4.5\text{mg}/\text{m}^3$. In 2000, James S. Grabbs[26] detected VOCs within four new Car, quantitative analyzed TVOC levels associated with the new car smell in a short period of time, and observations of VOCs associated with the new car's internal air was also provided. Found that the car's average concentration of VOCs exceeded the concentration of environment, was often more than the indoor concentrations of VOCs. a minivan TOVC concentration on the first day of sampling was $7,500\text{mg}/\text{m}^3$, after three weeks was reduced by approximately 90%. Other three cars in the sampling for the first time the TVOC concentration range was $300\sim 600\text{mg}/\text{m}^3$. They further estimated that contribution rate of time spent in car on VOCs exposure related gasoline reached 10% to 20%. You et al.[27] identified type and quantity of the compound inside new and used cars in the stationary state. Environmental conditions (temperature, horizontal and vertical flow speed, background VOC concentration) needed to be provided a stable and accurate control using an environmental chamber housing size in this experiment. The experimental results showed that the main VOC include toluene, xylene and other aromatic compounds as well as a variety of types of C7-C12 alkanes. Age in the car for five years, concentration for the top five compounds: toluene ($32.2\mu\text{g}/\text{m}^3$), D-limonene ($14.4\mu\text{g}/\text{m}^3$), butyl hydroxy toluene (BHT) ($14.1\mu\text{g}/\text{m}^3$), m/p-xylene ($10.2\mu\text{g}/\text{m}^3$) and n-undecane ($9.3\mu\text{g}/\text{m}^3$).

Fedoruk and Kerger[28] tested the kinds and concentration of VOC within five different cars. Car conditions were static (parking, no ventilation) and a particular state (such as open air conditioning, only rely on vent or the driver side window half-open). The results of the study showed that the level of TVOC under different conditions was lower 4~8 times (about 50 to $160\mu\text{g}/\text{m}^3$) than measured data when car parked in thermal state. The lowest concentration was often observed when the window ajar. These data suggested that concentration of VOCs detected in car had a relatively high level in extreme heat stationary state. Normal mode of operation can quickly reduce VOC concentration inside the car and even when the car air

conditioning system was set to recycle. So think concentration of VOC emitted from interior materials was the main cause of driver and passenger exposure. In 1999, Jo and Park[29] studied several VOCs concentration within four vehicles traveling in the two cities and a country in South Korea. The average concentration of aromatic VOCs ranges from $25\mu\text{g}/\text{m}^3$ of ethylbenzene to $181\mu\text{g}/\text{m}^3$ of toluene. In winter, the car's TVOC concentration is six times than that of the surrounding environment. This also reflected from the side the importance of a window ventilated for in-car air quality. The University of Hong Kong Andy T. Chan[30] spent six months to detect and analyze the relationship of concentration of nitrogen oxides and carbon monoxide inside and outside of a light-goods-vehicle at three modes of ventilation and four different road conditions. Research found that car IO (indoor and outdoor air quality) ratio was not specific only to the mode of ventilation, but also depends on the driving environment. When the traffic density increased, the car NO levels in the state of natural ventilation was also increased. When opened the air-conditioning, the car's NO and CO concentration were significantly affected by the outside air. This showed that the car in different road conditions should select the appropriate ventilation mode. LD Knibbs et al.[31] chose six passenger cars with different vehicle age (<1 year to 18 years), quantitative studied the exchange and flow rate of the air inside the vehicle when the window is closed in three speed and four different ventilation mode. When the speed was $60\text{km}/\text{h}$, the hourly air exchange rate (ACH) was 33.1; when the speed was $110\text{km}/\text{h}$, the hourly air exchange rate was 2.6 to 47.3, of which 2005 Volkswagen golf was the largest value, 1989 Mazda 121 was minimum value.

In 2010, Ralf-Peter Vonberg et al.[32] used air outer loop and the inner loop of automobile as the prerequisite, utilized a high volume air sampler impact method to measure the number of vehicle microorganisms and mold spores, and found that a few minutes after the start of the air conditioning system, a 81.7% decrease in the number of microorganisms and a 83.3% decrease in the number of fungal spores, in the different types of cars and the air-conditioning system operating mode, the results were not significant differences. When used the old filter and the car's air-conditioning system, the number of micro-organisms in the car was increased 67%, the number of fungal spores was also doubled, when replaced with a new filter, the car's air quality was improved, the number of micro-organisms reduced by 57%, fungal spores were completely removed, the number of particles was also reduced by 80%. So keep the car clean, replace the filter on a regular basis, pay attention to the natural ventilation can reduce the potential harm of micro-organisms and fungi on the human body.

Buters Jeroen TM et al.[33] chose two cars whose age was different (A car for a new car, a car for 3 years) and interior was the same, placed them on the $14,000\text{watt}$ light irradiation to test health effects of substances released by car materials. Results found: the concentration of TVOC in new car and old car were

10.9 mg/m³ and 1.2 mg/m³ respectively, and found no influence of the car compound on the fourth class of sensitizing (t cell-mediated immune response) and the potential stimulation, only a slight effect on immunoglobulin E-mediated immune response was detected in a new car. The survey showed that air inside the car parked well had no significant health hazards.

5. ANALYSIS METHODS

In 2009, Jiang Fan and Wang Chun et al.[34] from Guangzhou University used infrared detector to detect the concentration of carbon monoxide and carbon dioxide inside the car, pressure sensor to test the pressure field, and adopted the multicomponent model to simulate and analyze the flow of toxic gases in vehicle according to the normal driving conditions, also discussed the impact of different forms of ventilation on the flow of toxic gases inside. Multicomponent numerical simulation could obtain many parameters about air flow in the cabin, was an important tool to understand the transportation of VOC in the cabin. Through the multi-component numerical simulation, ventilated velocity and inlet number impact that the benzene transport in the cabin, when increases ventilated velocity, the benzene concentration could reduce, simultaneously increases number of inlet, will further reduce the benzene concentration in the cabin. The simulation found that air entering from the left and right sides of the vehicle and exhausted from the front of the vehicle can maintain an optimal air quality. Provide a useful reference for the control of air quality inside the car.

Tom Longa, Ted Johnsona and Will Ollisonb[35] studied the influence of air exchange rate, speed, window location, state of ventilation, air conditioning wind speed and air conditioning usage on in-car air pollution exposure. They created a database of 4715 car, 1905 marked as “open”, which means that windows, skylights or removable roof fully open or half-open. Linear regression analysis showed that “open” the windows status is affected by the wind speed, relative humidity, speed, cloud cover, temperature, number of days, hours, models, the age of the vehicle, vehicle color, number of windows, skylights, sampling points and air quality. When the relative humidity is high, the apparent temperature is lower than 10°C or when the car is relatively new, should reduce the frequency of the window.

The effects of temperature and humidity in the cabin and days lapsed after the delivery of the car on the airborne concentrations of the organic compounds were analyzed statistically by Yoshida, T and Matsunaga, I.[23]. The relations between the logarithms of interior concentration of the organic compounds and three factors, interior temperature, interior humidity and the number of lapsed days after delivery, were tested with a multiple linear regression model. Data analyses were performed using the software STATISTICA ver. 5.1 J (StatSoft, Tulsa, OK, USA) on a personal computer Sony Model VAIO PCV-J21M (Tokyo, Japan). The data of the logarithm of interior

concentration ($\ln C$, $\mu\text{g}/\text{m}^3$), interior temperature (T , °C), interior humidity (H , %) and days lapsed after the delivery of the car (D , days) at each air sampling were fitted simultaneously to the following equation, and the parameters (A , $B1$, $B2$ and $B3$) were estimated by a linear least-squares model for each compound.

$$\ln C = A + B1 \times D + B2 \times T + B3 \times H$$

where A is intercept of the regression line; $B1$, $B2$ and $B3$ are regression coefficients for lapsed days, interior temperature and interior humidity, respectively.

Kadiyala et al.[36] demonstrated the application of GA(Genetic Algorithm) to predict indoor pollutant levels inside a public transport bus operating on biodiesel fuel. Optimal conditions to run GA were determined and it was observed that for small population sizes, neither crossover nor mutation performed significantly better. The predicted in-vehicle pollutant levels were close to observed pollutant levels indicating that GA performed well. They[37] also compared the performance of two neural network techniques: back propagation(BP) and radial basis function networks(RBFN) in predicting vehicular pollutant levels inside a biodiesel(B20) bus and determined the RBFN model to perform better than the BP model on comparing the statistical performance measures. In this article, the use of GA was explored to predict pollutant levels of carbon dioxide (CO₂, in ppm), carbon monoxide (CO, in ppm), nitric oxide (NO, in ppm), and sulfur dioxide (SO₂, in ppm) inside a public transport bus running on B20 grade biodiesel.

In 2012, Kadiyala et al.[38] adopted the method of two-step to identify and sequence the important factors affecting the particle inside the car. They first used the method of regression tree to determine several important factors that affects the particles inside the car, including the weather, time, indoor source, ventilation road conditions. Then, with variable analysis as supplement the sensitivity analysis of the results of regression tree to sort the important factors affecting the particles inside the vehicle. The results showed that the concentration of particles of the car and the number of sub-micron particles are mainly affected by month and season, visibility and ambient PM_{2.5} will also affect the amount of submicron particles. They[39] also studied HGA(Hybrid Genetic Algorithm) based neural networks using regression trees for modeling air quality inside a public transportation bus. The objective of this research was to develop and evaluate an integrated approach to model the occupant exposure to in-bus contaminants using the advanced methods of data mining and artificial intelligence.

6. CONCLUSION

The more we know about air pollution in car where it comes from and exactly how it affects us, the more we realize how important it is to do something about it. Weschler and Nazaroff[40] found semi volatile organic compounds such as phthalates in indoor were widespread. Some phthalates mass production, it is widely used in plastic products. Phthalates in plastics is not restricted by chemical, is a free activity and

leaching. Therefore, over time, they can be released from the soft plastic products into the environment. Now, however, research materials about phthalates in automobile are quite limited, some past studies focused on the phthalates in indoor living environment. So the next focus of attention should be semi volatile organic compounds inside the car.

Auto pollution exists in two forms, one being exhaust emission and noise that pollute the environment outside the car, the other being harmful substances released by car body material that pollute the environment inside the car. Many articles either consider the influence of external factors on the air inside the car, or the car interior parts and decorative materials to the influence of the air inside the car. Therefore, future research should be put the two together, combined with pollutant distribution model, the fluid flow model, photocatalytic model to assess harmful volatile organic compounds degradation efficiency inside the car.

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DYNAMICALLY MATCHING THE LOAD OF SUPPLY AND DEMAND SIDE BASED ON THE HEAT UTILIZATION OF SEWAGE

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ABSTRACT

Research and analyze Sewage dynamic flow characteristics based on the sewage volume test data of Chongqing sewage pipe network; simulate the annual dynamic load of a shopping mall; Analysis the dynamic differences between them, seek the most unfavorable flow under dynamic matching, Raise matching methods in Various differences case.

Keywords: Sewage volume; Building load; Matching method.

1. INTRODUCTION

Effective use of urban sewage heat energy has significant energy-saving economic and environmental benefits [1], the sewage-source heat pump has become a hot topic. For sewage heat resource utilization at home and abroad, however, most studies are on sewage-source heat pump engineering application technology and equipment development [2] -[3], Such as the way of sewage abstraction and withdrawal, anticorrosive and anti-blocking of the sewage filter, fouling formation and heat transfer enhancement of the sewage heat exchanger, the feasibility of application and the energy efficiency evaluation of the sewage source heat pump and so on; It is lack of the study of matching characteristics between sewage's cold and heat resources and building's cooling and heating demands; and most of the existing researches are based on the "static" rather than "dynamic" consider, sewage cooling and heating resources available capacity is often calculated based on the minimum flow and the extremely temperature, neglect the dynamic changes of the temperature and volume of sewage throughout the year; The building load demand is based on the maximum load in the design temperature while ignoring the characteristics that building most of the time is at part load conditions. Existing research and engineering application experience can not establish a dynamic supply and demand matching relationship between urban sewage and buildings; it is not conducive to take full advantage of the cooling and heating resources of urban sewage and to improve the reliability. Therefore, this article based on dynamic concept, through the analysis of a sewage pipe network's sewage flow annual dynamic changes and a commercial building's summer cooling load dynamic changes, find out the difference between the supply and demand sides, and propose appropriate matching method.

2. METHODS

2.1 The dynamic characteristics of the sewage flow

The urban sewage quantity as the urban domestic water quantity hourly varies from month to month and the degree of variation relate to the type of sewage pipe network, living habits, weather conditions, holidays and so on. There are some certain differences in the flow rate variation between different types of sewage pipe network. From the perspective of the application of urban sewage cooling and heating resources in building, the urban sewage pipe network can be divided into three types[4]: ①the corresponding sewage main pipe, which collected sewage from vast areas along the way. In general, the diameter of the pipeline is from 1.2 to 2.5m and the buried depth is from 2 to 5m; ②the corresponding sewage sub main pipe, which collected sewage from many residential areas or branch pipes. In general, the diameter of the pipeline is from 0.8 to 1.2m and the buried depth is from 2 to 3m; ③ the corresponding sewage branch pipe, which collected sewage from individual residential area or other buildings. In general, the diameter of the pipeline is from 0.5 to 0.8m and the buried depth is from 1 to 2m. The influence of the living habits on the sewage flow is: every day, the use of water at morning, noon and afternoon will lead sewage flow to a peak. Due to the fluctuation of weather, the quantity of drainage is most in summer and least in winter; the displacement of rainy days is more than sunny days'. Without considering the climate mutation, daily variation of the urban domestic sewage flow rate is not obvious; Engineering applications often assume that the daily flow is constant in a month, while the flow appear difference between months. This is corresponding to the monthly statistics.

In fact, the variation of climatic is not obvious in a month; and the volume of sewage is basically stable [5]. Definition month flow coefficient:

$$k_{m,i} = \frac{12Q_{m,i}}{\sum_{i=1}^{12} Q_{m,i}}$$

Where, $k_{m,i}$ ——sewage pipe network month flow coefficient;
 $Q_{m,i}$ ——sewage pipe network month flow(m³/month)
 i ——month, $i = 1, 2, 3, \dots, 12$

Table 1 The distribution of a 3rd class sewage pipe network month flow coefficient

month	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec
$k_{m,i}$	0.99	0.78	0.95	0.96	0.97	1.04	1.17	1.22	1.08	1.01	1.00	0.83

Sewage flow can be considered to be uniform in one hour [4].Hour flow coefficient is defined:

$$k_{h,i} = \frac{24Q_{h,i}}{\sum_{i=1}^{24} Q_{h,i}}$$

Where , $k_{h,i}$ ——sewage pipe network hour flow coefficient;

The variation of sewage pipe network month flow coefficient is associated with the type of the network. The more sewage flow within the pipe network, the smaller the variation of the month flow coefficient, therefore, in order to realize maximum utilization of sewage cooling and heating resources, it needs to get the actual flow distribution at different time. In this paper, research and analysis was conducted about the variation of three types of sewage pipe networks' month flow coefficient in Chongqing, and the 3rd class sewage pipe network flow coefficient distribution throughout year is such as table 1.

$Q_{h,i}$ ——sewage pipe network hour flow(m³/h)

i ——hour, $i = 1, 2, 3, \dots, 24$

This paper monitored the dynamic water supply variation of a residential community in Chongqing, obtained the corresponding sewage pipe network hour flow coefficient as table 2, according to the analysis of the water supply test data.

Table2 The distribution of a 3rd class sewage pipe network hour flow coefficient

hour	0	1	2	3	4	5	6	7	8	9	10	11
$k_{h,i}$	0.79	0.70	0.60	0.61	0.65	0.75	0.82	1.11	1.16	1.05	0.96	1.10
hour	12	13	14	15	16	17	18	19	20	21	22	23
$k_{h,i}$	1.44	1.21	0.84	1.07	1.01	1.30	1.38	1.03	0.97	0.96	0.97	0.97

2.2 The dynamic characteristics of the building load

Air conditioning load characteristics has a great relationship with the type of building, each type of building's load fluctuations is different; it can be seen from the utilization of sewage source heat pump projects of domestic or abroad, most of the sewage source heat pump are used in office and commercial public building. Shopping mall as a typical construction, its cooling load has an obvious fluctuation throughout the year. Therefore, this paper chose a shopping mall building as an example, study dynamic load characteristics and match it with sewage dynamic cold quantity, explore the dynamic matching method based on sewage heat utilization at the supply and demand side.

All-air condition system is usually adopted as the air-conditioning system of shopping mall, the load typically include fresh air load, reheat load and indoor cooling load. In addition to the special commodity mall and underground mall with a high humidity, general air treatment of shopping mall has no reheat load. Indoor

cooling load is mainly composed of the human body load, envelope load, lighting load and equipment load; the shopping mall air conditioning operation indicates that various cooling load ratio range as follows: the human body load and lighting load accounts for about 60%~70% of the total load, fresh air load accounts for about 23%~30%, envelope structure, however, accounted for only 7%~10% [6]. Body load is mainly affected by the impact of occupant density, paper [7] points out that the peak of the occupant density in one day appears at 10:00~11:00 am and 15:00~16:00 pm according to the measured data of the fluctuation of occupant density, specific distribution is shown in Figure 1; the peak of the occupant density in a year appears on the Spring Festival and May 1st and October 1st, July and August is the off-season of shopping centers shown in Figure 2.

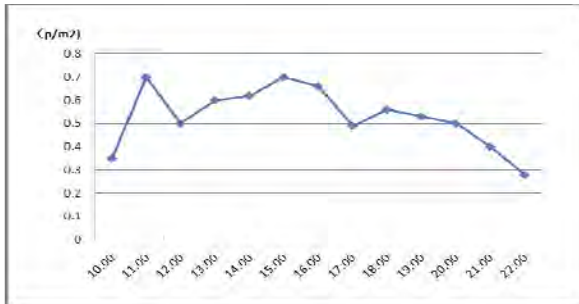


Figure 1 Day density distribution in mall



Figure 2 Year density distribution in mall

Based on the investigation of the situation of shopping mall in Chongqing area, a typical mall building model is set up that construction area is 6000 square meters which consists of 3 layers and every layer is 4.8 m high. The heat transfer coefficient of building envelope and doors and windows are set in accordance with the building energy efficiency standards, the fresh air is 20m³/(hp) , occupant density is shown in Figure 1,2. Other parameters are as follows, Indoor design temperature is 26℃ and humidity is 55%—65% in summer; lighting load is 40(W/m²);

The amount of equipment's heat dissipation: Showcase 10(W/m²), escalator (11kw/each). Due to building envelope load accounted for a smaller proportion for the mall, heating load is small and even become cooling load in winter, so this paper only considers the characteristics of summer cooling load and simulate cooling load from May to October in accordance with the above conditions by using the DEST simulation software; The peak load distribution is shown in figure 3, building load distribution of the typical day in the hottest month is shown in Figure 4.

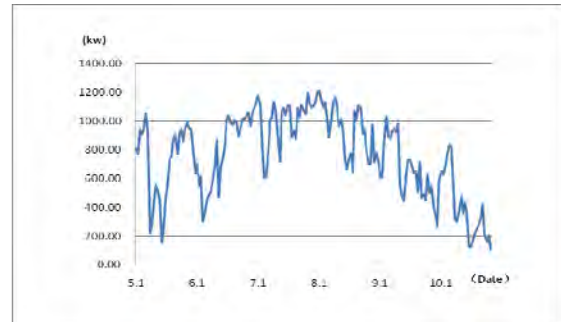


Figure 3 Peak load distribution in summer

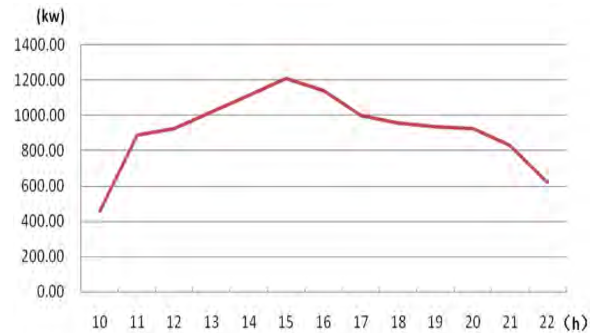


Figure 4 load distribution of the typical hottest day

It can be seen from figure 3 that the fluctuation of summer peak cooling load is drastic, the maximum occurs only in a few days from July to August; the air conditioning load is far less than the maximum load at most of the time. It shows that the number of days when the load is less than 80% of the maximum load is up to 70% of the total number of days through the statistics of all peak load. Figure 4 shows the August 3rd air conditioning load maximum value appeared around 15:00 when occupant density is large; higher outdoor temperatures leading fresh air load is very large.

2.3 Dynamically matching the load of supply and demand side

Because the sewage flow is dynamic changing; the heat that sewage can carry from building is also dynamic changing when the use of air conditioning in summer. Sewage temperature is usually in the range of 25-28 in summer in Chongqing, the variation of sewage temperature in a day is about 1~2℃, the temperature of the sewage is basically stable in the short term, fully meeting the requirements of the cooling temperature of the heat pump unit. We calculated sewage dynamic heat-carrying capacity according to the formula (1) :

$$Q_1 = C \times G \times \Delta T \quad (1)$$

G: sewage dynamic flow in summer, m³/h

Q₁: sewage dynamic heat-carrying capacity in summer, kW

ΔT: temperature difference of Sewage water side (it takes 5℃ in this paper) ,℃。

Calculate sewage dynamic heat-rejection capacity according to the formula (2):

$$Q_2 = Q_L \times (E + 1) / E \quad (2)$$

Q_L : Building dynamic cooling load, KW

Q_2 : Building dynamic heat- rejection capacity, KW

E : COP of the heat pump, KW/KW

Sewage source heat pump projects are usually statically analysis. We can Judge whether the most unfavorable sewage flow can meet the maximum load of the building by the analysis. Sewage heat utilization are often limited in many projects because of the

sewage flow is too small. However, the period that the sewage flow is unfavorable is not always that the building load is at a maximum value; we must find the most unfavorable flow based on dynamic matching between supply and demand sides. We graphed the sewage's average heat-carrying capacity and the building's maximum heat- rejection capacity from May to October, shown in Figure 5:

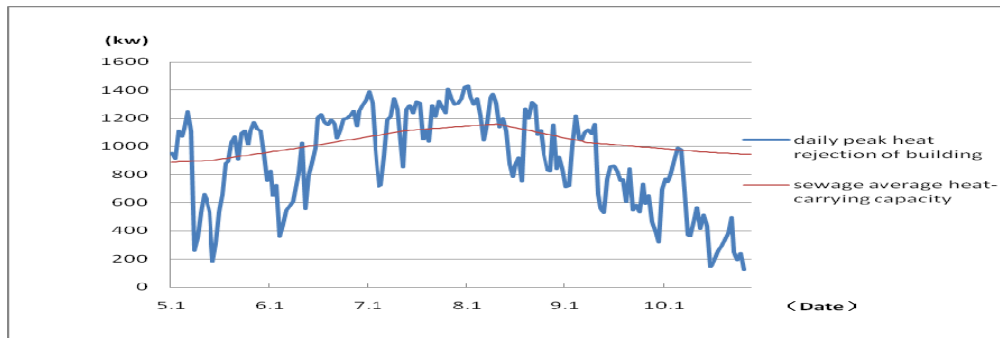


Figure 5 Distribution of Peak heat rejection of building and sewage average heat-carrying

The relative size of building's heat-rejection capacity and sewage's average heat-carrying capacity depends on the volume of sewage and the scale of the building; it can be seen from figure 5 that peak heat rejection of building is more fluctuant, the difference of peak and valley is up to 7 times; while the fluctuation of sewage average heat-carrying capacity is small, the difference of peak and valley is only 1.3 times; the average sewage flow reaches the minimum relative to

the building peak load at late June and early July, it can be considered that the most unfavorable sewage flow appears during this period. We selected one day that can reflect the variation of building load as the most unfavorable day from late June to early July, graphed the sewage dynamic heat-carrying capacity and the buildings dynamic heat- rejection capacity of a typical day, shown in Figure 6:

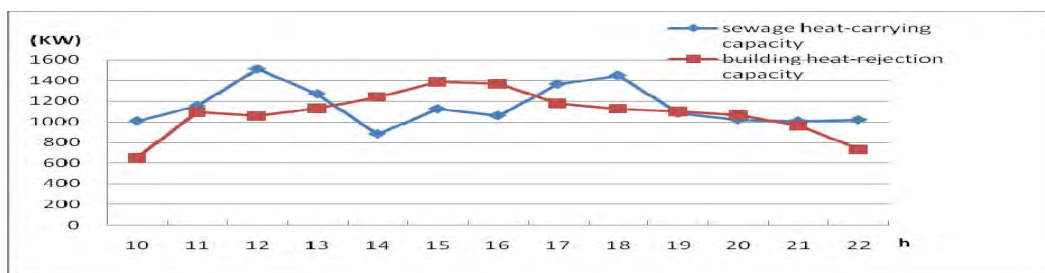


Figure 6 Distribution of the sewage heat-carrying and the buildings heat- rejection of a typical day

It can be seen from figure 6 that sewage heat-carrying capacity and buildings heat- rejection capacity all have a drastic fluctuation, the sewage flow often cannot meet the building's load requirements in the afternoon, the biggest difference appear at 2 pm, at this time, the sewage flow is the most unfavorable under the dynamic analysis. Designs of the sewage source heat pump in practical engineering are often based on the fluctuation of sewage flow at typical days. We matched the fluctuation of sewage's heat-carrying capacity and the fluctuation of building's heat- rejection capacity, and chose appropriate water withdrawal of sewage heat pump system and capacity of cooling and heating

sources. There are generally two kinds of relationships between the most unfavorable sewage flow and building load, as the following:

When the most unfavorable sewage flow meets the building load's requirements, the sewage's heat-carrying capacity is higher than building's heat-rejection capacity at any time, directly using of sewage source heat pump can meet the cooling requirements of building; The dynamic analysis method increases the reliability and scope of using sewage from the angle of the sewage flow and building load's day fluctuation and hour fluctuation compared with the static thought.

When the most unfavorable sewage flow does not meet the building load's requirements, but the accumulation of the sewage throughout the day can meet the building load's requirements, we can add sewage tank or use the energy storage technology to shift load to solve the problem that the sewage flow cannot meet the building load's requirements. But when use sewage tank to relieve the difference between supply and demand sides, we must first solve the problem of water storage. Sewage tank usually occupies a larger area, it is difficult to achieve in practical engineering applications. Therefore, most of projects often built composite system of sewage-source heat pump and conventional air conditioning to solve the contradiction that available sewage cannot match the demand of building cooling and heating load, and to improve the comprehensive utilization efficiency of the energy [9]. Basis and key of composite system's applications is to determine the load sharing coefficient (for existing buildings, in order to make the system more efficient operation, assume what percentage the sewage-source heat pump account for and conventional system account for in the design stage).

Figure 6 show the sewage's heat-carrying capacity reaches the minimum relative to the building's heat-rejection capacity at 14:00, at this time, the portion that building's heat-rejection capacity excess sewage's heat-carrying capacity can be borne by conventional system; the maximum of building's heat-rejection capacity in the morning and evening appears at 11:00, the capacity of sewage source heat pump system can be determined by building load of 11:00; Sewage source heat pump system and conventional systems work at the same time between 13:00 to 17:00, and only sewage source heat pump system work at the rest of the time, this can meet the requirements of the building load. This distribution mode makes the partial load rate of heat pump system and conventional system all at a higher level, can ensure that the unit is running at higher efficiency. We can control the on-off operation of the auxiliary system by controlling chilled water's temperature of supply and return. When the building's heat-rejection capacity at any moment is smaller than the sewage's heat-carrying capacity, just operating sewage source heat pump unit can meet the requirements of the building load throughout the day.

2.4 Conclusion

In order to realize the maximum use of the low grade energy of urban sewage, the relationship of the sewage's heat-carrying capacity and the building's heat-rejection capacity should be analyzed based on the dynamic ideological. This paper chose a shopping mall as an example, converted simulated building load to the building's heat-rejection capacity, and analyzed the heat-rejection capacity compared with sewage heat-carrying capacity of a 3rd class sewage pipe network, obtained the most unfavorable flow.

The differences between sewage's heat-carrying capacity and the building's heat-rejection capacity is

different at different time, the designed capacity of the conventional system should be based on this kind of difference which is occur at the time that the sewage's heat-carrying capacity reaches the minimum relative to the building's heat-rejection capacity in the most unfavorable day; the designed capacity of sewage source heat pump system should be based on the largest building load during the time that the sewage's heat-carrying capacity is higher than the building's heat-rejection capacity in the most unfavorable day, and by controlling the on-off operation of the conventional system to improve the efficiency of energy utilization, make the sewage source heat pump system meet the dual demands of energy efficiency and reliability.

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RESEARCH ON THE INDOORTHERMAL ENVIRONMENT AND WALL THERMALPERFORMANCE OF A NEW TYPE OF PHASE CHANGE MATERIAL ROOM

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ABSTRACT

For lots of areas in the world, the indoor thermal environment is terrible both in summer and in winter. The thermal environment needs to be improved for both the two seasons. A new type of phase change material (PCM) room is proposed in this paper. That is to lay two different kinds of PCM on room envelopes of different orientations. The two kinds of PCM experience the phase change process separately in summer and winter. In this paper, the new PCM room and room without PCM are all built. Experiment is done for the two rooms under the actual climate condition in summer. Indoor air temperature, wall interior surface temperature and heat flux are measured. Several meaningful conclusions are reached.

Keywords: phase change material room; experimental study; indoor thermal environment; thermal performance of the walls

1. INTRODUCTION

The use of phase change material (PCM) in building envelopment can increase the thermal inertia of the wall and decrease the fluctuation of the indoor air temperature, so as to improve the indoor thermal environment. It can also be used with cooling/heating system to storage energy during the night when the electricity price is low, and release energy during the day when the electricity price is high. So the use of PCM in building energy saving is of considerable interest in the last decades [1~2].

Scalat et al. [3] tested a full scale latent heat storage wallboard. Two identical rooms were used; one with ordinary gypsum wallboard and the other with PCM impregnated gypsum wallboard. For both heating and cooling operation the PCM wallboard room exhibited a better thermal behavior. P. Schossig et al.[4] investigated the effect of micro-encapsulated phase change materials in a real building. During the summer season, the air temperature difference between a room with PCM and the reference room without PCM was about 4 °C. Fang and Zhang [5] developed a new composite material by blending an organic PCM with an organic-modified montorillonite. The results showed a reduction of the indoor temperature variation, improving with the increase of PCM used in the board. Cabeza et al.[6] determined experimentally the benefits of a new composite material, mixing Micronal from BASF with concrete. The study in the framework of the European project MOPCON, demonstrated the improvements that can be achieved, reducing the temperature up to 4 °C. K. Frederic et al. [7] carried out experiments in a full-scale test room which is

completely controlled. A numerical modeling has been used to investigate energy storage. Five millimeters of PCM wallboard double the energy that can be stocked and destocked during the experiment. F. Berroug [8] analyzed and discussed the thermal performance of a north wall made with phase change material (PCM) as a storage medium in east-west oriented greenhouse. Results show that with an equivalent to 32.4 kg of PCM per square meter of the greenhouse ground surface area, temperature of plants and inside air were found to be 6-12 °C more at night in winter period with less fluctuation. A.G.Entrop et al. [9] presented an experimental research on a new use of PCMs in concrete floors. It is aimed to reduce the need for thermal energy of conventional heating in houses. The results show that the application of PCMs in concrete floors leads to a reduction of maximum floor temperature up to 16±2% and an increase of minimum temperatures up to 7±3%.

It can be seen that most researches have focused on the thermal performance of PCM wall in summer or winter, but in some areas of the world, the indoor thermal environment requires to be improved both in summer and in winter. The energy consumption needs to be reduced for both cooling and heating systems. For this purpose, new types of PCM wall appear.

A. Pasupathy [10] studied a double layer PCM concept in detail to achieve year round thermal management in a passive manner. It is concluded that for the purpose of narrowing indoor air temperature swing and to suit for all seasons, a double layer PCM incorporated in the roof is suggested and recommended. M. bogdan [11] proposed a new type of composite wall system incorporating PCM, the external layer has a

higher value of the PCM melting point and it is active during hot season and the internal layer with a PCM melting point near set point temperature for heating is active during cold season.

In this paper, a new type of phase change material room is proposed. That is to lay two different kinds of PCM in room envelopes of different orientations. Experimental and numerical studies are carried out for this new type of PCM room.

2. DESCRIPTION OF THE EXPERIMENTAL SET-UP

2.1 Test rooms and materials

The experimental installation consists of two identical small rooms as shown in Fig 1. The dimensions of the two rooms are all 1000mm×1000mm×1000mm. At the south side of each room one double glazing window is installed of 670mm×670mm. The compositions of the walls are described in Table 1.



Figure 1 Experimental test rooms

Two different kinds of PCM (SP29 and RT18) are attached on the interior surfaces of the combined PCM room. SP29 is attached on the floor, roof and west wall while RT18 is attached on the east and north wall. The physical characteristics of the two PCMs are summarized in Table 2. SP29 is inorganic and RT18 is organic. They are encapsulated by steel boxes.

Table 1 Compositions of the wall-materials are given from interior to exterior

Material	Thickness (mm)	Density (kg/m ³)	Capacity J/(kg.K)	Conductivity (W/(m.K))
Room with PCM				
color plate	1.2	7850	480	48
EPS board	75	20	1400	0.04
color plate	1.2	7850	480	48
PCM panel	10	—	—	—
Room without PCM				
color plate	1.2	7850	480	48
EPS board	75	20	1400	0.04
color plate	1.2	7850	480	48

2.2. Instrumentation and measurements

Twelve thermocouples are placed at the center of the interior surfaces of the walls of the two rooms to measure the temperatures of the walls. Another two thermocouples are placed at the center of each room to measure the indoor air temperatures.

Table 2 Physical characteristics of the PCMs

Physical characteristics	values
SP 29	
Phase change temperature (°C)	28-30
Latent heat (kJ/kg)	190
Capacity (J/(kg.K))	2000
Density (kg/m ³)	1530(solid) 1520(liquid)
Conductivity (W/m.k)	0.6
RT 18	
Phase change temperature (°C)	17-19
Latent heat (kJ/kg)	225
Capacity (J/(kg.K))	2000
Density (kg/m ³)	880(solid) 770(liquid)
Conductivity (W/m.k)	0.2

Two flux meters are placed on the interior surfaces of the same orientation of the two rooms. As the rooms are placed outside, the meteorological parameters are also recorded. The solar radiation is recorded by irradiance sensor and the outdoor air temperature is measured by thermocouple. All the parameters are recorded every 10mins.

The experiment instrument and measuring point distribution are shown in Tab.3 and Fig.2.

Table 3 Parameters and instruments

Parameter	instrument	accuracy
air and surface temperature	thermocouple	±0.3°C
heat flux	flux meter	±0.5W/m ²
solar radiation	irradiance sensor	±1.0W/m ²

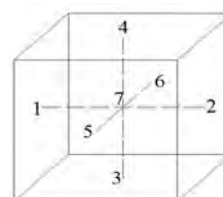


Figure 2 Measuring points distribution sketch

3. RESULTS AND DISCUSSIONS

3.1 Indoor air temperature

It can be seen from Fig.3 that, the maximum indoor air temperature of room without PCM reaches 48.0°C. For PCM room, the value is 42.5°C. The temperature difference is 5.5°C. The minimum indoor air temperature of room without PCM is 29.5°C. For

PCM room, the value is 30.7°C . The temperature difference is 1.2°C . The fluctuation of the indoor air temperature for room without PCM is 18.5°C . The value for PCM room is 11.8°C . The PCM room has a smaller indoor air temperature fluctuation.

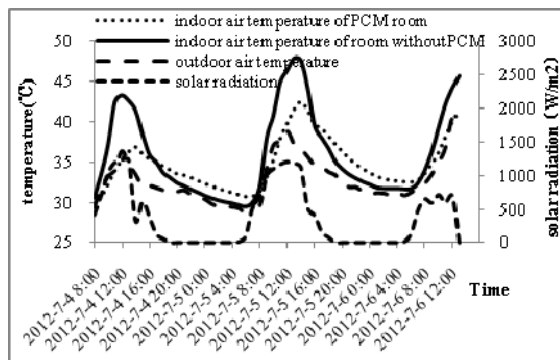


Figure 3 Indoor air temperature of combined PCM room and room without PCM

3.2 Interior surface temperature

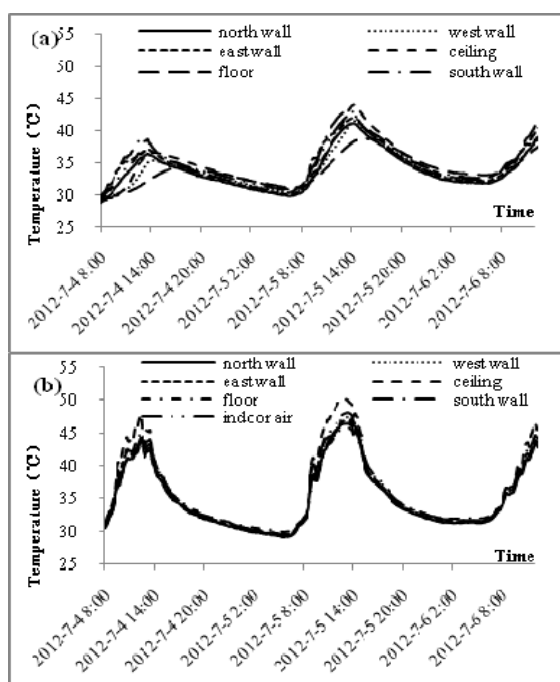


Figure 4 interior surface temperature of the PCM room (a) and room without PCM (b)

Fig.4 shows that the fluctuation of the interior surface temperature for PCM room is much smaller than that of room without PCM. The interior surface temperature for PCM room is lower during the day and higher at night as compared to that of room without PCM.

Table.4 showed the comparison of the maximum temperature differences of the walls of different orientation. The wall containing SP29 has a larger

surface temperature decrease as compared to the wall containing RT18. The reason is that the RT18 is liquid all the time during the experiment. The minimum temperature difference of the walls is very small for all orientations. That is to say that the surface temperature of the two rooms is close to each other at some time of the day.

Table 4 Interior surface temperature difference between the walls of the two rooms

	Maximum value($^{\circ}\text{C}$)	Minimum value($^{\circ}\text{C}$)	Mean value($^{\circ}\text{C}$)
East wall(RT18)	8.11	0	2.2
South wall	6.09	0.02	1.65
West wall(SP29)	11.26	0.03	2.91
North wall (RT18)	8.5	0.03	2.33
Floor (SP29)	12.33	0.02	3.23
Ceiling (SP29)	13.64	0.04	3.56

The average temperature difference between the walls of the two rooms is $1.65^{\circ}\text{C} \sim 3.56^{\circ}\text{C}$, this value is larger for walls containing SP29.

3.3 Heat flux of the interior surfaces

3.3.1 Heat flux of the ceiling

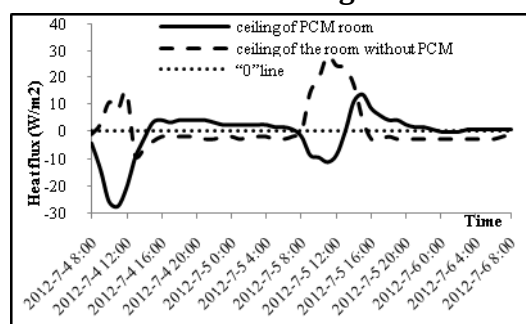


Figure 5 Comparison of the heat flux of the roof of the two rooms

Fig. 5 shows that, the heat flux of the ceiling of the PCM room is positive during the day and negative at night, that is to say the SP29 absorbs heat from the room during the day and releases heat to the room at night. However, the ceiling of the room without PCM releases heat to the room during the day and absorbs heat from the room at night.

It can be seen from Fig. 3 that the indoor air temperature is higher in the second day than the first day. The heat flux of the ceiling of the room without PCM is larger in the second day.

However, the heat flux of the ceiling of the PCM wall is larger in the first day. The reason is that SP 29 is solid at the beginning of the experiment and it absorbs much heat from the room during the first day. At the first night, the indoor air temperature is about 30°C which is very close to the phase change temperature of SP 29, so the heat it releases to the room is less. In the

second day, the solid SP29 is very limited and the thermal resistance of the liquid SP29 is larger than the first day which leads to the reduction of the heat flux.

3.3.2 Heat flux of the west wall

It can be observed from Fig. 6 that the heat flux of the west wall of the room without PCM is negative for most of the time. At about 14:00 of the second day, the heat flux becomes positive because the exterior surface of the west wall is exposed to the sun at this time which causes the interior surface temperature to increase and the wall releases heat to the room. The situation does not happen for PCM room because the west wall of the PCM room contains SP29 which absorbs the heat from the exterior surface and the interior surface temperature will not increase significantly when the wall exposed to the sun. The west wall of the PCM room absorbs heat from the room during the day and releases heat to the room at night. However, the absolute value of the heat flux of the west wall for PCM room is larger than that for the room without PCM.

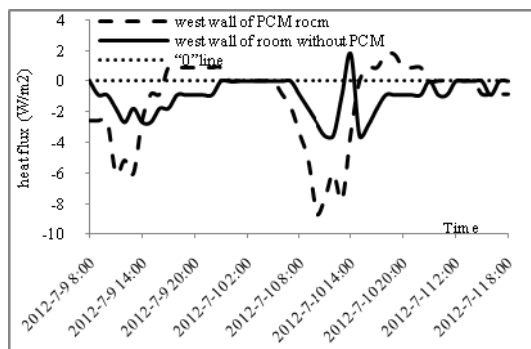


Figure 6 Comparison of the heat flux of the west wall of the two rooms

3.3.3 Heat flux of the floor

The heat flux of the floor for both the PCM room and room without PCM is positive during the day and negative at night. That is to say the two floors absorb heat from the room during the day and release heat to the room at night. The floor of the room without PCM has a similar thermal performance with the floor of the PCM room because the two rooms are all placed on the concrete ground which causes the floor of the room without PCM also have a large heat storage capacity.

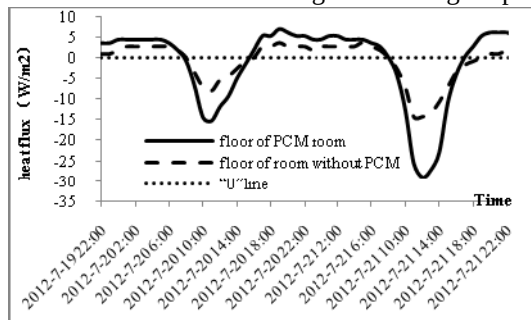


Figure 7 Comparison of the heat flux of the floor of the two rooms

3.3.4 Heat flux of the east wall

The variation rule of the heat fluxes of the east wall of the two rooms is similar. The positive peak of the heat flux appears at about 10:00 am because of the exposure of the exterior surface of the east wall to the sun.

The negative peak appears at about 12:00 am, because the indoor air temperature is much higher than the outdoor air temperature at this time. The heat flux of east wall for room without PCM is larger than that of PCM room.

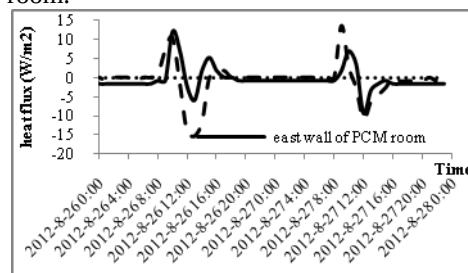


Figure 8 Comparison of the heat flux of the east wall of the two rooms

3.3.5 Heat flux of the north wall

Fig. 9 shows that the variation rule of the heat fluxes of the north wall of the two rooms is similar. The heat flux is negative for most of the time. The absolute value of the heat flux of the north wall for PCM room is smaller than that of the room without PCM.

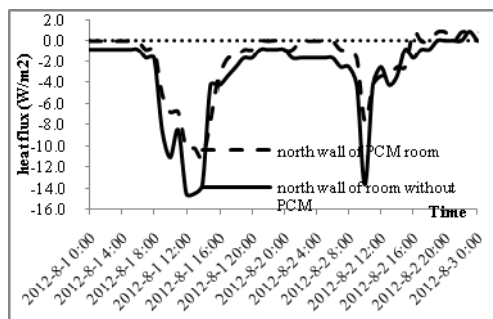


Figure 9 Comparison of the heat flux of the north wall of the two rooms

4. CONCLUSIONS

1. The new type of PCM room can improve the indoor thermal environment.
2. Thermal performance of the walls of room without PCM

- During the day, because the heat gains through the window is very large and the indoor air temperature is very high. The heat flux of the interior surface is negative. For east, west and ceiling, when they are exposed to the sun, the interior surface temperature will increase and release heat to the room.

- At night, the interior surface temperature of the floor is higher than the indoor air temperature and releases heat to the room. However, the indoor air temperature is higher than the outdoor air temperature

and the heat flux is negative for the walls except the floor.

3. Thermal performance of the walls of the PCM room

- During the day, the heat flux is negative for most of the time, but the heat flux of the east wall will be positive when it is exposed to the sun. However, the heat flux of the west wall and ceiling will not be positive even if they are exposed to the sun, because the west wall and ceiling which contain SP 29 will melt and absorb heat when exposed to the sun.

- At night, the heat fluxes of west wall, ceiling and floor which contain Sp 29 are positive and release heat to the room. For east wall and north wall which contain RT 18 that is all liquid during the experiment, the heat fluxes are negative and absorb heat from the room.

4. Common thermal performance of the heat transfer of the two rooms

During the day, only the heat flux of east wall, west wall and ceiling become positive for a short time. For most of the time, the heat fluxes of the wall are negative. However, the indoor air temperature increases quickly in the morning, so the heat transfer through the window is the main heat gains of the rooms. Meanwhile, the heat transfer the window is the main heat losses of the room in the afternoon.

5. ACKNOWLEDGMENTS

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APPLICATION OF CFD IN THE OPTIMAL DESIGN OF A CLIMATE-ADAPTIVE FAÇADE FOR RETROFIT OF EXISTING EXTERNAL WALL IN HSCW ZONE, CHINA

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ABSTRACT

In Hot Summer and Cold Winter (HSCW) zone of China, high performance envelope should consider heat conservation in winter and the same as heat insulation in summer. Air-condition load through the envelope takes a large part in building's total energy consumption. It's highly urgent to improve the thermal performance of existing building envelope.

Illuminated by the concept of double skin façade, in this paper a switchable façade for retrofit of existing buildings in HSCW region is investigated. The switchable façade retrofit system mainly consists of three layers. The first layer, attached to the existing wall called thermal resistance layer is made of PU or Stone wool, even VIP (Vacuum Insulated Panel), etc. The external layer is made of non-transparent materials such as cement board, etc. The air gap layer between those two layers is emphatically discussed in this paper. There are switchable venetian blinds at the top and the bottom of the air gap layer. In summer, the venetian blinds are switched on for nature ventilation. In winter the venetian blinds are switched off and the air cavity can function as thermal insulation.

In this paper, CFD (Computational Fluid Dynamics) calculations were used to optimize the design. Simulation result shows that in summer the heat transfer coefficient of the air gap is positive correlated with its thickness when thinner than 60mm and reach a steady value when the gap get thicker than 60mm; In winter, the air gap thickness has little influence to the thermal insulation of the total system. Overall considering the separated simulation result of summer and winter, we get a recommended thickness of the air gap in condition of a certain size wall. The basic data in this paper can be used for further design and project accordance.

Keywords: Computational fluid dynamics; hot summer and cold winter climate zone; thermal performance; multiple skin façade; building retrofitting;

1. INTRODUCTION

In the last 30 years, the rapid growth and development of the Chinese economy has led to a similar increase in energy consumption. By the end of 2010, the whole of China has over 43 billion square-meters of constructed area [1], however, only 4-5% meets the national building energy standard. In order to maintain occupant living comfort [2], heating in winter and cooling in summer is required in HSCW region (figure.1). To Improve the thermal performance of building envelop has a great significant to reducing the energy consumption.

However, few studies have been conducted on the retrofit of buildings in the HSCW zone different from cold climate zone. Most prevalent wall retrofitting method in HSCW zone is generally according to the External Thermal Insulation Composite System (ETICS) applied in cold climate region.

In this paper, a new retrofit system for existing buildings in HSCW region is introduced (figure.2). The main concept of the system was illuminated by the known double skin façades (DSFs) [3]. This multiple-layer retrofitting system mainly consists of two layers. From inside to outside, the first layer attached to the existing wall is thermal insulation layer which has a high heat resistance. The external layer is made of nontransparent material such as cement board we call it shading layer. The most important part of this system is the air gap between these two layers.

We make the ventilation of the cavity switchable. In summer, the venetian blinds at bottom and top are switched on. When the shading layer is heated by sun radiation, it heats the air in the cavity. By the buoyancy force the high temperature air in the cavity is taken out from the upper opening.

NOMENCLATURE

Abbreviation		D	Cavity width (m)
HSCW	The Hot Summer and Cold Winter climate zone	T_o	air temperature of the outlet (K)
DSFs	Double skin façades	T_i	air temperature of the inlet (K)
CFD	Computational fluid dynamics	I	Solar radiation (W/m^2)
		R	Heat resistance ($\text{m}^2 \cdot \text{K/W}$)
Symbols		Greek Letters	
A_{ff}	effective area (m^2)	κ	Turbulent kinetic energy (J/kg)
t_{am}	Ambient temperature ($^{\circ}\text{C}$)	ε	dissipation rate of turbulent kinetic energy ($\text{J/kg} \cdot \text{s}$)
t_{in}	Indoor temperature ($^{\circ}\text{C}$)	α	absorptance (dimensionless)
t_{sa}	Comprehensive temperature ($^{\circ}\text{C}$)	ρ	Density (kg/m^3),
Δp	pressure difference of the outlet and inlet (Pa)	λ	thermal conductivity ($\text{W/m} \cdot \text{k}$)
Q	volumetric flow rate of air (m^3/h)		



Figure 1 The climate regions of China

At the same time the membrane on the face of the thermal insulation will effectively protect the long wave radiation from the shading layer. On this way the external face temperature of the thermal insulation layer is reduced to near ambient temperature. In winter, the blinds are switched off. With high air tightness, the cavity acts as a thermal insulation layer and will increase the total heat resistance of the whole wall.

Mainly objective of this paper are as follows:

- (1) Introduce one kind of retrofit system adapted to HSCW region;
- (2) Simulate the thermal performance in various conditions;
- (3) Give a basic data for further design guidance and project accordance.

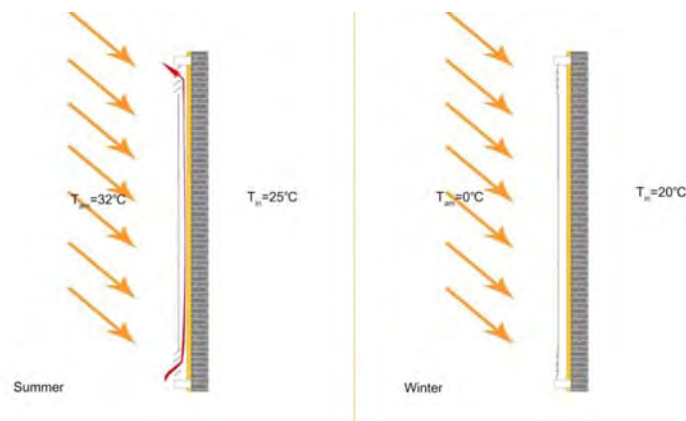


Figure 2 Diagrammatic drawing of the retrofit system

2. CASE DESCRIPTION

An ideal wall was modeled in this research. Assume the existing wall as 240mm thickness brick wall, and the thermal insulation layer as 50mm thickness PU layer with Teflon membrane on the external surface. This research was conducted in mainly three steps. In the first step, the main dimensions of the wall are 10m width and 3m height (equals one story height) discussing the influence of cavity thickness.

Getting the optimal thickness, in the following step, different height of the model was simulated with the certain cavity thickness.

3. NUMERICAL METHOD AND BOUNDARY CONDITION

The numerical results were obtained with the CFD software Flovent 8.1, which is based on the finite volume method to numerically solve the fluid dynamics

governing equations. The κ - ϵ standard turbulence model was used. Radiation was taken into consideration for an accurate simulation.

The boundary conditions are very important for the numerical simulation. The type and the numerical values must be carefully chosen especially for the turbulent quantities. In this research, three kinds of boundary conditions are interested: ambient temperature, external wall boundary conditions and inner wall surface temperature.

Theoretically, as shown in figure.3[4], due to the solar radiation, the comprehensive temperature of outdoors in summer would reach a peak tremendously high. A simplified equation to calculate the comprehensive temperature is:

$$\overline{t_{sa}} = \overline{t_e} + \frac{\overline{I} \cdot \rho_s}{\alpha_e} \quad (1)$$

However, it is unnecessary to simulate transient state of the thermal performance through the day in this study. We set external temperature of the shading layer 55°C based on the in-situ testing conducted by LU ling-huan[5](figure.4).

In summer condition, the ambient temperature was set as 32°C and the temperature of inner surface of the internal wall was set to 26°C.

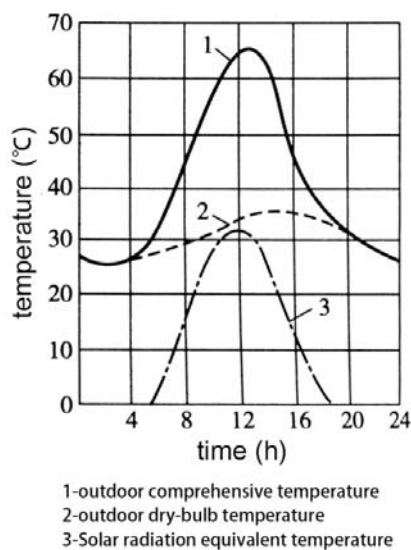


Figure 3 Comprehensive temperature of outdoors

Theoretically, winter condition is simpler, the air cavity acts as a layer of thermal insulation. However, the temperature difference of surface2 and surface3 would cause a circular air movement in the cavity, thus the heat convection would be enhanced. How the dimension of air cavity thickness influence the thermal performance in winter condition is almost as the same important. In the simulation boundary condition, the temperature of shading layer was set to 0°C the same as ambient temperature. The temperature of inner surface3 (marked in fig.) was set to 18°C. On these assumption various dimensions of cavity thickness was simulated.

To ensure the accuracy and authority of the result, much research had been referenced.

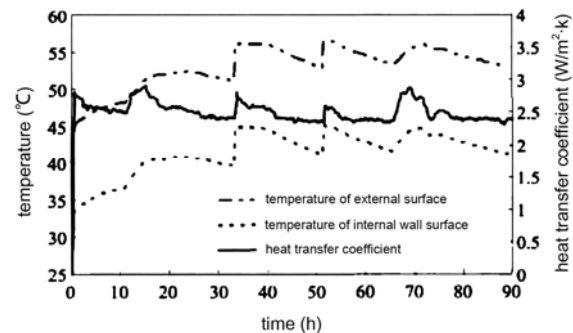


Figure 4 In-situ testing result of the external wall in HSCW region

4. RESULT AND DISCUSSION

2.3 Dimension of the cavity thickness

In summer condition, as shown in figure.5, the external panel was heated by sun radiation and the temperature of surface1 (marked in figure.5) get relatively high. The higher temperature panel then heats the cavity air and radiate of the surface3 (marked in figure.5). Heat was transferred through radiation from external panel and convection from the heated cavity air. At same time, the heated air was driven out by the buoyancy force. This nature ventilation drop down the cavity temperature efficiently.

In the first step, how the dimension of the cavity thickness influence the thermal performance of the total system.

Extract the date from cut plans we get the following figures (from figure.6 to figure.9). From these figures some valuable conclusion can be made.

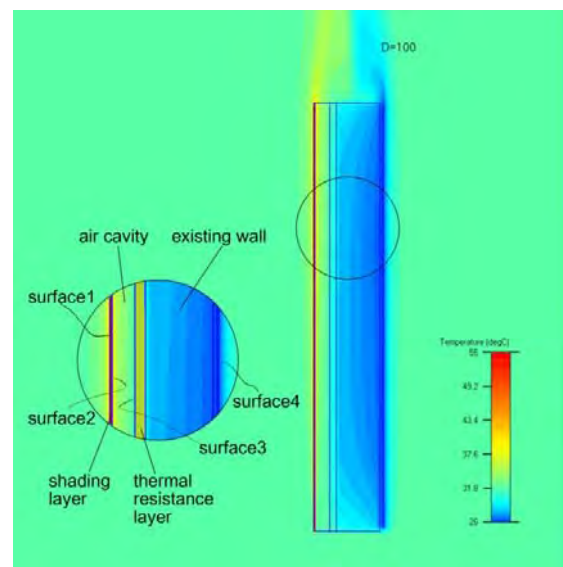


Figure 5 Section view of the simulated system

Figure.6 is the average velocity of the cavity. With D increasing, the average velocity of the air in the cavity increases. Figure.7 and figure.8 each shows mean

temperature of the air in the cavity and temperature of surface3. From figure.9 we get the conclusion that turning point is near D=60mm. Temperature increases rapidly with D's getting thinner. The most important index of the thermal performance of the system is heat flux through surface3. In accord with the temperature trends, heat flux through surface3 can be regarded as two parts from turning point of D=60mm. Thinner than 60mm, the heat flux increases rapidly with the air gap getting thinner. And get relative flat when the air gap gets thicker than 60mm.

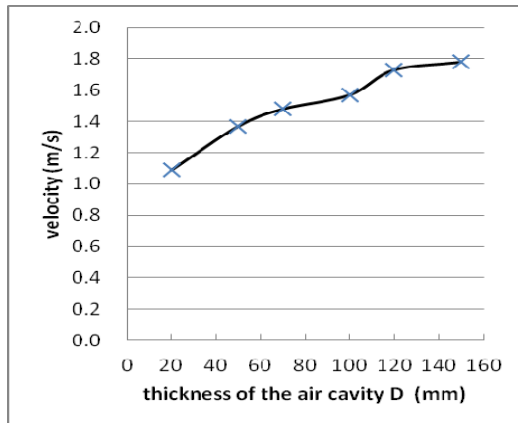


Figure 6 Average velocity of the air in the cavity

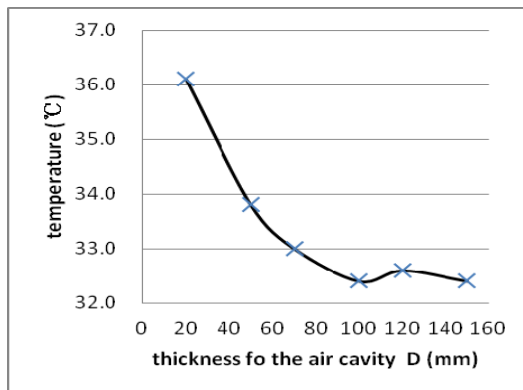


Figure 7 Mean temperature of the air in the cavity

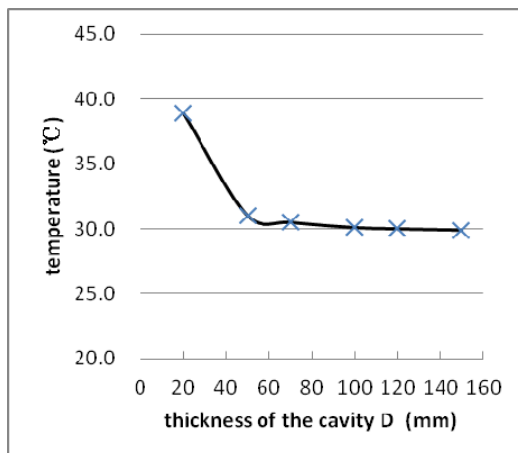


Figure 8 Mean temperature of surface3

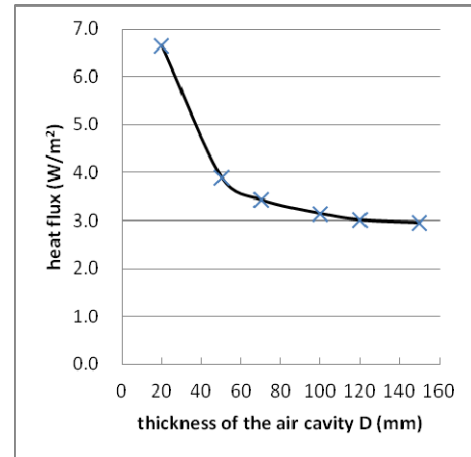


Figure 9 Heat flux through surface3

This is the accordance of the following simulation boundary condition to set the dimension of the cavity to D=60mm.

2.4 Height of the cavity

The vertical ventilation driven by buoyancy force can be calculated by formulas [6] as follow:

$$Q = A_{eff} \sqrt{2\Delta p / \rho_0} \quad (2)$$

$$\Delta p = -\rho_0 g h (1 - T_o / T_i) \quad (3)$$

Q: volumetric flow rate of air;

A_{eff} : effective area of the openings;

T_o : air temperature of the outlet;

T_i : air temperature of the inlet;

Δp : pressure difference of the outlet and inlet.

From the formula we know that Δp is in direct proportion to the $\Delta T / T_i$. The issue is that with the height increases the temperature of the air in the cavity increases. Can the increased air flue take out the heated air more efficiently? In this part research various height were simulated at same boundary condition.

In the simulation, ideal model was set to entirely open at the bottom and top. Except for the height of the panel changes, the other boundary conditions were the same as before. The thickness of the cavity was set to D=60mm according to the conclusion of the first step simulation.

From figure.10 we can find that with the height increase the velocity increases in linear correlation. However, the temperature of air in the cavity and surface3 increases much more sharply as we can see in figure.11. The consequence of the sharply increased temperature causes the increase of heat flux through surface3.

In conclusion, increasing the height at D=60mm can enhance the air flow in the cavity, on the whole, it increase the total heat flux through the wall. Thus, the lower cavity can be better.

Due to the low efficient volumetric flow rate, the temperature of the air in cavity and surface3 increases

rapidly. If possible mechanical ventilation in the cavity is highly recommended.

2.5 Opening Rate

In real practice field it is hard to open the venetian blinds or some other kinds of openings entirely. How the opening size influence the heat performance of the air cavity is an important issue. On the other hand, to shut down the openings absolutely is difficult, to minimize the size of openings can increase the possibility of the absolute air tightness. In this part of the simulation, varies of opening size is simulated.

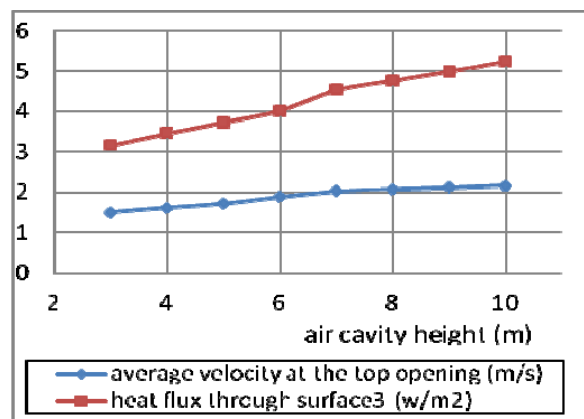


Figure 10 Velocity and heat flux of various cavity height

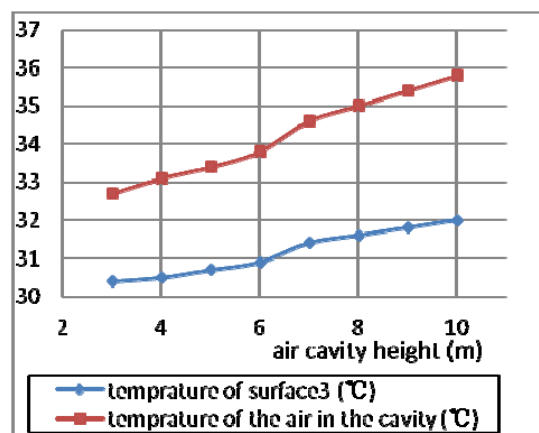


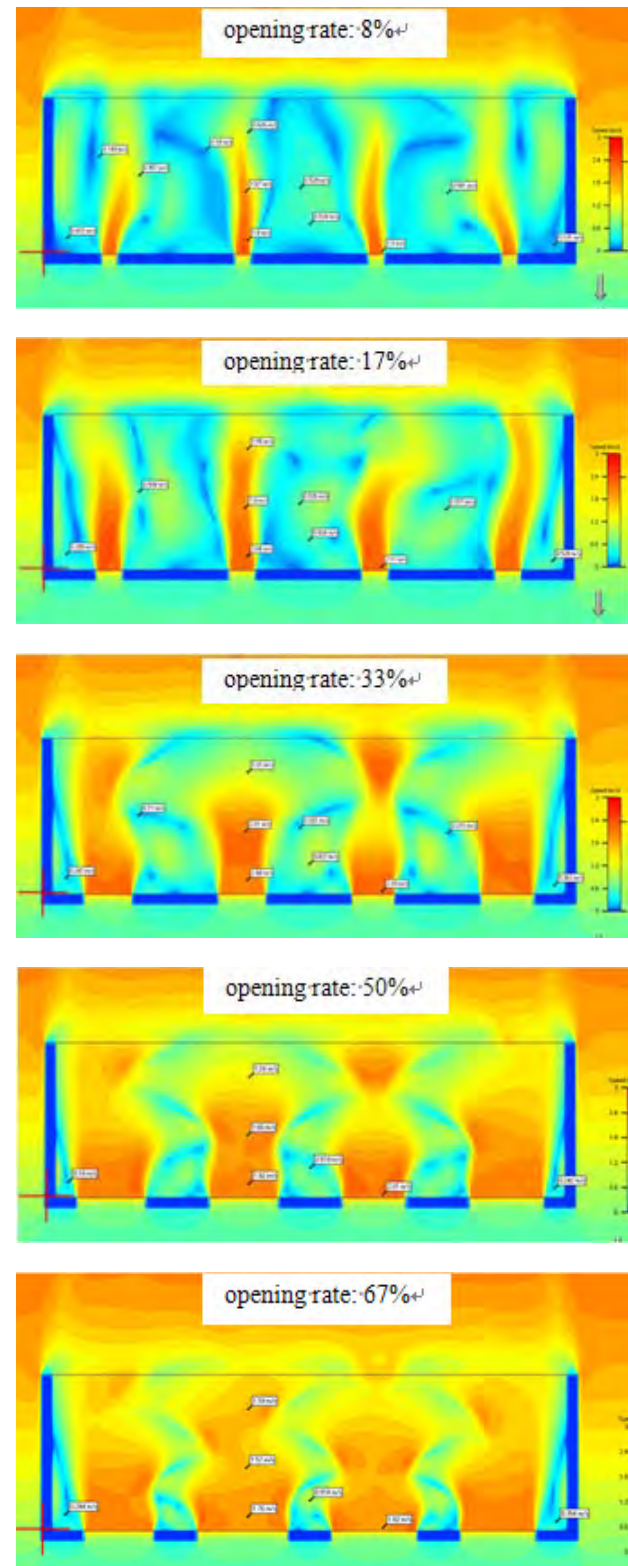
Figure 11 Temperature of air cavity and surface3

Visualized contours were showed in the followings figure.12. Directly perceived through the senses from the contours of the velocity, the more the obstruction exists the much disturbing of the air flow coursed.

Extract the velocity date from the cut plane in the middle of the cavity as shown in figure5. Conclusion can be made that the uniformity of the air flow (represented by velocity) is positive correlated with the opening rate. And the average velocity decreases with the opening rate increasing as shown in figure.13.

Figure.14 and figure.15 shows that with the decrease of opening rate both T_{ca} and T_{sur} increase. The cheering information we get from these two chat is that with the opening rate lager than 50% the temperature of surface3 and cavity get flat.

The last but most important index is the heat flux of the inner face. As shown in the figure.16, with the decrease of opening rate from 50% to 90% the heat flux through surface3 increase rapidly. And from 50% to 100% opening rate the heat flux goes relatively flat.



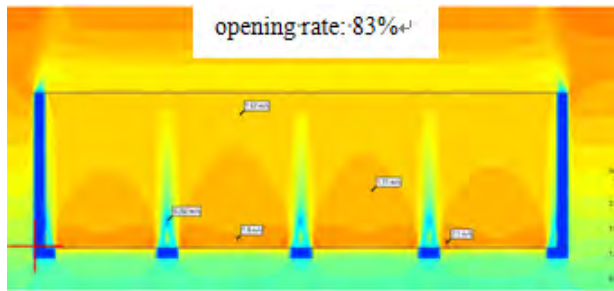


Figure 12 Velocity contours of various opening rate

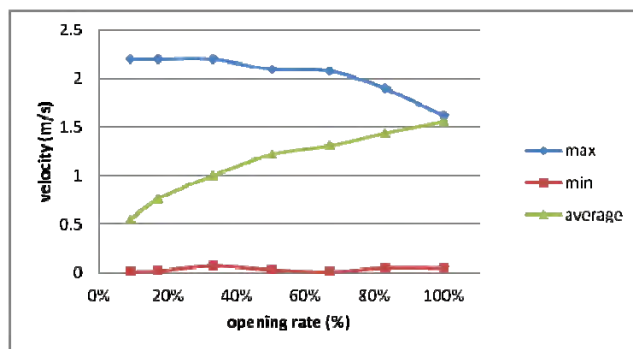


Figure 13 Velocity of the air in the cavity

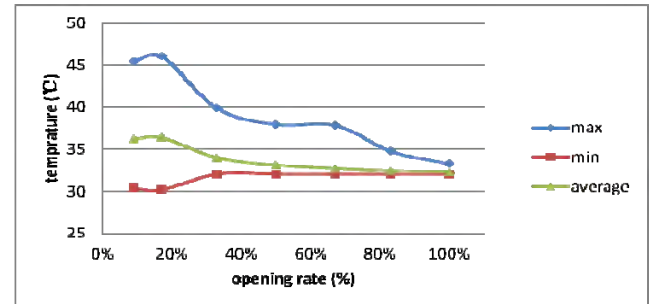


Figure 14 Temperature of the air in the cavity

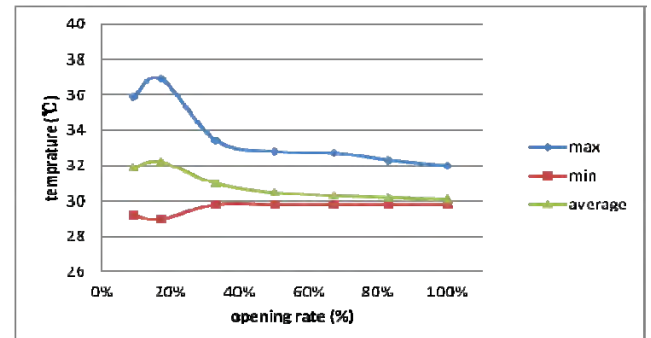


Figure 15 Temperature of surface3

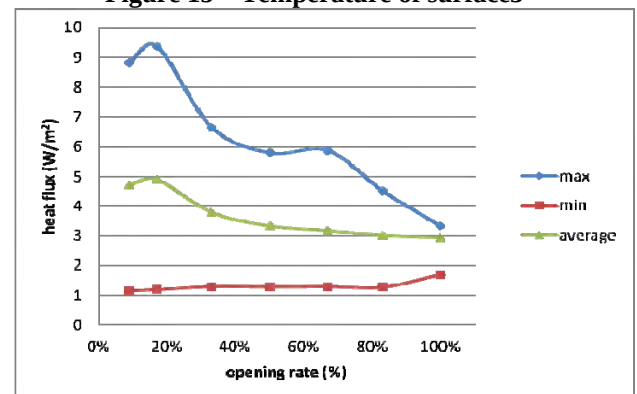


Figure 16 Heat flux through surface3

Table 1. Winter condition

Geometry, heat flow and material properties	Thickness of the air cavity D (mm)						
	10	20	30	40	50	60	70
Vertical air cavity in normal condition	0.14	0.16	0.17	0.18	0.18	0.18	0.18
Vertical cavity with aluminium foil on one side	0.26	0.39	0.44	0.47	0.49	0.50	0.50
Vertical cavity with aluminium foil on one both sides	0.31	0.49	0.59	0.65	0.69	0.71	0.72

2.6 Winter Condition

Theoretically, winter condition is simpler, the air cavity act as a layer of thermal insulation. However, the temperature difference of surface2 and surface3 would course a circular air movement in the cavity, thus the heat convection would be enhanced. How the dimension of air cavity thickness influence the thermal

performance in winter condition is almost as the same important.

From table 1,[7] we get the conclusion that with the air cavity getting thicker the heat resistance of it increases. In normal condition (the reflectivity of both surface2 and surfance3 are normal), the heat resistance gets steady when the cavity thicker than 50mm. another important conclusion we can get from table 1 is that the heat resistance of the cavity doubled or even tripled

when surface2 or surface3 were covered by aluminum foil.

By simply calculation, the R-value of the whole system can reach $1.78(\text{m}^2 \cdot \text{k/W})$ to $2.32(\text{m}^2 \cdot \text{k/W})$ when the cavity thicker than 60mm.

Consider improving the heat resistance in winter, we advise that

(1) Ensure the thickness of the cavity more than 50mm if possible.

(2) Increase the reflectivity of the inner surface (surface2 and surface3 in figure.5).

5. CONCLUSION

In the view of technical, this retrofit system performed well in both summer and winter conditions. As shown in figure.17, in winter, to increase the thermal insulation is a good way to improve the thermal performance. Summer condition is relatively complex, cavity thickness recommend to thicker than 60mm. the height of the cavity is not the higher the better, on the contrary, low air cavity is recommended. If possible, mechanical ventilation is recommended, too.

In the view of application, it is not a big issue to introduce such a retrofit system. But the system itself is worth to be deep introduced and improved. As the author investigated, some practical retrofit project had coincidentally applied some similar system structure without any switchable component. However, they may have not realized the prospect of this system. The purpose of this study is to get a theoretical guidance to this retrofit system. And push the purpose one step further: to get a climate-adaptive envelope system.

6. ACKNOWLEDGEMENTS

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NUMERICAL METHODS FOR POPULATION BALANCE EQUATIONS ON IFVM AND FVM

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ABSTRACT

Ice slurry is considered as a promising refrigerant and secondary refrigerant on cold thermal energy storage, where ice particle volume distribution and its evolution during cold thermal energy storage have a crucial influence on the performance of its thermodynamics and heat transfer. It is of significant importance to carry out investigations on ice crystal volume and distribution during cold thermal energy storage. Population balances, in their integro-differential equation form, are a widely used tool for simulating the dynamic behavior of a population of small particles in dispersed systems, including the ice slurry systems of production and storage. Since, population balance equations (PBEs) can only be solved exactly in simplified cases, numerical solutions are usually needed.

In this paper, both improved finite volume method (IFVM) and finite volume method (FVM) are used to solve the population balance equations, in which growth and aggregation processes are only involved. The PBEs are firstly transformed in the mathematical way, namely convert the quantity conservation equation into mass conservation equation. And then the non-uniform geometric grid is used to discrete computational domain and the finite volume method is used to calculate the transformed equation. The numerical results from analytical solutions, improved finite volume method and finite volume method are respectively compared.

The results show that in pure aggregation process, the numerical results of both IFVM and FVM are in good agreement with the analytical solutions. In the process of growth and aggregation, the numerical results of IFVM keep consistent with the analytical solutions and numerical distortion emerges in the result of FVM compared with the analytical solution. Instead of FVM, the IFVM results show a clear advantage where the advection term disappears from the governing equation.

Keywords: Cold thermal energy storage; Ice slurry; Population balance equations; Improved finite volume method; Growth; Aggregation.

1. INTRODUCTION

The dynamic behavior of a population of small particles is a subject of interest in various engineering fields. Ice slurry is a promising a binary fluid which contains a large number of small ice particles, where ice particle volume distribution and its evolution during cold thermal energy storage have a crucial influence on the performance of its thermodynamics and heat transfer^[1]. Population balances, in their integro-differential equation form, are a widely used tool for simulating the dynamic behavior of a population of small particles in dispersed systems, including the ice slurry systems of production and storage.

Since, population balance equations (PBEs) can only be solved exactly in simplified cases, numerical solutions are usually needed. Many researchers have developed a number of numerical method to solve the population balance equations, such as Monte Carlo method^[2], finite volume method^[3], Quadrature method of moments^[4] etc, the results showed that the numerical results using the methods above were somehow

numerical distortion or instability. However, the improved finite volume method (IFVM), combining method of characteristics and finite volume schemes, is capable of eliminating distortion. Qamar^[5] have described the improved finite volume method, and used improved finite volume method and finite volume method to solve population balance equations respectively, and compared the solutions of two kinds of two numerical methods with analytical solution. But Ref [5] focused on solving the distribution of particle for several dynamics events, without considering one or two kinds of transformation at the same time.

The aim of this current work is to compare IFVM and FVM to solve an integro-differential population balance equations, in which growth and aggregation processes are only involved. The PBEs are firstly transformed in the mathematical way, namely convert the quantity conservation equation into mass conservation equation. A self-developed Matlab program can be used to solve the resultant ordinary differential equations. Validation of the numerical solution will be made to available analytical solutions.

NOMENCLATURE

Symbols		N	the total number of cells,
$f(t, x)$	the number density of a particle of volume x at the time t	q	mesh factor
t	time,s	x_0	mean volume of particles at time $t = 0$
x, y	volume of particles	N_0	total number of particles at time $t = 0$
$x_{\min}$	volume of the smallest particle in total computational domain	Greek Letters	
$x_{\max}$	volume of the largest particle in total computational domain	$\beta(t, x, y)$	aggregation frequency between particles of volume x and y at time t

2. POPULATION BALANCE EQUATION

In this section, we will reformulate the underlying one-dimensional PBEs in a form which can be readily used to derive the proposed numerical methods. Consider a PBE of the form^[6]:

$$\frac{\partial f(t, x)}{\partial t} + \frac{\partial [G(t, x) f(t, x)]}{\partial x} = A_{agg}(t, x) + B_{break}(t, x) + S(t, x) \quad (1)$$

Where $t, x \in [0, +\infty]$, $f(t, x)$ is the number density of a particle of volume x at the time t ; $G(t, x)$ is the growth reaction term; $A_{agg}(t, x)$ is the aggregation reaction term; $B_{break}(t, x)$ is the breakage reaction term; $S(t, x)$ the rate at which particles of volume x are nucleated.

To compare FVM and IFVM, only the effect of growth and aggregation are under consideration in this paper. Thus, Eq.(1) can be rewritten as:

$$\frac{\partial f(t, x)}{\partial t} + \frac{\partial [G(t, x) f(t, x)]}{\partial x} = A_{agg}(t, x) \quad (2)$$

Where the aggregation reaction term, $A_{agg}(t, x)$, is defined as

$$A_{agg}(t, x) = \frac{1}{2} \int_0^x \beta(t, x-y, y) f(t, x-y) f(t, y) dy - \int_0^\infty \beta(t, x, y) f(t, x) f(t, y) dy \quad (3)$$

The first term in Eq.(3) accounts for the formation of particles with volume $x+y$ resulting from the merging of two particles with respective particle volume x and y . The second term is the death term due to loss of particle of particle volume x by aggregation with other particles of any volume. The aggregation coefficient $\beta(t, x, y)$ characterizes the rate at which the aggregation of two particles with respective particle volume x and y produces a particle of volume $x+y$.

Consider the mass balance. Multiplying both sides of Eq. (2)- (3) by x :

$$x \frac{\partial f(t, x)}{\partial t} + x \frac{\partial [G(t, x) f(t, x)]}{\partial x} = x A_{agg}(t, x) \quad (4)$$

Define $x A_{agg}(t, x) = -\partial F_{agg}(t, x) / \partial x$, then Eq. (3) can be rewritten as:

$$F_{agg}(t, x) = - \int_0^x \int_{x-u}^\infty u \beta(t, u, v) f(t, u) f(t, v) du dv \quad (5)$$

Define $\tilde{f}(t, x) = x f(t, x)$, then Eq. (4) can be

rewritten as:

$$\frac{\partial \tilde{f}(t, x)}{\partial t} + \frac{\partial [G(t, x) \tilde{f}(t, x)]}{\partial x} - \frac{G(t, x) \tilde{f}(t, x)}{x} = - \frac{\partial F_{agg}(t, x)}{\partial x} \quad (6)$$

Instead of the Eq. (2), we will discretize Eq. (6).

Due to the relation $\tilde{f}(t, x) = x f(t, x)$ one can easily recover $f(t, x)$.

3. NUMERICAL PROCEDURE

3.1 Domain discretization

In order to apply any numerical scheme, the first step is to discretize the computational domain. Let N be a large integer representing the total number of cells, we denote by $(x_{i-(1/2)})$ a mesh of $[x_{\min}, x_{\max}]$, where $x_{\min}$ is the minimum and $x_{\max}$ is the maximum value of the property variable. To capture the initial profile properly, in this paper it is useful to use a geometric grid. A typical one-dimensional geometric grid is given as, the parameter q is any positive integer.

$$x_{1/2} = x_{\min},$$

$$x_{i+1/2} = x_{\min} + 2^{(i-N)/q} (x_{\max} - x_{\min}), \quad i \in \{1, \dots, N+1\} \quad (7)$$

3.4 Numerical methods

After having a discretized computational domain and assigning the initial data to each grid cell, the next step is to solve the given Eq. (6). Finite volume method (FVM) was used to solve the PBE without any further modification^[7]. Therefore, in this section, we are interested to apply improved finite volume method (IFVM) for PBEs: the method of characteristics for the growth process and a finite volume scheme for the aggregation process. For a scalar linear conservation law, for example PBE in the present case, there exist unique characteristic curves along which information propagates. If the solution moves along the propagation path-line, the advection term $\partial G f / \partial x$ in the current PBE disappears. This drastically reduces numerical diffusion in the solution of the scheme in comparison to other schemes which use some discretization technique for approximating the advection term.

Substitute the growth rate $G(t, x)$ by: $dx/dt=G(t, x)$

Integration over the control volume $\Omega_i(t) = [x_{i-1/2}(t), x_{i+1/2}(t)]$ implies

$$\frac{d}{dt} \int_{\Omega_i(t)} \tilde{f}(t, x) dx = \int_{\Omega_i(t)} \frac{G(t, x) \tilde{f}(t, x)}{x} dx - (F_{agg(i+1/2)} - F_{agg(i-1/2)}) \quad (8)$$

Where $F_{agg(i+1/2)}$ according to Filbet and Laurencot^[8] are given as

$$F_{agg(i+1/2)}^m = \sum_{k=0}^i \Delta x_k(t) \tilde{f}_k^m \left\{ \sum_{j=\alpha_{i,k}}^N \tilde{f}_j^m \int_{\Omega_j} \frac{\beta(x, x'_k)}{x'} dx' + \tilde{f}_{\alpha_{i,k}-1}^m \int_{x_{i+1/2}-x_k}^{x_{\alpha_{i,k}-1/2}} \frac{\beta(x, x'_k)}{x'} dx' \right\} \quad (9)$$

Where the integer $\alpha_{i,k}$ corresponds to the index of the cell such that $x_{i+1/2}(t) - x_k(t) \in \Omega_{\alpha_{i,k}-1}(t)$.

Define $\tilde{f}_i = \tilde{f}_i(t)$ as the mean volume of the particles, $\tilde{f}_i = \Delta x_i^{-1} \int_{\Omega_i} \tilde{f}(t, x) dx$, Substitute the mean

volume of the particles into Eq. (6):

$$\frac{d}{dt} \tilde{f}_i = \frac{G_{i+1/2} \tilde{f}_i}{x_i(t)} - \frac{(F_{agg(i+1/2)} - F_{agg(i-1/2)})}{\Delta x_i(t)} \quad (10)$$

$$\frac{d\tilde{x}_{i+1/2}}{dt} = G_{i+1/2} \quad (11)$$

3.5 Initial and boundary condition

Initial condition:

$$\tilde{f}(0, x_i) = \tilde{f}_0(x_i) \quad (12)$$

As a boundary condition the number density is taken to be zero outside the computational domain. This means that there exist no smaller or bigger particle than our specified volume range of particles. The above system of ordinary differential equations can be solved by a self-developed Matlab program.

4 RESULTS AND ANALYZE

The numerical schemes derived in this article can be applied to solve various combinations of nucleation, growth and aggregation processes for different choices of nucleation, growth and aggregation rates. However, to validate the current schemes, it is needed to test the schemes for the test problems where analytical solutions are also available. In the literature, analytical solutions of the PBEs are available for the following combinations of processes: pure aggregation and simultaneous growth and aggregation.

4.1 Pure aggregation

Consider constant aggregation with $\beta(t, x, y) = \beta_0$ and an exponential initial distribution of the form:

$$f(0, x) = f_0(x) = N_0/x_0 \exp(-x/x_0) \quad (13)$$

Where N_0 and x_0 are the total number of particles and the mean volume of particles at time $t = 0$, respectively. In both methods we use a geometric grid with $q = 3$. Where, we take $\beta_0 = 1$, $N_0 = 5$, $x_0 = 0.01$ and 91 mesh points. The analytical solution is given by Kumar^[9]

$$f(t, x) = \frac{4N_0}{x_0(N_0\beta_0 t + 2)^2} \exp\left(\frac{-2x}{(N_0\beta_0 t + 2)x_0}\right) \quad (12)$$

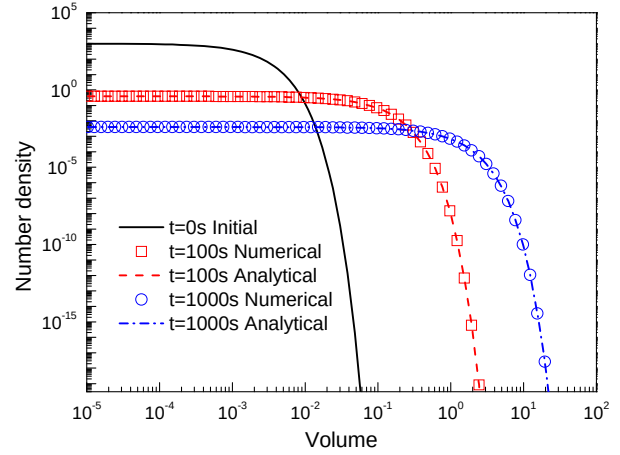


Figure 1 Number density of constant aggregation process on IFVM

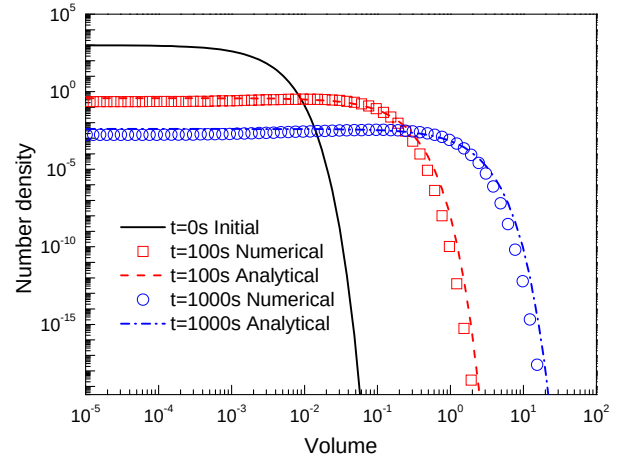


Figure 2 Number density of constant aggregation process on FVM

Figure 1 and Figure. 2 illustrate the numerical results of both methods are compared with the analytical solution. Note that both figures use a log scale on both axes so that the solutions are not merely shifted in the graph. The IFVM results in Figure 1 are in excellent agreement with the analytical solution. The stability and numerical diffusion of the numerical solution which marred the performance of the finite difference techniques are completely absent in the present scheme. The numerical results of FVM in Fig. 2 are also in agreement with the analytical solution. However, one can see a little numerical dissipation in the solution at the right of the curve. A comparison of both methods shows that FVM results are little bit diffusive. Instead of any other scheme which uses some discretization technique for the advection term, the IFVM results show a clear advantage where the advection term disappears from the governing equation.

4.1 Growth and aggregation

Consider constant aggregation with $G(x) = G_0$ and constant aggregation with $\beta(t, x, x') = \beta_0$ and an exponential initial distribution of Eq. (13). In both methods we use a geometric grid with $q = 4$. Where, we take $G_0 = 1$, $\beta_0 = 100$, $N_0 = 5$, $x_0 = 0.01$ and 91 mesh points. The analytical solution is given by Qamar^[6]

$$f(t, x) = \frac{M_0^2 / M_1}{1 - 2\Lambda x_0 (N_0 - M_0 / M_1)} \times \exp \left[-\frac{M_0 (x - 2\Lambda x_0 (N_0 / M_0 - 1))}{M_1 (1 - 2\Lambda x_0 (N_0 - N_0 / M_0))} \right] \quad (15)$$

Where,

$$\Lambda = G_0 / \beta_0 N_0 x_0, \quad M_0 = \frac{2 N_0}{2 + \beta_0 N_0 t}, \quad (16)$$

$$M_1 = N_0 x_0 \left[1 - \frac{2 G_0}{\beta_0 N_0 x_0} \ln \left(\frac{2}{2 + \beta_0 N_0 t} \right) \right]$$

The simulation results of constant growth and constant aggregation process on IFVM and FVM are shown in Fig. 3 and Fig. 4, respectively. Fig. 3 indicates that the numerical results of IFVM are in very good agreement with the analytical results. However, the numerical results of FVM are clear dissipative at the left discontinuity of the number density profile and are in good agreement in the other regions. There are some differences between FVM results and analytical on the far left side of the number density profile for $t = 0.01$ in Fig. 4 and similar but not visible differences in Fig. 3. This is because the analytical solution in this range breaks down. Kumar^[10] et al have pointed out that for approximately equal growth and aggregation rates, their analytical solution for $G(x) = G_0$ and $\beta(x, x') = \beta_0$ is valid for only large particle volumes. For the aggregation-dominated situation, the range of validity is increased, where as for the growth-dominated situation, the solution breaks down completely. The simulations given in Fig. 3 and Fig. 4 are somehow aggregation-dominated cases in order to allow analytical solutions to hold in a reasonably wide range.

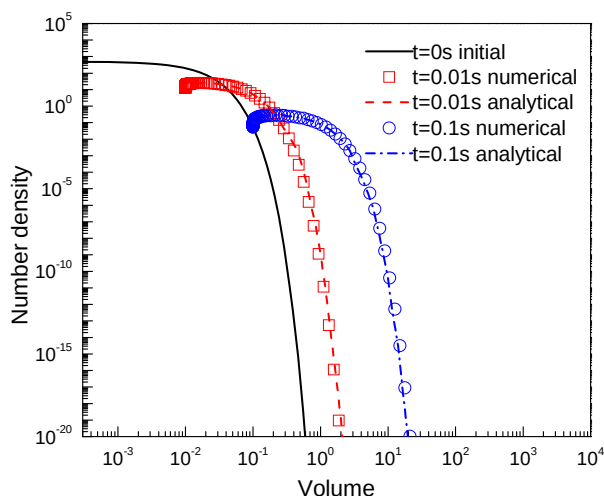


Figure.3 Number density of constant growth and constant aggregation process on IFVM

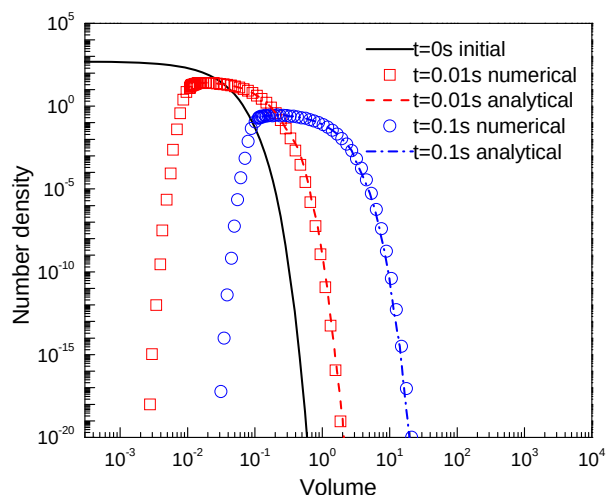


Figure.4 Number density of constant growth and constant aggregation process on FVM

5. CONCLUSIONS

In this article two numerical methods are compared for the solution of PBEs for pure aggregation and simultaneous growth and aggregation processes. The IFVM combines the finite volume scheme for aggregation process with a method of characteristics for growth process. The FVM purely uses a finite volume scheme for growth and aggregation of particles. The results show that In pure aggregation process, the numerical results of both methods are in good agreement with the analytical solutions. In the process of growth and aggregation, the numerical results of IFVM keep consistent with the analytical solutions and numerical distortion emerges in the result of FVM compared with the analytical solution. Instead of FVM, the IFVM results show a clear advantage where the advection term disappears from the governing equation.

ACKNOWLEDGEMENTS

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NUMERICAL STUDY ON THE THERMAL PERFORMANCE OF THE U-TYPE VACUUM SOLAR COLLECTOR TUBE

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ABSTRACT

The U-type vacuum solar collector is widely used in solar heating and solar air conditioning due to its high thermal efficiency. The study of U-type vacuum solar collector tube is necessary to better improve its thermal efficiency in high temperature condition. In this paper, the thermal performance of a single U-type vacuum solar collector tube is studied by numerical method. The physical model and three-dimensional mathematical model of two different geometric construction of single U-type vacuum solar collector tube are established and the coupled heat transfer of conduction, convection and radiation in the U-type vacuum solar collector tube is realized. The influence of air layer between absorber tube and the cooper fin and natural convection in cross section of U-tube on the thermal performance of U-type vacuum solar collector tube is analyzed. The effect of main parameters including fluid inlet velocity, inlet temperature and radiation intensity collected by solar collector on the heat transfer characteristics of U-type vacuum solar collector tube are researched. Results show that when considering air layer between the absorber tube and the copper fin, the thermal efficiency of the U-type vacuum solar collector tube reduces about by 6% and the surface temperature of absorbing coating increases by 40°C comparing with that of the assumption that the absorber tube and the copper fin is tightly connected. The natural convection in the U-tube increases the heat transfer coefficient of branch one of U-tube compared with pure convection. However, the heat gain of branch two of U-tube decreases and the total heat gain remain almost the same compared with pure convection due to the heat transfer characteristic of branch two of U-tube. The thermal efficiency falls by 9% and the surface temperature of absorbing coating increases by 56°C when fluid inlet temperature ranges from 20°C to 80°C. The heat gain of branch one of U-tube decreases with the increase of fluid inlet velocity, because the increase of inlet velocity weakens the natural convection in the branch one of U-tube. And the heat gain of branch two of U-tube increases with the increase of fluid inlet velocity and the overall heat gain barely changes. According to the analysis above, the heat transfer characteristics of U-type vacuum solar collector tube in different operating condition are better understood.

Keywords: U-type vacuum solar collector; heat transfer characteristic; numerical simulation

1. INTRODUCTION

As we all know, the all glass evacuated solar collector is widely used in China as a way of solar thermal utilization to provide domestic water or heating. The thermal performance of all glass evacuated solar collector is studied by many researchers. However, the use of all glass evacuated solar collector is restricted in occasion requiring high temperatures and high pressure for its shortcomings such as easy breaking ,frost crack and can't operating in high pressure. To eliminate these drawbacks of all glass evacuated solar collector, the mental-glass evacuated solar collector is studied and the U-type vacuum solar collector with copper fin is the most widely used in market for its pressure-bearing and favorable thermal performance in high temperature conditions. The investigation of the thermal

performance of the U-type vacuum solar collector is significant to improve the thermal efficiency and lay a foundation for the utilization of the U-type vacuum solar collector in air-conditioning and large-scale project.

In Ref. [1], the effect of air layer resistance between the absorber tube and copper fin on the thermal efficiency of solar collector is researched. Results show that the thermal efficiency of solar collector remarkably decreases with the increase of air layer thickness. Ma et al. employ one-dimensional analytical method to investigate the influence of air layer on the thermal performance of the solar collector [2]. The results indicate that the thermal efficiency of solar collector falls by 10% and coating temperature increases 30°C because of the air layer resistance.

NOMENCLATURE

Symbols

S	Solar energy absorbed by selective coating (W)
Q_u	Net heat gain of solar collector (W)
Q_L	Heat loss of solar collector (W)
U_e	Edge loss coefficient of header tube $W/(m^2 \cdot K)$
U_t	heat loss coefficient from absorber tube to ambient
h_{ga}	convection heat transfer coefficient $W/(m^2 \cdot K)$
h_{pgc}	heat transfer coefficient for conduction $W/(m^2 \cdot K)$
h_{pgr}	radiation heat transfer coefficient $W/(m^2 \cdot K)$
A_a	outer surface area of the absorber tube (m^2)
A_0	aperture area of collector (m^2)
L	Length of solar collector tube (m)
W	distance of adjacent collector tubes (mm)
D	Outer diameter of solar collector tube (mm)

b	Glass thickness (mm)
I_T	Radiation intensity (W/m^2)
E	intensity of inner heat source (W/m^2)
g	Acceleration of gravity (m/s^2)
u	Fluid velocity in x direction (m/s)
v	Fluid velocity in y direction (m/s)
w	Fluid velocity in z direction (m/s)
T_a	Ambient temperature (K)

T_g	Outer glass temperature (K)
T_p	Temperature of selective coating (K)
T_c	Inner wall temperature of absorber tube
T_u	Wall temperature of U-tube (K)
T_f	Fluid temperature (K)
T_{ai}	Temperature of copper fin (K)
T_{in}	Fluid inlet temperature (K)
T_0	Initial temperature of fluid (K)
c_p	specific heat at constant pressure $J/(kg \cdot K)$

Subscripts

-up	Upper wall of absorber tube
-in	inlet
-out	outlet

Greek letters

α	Absorptivity of the selective coating
η	Thermal efficiency
θ	Tilted angle of solar collector tube
τ	transmissivity of the glass
ρ	Fluid density in any temperature (kg/m^3)
ρ_0	Initial temperature of fluid (kg/m^3)
λ	Thermal conductivity of fluid $W/(m \cdot K)$
μ	Dynamic viscosity coefficient ($N \cdot s/m^2$)

Kim et al. researched four different shapes of absorber tube by numerical method and find that the U-tube welded tightly inside a circular fin has the best thermal characteristics for a single collector tube compared with the other three types of structures [3]. The bond conductance of solar collector with five different samples is studied by Badran et al experimentally [4]. The results turn out that the bond conductance of five samples used in experiment ranges from 1.8 to 6.3W/(m·K). According to the investigations above, we can conclude that the air layer resistance between the absorber tube and the copper fin is paramount to the thermal performance of U-type vacuum solar collector. However, the researches above are mostly one-dimensional analysis and the three-dimensional numerical study is necessary to have a better understand of the thermal performance and can realize the visualization of the temperature field and flowing field. In addition, the natural convection for laminar flow in collector tube is another important parameter needed to be studied. Ho et al analyzed the thermal efficiency of U-type vacuum solar collector with copper fin theoretically [5] and results show that the increase of copper fin can strengthen the heat transfer efficiency. But Hobbi et al indicate experimentally that the enhancement devices in flat-plate solar collector don't have effect on the improving the heat transfer in the tube when considering natural convection in the tube [6]. The flow in the solar collector is usually the combination of natural convection and forced convection especially for laminar flow. So the thermal performance of U-type vacuum solar collector in mixed convection for laminar flow is needed to be researched.

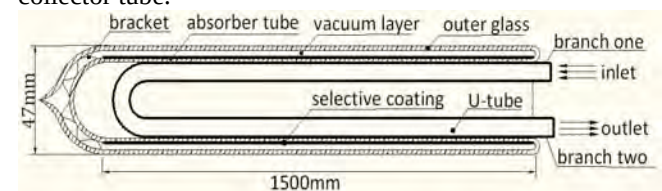
In this paper, the coupled heat transfer performance of single U-type vacuum solar collector

tube with copper fin is investigated numerically based on the three-dimensional model. The air layer between the absorber tube and the copper fin and the mixed convection in the collector tube are researched. Also the effect of the main elements such as the fluid inlet velocity, the fluid inlet temperature and radiation intensity on the thermal performance is analyzed.

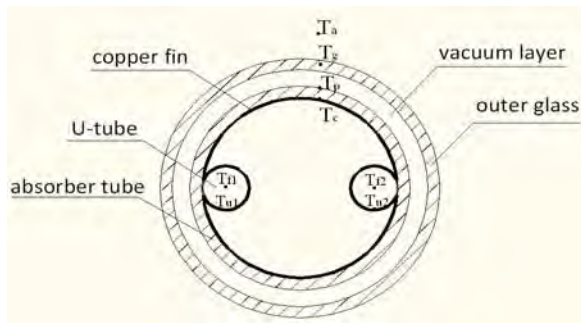
2. PHYSICAL AND MATHEMATICAL MODEL

2.1 Physical problem description

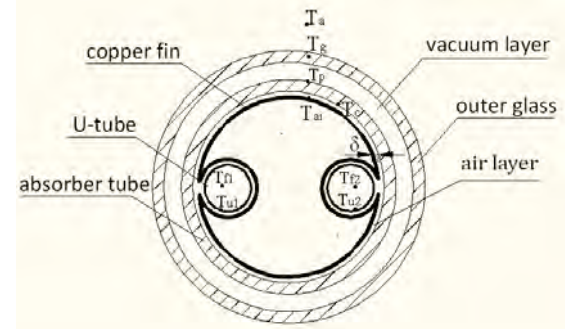
Two types of physical model of the individual U-type vacuum solar collector tube with copper fin are built. One type is that the absorber tube and the copper fin are tightly connected and the other is that there is air layer resistance between the absorber tube and the copper fin and also between the copper fin and the U-tube wall. The illustration of the U-type vacuum solar collector tube and its cross section are given in Fig 1. The inner and outer diameter of the solar collector tube is 37mm and 47mm, respectively. And the length of the collector tube is 1500mm and the air layer is assumed to be 1mm. The selective absorbing coating is deposited on the outer surface of absorber tube. And the air in the space between the two glass tubes is withdrawn, forming a vacuum. The water is used as the working fluid. Table 1 gives the main parameter of the solar collector tube.



(a) Axial structure of solar collector tube



(b) Cross section configuration with no air layer



(c) Cross section configuration with air layer

Figure 1 Geometry of U-type vacuum solar collector tube with copper fin

Table 1 Main parameter of solar collector tube

Material	Parameters	value	Unit
Outer tube	Thickness	2	mm
	Outer diameter	47	mm
	conductivity	0.74	W/(m·K)
Absorber tube	Outer diameter	37	mm
	Thickness	2	mm
	conductivity	0.74	W/(m·K)
Copper fin	Thickness	0.6	mm
	Conductivity	202	W/(m·K)

Air layer	Thickness	1	mm
	Conductivity	0.0204	W/(m·K)
U-tube	Thickness	0.7	mm
	Conductivity	398	W/(m·K)
	Outer diameter	8	mm
Absorbing coating	Absorptivity	0.92	
	Emissivity	0.08	

2.2 Heat transfer analysis

The thermal resistance network is given in Fig 2 to analyze the heat transfer in the solar collector tube. As we can see in Fig 2, one part of the solar energy absorbed by the selective coating is transferred to the working fluid by convection and conduction, the other part is lost to the ambient by radiation, conduction and convection. The net heat gain of the solar collector tube can be expressed as below.

$$Q_u = S - Q_L \quad (1)$$

where S is the solar energy amount absorbed by the selective coating and Q_L is the heat loss by conduction, convection and radiation. And Q_u is the net heat gain by the fluid. The over heat loss coefficient can be written as

$$U_L = U_t + U_e \quad (1)$$

where U_e is edge loss of the header tube. U_t is the heat loss coefficient from the absorber tube to the ambient through radiation, conduction and convection. U_t can be defined as below.

$$U_t = \frac{1}{\frac{1}{h_{ga}} + \frac{1}{h_{pgr} + h_{pgc}}} \quad (3)$$

where h_{ga} is the convection heat transfer coefficient from the outer glass tube to the surroundings. And h_{pgc} is the heat transfer coefficient due to the conduction of the bracket. And h_{pgr} is radiation heat transfer coefficient between the absorber tube and the outer glass wall.

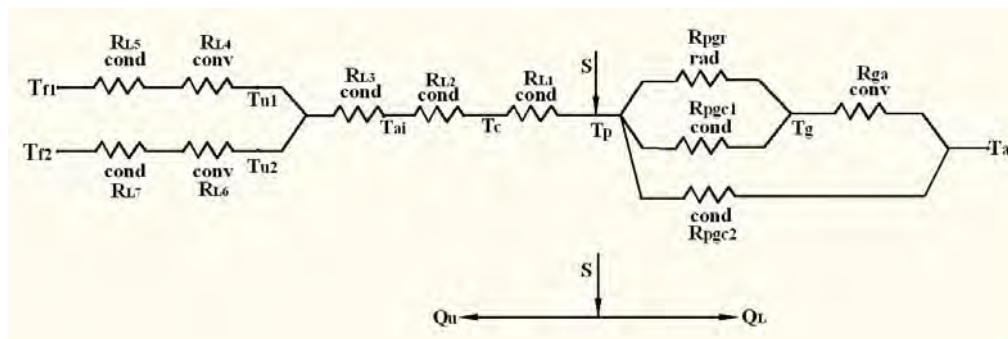


Figure 2 Thermal network of U-type vacuum solar collector tube with cooper fin

2.2 Governing equation and boundary condition

To analyze in detail the process of heat transfer from the absorbing coating to the working fluid and surroundings. The three-dimensional mathematical model is used to study on the thermal performance of

the U-type vacuum solar collector tube. The geometry and coordinate system is shown in Figure 3. In the present calculations, the Boussinesq approximation is adopted, so the physical properties of the fluid are assumed to be constant. Only the density in the buoyancy term of the vertical direction momentum equation is a variable of temperature. To simplify calculations without losing accuracy significantly, several assumptions are made. (1) The heat transfer process and fluid flow in the solar collector tube are steady. (2) The heat transfer coefficient for convection between the ambient and outer glass wall is constant. The governing equations are the three-dimensional, time-independent continuity equation, Navier–Stokes equations, and energy equation. In the Cartesian coordinates, they can be expressed as

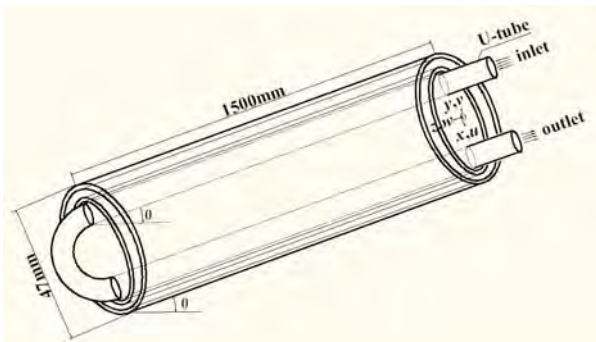


Figure 3 Geometry and coordinate system of collector tube

(1) Mass conservation equation

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \quad (4)$$

(2) Momentum conservation equation

$$\frac{\partial(\rho uu)}{\partial x} + \frac{\partial(\rho vu)}{\partial y} + \frac{\partial(\rho wu)}{\partial z} = -\frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (5)$$

$$\frac{\partial(\rho uv)}{\partial x} + \frac{\partial(\rho vv)}{\partial y} + \frac{\partial(\rho wv)}{\partial z} = -\frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + g \cos \theta (\rho - \rho_0) \quad (6)$$

$$\frac{\partial(\rho uw)}{\partial x} + \frac{\partial(\rho vw)}{\partial y} + \frac{\partial(\rho ww)}{\partial z} = -\frac{\partial p}{\partial z} + \mu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) + g \sin \theta (\rho - \rho_0) \quad (7)$$

(3) Energy conservation equation

$$\frac{\partial(\rho uT)}{\partial x} + \frac{\partial(\rho vT)}{\partial y} + \frac{\partial(\rho wT)}{\partial z} = \frac{\partial}{\partial x} \left(\frac{\lambda}{c_p} \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\lambda}{c_p} \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(\frac{\lambda}{c_p} \frac{\partial T}{\partial z} \right) \quad (8)$$

It can be seen from Fig. 2 that the solar energy absorbed by the selective coating is distributed to thermal losses through the outer glass and to useful energy gain. And the energy loss through the outer glass is the results of radiation between the inner glass and outer glass and the conduction due to the bracket. The radiation heat loss can be calculated by the discrete ordinates (DO) radiation model, which transforms

radiative transfer equation into a transport equation for radiation intensity in the spatial coordinates. In the direction $\vec{s}$, DO model can be written as

$$\nabla \cdot (I(\vec{r}, \vec{s}) \vec{s}) + (a + \sigma_s) I(\vec{r}, \vec{s}) = an^2 \frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r}, \vec{s}) \Phi(\vec{s}, \vec{s}') d\Omega' \quad (9)$$

where $\vec{r}$ is position vector, $\vec{s}$ is direction vector, $\vec{s}'$ is scattering direction vector, a is absorption coefficient, σ_s is scattering coefficient, n is refractive index, Φ is phase function, Ω' is solid angle, σ is Stefan-Boltzmann constant, T is local temperature, and I is radiation intensity, which depends on position $\vec{r}$ and direction $\vec{s}$.

The heat loss through the bracket is converted into the conduction of the gas layer which is supposed between the inner and outer glass. Using the thermal equilibrium equation

$$h_{pgc} A_a (T_p - T_g) = \lambda_{air} A_a \left(\frac{T_p - T_g}{\delta} \right) \quad (10)$$

The equivalent thermal conductivity of the gas layer can be written as

$$\lambda_{air} = h_{pgc} \delta \quad (11)$$

where h_{pgc} is the heat transfer coefficient due to the conduction of the bracket and is given to be 0.279 W/(m² · K) by experiment results [7]. And δ is the gas layer thickness.

According to the reference [7], the radiation energy absorbed by the selective coating is given as

$$S = I_T(\tau\alpha) \cdot 1.43DL \quad (12)$$

Due to the non-uniform of radiation, the practical radiation energy absorbed by the upper circumference of the collector is as:

$$I'_{T-UP} = \int_0^\pi I_T(\tau\alpha) \sin \theta \cdot RL \cdot d\theta = I_T(\tau\alpha) \cdot DL \quad (13)$$

According to equation (12) and equation (13), we can get that the radiation energy absorbed by the lower circumference of the collector tube can be defined as

$$I'_{T-low} = 0.43DL \cdot I_T(\tau\alpha) \quad (14)$$

In the setting of boundary condition, the solar radiation energy absorbed by the selective coating is regarded as inner heat source of the absorber tube. For the upper circumference of the collector, the intensity of inner heat source can be defined as

$$E = \frac{I_T(\tau\alpha)DL}{\pi DLb} \quad (15)$$

where b is the glass thickness, τ is the transmissivity of the glass, α is the absorptivity of the selective coating, D is the outer diameter of absorber tube, I_T is the solar radiation intensity exposed on the collector. The intensity of inner heat source acted on the lower circumference of the collector tube can also be written using the same method.

For the U-tube, inlet boundary conditions are

$$u|_{z=0} = 0, v|_{z=0} = 0, w|_{z=0} = w_0 \quad (16)$$

$$T|_{z=0} = T_0 = \text{const} \quad (17)$$

And outlet boundary conditions for U-tube are

$$\left. \frac{\partial u}{\partial z} \right|_{\text{outlet}} = 0, \left. \frac{\partial v}{\partial z} \right|_{\text{outlet}} = 0, \left. \frac{\partial w}{\partial z} \right|_{\text{outlet}} = 0 \quad (18)$$

$$\left. \frac{\partial T}{\partial z} \right|_{\text{outlet}} = 0 \quad (19)$$

Adiabatic boundary condition is applied on the wall of U-tube exposed in the external partition of vacuum glass tube (see Fig. 3).

$$-\lambda \left. \frac{\partial T}{\partial z} \right|_{\text{inlet-wall}} = 0 \quad (20)$$

$$-\lambda \left. \frac{\partial T}{\partial z} \right|_{\text{U-tube-bend}} = 0 \quad (21)$$

For cross section of inner glass tube and outer glass tube at $z=0$ and $z=1500$, the adiabatic boundary condition is also used.

$$-\lambda \left. \frac{\partial T}{\partial z} \right|_{z=0} = 0; -\lambda \left. \frac{\partial T}{\partial z} \right|_{z=1500} = 0 \quad (22)$$

Boundary condition of the outer glass wall and the ambient:

$$q_{\text{conv}} = h_{g-a}(T_g - T_a) \quad (23)$$

Adiabatic boundary condition of cooper fin and the air inner the collector:

$$-\lambda \left. \frac{\partial T}{\partial z} \right|_{\text{fin-wall}} = 0 \quad (24)$$

The FLUENT software is used to solve the above governing equations. And the SIMPLC algorithm is applied to resolve the pressure-velocity coupling. The residual error precise of the energy equation and momentum equation is 10^{-6} .

2.3 The thermal efficiency of the solar collector tube

The instantaneous efficiency of the solar collector is defined as the ratio of the heat gain by working fluid to the radiation energy in the aperture area of collector. The efficiency can be defined as below [8].

$$\eta = \frac{Q_u}{I_T A_0} \quad (25)$$

Where Q_u is the heat obtained by the working fluid, I_T is the radiation intensity, A_a is the area of the absorber tube, A_0 is the aperture area of collector and A_0 can be written as below.

$$A_0 = L \cdot W = L \cdot 2D \quad (26)$$

Where L is the length of the solar collector tube and W is the distance of adjacent collector tubes.

3 RESULTS AND DISCUSSION

3.1 Validation of the numerical model

To prove the accuracy of mathematical model of U-type vacuum solar collector tube, thermal efficiency of U-type vacuum solar collector tube is calculated by simulation method and the result is compared with that obtained by experiment [7] and that by others by numerical method [2]. Figure 4 gives the efficiency profiles of the U-type vacuum solar collector tube for the solar radiation intensity, $I_T=950\text{W/m}^2$. It is observed that the result is in accordance with the references [2,7], indicating that the three-dimensional mathematical model is reasonable and can be used to analyze the thermal performance of U-type vacuum solar collector tube. Figure 4 shows that there is a increasing error between the experiment result and simulation result with the increase of inlet temperature, because the conduction of header and bracket inner the glasses is ignored and the heat loss increases with the increase of inlet temperature.

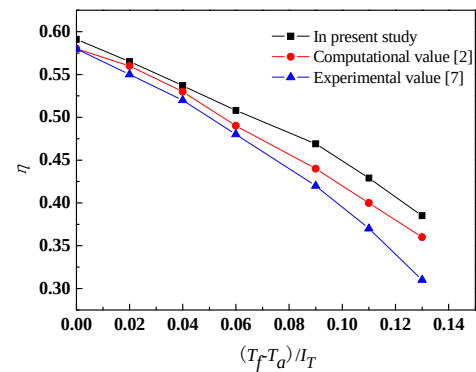


Figure 4 Thermal efficiency curve of the collector tube

3.2 Air layer effect

The thermal efficiency of U-type vacuum solar collector tube and surface temperature of absorbing coating for radiation intensity $I_T=1000\text{W/m}^2$ and inlet velocity $v=0.06\text{m/s}$ is given in figure 5 and figure 6. It can be seen from figure 5 that the thermal efficiency of U-type vacuum solar collector tube with air layer between the cooper fin and absorber tube is smaller by 6% than that of U-type vacuum solar collector tube without air layer. And the surface temperature of absorbing coating is 40°C higher compared with that of U-type vacuum solar collector tube without air layer in figure 6. The reason for the results above is that the increase of surface temperature of absorbing coating leads to the increase of the radiation heat loss and conduction loss and then the decrease of thermal efficiency of U-type vacuum solar collector tube. Figure 7 gives the temperature distribution of U-type vacuum solar collector tube in the two geometric constructions. It is observed that the fluid temperature increases constantly along the flowing direction and the inlet and outlet temperature difference is lower 0.6°C when considering the air layer than that of U-type vacuum solar collector tube without air layer.

It can also be seen that the thermal efficiency of U-type vacuum solar collector tube falls by 9% (in figure

5) and the surface temperature of absorbing coating of U-type vacuum solar collector tube enhances by 56°C (in figure 6) when inlet temperature of working fluid ranges from 20°C to 80°C . The increase of surface temperature of absorbing coating results in the increase of radiation and conduction heat loss of U-type vacuum solar collector tube.

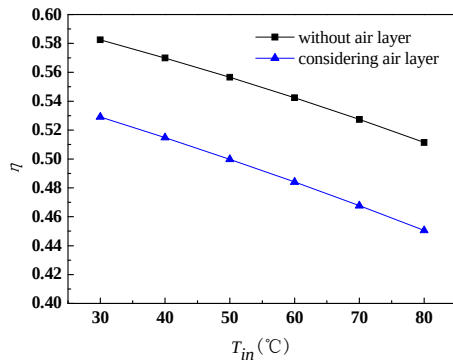


Figure 5 Influence of air layer on the thermal efficiency of collector tube

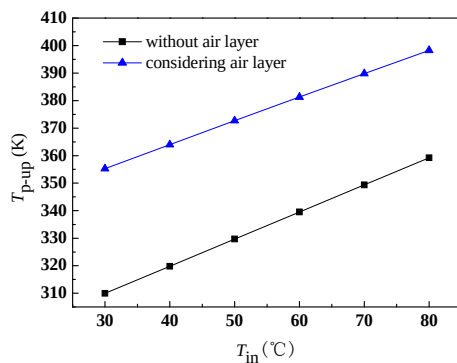
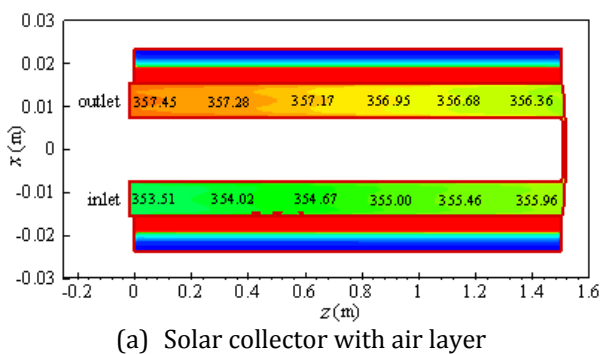
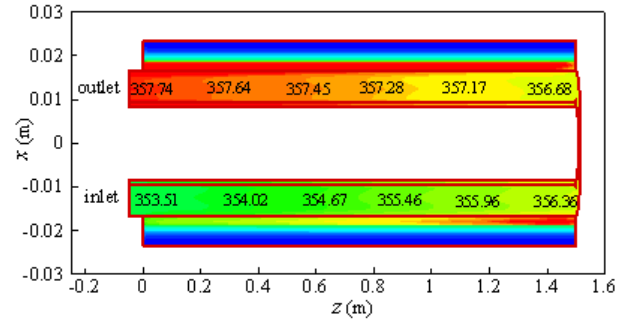


Figure 6 Influence of air layer on absorber surface temperature of collector tube



(a) Solar collector with air layer



(b) Solar collector without air layer

Figure 7 Axial temperature field of solar collector tube

3.3 Mixed convection effect

Figure 8 gives the variation of the heat gain of U-type vacuum solar collector tube with fluid inlet temperature in two occasions that the fluid in the U tube is pure convection and mixed convection for $IT=1000 \text{ W/m}$, $v=0.06 \text{ m/s}$. The temperature fields of U-type vacuum solar collector tube in mixed convection and in pure convection are given in figure 9. It can be seen that the heat gain of branch one of U-tube in mixed convection is higher than that of branch one of U-tube in pure convection, proving that the mixed convection reinforced the heat transfer in U-tube. However, the heat gain of branch two decreases in mixed convection compared with that in pure convection, which results in that the total heat gain of U-type vacuum solar collector tube is almost the same. The mixed convection in branch one of U-tube enhances the fluid temperature of flowing into the branch two of U-tube, so that the difference between the fluid and U-tube wall in branch two of U-tube decrease compared with that in pure convection and the heat transfer in branch two of U-tube is weakened. This is the reason why the mixed convection in U-tube does not improve the total heat gain of U-type vacuum solar collector tube.

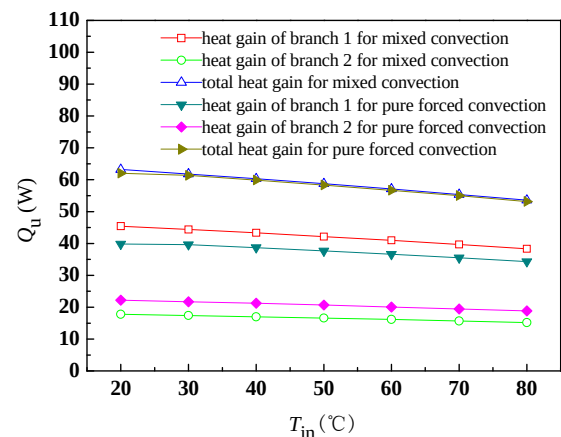
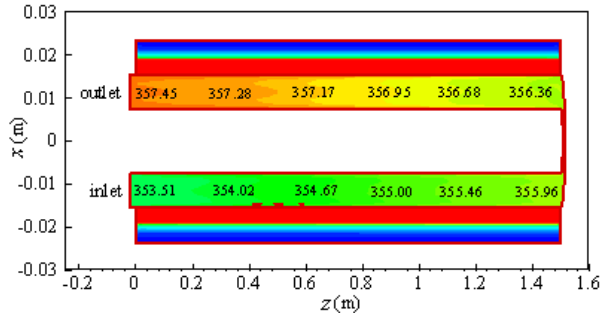
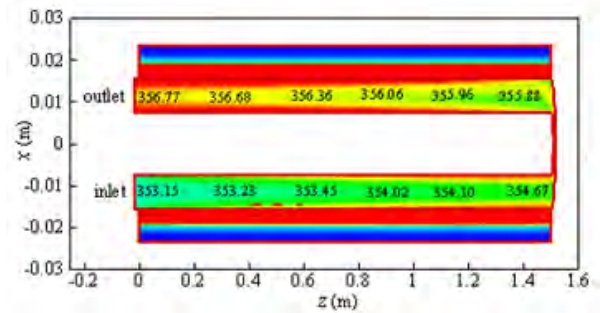


Figure 8 Mixed convection effect on the heat gain of the solar collector tube



(a) Mixed convection



(b) Pure forced convection

Figure 9 Axial temperature field of solar collector tube for mixed convection and pure convection

3.4 Main parameters effect

Figure 10 and figure 11 give the variation of inlet and outlet temperature difference and the heat gain of U-type vacuum solar collector tube with the inlet velocity of working fluid for inlet temperature $T_{in}=30^{\circ}\text{C}$, $I_T=1000\text{ W/m}^2$. It can be seen from figure 10 that the inlet and outlet temperature difference decreases with the inlet velocity of working fluid. The heat gain of branch one of U-tube in figure 11 decreases with inlet velocity of working fluid, because the increase of inlet velocity weakens the mixed convection in branch one of U-tube, which leads to the decrease of working fluid temperature flowing into branch two of U-tube and the increase of the temperature difference between working fluid and U-tube wall in branch two of U-tube. So the heat gain of branch two of U-tube increases with the increase of inlet velocity of working fluid and the total heat gain of U-type vacuum solar collector tube changes little.

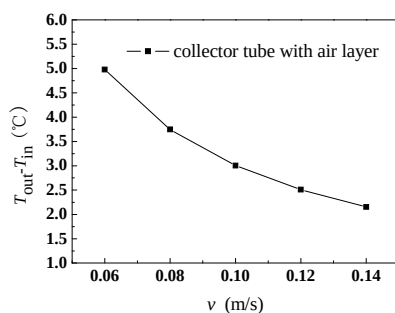


Figure 10 Variation of inlet and outlet temperature difference with inlet velocity

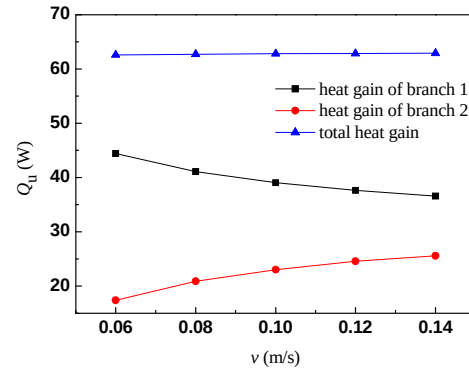


Figure 11 Variation of heat gain with inlet velocity

Figure 12 shows the effect of radiation intensity on the thermal efficiency of U-type vacuum solar collector tube for fluid inlet temperature $T_{in}=30^{\circ}\text{C}$ and $T_{in}=80^{\circ}\text{C}$ and $v=0.06\text{ m/s}$. It is observed that the thermal efficiency increases with the increase of radiation intensity and the thermal efficiency of U-type vacuum solar collector tube increases only 1% when fluid inlet temperature is 30°C , while the thermal efficiency of U-type vacuum solar collector tube increases about 8%. The radiation intensity has stronger influence on the thermal efficiency of U-type vacuum solar collector tube in high inlet temperature of working fluid.

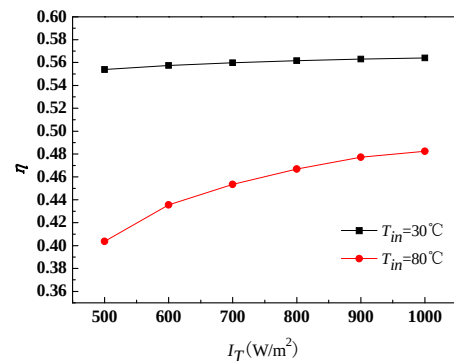


Figure 12 Effect of radiation intensity on thermal efficiency of solar collector tube

4. CONCLUSION

In this paper, the thermal performance of single U-type vacuum solar collector tube with copper fin is investigated in three-dimensional method numerically. According to the study of air layer and mixed convection in the solar collector tube, the results indicate that the air layer resistance leads to the decrease of the thermal efficiency by 6% and the increase of absorber surface temperature by 40°C . The mixed convection in the tube has no influence on the thermal efficiency of the solar collector tube, but affects the temperature field and flowing field greatly. The heat gain of the solar collector tube decreases with the increase of fluid inlet temperature, however, it changes little with the increase of fluid inlet velocity. And the radiation has stronger effect on the thermal efficiency in

high fluid inlet temperature. The study for the thermal performance of individual U-type vacuum solar collector tube with copper fin lays the foundation for optimizing operation and structural improvement of it.

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- Appendices

LITERATURE REVIEW AND RESEARCH NEEDS TO COUPLE BUILDING ENERGY AND AIRFLOW SIMULATION

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ABSTRACT

We have conducted a comprehensive literature review on motivations and methods for linking computational fluid dynamics (CFD) to building energy simulation (BES). The review begins with a focused evaluation on where and why a coupled simulation capability is needed. We then summarize methods for exchanging information, data, and model output between the CFD and BES tools, and comment on the suitability and performance of various data exchange strategies. We follow this with a review of possible software architectures employed for data exchange. Finally, we present typical use cases for the coupled simulation, and discuss critical research gaps for the coupled simulation.

Keywords: CFD; BES; Integrated Simulation.

1. INTRODUCTION

Due to the assumptions associated with simple multizone airflow models that are employed in typical BESs, most BES software may not provide satisfactory results for some applications. For instance, EnergyPlus, a common BES software, may not perform well in predicting localized comfort, the contaminate distribution within an occupant zone, or the control of HVAC systems (Crawley, Hand et al. 2008). In addition, multizone airflow models are often not suitable to simulate non-uniform airflow distribution within a zone, such as flow for displacement and natural ventilation. Moreover, it is challenging for a BES software to predict the correct convective heat transfer coefficient (CHTC). Most BESs estimate the CHTC with empirical formulas assuming the room air in room is instantaneously mixed. As shown by (Lomas 1996), the difference between the estimated annual heating energy using four different empirical formulas could differ by as much as 27%. These potential limitations of BESs may be overcome using CFD methods.

On the other hand, the CFD also has its own technical limitations. The CFD usually uses idealized static boundary conditions, such as fixed supply airflow rate and temperature, fixed wall temperature or heat flux through the wall. However, the actual boundary conditions are changing with weather condition and operation schedule of HVAC system that must be obtained from a BES (Djunaedy, Hensen et al. 2003).

As a result, it is possible to couple the BES and the CFD to provide missing information when a single program is not sufficient. This report reviews literatures on coupling the CFD with BES. The objective is to summarize the state-of-art research methods and provide guidance for the future research.

2. EXCHANGED DATA

This section is to summarize methods that have been reported to exchange information, data, and model predictions between BES and CFD tools, and discussed the benefits of the resulting simulation performance.

The information commonly transferred from a BES to a CFD include interior surface temperature of walls (Zhai, Chen et al. 2002; Zhang, Lam et al. 2012), boundary conditions at openings, including outdoor airflow velocity, pressure and temperature (Wang and Wong 2008; Ohba and Lun 2010), heat flux or air-conditioning load (Zhai, Chen et al. 2002; Fan and Ito 2012), supply airflow rate, temperature, humidity or pressure at supply diffuser (Ascione, Bellia et al. 2012; Fan and Ito 2012), and temperature or pressure at the outlet (Ascione, Bellia et al. 2012; Fan and Ito 2012). The possible information that may be transferred from the CFD to BES include: Indoor temperature distribution (Zhai, Chen et al. 2002; Ascione, Bellia et al. 2012), CHTC (Rong, Da et al. 2011; Zhang, Lam et al. 2012), flow rate through the openings, such as window, doors, outlet of the atrium (Wang and Wong 2008; Pan, Li et al. 2010; Zhang, Lam et al. 2012), and concentration of pollutants (Wang and Chen 2008; Goldsworthy 2012).

The selection of exchanged data can also impact the convergence, stability and computing speed of the coupled simulation. As concluded by Zhai and Chen (2005), sending the enclosure interior surface temperature from BES to the CFD and return the CHTC and indoor air temperature gradients to the BES can significantly improve the accuracy of the simulation.

3. DATA SYNCHRONIZATION STRATEGIES

We divide the strategies for data synchronization between the CFD and BES into three major categories, and describe them in subsection 3.1 to 3.3.

3.1. Static Synchronization Strategy

In the static coupling strategy the CFD exchanges data with the BES for only a few times, whether manually or automatically, in two methods: one-way and two-way exchange.

One-way static synchronization

The BES transfers data to the CFD at some specific time. Figure 1 shows the workflow of one-way static synchronization. The rectangle represents CFD and the ellipse represents the BES. The BES runs continuously while the CFD runs and receives the data only at a few moments. Since the CFD needs significantly more computing time than the BES, the one-way static synchronization can greatly reduce the total simulation time of a coupled simulation (Zhai, Chen et al. 2002). This method can be used when two programs are somewhat insensitive to the timing and the amount of exchanged data.

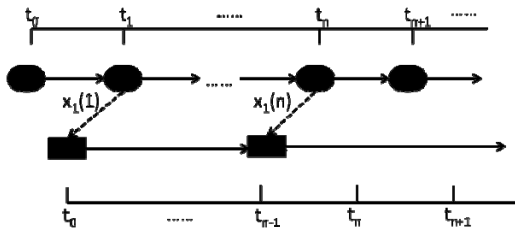


Figure 1 One-way static synchronization

Two-way static synchronization

The major difference between the one-way static synchronization and the two-way static synchronization (Figure 2) is that the later also sends the output of CFD to the BES (Zhai, Chen et al. 2002). The “feed-back/feed-forward” exchange is conducted in order to update the CHTC in the BES if it is significantly different from that calculated by the CFD.

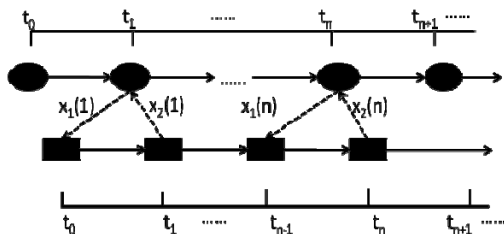


Figure 2 Two-way static synchronization

3.2. Dynamic Synchronization Strategy

By dynamic synchronization strategies two programs exchange data with each other at each predefined time step which indicates the communication frequency. Dynamic synchronization

strategy exchanges data more frequently than the static ones so that the data synchronization typically has to been implemented automatically. We can further divide the dynamic synchronization into quasi-dynamic synchronization and full-dynamic synchronization.

Quasi-dynamic synchronization strategy

Quasi-dynamic synchronization (Zhai, Chen et al. 2002) is also called loose coupling (Trcka, Wetter et al. 2007) or “ping pong” coupling (Trcka and Hensen 2006). It requires the two programs to conduct one exchange of data at every time step. The quasi-dynamic strategy has two sub-categories: loose quasi-dynamic strategy and cross quasi-dynamic strategy (Trčka, Hensen et al. 2009). In the loose quasi-dynamic strategy (Figure 3), the BES transfers data to the CFD which in turn runs the former time step and returns its output to the BES. In the cross quasi-dynamic strategy (Figure 4), the BES and the CFD run simultaneously and exchange data at the end of each time step. The received data are used for the next time step.

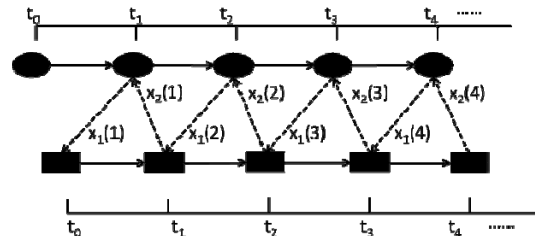


Figure 3 Loose quasi-dynamic synchronization

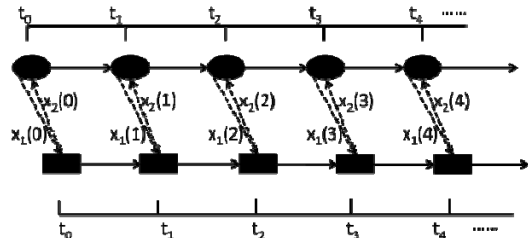


Figure 4 Cross quasi-dynamic synchronization

Fully dynamic synchronization strategy (Zhai, Chen et al. 2002) is also called onion coupling (Trcka and Hensen 2006) or strong coupling (Trčka, Hensen et al. 2009). It requires two programs to conduct multiple data exchanges within every time-step until convergent solutions are achieved for both simulations (Figure 5). Compared to the quasi-dynamic synchronization, the fully dynamic synchronization is more accurate but significantly slower, which can be a severe limitation for many applications (Zhai and Chen 2003).

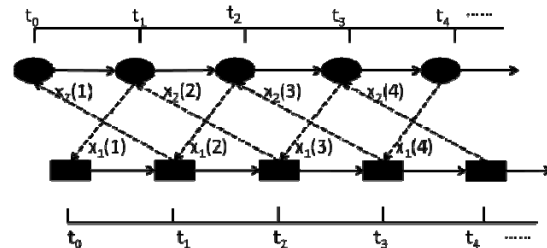


Figure 5 Fully dynamic synchronization

3.3. Bin Synchronization Strategy

The bin synchronization strategy (Zhai and Chen 2005) is also called virtual dynamic strategy (Zhai, Chen et al. 2002). As shown in Figure 6, it integrates indirectly the BES with the CFD through an intermediary, such as a database (Zhai, Chen et al. 2002), a neural network (Rong, Da et al. 2011), or a fitted formula (Pan, Li et al. 2010; Hiyama and Kato 2011; Zhang, Hiyama et al. 2013). The intermediary is established based on pre-computed CFD simulations in typical scenarios. For some typical scenarios the CFD will iterate with the BES, and the results will be transferred to a database, with which the BES then is coupled to simulate the building energy consumption. With no CFD run during the coupling, the bin synchronization strategy is computationally faster than the dynamic synchronization. However, the accuracy of the bin synchronization strategy may significantly drop if the flow conditions are outside the range of pre-computed CFD simulations (Zhai, Chen et al. 2002).

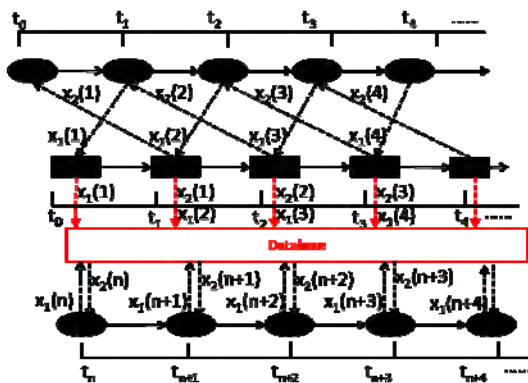


Figure 6 Bin synchronization

4. SOFTWARE ARCHITECTURE

The software architecture focuses on how to implement the software coupling. In general there are two methods: internal coupling and external coupling.

4.1. Internal Coupling

The internal coupling approach adds physical models of BES, such as the envelope heat transfer model and the envelope radiation model, into the CFD. As a result, the new integrated CFD program solves all the governing equations simultaneously until it reaches a converged solution. There are two major limitations in the internal coupling approach (Djunaedy, Hensen et al. 2003):

Solving heat transfer through the envelope by the CFD is computationally demanding due to the different time scales in the physical process. The heat transfer through the envelope may take hours while the airflow changes happen in a few seconds.

It is difficult to obtain a converged result due to the difference in the stiffness of fluid and solid equations (Chen, Peng et al. 1995).

4.2. External Coupling

Two methods for external coupling are reported in the literature. One is called discontinuity mechanism, which is defined as “exchanging data between two programs sequentially, where a model preprocessor transforms the output of one program into the input for a slave program after the master program completes its simulation” (Djunaedy, Hensen et al. 2005). The other is a continuity mechanism, by which two programs are called separately and run in parallel (Trčka and Hensen 2006). The external coupling has at least four advantages as to the internal coupling (Djunaedy, Hensen et al. 2005):

it is much faster than the internal coupling (Djunaedy, Hensen et al. 2004). Since it is not necessary to iterate two programs to converge at each time step, the run time of coupled simulation can be dramatically reduced.

there is no computational stiffness as the two programs run separately.

it can take the advantage of the state-of-art technology in either program as there is no need to rewrite the code.

the program can be optimized individually in order to solve some specific problems.

To implement data exchange by external coupling, researchers have developed different methods, such as using a self-developed interface for direct coupling, a data exchange platform, and a standard interface. The main function of the self-developed code is to transform the output from one program into a recognizable pattern for the other program (Liping and Hien 2007; Fan and Ito 2012).

The data exchange platform allows programs to exchange data with other programs after connected to the platform. (Trčka, Hensen et al. 2009) presented several data exchange platforms for co-simulation of building performance. A popular approach, the Building Controls Virtual Test Bed (BCVTB) (Wetter 2010) is one of the more advanced methods for building performance simulation. However, it does not provide any links to commercial CFD software. It is necessary to develop a code to bridge the CFD and BCVTB, such as FLOW+ which is used to connect the CFD program FLUENT to the BCVTB (Zhang, Lam et al. 2012).

Two standard interfaces are developed for coupling CFD and the BES: Functional Mockup Interface (Blochwitz, Otter et al. 2011) and building product model based on International Standard Organization standard (Lydon, Keane et al. 2005).

5. USE CASES

The coupled simulation of BES and CFD can be used for various applications. It was used to evaluate the performance of the advanced indoor ventilation method, such as personalized ventilation, displacement ventilation, and natural ventilation (Zhang, Lam et al. 2012). It was adopted for the design of the advanced air-conditioning methods, such as underfloor heating with a top return (Fan and Ito 2012) and local thermal

environment control (Steeman 2009). Moreover, it was applied for the study of thermal performance of double-skin facades (Zeng, Li et al. 2012), double-skin wooden roof (Villi, Pasut et al. 2009), and membrane (Devulder, Wilson et al. 2007) where flow within and around complex geometry is involved. Furthermore, (Goldsworthy 2012) used it to investigate mechanical ventilation for smoke control and (Bouyer, Inard et al. 2011) applied it to improve the microclimate condition around the building for BES.

6. CONCLUSION

By coupling the CFD and BES, we can obtain more complementary and accurate information about the indoor environment and building energy performance than single program alone. The exchange data can be synchronized by mainly three ways: static coupling, dynamic coupling and bin coupling. These strategies vary in time cost, simulation speed and accuracy. To implement the data exchange between the two programs, internal and external coupling are feasible. External coupling, which exchanges data externally out of programs by a self-developed code, a standard interface, or a data exchange platform, is preferred for it is faster and easily converged performance.

The coupled simulation can achieve satisfactory simulation for some applications. However, the coupled simulation still needs to be improved for the design and performance evaluation of indoor environment control. For instance, model-based HVAC control to prevent the spread of smoke when there is a fire emergency in buildings. This requires a fast computing speed for indoor environment simulation and dynamic simulation capability for HVAC control. The fast fluid dynamics (Zuo and Chen 2009) model is about 50 times faster than the CFD and can provide sufficient results for the smoke control. The Modelica models are dynamic and well-suitable for HVAC control (Wetter, Zuo et al. 2013). Coupling the FFD and Modelica may enable a fast and dynamic simulation for the indoor environment control.

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SIMULATION STUDY ON SPRAY CHARACTERISTICS FOR EVAPORATIVE CONDENSER

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ABSTRACT

Evaporative cooling is a highly efficient way of heat exchange, which can reduce the energy consumption of air conditioning units, it has been widely used in petroleum, chemical, air conditioning refrigeration, medicine, food and other fields. However, in the actual operating conditions, the simulation study of air - water on convective heat and mass transfer is still a few, the research information and data is scarce, which need for in-depth research. In this article, numerical simulation method is used to study the effect of the nozzle spacing and spray water speed on the evaporative condenser, and the results show that: when the nozzle spacing is 200mm and spray water speed is 1.8m/s, the effect of heat transfer on evaporative condenser is better.

Keywords: Evaporative condenser; Nozzle spacing; Spray water speed; Simulation study

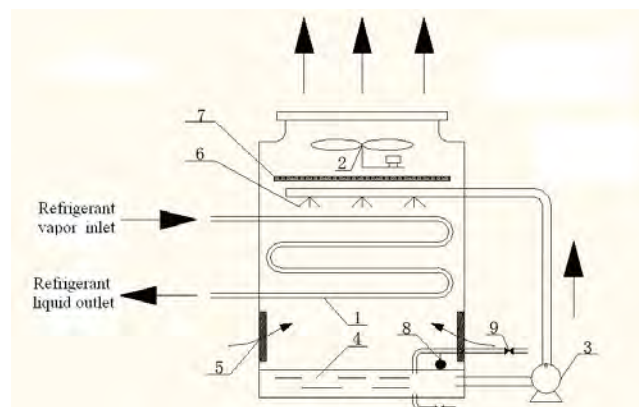
1. INTRODUCTION

In China, the air conditioning is a large energy-consuming in buildings, according to the statistics show that, air conditioning energy consumption accounts for about 50%~60%^[1] in the energy consumption of the whole building, and there is a rising trend, therefore it is very important to reduce the energy consumption of the air conditioning units. The condenser as an important component equipment of the refrigeration system is a key components to achieve the energy conversion, which account for a large proportion in power consumption and investment. Therefore, enhanced heat transfer and energy saving of the condenser has become a hot research topic. Evaporative condenser has been widespread concerned by the researchers^[2~3] for its unique form of heat transfer, higher heat transfer efficiency, saving energy and water, etc.

2. WORKING PRINCIPLE OF EVAPORATIVE CONDENSER

The evaporative condensers is mainly composed by three parts, that are heat exchanger, water circulation system and blower, which the structure shown in Figure 1. By the pump 3 and through the nozzle 6, cooling water sprayed the surface of the heat transfer coil and formed water film, the temperature of water film rising after absorbed the heat of the refrigerant vapor in tube, and direct contact with the rapid flow of air in the outside of the tube, which the process is latent heat-based and sensible heat -supplemented. The evaporated water exhaust with the air and take away the heat, no evaporated water flows to the lower tube bundle, which

gathered in the water tray, then recycled by the pump to the nozzle.

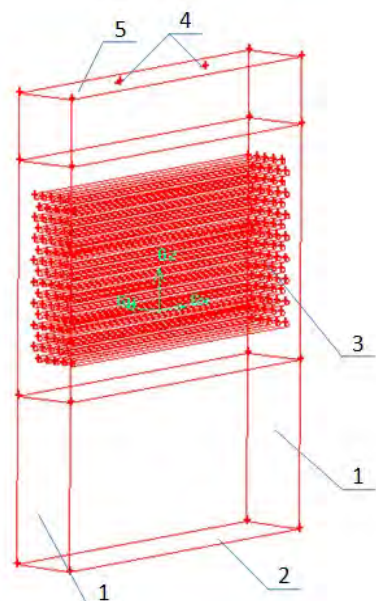


1- Heat transfer coil; 2- Axial fan;
3- Centrifugal pump; 4- Water tray; 5- Grill;
6- Nozzle; 7- Flap; 8- Float valve; 9- Valve
Figure 1 Evaporative condenser diagram

3. NUMERICAL SIMULATION AND RESULTS

Analysis of the evaporative condenser

The structure size of circular tube that evaporative condensers uses is $\Phi 8 \times 1$ mm, due to the heat and mass transfer process of evaporative condensers is very complex, it is very difficult to adequately describe the flow characteristics by using the two-dimensional mode, so using a three-dimensional model, the model structure is shown in Figure 2, Model box is rectangular, which symmetrically to the center position of the coordinates of the origin (0, 0, 0).



1-Air inlet;2- Spray water exports;3- Heat transfer coil;4- Nozzle (spray water inlet);5- Air outlet
Figure 2 Geometric model of the evaporative Condenser

Simulated conditions: The amount of spray water is 0.08kg/s, the amount of air is 0.14m³/s, wind speed is 3m/s, nozzle spacing respectively is 100mm, 150mm, 200mm and 250mm, spray water speed respectively is 1m/s, 1.2m/s, 1.5m/s, 1.8m/s and 2.0m/s.

To make uniform airflow on both sides of the heat exchanger at the air inlet, a stabilization stage is set in the air flow inlet section and the coil section^[4], which the height is 0.05m.

3.1. The effect of nozzle spacing on heat transfer performance of evaporative condenser

When the nozzle spacing is 100mm, 150mm, 200mm and 250mm, simulation the effect of nozzle spacing on heat exchange of evaporative condenser, the results shown in Figure 3:

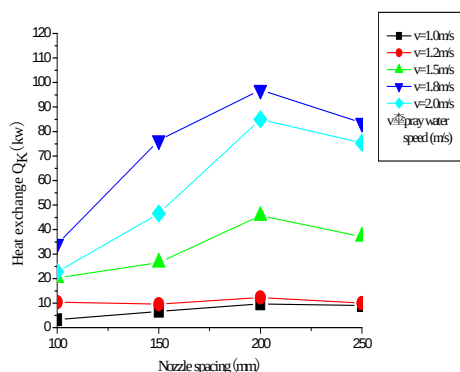


Figure 3 The relationship between heat exchange and nozzle spacing

Figure 3 shows that, with the nozzle spacing increases, the heat exchange first increases then decreases. If the nozzle spacing is too small, on the one hand, the overlapping area of the spray water spraying on the heat transfer coil is increased, on the other hand, the number of nozzles is more, this will increase the pressure drop which the air flow through the nozzle and causing the increase of fan load; If the nozzle spacing is too large, it will lead to dry spots phenomenon on the surface of heat transfer coil. Therefore, there is an optimum value for nozzle spacing, When the nozzle spacing is 200 mm, the effect of heat transfer on evaporative condenser is better.

3.2. The effect of spray water speed on heat transfer performance of evaporative condenser

When the spray water speed is 1m/s, 1.2m/s, 1.5m/s, 1.8m/s and 2.0m/s, simulation the effect of spray water speed on heat exchange of evaporative condenser, the results shown in Figure 4:

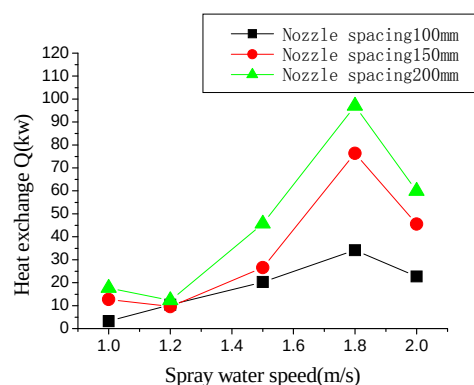
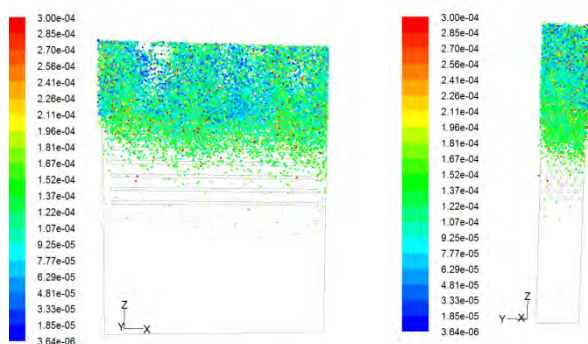


Figure 4 The relationship between heat exchange and spray water speed

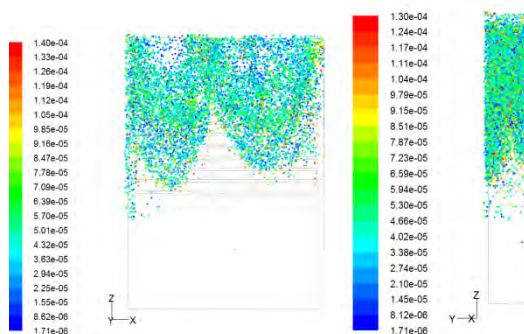
Figure 4 shows that, the best matching value between the heat exchange and spray water speed is exist, When the spray water speed is 1.8m/s, the effect of heat transfer on evaporative condenser is better.

3.3. The trajectory of droplets when nozzle spacing respectively is 150mm and 200mm

Based on the existed simulation relationship between heat exchange and nozzle spacing, when the nozzle spacing respectively is 150mm, 200mm, simulated the trajectory distribution of droplets, the results shown in Figure 5:



Sectional $y = 0$ Sectional $x = 0$
(a) Nozzle spacing is 150mm



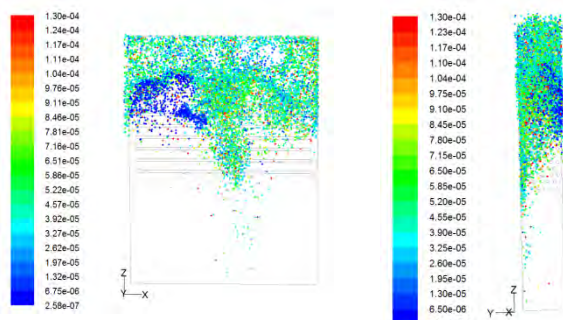
Sectional $y = 0$ Sectional $x = 0$
(b) Nozzle spacing is 200mm

Figure 5 The trajectory distribution of droplets when the nozzle spacing respectively is 150mm, 200mm

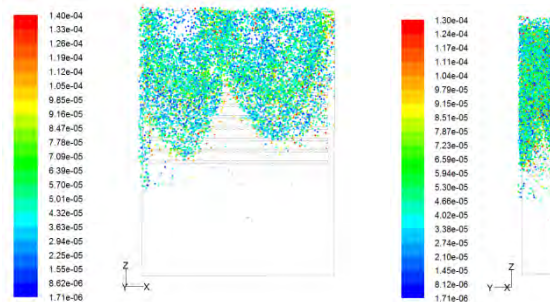
Figure 5 shows that, when the nozzle spacing is 200mm, spraying water can be uniformly covered the surface of the heat transfer coil, thereby enhancing the heat transfer, which is same as the variation of heat exchange from the Figure 3.

3.4. The trajectory of droplets when spray water speed respectively is 1.5m/s、1.8m/s and 2.0m/s

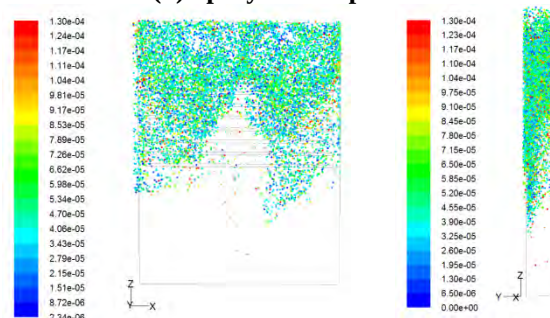
Based on the existed simulation relationship between heat exchange and spray water speed, when the spray water speed respectively is 1.5m/s、1.8m/s and 2.0m/s, simulated the trajectory distribution of droplets, the results shown in Figure 6:



Sectional $y = 0$ Sectional $x = 0$
(a) Spray water speed is 1.5m/s



Sectional $y = 0$ Sectional $x = 0$
(b) Spray water speed is 1.8m/s



Sectional $y = 0$ Sectional $x = 0$
(c) Spray water speed is 2.0m/s

Figure 6 The trajectory distribution of droplets when the spray water speed respectively is 1.5m/s, 1.8m/s and 2.0m/s

As shown in Figure 6, when the spray water speed is 1.8m/s, the distribution of droplets is ideal, thereby enhancing the heat transfer, which is same as the variation of heat exchange from the Figure 4.

4. Conclusion

By numerical simulation, the obtained conclusions as follows:

1) There is an optimal value between nozzle spacing and the heat transfer of evaporative condenser, When the nozzle spacing is 200mm, the effect of heat transfer on evaporative condenser is better, which the heat exchange is 96.99kw.

2) when the spraying water speed is 1.8m/s, Droplets can be uniformly covered the surface of the heat transfer coil l, which enhanced the heat transfer

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STUDY OF THE PERFORMANCE OF THERMOELECTRIC MODULES FOR USE IN BUILDING INTEGRATED PHOTOVOLTAIC AND THERMOELECTRIC HEAT PUMP SYSTEM

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ABSTRACT

Building integrated photovoltaic (PV) and thermoelectric (TE) system (BIPV/TE) are a new enclosure technology which achieve the electrical and thermal comprehensive utilization of solar energy. In BIPV/TE systems, the PV system transfers solar energy into electrical energy, which is directly used to power TE system to cool the PV panel, so as to maintain the PV in a lower temperature for high electrical efficiency; meanwhile, the TE modules absorb the heat from the PV panel and through heat pump effect to get a higher temperature thermal output for indoor heating. In the current stage of the study, a 600x600mm BIPV/TE prototype wall system was constructed by using commercial TE modules, and carried out tests on the performances of single TE module in off-design condition states. On this basis, a series of experiments were carried out to determine the performances of TE system under different voltages, different temperature of air channel, and in a step change of voltage. Based upon the measured data, the most suitable voltage and current, the relation between TE modules and PV system, and the thermal insulation of BIPV/TE wall are discussed.

Keywords: Photovoltaic; Thermoelectric; Building integrated; Comprehensive utilization

1. INTRODUCTION

Building integrated photovoltaic thermal (BIPV/T) systems have been growing with the rapid development over past decades and have the potential to become a major source of renewable energy in the urban environment [1]. Compared to installing stand-alone photovoltaic (PV) modules into a building which have been illustrated by Kimura [2] and Taleb and Pitts [3], BIPV/T system produces thermal and electrical energy simultaneously, which has higher overall output than that of two separated PV and solar thermal systems and potentially reduces the effective system costs. Besides, the merit of a BIPV / T system lies in the reduction of the demands on physical space and the equipment cost through the use of common frames and brackets [4-5].

The main part of a BIPV/T system is a PV / T collector. Up to date, many PV / T collectors have been designed to remove the waste heat and used as a source of energy for building applications [6-8]. Most of the above studies were using air and water for the cooling of PV and found water was better than air in terms of PV cooling. But generally, only when water is heated to a higher temperature (above 40°C), can it be effectively utilized, so the cooling effect of water for the PV will be inevitably reduced [9]. The temperature of the refrigerant in the evaporator of a heat pump system is lower and shows a small range of wave, so to

use refrigerant in cooling of PV cells can not only maintain the lower temperature of the PV cells to improve the conversion efficiency of the cells, but also absorb the heat on the base plates of PV cells to achieve the utilization of heat [10-11]. However, this kind of vapor compression heat pump system requires the components like compressor, expansion valve, evaporator, condensers, so it is too bulky and difficult to achieve a better organic combination of PV/T systems and building envelopes as well as the integration of buildings and solar application. It is worth noting that the solar PV systems can generate DC power with a cost of only 0.7 yuan / kWh, but if the DC power is exported to the photovoltaic grid-connecting inverter and converted into AC power and finally transmitted into the electric grid, the cost of grid-connecting power generation will be increased to 2 yuan / kWh or so [12].

Different from the vapor compression heat pump systems, the TE module can work directly using the DC power generated from PV system. Moreover, it has no mechanical moving parts, and does not need working fluids or utilizes any moving parts. Because of the advantages such as high reliability, flexibility in packaging and integration, and low weight, the TE system are regarded as clean and active cooling methods which have been widely used in military, aerospace, instrument, and industrial products [13-14]. In this paper, TE and PV system are integrated into

building envelope to form a new type BIPV/TE system. In the BIPV / TE system, the PV system generate electric energy to directly drive the TE heat pump to cool the PV system themselves, so as to effectively reduce the temperature of the PV cells, and increase the power-generating efficiency of the PV cells; at the same time, the TE heat pump absorb the energy of heat from the base plates of the PV cells, and then enhance the energy through heat pump effect to achieve the indoor heating, and achieve the solar-electricity and solar-heat comprehensive utilization of solar energy.

In the previous research, the project teams carried out researches on the performance of TE systems [15], and they also launched systematic researches on the application of TE systems in the waste heat recovery of buildings at the same time, and so far, TE domestic-ventilator, TE heat pump dryers, TE heat pump water heaters and other products adopting waste-heat-recovery technology have been developed [16-17]. In other experimental studies on TE modules power by a PV system, Dai et al. [18] have studied the relationship between the coefficient of performance of a TE refrigerator, and the intensity of solar radiation. Sabah A. Abdul-Wahab et al [19] fabricated and tested a portable solar TE refrigerator unit, and found that system performance was strongly dependent on the temperature difference of hot and cold sides between the TE modules. N.Rahbar et al [20] design a new type portable TE solar still. XuXu et al [21] fabricated and tested an active building envelope window system. Although these studies also combined TE and PV Technologies similar to our research, the proposed systems are not considered to use TE heat pump to cool PV base plates, so as to achieve the comprehensive and high-efficiency solar utilization.

In conventional TE system, the TE modules are worked in design condition states, however, In BIPV/TE system, the PV system transfers solar energy into electrical energy, which is directly used to power TE system, the voltage and current are affect by the solar radiation and show a big range of wave, the TE system need work in off-design condition states under different temperature of air channel temperature in long time. In the current stage of the study, a 600x600mm BIPV/TE prototype wall system was constructed by using commercial TE modules, and carried out tests on the performances of single TE module in off-design condition states. On this basis, a series of experiments were carried out to determine the performances of TE systems under different voltages, different temperature of air channel, and in a step change of voltage. Based upon the measured date, the most suitable voltage and current, the relation between TE module and PV system, and the thermal insulation of BIPV/TE wall are discussed.

2. BIPV/TE WALL SYSTEM

As shown in Figure 1, the BIPV/TE wall system from the outside to the inside consists of photovoltaic cell, outer air channel, insulation layer, inner air channel, heat sink, TE modules, indoor radiant panel,

and air channel switches, etc. The TE modules are connected in series and sandwiched between the radiant panels and heat sinks. By controlling the switches of the air channels, a BIPV / TE wall system can achieve the functions of heating, cooling, ventilation and heat dissipation, and its working principle is as follows:

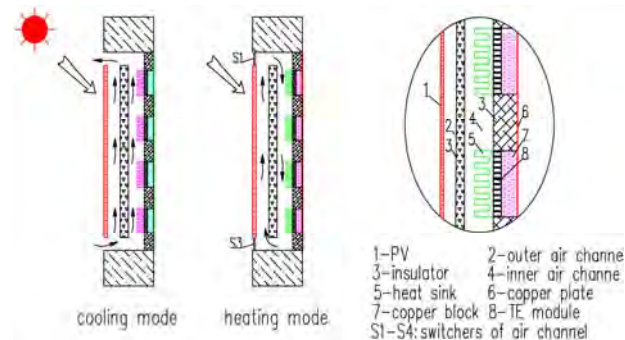


Figure 1 Schematic diagram of new type of BIPV / TE wall system

In heating mode, air channels switchers s1 and s3 are closed, while air channels switchers s2 and s4 are opened, the PV system converts a part of solar energy into electricity, and most of which converts into heat to the outer air channel s2 and inner air channel s4 through the PV base plates, and the PV cells generate electricity to directly drive the work of the TE system, at the same time, the cold-side heat sinks on the outdoor side of the TE system can just recover the heat generated by the PV base plates, so while lowering the temperature difference of the TE modules and improving the efficiency of the TE system, the PV modules can work at lower temperatures, so as to achieve the solar-electricity and solar-heat comprehensive utilization of solar energy.

In cooling mode: all of the air channels are opened on to dissipate heat, to change the direction of current input the TE system and use the PV cell to drive TE system to achieve cooling; in winters, when solar light is weak, all of the air channels are closed, so that the wall insulation can be achieved; in the summer nights, when the TE system does not work, all of the air channels are opened to achieve the ventilation and cooling.

3. EXPERIMENTAL RESEARCHES AND CALCULATING METHODS

3.1 Preparation for a BIPV / TE wall experiment

A 600X600mm performance test platform on which nine TE modules are fixed for BIPV / TE prototype wall system is constructed as shown in Figure. 2. The wall is composed of a radiation copper plate, a copper block, TE module, heat sink and insulation materials, and a constant temperature water tank is adopted to simulate the heat dissipation of PV cells. TE modules used in this study were manufactured by hualeng, china. Size of TE modules was 40 x 40x3.8mm, with 127

thermoelectric couples of bismuth telluride and ceramic surface, type of 12706.

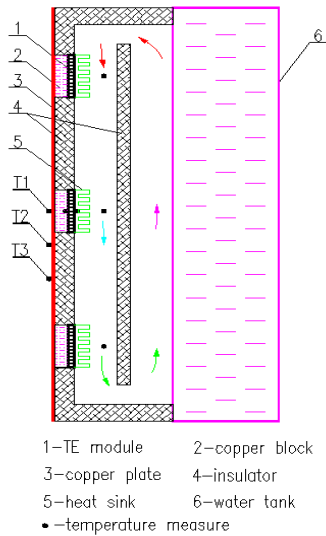


Figure 2 BIPV / TE prototype wall testing system

The experiments include testing of single TE module in off-design condition states and testing TE system performance for use in BIPV/TE system. The former experiments including three groups of sub-experiments: (1) Single TE module's performances under off-design voltages; (2) Single TE module's performances under off-design dramatic changes voltages.

In order to proceed with the full scale testing of the BIPV/TE system prototype, a more detailed study of the TE subsystem is needed. And then a series of experiments are carried out to test the performance of the nine TE modules in the test platform, namely: (1) different voltages; (2) different air channel temperatures; (3) Step-change voltages and longer time. The connections between the TE modules mainly include two methods shown in Figure 3. One method is serial connections between the 9 TE modules, another method is parallel-serial connections between the 9 TE modules, and considering the actual situation of photovoltaic cells voltage and current, the method of parallel-serial connections is adopted in this experiment.

The data collected in the experiment mainly include the cold-side (T_c) and hot-side temperatures (T_h) of the TE module, air channel temperatures (T_s), temperature of the radiation copper plate (T_1, T_2, T_3), the inputs of operating voltage and current. All temperature values are tested and recorded with a paperless recorder and a PT100 temperature sensor, and the accuracy of this temperature PT100 is $\pm 0.1^\circ\text{C}$, and the specific arrangement of testing points is shown in Figure 1; The input operating voltages and currents of the TE module are read through the DC power display.

3.2 Calculation of thermoelectric heat pump performance

Provide sufficient detail to allow the work to be reproduced. Methods already published should be indicated by a reference: only relevant modifications should be described.

$$Q_c = SI(T_c + 273.15) - \frac{1}{2}I^2R - K(T_h - T_c) \quad (1)$$

$$Q_h = SI(T_h + 273.15) + \frac{1}{2}I^2R - K(T_h - T_c) \quad (2)$$

$$COP = \frac{Q_h}{VI} \quad (3)$$

Where I is the current; T_c is the cold-side temperature; T_h is the hot-side temperature; S is the module's electrical resistance ($R = 2Na$); R is the module's electrical resistance ($R = 2N\sigma / G$); and K is the module's thermal conductance ($K = 2NkG$). S , R and K are the temperature dependent parameters. N refers to the total number of TE elements used in each module. G is the so called geometric factor, and a , σ , K are the material properties for the specific the type of the TE element.

Because the values of S , R , K are related to the cold-side and hot-side temperatures of the TE modules, so the values of S , R , K change with the changes of cold-side and hot-side temperatures, and in the paper, comprehensively considering the factors of S , R , K changing with temperature, the authors calculate the COP value of the TE heat pump according to the method in reference [22].

4. EXPERIMENT RESULTS

4.1. The performances of single TE module under different voltages Theory/calculation

Figure.. 3 and 4 show the changes of the temperature difference between the hot-side and cold-side of the TE module (ΔT) under different voltages in heating and cooling modes, respectively. In the experiment, the ambient temperature(T_e) is about 18°C and the temperature of air channel(T_s) is about 30°C .

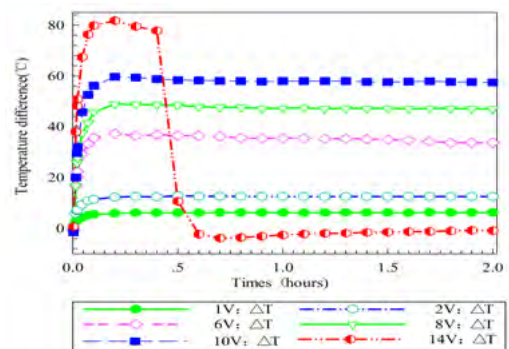


Figure 3 ΔT for single TE module under different voltages in heating mode

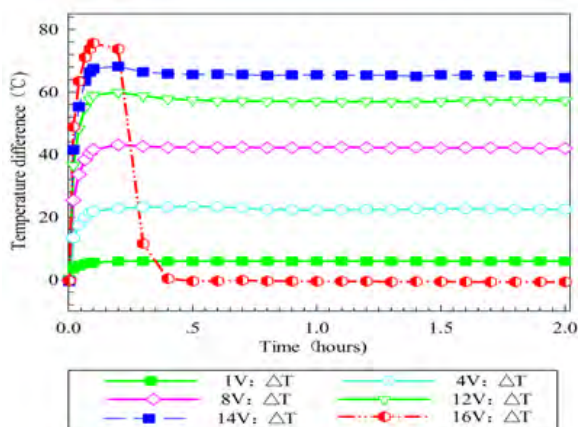


Figure 4 ΔT for single TE module under different voltages in cooling mode

It can be seen from Figure. 3 and 4, the ΔT increases rapidly after the TE module begins to work, and about after ten minutes the ΔT is stabilized at a certain value when the TE module work under normal voltages in heating and cooling mode. The higher the voltage is, the bigger the ΔT is. However, when the voltage increases to 14V, the TE module is broken down after forty ten minutes when the ΔT reached to 82°C, then the TE module is out of service, and after that the ΔT decreases to zero quickly in heating mode. Compared to heating mode, the TE module is broken down when the voltage increases to 16V and the ΔT is 76°C when the TE module is out of service in cooling mode.

4.2 The performances of single TE module under dramatic change voltage

Figure. 5 and 6 depict the performances of a TE module under dramatic change voltage for the heating and cooling modes. During the experiments, voltage changes every 15 minutes, the specific sequence of change is 2V-4V-6V-8V-6V-4V-2V. In the experiment, the ambient temperature is about 24 °C, and the T_s is about 30 °C. It can be seen that when the voltage increases (voltage from low to high), the TE module current rapidly increases to a high point and then, with the further increase in the temperature difference of the TE module, the current slightly falls and then maintains at a certain value. When the voltage decreases (voltage from high to low), the current rapidly falls to certain value and then slightly increase to a certain value and keep stable.

At the same time, the figures also show that the impact of voltage on the T_c is smaller than the impact on the T_h . In the processes of voltage increases from low to high (2V-4V-6V-8V) and voltage decreases from high to low (8V-6V-4V-2V), at the same voltages in the two processes above-mentioned, the T_c and T_h in the process of voltage increasing are lower than the temperatures in the process of voltage decreasing.

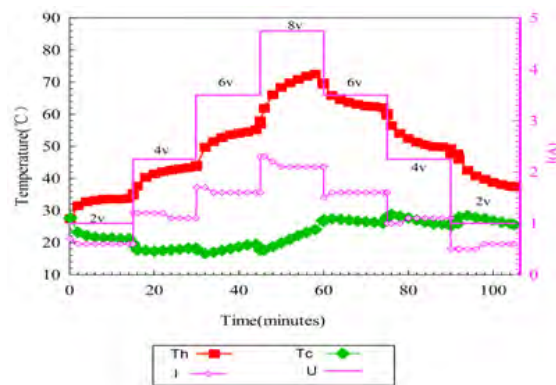


Figure 5 Performances of the single TE module under dramatic change voltage in heating mode

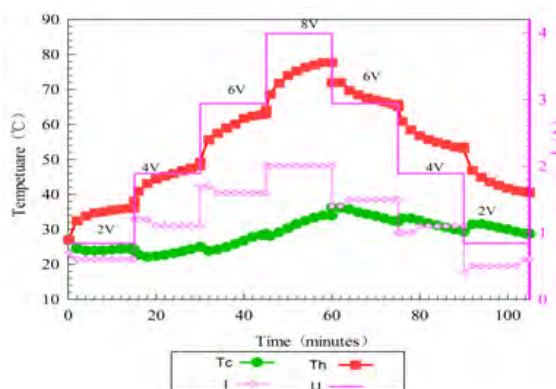


Figure 6 Performances of the single TE module under dramatic change voltage in cooling mode

4.3 Performances of TE system under different voltages

It can be seen from the tests of single TE module in off-design condition, voltage plays an important role in the safe and efficient operation of TE module. In Fig. 7, T_h and T_c under different voltage and at T_s of 30 °C are compared. It can be seen from the figure, the higher the voltage is, the higher the T_h is. It can also be seen from the Figure 7, compared to T_h , the T_c is affected less by the voltage.

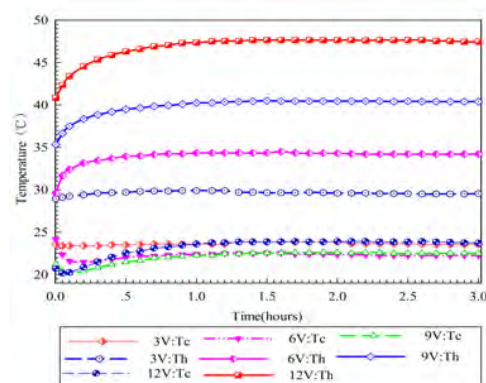


Figure 7 Performances of TE systems under different voltages

The performance of TE system under different voltages after 2 hour continuous work and entering a stable state are shown in Table 1. It can be seen from the table 1, the higher the voltage is, the higher the temperature of the radiation copper plate (T1, T2, and T3), and the greater the temperature differences between the points on the copper plate are.

Table 1 Performances of TE system under different voltage

V	Ts	Tc	Th	T1	T2	T3	COP
3V	29.3	23.9	29.6	28.1	27.6	27.0	4.10
6V	29.9	22.4	34.3	32.2	31.3	30.6	2.40
9V	31.1	22.6	40.4	37.3	36.2	35.5	1.88
12V	30.0	23.9	47.6	43.3	41.7	41.0	1.59

Increasing of voltage applied on TE modules meant more power was input to cause the higher heat flux, which was hardly to be transferred timely through indoor radiant panels. The higher the voltage is, the greater the ΔT , and the higher the temperature of the radiation copper plate are. From table 2, the voltage of 6V can reach the demand of BIPV/TE wall on the temperature of indoor radiant panels and get a high coefficient of permanence.

4.4 Performances of TE system under different air channel temperatures

In the actual operation of a BIPV/TE wall system, the PV system converts a part of solar energy into electricity, and most of which converts into heat to the air channel in the form of heat, and in this experiment, the air channel temperatures under different conditions are simulated by adjusting the temperature of the constant temperature water tank, to test the performance of the TE system. Figs.8-10 respectively show the TE system performances tested at air channel temperatures of 10 °C, 20 °C and 30 °C, at an indoor ambient temperature around 18 °C, at operating voltage of 6V, the test time is 3 hours.

It can be seen from the figures, after the TE system start to work, the Th rises rapidly while the Tc quickly falls. The temperature of the radiation copper plate (T1, T2, T3) increases with the increase in the Th. The Ts have major impact on the Tc, and the higher the Ts is, the higher the Tc is. At the same time, the higher the Ts is, the greater the difference between the cold-side temperature and thermal channel temperature is, and the more heat of TE system absorbed from the air channel

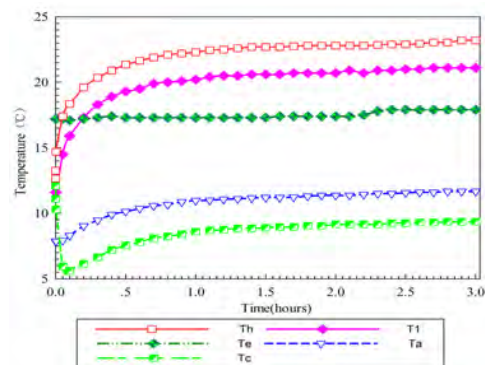


Figure 8 Performance of TE system at a cooling channel temperature of about 10 °C

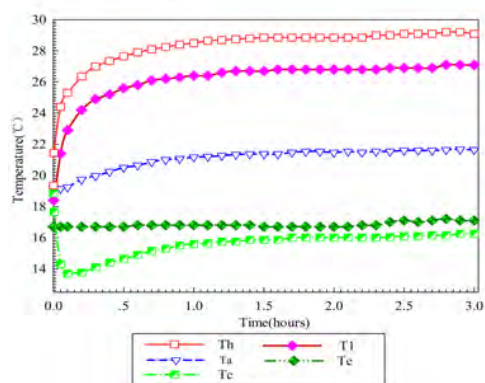


Figure 9 Performance of TE system at a cooling channel temperature of about 20 °C

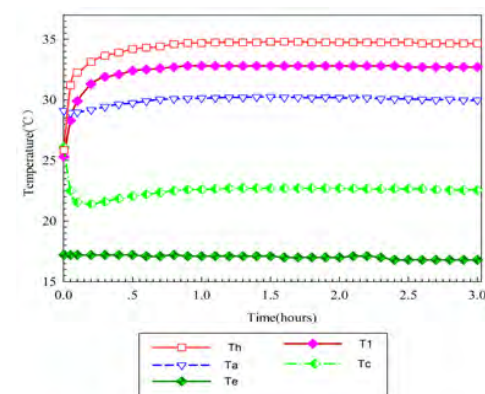


Figure 10 Performance of TE system at a cooling channel temperature of about 30 °C

When the Tc is 10 °C and the working voltage is 6V, the copper temperature(T1) is stabilized at the temperature of 20 °C; the copper temperature rises with the increase of Ts, and when the Ts is 40 °C, the copper temperature(T1) is about 38 °C. The temperature of the radiation sheet with in BIPV / TE wall system is affected by the Ts, so to maintain a comfortable indoor environment, we can regulate the operating voltage of the TE module to achieve a proper temperature.

The Performance of TE system after a 2-hour continuous work and entering a stable working state are shown in Table 2, and it can be seen from the table that

the T_s has a major impact on the heating coefficient of the TE heat pump and the copper temperature.

Table 2 Performances of TE system under different air channel temperatures

V	T_s	T_e	T_1	T_2	T_3	COP
6V	8.1	7.2	12.1	12.0	11.9	2.62
6V	11.4	7.3	17.3	17.0	16.9	1.84
6V	20.0	7.3	23.4	23.1	22.8	2.38
6V	30.7	7.3	31.0	30.6	29.9	3.10
6V	8.2	17.3	17.3	17.2	17.0	2.03
6V	11.4	17.3	20.7	20.3	20.0	1.59
6V	20.0	16.7	26.8	25.9	25.0	1.92
6V	29.9	17.2	32.2	31.3	30.6	2.40

The lower the T_e is, the lower the copper temperature is, and the smaller the ΔT is, the lower the heating coefficient is. At the same ambient temperature, the higher the T_s is, the smaller the ΔT is, and bigger the heating coefficient is. When the voltage is 6V, the T_e is 7.2°C, the T_s is 8.2°C, the temperature of the copper plate (T_1) is 12.1°C, and the COP is 3.28; when the T_s rise to 40.2°C, the temperature of the copper plate (T_1) is 37°C, and the COP is 4.9.

4.5. Performances of a TE system under continuous change of voltage

Fig. 11 shows the T_c and T_h , the copper temperature (T_1) changes, and the current changes under the continuous changes of voltage.

In a BIPV / TE wall system, the PV power the TE system to work long hours, the experiment simulates and tests the performance of a wall system in 7 hours of continuous work under continuous changes of voltage. During the test, the T_s is 30°C, the T_e is 18°C. The TE system is adjusted in the order of 1V-2V-3V-4V-3V-2V-1V. Compared to the performance of a single TE module in the continuous changes of voltage; the two changes are essentially similar.

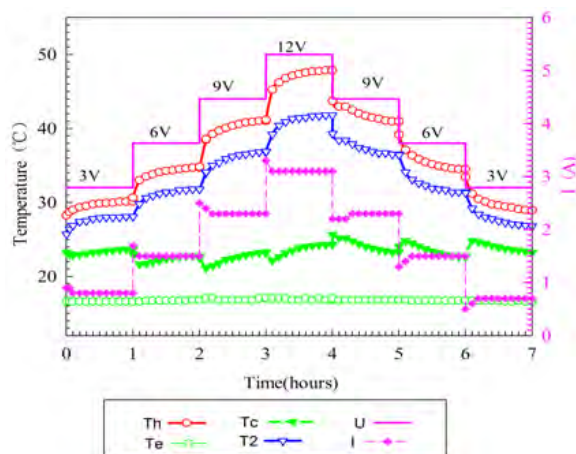


Figure 11 Performance of TE system in a continuous change of voltage

5. DISCUSSION

It can be seen from the above experimental results that when the T_s is 11.4°C and the T_e is 7.3°C, to maintain the temperature of radiation copper plate (T_1) at 17.3°C, the input voltage is 6V, and the current is 6A, namely the wall with a area of 0.36 m² needs the power of 9.6 W per hour, and a wall with a area of 1 m² needs the power of 26.7W per hour, so it needs the power of 641W per day. For example, the average solar radiation for Harbin in China over January is about 5.3kWh / (m²-day), and a typical crystalline silicon PV panel has an efficiency of approximately 18%, so the electric quantity per day generated by a PV with an area of 1m² is 954W, which can meet the power needs of the wall system. With the increase of T_s , the temperature of radiation copper plate increases as well, the higher the T_s is, the lower voltage and the less electricity are needed to maintain the radiation copper plate at a certain temperature.

In the actual operations, in order to maintain the 24-hour operation of the wall system, the methods of energy storage include battery power storage and wall heat storage. The former method is to control the temperature of the radiation copper plate by controlling the output voltage of the battery (photovoltaic cells), and the matching of PV and TE systems relates only to the issue - whether the annual generating capacity of the PV can meet the needs of the TE system. The latter method (wall heat storage) is to add a heat storage layer on the inner side of the radiation copper plate, in addition, phase-changing thermal storage materials or sensible heat storage method can be adopted to increase the thermal insulation property of the wall of under non-working state.

Compared to ordinary walls, even when the TE system stop working, BIPVTE walls can effectively achieve the thermal insulation function through the rational design of the thicknesses of the air layer and of insulation board in the walls structure, for example, when the air channels thickness are both 60mm (air thermal conductivity is 0.0266 W / (m • °C), the thickness of insulation backplane between the air layer is 20mm (thermal conductivity is 0.046 the W / (m • °C), ignoring the thermal resistances of the thin photovoltaic cells and the radiation copper plates, the comprehensive heat-transfer coefficient of the BIPVTE wall in non-working state is about 0.20 W / (m² • °C) which is lower 44.9% than the lower limit value of the outer wall heat transfer coefficient - 0.45 W / (m² • °C) set forth in China Design standard for Energy Efficiency of Residential Buildings in severe Cold and cold Zones. Obviously, to substitute BIPV / TE walls for original walls can greatly improve the thermal performance of building envelopes. According requires of the thermal insulation, we can decide the insulation backplane between the air layer is need or not. In this paper, the heating function and characteristics of the BIPVTE wall systems are mainly researched, but the walls also have a function of cooling in summers, and functions of ventilation and

cooling in transitional seasons, the relevant researches on the cooling and ventilation of walls in summers will be carried out in the next experiment when the BIPV/TE wall installed and the TE directly powered by the PV.

6. Conclusions

To research the new-type BIPV/TE wall system, a 600x600mm performance test simulation platform for BIPV / TE wall system is constructed, and carries out tests and study on the performances of single TE module in off-design condition states. On this basis, a series of experiments are carried out to determine the performances of TE systems under different voltages, different air channel temperatures (T_s), and in a step change of voltage.

Voltage has an important impact on the normal operation of TE modules. In certain heat dissipation conditions, the higher the voltage is, the bigger the ΔT is, and when the voltage increases to a certain value, the ΔT increase to extreme and then The TE module is broken down and out of service. The voltage of 6V is considered as the best voltage when BIPV/TE systems working in heat condition.

A change-step voltage regime was used to simulate the practical condition of changing PV output power. It was found that changing the input voltage has a major impact on T_h (more than 5°C in a heating mode), but has little impact on T_c with 1to 2°C difference with each step. In the processes of a voltage gradually rising from 2V to 8V and gradually falling from 8V to 2V, for the TE modules at the same voltages in the two processes above-mentioned, the T_c and T_h in the former process are lower than the temperatures of TE modules s in the latter process.

Air channel temperature (T_s) has a major impact on the cold-side temperature of TE modules (T_c), and the higher the T_s is, the higher T_c is. The temperature of the internal radiation sheets in BIPV / TE wall systems is affected by the air channel temperature (T_s) and indoor ambient temperature (T_e) and other factors, so we can adjust the temperature of radiation copper plates by controlling the voltage of TE modules, to meet the requirements of indoor comfort.

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A SIMPLE METHOD TO DETERMINE ENERGY CONSUMPTION RATION —A CASE STUDY FOR OFFICE BUILDINGS

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ABSTRACT

Energy efficiency management of large-scale public buildings is the breakthrough point of building energy saving in China. Energy consumption ration (ECR) as a benchmark of energy saving is one of the useful methods to promote efficient use of energy, and the method for determining ECR is an issue of the utmost importance. In this paper, the mean value method, factor normalization method and cumulative frequency distribution method are proposed firstly to determine the energy consumption ration. In the factor normalization method, energy consumption ration is defined as the maximum value of energy use intensity by regulating the value of each explanatory factor within its scope. The large-scale office buildings in Shanghai are selected for case study. Detailed energy audits and on-site surveys were conducted to obtain building information, operation and energy use data. Based on the energy consumption survey data, the ECRs under the proposed three methods are compared, and the ECR value from the factor normalization method is approved to be reasonable and practicable. In addition, it is found that three dependent variables, gross floor area, building usage rate and having data center or not, play a big role on the office building energy consumption. The ECR for office building in Shanghai is determined to be 60.3 kgce/(m².a), and only 20% of the samples have higher energy use intensity than the fixed ration. It is suggested to use the factor normalization method as a simple method to determine energy consumption ration. The proposed method can also shed some lights on the determination of energy consumption ration for other buildings.

Keywords: Large-scale office buildings, Energy audits, Building energy benchmarking.

1. INTRODUCTION

Energy consumption of large-scale public buildings should be paid more attention due to it accounting for almost 30% of the total electricity consumption by civil buildings in China. At present there are 500 million large-scale public buildings, and the electricity use intensity is 70~300 kWh/ (m².a), 10~20 times bigger than that of the residential buildings. Therefore, energy management of public buildings especially for the large-scale public building is one of the most important ways to promote energy conservation [1]. It will effectively constrain the growth rate of building energy consumption. In order to improve the level of building energy management and to turn the desire and demand of energy saving into a factual action, a benchmark of building energy consumption should be established. Energy consumption ration (ECR) of large public buildings is one of government's tools to promote energy efficiency

and to guide activities of building energy saving [2]. Government progressively adds the price to the unreasonable part of energy consumption based on the ration of building energy consumption, which helps to punish or reward energy users and encourage management and maintenance of large-scale public building energy saving. Therefore, this ECR value shall serve a standard for building energy saving, and the method for determining ECR is an issue of the utmost importance.

In this paper, a simple method to determine energy consumption ration for large-scale public buildings is developed. The large-scale office buildings in Shanghai are selected for case study. By combining the proposed ECR method with energy survey data, the recommended value of ECR for large-scale office buildings is obtained. The proposed method can be used to determine the ECR values for other buildings and guide management of building energy saving.

NOMENCLATURE

Abbreviation		Q	The building energy use (kgce)
ECR	Energy consumption ration	A	The gross floor area (m ²)
EUI	Energy use intensity	R _r	The building usage rate
Symbols		L _{dc}	The nominal variable to determine a building having data center or not, 1 represents YES, 0 represents NO
Y	The independent variable in regression model	Greek Letters	
X _k	The dependent variable in regression model	β _k	The regression coefficients
$\bar{X}_k$	The sample mean of the dependent variables	β ₀	The intercept
Y ₀	The mean building energy use		

2. ENERGY CONSUMPTION RATION METHOD

2.1 Consideration

To determine ECR of a building, several work has been done in [3-7]. In literature [3], some considerations about the determination of ECR were given. To sum up, there are two keys to obtain an objective and reasonable energy consumption ration for public buildings. Firstly, the current building energy simulation still has some difficulties to accurately describe the reality. The determination of ECR should be based on the actual operation data from energy survey or energy monitoring system, which can be easily accepted and recognized by the industries. Secondly, the determination of ECR should consider the natural and social factors such as local climate characteristic, economic level, living habit, and trend of energy consumption. Along with the above factors, building age, type and energy consumption level will be simultaneously taken into account to estimate an ECR value. So it ensure that energy consumption levels of the most of buildings are below or close to the value of ECR, and some high energy consumption buildings after factor modifications are identified with big energy-saving potentials.

Therefore, from the view of reality, energy consumption levels of the most of buildings are regarded as reasonable, and the determination of ECR is firstly to obtain a value expected to be close to most of current building energy use levels, and then to consider the factor modifications.

2.2 Method

2.2.1 Method A: mean value

From the aspect of "social equity" principle, each building or system shall consume the same energy when different buildings or systems achieve the same function. Ignoring special energy use or the effect of the

differences in actual process conditions for the same type buildings, the larger the value of energy consumption intensity is, the worse energy efficiency a building has. Then the social mean value of energy consumption intensities severing as ECR is the simple and intuitive characterization index to evaluate the energy efficiency for the same type buildings.

2.2.2 Method B: factor normalization

Building-energy-use factors could be divided into two categories according to their effect on energy use, i.e. objective/uncontrollable factors and subjective/controllable factors. The former includes those factors that cause the objective and reasonable differences of building energy consumption, such as climate conditions, building size and type. The latter includes the operation and management level factors that cause the subjective differences of building energy consumption, such as building envelope structure, equipment system performance and operation strategy. The latter factors' improvements are also the direction of building energy conservation. Therefore, a factor normalization method to determine ECR is proposed to eliminate or reduce the differences of building energy consumption caused by the objective/uncontrollable factors, and reflect the fairness and rationality of ration values.

The concept of the factor normalization method is to normalize the values of the objective/uncontrolled factors and keep the values of the subjective/controllable factors unchanged for each building, as shown in Figure 1. Then the normalized building energy consumption reflects the differences caused by the subjective/controllable factors. If a building energy consumption ration after normalization is higher than others, we can conclude that the operation and management level of this building is accordingly poor. It is known that there are many various factors affecting building energy use, and some factors are not easily obtained or determined. So the accurately accessed and important objective/uncontrolled factors are chosen for analysis in this paper.

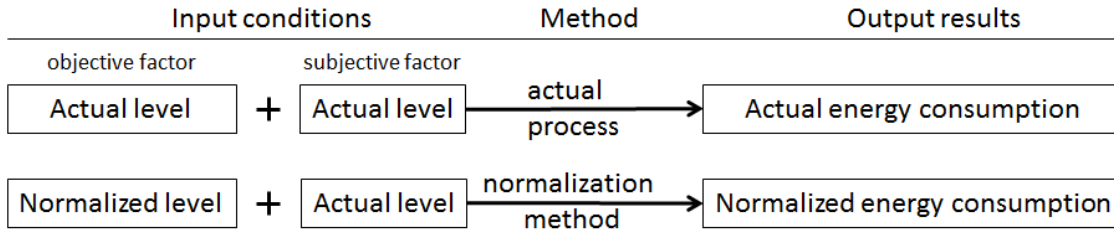


Figure 1 The concept of the factor normalization method

In order to determine the normalized energy consumption ration, a building energy consumption ration model is developed based on abundant building sample data from field survey and multiple regression analysis method. For simplicity, we assume the final model is expressed as,

$$Y_i = \beta_0 + \sum_{j=1}^n X_{j,i} \beta_j \quad (1)$$

Where Y is the independent variable, building energy use. X_k ($k=1,2,\dots,n$) are the dependent variables, the significant objective/uncontrolled factors; β_k ($k=1,2,\dots,n$) are the regression coefficients; β_0 is the intercept.

In order to illustrate the effect of each objective/uncontrolled factor on building energy use, the following equation is derived,

$$Y_i = \sum_{j=1}^n (X_{j,i} - \bar{X}_j) \beta_j + Y_0 \quad (2)$$

Where $\bar{X}_k$ ($k=1,2,\dots,n$) are the sample mean of the objective/uncontrolled factors; Y_0 is the mean building energy use.

Building energy use intensity can be obtained by

$$EUI(X_k) = Y_i / A \quad (3)$$

When X_k takes the value of $\bar{X}_k$, $EUI(\bar{X}_k)$ is the sample mean of building energy use intensity.

For any X_j , when the other objective/uncontrolled factors value are the average values. we look for a X_j to make $EUI(\bar{X}_k)$ reach the maximum. At this point, correction coefficient C_j is defined as:

$$C_j = \max \{ EUI(X_j) \} / EUI(\bar{X}_k) - 1 \quad (4)$$

$$X_j \in [X_j^{\min}, X_j^{\max}], X_{i \neq j} = \bar{X}_{i \neq j}$$

Then, the following formula of the maximum energy use intensity by multiplying the sum of the correction coefficient of the dominant objective factors with the sample mean of building energy use intensity (EUI).

$$EUI_b = \left[\sum_{i=1}^n C_i + 1 \right] EUI(\bar{X}_k) \quad (5)$$

From Eq.(5), it can be seen that, for the same type buildings, the actual energy use intensity of each building should not be higher than the maximum, i.e.

the modified energy value EUI_b . In addition, considering the implementation of price increase beyond ration by government at the early stage, the relaxed ECR values help to the implementation of the policy, so EUI_b is suggested to be the value of ECR.

2.2.3 Method C: cumulative frequency distribution

Energy consumption ration refers to the basic energy requirements to guarantee building's function. Frequency rating procedures define energy classes based on the frequency distribution of buildings and by considering an equal number of buildings for each class. Thus, in case that ten classes have to be defined, the upper class involves the top 10% of the building stock, the second class buildings between the top 10% and 20% of the buildings, and the last class the worst 10% of the stock. Based on the cumulative frequency distribution of annual building energy use intensity, the corresponding energy-saving feasibility under different classes can be directly evaluated. In order to encourage public buildings to high efficiency, low energy consumption and improve the feasibility of the implementation of price increase beyond ration by government, the ECR value is regarded as reasonable when the cumulative frequency distribution is between 75%-90%. Of course, the final value of the ECR depends on the degree of building energy-saving enforcement, policy guidance and so on.

3 CASE STUDY FOR OFFICE BUILDING

3.1 Building energy survey

Many factors affect the energy consumption of the buildings. These factors not only include thermal characteristics of building envelop and equipment system, but also contain the parameters of operation mode, management level, user awareness of energy conservation and others. In order to obtain enough data for ECR determination, a field survey on the basic information and the actual operation parameters of 20 office buildings in Shanghai was conducted (see Table 1). These buildings are selected from the wide statistics sample, and have the similar distributions on the mean and standard deviation (SD) values.

Table 1 Survey information on office buildings

Item	Sample number	Building area,m ²	Energy use intensity, kgce/(m ² .a)			
			statistics sample		survey sample	
			mean	SD	mean	SD
Value	20	918830	45.2	17.5	46.4	16.9

3.2 Building energy consumption factors

The current survey items of each building include building name, completion time, gross floor area, functional zone area, number of layers, thermal parameters of wall, window, HVAC, lighting, building automation, work time, air-conditioning time, electricity, oil or gas bills, special equipment etc.. In section 2.2.2, building-energy-use factors are divided into the objective/uncontrollable factors and subjective/controllable factors. The objective factors such as climate conditions, building size and type directly affect the fairness of ECR value. For example, the EUI of a hotel is usually bigger than that of an office due to building function difference when other factors are the same. All the buildings are located in Shanghai, so the effect of climate on building energy consumption can be ignored.

According to the requirement of the factor normalization method, the construction completion time, gross floor area, garage area, number of layers, building usage rate, work time and having data center or not are the objective/uncontrollable factors and taken into the building energy use model.

3.3 Stepwise regression model of building energy use

Stepwise regression method is used to obtain the relationship between building energy consumption data and the dependent variables listed in the table 2. As the significant level is 0.05, the calculation results are

shown in Figure 2. By introducing and deleting of the dependent variables three times, the goodness-of-fit index, R^2 is 0.845, and the modified R^2 is 0.816. The p-value for the F-test of the regression model is less than 0.0001, and the regression model is proved to be seasonable.

After the factor selection, only three dependent variables, gross floor area, building usage rate and having data center or not, are included in the regression model. The size of the model reduces a lot, and the correlation coefficients between each variable are significant. The Durbin-Watson statistic is within the range of 1.5~2.5, there is no significant auto-correlation. The auto-correlation coefficients among variables are less than 0.3.

After the selection criteria test, we obtain the following regression model for office building:

$$EUI = \left[\begin{array}{l} 0.069*(A_i - 45942) + \\ 4187.829*(R_i - 0.89) + \\ 939.543*L_{dc} + 2034.909 \end{array} \right] * 1000/A_i \quad (6)$$

From Eq. (6), we know that three dependent variables, gross floor area, building usage rate and having data center or not, play a big role on the office building energy consumption. When these three factors are set as the social average value, building energy intensity can be calculated with the result of 48.4kgce/(m².a), and the value is very similar to that of survey samples, 46.4 kgce/(m².a) with the difference 4.3%. It can be seen that the regression equation is effective.

Model summary

model	R	R ²	Corrected R2	estimated error	Durbin-Watson
1	.832 ^a	.693	.676	940.826	
2	.894 ^b	.799	.775	784.064	
3	.919 ^c	.845	.816	709.276	2.044

a. predictive variable, gross floor area; b. predictive variable, gross floor area, usage rate; c. predictive variable, gross floor area, usage rate, having data center or not; d. Independent variable: building energy use.

Coefficients

Model	Non standardized coefficient		Standard coefficient	t	Sig.
	B	Standard error			
constant	-4888.696	1537.308		-3.180	.006
gross floor area	.069	.009	.780	7.497	.000
usage rate	4187.829	1602.843	.267	2.613	.019
having data center or not	939.543	430.005	.233	2.185	.044

Anova^d

Model	quadratic sum	df	mean square	F	Sig.
3 regression	43832695.202	3	14610898.401	29.043	.000c
residual	8049169.800	16	503073.113		
Total	51881865.002	19			

Figure2 stepwise regression model with the significant level of 0.05

Table 2 Comparisons of ECRs by different methods for office buildings

Method	A	B	C			
	mean value	factor normalization	cumulative frequency distribution			
			60%	70%	80%	90%
Building energy use intensity, kgce/(m ² .a)	48.4	60.3	50	54	60	68

3.4 Comparison between different ECR methods

Table 4 shows the comparisons of building energy consumption rations by the proposed three methods. The following discussion is provided.

Method C gives the corresponding ECRs under different cumulative probabilities. Through the results, we can directly access the degree of building energy-saving enforcement under different ECRs.

The ECR obtained by Method A reflects the social average building energy level without considering the correction factors. If ECR is the social average value of building energy use intensity, more than about 50% of the samples have higher energy use intensity than the fixed ration. The actual enforcement of building energy management is burdensome.

On the contrary, method B takes the maximum value of the energy use intensity as the ECR by considering the effect of some dominant objective factors on energy use intensity. We can see that only 20% of the samples have higher energy use intensity than the determined ration from method B. The actual enforcement of building energy management is moderate.

From the comparison analysis, it can be concluded that the factor normalization method is approved to be reasonable and practicable, and is suggested as a method to determine energy consumption ration. From this case study, The ECR value for office building in Shanghai is suggested as 60.3 kgce/(m².a).

4 CONCLUSION

In this paper, three proposed methods, the average value method, factor normalization method and cumulative frequency distribution method, to determine the energy consumption ration for public office buildings are discussed.

Based on the energy consumption survey data of office buildings in Shanghai, the ECRs under different methods are compared, and the ECR value from the factor normalization method is approved reasonable and practicable. The ECR for office building in Shanghai is determined to be 60.3 kgce/(m².a) with only 20% of the samples having higher energy use intensity than the fixed ration. Therefore, the factor normalization method is suggested as a simple method to determine energy consumption ration.

However, this study the number of survey samples is not too large, and the number of the factors considered in the regression model of building energy use is also limited. More comprehensive work is still needed in the future to get a more scientific ECR. It is expected that the proposed method can shed some lights

on the determination of energy consumption ration for other buildings.

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LATTICE BOLTZMANN SIMULATION FOR HEAT TRANSFER IN POROUS SOIL BASED ON FRACTAL RECONSTRUCTION

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ABSTRACT

The processes of heat and moisture transfer in soil are important to improve the efficient of ground source heat pump and increase production from geothermal reservoirs. It is well known that soil and rock can be considered as porous media. Heat transfer across matrix of granules and pore filled by air, water and vapor is affected by microstructure of soil. It has been shown that granules shape and size distribution of soil granules have fractal characteristic in most studies. Based on fractal theory, this paper presents a stochastic method to reconstruct porous media from limited morphological and statistic information obtained from two-dimensional microscope slice images of real porous soil. The heat transfer in reconstructed random fractal porous soil is simulated thought thermal lattice Boltzmann method. The effect of soil structure parameters which include porosity, fractal dimension on thermal storage performance of reconstructed soil is analyzed. The effect of porosity on heat transfer process in soil is observed.

Keywords: Porous media; Fractal; Lattice Boltzmann method; Geothermal.

1. INTRODUCTION

Building energy consumption takes up a great proportion in total energy consumption of our country. Reducing energy consumption in process of building operation is beneficial to construct low carbon town. Ground-Couple Heat Pump (GCHP) is new technology of energy conservation and environmental protection, which has been developed with energy crisis in recent years. Soil of ground is considered as heat source/sink of heat pump, which is a major characteristic for GCHP system. Surface soil is a huge solar energy collector and dynamic energy balance system for earth. The temperature of surface soil have smaller fluctuate in whole year, that is very useful to improve the efficiency of refrigeration cycle, because the condensation temperature is decreased in summer and evaporation temperature is increased in winter. Further more; if heating boiler is replaced by GCHP, atmospheric environmental quality will be improved, which is urgent problem to be solved for china. The thermophysical properties of soil have great effect on operation performance of GCHP [1], which are the most key parameter in the design and research process of ground source heat pump system.

It is well known that the soil is typical porous media which is constituted with soil granules, air and moisture. The mechanical and heat transfer properties of soil are determined by microscopic structure characteristics of porous media, therefore, Many researchers have devoted to propose series methods to describe the structure of porous media[2-4]. Since

fractal properties of soil were revealed from 80s of last century [5-6], a new analysis tool was provided to study the structure parameters of soil.

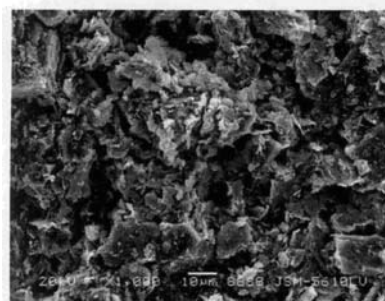
In this work, the binary two-dimensional microscope slice image of soil section was analyzed base on fractal theory, and the fractal dimension was obtained used by box-counting method. The random fractal porous media is reconstructed by 2D midpoint displacement method. The numerical heat model in porous media was established through lattice Boltzmann method of D2Q9. The thermal storage performance of reconstructed porous media were simulated with different key parameters of porous soil, such as porosity, fractal dimension and thermophysical properties. The effect of soil characteristics on thermal storage process was presented.

2. FRACTAL RECONSTRUCTION OF SOIL

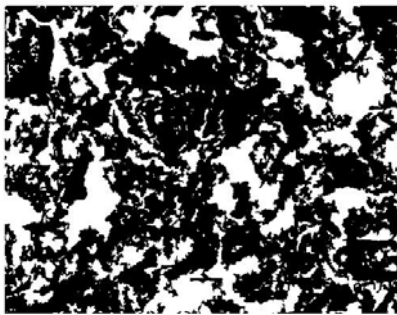
2.1 Fractal dimension of soil

A wide range of fractal models have been proposed to measure fractal dimension and other fractal properties, however there are some of contradictions also in those methods, which is systematic researched by Tao Gaoliang [7]. Box-counting method or Box dimension is one of the most widely used method for calculating fractal dimension, in that this mathematic method is easy to estimate the dimension of porous media, and the reliability of the results can be ensured. In this paper, the fractal dimension of soft clay is

calculated by box-counting method, the microscope image of which is obtained by Tao Gaoliang [7], as shown in Figure 1. The color image was transferred to binary image, where the white pixel is pore of soil and the black pixel is solid granules.



(a) color image



(b) binary image

Figure 1 Microscope slice image of soft clay

The process of box-counting method can be briefly expressed as: (1) The measured image is overwritten by several small-square that have equal side-length (r_1); (2) If there is at least one pixel of pore in one of small-square, the small-square is remarked as effective square, The total number of which is calculated as $N(r_1)$; (3) The side-length of small-square is changed to r_2 , where r_2 is a half of r_1 , and the new effective square is recounted as $N(r_2)$; (4) If the side-length is reduced step by step, a series of N can be obtained. Based on fractal theory, the relationship between N and r follow as:

$$N(r) \sim r^{-D} \quad (1)$$

According to binary image of Figure 1, the fractal dimension of soil D can be calculated by box-counting method. The logarithm of side-length and effective square presents linear relationship, as shown in Figure 2. Though linear fit for all of data, the fractal dimension of soft clay equals the value of 1.913, and the porosity of soil can be obtained from pixel information.

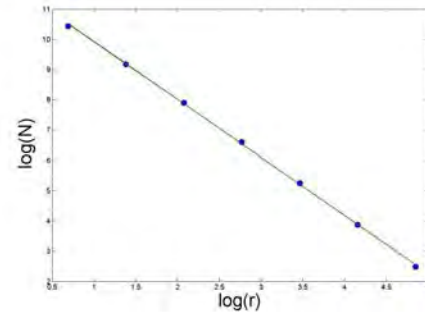


Figure 2 The logarithmic relationship between N and r

2.2 Random midpoint displacement algorithm

As mentioned above, reconstruction technology of porous media have been developed in recent years [8]. Random reconstruction is one of reconstruction method, which generates a random porous media based on statistic information of real porous media. This method can describe typical characteristic of a class of porous media, while the stochastic feature that consist in nature is retained. If the fractal theory is introduced to random reconstruction as statistic information of porous media, the stochastic fractal reconstruction is proposed [9].

Random midpoint displacement algorithm (RMD) is derived from fractal Brown motion theory which is common algorithm to generate fractal media such as terrain, rough surfaces, and time-sequence curve. This algorithm has been applied to replicate fractal porous media by E. S. Kikkinides et al [10, 11], and has been developed in recent years [12]. As an example, one dimension fractal brown curve is generated by RMD, as shown in Figure 3. For a straight line, the position of endpoint A and B is decided by random number between -1 and 1. The position of midpoint C is displaced by a random number, base on a determined variance, as follow as [12]:

$$F(C) = (F(A) + F(B)) + A \cdot R \cdot 2^{-H/n} \quad (2)$$

where A is a constant. H is Hurst exponent, R is also a random number which can be express as:

$$R = N(0, \sigma_n^2) \quad (3)$$

$$\sigma_n^2 = \frac{\sigma_0^2}{2^{nH}} \quad (4)$$

σ_n^2 is variance of random number. After the last step, two new straight lines AC and BC can be obtained. The position of new midpoint in the two straight lines is displaced with random number. After several repeated displacement, a fractal Brown curve is generated. For 2D porous media, creating fraction brown surface is the key procedure of reconstruction process. According to research results of other researchers, the Hurst exponent equal the value of 2-D. Kikkinides [10] found that the correlation length of Brown curve generated by RMD is compare to the calculated region, that results in the reconstructed media cannot represent the real porous media. Aiming at these

problems, the multi-cell RMD was proposed [10]. In this method, the calculated dominate is divided into several small calculated dominate, and RMD is implemented in every small calculated dominate.

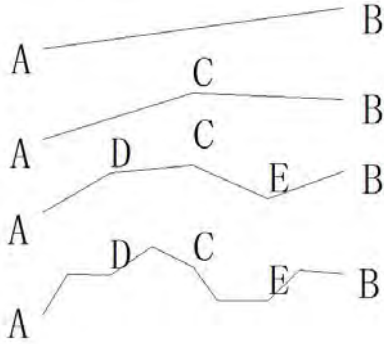


Figure 3 Schematic diagram of RMD

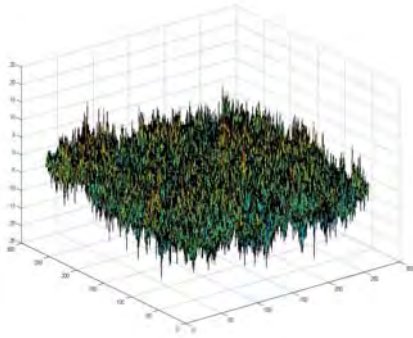


Figure 4 The fractal Brown surface of soil

In the order to ensure the porosity of generated porous media conformed to the real porous media, an appropriate cutoff value should be choose. The proportion of the number of nodes, the height of which is lower than cutoff value, is equal to the porosity of real porous media, then reconstructed porous media is generate by binarization processing. For the fractal soil image is analyzed in previous section, the fractal Brown surface can be depicted as shown in Figure 4, and the reconstructed porous media is illustrate in Figure 5, where the number of nodes is 513×513.

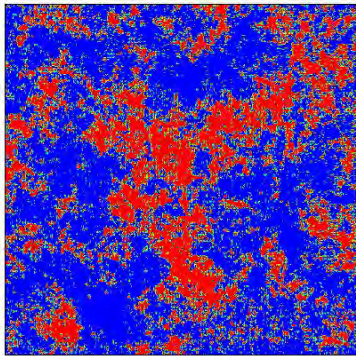


Figure5 Reconstructed soil image by RMD

3. LATTICE BOLTZMANN SIMULATION

Lattice Boltzmann method is hot topic in flow, mass and heat transfer fields now. This mesoscopic simulation method has displayed great advantage for transport process in complex media [13]. In this paper, we just consider the heat transfer in reconstructed porous media for purpose of theoretical study.

Based on CLBGK model [14], the temperature evolution equation and equilibrium distribution can be expressed as:

$$g_i(\mathbf{x} + \mathbf{e}_i \Delta t, t + \Delta t) - g_i(\mathbf{x}, t) = -(g_i(\mathbf{x}, t) - g_i^{eq}(\mathbf{x}, t)) / \tau_g \quad (5)$$

$$g_i^{eq} = \omega_i T [1 + 3 \frac{\mathbf{e}_i \cdot \mathbf{u}}{c^2} + 4.5 \frac{(\mathbf{e}_i \cdot \mathbf{u})^2}{c^2} - 1.5 \frac{u^2}{c^2}] \quad (6)$$

where τ_g is relaxation time, e is lattice velocity and ω is weight coefficients of D2Q9 model. The temperature and thermal diffusivity is obtained as:

$$T = \sum_i g_i \quad (7)$$

$$\alpha = \frac{1}{3} c^2 (\tau_g - \frac{1}{2}) \quad (8)$$

As preliminary study for reconstructed porous soil, the thermal storage process of porous media was simulated. The upper boundary of Figure 5 was set as isothermal boundary condition with temperature was T_1 , and other boundaries were employed as adiabatic boundary. The calculated dominate had uniform temperature T_0 at the initial time. In the order to make the simulation results have generality, these dimensionless numbers is defined as follows:

$$Fo = \frac{\alpha_1 \tau^2}{L^2}, \Theta = \frac{T - T_0}{T_1 - T_0} \quad (9)$$

where Fo is dimensionless time and Θ is dimensionless temperature. L is length of calculation region, τ is computing time, α_1 is the thermal diffusivity of soil granules. Considering the actual thermal properties of soil and air, the ratio of the thermal diffusivity of soil and air was assumed as 3. The dimensionless temperature distribution with different Fo numbers is shown in Figure 6. The rising process of temperate could be presented. The tortuous isothermal is found, because the difference of thermal diffusivity between air and soil granules and the stochastic characteristic of porous media.

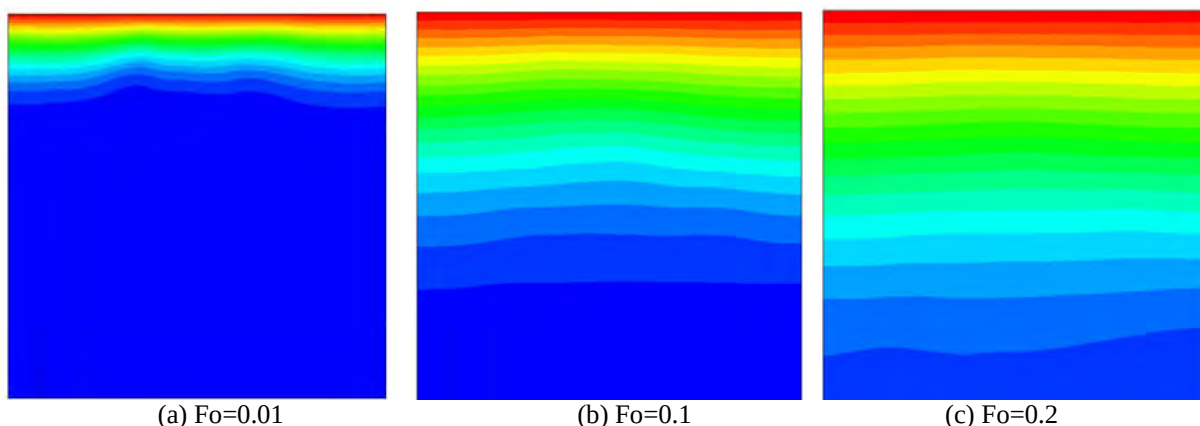


Figure 6 The dimensionless temperature distribution of reconstructed porous soil

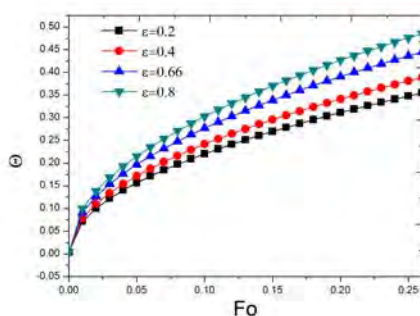


Figure 7 The average dimensionless temperature with different porosity

The compactness of soil can be considered as the porosity of reconstructed soil. The effect of porosity on the process of heat transfer is analyzed with the porosity from 0.2 to 0.8. As shown in Figure 7, the average dimensionless temperature of porous media increase as the increase of porosity of soil at the same Fo number. It is due to the difference of thermal diffusivity between soil and air, the performance of thermal storage decline with air rise in porous media.

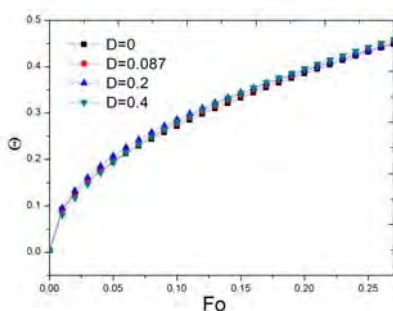


Figure 8 The average dimensionless temperature with different fractal dimension

The temperature rise process with different fractal dimension is examined through LBM. The rise rates of average temperature are almost uniform with different fractal dimension at the same porosity (see Figure 8), which indicate that the fractal dimension has few effect on heat transfer in soil.

4. CONCLUSIONS

The reconstruction process of soil microscope slice image was presented by random fractal algorithm. The fractal dimension and porosity of soft clay was obtained. Based on lattice Boltzmann method, the performance of thermal storage was simulated in reconstructed porous media. The average dimensionless temperature increase with the increasing porosity in calculation region. The obvious effect of fractal dimension on thermal storage of reconstructed soil has not observed in this work, while the flow and diffusion process in soil with different fractal dimension will be researched in future.

ACKNOWLEDGEMENTS

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MUSEUM OF HOT WET ENVIRONMENT AND BUILDING ENERGY CONSUMPTION SIMULATION

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ABSTRACT

Museum building as one of public buildings, the energy consumption is mainly for the power consumption, power consumption is mainly used for heating, air conditioning, lighting, etc. And air conditioning system is museum main thermal environment control system, its energy consumption is to guarantee system can long-term functioning of one of the key factors. Adverse temperature and humidity of the environment will not only damage to cultural relics, but also harmful microorganism breeding hotbed. Therefore, appropriate temperature and humidity of the environment is an effective way to protect the rare and precious cultural relics.

Based on Liaoning province museum of thermal and moisture environment air conditioning system (visit and storefront environment) as the research object, mainly studies different kinds of cultural relics preservation specification requirements of thermal and moisture environment conditions, and simulation analysis was carried out on the hot wet environment and system energy consumption. Air conditioning system energy consumption, in order to better analyze museum using Energy plus graphics processing software DesignBuilder energy consumption simulation, simulation analysis was carried out on the different air conditioning system running status; Evaluate the building indoor thermal environment; Analysis of building indoor environment's impact on building energy efficiency. Under the precondition of protecting cultural relics, the realization of energy saving air conditioning system, air conditioning system for museum operation and control thermal environment are put forward feasible suggestions.

Key words: Museum, Hot and humid environment, Energy simulation

1. INTRODUCTION

Energy is the world's economic development veins, Trend of world energy increasingly nervous, Energy consumption is increasing at a time. Museum, as serving the society and its development, a non-profit permanent institution, open to the public. For the purposes of the research, education, appreciation and as the evidence of collection, protection, research, dissemination and display human environment. The museum different from library, archives and other public buildings, it has its own special function requirements. Buildings must conform to the requirements of the museum of cultural relics collection, research, exhibition, education and management of the work and activities, especially the control of cultural relics protection environment, building energy consumption problem that not allow to ignore. In the process of the analysis of energy consumption in the museum, the DesignBuilder this more mature energy consumption simulation software to the museum annual hourly dynamic energy consumption are analyzed. Software simulation can quantify the energy saving effect of various technologies, each with different technical adjustment can be obtained from the software simulation results for

the construction of overall energy saving performance, thus confirm the best air conditioning energy saving scheme.

2. THE ESTABLISHMENT OF THE MUSEUM BUILDING MODEL

Museum is located in the middle of Shenyang, liaoning province, longitude 123.4° and latitude 41.8°, building orientation: due south. Total construction area of 31095 m², which covers an area of 16000 m², building height of 24 m. On the ground, three layers for the visit, underground for storage. The interior of the building is ring, the middle one is the hall, on both sides are cultural relics exhibition hall, There is no outside window in cultural relics exhibition hall, light Wells is set on the third floor roof. Figure 2.1 is the architectural model diagram of liaoning province museum by using DesignBuilder software.

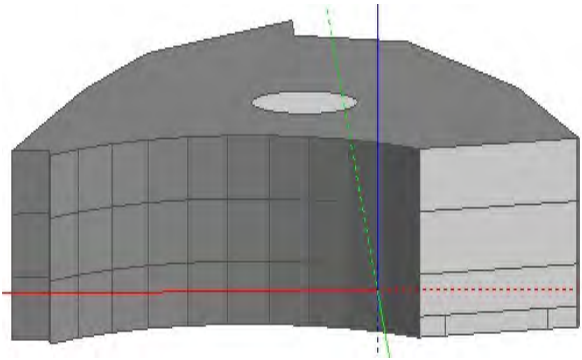


Figure 2.1 Museum architecture model diagram

Regional function is relatively simple in the museum, the museum is divided into the exhibition hall, shop, bathroom, stairs, multi-function hall, etc. There are 12 exhibition halls in the museum, including three exhibition halls on the first floor, four exhibition halls on the second floor, five exhibition halls on the third floor. Figure 2.2 is the interior partitions of the museum.

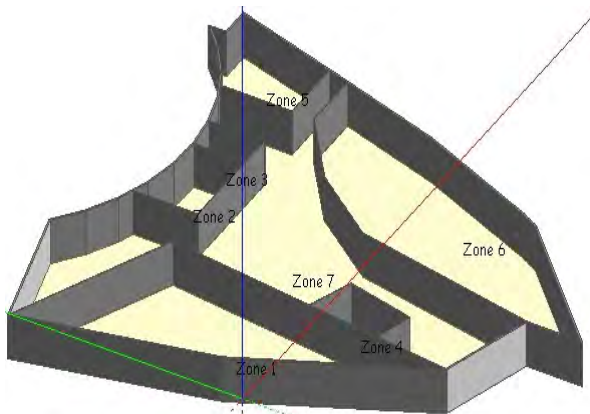


Figure 2.2 Building indoor partition diagram

2.1 LOCATION AND WEATHER DATA

(1) The geographical position

Software comes with a geographic location of the database, the database information is provided by the ASHRAE, including Shenyang. Specific have the following information: latitude, longitude, altitude, standard atmosphere, the beginning and end of the winter and summer time, and the design temperature of winter and summer.

(2) Meteorological data

EnergyPlus on China's meteorological data adopt international energy consumption calculation meteorological data IWEC. The meteorological data included 227 regions in the world except the United States, Canada's hourly weather data, Shenyang of China was among them. It mainly suitable for hourly simulation of energy consumption and can be used to simulate the typical energy consumption of years.

Weather files included such as outdoor dry bulb temperature, dew point temperature outdoor, outdoor relative humidity, atmospheric pressure, direct radiation, diffuse radiation intensity, intensity of illumination, wind direction, wind speed of 28 kinds of hourly meteorological parameters. This paper uses Shenyang typical meteorological year data which provided by ASHRAE to evaluate the building energy consumption.

2.2 BUILDING PALISADE STRUCTURE

Museum building palisade structure is mainly composed of outside walls, roof, outside the window, door, floor, etc. In the simulation model of the museum, in terms of proportion, the area of the outer door is very small relative to the whole building plane, so the outside door of calculation can be included in the outside the window, so the palisade structure heat transfer heat consumption is mainly composed of roofing, external wall and outside window. DesignBuilder provides the commonly used template of palisade structure of most countries. If it comes with the database without simulate building palisade structure, the user can define and change ourselves. Required parameters are: materials, thickness of palisade structure, density, specific heat, coefficient of thermal conductivity. So that we can customize palisade structure to meet the needs of users. Existing building energy saving and specification in our country, through prescriptive indexes stipulated in different areas, public building or residential building palisade structure heat transfer limits, this article museum retaining structure with reference to the "public building energy efficiency design standard" (GB50189-2005) set out in the cold region of area B public building palisade structure heat transfer standard, as specified in table 2.1 for the museum of palisade structure main parameters.

Outside the window structure: the thickness of the glass (5 + 6 + 5 a), using low-e hollow toughened glass, heat transfer coefficient is 2.72 W/m², K.

2.3 ONE SET AND AIR CONDITIONING DESIGN PARAMETERS

(1) Within the one set

Liaoning province resident office number is about 150 people, the museum's maximum attendance people is 2000, visited convert to permanent staff, personnel density take on average 0.1 / m²; Building interior lighting lamps and lanterns is about 50 kw total power, illumination density is 2.5 W/m², total power equipment is about 3000 W.

(2) The air conditioning design parameters

Shown on : Table 2.2 outdoor design parameters

Table 2.3 interior design parameters

Table 2.1 Palisade structure main parameters

Name of the wall	Wall structure	Thickness (mm)	Heat transfer coefficient (W/m ² • K)
Exterior wall	Brick wall		
	Foam concrete	395	0.42
	Fiberboard		
	With white paint		
Roof	Precast concrete with fine stones, surface spray of white cement	25	
	Ventilated layer, coiled material waterproof layer, leveling layer	240	
	Expanded perlite insulation: asphalt	200	0.35
	Steam trap layer, cast-in-situ reinforced concrete	90	
	In the painting		

Table 2.2 Outdoor design parameter

Winter outdoor design parameters					Summer outdoor design parameters				
Ventilation calculation on temperature (°C)	Air conditioning calculation on temperature (°C)	Relative humidity of air conditioning (%)	The atmospheric pressure (hPa)	Average wind speed (m/s)	Ventilation calculation on temperature (°C)	Air conditioning calculation on temperature (°C)	Air conditioning calculation on wet bulb temperature (°C)	The atmospheric pressure (hPa)	Average wind speed (m/s)
-16.2	20.6	69	1023.3	2.0	28.2	31.4	25.2	998.5	2.8

Table 2.3 Interior design parameter

Air conditioning room	Summer		Winter		The new air volume standard	Ventilated take a breath	Noise
Name	Temperature °C	Relative humidity %	Temperature °C	Relative humidity %	m ³ /h • person	Times/h.	dB (A)
Exhibition room	26	≤60	20	≥40	20	-	40
Office	26	≤60	20	≥40	30	-	50
Hall	26	≤60	20	≥40	20	-	45
Lecture hall	26	≤60	20	≥40	20	-	45
Conference room	26	≤60	20	≥30	30	-	55
Constant temperature and humidity showcase	23±2	60±5	24±2	60±5	-	-	-

2.4 THE ESTABLISHMENT OF THE MUSEUM OF AIR CONDITIONING SYSTEM

Liaoning provincial museum actually uses a return air system with constant temperature and humidity air conditioning system. One new return air system air volume adjusted according to the seasons, winter and summer new air volume adjustment in 10% ~ 20%, full

fresh air in transition seasons. System has set up 13 air conditioning units for the exhibition hall, 2 cases of the constant temperature and humidity unit to supply air for calligraphy and painting area.

The museum exhibition hall with the glass case will be closed, and isolation with the audience, exhibition hall and the calligraphy and painting area use two different sets of air conditioning system control the indoor environment, exhibition hall area using central

air conditioning system control room internal heat wet environment .air-conditioning running time: from 8:00 to 16:00 . Due to the calligraphy class display belong to environmental sensitive kind of cultural relics, the conservation environment stability of the demand is higher, so the painting area adopt constant temperature and humidity air conditioning control system, run 24 hours a day, and other cultural relics of the closed glass enclosure is no air conditioning.

The museum is set constant volume or dual duct VAV air conditioning system , we will contrastive analysis building energy consumption .Constant temperature and humidity air conditioning system running independently, not included in the analysis of energy consumption simulation. Simulation time is January 1, 2012 to 31 December 24.

3.BUILDING ENERGY CONSUMPTION SIMULATION RESULTS AND ANALYSIS

Simulated by DesignBuilder, museum annual maximum heat load is 2235.16 KW, maximum cooling load is 3175.67 KW, construction of various energy consumption simulation results as shown in figure 3.1 and figure 3.2, the museum uses constant volume or dual duct VAV air conditioning system comparison, comparing the total power consumption of the building, the simulation results as shown in figure 3.3

The figure shows that the change trend of the constant volume air conditioning system and dual duct VAV air conditioning system construction power consumption is basically the same .All the year round ,the load of power equipment, lighting lamps and lanterns in the museum are basically stable. Winter heating energy consumption and summer air conditioning refrigerating capacity varies with the seasons. One year, heating and cooling of the air conditioning power consumption is significantly higher than the transition season, the summer power consumption peak in July and August, the winter peak electricity consumption focus in December and January.

Constant volume air conditioning system in summer air conditioning refrigeration and power consumption of building energy consumption ratio is greater than the proportion of winter heating energy consumption, dual duct air conditioning system of winter air-conditioning system heat building power consumption ratio is greater than the proportion of summer air conditioning refrigeration power consumption.

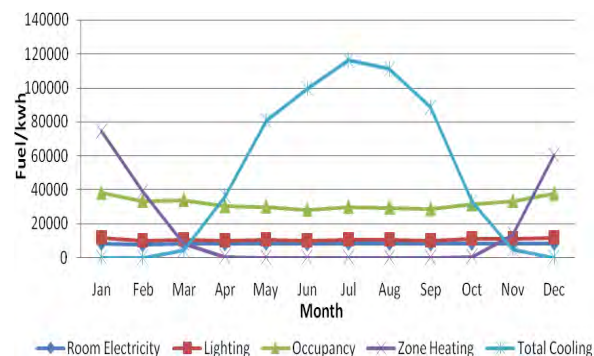


Figure 3.1 Various power consumption of building with Constant volume air conditioning system simulation results by DesignBuilder

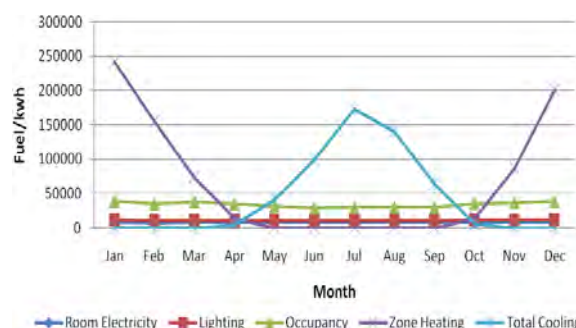


Figure 3.2 Various power consumption of building with Dual duct VAV air conditioning system simulation results by DesignBuilder

Can be seen from figure 3.3 for building power consumption situation, simulated values is obviously difference between constant volume air conditioning system and dual duct VAV air conditioning system. Dual duct VAV air conditioning system construction power consumption simulation is higher than constant volume air conditioning system about 40%. Transition from season: Dual duct VAV air conditioning system construction power consumption was slightly lower than the constant volume air conditioning system; Cooling season and heating season: constant volume air conditioning system power consumption significantly lower than the double duct variable air volume air conditioning system.

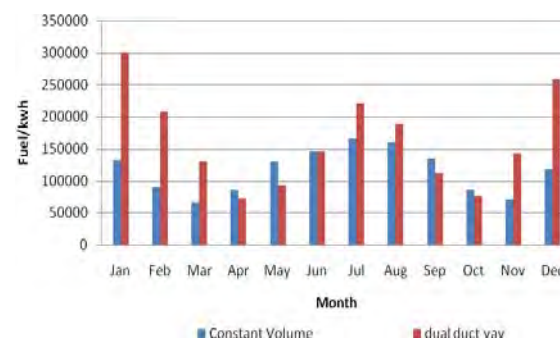


Figure 3.3 Constant volume air conditioning

system and Dual duct VAV air conditioning system energy consumption simulation value comparison

Figure 3.4 and figure 3.5 is the proportion of power consumption of constant volume air conditioning system and dual duct VAV air conditioning system in the museum on lighting, equipment, air conditioning, etc.

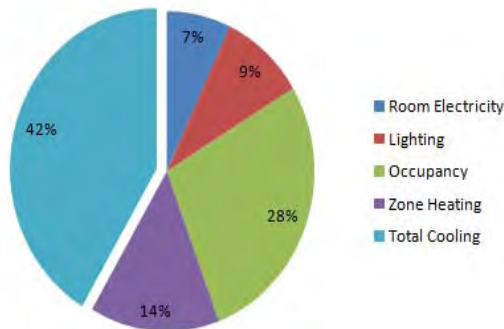


Figure 3.4 The constant volume air conditioning system energy consumption proportion

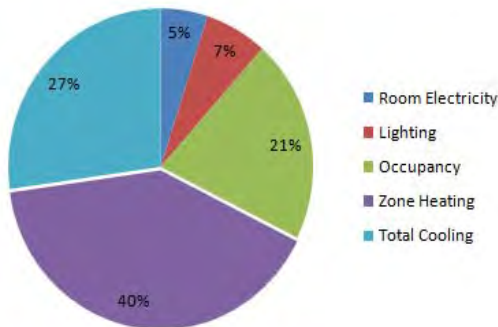


Figure 3.5 The dual duct VAV air conditioning system energy consumption proportion

The figure shows that the constant volume air conditioning system annual electricity consumption accounted for 56% of whole year total power consumption, dual duct VAV air conditioning system annual electricity consumption accounted for 67% of whole year total power consumption. Constant volume air conditioning system energy consumption is mainly used for summer cooling, dual duct VAV air conditioning system energy consumption is mainly winter heating, but summer refrigerating capacity is also high.

4. CONCLUSIONS

By using DesignBuilder for simulating museum run different air conditioning system, get:

Annual maximum heat load is 2235.16 KW, maximum cooling load is 3175.67 KW. Museum adopts constant volume system for power consumption is 1386484 kw, with dual duct VAV air conditioning system of annual electricity consumption is 1949164 kw, difference is 40%; Under the same hot wet environment, the museum of constant air volume air conditioning system is energy saving.

Annual electricity consumption air conditioning system of building throughout the year about 60% of the total power consumption, is the main part of the museum system operation cost, are important factors for museum building energy efficiency.

Study of air conditioning system energy saving potential in the museum analysis concluded that: strengthen the operation management, set up reasonable room air temperature and humidity, reasonable control the new air volume, enhance natural ventilation measures and transitional season, summer night on exhaust air heat recovery technology and reduce fresh air load, etc.

ACKNOWLEDGEMENTS

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SIMULATION PERFORMANCE OF A SOLAR POWERED DOUBLE BED NOVEL COMPOSITE ADSORBENT – WATER ADSORPTION CHILLER WITH THERMAL STORAGE

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ABSTRACT

Building energy efficiency and sustainability has played a major role in battling global warming. The energy consumed on cooling system represented about 30% of total energy consumption for building located under subtropical climate. The vapor compression chiller is the most commonly used technology for cooling generation but it consumes large amount of electricity. Adsorption chiller utilizing renewable or waste heat as the driving energy is an attractive alternative to vapor compression system. This paper simulated the performance of pairing three different types of solar collectors (flat plate, evacuated tube and parabolic trough) with a double bed adsorption chiller. The modeled chiller applied a novel composite adsorbent, a mixture of activated carbon, silica gel and calcium chloride. The mathematical model for the solar collectors, hot water thermal storage tank, adsorbent bed, evaporator and condenser were solved. The optimization of the thermal storage system and the overall system control is the critical next step for this application to be succeeded.

Keywords: Adsorption chiller; simulation; solar collectors.

1. INTRODUCTION

Climate change and increase in energy cost has motivated industries and research leaders to focus on renewable and energy efficiency. Buildings are the major electricity consuming element for urbanized cities. Under subtropical climate, air-conditioning and especially the cooling system demands the most energy and it represent about 30% - 40% of a typical commercial building's electricity consumption. Vapor compression chillers (VCC) operate under a Carnot vapor refrigeration cycle is the major part of the cooling system. Operating VCC not only carbon unfriendly with high global warming potential, most of the refrigerants are ozone depleting. Solar powered solid adsorption chiller (SPAC) can become an excellent replacement for the VCC since most of the work pairs for SPAC are using water as the adsorbate [1]. Studies of adsorption cooling systems using solid adsorbents have been ongoing since the Energy Crisis in the 1970s but have not received enough attention compared to systems that use liquid absorbents. Adsorption cooling systems require a heat source and little other electricity supply and have great potential of having humidity and temperature control decoupled. Integrating adsorption thermal systems with waste heat or solar energy can substantially reduce the dependence on fossil fuels. However the efficiency for SPAC is still very low and there are a few technical issues that require further

R&D work. For example the bulky size of the machine, fluctuating chilled water supply temperature due to the switching cycle, system optimization under unsteady solar radiation to name a few. One of the application issues is the SPAC performance driven by different types of solar thermal collectors. From the commercially available solar thermal collectors, the major types for building application are the flat plate (FPC), evacuated tubes (ETC) and parabolic trough collectors (PTC). The FPC is the most fundamental design to capture the heat from solar radiation. The design is very simple and the price is the most economy amount all. However, due to the low thermal resistance from the design the thermal efficiency is lower than the others. By using vacuum as the insulation, the ETC has a much higher thermal resistance and efficiency by design. However the absorbing area to aperture area ratio is lower than FPC since vacuum insulation take up more space than the other insulating materials. PTC is the only concentrating type of collector amount the three collectors selected. Due to the concentrating and reflecting mechanism, the PTC is able to operate at high temperature (>100°C). The application of PTC is usually for high quality heat operation, such as steam power generation. Similar to ETC, PTC suffered from low aperture area to collector area ratio but the thermal efficiency is higher due to the radiation concentrating design. The performance of SPAC is depending on the operating hot water temperature [2].

NOMENCLATURE

Abbreviation		Subscripts	
VCC	Vapor Compression Chiller	C	Solar Collector
SPAC	Solar Powered Adsorption Chiller	a	Ambient
FPC	Flat Plate Solar Collector	t	HWT
ETC	Evacuated Tube Solar Collector	b	Direct
PTC	Parabolic Trough Solar Collector	d	Diffuse
HWT	Hot Water Thermal Storage Tank		
COP	Coefficient of Performance		
Symbols		Greek Letters	
Q	Heating Power (W)	τ	Transmittance
A	Aperture Area (m^2)	α	absorptance
G	Solar Irradiation (W/m^2)	θ	Incident angle
U	Overall heat transfer coefficient ($W/m^2\text{ }^\circ\text{C}$)		
T	Temperature ($^\circ\text{C}$)		
C	Specific heat capacity ($W/kg\text{ }^\circ\text{C}$)		
K	function of the angle of incidence		
F'	Heat removal factor		

In this study, a model is established to investigate the performance of a SPAC system driven by the three different types of solar collectors. The system utilizes a novel composite adsorbent which is synthesized from three different kinds of the adsorbent materials as detailed in [2]. They are activated carbon, silica gel and calcium chloride. As we all know that the solar radiation is an unstable energy source. Hence, the SPAC system may not generate a stable cooling performance, particularly at night and/or in a cloudy day. With this regard, it needs to have a thermal energy storage tank for the SPAC system. This is because it not only can solve the problem of unstable solar energy from the sun, but also can provide a more stable desorption temperature to the adsorption cooling system. Therefore, a mathematical model for the solar collectors, a hot water thermal storage tank, two adsorbent beds, evaporator and condenser are considered in the simulation.

2. SIMULATION MODELING

2.1 System Description and operating principle for SPAC

The SPAC simulation model consists of the solar thermal collectors, a hot water thermal storage tank (HWT), two adsorbent beds, an evaporator and a

condenser as shown in figure 1. The solar collectors capture the heat from the solar radiation and the heat is transferred and stored at the HWT. The operations of the adsorbent bed take turns to execute the adsorption and desorption process. At the initial stage, all valves remained closed. At startup, one of the adsorbent beds will execute the adsorption process; the valve connecting the evaporator will open and allow the adsorbate (water vapor) to enter the adsorbent bed and adsorb by the adsorbent. The heat of adsorption will be removed by the cooling water circulating the adsorbent bed. On the other hand during desorption process, heat (hot water from the HWT) will apply to the adsorbent and release the adsorbate. The valve between the condenser and the adsorbent bed will be opened and allow the desorbed vapor to enter the condenser. In the condenser, the refrigerant (water vapor) condenses and transfers the heat to the cooling water. The condensed water flow naturally to the evaporator through the pressure controlled mass transfer valve. In the evaporator, the refrigerant (water) vaporizes and removes the heat from the chilled water. The water vapor will enter the adsorbent bed for adsorption process to complete the cycle. The adsorption/desorption cycle switches at a cycle time that the water uptake of the adsorbent reaches equilibrium.

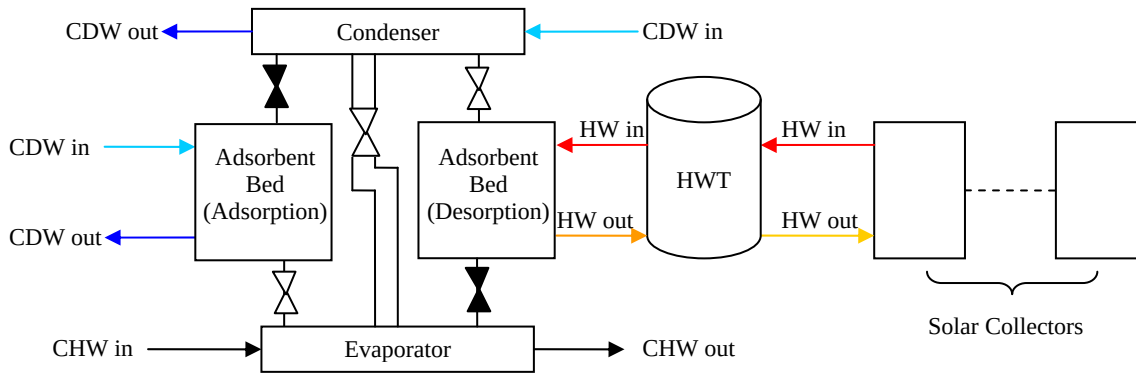


Figure 1 – A schematic diagram of a SPAC System

2.2 Simulation Equations

The performance for the FPC and ETC are usually represented by the Hottel-Whillier-Bliss equation [3], where the heat collected by the collectors is equal to the solar radiation received by the absorbing area of the collector minus the heat loss to the surrounding due to the temperature difference. Since the architecture of the PTC is different from the other two, there is an additional term called the concentration ratio to describe the characteristics of the concentrating design. For a more practical approach, the quasi-dynamic collector model is used to describe the thermal performance according to EN 12975 [4]. This model is represented by:

$$\begin{aligned} Q / A_c = & F'(\tau\alpha)K_1G_b + F'(\tau\alpha)K_2G_d - \\ & c_6uG - c_1(T_c - T_a) - c_2(T_c - T_a)^2 - \\ & c_3u(T_c - T_a) - c_4(E - \sigma T_a^4) - c_5 \frac{dT_m}{dt} \end{aligned} \quad (1)$$

where the c in this equation is representing the regression coefficient for different element of this model. This model can be simplified to a second order fit to data model as below:

$$\dot{Q} = A_c G_{solar} \left(\eta - a_1 \frac{(T_m - T_a)}{G_{solar}} - a_2 \frac{(T_m - T_a)^2}{G_{solar}} \right) \quad (2)$$

The modeling data for all three types of collectors are listed in table 1.

Table 1 – Modeling parameters for the solar collectors simulation [4-6]

	Brand 1	Brand 2	Brand 3
Type	FPC	ETC	PTC
η	0.788	0.754	0.674
a_1 (W/m ² K)	2.899	1.405	0.211
a_2 (W/m ² K ²)	0.026	0.005	0.002

The HWT plays as the buffer heat source for the entire SPAC system to stabilize the effect of the

unsteady solar radiation available. The HWT is gaining heat from the solar collectors and losing heat from the adsorption chiller and the ambient. The assumption for this simulation is the water inside the HWT is well mixed. The energy balance equation for the HWT is:

$$\begin{aligned} (m_w C_{pw} + m_{HWT} C_{p,t}) \frac{dT_w}{dt} = & \dot{m}_C C_{pw} \int dT_C - \\ & \dot{m}_{AC} C_{pw} \int dT_{AC} - U_t A_t (T_a - T_w) \end{aligned} \quad (3)$$

The simulation parameters for the HWT were studied by Cynthia et al. in [7]. The area of the HWT is 2.31 m² and the overall U-value is 0.86 W/m²-K. The simulation equations for the adsorption chiller by using the novel adsorbent was previously discussed by Tso et al in [3] and summarized in the Appendices.

3. RESULTS & DISCUSSION

The solar COP was simulated for the SPAC at hot water operating temperature at 60°C and 80°C. The result is shown in figure 2-3. The efficiency of the solar collectors decreases with higher hot water temperature due to the heat loss to the ambient but the efficiency for the adsorption chiller increases. From the results, the PTC has the highest solar COP at both temperatures. The ETC and FPC have similar efficiency at lower temperature with ETC has a slim edge during the winter period.

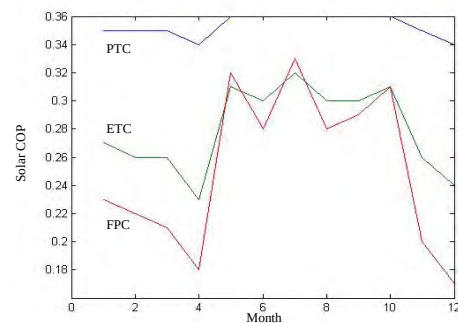


Figure 2 – Monthly Solar COP of the SPAC System with 60°C hot water temperature

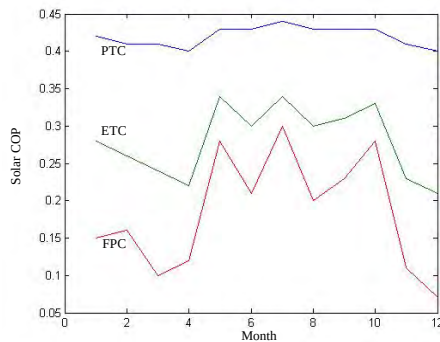


Figure 3 – Monthly Solar COP of the SPAC System with 80°C hot water temperature

Each of the three simulated solar collectors has different characteristics when driving the SPAC. The solar COP comparisons for each collector running at 60°C and 80°C are shown in figure 4-6. From figure 4, the efficiency difference for FPC during summer time is closer than in winter since the FPC has low thermal resistance. For the ETC, with the high thermal insulation, the solar COP are very similar with different operating temperature. Since PTC is designated for high temperature operation, the thermal efficiency decreases with lower operating temperature.

The temperature profile of the HWT was simulated by setting the initial temperature at the nominal operating temperature as shown in figure 7-10. As expected, the temperature within the HWT is very stable throughout the operation. The temperature varies within 1.5°C from the nominal operation. When the SPAC is operating at 60°C, the heat captured is more than the chiller required, therefore the temperature is going up from the starting point. Vice versa, when the SPAC is operating at 80°C the chiller is consuming more heat than the collectors can captured therefore the temperature is going down from the starting point.

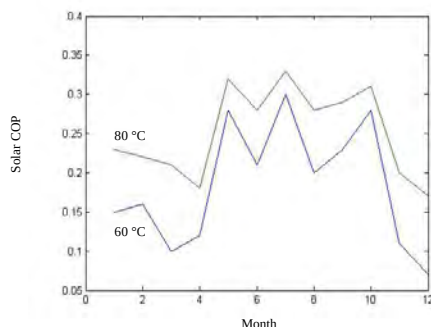


Figure 4 – FPC Monthly Solar COP Comparison

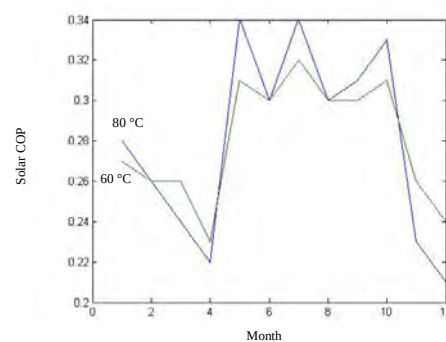


Figure 5 – ETC Monthly Solar COP Comparison

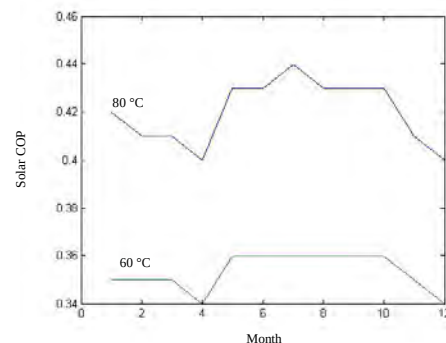


Figure 6 – PTC Monthly Solar COP Comparison

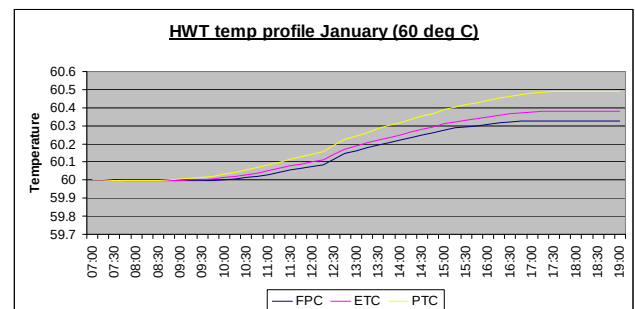


Figure 7 HWT temperature profile for January at 60 °C

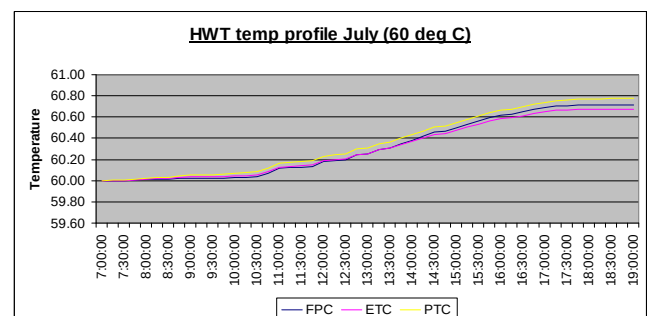


Figure 8 HWT temperature profile for July at 60 °C

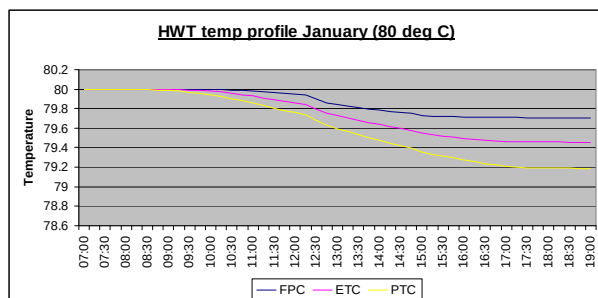


Figure 9 HWT temperature profile for January at 80 °C

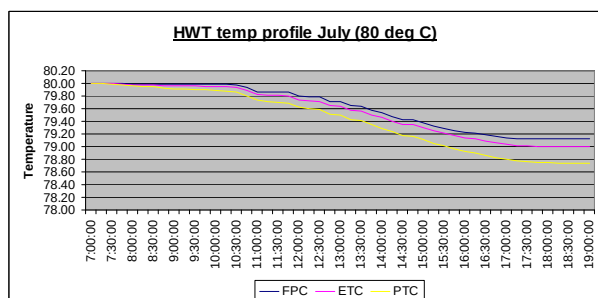


Figure 10 HWT temperature profile for July at 80 °C

4. CONCLUSION

The SPAC system driven by three types of solar collectors were simulated. The solar thermal system and the adsorption chiller have different efficiency at different operating parameters such as hot water supply temperature and flow rate. These parameters can be optimized under certain ambient environment and the operation conditions [8]. By using the ETC, the thermal efficiency is more consistent throughout the range due to the high thermal resistance, therefore it is the most suitable collectors when driving the SPAC. The PTC has the highest thermal efficiency thanks to the concentrator reflective surface for concentration and reflection of the solar radiation. However, due to the low aperture area ratio, PTC requires the most space to install and this is not an ideal solution for urbanized cities. By using the HWT, the SPAC system is able to provide stable cooling supply with unsteady solar radiation available. It is recommended to investigate the application of advance control theories, e.g. adaptive control to improve the efficiency of the SPAC system by optimizing the entire system operation under rapidly changing conditions.

ACKNOWLEDGEMENT

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APPENDICES

Table A – Nomenclature and values of parameters adopted in the simulation

Symbol		Value	Unit
A_{ad} (Heat transfer area in adsorber)	in	2.46	m ²
A_{de} (Heat transfer area in desorber)	in	2.46	m ²
A_{cond} (Heat transfer area in condenser)	in	3.73	m ²
A_{eva} (Heat transfer area in evaporator)	in	1.91	m ²
$C_{p,cu}$ (Specific heat of copper)		386	J/kg.K
$C_{p,al}$ (Specific heat of aluminum)		905	J/kg.K
$C_{p,w}$ (Specific heat of water in liquid phase)		4180	J/kg.K
$C_{p,v}$ (Specific heat of water in vapor phase)		1900	J/kg.K

vapor phase)		
$C_{p,c}$ (Specific heat of chilled water)	4200	J/kg.K
$C_{p,com}$ (Specific heat of composite adsorbent)	930	J/kg.K
ΔH (Heat of Adsorption)	2.30×10^6	J/kg
Kads (Adsorption rate coefficient)	1.30×10^{-3}	1/s
K_{des} (Desorption rate coefficient)	1.20×10^{-3}	1/s
L_w (Latent heat of vaporization)	2.6×10^6	J/kg
U_{ad} (Overall heat transfer coefficient of adsorber)	1602.56	W/m.K
U_{de} (Overall heat transfer coefficient of desorber)	1724.14	W/m.K
U_{cond} (Overall heat transfer coefficient of condenser)	4115.23	W/m.K
U_{eva} (Overall heat transfer coefficient of evaporator)	2557.54	W/m.K
$W_{al,ads/des}$ (Weight of aluminum fin in adsorber/desorber)	51.2	kg
$W_{cu,ads/des}$ (Weight of copper in adsorber/desorber)	64.04	kg
W_{com} (Weight of composite adsorbent)	47	kg
$W_{cu,eva}$ (Weight of copper in evaporator)	12.45	kg
$W_{cu,cond}$ (Weight of copper in condenser)	24.28	kg
W_{cw} (Weight of condenser cooling water)	15	kg
$W_{eva,initial}$ (Initial weight of water in the chilled water tank)	50	kg

Simulation equations for the adsorption chiller [2]

Adsorption equilibrium for composite adsorbent/water vapor:

$$\omega_{eq} = 0.375 \left[\frac{P_s(T_w)}{P_s(T_s)} \right]^{1.6}. \quad (A1)$$

$$P_s(T) = \exp(23.5 - \frac{3984.9}{T - 29.7}). \quad (A2)$$

Adsorption rate:

$$\frac{d\omega_{ads}}{dt} = K_{ads}(\omega_{ads eq} - \omega_{ads}). \quad (A3)$$

$$\omega_{ads} = \omega_{ads eq} - (\omega_{ads eq} - \omega_{ads o}) \exp(-K_{ads}t). \quad (A4)$$

Desorption rate

$$\frac{d\omega_{des}}{dt} = -K_{des}\omega_{des}. \quad (A5)$$

$$\omega_{des} = \omega_{des eq} - (\omega_{des eq} - \omega_{des o}) \exp(-K_{des}t). \quad (A6)$$

Adsorption energy balance:

$$\frac{d}{dt}[(m_{com-ads}C_{p,com-ads} + m_{com-ads}C_{p,w}\omega_{ads} + C_{pal}m_{al fin} + m_{cu-ad}C_{p,cu})T_{ads}] = \Delta H m_{com-ads} \frac{d\omega_{ads}}{dt} + m_{com-ads}C_{p,v} \quad (A7)$$

$$(T_{eva} - T_{ads}) \frac{d\omega_{ads}}{dt} + \dot{m}_{cwa} C_{p,w} (T_{cwa in} - T_{cwa out})$$

Cooling water leaving temperature (from adsorber):

$$T_{cwa out} = T_{ads} + (T_{cwa in} - T_{ads}) \exp\left(\frac{-U_{ad}A_{ad}}{\dot{m}_{cwa} C_{p,w}}\right). \quad (A8)$$

Desorption energy balance:

$$\frac{d}{dt}[(W_{com}C_{p,com} + W_{com}C_{p,w}\omega_{des} + C_{pal}W_{fin al} + C_{p,cu}W_{cu des})T_{des}] = \Delta H W_{com} \frac{d\omega_{des}}{dt} + \dot{m}_{hot} C_{p,w} (T_{hw in} - T_{hw out}) \quad (A9)$$

Hot water leaving temperature:

$$T_{hw out} = T_{des} + (T_{hw in} - T_{des}) \exp\left(\frac{-U_{de}A_{de}}{\dot{m}_{hot} C_{p,w}}\right). \quad (A10)$$

Condenser energy balance:

$$\frac{dT_{cond}}{dt} (C_{p,cu}W_{cu,cond} + W_{cw}C_{p,w}) = -L_w W_{com} \frac{d\omega_{des}}{dt} - W_{com}C_{p,w} (T_{des} - T_{cond}) \frac{d\omega_{des}}{dt} + \dot{m}_{cwc} C_{p,w} (T_{cwc in} - T_{cwc out}) \quad (A11)$$

Cooling water leaving temperature (from condenser):

$$T_{cwc out} = T_{cond} + (T_{cwc in} - T_{cond}) \exp\left(\frac{-U_{cond}A_{cond}}{\dot{m}_{cwc} C_{p,w}}\right). \quad (A12)$$

Evaporator energy balance:

$$\frac{d}{dt}[(W_{eva}C_{p,w} + C_{p,cu}W_{cu,eva})T_{eva}] = -L_w W_{com} \frac{d\omega_{ads}}{dt} - W_{com}C_{p,w} (T_{cond} - T_{eva}) \frac{d\omega_{des}}{dt} + \dot{m}_{ch} C_{p,c} (T_{chw in} - T_{chw out}) \quad (A13)$$

Chilled water leaving temperature:

$$T_{chw out} = T_{eva} + (T_{chw in} - T_{eva}) \exp\left(\frac{-U_{eva}A_{eva}}{\dot{m}_{ch} C_{p,c}}\right). \quad (A14)$$

Mass balance in evaporator:

$$\frac{dW_{eva}}{dt} = -W_{com} \frac{d\omega_{des}}{dt} - W_{com} \frac{d\omega_{ads}}{dt}. \quad (A15)$$

Evaluation of performance:

$$SCP = \frac{\dot{m}_{ch} C_{p,c} \int_0^{t_{cycle}} (T_{chw,in} - T_{chw,out}) dt}{W_{com} t_{cycle}}. \quad (A16)$$

$$COP = \frac{\dot{m}_{ch} C_{p,c} \int_0^{t_{cycle}} (T_{chw,in} - T_{chw,out}) dt}{\dot{m}_{hot} C_{p,w} \int_0^{t_{cycle}} (T_{hw,in} - T_{hw,out}) dt}. \quad (A17)$$

NUMERICAL SIMULATION OF INDOOR AIRFLOW BY PARALLEL METHOD

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ABSTRACT

Parallel computing plays an essential role in the development of CFD technology. It is necessary to apply parallel computing to the numerical simulation of indoor airflow, since serial computing on a single CPU cannot meet the demand for data size and time, especially in large space buildings. In this paper, a two-dimensional ventilation model was taken for example, and its indoor airflow was simulated numerically by the parallel algorithm of traditional software ANSYS FLUENT at first. The computed results indicated that parallel strategy reduced the computational cost significantly under same computing precision. Then, its performance of parallel computing was discussed, and a new parallel algorithm for the problem was analyzed.

Keywords: Indoor airflow; numerical simulation; parallel algorithms; Schwarz domain decomposition method.

1. INTRODUCTION

With the rapid development of modern science and technology nowadays, computing has become the third important way after theory and experiment. High performance computing plays a foundational role in all areas of economic and social development. Fast computing speed and short response time has become the core needs of many industries.

The demand of industrial design in terms of data size and time is rapidly expanding, which the calculation in Computational Fluid Dynamics (CFD) by a single CPU can not meet. To be more exact, the demand is reflected in the following two aspects.

On the one hand, there are some large-scale computing problems in industrial applications, such as supersonic flow simulation in aircraft design, shape design of high performance car with low resistance, taking off and landing process of aircraft. With the vigorous development of architecture industry, various kinds of large space buildings, such as theaters, stadiums, exhibition halls and airport terminals, have been built or used in recent years. Generally, these large space buildings are more than 5 meters high and 10000 cubic meters, and their indoor environments are more complicated [1]. Numerical simulation of indoor air flow is carried out to understand whether the indoor environment status meet the comfort requirements before the design and construction of building. CFD simulation of these typical scale industries needs nearly five million grids and more than 5GB computer memory. It needs up to tens, hundreds or even thousands of hours to complete the computation. The

computing of such a large-scale problem is difficult to achieve without the support of high performance computer. In addition, it puts forward further demand for CFD high performance computing to obtain results in a short period. On the other hand, computing accuracy needs to be increasingly improved. In order to meet the accuracy requirement, people try to give up hypothesis to solve problems directly, such as the direct numerical simulation method for turbulence which is different from simplified turbulence model based on statistical theory.

Based on the above two reasons, despite the improvement of numerical algorithms for serial programs, the CFD researchers find that parallel computing is almost the only method to improve the computational efficiency and optimize product design greatly. In the past 10 to 15 years, the development of computer technology, especially network technology, has provided good conditions for the realization of CFD high performance computing. The computing platform develops from initial large supercomputers to minicomputer-main clusters today. Grids and network computing provide an unprecedented super computing power, while the development of parallel algorithm and software is hysteretic seriously. Some investigations show that the efficiency of existing parallel program usually can only achieve 6% of the floating-point peak performance, which is very low, and the development cycle is long. It is a challenge for CFD researchers to develop high performance CFD tools by the use of current computer and network technology [2].

CFD parallel computing has been applied in aerospace field [2]. In 1998, Veer. N. Vatsa, et al. [3]

studied the parallel computing of TLNS3D large CFD software and calculated viscous fluid flow of complex size. In 1999, J. Reuther et al. [4] adopted domain decomposition algorithm and parallel technology in the shape design of supersonic aircraft. The results indicated that the parallel computing greatly accelerated the computing speed. In 2004, R. Consul, et al. [5] studied fluid flow and heat transfer in large storage tanks by parallel computation. In 2005, based on Windows operating system and CFD software FLUENT, Ting Luo [6] researched different methods to construct hardware platform of parallel computing, and he calculated the turbulence flow field of a typical pollutant dispersion process in a valley. The advantages and disadvantages of parallel computing method were discussed by the analysis of two kinds of parallel processing method, Symmetrical Multi-Processing (SMP) and Massively Parallel Processing (MPP). Xiaohua Xin [7] calculated the flow field outside a moving car by parallel computing on commercial CFD software FLUENT. The results indicated that parallel strategy greatly shortened the computation time. However, with the increase of the number of processors, the utilization efficiency of a single CPU decreased gradually, so it's a waste of resources to adopt parallel algorithm in the case of less grid number. In 2006, J. Tsuchida, et al. [8] adopted parallel technology in the direct numerical simulation of the sound transmission in the air. In Nanjing University of Aeronautics & Astronautics, Xuede Wang [9] researched parallel direct numerical simulation on unstructured grids by autonomic programming using Fortran90 language, based on parallel architecture of PC-CLUSTER cluster and MPI. Parallel computation was carried out on the hypersonic flow around a body, and the ideal speed-up ratio and computational efficiency were obtained. In 2008, in Tsinghua University, Zhi Wang[10] carried out the multi-dimensional simulation of Homogeneous Charge Compression Ignition (HCCI) engine by using parallel computation and reduced mechanisms, based on 16-CPU computer cluster system and MPI for computing. The results showed that the maximum profit can be obtained by 8 CPUs in 3D simulation of HCCI engine.

The current research on the application of parallel algorithm to indoor airflow simulation is less, and the application of new parallel algorithm is even blank. In this paper, taking a two-dimensional ventilation model for example, we would conducted its indoor airflow simulation by parallel algorithm of traditional software ANSYS FLUENT and discussed the performance of parallel computing method.

2. MATHEMATICAL PHYSICAL MODEL

2.1 PHYSICAL MODEL

In order to validate the computational results, we took a two-dimensional forced convection model [11] as an example to carry out numerical simulation of

indoor airflow, as shown in Figure 1. The velocity of supply air is $u_{in}=15\text{m/s}$; the height of computational domain is $H=89.3\text{mm}$; the length is $L=3H$; the height of inlet is $h=0.056H$; the height of outlet is $h'=0.16H$.

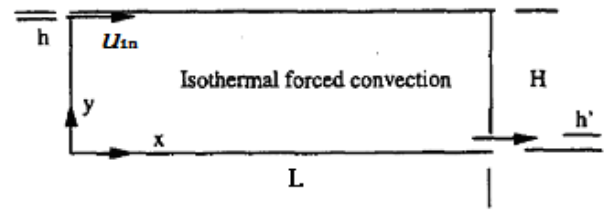


Figure 1 Sketch of the forced convection case

2.2 MATHEMATICAL MODEL

Indoor air is a kind of low speed flow at normal temperature and pressure and can be regarded as an incompressible fluid. But the air flow is often turbulent. Its governing equations are expressed as

Mass continuity:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} = 0 \quad (1)$$

Momentum:

x-direction:

$$\begin{aligned} & \frac{\partial (\rho u)}{\partial t} + \frac{\partial (\rho u u)}{\partial x} + \frac{\partial (\rho u v)}{\partial y} \\ &= -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left[\mu_e \left(\frac{\partial u}{\partial x} + \frac{\partial u}{\partial x} \right) \right] + \frac{\partial}{\partial y} \left[\mu_e \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right] \end{aligned} \quad (2)$$

y-direction:

$$\begin{aligned} & \frac{\partial (\rho v)}{\partial t} + \frac{\partial (\rho v u)}{\partial x} + \frac{\partial (\rho v v)}{\partial y} \\ &= -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left[\mu_e \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right) \right] + \frac{\partial}{\partial y} \left[\mu_e \left(\frac{\partial v}{\partial y} + \frac{\partial v}{\partial y} \right) \right] \end{aligned} \quad (3)$$

where u = mean velocity component in x-direction; v = mean velocity component in y-direction; x , y = perpendicular axe coordinates, ρ = air density; p = pressure; μ_e = effective viscosity; β = thermal expansion coefficient of air.

The turbulent influences are lumped into the effective viscosity as the sum of the turbulent viscosity, μ_t , and laminar viscosity, μ :

$$\mu_e = \mu_t + \mu \quad (4)$$

3. PARALLEL COMPUTING RESULTS ON ANSYS FLUENT

In WINDOWS operating system, the airflow in the ventilation model in Figure 1 was simulated on ANSYS FLUENT software. We adopted k- ϵ two-equation turbulence model to the numerical simulation, and the value of residual was 10^{-5} . The CPU was core i7-3770; the dominant frequency of CPU was 3.4GHz; the number of cores is four; the memory capacity is 4G. Figure 2 shows the computational domain meshed by 10740 structured grids.

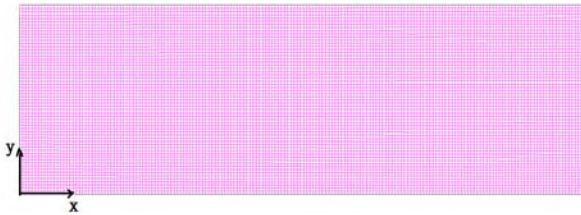


Figure 2 Sketch of grids

ANSYS FLUENT software owns a parallel module which can be used for parallel computing. In this case, the computational space was divided into two non-overlapping domains. Figure 3 shows the two domains.



Figure 3 Sketch of grid partitions

ANSYS FLUENT assigned the two divided subdomains to two compute nodes. The boundary and initial conditions and geometric information of the grids were loaded into the memory of the compute node corresponding to each subdomain. The computational process started in each compute node, and the main process scheduled the computing of each compute node. In the iteration of a full scanning process, the data was exchanged on the artificial boundary between each compute node, namely the communication among compute nodes. The main process conducted the all data collection, convergence criterion check, disk writing or other operations as needed [2]. Figure 4 shows the parallel computing process.

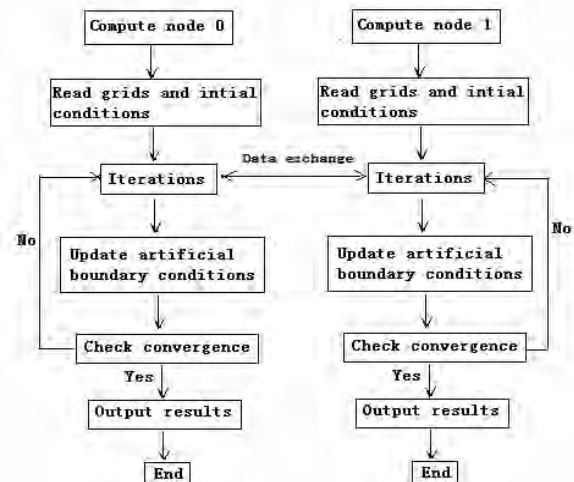


Figure 4 Flow chart of parallel computing

On ANSYS FLUENT, the airflow in the model was simulated respectively by serial and parallel computing method. Figure 5 shows the results of velocity field. The computed velocity profiles are compared in Figure 6 with experimental data [12] in the vertical line $x/H=2$. The numerical results of serial and parallel computing are basically consistent with experimental data, so parallel computing can ensure the same precision with serial computing.

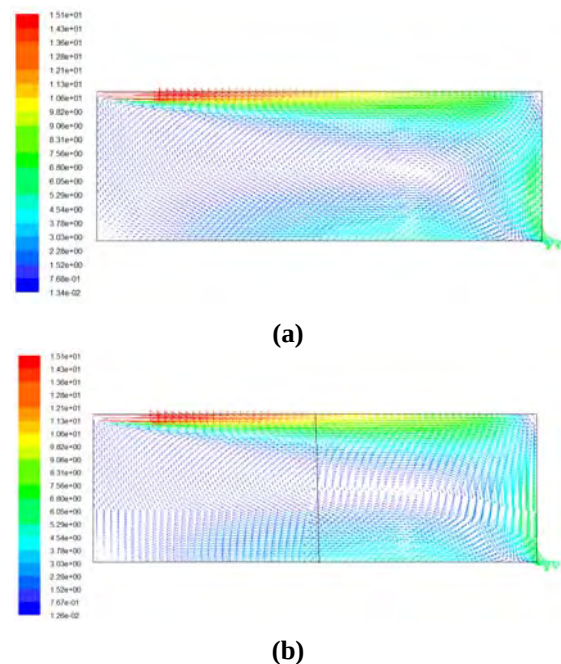


Figure 5 Comparison of velocity field: (a) serial computing, (b) parallel computing.

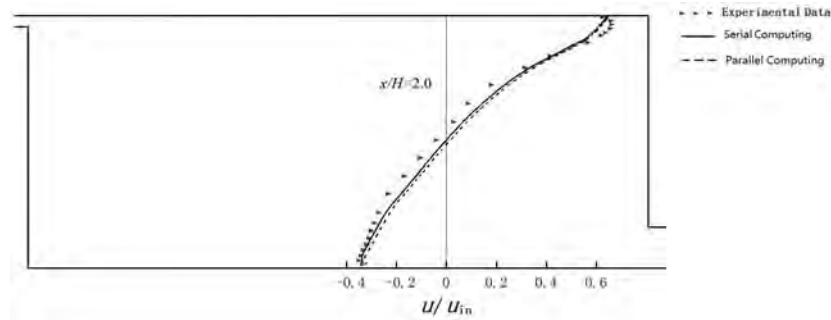


Figure 6 Comparison of velocity profiles in the vertical line $x/H=2$

In table 1, the iterations and cost time between serial and parallel computing are compared.

Table 1 Comparison of computing performance of serial and parallel computing

Calculation type	Serial computation	Parallel computing
The number of compute node	1	2
Iterations	1951	1294
Average wall-clock time per iteration	0.026 sec	0.014 sec
Data transfer per iteration	0 MB	0.019 MB
Total wall-clock time	50.838 sec	18.261 sec
Total CPU time	95.176 sec	54.778 sec

Compared with serial computing, parallel computing reduced the iterations by 33.7% and shortened the total CPU time by 42.4%. The example shows that parallel computing accelerates the computing speed significantly based on a certain precision.

4.DISCUSSION ON A NEW PARALLEL METHOD FOR INDOOR AIRFLOW

In this section, we will discuss a kind of parallel algorithm for the indoor airflow simulation. For simplicity, energy and species concentrations will not be included in the model. Noting that in classical SIMPLE algorithm, two momentum equations need to be solved to obtain velocity components in x-direction and y-direction during the iterations in SIMPLE algorithm. Since the scale of the problem is always very large in practical problems, then we can adopt a kind of domain decomposition method, Schwarz methods, to solve momentum equations in parallel in order to improve the computational efficiency.

Schwarz methods are the oldest domain decomposition methods, which were invented by Hermann A. Schwarz in 1896 as an analytical tool to prove results obtained by Riemann through a minimization principle. By the arrival of parallel computers in the last century, various Schwarz methods have been presented and analyzed. Optimized Schwarz methods grew out of the comments by Lions and

Hagstrom et al, which used more general operators in the robin transmission conditions, see [13, 14]. Nataf also stated in [15] that “The rate of convergence of Schwarz and Schur type algorithm is very sensitive to the choice of interface conditions. ...It has been remarked that absorbing (or artificial) boundary conditions are a good choice.” Over the last 15 years, a lot of research has studied the optimal choice of parameters in the transmission conditions of optimized methods, and now plenty of results are available for many kinds of PDEs, see [16, 17].

4.1 Schwarz methods for the model problem

Noting that the mathematical model discussed in the last section, the momentum equations in the SIMPLE iterations can be written as (take the equation in x-direction for example)

$$\frac{\partial(\rho u)}{\partial t} + u^{old} \frac{\partial(\rho u)}{\partial x} + v^{old} \frac{\partial(\rho u)}{\partial y} = \mu^{old} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) + \mu \left(\frac{\partial^2 u^{old}}{\partial x^2} + \frac{\partial^2 v^{old}}{\partial y^2} \right) - \frac{\partial p^{old}}{\partial x}, i = 1, 2.$$

Therefore, these momentum equations can be rewritten as the following uniform form.

$$L(u) \triangleq \bar{a}(x, y) \cdot \nabla u - \mu(x, y) \Delta u = f \quad \text{in } \Omega,$$

$$C(u) = g \quad \text{on } \partial\Omega,$$

where $\Omega \subset \mathbf{R}^2$, $\bar{a}(x, y) = (a_1(x, y), a_2(x, y))$ is known velocity components in both directions, f is the source term, C is a linear operator such as the identity and the normal derivative, and u is velocity components which needs to be computed. For simplicity to state the methods, we suppose that the coefficients $a_1(x, y)$, $a_2(x, y)$ and $\mu(x, y)$ in the above equation are all constants, i.e.

$$L(u) = \bar{a} \cdot \nabla u - \mu \Delta u = f \quad \text{in } \Omega,$$

$$C(u) = g \quad \text{on } \partial\Omega.$$

4.1.1 Overlapping Schwarz methods

The spatial domain is divided into two overlapping subdomains $\Omega_1 = (0, L_{x1}) \times (0, L_y)$ and

$\Omega_2 = (L_{x2}, L_x) \times (0, L_y)$, where $\delta = L_{x1} - L_{x2}$ is

overlapping value, see Figure 8. Then overlapping Schwarz method is

$$\begin{aligned}
 Lu_i^{(k+1)}(x, y) &= f(x, y), \text{ in } \Omega_i, \\
 B_i u_i^{(k+1)}(L_{xi}, y) &= B_i u_j^{(k)}(L_{xi}, y), \text{ on } \Gamma_i, i, j = \{1, 2\}, i \neq j, \\
 C(u) &= g \quad \text{on } \partial\Omega,
 \end{aligned}$$

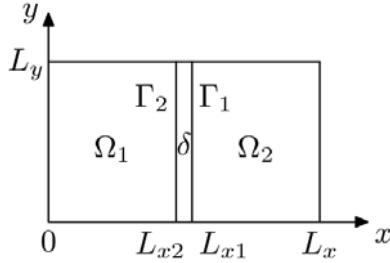


Figure 7 Overlapping case

where $\Gamma_i = L_{xi} \times (0, L_y) (i=1, 2)$ are artificial boundaries, operators $B_i (i=1, 2)$ are called boundary transmission conditions. When $B_i = I (i=1, 2)$, i.e., identity operator, the boundary condition is called Dirichlet condition. In order to improve the efficiency of Schwarz methods, order 0 Robin condition $B_i = \partial_{n_i} + p_i (i=1, 2)$ and order 2 condition $B_i = \frac{\partial}{\partial n_i} - \alpha_i - \beta_i \frac{\partial}{\partial y} - \gamma_i \frac{\partial^2}{\partial y^2}$ are introduced, where $p_i, \alpha_i, \beta_i, \gamma_i (i=1, 2)$ are parameters.

4.1.2 Non-overlapping Schwarz methods

In this case, the spatial domain is divided into two non-overlapping subdomains $\Omega_1 = (0, L) \times (0, L_y)$ and $\Omega_2 = (L, L_x) \times (0, L_y)$, see Figure 8. Then overlapping Schwarz method is

$$\begin{aligned}
 Lu_i^{(k+1)}(x, y) &= f(x, y), \text{ in } \Omega_i, \\
 B_i u_i^{(k+1)}(L, y) &= B_i u_j^{(k)}(L, y), \text{ on } \Gamma, i, j = \{1, 2\}, i \neq j, \\
 C(u) &= g \quad \text{on } \partial\Omega,
 \end{aligned}$$

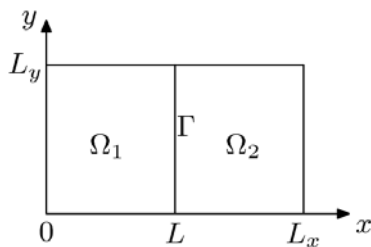


Figure 8 Non-overlapping case

where $\Gamma = L \times (0, L_y)$ is artificial boundary and operators $B_i (i=1, 2)$ are boundary transmission conditions as those in the overlapping case. Especially, when $B_i = I (i=1, 2)$, Schwarz method is not convergent for the non-overlapping case.

4.1.3 A new parallel method

In this section, we take SIMPLE method and Schwarz method as the outer iteration method and the inner iteration method, respectively, in order to achieve the simulation in parallel for indoor airflow model. The basic framework of this method is as follows.

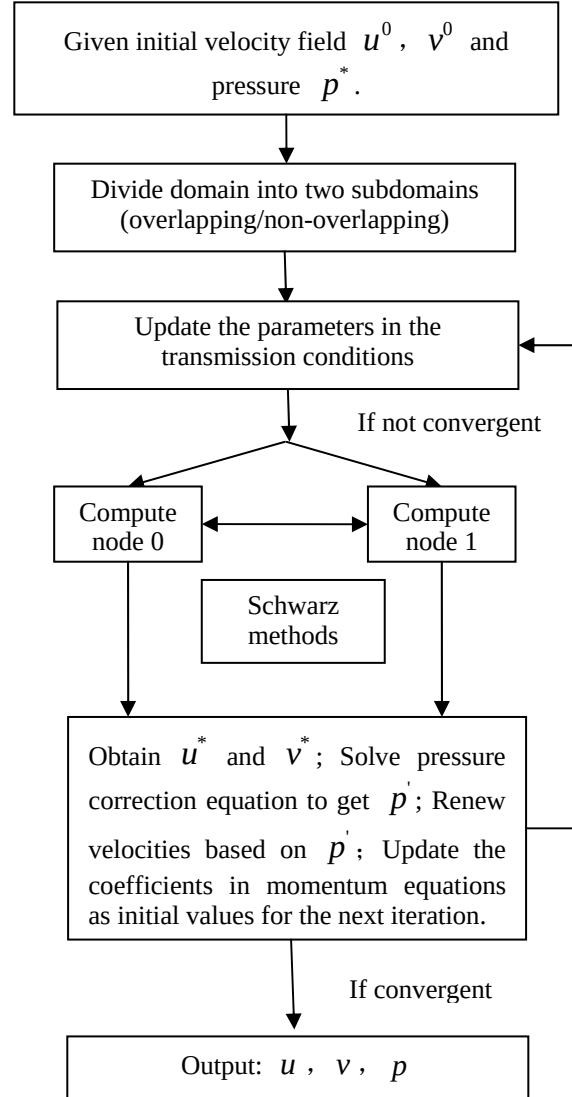


Figure 9 Framework of the new Parallel method

In our future work, we will implement this new parallel method for the indoor airflow, and we believe that this strategy will work well and improve computational efficiency. We will also take our effort in developing some new parallel techniques to improve the parallel efficiency.

5. SUMMARY

In this paper, a two-dimensional ventilation model was taken for example, and its indoor airflow was simulated numerically by the parallel algorithm of traditional software ANSYS FLUENT in WINDOWS

operating system. Under the computational conditions, the numerical results of serial and parallel computing are basically consistent with experimental data. Parallel computing reduced the iterations by 33.7% and shortened the total CPU time by 42.4% compared with serial computing. We conclude that parallel computing accelerates the computing speed significantly based on a certain precision.

This paper also proposed a new parallel algorithm, which was based on the SIMPLE method as the outer iteration and took the Schwarz domain decomposition as the inner iteration. This new algorithm will be implemented in our future work to improve the parallel computational efficiency.

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EXPERIMENTAL RESEARCH OF ON-SITE DETECTION OF WALL HEAT TRANSFER COEFFICIENT WITH COLD BOX-HEAT FLOW METER METHOD

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ABSTRACT

Due to the thermal performance of the self-thermal insulation wall are influenced by the main material, temperature, moisture content and many other factors, the wall heat transfer coefficient detection facing a difficulty, which is how to determine the heat transfer coefficient of the optimum detection time. In this paper, using the cold box-heat flow meter method on the wall heat transfer coefficient testing was experimentally studied. Through the establishment of model experiment, and field measurement of each physical quantity needed with experimental analysis, analysis of the measured physical quantity, in order to determine the time and the laws of moisture balance, provide the theoretical basis for determining the heat transfer coefficient optimum testing time of the self-thermal insulation wall.

Keywords: Self thermal insulation wall, heat transfer coefficient, hot and cold boxes-heat flow meter method, heat flow meter method, the optimum detection time.

1. INTRODUCTION

Consumption of non-renewable energy like petroleum and coal, and a weak popularization of alternative energy is now reminding people with a huge problem- building energy saving, and the way to conserve energy in a building has gained more attention from society. Under the push of national policy and energy saving standard, wall thermal insulation technology is developing rapidly, and some technical systems have been seen in recent years. However, a further verification is needed when evaluating the practical application of wall thermal insulation technical system and determining whether the thermal insulation wall meet the building energy saving standard[1].

The detection of wall heat transfer coefficient if an important index to evaluate the building's thermal insulation features. For the coefficient of wall structure, cold box-heat flow meter method can give an effective detection. However, here is a difficulty no matter it is practical or experimental detection that is the determination of optimal detection time. A traditional experience used to air dry the wall more than 15 days under the environment of natural ventilation. However, with too many uncertain factors in natural ventilation, it directly impacts the heat and moisture transfer process between wall structure and environment.

Wei Tan and Jun Ren[2] have put out ways to employ microwave humidity test system to determine the humidity ratio of the main material of the wall, leading a humidity tracking measurement, which includes slightly destroy, on the wall's long natural air dry process in order to determine the optimum time to

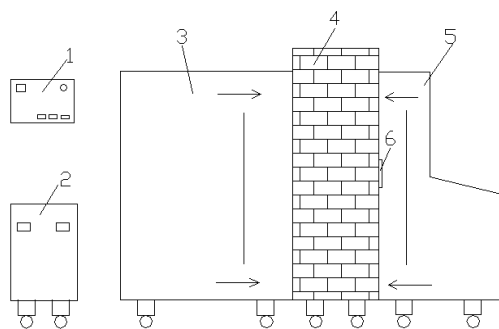
measure the heat transfer coefficient. Since the MOIST microwave humidity test system can only measure the value of humidity of a single material, it can't lead a tracking measurement on the humidity of the entire envelope. In this paper, we mainly use cold box-heat flow meter method and wall humidity measuring instrument to test the heat flux and humidity ratio of entire main material of different type of building envelope, and it has little destroy on the wall. Leading a tracking measurement on the heat flux and humidity value in the wall's natural air dry process, we can find out the law and time when it achieves thermal and humidity balance and provide theoretical basis to determine the optimum time to test the heat transfer coefficient.

2. PRACTICAL MEASURING INSTRUMENT AND PRINCIPLE OF COLD BOX-HEAT FLOW METER METHOD

2.1 Instruction of practical measuring instrument of cold box-heat flow meter method

The instrument mainly includes[3]: cold box, heat box, distribution box, controller, heat flow meter, thermocouple, JTRG-II building temperature and heat flow automation testing system and so on. JTRG-II building temperature and heat flow automation testing system is a new instrument to test temperature and heat flow, it can be used cooperatively with PC and is externally connected with several temperature and heat

flow sensor. It is used to observe and record temperature and heat flow, and has features like high precision, friendly man-machine interface, fewer human interfere and public used AC&DC. The instruction use JTRG-II to assemble data tested by heat flow meter and thermocouple; employing electric heating and compression refrigeration to form a stable and adjustable thermal field in both cold box and heat box; utilizing PID control module to improve heating power so as to precisely control the calefaction heat quantity of electric heater. Using small refrigeration compressor to supply cold in cold box; vertical direction of both in cold box and heat box are all installed with flow guiding screen which can help to form a stable convection on the wall surface; the surface of internal wall of cold box is attached with heat flow meter and the surface of internal wall, both in cold and heat box, are attached with thermocouple, and the thermocouple stays closely to heat flow meter. In addition, sensors are installed both inside the cold and heat box and they are used to measure the air temperature inside the box. The mechanism of cold heat-box flow meter method wall heat transfer coefficient measuring instrument is shown as below:



1 JTRG- II building temperature and heat flow automation testing system; 2 power distribution box; 3 hot box; 4 wall; 5 cold box; 6 heat flow meter.

Figure 1 structure scheme of cold heat-box flow meter method wall heat transfer coefficient measuring devise

2.2 Principle of measurement

Heat flow method[4] mainly use both heat flow meter and thermocouple to measure the wall's heat flow and temperature outside and inside of the wall, calculating the heat transfer coefficient through analyzing these data and consequently find out whether this building can meet the energy saving standard. Cold heat-box flow meter is based on the measuring principle of heat flow meter, precisely controlling the inside and outside temperature of the wall through creating a man-made environment by cold and heat box, and then determining the heat transfer coefficient by heat flow meter method. Cold and heat box is a automatic temperature control devise, if has functions of cooling and heating, it use temperature sensor to assemble the air temperature of cold and heat box, and setting temperature according to the request in order to create a environment of one-dimensional stable heat transfer, then we can test the wall heat transfer coefficient by

using the heat flow meter and it effectively overcome problems related with region and season.

3. DETERMINATION OF DAMPNES VALUE

When wall material is in heat and humidity balance with air in a certain situation (a certain temperature and certain humidity), the dampness value of that material is balance humidity, and it is considered as a optimum time to measure the heat transfer coefficient, the heat transfer coefficient measured can most accurately reflect envelope's practical heat insulation capacity. According to the research, by leading a tracking measurement until the dampness value of the wall reaches the range of balance humidity, we can start measure of heat transfer coefficient.

Take the currently widely used shale porous brick as an example, by leading a liner fitting treatment between measurement value of wall humidity measuring instrument and the practical humidity ratio at a correspondingly time point, we can achieves a goal to determine the humidity ratio for different type of materials. In this experiment, the wall humidity measure devise mainly use electromagnetic wave to penetrate the internal wall in order to measure, this method excludes destroyable measurement like burrow and inserting, it lead a nondestructive testing, it also can draw a more accurate measure result.

3.1 Liner conversation of measure value and mass humidity bearing rate

To accurately reflect the internal situation of humidity, we use samples in size of (240×190×115) mm and (600×240×200) mm.



Figure 2 Samples of (240×190×115) mm and (600×240×200) mm

Firstly, immerse the sample in the water of 25℃ until it reaches the situation of saturation and moisture absorbed (after a long immersion, take out the sample two times every one hour and weighting it until its weight does not change any more), taking out the sample at this time, absorbing the moisture left on the samples' surface by sponge, then prevent the internal moisture, when the sample is in saturated and moisture absorbed state, from spreading out to air through using

fresh-keeping film to cover the sample, recording the weight at this time as m_1 , using wall humidity measuring device to record the relative humidity of the sample as w_1 . Subsequently putting the samples into drying oven of 90°C and starting a drying process until the samples are completely dried (after a long drying process, take out the sample two times every one hour and weighting it until its weight does not change any more), taking out the sample, to prevent the sample from absorbing moisture from air when it is in bone dry state, recording the weight of the sample as m_2 , and then using wall humidity measuring device to record the relative humidity as w_2 after the temperature of the samples are the same as indoor air's through cooling the samples in environment of constant temperature and humidity.

As stated above, we can start a liner fitting process between measured humidity value and mass humidity bearing rate, and getting the relationship between them. Specific steps are as following: when the sample is in saturated and moisture absorbed state, the quality is m_1 , and the measured humidity value at this time is w_1 . When the sample is bone dry, its quality is m_2 , and the corresponding measured humidity value, which is a dimensionless model, at this time is w_2 . Thus, the mass humidity bearing rate is $(m_1 - m_2) / m_1$, and the mass humidity bearing rate, when it is w_2 , is 0. Conducting a liner fitting at these two points, a relationship between them can be drawn.

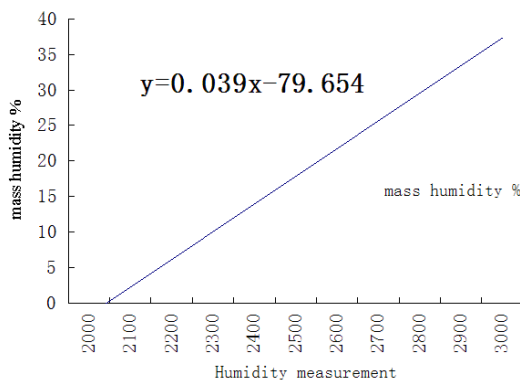


Figure 3 (240×190×115) mm fitted relationship of sample

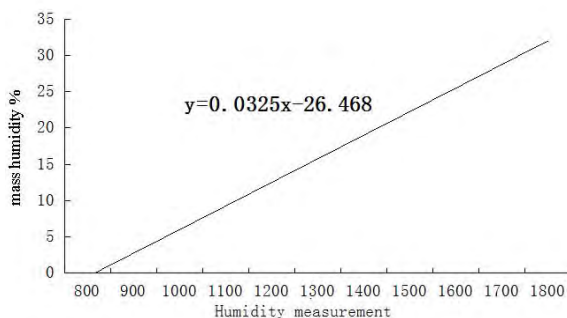


Figure 4 (600×240×200) mm fitted relationship of sample

After fitting as the liner ways stated above, the sample of (240×190×115)mm has a mass humidity bearing rate $y=0.039x-79.654$, the sample of (600×240×200) mm has mass humidity bearing rate which is represented by equation $y=0.0325x-26.468$, in this equation x refers to measured humidity value. Thus, it will be much easier to transfer its mass humidity bearing rate after we measure the material's humidity value by using wall humidity measuring devise.

3.2 Analysis of error

To prove that the liner fitting between measured humidity value and mass humidity bearing rate is applicable and precise, we lead a analysis of errors. Take the curve fitting of sample of size (240×190×115) mm as an example, excepting its quality and humidity value when it is in saturated and moisture absorbed state and in bone dry state, we randomly selecting a point and measure it between these two extreme state, and we put the measured value into the equation to analyze the precision.

For example, we take a sample after it has been dried for 6 hours, the quality m_1 of sample is 7664g when it is saturated,, and the humidity value is 2947; the quality m_2 of sample is 5267g when it is bone dry,, w_2 is 2042. As stated above, when measuring the average value, the mass humidity bearing rate is $(7664-5367) \times 100 / 7664 = 29.97\%$ when the sample is in saturated and moisture absorbed state ; when the average measurement value is 2042, mass humidity bearing rate is 0 when it is in bone dry state. The curve fitting is shown as below, and the equation is, in which x refers to humidity value measured.

The quality and humidity value of sample are respectively 6873 and 2596 when sample is dried for 4 hours, and mass humidity bearing rate, which is calculated by fitting equation, is $y=0.039 \times 2596 - 79.654 = 21.59\%$. The practically measured value is $(6873-5367) \times 100 / 6873 = 21.91\%$.

The absolute error between theoretical value and practical value is $(21.91-21.59) \times 100 / 21.91 = 1.46\%$.

As the ways stated above, we analyzed errors in several different time points, the absolute errors between theoretical value and practical measured value can all be controlled within a slightly range, and thus we can draw that the equation of linear function between measured humidity value and mass humidity bearing rate of block material is precise. It is applicable to use linear fitting to treat the two extreme stage (saturated and bone dry) of building materials.

4. COLD HEAT-BOX FLOW METER METHOD FOR TRACKING MEASUREMENT OF TEMPERATURE, HUMIDITY AND THERMAL FLUX OF WALL

In the process of tracking measurement of a wall through using cold heat-box flow meter, we exclude the complicated exchange coupling interaction of heat and humidity, and purely considering temperature and

humidity of the building's main material. Distributing points by setted thermocouple and humidity measuring sensor, leading a tracking measurement of a shale porous brick and by analyzing the measure date, we expect to find out the law and time of humidity stability and thermal stability of a wall, providing theoretical basis for finding out the optimum time to measure heat transfer coefficient. The measurement points of temperature and humidity of wall are shown as below.

In this experiment, we mainly use shale porous brick as a sample, its main material's equation drew by using fitting ways mentioned above is as below: sample of size (240×190×115) mm, : $y=0.039x-79.654$; we conduct 5 measurements for the envelope by the thermocouple and humidity sensor, both of which have been determined. The time of measurement is as follows: the day of wall approach; 7 days after approach; 15 days after approach; 30 days after approach; 40 days after approach.

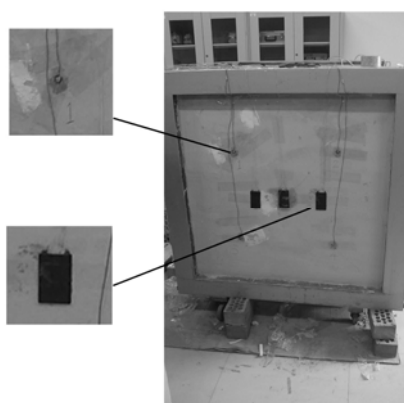


Figure 5 Wall temperature and humidity tracking points of thermal flux

After a systematic analysis of data from tracking measurement, we draw the following conclusion:

(1) For the measurement when building envelopes firstly approach, the average value of data assembled from humidity sensor is 2318, the average temperature is 27°C, and the mass humidity bearing rate is 10.75% through using the equation calibrated by fitting ways setted above, the fluctuation of thermal flux measured by cold heat-box flow meter is large, it means that the humidity is high at this time and it is not a proper time to measure the heat transfer coefficient.

(2) The second measurement is 7 days after approach, average humidity assembled from sensors is 2286, average temperature is 27.5 °C , and mass humidity bearing rate is 9.5% through using the equation calibrated by fitting ways setted above, the range of thermal flux fluctuation is still obvious, which means the humidity is still high at this time, it is not proper to measure heat transfer coefficient.

(3) 15 days after approach, the average value of data from humidity sensor is 2173, average temperature is 28.2°C, and mass humidity bearing rate is 5.21% through using the equation calibrated by fitting ways setted above, the thermal flux is still fluctuating within

a small range, which means the humidity of envelope is still large, it is still not a proper time to measure the heat transfer coefficient.

(4) 30 days after the envelope remains in a natural environment, the average value of data from humidity sensor is 2106, average temperature is 28.6°C, and mass humidity bearing rate is 2.48% through using the equation calibrated by fitting ways setted above, the value of thermal flux measured by cold heat-box flow meter is approaching stable, which means that the wall's internal humidity reaches the theoretical balance, and can be used to measure heat transfer coefficient, after the measurement, we found out the coefficient is 1.41 (W/m² · K), which meet the building energy-saving design standard.

(5) 40 days after the material remains in natural environment, the average value of data from humidity sensor is 2093 average temperature is 28.8°C, and mass humidity bearing rate is 1.97% through using the equation calibrated by fitting ways setted above, at this time, the thermal flux is very stable, and the distribution of value of every point is very even, which means that it is the optimum humidity balance and optimum time to measure the heat transfer coefficient.

5. CONCLUSION

Measuring and analyzing comprehensive humidity ratio and thermal flux of main materials of envelope in different types through using cold heat-box flow meter can help to precisely find out the envelope's thermal and humidity balance and so as to provide theoretical basis for the measurement of envelope's heat transfer coefficient.

For measurement of envelope's heat transfer coefficient, the main steps are as follows: firstly, immersing and drying the sample, working out an linear function equation between measure humidity value and mass humidity bearing rate through using two experimental humidity values in two extreme stages. Then the diagram is drawing according to the fitted linear function equation in order to find out the corresponding measured humidity value and to set points on the envelope to lead a tracking measurement, finding out the time period that is close to humidity practical value. Finally, measuring the envelope's heat transfer coefficient, and drawing parameters like temperature, humidity and thermal flux of envelope's main material at this state.

ACKNOWLEDGEMENTS

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STUDY ON THERMAL INSULATION AND AIR DIFFLUENCE FOR REMOVING HEAT AND COOLING BASED ON NUMERICAL SIMULATION IN EXCAVATION ROADWAY

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ABSTRACT

In the paper, a method of thermal insulation and air diffuence for removing heat and cooling in excavation roadway is proposed. According to the standard $k - \varepsilon$, the physical models of thermal insulation and air diffuence for removing heat and cooling in excavation roadway are established. Law of heat transfer between surrounding rock and main airflow channel in excavation roadway is analyzed, and the temperature field is studied by numerical simulation. The results show that the cooling effect of the method proposed in this paper is quite effective and the optimum air supply velocity is obtained: the optimum air supply velocity is between 7m/s and 8m/s.

Keywords: Excavation roadway; Thermal insulation; Removing heat and cooling; Temperature field; Numerical simulation.

1. INTRODUCTION

As the mining of mines in China are gradually deeper, the problem of high temperature and high humidity becomes more and more serious, which has become one of the major problems restricting mine safety and high efficiency mining [1, 2]. The environment of high temperature and high humidity in mine will cause miners to produce heat stroke, thermal blooming and other diseases, seriously endangering the health of workers. At the same time, the environment of high temperature and high humidity will greatly lower labor productivity, thereby affecting the economic benefits of the mine. Heat harm and high humidity has become a disaster of the mine, which need to be solved urgently.

With the exception that high temperature water gushing and strong oxidation of a few mines ore is main factors[3,4] forming mine heat harm, the main factors to cause heat harm roadway in deep mine are surrounding rock the exothermic, air since compression exothermal, exothermic when the operation of the electrical and mechanical equipment, transport of ore and rock exothermic. In the excavation roadway, the high-temperature heat sources are mainly the exothermic original rock and tunneling equipment [5]. Many scholars have made the research work on airflow temperature and surrounding rock exothermic in excavation roadway [6-9], and proposed temperature prediction and numerical simulation method for roadway airflow and temperature field of surrounding rock, which is of great significance for the prediction of

laneway heat injury. For the existing of rock temperature, some scholars proposed cooling methods of isolating heat source. The domestic measures are mainly to use thermal insulation materials to spray roadway surface or use line insulation. There are a lot of researches on thermal insulation materials, which have made great progress, but the cooling effect is not ideal [10-12].

Thermal insulation and air influence for removing heat and cooling method in excavation roadway is presented in this paper. The method of numerical simulation is used to research the cooling effect.

2. METHOD OF THERMAL INSULATION AND AIR DIFFLUENCE FOR REMOVING HEAT AND COOLING

According to the characteristics of high temperature of rock in deep mine, the insulation shunt exhaust heat cooling method of roadway is proposed, as shown in figure 1. The insulation board, imitating the shape of roadway surface and airtight, is set at a distance of certain size B from roadway surface, along the direction of roadway excavation; The roadway airflow is divided by insulation board into main tube airflow in the middle of roadway, and side tube airflow between insulation board and roadway surrounding rock; The fan drum connected with the local fan is arranged on the main road; The airflow sent by local fan is divided into two airflows after cleaning the working face. Part of the airflow discharged the harmful gas dust and heat and humidity in tunneling working surface and roadway surrounding rock is discharged through the

side channel. The other part of airflow along the main channel returned the return air lane connected with roadway. The convective heat and humidity exchange between ambient air in roadway surrounding rock and air in the middle of roadway is cut off by insulation board. Heat insulation board is made of insulating materials with thermal conductivity less than 0.05W/(m·K). A guiding plate is connected to insulation board close to head-on area in the excavating face.

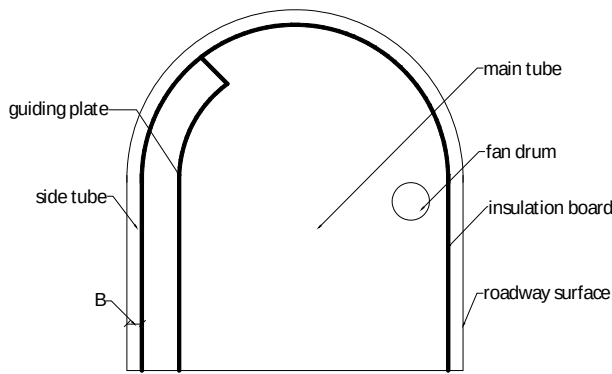


Figure 1 Schematic diagram of insulation diversion in excavation roadway

3. NUMERICAL SIMULATION OF THERMAL INSULATION AND AIR DIFFLUENCE FOR REMOVING HEAT AND COOLING METHOD

3.1 Physical model

According to the above description, a simplified model of semi-circular arched roadway is shown in figure 1. The wall rock of semi-circular arched tunnel is homogeneous, the width of roadway is 4.5m, the height is 4.8m (wall height is 2.6m), and the length is 200m. The thickness of heat insulation board is 0.03m, the distance B between heat insulation board and walls of the tunnel is 0.2m, and the distance between heat insulation board and the tunneling head is 5.0m. The local ventilation of the excavating roadway is press-ventilated. The canvas ventilator is used, and the diameter of air duct is 0.5m. The air duct is installed along the horizontal direction close to the heat insulation board, at a height of 2m from the roadway floor, and the distance between the duct outlet and the working face is about 10m. Software fluent is used to carry on the numerical simulation, the roadway entrance is set as the origin.

3.2 Governing equations

When setting up the governing equations, the following assumptions are made:

- (1) The airflow in the tunnel is incompressible; the heat consumption caused by fluid viscous force working is ignored.
- (2) The turbulent viscous of fluid is isotropic.
- (3) The flow is steady turbulent and meets the Boussinesq hypothesis.

(4) Only the rising temperature of air inside caused by heat in surrounding rock roadway is considered, ignoring the influence of excavating equipment and other heat sources.

The roadway air flow and heat transfer turbulent model adopts the standard $k-\varepsilon$ model, the governing equations includes continuity equation, momentum equation, energy equation, k equation, ε equations, the equations can be expressed by the general form as follows [13,14]:

$$\frac{\partial(U_j \Phi)}{\partial X_j} - \frac{\partial}{\partial X_j} \left[\Gamma_\Phi \frac{\partial \Phi}{\partial X_j} \right] = S_\Phi \quad (5)$$

Where, Φ is universal variable; Γ_Φ is diffusion coefficient; S_Φ is the source term for general equations; U_j is velocity vector, m/s.

3.3 Boundary conditions

While the software Fluent is used to simulate the airflow temperature in tunnel, two semi-circular arched roadway models are established with the software Gambit. One is the original roadway, denoted as roadway1; the other is the roadway setting insulation board, denoted as roadway2. The airflow section of the air duct outlet is set as the inlet boundary of model; the roadway free section is set as the outlet boundary.

Inlet: the temperature T is 302K, the velocity v is between 2m/s and 12m/s, the turbulent kinetic energy $k = 0.05v^2$, the dissipation rate $\varepsilon = C^{\frac{3}{4}} k^{\frac{3}{2}} / (0.07D)$, the constant C is 0.09[5], D is the duct diameter.

Outlet: outflow.

The wall of surrounding rock: without slip, the wall roughness is default, the initial temperature of surrounding rock is set 318K, the heat generation rate is 150W/m³; the initial value of roadway floor is set 305K, the heat generation rate is 120W/m³.

The wall of heat insulation board: the interface of heat insulation board and the main, side channel is the solid-fluid coupled heat transfer boundary, selecting the coupling conditions, the thickness of heat insulation plate is 0.03m, and the heat production rate is zero.

The air duct wall: the interface of air duct and the inlet air, the main channel return air flow is the solid-fluid coupled heat transfer boundary, the selected coupling conditions, the thickness of air duct wall and the heat production rate is zero.

4. RESULTS AND ANALYSIS OF AIRFLOW TEMPERATURE IN EXCAVATING ROADWAY

4.1 The cooling effect of heat insulation shunt method

Through the numerical simulation on the model of roadway1 and roadway2, the change of airflow temperature in the air duct, the main, side tube is

calculated. After extracting the temperature data, the temperature curves in the process of airflow in the duct and the main tube are shown in figure 2 and figure 3.

(1) The comparison of figure 2 and figure 3 shows that, the air outlet temperature of roadway2 air duct is lower about 3°C than roadway1; the return airflow temperature of roadway2 main channel is lower 4°C than roadway1, as a result of roadway2 setting insulating board. It is visible that setting heat insulation board increases the heat resistance between the surrounding rock and the main channel flow, reduces the heat flow from rock to main channel, and the cooling effect of heat insulation is obvious.

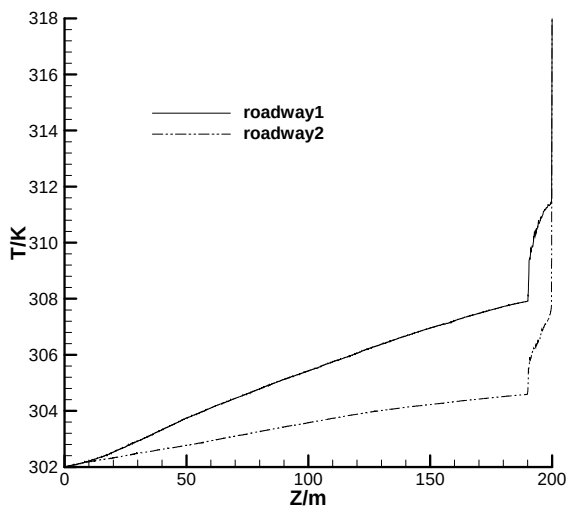


Figure 2 Comparison of airflow temperature in hairdryer in excavation roadway

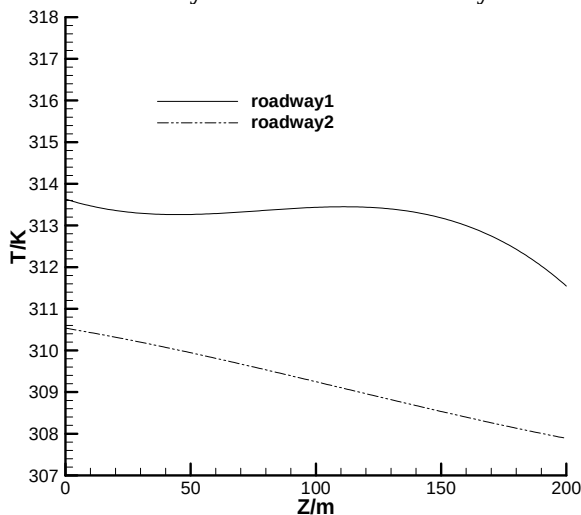


Figure 3 Comparison of airflow temperature in main tube in excavation roadway

(2) In figure 2, the airflow temperature of roadway1 and roadway2 main channel is rising sharply, which is caused by the close direct contact between the regional air and the rock newly exposed, and the occurrence of strong heat exchange. Therefore, the staff

should reduce working time at the tunneling working surface as much as possible.

4.2 Distribution of airflow temperature in tunnel

Through the numerical simulation, the airflow temperature distribution of roadway1 and roadway2 in tunnel is shown in figure 4 and figure 5. When the other conditions remain unchanged, changing the of supply air velocity based on the roadway2 model, taking the section of 25m, the variation of return airflow temperature with the inlet air velocity is shown in figure 6.

(1) Figure 4 and figure 5 show that, there is strong heat exchange between flow and wall near the tunnel wall, and the air temperature increases continuously along the direction of air flow, also the air temperature of the air duct increases gradually along the flow direction. In the same section of tunnel airflow, the intermediate temperature is the minimum. The temperature near the tunnel wall is high, close to the temperature of surrounding rock. In the head area of excavating roadway, there is strong heat exchange between the air and the roadway surrounding rock, and air temperature increases for larger gradient.

(2) In figure 5, because of the insulation effect of heat insulation board, the return airflow temperature of side channel is higher than that of main channel, and the increase gradient of airflow temperature of side channel is higher than that of main channel. In the main channel, the increase gradient of airflow temperature close to the tunneling working surface and not in the protection area of insulation board is higher than that of the inner insulation board airflow temperature.

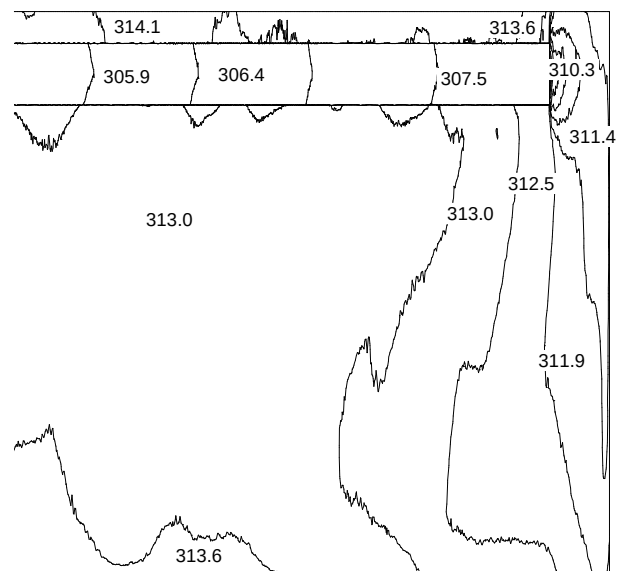


Figure 4 The airflow temperature in roadway1

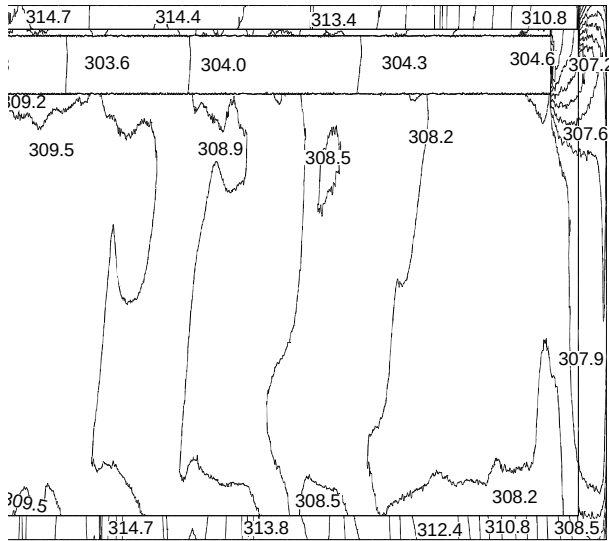


Figure 5 The airflow temperature in roadway2

(3) Figure 6 shows that, with the increase of air velocity, the return airflow temperature of the main channel gradually decreases, but when the air velocity is between 7m/s and 8m/s, the temperature is oscillating; when the air velocity is bigger than 8m/s, the air temperature increases in return. So in the roadway, the appropriate velocity of inlet air is between 7m/s and 8m/s.

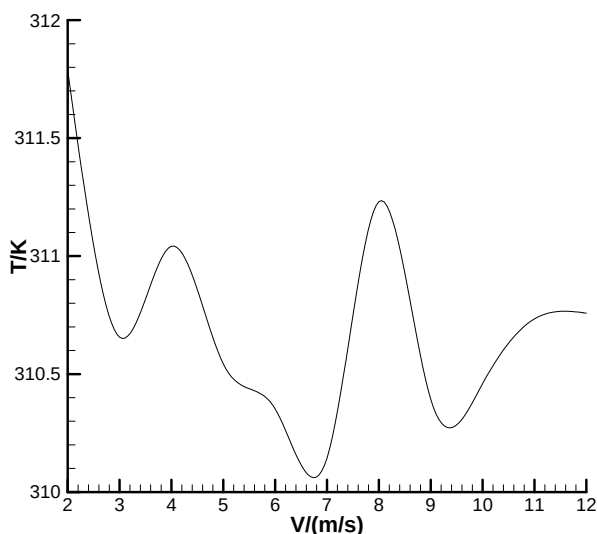


Figure 6 Variation of return airflow temperature with the inlet air velocity

CONCLUSIONS

(1) A cooling method of tunnel insulation shunt exhaust heat is proposed, numerical simulation show that the average temperature of airflow in main runner 37°C, the average temperature is reduced by 4 °C, compared to tunnel that not set insulation board. It verifies obvious cooling effect of the cooling method of tunnel insulation shunt exhaust heat.

(2) After the insulation shunt exhaust heat cooling method is applied in excavation roadway, on the same

section of tunnel airflow, the intermediate temperature is the minimum; the temperature near the tunnel wall is high. The return airflow temperature of side channel is higher than that of main channel, and the increase gradient of airflow temperature of side channel is higher than that of main channel. In the head area of excavating roadway, air temperature increases for larger gradient for the heat insulation board could not be provided timely.

(3) The return airflow temperature of excavation roadway decrease with increasing air velocity, so appropriately increasing the air velocity can effectively reduce the temperature of the laneway; the optimum air supply velocity is between 7m/s and 8m/s.

ACKNOWLEDGEMENTS

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NUMERICAL SIMULATION OF AIRFLOW DISTRIBUTION IN A HIGH-PRESSURE CENTRIFUGAL BLOWER

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ABSTRACT

Based on steady compressible Reynolds-averaged N-S equations, standard $k-\varepsilon$ turbulent model, computational fluid dynamics software Fluent is applied, and the airflow distribution in a double suction high-pressure centrifugal blower is numerically simulated. The internal pressure, velocity, temperature field and the internal vortex has been obtained. The internal flow characteristics of the centrifugal blower is analyzed, and the simulation results provide the theoretical foundation for the modification of the blower.

Keywords: Centrifugal blower, Numerical simulation, Airflow distribution, Compressible fluid

1. INTRODUCTION

At present, most of the centrifugal blowers in our country were designed in 1970s and 1980s, and the efficiency of blower needs to be improved. Impeller is the heart of blower, and the airflow distribution greatly affects the efficiency of blower [1-3]. So the optimization of impeller is an important research direction to improve the efficiency of centrifugal blower. Numerical simulation is used as a high efficient method in blower. The development process of numerical simulation includes two-dimensional to three-dimensional flow, inviscid to rotating flow. Nowadays, full three-dimensional viscous flow can also be calculated [4-6]. In this paper, the compressible fluid model in a high-pressure centrifugal blower is numerically simulated by software Fluent, and the airflow field is obtained in standard working condition. The results show us the internal flow law of this blower, and it provides theoretical basis for further reformation of this centrifugal blower.

2. MODEL

2.1 Physical model

This is a double suction centrifugal blower with a high pressure. The impeller's shape is arc, with 18 backswept blades. The impeller inlet diameter is 610 mm, and outlet diameter is 1120 mm with an outlet width of 42 mm. The impeller is designed to operate at 4750 rpm. The simplified physical model of centrifugal blower is shown in Figure 1.

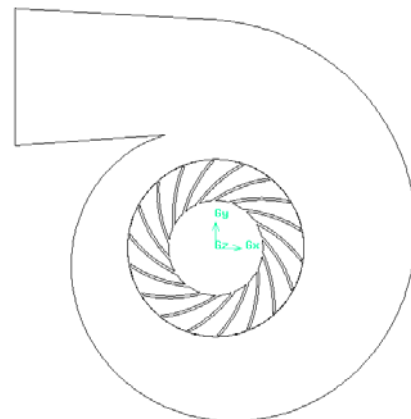


Figure 1 Physical model of the centrifugal blower

2.2 Governing Equations

Air flowing in the centrifugal blower is complex turbulent flow. The steady Reynolds averaged N-S equations and standard $k-\varepsilon$ turbulence model is applied. The numerical simulation is carried out with a multiple frame of reference approach because the impeller flow field is within a rotating frame [7-10]. The governing equations for compressible mean-flow in the centrifugal blower can be written as following in Cartesian coordinates.

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}(\mu \frac{\partial u_i}{\partial x_j} - \rho \overline{u_i u_j}) \quad (2)$$

$$\frac{\partial(\rho u_j T)}{\partial x_j} = \frac{\partial}{\partial x_j}(\frac{K}{C_p} \frac{\partial T}{\partial x_j}) + S_T \quad (3)$$

Where, ρ is fluid density, kg/m^3 ; u_i is fluid velocity components, m/s ; p is fluid pressure, Pa ; x_i is space coordinates, m ; μ is dynamic viscosity, $\text{Pa}\cdot\text{s}$; T is fluid temperature, K ; S_T is energy equation resource item; $-\overline{u_i u_j}$ is Reynolds stress; K is fluid heat transfer coefficient, $\text{W}/(\text{m}^2\cdot\text{K})$; C_p is air specific heat at constant pressure, $\text{kJ}/(\text{kg}\cdot\text{K})$.

Additional, the standard $k-\varepsilon$ equations is used. The equations are as follows:

$$\frac{\partial}{\partial x_j}(\rho k u_i) = \frac{\partial}{\partial x_i}[(\mu + \frac{\mu_t}{\sigma_k}) \frac{\partial k}{\partial x_j}] + G_k - \rho \varepsilon \quad (4)$$

$$\frac{\partial}{\partial x_j}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_i}[(\mu + \frac{\mu_t}{\sigma_\varepsilon}) \frac{\partial \varepsilon}{\partial x_j}] + C_{1\varepsilon} \frac{\varepsilon}{k} G_k - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad (5)$$

Where, μ_t is eddy viscosity coefficient

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (6)$$

G_k is turbulent pulsating kinetic energy generated item

$$G_k = -\rho \overline{u_i u_j} \frac{\partial u_j}{\partial x_i} \quad (7)$$

The universal constant values are as follows: $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $C_\mu = 0.09$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$.

3. NUMERICAL METHODS

3.1 Meshing

In this paper, the meshing tools ICEM-CFD was applied, using the unstructured grid. The grid size of some region is a higher density in volute tongue to meet the precision. The total number of grid is 277164. The total number of faces is 9545; the total number of nodes is 280771. Meshing is shown in Figure 2.

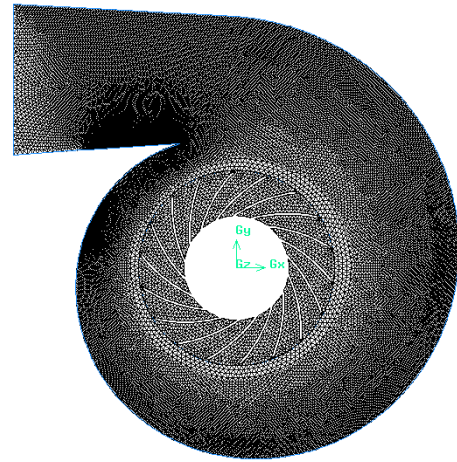
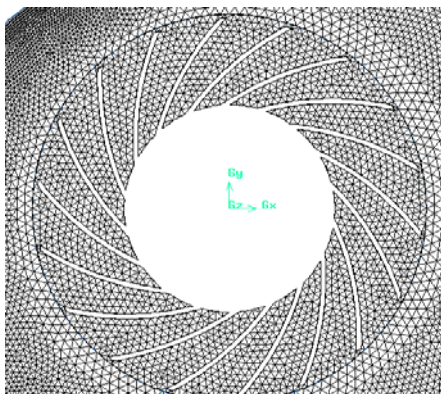


Figure 2 meshes for the centrifugal blower

3.2 Boundary Conditions

(1) Inlet: pressure inlet, total pressure is 100.7kPa; the total temperature is 315K.

(2) Outlet: pressure outlet, the static pressure is 49.7kPa, the temperature is set to default.

(3) The region of impeller: the rotating flow rate is 4750 rpm; the blades are arranged as a rotating wall.

(4) The solid walls: no-slip boundary condition is applied in walls; the near wall regions use the standard wall function method.

(5) The differential method: pressure uses standard format, and velocity, turbulent kinetic energy and turbulent viscosity coefficient uses the second order upwind format.

4. RESULTS AND ANALYSIS

After the post-processing of simulation results, some diagrams of impeller and volute are obtained. The absolute pressure, dynamic pressure, static pressure, velocity, velocity vector and temperature distribution of the blower are shown from Figure 3 to Figure 8.

Figure 3 shows that the absolute pressure in the blower mainly increases in the impeller in which the gas forced rotation by leaf blades. Absolute pressure in the impeller is approximate symmetric distribution, and it increases from inlet, but it descends at the exit due to vortex's dissipation. Figure 4 shows, near the volute tongue's wall, the dynamic pressure is very low. From the inlet to the outlet of the impeller, dynamic pressure increases. While for the volute part, the more close to the outside of the volute the lower dynamic pressure is, because of the dynamic pressure which is gradually converted to static pressure. Part of the high dynamic pressure appears at the outlet due to part of the fluid flow discharging directly and the high static pressure of the volute which is converted to dynamic pressure at the outlet, and the simulated outlet static pressure 48500 Pa is near the actual static pressure 49000 Pa. As figure 5 shows, the static pressure in impeller's inlet is the lowest, and it increases gradually along streamwise direction in the impeller. The static pressure in impeller is approximately symmetrical. In the suction side of the

impeller, the pressure is low, and the static pressure at the external volute is high, the internal is low.

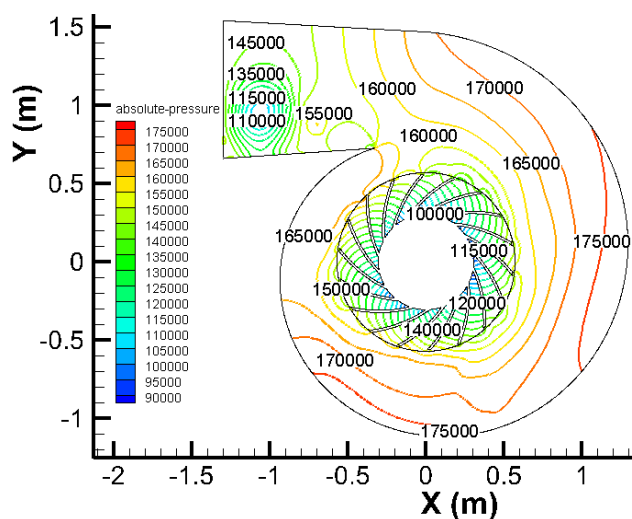


Figure 3 Absolute pressure of the blower

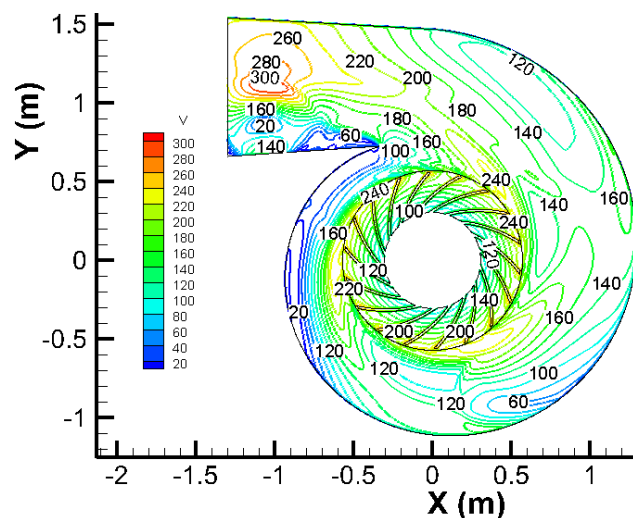


Figure 6 Velocity distribution of the blower

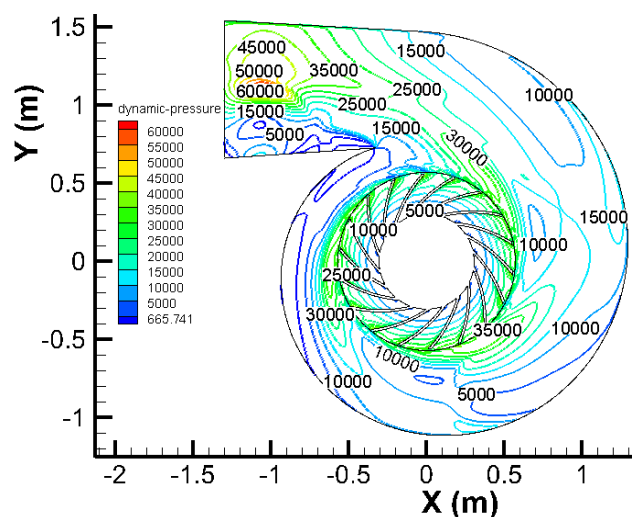


Figure 4 Dynamic pressure of the blower

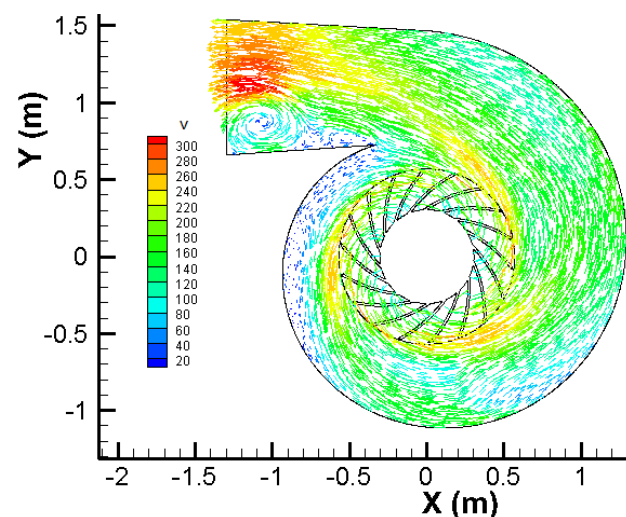


Figure 7 Velocity vector of the blower

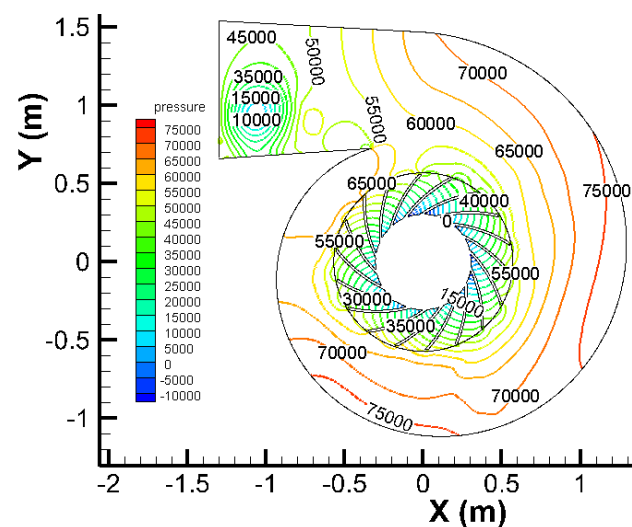


Figure 5 Static pressure of the blower

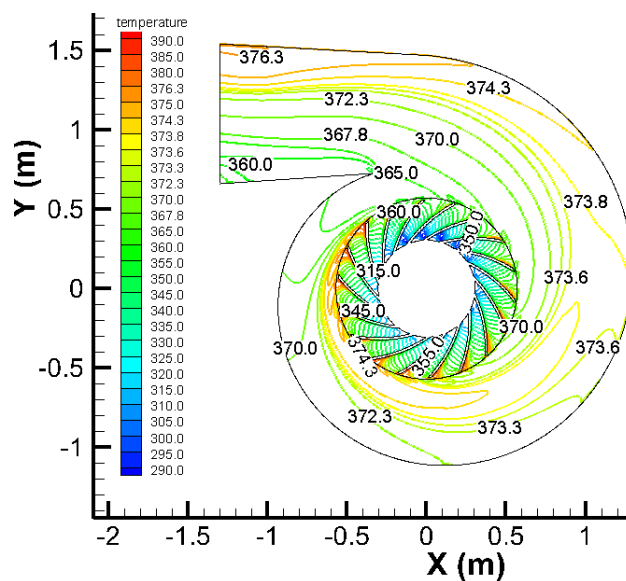


Figure 8 Temperature distribution of the blower

As figure 6 shows, because of the asymmetric volute, the internal airflow field is also not symmetrical. But flow in the volute is relatively smooth and the flow loss is smaller. A portion of the fluid flows out directly to the export from the impeller, which creates a relatively high speed fluid field. While the speed of volute tongue and near volute wall is relatively low, the export rate of about 180m/s is close to the actual outlet speed. Figure 7 shows that the velocity distribution is not uniform. In the volute tongue region, the speed is very low with a backflow. On the contrary, the speed of the side far from volute tongue is high, and a vortex appears.

Figure 8 shows the temperature increasing with the compression and polytropic process in the impeller region. The air is forced to rotate with a high speed by the impeller, and then compressed. The temperature field distribution is almost symmetry, and minimum value of temperature about 315K appears on the inlet region. The temperature outside area of volute is highest; the outlet temperature near 360K is very close to the actual outlet temperature.

5. CONCLUSIONS

Through the numerical simulation of the high-pressure centrifugal blower, the internal pressure, velocity, temperature distribution has been obtained. The results show that the simulation flow field is close to real flow field under the same parameters. Based on the above research, we will get a better flow field by changing the shape of the impeller and volute tongue. In this way, a higher efficiency blower and energy saving can be achieved. The results provide a theoretical basis for future improvement and optimization.

6. ACKNOWLEDGEMENTS

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RESEARCH PROCESS ON THERMAL MODELING OF HOLLOW BLOCK VENTILATED WALL

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ABSTRACT

The energy demand in China is rapidly growing due to increasing population, urbanization and the improvement of living standards. The development and applications of low energy technologies for green buildings have been paid much concern. The hollow block ventilated wall is proposed in this paper, this, in turn, allows the use of heat recovery and low energy cooling or heating sources, such as the ground, outside air or recovered process heat. The principle of the hollow block ventilated wall is described; the calculation models of many types of ventilated walls as well as the performance evaluation applied in building systems for air conditioning are also critically reviewed. Finally, this review shows that the available dynamic thermal network model of the hollow block ventilated wall for easy practical applications with good accuracy need to be investigated to evaluate the dynamic thermal performance and to make this structure to be used in low energy architecture for improving energy efficiency and occupant thermal comfort.

Keywords: Ventilated wall; Low energy sources; Calculation models; Performance evaluation.

1. INTRODUCTION

The energy demand in China is rapidly growing due to the increasing population, urbanization and the improvement of living standards. China's buildings sector currently accounts for 27.6% of total energy use and is projected to increase to 35% by 2020 [1, 2]. Building energy conservation has become an important issue of building sustainable development and will continue growing in importance in future [3]. It was estimated that energy consumed by the heating, cooling and air conditioning account for 65% of total building energy consumption [4]. Building cooling and heating requirements are mainly caused by the building envelope and air infiltration through windows and doors, and the wall occupied a large proportion, the insulation performance of wall directly affects the level of the building's energy consumption [5]. Therefore, the development of new energy-saving wall, and the study of heat transfer characteristics and energy-saving potential of this structure have important significance.

Currently, many types of new wall structures have been developed and produced in the market, and the hollow concrete block accounted for a large proportion. The thermal performance of hollow block has been improved to some extent compared with traditional solid clay brick, however, the heat transfer coefficient is still not up to building energy efficiency standards requirements for ordinary single and double row hollow block wall [6]. According to the characteristic of the hollow part from up to down, a ventilated wall is proposed which use fans for heat/coolth transportation,

and use the thermal storage of the block and large contact area between the air flows and interior surface of the cavity to achieve the heat transfer and remove.

2. PRINCIPLE OF CONCRETE HOLLOW VENTILATED WALL FOR AIR CONDITIONING

Fig.1 is the schematic view of the hollow block ventilated wall. The supply air passes through the hollow part of the block, and then discharges to the outside after the heat exchange with the concrete hollow block, the cooling and heating requirements of envelope caused by outdoor environment can be partially removed; the building cooling/heating load can be greatly reduced or even eliminated, the inner surface temperature of the enclosure structure is much closer to room temperature, the radiation heat transfer between the human body and the inner surface is reduced, and thus, the thermal comfort of the human body is improved. The air flow forth through the wall cavity, the heat/coolth removed depends on the heat transfer area between the passed air and the interior surface of the cavity and increasing the contact time allowing for substantial heat flows between the passed air and the structure even for relatively small temperature differences. This, in turn, allows the use of low cooling or heating energy sources, such as natural ground source and ambient air source.

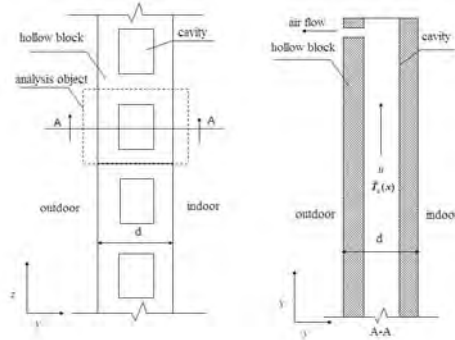


Figure 1 the schematic view of the hollow block ventilated wall

In summer, the supplied air for ventilation wall may use the exhaust of the air-conditioning system, ground tunnel air and the night air and so on; in winter, the exhaust of the air-conditioning system can be used as the supplied air for ventilated wall. The exhaust of the air-conditioning system is about 10% of the total amount of supply air, there may cause a great waste of energy if the exhaust of the air-conditioning system is discharged directly to the outside. The air through ground tunnel can be cooled by the heat exchange between the air and the surface of ground tunnel or duct, and can be supplied as the cooling air [7] and fresh air [8]; summer night ventilation also has a large energy saving potential [9]; so the exhaust of the air-conditioning system, the air through ground tunnel and the night air can be used for cooling and heating the hollow block ventilated wall. The ventilation of hollow block wall combines the heat recovery of air conditioning system or the use of renewable energy with the reduction of building energy consumption, the effect of outdoor environment on indoor can be significantly weakened by full use of waste heat and low-grade renewable energy, and it is of great significance for building energy efficiency.

3.CALCULATION MODELS AND THE THERMAL PERFORMANCE OF VENTILATED WALL

A ventilated wall is usually a double skin envelope but it differs in its construction and operational mode. Ventilated wall that appeared in the 1980s is generally used to prevent heat transfer from the indoor to the outdoor in winter in cold areas. The outdoor air is introduced to the ventilation gap that has been pre-formed in the building wall, and then heated by the solar or heater so as to exchange the heat with the wall and cut off the heat loss of the wall. For instance, we can make the high temperature gas combusted in the furnace flow through the wall/floor with ventilation gaps; or make use of the solar collector system to heat the air in the gap to take away the cold of the wall/floor; or make the air in the ventilation gap be heated and then flow into indoor through the porous permeable material layer to supply indoor fresh air. At the present time, the

air gap ventilated wall has been widely applied abroad especially in the Nordic countries [10-12].

The domestic researches on the thermal characteristics and energy efficiency of the ventilated wall are mostly in the beginning stage, and it mostly reflected in two aspects of the experimental research and the heat transfer model of the ventilated wall. Feng [13] built two experimental rooms and a gap ventilated wall, tested the inlet and outlet temperatures of the exhaust air of air conditioning system through the air gap ventilated wall in winter in Shanghai, and took a contrastive measure about the electric consumption of a simulation room with the air gap ventilated wall and another simulation room with the ordinary wall. The results showed that the ventilated wall had remarkable energy saving effect. Jian [14] measured the average heating and cooling load of the composite ventilated wall in the coldest month and hottest month, and pointed out that it could greatly reduce the load caused by wall heat conduction by using the solar energy to heat the air in the wall gap in winter and utilizing the pipe buried in the ground to make the cold air flow through the gap to reduce the temperature of the wall. Tan[15] considered the thermal resistance of wall and established the simplified one-dimensional steady-state heat transfer model of the solar panels composite wall. The thermal resistance of this structure were calculated when the air in the solar composite panel gap was in the states of static, the natural convection and forced convection. Zou [16] studied the thermal performance of dynamic insulation which is to draw the exhaust of air conditioning to outside through air permeable insulation, the one-dimensional steady-state heat transfer mathematical model of the ventilated wall was established and the steady-state heat transfer characteristics under the indoor and outdoor disturbance were calculated by numerical methods, and the effects of indoor air-permeability speed and the thickness of the penetrate material on heat transfer were analyzed. Then the one-dimensional dynamic mathematical model of the model was investigated on the basis of previous study [17]. Chen [18] performed a numerical simulation study on indoor ventilation effect when the fresh air was introduced into the room whose envelope structure is hollow block ventilated wall during the transition season by establishing a three-dimensional steady-state model. Zhao [19] established a two-dimensional steady-state model and studied the thermal performance of ventilated wall in time domain by numerical simulation.

The concerns of foreign researches on ventilated wall originated in the early eighty's. The theoretical study on heat transfer model and thermal characteristics of the ventilated wall began in the 1990s. Taylor et al [20] established a one-dimensional steady-state model of the three-layer dynamic insulation wall (two gaps) and gave the one-dimensional steady-state equation which including the wall temperature, heat flux, vapor concentration and convective vapor flow. Then, this researcher adopted this model to study the influence of the airflow rate on the wall surface temperature, and compared and analyzed the model result with hot box experiment data [21]. Akander [22] proposed a new

model by employing the resistance-capacity network method, one-dimensional heat transfer model of the multilayer components wall/roof was established. Based on it, Ren and Wright [23] used this method to investigate the simplified dynamic thermal network model of hollow precast slab ventilated roof. In the model, each floor component connected a thermal resistance and a heat capacity (namely, the first-order model), the heat capacity and thermal resistance were determined manually based on the geometry description of the roof. The results were compared with the test data, the maximum error of the ventilation outlet temperature is 1.1°C . It pointed out that the model could also be directly embedded in the built environment and energy simulation platform. Pottler [24] established a two-dimensional steady-state heat transfer model and utilized the finite difference method to study the heat transfer process of the air layer between the glass and the wall in time domain which was heated by the solar collector, and then simulated the outlet air temperature, the glass surface temperature, and the temperature in different parts of the absorbing device, and the simulation and experimental results are basically consistent.

Over the past decade, the theoretical researches on the heat transfer model and thermal characteristics of ventilated wall are still mostly about the steady-state. Schmidt [25] established a two-dimensional simplified steady-state model of the double-layer ventilated wall and studied the change of temperature along the airflow in the cavity. The cavity was suggested as close as possible to the inner surface in order to reduce heat loss. But this method is only suitable for solving the steady-state heat transfer and linear heat transfer coefficient. Balocco[26] established a one-dimensional steady-state model of the ventilated wall and utilized the finite element method to calculate each step of the wall surface and heat flow temperature, and the thermal characteristics of different cavity thickness were studied in summer and winter when the outlet opened and closed, respectively. Ciampi [27] took the wall thermal resistance into consideration and established the simplified two-dimensional steady-state heat transfer model of the ventilated wall under the chimney effect. the heat took away by the airflow in the ventilated wall in summer was calculated, and the influences of surface roughness, thermal resistance, solar radiation intensity, cavity layer thickness as well as different indoor and outdoor temperatures on the energy-saving performance were analyzed. Sanjuan[28] analyzed the thermal characteristics of a typical multi-opening laminated ventilated facades in Madrid of Spain, the three-dimensional steady-state and quasi-steady-state models were established and the numerical simulation method was used for solving the models. Then, they studied the temperature distribution, heat transfer and energy consumption in the cavity in the closed and multi-opening interlayer ventilated wall in summer and winter. Mavromatidis [29] established a one-dimensional dynamic model of the heat transfer process for the multi-layer insulated wall which was made up of the air gaps. The finite element method was adopted to

study the heat transfer process of convection and radiation, the internal temperature distribution characteristics and heat transfer characteristics of the multi-layer insulated member.

Some other researchers studied the heat transfer performances of the ventilated wall through the experimental tests. The researches in the 1990's focused on the test under the steady-state parameters of environmental chamber, Roots [30] used the pressure tanks to measured the dynamic U-values and air tightness at different air flow rates through the ventilated wall. Crowther [31] performed an experimental study of the hot-box on permeable insulation ventilated wall and analyzed the effect of the airflow velocity on the wall surface temperature. After 2000, the experimental study on the thermal characteristics of the ventilated wall under the outdoor natural climate conditions became a hot research. Clare and Etheridge [32] measured the heat transfer characteristics of the infiltrated ventilation wall by establishing the geometrically similar model. In 2003, Baker [33] tested the effect of the different penetration air speed and solar radiation heat gain on the dynamic U-value, the inner surface temperature and the heat flow of the monolayer permeable insulation ventilated wall in the actual outdoor weather conditions and comfortable indoor conditions, and then used the steady-state algorithm to calculate the recovered heat of the air that flowed into the indoor through the permeable ventilated wall from the outdoor. Dimoudi[34] tested the inner surface temperature and the heat conduction of the permeable ventilated wall with a 5cm air gap under different indoor and outdoor differential pressure and with or without a heat source in the cavity, and then carried out the wall temperature distribution curve along the thickness direction and the temperature distribution in the cavity along the thickness and the height direction in a certain time. In 2001, Stazi [35] designed the air gap ventilated wall whose out surface was clay and applied it to an office building in Ancona, Italy. Then, the airflow temperature, flow rate, heat flux in the cavity and the internal and external wall surface temperature of the air gap ventilated wall at different heights and orientations were tested, and the evaluation of the physical thermal characteristics of the wall was carried out.

In summary, there are many researches directed towards the thermal performance of the ventilated wall, which mainly included heat transfer model and experiment study. Modeling study mainly adopts the following four heat transfer models, the first ones are the steady heat transfer models, including one-dimension, two-dimension, and three-dimension, the numerical method such as the finite difference was applied to solve the model; the second ones are the simplified steady-state thermal network models which only consider the thermal resistance; the third ones are dynamic heat transfer models using numerical method, and the fourth are simplified dynamic thermal network models which not only consider the thermal resistance but also the heat capacity. The majority of the current studies are the steady-state models; however, the

instantaneous heat transfer characteristics of wall are essential important to cooling/heating loads and indoor thermal comfort. In general, finite element method or detailed finite difference method are computationally too extensive to allow the modeling of the major thermal disturbances on a building. Simplified dynamic thermal network model can integrate them in existing whole building simulation packages for energy performance evaluation and project design, while, the parameters of these thermal network models were determined manually based on the geometry description of the component which may deteriorate the modeling accuracy. So the available dynamic thermal network model of the ventilated wall for easy practical applications with good accuracy needs to be investigated.

4. CONCLUSION

The hollow block ventilated wall is proposed according to the characteristic of the hollow part from up to down in this paper; the principle of the hollow block ventilated wall is described, the cooling and heating requirements of envelope caused by outdoor environment can be reduced or even eliminated by the supplied air which can use the heat recovery and low energy cooling or heating sources. The calculation models of ventilated walls as well as the performance evaluation applied in building systems for air conditioning are also critically reviewed.

The modeling of ventilated wall include one-dimension, two-dimension, or three-dimension using simplified RC model, finite difference model or finite element model with different assumptions. The studies on the heat performance of concrete hollow ventilated wall have not been found yet, the heat transfer model of this structure are very complex which is affected by the outdoor temperature, indoor temperature and the air temperature and velocity inside the cavity. The available dynamic thermal network model of the ventilated wall for easy practical applications with good accuracy needs to be investigated to evaluate the dynamic thermal performance, which can calculate the heat transfer in the wall mass, the heat transfer between the ventilated air and the interior surface of cavity, the air temperature gradient along the cavity simultaneously, especially can integrate them in existing whole building simulation packages for energy performance evaluation and project design.

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CALCULATION METHOD OF HEAT-CARRYING CAPACITY OF STATIC WATER IN OPEN-LOOP SWHP SYSTEM

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ABSTRACT

Chongqing, China, which is in the hot summer and cold winter zone was taken as an example, the mathematical relationship between the water heat carrying capacity and the water characteristic parameters was achieved. Firstly, based on the N-S equations, by respectively adding the quality and energy source terms into the water temperature equations and the control equations, the two-dimensional mathematical model of the water temperature was established in the conditions of inputting different load, and the numerical solution of the water temperature was achieved by the coupled solution. Secondly, according to the limit of the inlet water temperature in summer and winter, the maximum cooling or heating load was obtained, which can be carried by the water in the conditions of different load characteristics, water depth and initial water temperature during the building cooling or heating period. Finally, by the regression method, the regression equations of the water heat carrying capacity was obtained. Through the practical engineering validation, the reliability of this calculation method can be proved.

Keywords: Open-loop lake water source heat pump system; Static water; 2-D water temperature model; Water temperature limit; Heat carrying capacity.

1. INTRODUCTION

As the global energy crisis and the trend of energy saving, the surface water heat pump system (SWHP) with large energy saving potential has attracted more and more attention in HVAC industry. All the rivers, lakes and seas can be available for the SWHP system. Most of them with large heat-carrying capacity can timely take away the heat released by heat pump, while the relatively static water with small heat-carrying capacity, such as reservoir and lake can not make it. If the indoor load is beyond the water heat-carrying capacity, some problems will appear. Therefore, it's necessary to research the relationship between the indoor load and water heat-carrying capacity.

The SWHP system developed earlier in Europe. In the 1970s, European and American countries began to promote the SWHP system (Enstrom, Karstrom et al. 1977), their main application object is closed-loop SWHP system (Rackliffe and Schabel 1986; Chiasson, Spitler et al. 2000; Stephen and Kevin 2002; Kevin and Rafferty 2004; Hattemer and Kavanaugh 2005). In the aspect of research on the water heat-carrying capacity, M.c.Pezent (Pezent 1990), Cartrell and Wepfer (Cartrell 1984), Kavanaugh (Kavanaugh and Rafferty 1997) have done some research.

Recently China also began to massively apply the open-loop lake SWHP system (Li and Li 2003; Chen and Zhang 2006; Feng and Xiong 2006; Chen and Fu 2008), but researched less on the water heat-carrying

capacity. The water heat-carrying capacity is subject to the practical projects. The detailed calculation method of the water heat-carrying capacity has not yet reported. Therefore, taking the reservoir as the research object, combining the building load and heat pump system, the calculation method of the water heat-carrying capacity is researched in this paper.

2. MATERIALS AND METHOD

2.1 Materials

The project described in this paper is the Kai County people's hospital in hot summer and cold winter zone in Chongqing, China. It has 23 floors with 54411m² in covered area, 24250m² in air-conditioning area and 2494kW/995kW in the maximum cooling/heating load. Using SWHP system, Ankang reservoir in the east of the building is taken as the low level cooling/heating source of the system. Ankang reservoir is about 300m away from the machine room with 160000-220000m³ in water capacity, about 34000m² in water surface area and 5-7.5m in water depth throughout the year. The horizontal distance between water intake and water drain is about 210m. The water drain is by means of surface drain and the water depth around it is about 2m; while the water intake of 6m below the water surface is by the way of scattered water intake and the water depth around it is about 7m. The relative position of Ankang reservoir, water intake and water drain is shown in Figure 1.

NOMENCLATURE

Abbreviation			
SWHP	surface water heat pump system	$f(v)$	wind function, $f(v)=9.2+0.46v^2$ (W/m ² x mmHg)
EER	the cooling/heating energy efficiency ratio of SWHP unit (using “+” when in cooling period and “-”when in heating period) (W/W)	D_z	the thermal diffusion coefficient(m ² /s)
		$\bar{q}$	maximum released/absorbed heat carried by water in unit water surface area(W/m ²)
Symbols		$\bar{h}$	average water depth, $4 \leq \bar{h} \leq 10$ (m)
$q(x,z,t)$	quality source term	$\bar{t}$	average initial water temperature in vertical direction (°C)
$S(x,z,t)$	energy source term		maximum air conditioning cooling/heating load carried by water in unit surface water (W/m ²)
$m(x,z,t)$	mass flow of water intake and water drain(kg/s)	q	regression coefficients
$V(x,z)$	computing unit volume nearby the water intake and water drain (m ³)	a_1, a_2, a_3, a_4, a_5	
ΔT	the temperature difference between water intake and water drain (°C)	Subscripts	direction of axis
Q	cooling/heating load carried by SWHP system (W)	x,z	wind
p	pressure (Pa)	w	air
u	velocity in x direction (m/s)	a	
w	velocity in z direction (m/s)	Greek Letters	
E	turbulent diffusion coefficients (m ² /s)	μ	turbulent viscosity (m ² /s)
v	wind velocity of 10m above the water surface (m/s)	τ	shear stress on the water surface (Pa)
v_w	wind velocity component in x direction (m/s)	ρ	density (kg/m ³)
		φ_n	net heat absorbed by water(W/m ²)

2.2 Method

In summer operation process, the cooling load is constantly released to the water through system, lead to the excessive increase of water temperature. When the water intake temperature is higher than a limit, the EER of SWHP system will be lower than the traditional air-conditioning system. Therefore, the water intake temperature limit is used to characterize the water heat-carrying capacity. The maximum released heat carried by water is defined as the summer water heat-carrying capacity. It is opposite in winter.

Ignoring the secondary factors of EER of unit, temperature difference between water intake and water drain, water drain mode, etc, base on the typical meteorological parameters in Chongqing, China, this paper mainly researches the mathematical relationship between the heat-carrying capacity of static water in open-loop SWHP system and the water characteristic parameters such as water depth, initial water temperature.

2.2.1 The vertical 2-D water temperature model

In Europe, America and China, the vertical 2-D natural surface static water temperature model has been mainly researched, and it is relatively mature (Joseph 1983; Chen 1998; Andrew 2000; Vassilis and Soultana 2003).The vertical 2-D water temperature model in this paper is established base on the front model in the previous research.

2.2.1.1 Governing equations

Base on the fluid motion equations N-S equations, the vertical 2-D water temperature model under the condition of inputting load by respectively adding the quality and energy source terms in the continuous and temperature equations is established. The governing equations can be expressed in the equations (1)-(3). They can be expressed in the formulas (4)-(6).

2.2.1.2 Boundary and initial conditions

(1)Free surface

Using the steelmaking cap supposition, namely the water depth and liquid level have no change over time, there are $\partial u / \partial z = 0$, $w = 0$.

It can be expressed in formulas (7)(8) and (9).

For temperature equation, the free surface boundary condition reflects the heat transfer between the water surface and air, which can be expressed in formula (10).

(2) Reservoir bottom and surrounding walls

For flow equation, it is set as no-slip boundary, $u=w=0$; for temperature equation, it is set as adiabatic boundary.

(3)Inlet boundary

In open-loop SWHP system, inlet boundary is just the water drain of system, namely the computing unit of water drain. For flow equation, it needs to give the mass flow of water drain; for temperature equation, it needs to give the water drain temperature.

(4)Outlet boundary

In open-loop SWHP system, outlet boundary is just the water intake of system, namely the computing

unit of water intake. For flow equation, it needs to give the mass flow of water intake; for temperature equation, it needs to give the water intake temperature.

(5) Initial condition

The temperature and velocity fields in the beginning operation of open-loop SWHP system which can be achieved by experimental test are given, namely $T(x,z,0)=T_0(x,z)$; $u(x,z,0)=u_0(x,z)$; $w(x,z,0)=w_0(x,z)$.

2.2.1.3 Other setting conditions

According to the climate conditions in Chongqing, China, the cooling/heating period is set as follows.

(1) The summer cooling period is from Jun. 1st to Aug. 31st. Temperature difference between water intake and water drain is set as 5°C by the method of water intake from the bottom and water drain to the constant temperature layer.

(2) The winter heating period is from Dec. 15th to Feb. 15th. Temperature difference between water intake and water drain is set as 4°C by the method of water intake from the bottom and water drain to the surface layer.

Scatter the equations by alternating direction implicit difference scheme, and solve the discrete equations in MATLAB program by forward elimination and backward substitution.

2.2.2 Set the summer and winter water intake temperature limits

The energy-saving performance of open-loop SWHP system in summer is relative to the traditional air-conditioning system. The energy consumption of intake pump in open-loop SWHP system is generally greater than that of cooling water circulation in cooling tower system. The open-loop SWHP system will be not energy-saving when its water intake temperature is higher than the standard cooling tower outlet temperature, so 32°C is defined as the highest water intake temperature limit of open-loop SWHP system.

In winter operation of system, the heat which is constantly absorbed from the water results in the continuous decrease of the bottom water temperature. When the absorbed heat from water reaches a limit, the bottom water intake temperature will also reaches a limit at this moment, and then the unit inlet temperature will reach 0°C so that the unit can not operate normally in open-loop SWHP system. The winter water intake temperature limit is set as 4°C by the temperature difference between water intake and water drain 4°C.

2.2.4 Confirm the water heat-carrying capacity

This paper mainly studies the mathematical relationship between the heat-carrying capacity of static water in SWHP system and the water characteristic parameters (water depth, initial water temperature) which can be expressed in formula (11).

Using formula (11), according to the relationship between the released/absorbed heat carried by water and air-conditioning cooling/heating load, the

mathematical relationship between the maximum air-conditioning cooling/heating load carried by water, water characteristic parameters and EER of SWHP unit can be achieved, as shown in formula (12).

Firstly, respectively confirm the maximum released/absorbed heat carried by water in different average water depths and different average water temperatures; Secondly, achieve the regression equation form with the minimum residual sum of squares from the results by least square method; Thirdly, achieve the regression equation between the maximum released/absorbed heat carried by water and the water characteristic parameters by compiled program in MATLAB; Finally, using formula (12), achieve the regression equation between the maximum air-conditioning cooling/heating load carried by water and the water characteristic parameters.

3. RESULTS AND DISCUSSION

The small thermally stratified static water with 4-10m in depth in SWHP system is mainly researched in this paper. Taking Chongqing, China in hot summer and cold winter zone as an example, the mathematical relationship between water heat-carrying capacity and water characteristic parameters is achieved.

3.1 Calculate the summer water heat-carrying capacity

The summer water heat-carrying capacity is achieved by calculating the bottom water temperature changes in different average water depths (4-10m), different average water temperatures and different released heat which is inputted into the vertical 2-D water temperature model during cooling period. Figure 2 shows the bottom water temperature changes with 4m in average water depth, 26.35°C in average water temperature, separately 0, 14, 16W/m² in released heat carried by water. Figure 3 shows the bottom water temperature changes with 5m in average water depth, 26.02°C in average water temperature, separately 0, 20, 22W/m² in released heat carried by water.

As shown in Figure 2: when the released heat carried by water is 16W/m², the bottom water temperature is below 32°C from the beginning of June to early August, and it is higher than 32°C from the middle August; when the released heat carried by water is 14W/m², the bottom water temperature is below 32°C from the beginning of June to late August, and it is slightly higher than 32°C at the end of August which is also the end of the cooling period, so 14W/m² is confirmed the maximum released heat carried by water with 4m in average water depth and 26.35°C in average water temperature. Using the same calculation method, the maximum released heat carried by water with different average water depths and different average water temperatures can be respectively confirmed. Using the above-mentioned mathematical method, the regression equations can be achieved.

The summer regression equation form with the minimum residual sum of square can be expressed in formula (13).

The regression equation between the maximum released heat carried by water and the water characteristic parameters can be expressed in formula (14).

The regression equation between the maximum air-conditioning cooling load carried by water and the water characteristic parameters can be expressed in formula (15).

3.2 Calculate the winter water heat-carrying capacity

The winter water heat-carrying capacity is achieved by calculating the bottom water temperature changes in different average water depths (4-10m), different average water temperatures and different absorbed heat which is inputted into the vertical 2-D water temperature model during heating period. Figure 4 shows the bottom water temperature changes with 5m in average water depth, 14°C in average water temperature, separately 0, 24, 28W/m² in absorbed heat carried by water. Figure 5 shows the bottom water temperature changes with 6m in average water depth, 13°C in average water temperature, separately 0, 26.6W/m² in absorbed heat carried by water.

Using the same calculation method as summer, the maximum absorbed heat carried by water with different average water depths and different average water temperatures can be respectively confirmed. Using the same mathematical method, the regression equations can be achieved.

The winter regression equation form with the minimum residual sum of square can be expressed in formula (16).

The regression equation between the maximum absorbed heat carried by water and the water characteristic parameters can be expressed in formula (17).

The regression equation between the maximum air-conditioning heating load carried by water and the water characteristic parameters can be expressed in formula (18).

Using the parameters of average water depth, average water temperature, system EER, etc, by formula (15) and (18) the maximum air-conditioning cooling/heating load carried by water in open-loop SWHP system under summer and winter conditions can be calculated. Base on this, the adaptability of the static water such as reservoir used as the low level cooling/heating source of open-loop SWHP system can be evaluated.

3.3 Validation

Taking the Chongqing Kai County people's hospital using open-loop SWHP system which takes Ankang reservoir as the lower level cooling/heating source as the experiment table, the operation condition of system had been tested in the summer of 2008. The

average water temperature of the reservoir is 22.31°C before the system operating, and the summer cooling period of the system is from Jun. 1st to Aug. 31st. Due to some departments of the hospital was not used, the system load was relatively low. Table 1 shows the comparison of the released heat and bottom water temperature respectively achieved by experiment test and theoretical approach presented in this paper.

As shown in Table 1, it is similar but slightly different between the results by experiment test and theoretical approach. The reasons may be as follows: (1) in actual operation process, the system flow, temperature difference between water intake and water drain, etc, are constantly changing; (2) the unit is in discontinuously operation; (3) different rainfall process will influence the reservoir water temperature. These factors are ignored in the calculation process. But overall, the experiment test results can better prove the reliability of the calculation method of water heat-carrying capacity.

4. CONCLUSIONS

This paper presents the detailed calculation method of the water heat-carrying capacity. Firstly, taking examples from the current relatively mature vertical 2-D natural water temperature model, base on the fluid motion equations-N-S equations, establish the vertical 2-D water temperature model under the condition of inputting load by respectively adding the quality and energy source terms in the continuous and temperature equations. Secondly, according to the summer and winter water intake temperature limits which base on the SWHP unit operational characteristic, achieve the water temperature changes in different water depths (4-10m), different initial water temperatures and different released/absorbed heat during cooling/heating period; Finally, by numerically analyzing the water temperature changes, achieve the numerical relationship between the water heat-carrying capacity and the water characteristic parameters (water depth and initial water temperature) by regression method. This calculation method can be applied in the practical project, as the criterion to evaluate the adaptability of the surface static water as the low level cooling/heating sources of the open-loop SWHP system.

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TABLES

Table 1 Comparison of the results by experiment test and theoretical approach

Theoretical heat-carrying capacity (w/m ²)	Experimental released heat (W/m ²)	Theoretical temperature limit (°C)	Experimental bottom water temperature at the end of August (°C)
30	33.4	32	31.56

FIGURES

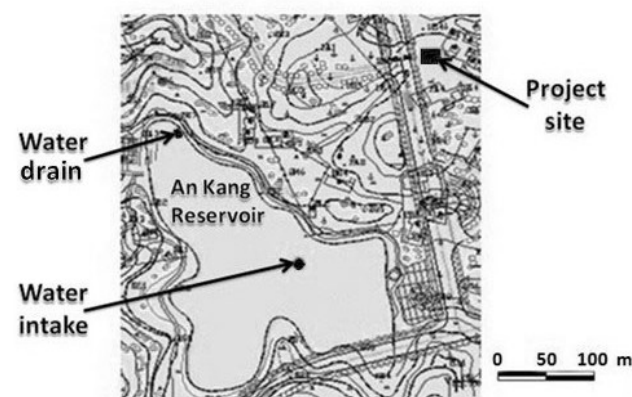


Figure 1 The relative position of Ankang reservoir, water intake and water drain in satellite image

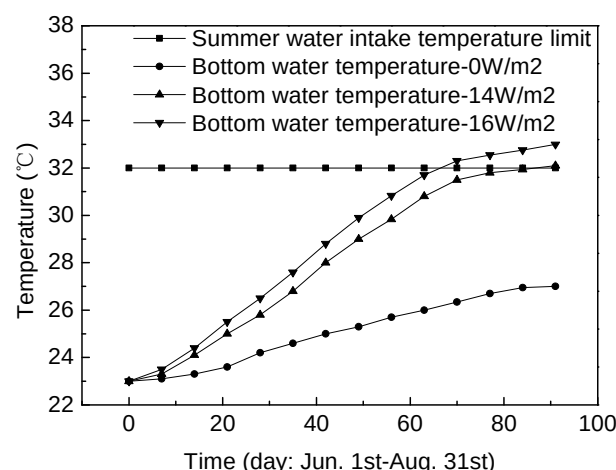


Figure 2 The bottom water temperature changes with 4m in average water depth, 26.35°C in average water temperature, separately 0, 14, 16W/m² in released heat carried by water

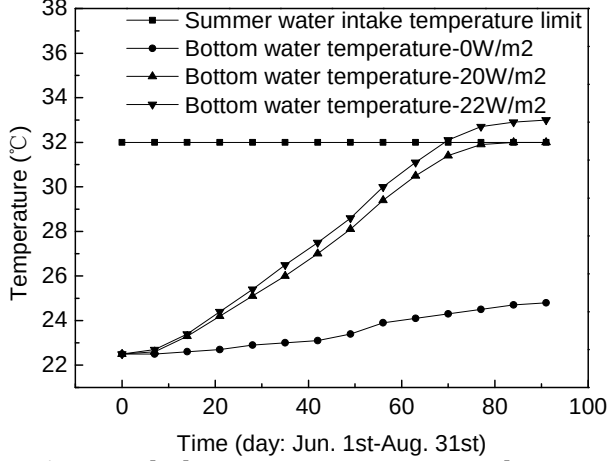


Figure 3 The bottom water temperature changes with 5m in average water depth, 26.02°C in average water temperature, separately 0, 20, 22W/m² in released heat carried by water

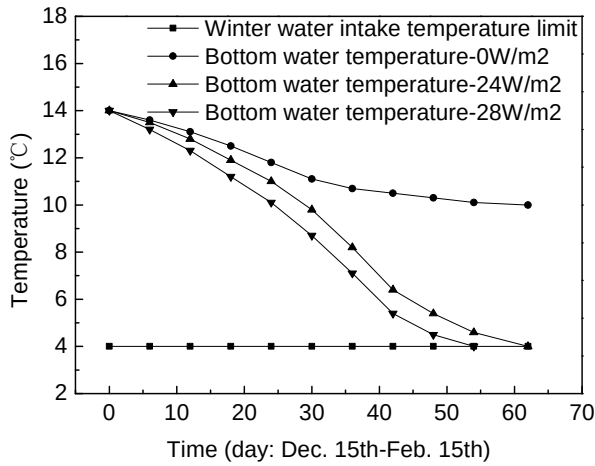


Figure 4 The bottom water temperature changes with 5m in average water depth, 14°C in average water temperature, separately 0, 24, 28W/m² in absorbed heat carried by water

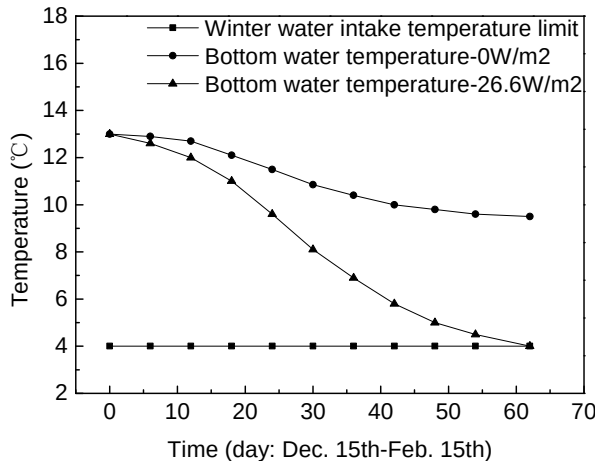


Figure 5 The bottom water temperature changes with 6m in average water depth, 13°C in average water temperature, separately 0, 26.6W/m² in absorbed heat carried by water

FORMULAS

Continuous equation:

$$\frac{\partial u}{\partial x} + \frac{\partial w}{\partial z} = \frac{1}{\rho} q(x, z, t) \quad (1)$$

Momentum equation:

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + w \frac{\partial u}{\partial z} + \frac{1}{\rho} \frac{\partial p}{\partial x} = \frac{\partial}{\partial x} \left(\mu_x \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial z} \left(\mu_z \frac{\partial u}{\partial z} \right) \quad (2)$$

Water temperature equation:

$$\frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left(E_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial z} \left(E_z \frac{\partial T}{\partial z} \right) - u \frac{\partial T}{\partial x} - w \frac{\partial T}{\partial z} + \frac{1}{\rho c_p} \frac{\partial \phi_z}{\partial z} + \frac{1}{\rho} S(x, z, t) \quad (3)$$

$$q(x, z, t) = \begin{cases} +m(x, z, t)/V(x, z) & \text{the computing unit of water drain} \\ -m(x, z, t)/V(x, z) & \text{the computing unit of water intake} \\ 0 & \text{other computing units} \end{cases} \quad (4)$$

$$S(x, z, t) = \begin{cases} q(x, z, t) \Delta T & \text{the computing unit of water drain} \\ 0 & \text{other computing units} \end{cases} \quad (5)$$

$$m(x, z, t) = \frac{Q}{c_p \Delta T} \left(\frac{EER \pm 1}{EER} \right) \quad (6)$$

with wind:

$$\mu_z \frac{\partial u}{\partial z} = \frac{\tau_{wx}}{\rho} \quad (7)$$

$$\tau_{wx} = \rho_a f(v) |v| v_w \quad (8)$$

Without wind:

$$\tau_{wx} = 0 \quad (9)$$

$$\frac{\partial T}{\partial z} = - \frac{\phi_n}{\rho c_p D_z} \quad (10)$$

$$\bar{q} = f(\bar{h}, \bar{t}) \quad (11)$$

$$q = f(\bar{h}, \bar{t}) \times \frac{EER}{(EER \pm 1)} \quad (12)$$

$$y = a_1 + a_2 x_1^{a_3} + a_4 x_2^{a_5} \quad (13)$$

$$\bar{q}_{cooling} = 130.7321 + 1.0025(\bar{h})^{1.7509} - 11.8729(\bar{t})^{0.7333} \quad (14)$$

$$q_{cooling} = (130.7321 + 1.0025(\bar{h})^{1.7509} - 11.8729(\bar{t})^{0.7333}) \times \frac{EER}{(EER+1)} \quad (15)$$

$$y = a_1 + a_2 x_1 + a_3 x_2 + a_4 x_1^2 + a_5 x_2^2 \quad (16)$$

$$\bar{q}_{heating} = -94.2202 + 9.231(\bar{h}) + 9.9024(\bar{t}) - 0.4619(\bar{h})^2 - 0.2694(\bar{t})^2 \quad (17)$$

$$q_{heating} = (-94.2202 + 9.231(\bar{h}) + 9.9024(\bar{t}) - 0.4619(\bar{h})^2 - 0.2694(\bar{t})^2) \times \frac{EER}{(EER-1)} \quad (18)$$

STATE OF CHARGE ESTIMATION WITH LEAST SQUARES SUPPORT VECTOR MACHINE

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ABSTRACT

In order to enhance the precision of SOC estimation of LiFePO₄ battery, this paper uses Least Squares Support Vector Machine (LSSVM) to establish a prediction model based on the estimator with radial basis function kernel. According to the influence of SOC, the main Influencing factors include OCV, time and discharge rate was selected as the input variable, State of charge as the output variable. Data from another running period was used to verify model prediction accuracy. The results show that the precision of the model can meet the requirements. The results show that the maximum relative error (MRE), the mean absolute percentage error (MAPE) and the root mean square error (RMSE) are 5.784 % and 5.57 %,1.35%,respectively. The results show that the LS-SVM model is pithily, and it has better extensive capability than traditional methods. The new method can be viewed as a new approach to advance the development of SOC model and improve the prediction accuracy.

Keywords: Least Squares Support Vector Machine, state of charge, Kernel function, Prediction Model

1. INTRODUCTION

An intermittent energy source such as wind and solar is any source of energy that is not continuously available due to some factor outside direct control. For solve this difficulty, Battery Energy Storage System, include EV, HEV and UPS especially in the building, can be used to offset the adverse effects of intermittent renewable energy sources. LiFePO₄ battery is a member of a family of secondary battery in which lithium ions move from the positive electrode to the negative electrode during charging, and back when discharges.

Although the LiFePO₄ battery energy storage system is helpful to solve the air pollution, yet its energy density far less than gasoline. Thus, how to effective use of limited energy, avoiding unnecessary waste and accurate indication of battery energy is especially important. To explore the hidden energy of battery as soon as possible, one has proposed the concept of "State of Charge".

An assortment of methods is proposed in the literature or patent to measure or estimate the State of Charge(SOC). Ampere-hour integration model (Ah) is an iterative algorithm and is one of the most widely used models that is required dynamic current with time integral[1-4]. But it has a fatal weaknesses that is the initial state of charge (SOC₀) is difficult to estimate. The similar things happened to a king of the intelligent algorithm, such as Kalman Filter[5-7], unscented Kalman filter[8]. However, these models do not provide much attention to uncertainty in the SOC₀. The

incorrect SOC₀ leading to further deterioration of the accuracy of SOC estimation. Open Circuit Voltage (OCV) can be used to calibrate the SOC₀, nevertheless, it needs a long time in order to achieve a steady voltage.

Least Squares Support Vector Machine (LSSVM) [9-10]method is used for battery SOC estimation, LSSVM as a kind of neural network, describes the relationship between the input and output that don't need a precise formula, as long as reasonable selection of input and output parameters, the relationship can be determined in the process of network training. At the same time, the LSSVM estimation method with adaptive characteristics. New data can be used again to the training of the network, so as to realize the compatible with the new data.

2. TEST BENCH AND TESTING STEPS

The Battery test bench is built and the experimental schedule is designed. In order to build the relationship of SOC₀ with various parameters contain temperature, open circuit voltage(OCV) and resting time, a series of experiments is specially designed and correlation parameters are measured according to designed SOC point.

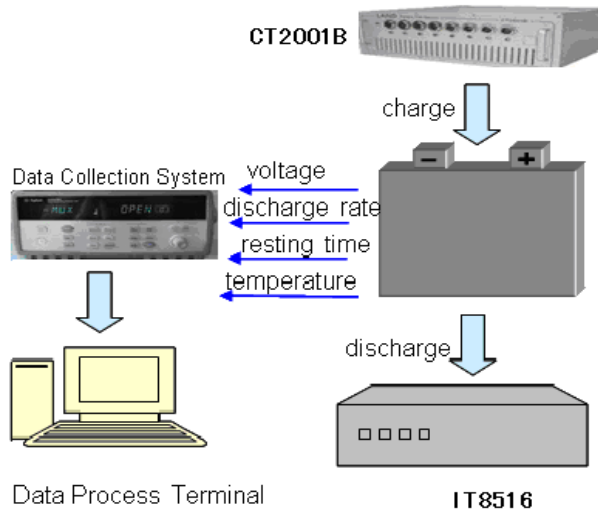


Figure 1 Schematic of the battery test bench.

LiFePO₄ batteries with nominal of 3.20 V/20 Ah are selected for the experiments. The test bench is shown in Fig.1, which consists of a battery testing system, data acquisition system and a data process terminal. The batteries are charged by Charging instrument LAND-CT2001B and discharged by a programmable electronic load ITECH-IT8516. The signals including operating time, temperatures, currents and voltages are monitored and recorded by data acquisition modules, then processed in a personal host computer. Furthermore, the data collection system, whose measurement accuracy of DC voltage is as high as $\pm 0.002\%$ with a 6.5 digit resolution, applied for voltage measurement in order to improve the sample precision of voltages.

3. SOC MODEL

In the experiment, the battery is first charged with a constant current of 0.3C to a threshold of 3.65 V, and then by a constant-voltage of 3.65V charge until the current density decreased to 0.002C.

Following and tracking some useful methods of the traditional testing steps, discharge-resting-discharge, an effective method for OCV test is proposed. In order to explore the changes of OCV at the different point of SOC and resting time, the following experiment was carried out.

With constant discharging rates (0.1C-1C), the battery voltage decreases gradually and reaches the designed capacity (SOC=10%-90%). Then, the battery needs to rest for several minutes (30 minutes in this experiment). That is to say, no current through the electrodes and ohmic resistance will instantly disappear, however, the polarization of electrochemical and concentration is gradually weakened resulting in terminal voltage slowly rising.

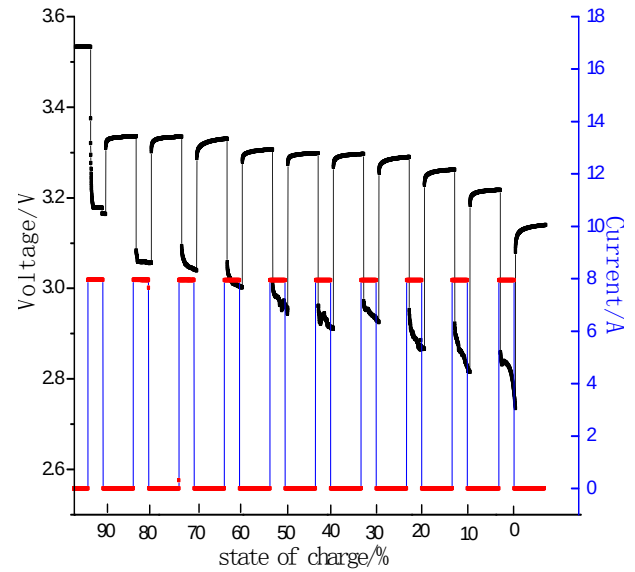


Figure 2 The discharge-resting experiment (0.4C)

In the practical application, the interrelationship between SOC and OCV was estimated and used as the initial SOC for Ah model. However, the long waiting time for the battery to reach a steady state must be considered if the OCV model was used, otherwise, the OCV will cause the illusion of "virtual-high" capacity in the SOC estimation. Whereas OCV cannot be measured quickly, but estimated from other correlation parameter. To improve the accuracy and the velocity of SOC₀, an enhanced SOC₀ model was proposed and also takes into account the effect of current rates, time and temperature.

4. ALGORITHM OF LS-SVM

The least square support vector machine (LSSVM) is first proposed by Suykens and Vandewalle (1999). It is an alternate formulation of SVM regression. Support vector machine is a new machine learning method based on statistical learning theory, and is specialize small sample situation. The kernel idea of SVM is control learning machine's extend ability through control learning machine's complexity. It is a powerful methodology for solving problems in density estimation non-linear classification, and function estimation.

The basic principle of SVM is that consider given inputs and output series to estimation can be stated briefly as follows: first, map the training set into the nonlinear feature space with a higher dimension; then, for the problem with the optimal separating hyperplane control function has also been constructed in the feature space explicitly. All of the separating hyperplanes can be formulated in the feature space:

$$f(x) = w \cdot \phi(x) + b \quad (1)$$

Where f indicates the relationship between the input and the output variable, w is the m -dimensional weight vector $\phi(\bullet)$ is the mapping function between the input space to the output space and b is the corresponding bias term.

Taking into account the complexity of the function, the regression function can be given according the structural minimization principle as

$$\min R(w, \xi, \xi^*) = \frac{1}{2} \|w\|^2 + c \sum_{i=1}^k (\xi + \xi^*) \quad (2)$$

Subject to

$$\begin{cases} f(x_i) - y_i \leq \xi^* + \varepsilon, i = 1, 2, \dots, k, \\ y_i - f(x_i) \leq \xi + \varepsilon, i = 1, 2, \dots, k, \\ \xi, \xi^* \geq 0, i = 1, 2, \dots, k, \end{cases} \quad (3)$$

where c is penalty parameter that controls the degree of the punishment beyond the sample error; ξ_i^*, ξ_i is relaxation factor; ε is loss function.

The resulting optimization problem of LSSVM can be formulated as the following:

$$\min_{w, b, \xi} \frac{1}{2} \|w\|^2 + c \sum_{i=1}^k \xi_i^2 \quad (4)$$

Subject to

$$f(x_i) = w \cdot \phi(x_i) + b + \xi_i, i = 1, 2, \dots, k \quad (5)$$

To solve the optimization problem given in equation(2). The constraint problem By converted the constraint problem into unconstraint one, the objective function can be obtained. The Lagrangian comes in the form:

$$L = \frac{1}{2} \|w\|^2 + c \sum_{i=1}^k \xi_i^2 - \sum_{i=1}^k a_i \{y_i [w \cdot \phi(x_i) + b] - 1 + \xi_i\} \quad (6)$$

Where a_i is the Langrange multiplier. According to the conditions of the optimality theory, the following equations must be satisfied: $\partial L / \partial w = 0$ $\partial L / \partial b = 0$ $\partial L / \partial \xi_1 = 0$; $\partial L / \partial a_1 = 0$. Subsequently, according to the condition of the Karush-Kuhn-Tucker the optimal conditions can be obtained by taking the partial derivatives. Thus the linear equations are obtained as

$$\begin{bmatrix} 0 & -Y^T \\ Y & QQ^T + \gamma^{-1}I \end{bmatrix} \begin{bmatrix} b \\ a \end{bmatrix} = \begin{bmatrix} 0 \\ \vec{I} \end{bmatrix} \quad (7)$$

Where $Q = [\phi(x_1)^T y_1, \dots, \phi(x_m)^T y_m]$;

$\vec{I} = [1, \dots, 1]^T$; $Y = [y_1, \dots, y_m]^T$; $a = [a_1, \dots, a_m]^T$; Then regression function of LSSVM is obtained.

$$f(x) = w \cdot \phi(x) + b = \sum_{i=1}^k a_i k(x_i, x) + b, i = 1, 2, \dots, k \quad (8)$$

Where kernel function can be expressed as $k(x, x_i) = \phi(x)^T \phi(x_i)$ and it meets Mercer' condition. It only consider the dot product of the high-dimensional feature space rather than directly using to avoid the complex calculations. Commonly used kernel functions are the polynomial kernel function, Gaussian

function, the radial basis function(RBF) and Sigmoid function. The RBF is a powerful in the application, so it is selected and given as follows:

$$k(x_i, x) = \exp \frac{-(x_i - x)^2}{2\sigma^2} \quad (9)$$

Where σ is the width of the RBF kernel.

5. SOC ESTIMATION OF LSSVM MODEL

The initial state of charge of LiFePO4 battery is a multi-parameter function involved in the theory of electrochemical reaction and thermodynamic, and it also shows the strong feature of non-linearity and time-varying to further increased the difficulty of the prediction. Least squares support vector machine (SVM) is an improved algorithm based on support vector machine (SVM), and application of the theory of structural minimization risk. It chooses feature subsets from a set of training set, and made a linear divided to the feature subsets that is equal to the segmentation of the entire data set. LSSVM is a form under quadratic loss function, and put the secondary optimization of support vector machine (SVM) algorithm transformed into linear equations, so as to solve the complex computing problems, and effectively overcome the dimension disaster and local minimum problem.

Based on correlation analysis of the SOC and main parameters, this paper selects four parameters as input that are OCV, resting time, discharge rate, and temperature and sets SOC0 as output to establish prediction model by LS-SVM. Factors parameters are as shown in Tab.1. Radial basis function is better than other kernel function on dealing with the nonlinear sample. Complex nonlinear relationship is shown between fouling and water quality parameters. Therefore, this paper selects radial basis function as kernel function and makes use of cross-examination to determine the parameters, regularization parameter 0.1, kernel width Gamma 500.

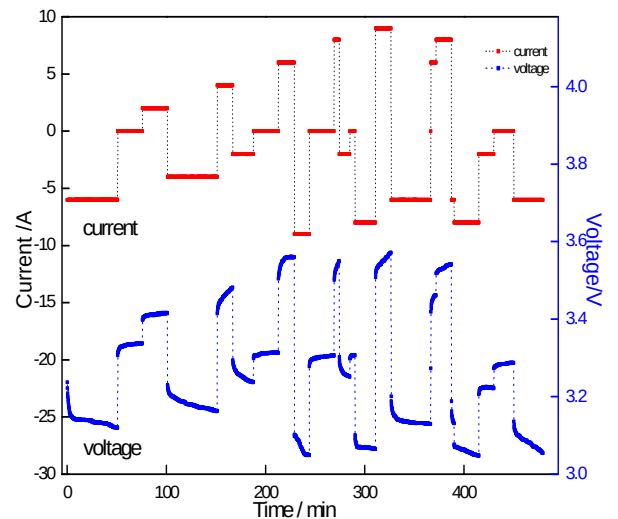


Figure.3 Voltage, current, of dynamic Discharge

6. RESULTS AND DISCUSSION

In this paper, we adopted two cycle of experimental data, one cycle of the data for charged state estimation model is set up, The other period was used for testing the accuracy of model. The comparison results between the actual value and the prediction value as shown in figure 4.

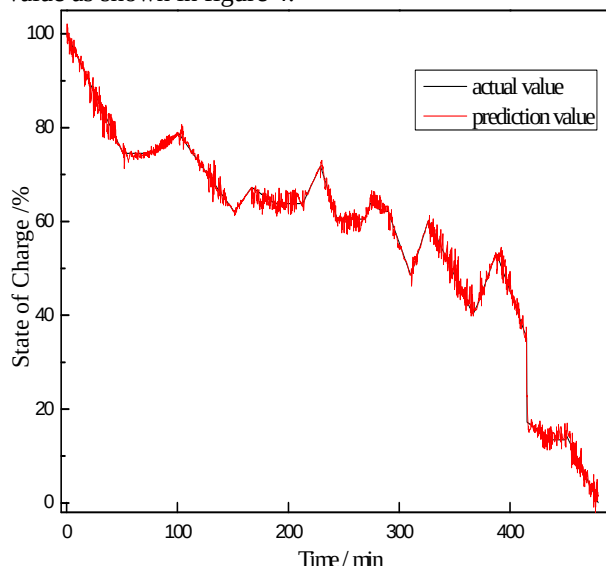


Figure.4 The comparison curve of experiment and prediction value

In order to quantitatively evaluate accuracy of the model, the maximum relative error (MRE), the mean absolute percentage error (MAPE) and the root mean square error (RMSE) are all used to evaluate the accuracy of model prediction. Among them, the MRE used to calculate the error between the predicted values and the measured values. Mean absolute percentage error (MAPE) and the root mean square error (RMSE) to determine the prediction model. The error (actual value and the prediction value)curve as shown in Figure.5.

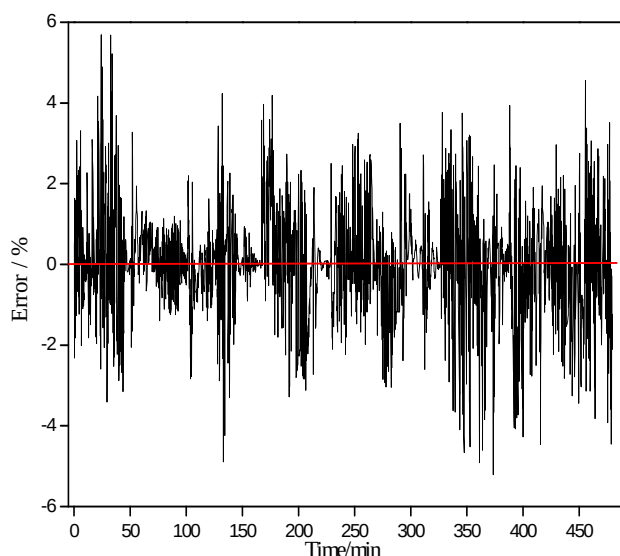


Figure. 5 Error curve of LSSVM prediction model

Table 1 Error of the LSSVM method

Prediction model	MAPE	RMSE	MRE
LSSVM	5.57	1.35	5.784

The result of practical application indicates that the performance of LSSVM has a good coordination capacity between the model complexity and generalization ability can overcome the problem of local optimum and over learning of neural network model excellently. It provides an accurate and reliable method for the prediction of SOC.

7. CONCLUSIONS

Least squares support vector machine (SVM) can be used to estimate SOC. Intermittent discharge dynamic experiment was conducted and the main variable contain open circuit voltage, discharge rates and resting time were measured. Prediction model of SOC based on the LSSVM was established and the result indicted that it has a good accuracy. The results show that the maximum relative error (MRE), the mean absolute percentage error (MAPE) and the root mean square error (RMSE) are 5.784 % and 5.57 %, 1.35%, respectively.

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COMPARISON OF BOREHOLE THERMAL RESISTANCE UNDER various TEMPERATURE PROFILE APPROXIMATIONS IN THERMAL RESPONSE TEST

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ABSTRACT

The thermal parameter estimation of the soil and borehole is of great importance in the Ground Coupled Heat Pump (GCHP) system design procedure. In general, the arithmetic mean temperature is used, which may lead to an overestimation of the borehole resistance. In this paper, a well-organized thermal response test was performed in Hunan University, Changsha. To improve the prediction accuracy of the vertical temperature profile for a vertical pipe, various vertical temperature profile approximations were proposed. Various borehole thermal resistances were obtained using six different average assumption approximations of the inlet and outlet circulating water temperature. A recommended p value was proposed by the comparison between the theoretical results which were based on the quasi-three dimensional model and various assumption vertical temperature profiles. Comparing to the recommended p value, the borehole thermal resistance with the arithmetic mean method has a 7.70% overestimation. Finally, the exactly p value calculated by the curve fitting method is achieved at three special times. The exactly p value varying with time is also illustrated in a figure.

Keywords: thermal parameter estimation, p -linear averaging method, vertical temperature profile, in situ thermal response test

1. INTRODUCTION

Ground coupled heat pump (GCHP) systems have been considered as renewable energy systems for their carbon reduction and energy conservation. However, the GCHP systems are not always operated at the efficient mode because of the poor system design. As the heating and cooling load of the building was transfer into the ground through the deep borehole heat exchangers (BHEs), the effective ground thermal properties such as ground thermal conductivity and the borehole thermal resistance were the key parameters during the GCHP system design period.

At the very beginning, the thermal properties were analyzed in the laboratory, which was estimated by the known geology. However, it was updated by the on-site thermal response test (TRT) due to the low accuracy thermal properties prediction. The TRT was first presented by Mogensen(Mogensen 1983), and followed by AUSTIN(AUSTIN III 1998), Gehlin(Gehlin 2002), Raymond(Raymond, Therrien et al. 2011), shonder, JA and Beck, JV(Shonder and Beck 2000) and so on. With a constant heat flow, the circulating water flowed into the HDPE U-pipe, and was cooled by the surrounding ground along the pipe. The circulating water flow rate, the inlet and outlet temperature should be record every five or ten minutes, and the test would last no less than 48 hours(AUSTIN III 1998). In the test, the borehole heat exchanges can be modeled as a heat source in the forms of line or cylinder. Although numerous of heat

source models were proposed, the infinite line source model was the most conventional one applied in TRT analysis, as it was very easy to calculate and provided an acceptable accuracy result.

Most of approaches to interpret the TRT were based on the non-linear curve fitting method, either by least square method (LSM) or Manual adjustment. The curve fitting method in the TRT was determined by the infinite line source equation, and it can be written as a function of time. The y data and x data for this curve fitting equation is the average temperature of heat carrier fluid along the HDPE U-pipe and the time, respectively. Because only the inlet and outlet heat carrier fluid temperature was measured, the vertical assumption fluid temperature profile along the HDPE

U-pipe played an important role in the TRT thermal properties estimation, and it was in great connection with the accuracy of the estimation results. In most of the conventional TRT, the arithmetic mean temperature of the inlet and outlet fluid temperature was taken into consideration as the average fluid temperature which was reported to lead to an overestimate of the borehole thermal resistance. Marcotte and Pasquier(Marcotte and Pasquier 2008) proposed a more precise temperature profile approximation based on the finite element method, it was defined as p -linear average method. The arithmetic mean method as well as other special average methods can be included with different p value.

NOMENCLATURE

Symbols		out	outlet
R	thermal resistant (m.K/W)	b	borehole wall
r	Radius(m)	s	soil
T	Temperature (°C)	f	fluid
t	Time(s)	p	Weight of the temperature assumption
k	thermal conductivity (W/m.K)	Greek Letters	
q	heat flow per meter (W/s)	α	Thermal diffusivity (m ² /s)
2D	The distance between the two leg of U pipe(m)	ρ	Density (kg/m ³)
Subscripts		$\Delta\bar{T}$	temperature excess °C
in	inlet	r	Euler's Constant

Du and Chen(Du and Chen 2011) and Lamarche et al. (Lamarche, Kaji et al. 2010) gave a modified p-linear approximation based on the quasi-three-dimensional model analytical solution, it has not been widely used because the reference temperature was the borehole wall temperature which cannot be obtained by the measurement. Beier(Beier, Acuña et al. 2012) provided different p value assumption base on the Marcotte and Pasquier's equation.

Although so many fluid temperature average methods were recommended, no p values were verified due to the vertical temperature profiles cannot be measured in the circulating fluid along the borehole. In this paper, a well-organized TRT was operated in Changsha, the data set was measured by two thermocouples and a flowmeter with high accuracy, and collected by a data logger. Six different fluid temperature profile approximations are taken into consideration, and the thermal properties estimated by various fluid temperature average methods are applied into a new verification method base on the quasi-three-dimensional model. The recommend p value is also proposed in this paper for this TRT case based on the curve fitting method.

2. THEORETICAL BACKGROUND

We have briefly reviewed the theoretical models which are related to the ground thermal properties estimation base on the data collected by the TRT. A lot of analytical models for evaluation of the TRT data under constant heat input rate test were proposed such as the infinite line source, finite line source and the cylinder heat source, it was well reviewed by Yang(Yang, Cui et al. 2010). Some more sophisticated models which were analysis by numerical models is suggested by Austin and Yacuzturk(Austin 2000), Gehlin and Spiliter(Gehlin and Spitler 2003). Among all these models, the infinite line source (ILS) was the most widely used because it is easy to operate and give an acceptable result.

In the ILS model, the undisturbed ground temperature T_s , which was assumed to stay uniform along the borehole, was set as the reference temperature, the average temperature excess is defined as equation 1,

$$\Delta\bar{T}(r,t) = \bar{T}(r,t) - T_s = \frac{q}{4\pi k_s} \int_{r^2/4\alpha t}^{\infty} \frac{e^{-u}}{u} du$$

$$\approx \frac{q}{4\pi k_s} \left(\ln \frac{4\alpha_s t}{r^2} - \gamma \right) \quad (1)$$

The previous work has indicated that if $4\alpha_s t/r_b^2 > 5$, the natural logarithm approximation is accurate to within 10%(Lee, Park et al. 2011), and the γ is the Euler's Constant, and it is approximated as 0.57722. The temperature at the face of the borehole wall is shown as equation 2, as $r = r_b$. And the mean fluid temperature is calculated by equation 3.

$$\bar{T}_b = T_s + \frac{q}{4\pi k_s} \left(\ln \frac{4\alpha_s t}{r_b^2} - \gamma \right) \quad (2)$$

$$\bar{T}_f(t) = \bar{T}_b + qR_b = T_s + q \left(R_b + \frac{1}{4\pi k_s} \left(\ln \frac{4\alpha_s t}{r_b^2} - \gamma \right) \right) \quad (3)$$

Equation 3 indicates that the average fluid temperature rise follows a linear trend in a plot with the nature logarithm of time on the horizontal axis, it can be written as equation 4. However, the equation cannot be applied to the very beginning of the TRT, because the fluid temperature is not steady and affect by the fluid and pipe thermal storage, this period is usually last for about 10 hours.

$$\bar{T}_f(t) = a \ln(t) + b \quad (4)$$

Where

$$a = q / (4\pi k_s), b = q \left(R_b + \frac{1}{4\pi k_s} \left(\ln \left(\frac{4\alpha_s}{r_b^2} \right) - \gamma \right) \right) + T_s$$

Base on the above equations, the effective ground thermal conductivity and the borehole thermal resistance can be evaluated by the linear curve fitting with the LSM. Sometimes, some other thermal parameters should be evaluated, such as the ground thermal diffusivity, the initial ground temperature and so on. A non-linear curve fitting with the LSM should be taken into consider, all this work should be completed by Matlab.

In equation 3, the arithmetic mean of inlet temperature and outlet fluid temperature was applied to represent the internal temperature along the borehole. However, this is not realistic as the heat flux is not constant along the borehole. The log average difference is proposed by Incropera et al.(Incropera and DeWitt

1985) as the temperature at the borehole wall is staying constant, this is not unlikely to occur, too.

$$T_{fm} = \frac{(T_{in} + T_{out})}{2} \quad (5)$$

$$\Delta \bar{T}_{fp} = \frac{\Delta T_{out} - \Delta T_{in}}{\ln(\Delta T_{out} / \Delta T_{in})} \quad (6)$$

A new fluid temperature profile approximation along the U-pipe was proposed by Marcotte and Pasquier (Marcotte and Pasquier 2008) based on the 3D numerical model. The formula for the approximated fluid temperature along the U-pipe and the internal average temperature was shown as equation 7 and 8, respectively. Some special average situation, such as arithmetic, geometric, log and harmonic average of the entry and exit temperature, was comprised by the p-average approximation as the weight of the p value is 1, 0, -1/2 and -2, respectively. Marcotte and Pasquier recommend p value as -1 base on 3D numerical model while Beier (Beier, Acuña et al. 2012) recommend -3

base on the vertical temperature profile model. All this values were summarized in table 1.

$$\Delta T_{fp}(x)^p = \Delta T_{in}^p + \frac{x}{L} (\Delta T_{out}^p - \Delta T_{in}^p) \quad (7)$$

$$\Delta \bar{T}_{fp} = \frac{1}{L} \times \int_0^L \Delta T_{fp}(x) dx = \frac{p}{1+p} \cdot \frac{\Delta T_{in}^{p+1} - \Delta T_{out}^{p+1}}{\Delta T_{in}^p - \Delta T_{out}^p} \quad (8)$$

However, six p values were recommended, none of the thermal parameter evaluation results were verified. In this paper, the quasi-three-dimensional model (Zeng, Diao et al. 2003) was employed to simulate the vertical dimensional temperature distribution along the U-pipe. The vertical fluid temperature profile at time t=24h, t=48h, t=70h were simulated, and compared to the theoretical value calculated by the quasi-three-dimensional model, the most reliable result was obtained with the statistics method. The dimensional outlet temperature express will be calculated by the following equation.

Table 1 p-linear temperature excess and average temperature excess

p value	Temperature excess	average temperature excess
$p=1$	$\Delta T_{fp}(x) = \Delta T_{in}^p + \frac{x}{L} (\Delta T_{out} - \Delta T_{in})$	$\Delta \bar{T}_{fp} = \frac{\Delta T_{in} + \Delta T_{out}}{2}$
$p \rightarrow 0$	$\Delta T_{fp}(x) = \Delta T_{in} \left(\frac{\Delta T_{out}}{\Delta T_{in}} \right)^{x/L}$	$\Delta \bar{T}_{fp} = \frac{\Delta T_{out} - \Delta T_{in}}{\ln(\Delta T_{out} / \Delta T_{in})}$
$p=-1/2$	$\Delta T_{fp}(x) = \left(\frac{1}{\sqrt{\Delta T_{in}}} + \frac{x}{L} \left(\frac{1}{\sqrt{\Delta T_{out}}} - \frac{1}{\sqrt{\Delta T_{in}}} \right) \right)^{-2}$	$\Delta \bar{T}_{fp} = \sqrt{\Delta T_{in} \Delta T_{out}}$
$p \rightarrow -1$	$\Delta T_{fp}(x) = \left(\frac{1}{\Delta T_{in}} + \frac{x}{L} \left(\frac{1}{\Delta T_{out}} - \frac{1}{\Delta T_{in}} \right) \right)^{-1}$	$\Delta \bar{T}_{fp} = \frac{\Delta T_{in} \Delta T_{out} \ln(\Delta T_{out} / \Delta T_{in})}{\Delta T_{out} - \Delta T_{in}}$
$p=-2$	$\Delta T_{fp}(x) = \left(\frac{1}{\Delta T_{in}^2} + \frac{x}{L} \left(\frac{1}{\Delta T_{out}^2} - \frac{1}{\Delta T_{in}^2} \right) \right)^{-1/2}$	$\Delta \bar{T}_{fp} = \frac{2 \Delta T_{in} \Delta T_{out}}{\Delta T_{in} + \Delta T_{out}}$
$p=-3$	$\Delta T_{fp}(x) = \left(\frac{1}{\Delta T_{in}^3} + \frac{x}{L} \left(\frac{1}{\Delta T_{out}^3} - \frac{1}{\Delta T_{in}^3} \right) \right)^{-1/3}$	$\Delta \bar{T}_{fp} = \frac{3 \Delta T_{in} \Delta T_{out} (\Delta T_{in} + \Delta T_{out})}{2 \Delta T_{in}^2 + \Delta T_{out}^2 + \Delta T_{in} \Delta T_{out}}$

3. RESULTS AND DISCUSSION

In this paper, a well-organized Thermal Response Test (TRT) was performed in Hunan University, which is located at Changsha City. The circulating water was heated by a 3kW electric heating element, however, the heat flow is not constant because of the voltage fluctuation, the heat carried water was pumped into the U-pipe and cooled by the surrounding ground, two thermocouples and a flow meter were used to measure the entry and exit fluid temperature of the pipe and the volume flow rate of the circulating water. The TRT lasted about 70 hours, the major information about the TRT parameters were list in table 2.

Table 2 Major parameters of the TRT in Changsha

Parameter	Value
Diameter of the borehole(rb)	0.075m
The length of the borehole(H)	94m
The outer radius of the U pipe(ri)	0.014m
The inner radius of the U pipe(ro)	0.016m
The distance between the two leg of U pipe(2D)	0.06m
Thermal conductivity of pipe(kp)	0.35 W/m.K
Average fluid mass flow rate(m)	0.228kg/s
Average input heat rate(q)	2715W

Because the ground thermal diffusivity was not obtained, the non-linear curve fitting method with LSM was employed; A weighted average temperature measured at the entry and exit pipe was assumed to be

the mean fluid temperature along the pipe, the weight in the TRT is defined as p average by Marcotte and Pasquaier, the ground thermal properties evaluation results were list in Table 3 under different p value. Table 3 indicated that the ground thermal conductive and thermal diffusivity various little with different p value, the difference is within 5%, and have little effect on the evaluation results. The borehole thermal resistance changes a lot under different fluid average weight. The thermal properties evaluation process is illustrated in Figure 1 under the arithmetic fluid average method, others are concealed as they were similar with the figure 1.

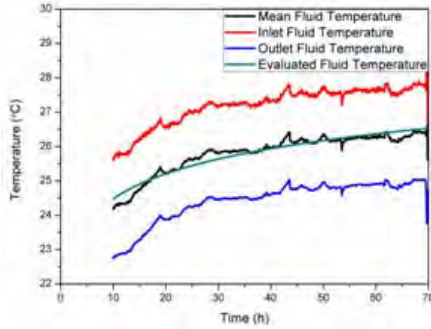


Figure 1 Fluid Temperature under arithmetic mean average method

All these fluid temperature average methods were proposed by different authors, none of them were verify by the on-site TRT. In this paper, the vertical fluid temperature profile was simulated with the quasi-three-dimensional model at times $T_1=24h$, $T_2=48h$ and $T_3=70h$. These six fluid temperature profile approximations also had been calculated at the three time points with the known inlet and outlet temperature. They are shown in figure 2-4.

Figure 2-4 illustrated that in our TRT located in Changsha, the weight of p value is not constant, it various between -2 and -3. At $T_1=24$, the weight of p is closed to -2, while it is closed to -3 at the $T_3=70$, and it is different from Marcotte and Pasquier's recommended value $p=-1$. The more precise assumption in this TRT case should be recommended roughly at $p=-2$, for the borehole thermal resistant will be underestimated at $p=-3$. However, the p value should not always come as integral number, it can be obtained by simple curve fitting. Three recommended p values were proposed at the time points $T_1=24h$, $T_2=48h$ and $T_3=70h$, which is shown in Table 4. From Table 4 and Marcotte and Pasquier's result, we can see that the curve fitting weight is changed as the time flows through, and it is shown in figure 5, as the time flows through, the p value is declined in a line trend.

Table 3 the ground thermal properties evaluation results under different p value

p value	Rb	ks	as	Rb difference	Ks difference
1	0.118	2.177	2.03E-06	0.00%	0.00%
0	0.115	2.149	1.94E-06	2.67%	-1.25%
-0.5	0.114	2.136	1.89E-06	3.98%	-1.86%
-1	0.112	2.123	1.85E-06	5.26%	-2.46%
-2	0.109	2.099	1.76E-06	7.70%	-3.58%
-3	0.106	2.077	1.69E-06	9.97%	-4.59%

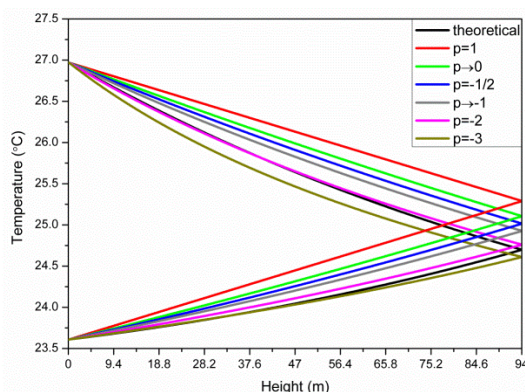


Figure 2 Comparison of temperature profile from quasi-three-dimensional model and other assumption approximation at $T_1=24h$

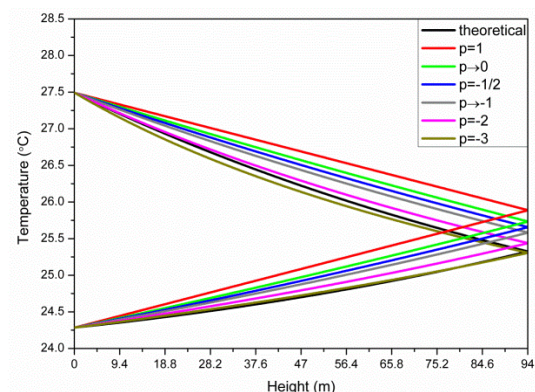


Figure 3 Comparison of temperature profile from quasi-three-dimensional model and other assumption approximation at $T_1=48h$

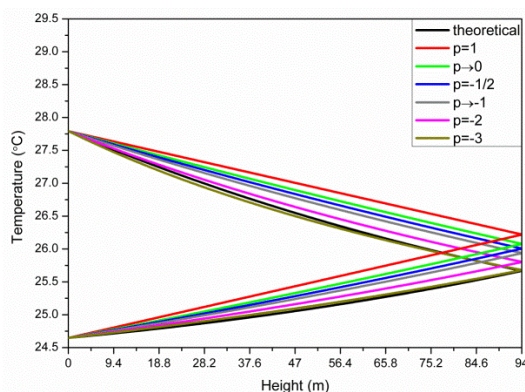


Figure 4 Comparison of temperature profile from quasi-three-dimensional model and other assumption approximation at T1=70h

Table 4 Recommend weight by curve fitting method

Time	T1=24	T2=48	T3=70
h	-	-	-
p value	2.2368	2.6698	2.8969

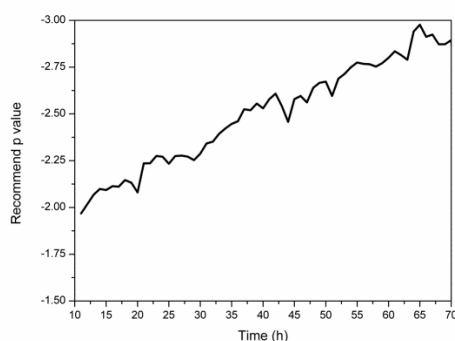


Figure 5 the recommended p value changes as the times flow

4. CONCLUSION AND FUTURE WORK

An on-site Thermal Response Test was well performed in Changsha in this paper, the thermal properties, such as effective ground thermal conductivity, thermal diffusivity and borehole thermal resistance, also have been evaluated under various average assumption method. The comparison results between the theoretical vertical temperature profile based on the quasi-three-dimensional models and various recommended average methods are obtained. In this paper, some conclusions have been made as follows,

The arithmetic average fluid temperature will lead to an over estimation of the borehole thermal resistance and the effective ground thermal conductivity is in an acceptable condition.

The recommended weight of p value which is defined by Marcotte and Pasquier is not always stay constant, it is various as the time changes, and varies with different TRT.

In this TRT in Hunan University, the recommended p value is not the same as previous work's suggestion,

the harmonic average assumption performs the best, and it have a 7.70% reduction in borehole thermal resistance comparing to the arithmetic average method.

However, there are a lot of work should be completed in the coming future. As the roughly analysis about the p value, we know that the weight of p value changed with different case and different time. The relationship between the geometric arrangement of the borehole, the ground properties and the weight of the p value should be analysis in the future work, and an analytical or empirical formula should be proposed.

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SIMULATION STUDY OF A STANDALONE HYBRID SOLAR-WIND SYSTEM WITH PUMPED STORAGE

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ABSTRACT

In this study, the pumped hydro storage (PHS) was introduced for microgrid hybrid solar-wind systems. It explores a new solution for the energy storage serving for the intermittent solar and wind energy systems. This research developed a simple mathematical model for the system, and the applied the proposed system a remote island in Hong Kong, which requires on average 200kWh/day electrical load. The simulation results indicate that the PHS can effectively compensate the intermittent nature of the renewables, although the overall system energy efficiency is low. However, a reliable and environmentally friendly power supply can be continuously provided by the PHS based renewable energy system, demonstrating that PHS is one of the ideal partners for renewable integration, especially for standalone applications.

Keywords: Hybrid solar-wind system, Pumped storage, Simulation.

1. INTRODUCTION

Currently, more than 1.5 billion inhabitants worldwide including 5 million people in China still lack access to electricity, most live in the small remote villages or isolated islands far away from the national utility grid (IEA 2006; WHO 2009). In such areas the population is dispersed and the terrain rugged, hence utility grid extension to these locations is not only impractical for these reasons but also because of the prohibitive capital cost. Constructing long transmission and distribution lines to such comparatively inaccessible places is thus both impractical and uneconomical (Hansen 1998; Chakrabarti and Chakrabarti 2002; Qoaidar and Steinbrecht 2010; Raman, Murali et al. 2012). As a result, the electrical demand is commonly powered by diesel generators or even no power is supplied (Sopian, Zaharim et al. 2008; Lau, Yousof et al. 2010; Bekele and Tadesse 2012). Low investment costs and lack of a feasible alternative are the main reasons behind the widespread use of diesel generator systems. The significant rise in diesel price and subsequent environment pollution since the oil crises of the 1970s (Elhadidy and Shaahid 2000), however, now make the further employment of these systems expensive and hence less favourable. Additionally, the negative environmental effects resulting from the employment of diesel, noise and water, soil and air pollution, damages the local ecological system (Qoaidar and Steinbrecht 2010).

Power supply for places with limited access has drawn extensive public attention, particularly after investigation into the viability of natural sources of energy, such as photovoltaics (PV), wind, micro hydro,

biomass or their hybrids. The standalone renewable energy systems (RES), have a reputation for being inexhaustible, environmentally benign, reliable, efficient, and with cost-effective characteristics, and are hence seen to be the most likely viable energy supply solution for such areas. In addition, renewable energy is a promising key to solve the problems of global environmental pollution and hence reduce the subsequent negative impact on worldwide prosperity due to the depletion of such as nonrenewable resources and also the negative impact on human health caused by the ensuing atmospheric degradation (Himri, Boudghene Stambouli et al. 2008; Shaahid and El-Amin 2009). An awareness of the above has caused the last decade to witness a dramatic expansion in the use of RES as replacements for fossil-based energy.

Technical renovation and the rapid development of PV systems, high initial cost is no longer the key barrier to their usage. However, the intermittent and unpredictable characteristic of solar energy availability prevents solar power from being fully reliable and economically viable without a suitable energy storage subsystem/capability complementing a standalone application. At present, this is the most challenge for remote renewable energy applications.

Standalone RESs usually employ rechargeable lead-acid batteries to store excess electricity. However, batteries have well known limitations: their initial capacity costs are too high; their lifespans are relatively short, usually 2-8 years, and frequent replacement imposes an additional financial burden; In addition, they incur environmental pollution in containing toxic substances such as lead, which can cause problems during shipment, installation, and particularly in

ultimate disposal; and they tend to be dangerous with sulfuric acid and in the possibility of explosion.

Potential storage techniques for microgrid RESs have been extensively studied by researchers in recent years, among which pumped hydro storage is considered to be the most promising alternative to batteries. Pumped storage, has been utilized in association with thermal and nuclear power plants for the past hundred years, and remains the most commonly used and commercially viable electricity storage technology due to its low cost and maturity. The total installed pumped storage is over 127GW worldwide, accounting for 99% of all storage capacity, according to a report by the USA Electric Power Research Institute in 2010(Rastler December, 2010. USA). The pumped storage system, based on two vertically-separated water reservoirs, stores potential energy by using low-cost off-peak electricity to pump water from the lower reservoir to an upper reservoir. During the periods of peak load demand, stored water is released to flow downhill through a turbine to redeliver power. In this way it can recover almost 76%–85% of the power used in pumping the water(Rastler December, 2010. USA).

The large scale renewable energy grid-connection system is becoming increasingly popular, but stability of an electricity network and the utilization safety of terminal users can be greatly affected once the PV share exceeds 20%(Chen, Zhang et al. 2012) or the PV capacity over 800 MW(Gardner, Mcgoldrick et al.). The study (Weisser and Garcia 2005) also suggested that for larger autonomous systems, a rational penetration limit for renewables should be kept lower than 35%. The main reason is that the output from PV fluctuates greatly, bringing real challenges in voltage regulation. The pumped storage system, the only proven technology for large scale RES integration, can potentially control sags in voltage and alleviate the effects of rapid swings on the distribution grid.

To mitigate the imbalance between power production and load demand, pumped storage power plant was proposed in (Aihara, Yokoyama et al. 2010; Aihara, Yokoyama et al. 2011), to effectively solve the power supply reliability problems associated with excess PV power. With the same objective, a wind-solar and pumped-storage hybrid power supply system has also been studied(Ruisheng, Bingxin et al. 2010). The papers (Ren and Li 2011; Yan, Yuan Zheng et al. 2011) have established simple models of the main components of a hybrid wind-solar-pumped-storage power system. In addition, substantial studies have been conducted on pumped storage systems to increase the penetration level of wind power in isolated power systems (Katsaprakakis, Christakis et al. 2011; Padrón, Medina et al. 2011; Tuohy and O'Malley 2011; Varkani, Daraeepour et al. 2011; Caralis, Papantonis et al. 2012).

To date there is little literature reporting real applications of pumped storage in small to medium sized RESs, even though the subject has attracted the attention of many scholars. Manolakos et al. (Manolakos, Papadakis et al. 2004) implemented a pumped water storage system to partially replace the battery storage system in a practical stand-alone 18

kWp PV plant in Greece. The preliminary results showed that the efficiency of the system was low at about 28% and accompanied by a high capital cost, while the social impacts, such as both electricity and water supply, would be significant. However, a single project example alone cannot fully reflect all possible operating conditions and more experiments and real demonstration projects are of great importance.

This study is a subpart of a research project conducted in collaboration with a utility company in Hong Kong. The rest of paper is organized as follows: Section II and III introduce the physical and mathematical model of this system, respectively. A case study based on the proposed model is carried out in Section IV. The results are presented and analyzed in Section V. Finally, conclusions are drawn in Section VI.

2. DESCRIPTION OF THE SYSTEM AND ITS OPERATION PRINCIPLE

The proposed stand-alone hybrid solar and wind system with pumped storage is illustrated in Figure 6, using the pumped storage subsystem as the main energy storage device. The local solar energy matches well with the load demand and a microgrid system using solar PV panels to convert sunlight for power generation. The wind energy output is used to produce power in the night and rainy days. An inverter transforms the output DC to supply 220V AC load via the power distribution network while any surplus power is used to charge the energy storage subsystem. A pump-turbine set is used as the energy storage device, usually operated in generating mode in the daytime and in pumping mode at night. The entire system would be regulated by a control centre to manage the whole microgrid system.

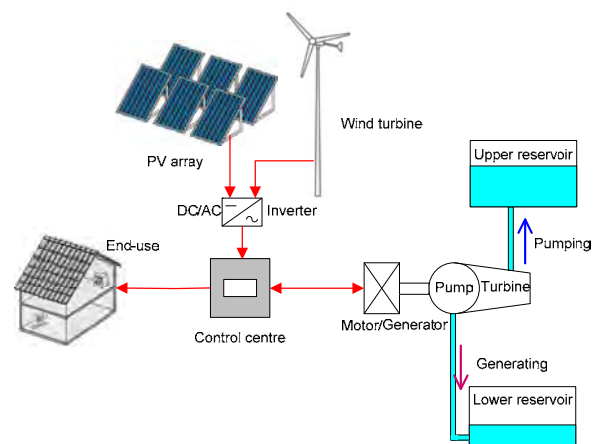


Figure 6 Hybrid solar-wind system with PHS

The operating principle of the hybrid solar/wind/pumped storage system can be briefly described as follows: The pumped storage system elevates water to the upper reservoir using solar pumps when PV power is available and greater than the load demand. The stored water is then allowed to flow back

to the lower reservoir enabling the production of electricity by a turbine/generator during periods of high electrical demand or lower solar energy availability. In this way, a sustained energy supply from the hybrid system would be guaranteed throughout day and night. This activity will suffer some conversion losses but the electric network can be stabilized and the related problems from batteries can be solved

3. MODELLING

To predict system operating performance, individual components should be modeled first and their combinations can then be better evaluated.

3.1 Modelling of the PV array

The mathematical model of the PV array can be described as (five-parameter model):

$$I = N_p I_{ph} - N_p I_0 \left(e^{\frac{1}{V_t} \left(\frac{V}{N_s} + \frac{I}{N_p} R_s \right)} - 1 \right) - \frac{N_p}{R_p} \left(\frac{V}{N_s} + \frac{I}{N_p} R_s \right) \quad (2)$$

where I_{ph} is the photo current (A); I_0 is the diode saturation current (A); R_s is the series resistance (Ω); R_p is the shunt/parallel resistance (Ω); $V_t = nKT / q$ is the diode thermal voltage (V); n is the diode ideality factor; T is the cell temperature (K); K is the Boltzmann's constant ($1.381 \times 10^{-23} \text{ J/K}$); q is the electron charge ($1.602 \times 10^{-19} \text{ C}$); N_s is the number of PV cells in series for the studied array and N_p is the number of PV module in parallel.

The power output from the PV generator can be expressed by (HOMER):

$$P_{PV}(t) = Y_{PV} \cdot f_{PV} \cdot \frac{\bar{G}(t)}{G_{STC}} \cdot [1 + \alpha_p \cdot (T(t) - T_{STC})] \quad (3)$$

3.2 Modelling of the wind turbine

Through a through literature review, it was found that the following equation was mostly used to describe the power output of a WT (characteristic curve):

$$P_{WT}(v) = \begin{cases} P_R \frac{v^k - v_c^k}{v_R^k - v_c^k} & (v_c \leq v \leq v_R) \\ P_R & (v_R \leq v \leq v_F) \\ 0 & (v < v_c \text{ and } v > v_R) \end{cases} \quad (4)$$

where P_R is the rated electrical power; v_c is the cut-in wind speed (the wind speed at which the turbine starts to generate usable power); v_R is the rated wind speed; v_F is the cut-off wind speed (the speed at which the turbine hits the limit of its alternator and can no longer put out increased power output with further increases in wind speed); k is the Weibull shape parameter.

3.3 Modelling of the energy storage subsystem

The model for the pump turbine set (PTS) has been well developed in (Schoppe 2010; Li, Yang et al. 2012; Tao, Chengzhi et al. September 2-5, 2012).

The water flow rate sucked from the lower reservoir by the pumps is expressed as:

$$q_p(t) = \frac{\eta_p \cdot P_{h \rightarrow p}(t)}{\rho g h} = c_p \cdot P_{h \rightarrow p}(t) \quad (5)$$

where c_p is the water pumping coefficient of the pump motor unit (m^3/kWh);

When the PHS system is under the discharging model, the output from the water turbine generator unit is:

$$P_t(t) = \eta_t \rho g h \cdot q_t(t) = c_t \cdot q_t(t) \quad (6)$$

where c_t is the turbine generating coefficient (kWh/m^3).

A. Upper reservoir(UR)

The volume of water stored in the upper reservoir (i.e. the state of charge of the storage tank) should be adequate to meet the power consumption of this island for the several expected days of autonomy. The gravitational potential energy stored in the UR can be derived from the following equation:

$$E_c = n_{day} \cdot E_{load} = \eta_t \cdot \rho \cdot V \cdot g \cdot h \quad (7)$$

where E_c is the energy storage capacity of a water reservoir (Joules); n_{day} is the number of days of autonomy; E_{load} is the daily load energy consumed; V is the volume or storage capacity of the water reservoir (m^3).

The total quantity of water stored in the UR at any time t is determined by:

$$Q_{HR}(t) = Q_{HR}(t-1)(1-\alpha) + \int_{t-1}^t q_p(t)dt - \int_{t-1}^t q_t(t)dt \quad (8)$$

where α is the evaporation and leakage loss. It can be compared to the self-discharge of a battery bank. For simplification, this study has ignored these losses in the above equations.

4. SIMULATION STUDY

The case study regarded a small remote island far away from the power grid in Hong Kong. The hybrid system was implemented to provide the power supply and energy storage for the local residents of about 100 peoples.

4.1 Model inputs

Load demand profile

Currently, this island is powered by a diesel generator and a small PV plant (20kWp). The daily and seasonal load profile is illustrated in Figure 7.

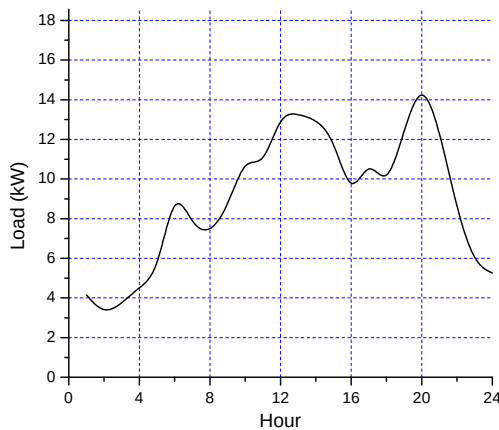


Figure 7 A sample of 24-hour load profile for this island

Renewable energy resource

The meteorological data on this island was collected by Hong Kong Observatory, including solar radiation, wind speed and ambient temperature. The solar radiation and wind speed distribution are presented in Figure 8. Their hourly weather data was synchronized through the year of 2009.

The polycrystalline panel (model: STP200-18/Ub-1) from Suntech Ltd. was employed in this study. The wind turbine Proven 11 (aka KW6) from Kingspan Renewables Ltd. was employed in this study. The solar pump efficiency is bit low in this project at 50%, and the efficiency of the turbine from Ebara Japan is 65%. The water heat was assumed at 60m in this study. Therefore, the overall energy storage system efficiency is 32.5%. The water pumping coefficient and the turbine generating coefficient is 3.06m³/kWh and 0.106kWh/m³.

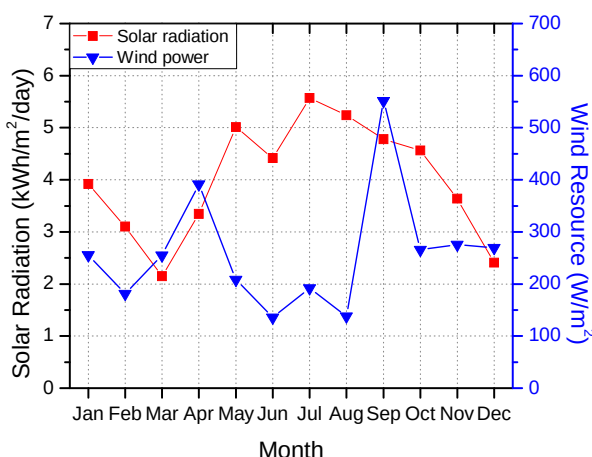


Figure 8 The monthly solar radiation and Wind speed distribution at the island

System Components information

4.2 Evaluation criteria

In this study, the loss of load probability (LPSP) is employed to evaluate whether a system is able to cater for the load demand and what is the time percentage with insufficient power. This index is widely used by the literature (Yang, Lu et al. 2003; Yang, Lu et al. 2007), defined as the total power supply failure hours divided by the number of sample hours over the studied period.

$$LPSP = \frac{\sum_{t=1}^{8760} \text{hours}[(P_{PV_L}(t) + P_{V_{WT_L}}(t) + P_t(t)) < P_L(t)]}{8760} \quad (9)$$

The LPSP index not only can help to size the generator (PV and wind turbine) and UR capacity, but also it can be used to assess the reliability of a specific system.

5. RESULTS AND DISCUSSION

5.1 Overall results

Along with the WT number varying from zero to four and PV size from 70 to 150 kWp at an interval of 10 kWp, 34 cases in total were simulated in this study. In these cases, the LPSP was set at zero, meaning no power supply failure, and the initial water in the UR was assumed equal to the maximum capacity. Based on the simulation program, the required minimum UR capacity can be derived for each combination of PV and wind turbine. For example, if the number of wind turbine is two and PV size is 120kWp, the required minimum UR size was calculated at 3,614 m³ in theory.

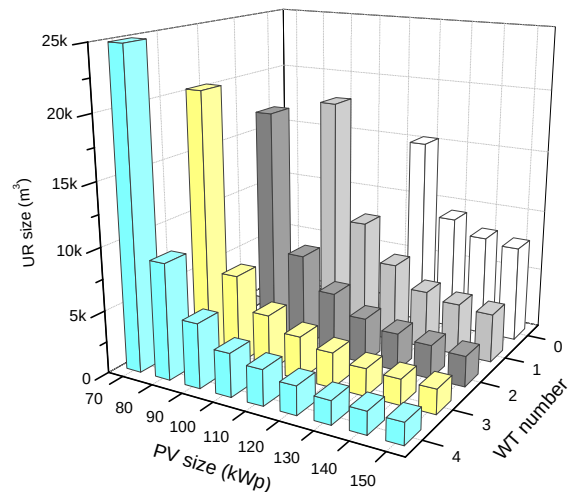


Figure 9 Overall simulation result

The overall results are presented in Figure 9. For the first scenario, no wind turbine was taken into account, i.e. this is a PV-only system with micro pumped storage. The lowest UR capacity (7750 m³) can be achieved when the PV size is 150 kW. The UR capacity increases gradually as the PV size decreases. For the PV at 120kW, the UR soars up to 15316 m³. If

PV size is further decreased, the UR will be over 25000 m³.

For the second scenario, one WT was considered. Due to the complementary characteristic between solar and wind energy, the required UR size can be reduced greatly. For the same PV size, about 50% of UR size could be abated when compared with the PV only system. Similar trends can be found for the wind turbine number at two to four. The more the wind turbine number, the more choices for the PV capacity. For instance, 70kW PV with four wind turbines can also achieve the zero power supply target, however, the UR reach to almost 25000 m³ for this case.

The minimum UR capacity required to achieve 0% LPSP can be plotted to generate the sizing curve. As shown in Figure 10, similar characteristics can be found between the sizing curves of different system configurations (one to four wind turbines). For the scenario with the same number of wind turbine, an increase of PV can result in UR size reduction, significantly in the beginning and less obviously after a certain point (inflection/critical point). On the other hand, for the same size of PV module, the increase of wind turbine number can also decrease the UR size. The curves in Figure 10 also shows that the space over the each sizing curve is the feasible design space for each scenario, which also means the space below is the infeasible region. If the economic performance curve (such as minimum cost of energy) was also combined into this figure, then the optimum system configuration can be found easily.

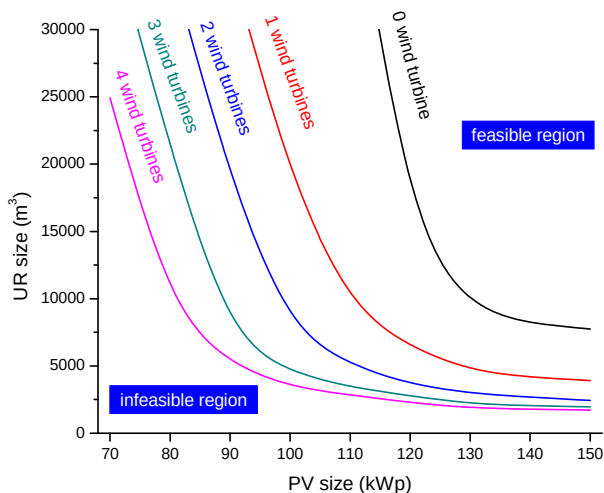


Figure 10 sizing curve for pumped storage based renewable energy system (design space)

The wind turbine only system with the UR of 20000 m³ was also studied. As illustrated in Figure 11, the LPSP value decrease from 17.4% to 0 %, when wind turbine number increases from 10 to 60. A significant improvement of the power supply reliability can be observed when the turbines changes from 10 to 20. Further increase in wind turbine number can also reduce power supply failure but less effectively. It can be seen that without PV panels a great deal of wind

turbines are required to have a zero LPSP power supply system.

Since the wind turbine output pattern cannot follow the daily and season load consumption profile, the excess power also grows with wind turbine increase. Up to 87.5% (2044kWh/day) of the electricity production is wasted for the case of 60 wind turbines due to the limited energy storage capacity, the excess energy figure is much higher than the energy required. From this point of view, the energy storage system is more important in the wind turbine only system, which can shift the excess energy during low demand period to cover the load when the demand is high.

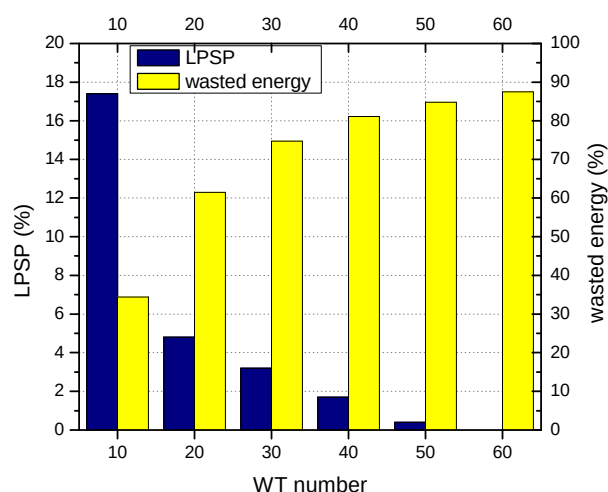


Figure 11 LPSP and excess energy percentage of the wind turbine only system

6. CONCLUSION

In this study, a hybrid solar and wind system with pumped storage was investigated in simulation. A mathematical model for this system was successfully developed and the basic criterion was discussed for assessing the system performance. The proposed system was applied in a case study for a remote island in Hong Kong, 34 cases in total were studied and their performance was then evaluated, which demonstrates that the effectiveness of the proposed system. The overall result and sizing curve shows that for same number of wind turbine, an increase of PV can result in UR size reduction, significantly in the beginning and less obviously after a critical point. In addition, complementary characteristic between solar and wind energy output was presented in this study.

The overall efficiency of this pumped storage scheme was only 32.5%. Such a low efficiency might result from the conversions losses of the micro pump-turbine set. The system performance could be potentially enhanced by introducing superior solar pumps or a small battery bank. This study demonstrates that PHS is the ideal partners for the standalone renewable energy power supply systems.

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THE RESEARCH ON THE INSULATION LAYER THICKNESS OF TRANSMISSION AND DISTRIBUTION NETWORK IN SMART ENERGY GRID

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ABSTRACT

Based on the transmission and distribution network of cold and hot logistics, the constraint condition of fixed allowable total cooling loss is proposed, this paper study the synergic relation between the insulation layer thickness and distribution distance, and the network economic characteristics of different insulation materials. Verified as an engineering example, it is shown that, under certain constraints, there exists an optimal synergic relation between insulation layer thickness and transmission distance for any insulation materials. This study provides theoretical guidance and research ideas for the research of optimal design of transmission and distribution network and the selection and distribution of product and storage energy systems in smart building energy grid.

Keywords: Smart building energy grid, optimal design of transmission and distribution network, the optimization of the thickness of insulation layer, synergistic effect

1. BACKGROUND

For the deficiency of low efficiency of comprehensive energy use, high investment cost of transmission and distribution, poor reliability and flexibility of power supply and easy to cause environmental pollutions in the conventional energy supply way of large power grid, central heating etc, this paper proposes a smart building energy grid based on natural gas (NDES-SG) in the early research. NDES-SG combines the natural gas distributed energy systems, advanced energy storage systems as well as building energy consumption monitoring system through the multi-layer smart grid and forms building energy production, storage, transport, energy four-in-one within the scope of city, to maximize the energy utilization.

The key scientific problems in NDES-SG including: 1) energy generation systems--the basic principles of distribution, system capacity configuration, system integration mechanism, variable conditions operating characteristics of natural gas distributed energy systems(NDES);2) energy storage systems -- the capacity configuration, distribution principles, the integration of energy storage/release characteristics and the integrated system, in the advanced and medium-sized cold thermal energy storage system;3) the transmission and distribution systems -the optimization and dissipation analysis of cold, heat and electricity transmission;4) building energy consumption systems - the real-time monitoring and forecast of energy consumption, and information

feedback in building groups;5) intelligent control system -- sharing and scheduling control of the information of energy production, storage, transport, and consumption. Through the optimization analysis of the thickness of variety thermal insulation materials, this paper proposes a design principle of the fixed total cold loss and analyses the design thickness of different thermal insulation materials, considering the diversity of cooling (heating) transmission loss caused by different transport distance in grid energy network, and provides a reference for the pipeline network optimization, to achieve the highest efficiency of total energy system with certain constraints, while it has a great significance to provide reference for the distribution of natural gas distributed energy systems and energy storage systems.

2.THE KEY FACTORS OF ENERGY DISSIPATION OF COLD AND HOT LOGISTICS NETWORK

In NDES-SG, the energy supply distance (energy supply radius) has a great impact on the efficiency of total energy transport systems. It is closely related to energy efficiency and the distribution of energy production and storage systems. So design and optimization of cold, heat, and electricity transmission and distribution grid is an important research in NDES-SG.

Cold and hot energy dissipation is the key issues on optimization and design of hot and cold transmission and distribution grid, the supply and return temperature of water, pipe diameter, pipe spacing, pipeline depth,

the thermal conductivity, thickness, transport distance of the insulation material, and surrounding environment etc have varying degrees impact on cold and hot loss in the delivery system. Yanjie Li of Harbin Institute of Technology insists that the main factor of heat loss is the structure of insulation layer, followed by thermal conductivity of soil, and its resistance have greater impact on low-temperature heat medium and large diameter pipe. Huan Zhang etc [3] of Tongji University analysis the unit length cold loss of directly-buried pipe by Antipodal method and obtain the relationship between the cold loss and the thermal conductivity, thickness of the insulation material layer, and the temperature, thermal conductivity of soil.

By analyzing the review of relevant literature, it can be known that whether to take appropriate insulation measures plays an important role on cold and hot energy dissipation, and selection of materials and thickness of insulation is the key point.

3.OPTIMIZATION ANALYSIS OF THE THICKNESS OF INSULATING LAYER IN COLD AND HOT LOGISTIC NETWORK

3.1 Analysis method of cold energy loss in pipeline network

Firstly, assuming a cold energy dissipation calculation model in cold pipeline network using cold energy dissipation of directly buried single pipe as the theoretical analysis model. Illustration of pipeline is shown in figure 1.

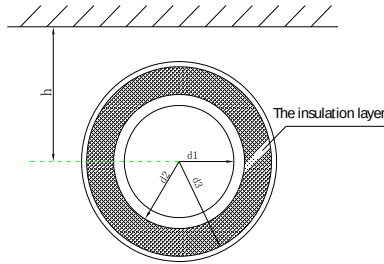


Fig.1 Schematic diagram of cooling pipe structure

Based on the basic principle of heat transfer, heat transfer thermal resistance of the single pipe includes steel pipe thermal resistance, insulating layer thermal resistance, external stent thermal resistance and soil thermal resistance, in which external stent thermal resistance is relatively small [4], so it is often neglected.

The heat transfer thermal resistance of each part is calculated as follows [5]:

$$R_1 = \frac{\ln \frac{d_2}{d_1}}{\lambda_1} \quad (1)$$

Where R_1 is steel pipe thermal resistance, $\text{m}^2\text{K/W}$; d_1 , d_2 is the inner and outer diameter of the

steel pipe respectively, m ; λ_1 is thermal conductivity of steel pipe, $\text{W/(m}^2\text{K)}$.

$$R_0 = \frac{\ln \frac{d_3}{d_2}}{\lambda_0} \quad (2)$$

Where R_0 is insulating layer thermal resistance, $\text{m}^2\text{K/W}$; d_3 is the outer diameter of the insulating pipe, m ; λ_0 is thermal conductivity of insulating layer, $\text{W/(m}^2\text{K)}$.

$$R_{soi} = \frac{\ln \frac{h}{d_3}}{2\pi\lambda_{soi}} \quad (3)$$

Where R_{soi} is soil thermal resistance, $\text{m}^2\text{K/W}$; h is the buried depth of pipe, that is the distance between the center of buried pipe and the ground, m ; λ_{soi} is thermal conductivity of soil, $\text{W/(m}^2\text{K)}$.

The total thermal resistance of buried cold water pipeline is given as: $R_{cool} = R_1 + R_0 + R_{soi}$

Cooling transmission losses formula [5-6] is given as:

$$\Delta Q_c = \frac{(t_{c,soi} - t_{c,s}) + (t_{c,soi} - t_{c,r})}{\frac{\ln \frac{d_2}{d_1}}{2\pi\lambda_1} + \frac{\ln \frac{d_3}{d_2}}{2\pi\lambda_0} + \frac{\ln \frac{h}{d_3}}{2\pi\lambda_{soi}}} L \quad (4)$$

Where $t_{c,soi}$ is average temperature of soil, $^{\circ}\text{C}$; $t_{c,s}$, $t_{c,r}$ is the average temperature of supply and return water in pipe respectively, $^{\circ}\text{C}$; L is length of cold water pipeline, m .

3.2 Analysis method of heat loss in pipeline network

The heat transport loss of heat supply pipe is calculated as follows [7]:

$$\Delta Q_{h,s} = \frac{(t_{h,s} - t_{soi,w})(R_{h,r} + R_{soi}) - (t_{h,r} - t_{soi,w})R_a}{(R_{h,s} + R_{soi})(R_{h,r} + R_{soi}) - R_a^2} L \quad (5)$$

$$\Delta Q_{h,r} = \frac{(t_{h,r} - t_{soi,w})(R_{h,r} + R_{soi}) - (t_{h,s} - t_{soi,w})R_a}{(R_{h,s} + R_{soi})(R_{h,r} + R_{soi}) - R_a^2} L \quad (6)$$

Where, $\Delta Q_{h,s}$ and $\Delta Q_{h,r}$ is the heat transport loss of supply and return water pipe respectively, W ; $t_{h,s}$ and $t_{h,r}$ is the temperature of hot water in supply and return water pipe, $^{\circ}\text{C}$; $t_{soi,w}$ is the temperature of the ground, $^{\circ}\text{C}$; $R_{h,r}$ and $R_{h,s}$ is the insulating layer thermal resistance of supply and return water pipe, $\text{m}^2\text{K/W}$;

Due to a large temperature difference of fluid between supply water pipe and return water pipe, so it adds a thermal resistance R_a in consideration of the additional thermal resistance about the heat transfer influence of supply and return water pipe, $\text{m}^2\text{K/W}$.

$$R_{h,s} = R_{h,r} = \frac{1}{2\pi\lambda_0} \ln \frac{d_3}{d_2} \quad (7)$$

$$R_a = \frac{1}{2\pi\lambda_{soi}} \ln \sqrt{\left(\frac{2H}{r}\right)^2 - 1} \quad (8)$$

$$H = h + \frac{\lambda_{soi}}{\alpha_{soi}} \quad (9)$$

Where r is the distance of supply and return pipe center lines, m; H is conversion buried depth, m; α_{soi} is soil heat transfer coefficient, W/(m²·K), it takes 12~15 W/(m²·K).

3.2 The optimization principle of the insulating layer thickness

To decrease the heat loss of pipeline network and to meet the relevant need, It is necessary to calculate the insulating layer thickness. Usually, we use the cost-efficient thickness method or the method of the temperature drop (rise) per unit length. It is the cost-efficient thickness method, considering two factors that are the investment cost of the insulating material of pipe and the investment cost of the heat loss of pipe. Then, it gets a reasonable insulating thickness, when the annual operating cost achieves the minimum value. The insulating thickness doesn't change with the length, when calculating with the method of the temperature drop (rise) per unit length. Therefore, the heat loss increases with the increases of distance. Usually, the loss ratio of transportation cooling is less than transportation heat. So this paper determines the constraints of the total loss of cooling firstly (i.e. $\Delta Q_c < (1 - \eta) Q_c$), and analyses the cooperative relationship between transportation distance and the insulating thickness. Generally speaking, the rising of the temperature of cooling water is caused by electricity consumption of water pump and the thermal exchange of pipe. This paper only considers the thermal exchange loss of pipe, and the percentage is 1%.

$$1 - \eta = \frac{\Delta Q_c}{Q_c} \quad (10)$$

Where η is transport efficiency; Q_c is cooling load, W.

According to equation (4) and (5), we get:

$$d_3 = \left\{ d_2^{\frac{1}{\lambda_0}} h^{\frac{1}{\lambda_{soi}}} \exp \left[2\pi \frac{(2t_{c,soi} - t_{c,s} - t_{c,r})}{(1 - \eta)Q_c} L - \frac{\ln \frac{d_2}{d_1}}{\lambda_0} \right] \right\}^{\frac{\lambda_0 \lambda_{soi}}{\lambda_{soi} - \lambda_0}} \quad (11)$$

$$\delta = \frac{d_3 - d_2}{2} \quad (12)$$

Where δ is the insulating thickness, m

In the perspective of economic calculation, investment costs of insulating layer are given as:

$$M = price \times \pi (d_3 - \delta_0) \delta L \quad (13)$$

Where M is investment cost of the insulating layer, Yuan; price is the per unit price of the insulating layer, Yuan/m³.

Compared with the case that the insulating layer doesn't change with the distance, the increased investment cost of the insulating layer is given as:

$$M_{add} = M - price \times \pi \times (d_{3,0} - \delta_0) \delta_0 L \quad (14)$$

Annual depreciation expense is given as: $M_{annual} = \alpha M_{add}$, where α is the ratio of annual depreciation expense, $\alpha = 0.15$; $d_{3,0}$ and δ_0 is the external diameter and thickness of the insulation layer of reference system respectively;

As the transportation distance increasing, compared with the case that the thickness of the insulating layer doesn't change, when the insulating layer is increased, the decreased loss of the cooling load is calculated as follows:

$$\Delta Q_{save} = \frac{(t_{c,soi} - t_{c,s}) + (t_{c,soi} - t_{c,r})}{\ln \frac{d_2}{d_1} + \frac{\ln \frac{d_{3,0}}{d_2}}{2\pi\lambda_1} + \frac{\ln \frac{h}{d_{3,0}}}{2\pi\lambda_{soi}}} L - \Delta Q_c \quad (15)$$

Convert to the loss of the power consumption is given as:

$$M_{save} = price_e \times \Delta Q_{save} \times t / (10^3 \times COP_{zs}) \quad (16)$$

Where $price_e$ is electrovalence, Yuan/(KW·h); t is annual cooling hours, h; COP_{zs} is the equivalent cooling of COP .

4. CASE STUDY

4.1 Boundary conditions in Case Study

This paper takes an engineering example for calculation and analysis, using a CCHP to satisfy the cooling, heating and power demand of a 19200m² scientific research office building, where the cooling load is 1766KW, the heating load is 1348KW, and the electrical load is 788 KW. Gas turbine is used to produce power, which connects the waste heat for lithium bromide absorption conditioning units for cooling and heating. The key parameters of pipe network transmission and distribution energy loss is shown in Table 1.

Table 1 The key parameters of pipe network transmission and distribution energy loss

The supply and return temperature of cold water/°C	5/11	Thermal conductivity of soil/(W/(m·K))	1.5
Cold water flow rate/(m ³ /h)	250	The average soil temperature in summer/°C	20.6
The supply and return temperature of hot water/°C	95/70	The average soil temperature in winter/°C	10
Hot water flow rate/(m ³ /h)	50.3	Isobaric specific heat capacity of water/(kJ/kg·°C)	4.2
The buried depth of pipeline/m	1.5	Electricity price/(yuan/(KW·h))	0.7
The inner diameter of steel pipe/m	0.207	COP _{zs}	1.4

The outer diameter of steel pipe/m	0.219	α	0.15
Thermal conductivity of steel pipe/(W/ (m·K))	45	Cooling hours t/h	1500
Pipe spacing of supply and return water r/m	1	α_{soi} (W/ (m ² ·K)	15

4.2 Results and Discussions

T The transport distance is 950 m in actual engineering, and at the same time the glass cotton is used as thermal insulation material, where $\lambda_0 = 0.04$ W/ (m·K) , $\delta = 45$ mm, $d_3 = 0.309$ m respectively. According to formula (1)~(4), the dissipation of the pipe $\Delta Q_c = 15.6$ KW . In this case, there are three kinds of insulation materials, polyurethane (950yuan/m³), glass wool (300yuan/m³) and polystyrene (150yuan/m³). The thickness and investment cost of different insulating layer materials are shown in figure 2-3, where for polyurethane $\lambda_0 = 0.02$ W/ (m·K) and for polystyrene $\lambda_0 = 0.067$ W/ (m·K)

Known by the curve in Figure 2, when the allowable value of total cooling is the constraint condition, the thickness of insulating layer also increases with a non-linear trend as the conveying distance varying, and at the same time the thickness increase rate is gradually increased. The more the thermal conductivity (thermal resistance) of insulation materials are, the more the thickness of insulation layer is, and the increasing rate of the insulation layer thickness is greater, the tendency of non-linear is more obvious.

When the transmission radius of pipe network $3100 \leq L \leq 5000$ m, the increasing trend between the thickness of three insulation layer materials (polyurethane, glass wool, polystyrene) and the conveying distance is nearly exponential power .Using polyurethane as insulation material when $L \geq 3300$ m then $\delta > 100$ mm; Using glass cotton as insulation material when $L \geq 1700$ m then $\delta > 100$ mm, Using polystyrene as insulation material when $L \geq 1100$ m then $\delta > 100$ mm. According to the design standard of energy efficiency, when the nominal diameter of pipe is between 125mm and 250mm, the recommended value of the cold water pipe insulation is 45 mm. When $\delta > 100$ mm, it has exceeded the conventional value of pipeline corresponding to the external diameter of steel pipe, which will bring inconvenience packaging of pipeline and construction.

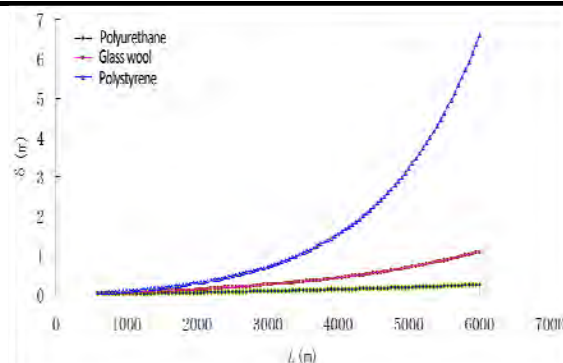


Figure 2 The changing chart of the thickness of different insulating layer materials varying with the conveying distance

Therefore, when we calculate the thickness of insulation layer using the principle of allowing cold loss, we have to take some factors such as distance, coefficient of thermal conductivity of insulation layer and construction feasibility into account.

In figure3, under the determination of cold loss and the diameter of steel pipes, the initial investment costs of insulation materials of pipes are related to the thickness of the insulation layer, Section length and the unit price, showing a trend of nonlinear increase along with the increase of section length and the growth rate of costs.

When the radius of the transmission and distribution network $L \geq 2050$ m, the investment costs of materials increased sharply, leading to the high cost of the pipe network system and the economy deteriorated sharply. Comparing the three insulation materials, we see that when $L < 700$ m, the investment cost of polystyrene insulation is minimum, which has a great relationship with its low unit price; when $L \geq 700$ m, the economic performance of polystyrene is inferior to glass wool, when $L \geq 1350$ m, the material cost of polystyrene is higher than polyurethane, which is due to that the polystyrene insulation thickness increasing sharply with the conveying distance longer; when $L < 2100$ m, the economic performance of glass wool is better than polyurethane, though the insulation layer thickness of polyurethane is less than glass wool, the high unit price is a greater impact on the investment costs; when $L \geq 2100$ m, the investment costs of polyurethane is lowest because its insulation layer thickness is minimum and the increase rate is slowest.

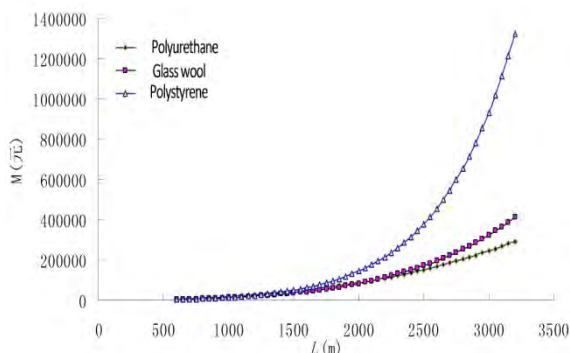


Figure3 The investment costs of insulation materials of pipes varying with the transportation distance

This proves that we can consider polystyrene and glass wool as the insulation material for short-distance transport and polyurethane for long-distance transport because it has better economic performance.

According to formula (13) to (15), which uses the insulation layer design at 950m of the three materials as a reference object for cost comparison. The relationship of the increased insulation layer investment in depreciation charges and reduced cold loss discounted fee and the transport distance is shown in Figure 4.

Seen from Figure 4, the increased investment costs and the reduced cold loss costs of the three insulation materials show the trend of nearly exponential increasing rapidly with the transport distance, the economy deteriorating sharply, the rise rate of investment costs of the materials increasing acceleratly, so do with the increase of thermal conductivity of the insulation material. When $L \leq 1450\text{m}$, the increased investment in years discounted fee of glass wool is minimum; When $L > 1450\text{m}$, the increased investment costs of polyurethane is minimum, polystyrene is at the highest level, which is due to the increase rate of insulation layer thickness sharply. Comparing the return on investment of the three insulation materials, when $L \leq 1800\text{m}$, the increased investment costs of polystyrene is less than the benefits of the reduced cold loss, which have contributed to the energy-saving rate and economy of the conveyor system; while when $L \geq 1800\text{m}$, the increased investment costs for thicker insulation layer is far greater than the reduced cold loss discounted fee and the conveyance distance is longer, the difference is greater; When $L > 2400\text{m}$, the increased investment costs of glass wool is over the cold loss; when $L > 2750\text{m}$, the increased investment costs of polyurethane is greater than the benefits of cold loss.

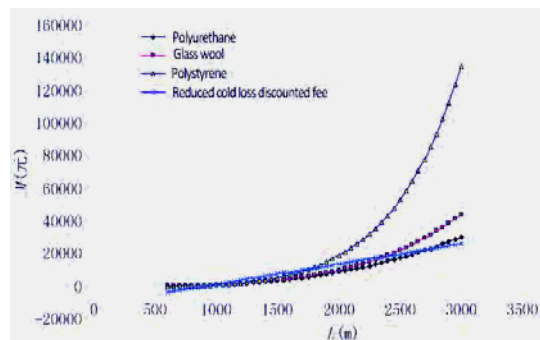


Figure4 The increased insulation layer investment in depreciation charges and reduced cold loss discounted fee varying with the transport distance

The conclusion is the same with Figure 3, that for short-distance transport, glass wool and polystyrene are more economical, while for long-distance transport, we had better choose polyurethane as insulation materials.

Figure 5 shows the pipe heat loss rate of different insulation materials, under the corresponding tube length and the thickness of the insulation layer, according to the national standard; we must ensure that the heat loss rate is less than 10% at the design of the heating pipe insulation layer.

Seen from Figure5, when $600\text{m} \leq L \leq 5000\text{m}$, the heat loss rate of the three insulation materials is less than 10%; And it tends to increasing with the increased of transport distances and insulation. The heat loss rate of polyurethane is minimum, and increased rapidly at the interval of 600m to 1850m, to flatten after 1850m; The ascensional range of the heat loss rate of glass wool and polystyrene is larger at the interval of 600m to 1350m, increased slowly after that, and when $L < 1700\text{m}$, the heat loss rate of polystyrene is greater than glass wool, the heat loss rate of glass wool insulation layer is largest when $L > 1700\text{m}$, which is because that the heat loss of pipe is associated with many factors, the thickness and thermal conductivity of the insulation layer has the different effects on the heat loss and cold loss. Analysis the reasons of pipe heat loss changes, there are the temperature difference of hot water in heating pipe and soil is large, the insulation effect is less than the cooling pipe, so the loss rate is large, but it can be guaranteed within the allowable range.

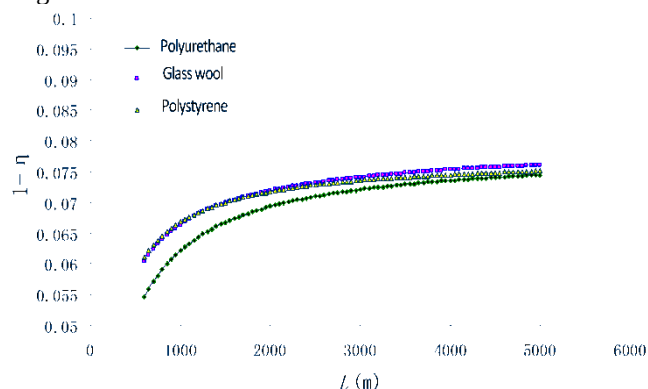


Figure 5 The heat loss rate varying with the transport distance

Comprehensive Figures 2-5, in case, we should control the maximum transport distance between 200-1900m. Among the range, using glass wool or polyurethane as insulation materials has the best economic and construction feasibility. Also, the return on investment of the system is higher, at the same time we can ensure that the hot and cold transmission efficiency reaches the predetermined value.

5. CONCLUSIONS

By analyzing the synergic relationship between the insulating layer thickness of hot and cold logistics distribution pipeline and the cold (hot) supply radius, we can get the following conclusion:

1) The distribution loss of a large-scale cold and hot network is closely related to the insulating layer thickness and the cold (hot) supply radius. Under certain constraint conditions, the insulating layer thickness and the total cost of pipeline network nonlinearly increase with transportation distance, when surpassing the synergy inflection point of the insulating layer thickness and the transmission distance, the energy efficiency and the economy of cold and heat pipeline network deteriorate. When calculating the insulating layer thickness, we should not only consider cold(hot) loss per unit area or per unit length, but also take into account the loss rate or temperature rise(fall) from the whole distribution network. This case study shows that the optimal cold (hot) transmission distance is 1.2 ~ 1.9 km, and the insulating layer thickness is 28 ~ 49 mm for polyurethane ; 61 ~ 80 mm (1.2 ~ 1.45 km) for glass cotton material.

2) The heat conductivity coefficient and price of thermal insulation material have a great influence on economy and energy efficiency; therefore one should take many factors into account when choosing thermal insulation material. This case study shows that: the cold (hot) supply radius within 1.2-1.9 km, glass cotton is the optimal economy choice for thermal insulation material; in a relatively short transmission distance (<0.7km), polystyrene is optimal economy choice; in a longer transmission distance (>2.1km), polyurethane would have better economic performance.

The above conclusion can be used as reference basis when dealing with a pipeline network design, the thermal insulating layer calculation and the location principles of energy production and storage, the application pace of NDES-SG will be further improved.

6. ACKNOWLEDGEMENTS

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ENERGY PERFORMANCE OF VAV AIR SYSTEM WITH TWO CONTROL SCHEMES

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ABSTRACT

This paper presented a new optimal control scheme of the variable air volume (VAV) system, sum flow rate-damper signal and highest load ratio-minimum flow rate, to reset supply air static pressure and temperature, respectively. This optimal control scheme aims at saving energy consumption of Air Handling Unit (AHU) fan and maintaining indoor thermal comfort. Unlike the conventional scheme that changes the static pressure set point only basing on the number of dampers those fully open, it also tends to take into account the difference between the sum flow rates measured and required. When the numbers of damper those fully open meet the predefined requirements, whilst the difference between the sum flow rates measured and required maintains within a certain range, it suggests that the VAV system reaches a steady state, and the static pressure set point maintains a fixed value, or it should be regulated to achieve this steady state. This scheme was written into software with Visual Basic (VB) language and applied to the Building Management System (BMS) of an office building in Hong Kong, whose original control scheme was constant static pressure and constant air temperature control. Serious of experimental tests were conducted to compare the performance of both control schemes. Typical days with identical climate conditions in winter, summer and transition season were selected, respectively. Both schemes were tested and the measured data in related were written into BMS every five minutes. The operations of the system with two control schemes were compared during work hour from 8:00 to 18:00 in work days. When optimal control scheme was adopted, the supply air static pressure would vary with the changing of outdoor temperature, while it maintained a fixed value with the original scheme. Coefficient of transformation is introduced to evaluate the energy saving impact. It reached 17.78% and 15.71% in winter and transition season, separately, when the indoor cooling loads is relatively low. It was 0.70% in summer, for the sake of higher outdoor temperature causing higher static pressure set point. Air Diffusion Performance Index (ADPI) is adopted to evaluate the indoor thermal comfort. The number of times (120 times in total) that ADPI is larger than 80% in winter and transition season with optimal control scheme is more than those with original scheme by 13 and 55, separately, while they are almost the same in summer. Results show that the optimal control scheme used in this paper can not only reduce the AHU fan energy consumption efficiently, but also improve indoor thermal comfort apparently, especially in part load situation. The optimal control scheme is therefore feasible.

Keywords: VAV system; experimental test; energy consumption; thermal comfort.

1. INTRODUCTION

Variable air volume (VAV) system was born in the United States in the 1960s, which tends to change the supply air volume in accordance with the changing cooling load indoor [1]. There are two control schemes mainly applied—constant static pressure control and static pressure reset control [2] Compared with constant static pressure control, the energy saving ratio of latter one is particularly significant in part load situation. However, the conventional scheme of static pressure reset control changes the static pressure set point only basing on the number of dampers those fully open [3-5],

namely that the static pressure set point will be decreased in condition that the parameter is less than the threshold (usually it is set as 1), and vice versa. Given the limitations of on-site defects of actual VAV system, such as malfunction of VAV damper actuator, unreasonable indoor air temperature set point and partitions modification, the conventional scheme that modifies the static pressure set point by identifying.

only one fully opening VAV damper may fail under these conditions, resulting in over cooling or overheating.

NOMENCLATURE

Abbreviation		Symbols	
BMS	Building Management System	t_i	measured temperature (°C)
AHU	Air Handling Unit	t_n	temperature set point (°C)
ADPI	Air Diffusion Performance Index	v_i	air velocity (m/s)
ET	Effective Temperature		

Focus of this paper will be given to propose an optimal control scheme that meets the requirement of supplying sufficient cooling air to maintain predefined indoor thermal comfort, whilst save energy as much as possible. Experimental tests are conducted to investigate how well optimal scheme is compared with original scheme concerning energy consumption of AHU fan and indoor thermal comfort. With the development of automatic control technology, Building Management System (BMS), including construction equipment monitoring system, fire system and security system, is widely employed in buildings to achieve real-time monitoring and management. The concerns will be evaluated basing on on-site experiment.

2. OPTIMAL CONTROL SCHEME

The optimal control scheme proposed in this paper contains two modules-static pressure reset control and supply air temperature reset control, aiming at achieving both of energy saving and indoor thermal comfort maintenance.

2.1 static pressure reset control

Unlike the conventional scheme, the optimal scheme tends to take into account the difference between the sum flow rates measured and required, as well. When the numbers of damper those fully open meet the predefined requirements, whilst the difference between the sum flow rates measured and required maintains within the setting range, it suggests that the VAV system reaches a steady state, and the static pressure set point maintains a fixed value, or it should be regulated to achieve this steady state. The flow chart of the developed static pressure reset control is shown in Figure 1. This module intends to reset static pressure as low as possible by providing the minimum cooling air to maintain the predefined thermal comfort. As for actual operation, certain fault or unexpected condition occurring in the system may cause insufficient supply flow rate to meet requirement. A fault-preventive conditional statement is defined as the number of times cumulating continuous insufficient sum flow rate. If the cumulative time exceeds the predefined set point, the control mode will be switched to the original constant static pressure control mode. All the parameter settings may vary case by case basing on the actual operating data. This makes the strategy more robust in terms of defect-tolerance.

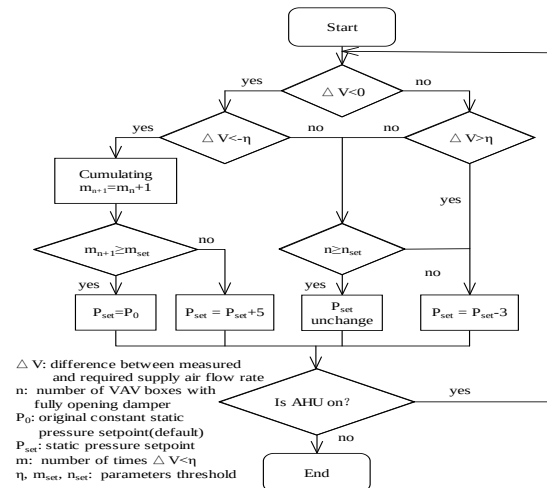


Figure 1 Flow chart for optimisation of supply air static pressure reset

2.2 supply air temperature reset control

The supply air temperature reset control module optimizes a reasonable supply air temperature set point by maintaining the predefined thermal comfort.

The thermal comfort in this module is determined from two aspects--the number of VAV boxes suffering less supply air flow rate than the preset minimum air flow rate, as well as the number of VAV boxes whose load ratio exceed 90%. Load ratio is defined as "required air flow rate/preset maximum air flow rate". Theoretically, both set points of j and i are supposed to be 1, for the sake of avoiding subzone overheating or overcooling. However, due to the malfunction of individual VAV boxes or special requirement from office tenants, the ideal setting will result as ineffective, even not working. Therefore, the parameter settings should be determined basing on the actual operating data and defect-tolerance considerations. The flow chart of supply air temperature reset is shown in Figure 2.

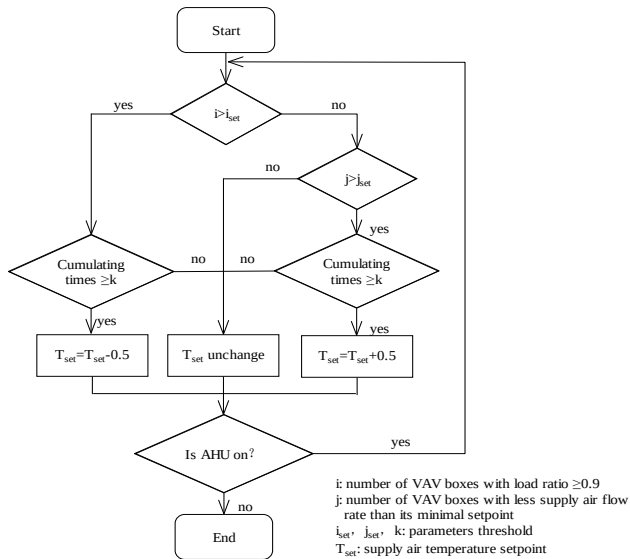


Figure 2 Flow chart for optimization of supply air temperature reset

3. BUILDING DESCRIPTION

The objective building is a 36-storey office building located in Hong Kong Island. The typical floor plan is shown as Figure 3. Its load is handled by an air handling unit (AHU), whose cooling capacity is 118.9KW. Air volume of the fan is up to 20160m³/h in design, whose power is 11KW. The supply duct is branching pipeline with a total of 37 VAV boxes. There are 4 static pressure sensors installed, the lowest signal of which is used as a feedback signal to adjust the fan speed. Before optimization, constant static pressure & constant temperature was applied as the control scheme. The static pressure set point was set to 200Pa in summer and 180Pa in winter, and temperature was 15°C. After optimization, static pressure reset and temperature reset control is applied. Static pressure set point changes in the range of between 100 and 250Pa, whilst temperature set point varies from 13.5 to 18°C.

The office building is equipped with BMS, which is able to monitor the operation conditions of duct system, water system, as well as refrigeration units on line and real-time. The data is accessible, including supply air temperature, return air temperature, indoor temperature, fan speed control signal, static pressure set points, air volume measured and required of each VAV box, opening signal of VAV boxes, etc. Data of BMS is updated every 5 minutes.



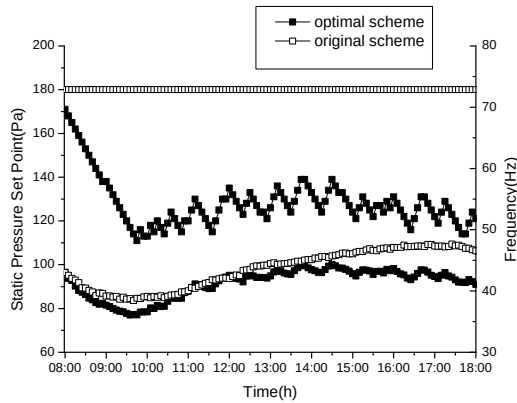
Figure 3 Schematic of typical VAV system

4. RESULTS AND ANALYSIS

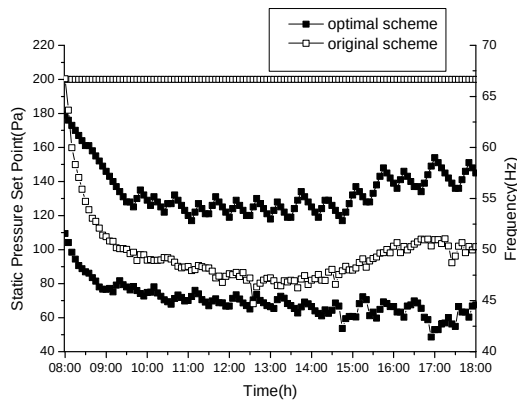
This optimal scheme was written into software with Visual Basic (VB) language and applied to BMS with OPC data communications technology, which provides convenient for the experimental tests aiming at comparing the energy consumption and indoor thermal comfort when both schemes are applied individually. Typical days with identical climate conditions in winter, summer and transition season were selected, respectively. Given the test days were weekdays; the cooling loads of personnel and equipment indoor were identical. Therefore, the cooling load of test days can be assumed to be the same. Both schemes were tested during work hour from 8:00am to 6:00pm, and the data in related were marked down by BMS every five minutes.

4.1 Impact of Optimization

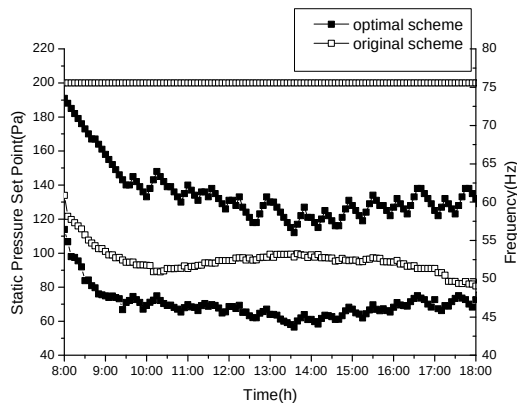
The trends of static pressure set point and frequency of AHU fan with two control schemes are shown as figure 4. They reveal that the static pressure set point remained a fixed value when original scheme was applied, while it varied when optimal scheme was adopted. It is noted that at the very beginning, a larger amount of air volume was required to remove the heat storage of building envelope and furniture when the indoor cooling load was relatively high. Consequently, the static pressure set point remained relatively high. With the decreasing cooling load, the value then fluctuated at a relatively low range. It can be seen that the frequencies of optimal scheme were lower than those of original scheme.



(a) Winter



(b) Summer



(c) Transition season

Figure 4 Operation with two control schemes

The cooling load in winter was so low that the supply air temperature remained 18°C, the maximum set point, to increase flow rate in case of insufficient fresh air when optimal scheme was adopted, while it was 13.5°C in summer, the minimum set point, for the sake of decreasing flow rate to save energy. The supply

air temperature in transition season is shown as figure 5. In transition season, the climate is so mild that the set point is not required to remain the maximum or minimum. It is noted that the set point varied in accordance with the changing cooling load.

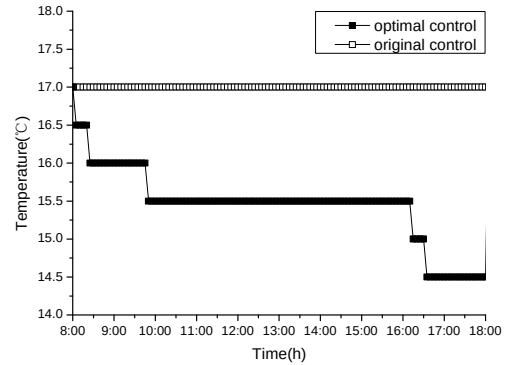


Figure 5 Supply air temperatures in transition season

4.2 Impact of Energy Conservation

Coefficient of transformation is introduced to evaluate the impact of energy conservation, which is defined as “total power consumption of fans/total cooling load”, namely the less the coefficient of transformation is, the less power is consumed to transform unit cooling load. The total cooling load can be calculated by chiller water supply temperature, water return temperature and air flow rate. The data mentioned above can be acquired from BMS.

Calculation results of test days are shown in Table 1. It can be found out that, in test days of winter and transition season, coefficients of transformation with original scheme were less than those with optimal control. The energy saving ratio is up to 17.78% and 12.31% respectively. While in test days of summer, the energy saving ratio is approximately 0.70%. It is not as ideal as test days of winter and transitional season. The reason is that the cooling load is much higher in summer. More air flow rate is required to remove indoor residual heat and wet. Static pressure set point is needed to maintain relatively high. In winter and transitional season, VAV system works in part load situation at most of the time and the requirement of air flow rate is met even though the fan maintain a relatively low speed. Impact of energy conservation is more obvious.

Table.1 Comparison of energy consumption in typical test days

		Total cooling load(kw)	Total power consumption(kwh)	Coefficient of transformation	energy saving ratio
winter	original control	9059	1508	0.166	17.78%
	optimal control	8534	1168	0.137	
summer	original control	33725	1763	0.0523	0.70%
	optimal control	31650	1643	0.0519	
transition season	original control	22043	1800	0.0817	15.71%
	optimal control	23130	1592	0.0688	

4.3 Impact of Thermal Comfort

Thermal comfort is a subjective feeling of the thermal environment which cannot be described accurately or measured directly, but it can be assessed under laboratory condition. Regarding comfortable air conditioning, impact of the relative humidity within the range of 30% -70% is insignificant, so wind velocity and air temperature can be taken into account to evaluate the combined effect to human body. According to the experimental result, the following relationship exists [6]:

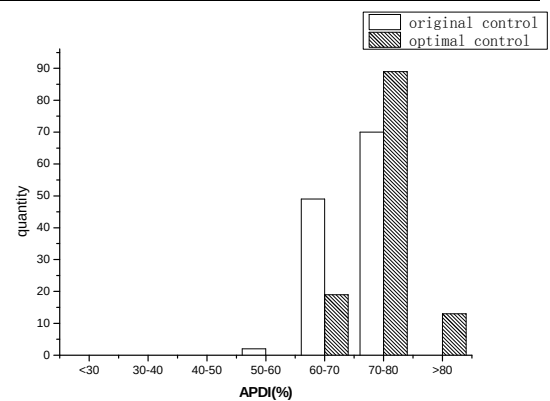
$$\Delta ET = (t_i - t_n) - 7.66(v_i - 0.15) \quad (1)$$

Where, ΔET is Effective Temperature, t_i is measured temperature(°C), t_n is temperature set point(°C), v_i is air velocity(m/s).

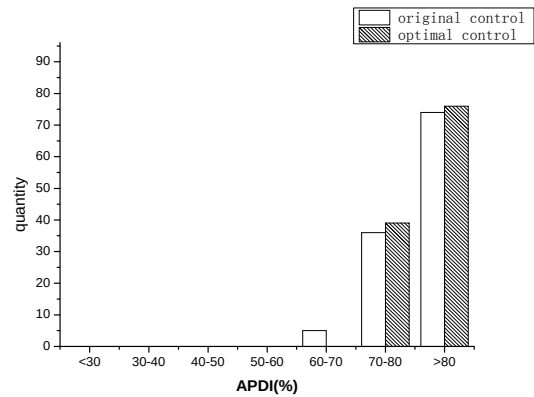
Air Diffusion Performance Index (ADPI) is defined as the percentage of test points meeting the criteria that ΔET is between -1.7 and +1.1^[5].

ADPI indicates that the impact of airflow to the comfort in occupied area. The higher ADPI is, the higher percentage of people is satisfied with the environment. Under normal circumstances, the thermal comfort is deemed satisfaction when $ADPI \geq 80\%$, and it is optimum when $ADPI = 100\%$.

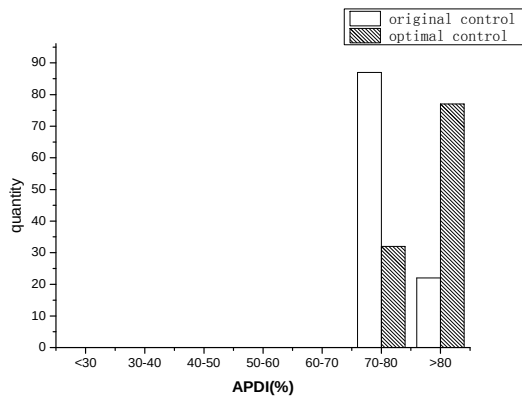
Data of Measured temperature and set point of each VAV box were acquired from BMS. ADPI was computed every five minutes. Figure 6 illustrates the distribution of ADPI with two control schemes in test days of winter, summer and transition season. It can be seen that the total number of times that ADPI is larger than 80% in winter and transition season with optimal control scheme is more than those with original scheme by 13 and 55, separately, whilst it is slightly better than the latter one in summer. These three cases show that the optimal control scheme used in this paper improve indoor thermal comfort remarkably, especially in part load situation.



(a) Winter



(b) Summer



(c) Transition season
Figure 6 ADPI with two control schemes

5. CONCLUSIONS

The optimal control scheme was proposed and applied to an office building located in Hong Kong. Typical days in winter, summer and transition season were selected. Serious of experiments were conducted to explore energy conservation and thermal comfort between optimal control scheme and original scheme. Following conclusions can be got.

1. Coefficient of transformation is introduced to evaluate the energy saving impact. It reached 17.78% and 15.71% in winter and transition season, separately, when the cooling loads indoor is relatively low. It was 0.70% in summer.

2. Air Diffusion Performance Index (ADPI) is adopted to evaluate the indoor thermal comfort. The total number of times that ADPI is larger than 80% in winter and transition season with optimal control scheme is more than those with original scheme by 13 and 55, separately, while they are almost the same in summer.

3. Three cases show that the optimal control scheme used in this paper can not only reduce the AHU fan energy consumption efficiently, but also improve indoor thermal comfort apparently, especially in part load situation. The optimal control scheme is therefore feasible.

ACKNOWLEDGEMENTS

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MODEL-BASED PREDICTIVE CONTROL FOR A HYDRONIC HEATING AND COOLING SYSTEM

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ABSTRACT

Buildings consume a significant amount of energy to maintain the indoor thermal comfort. One way to reduce the energy consumption of mechanical conditioning is to improve the overall energy efficiency through integrated advanced controls. It is undoubted that incorporating a model and utilizing future information in building operation offers great energy saving potential. Thermal comfort and energy conservation are the two basic reasons for having a building automation system in operation in buildings. However, neither of the two objectives can be explicitly included in conventional single-input-single-output (SISO) control laws. In addition, a building system is normally a nonlinear multiple-input-multiple-output (MIMO) system which could have coupling issues among the controlled objects. This paper presents the findings of applying classical linear model-based predictive control (MPC) in a nonlinear hydronic heating and cooling system. Wiener nonlinear MPCs and Hammerstein-Wiener nonlinear MPCs are studied for the space thermal conditioning. The approach remains the advantages of linear classical MPC and solves the nonlinearity issues involved in thermal comfort oriented control. With a linear time-invariant first-principle model for the building and linear MPC as the core, the controllers achieve the optimal solution for IWN in about three minutes for five days simulation on a Duo T6400 2.0GHz laptop. The fastest scenario takes only thirty more seconds to accomplish, which makes the approach practical for online implementation. The techniques of using MPC for achieving smooth day-night switch, band control, dynamic constraints, and dynamic weighting are discussed. In addition, the energy saving potential of applying the proposed MPCs is found to be between seven to forty two percent compared to a temperature-oriented conventional control. The techniques are applicable to other space conditioning systems other than a hydronic system.

Keywords: Model-based predictive control (MPC); Hydronic heating and cooling; Thermal comfort, Energy savings.

1. INTRODUCTION

Buildings account for about 40% of the total energy consumption in the U.S., and most of the energy is used to heat, ventilate and air-conditioning the space [1]. Although the efficiency of individual equipment has increased considerably with better design, manufacturing, and engineering in general, the energy use has continued to rise because of increased use of energy consuming devices and evolved demand on indoor thermal comfort. Given the significance of energy consumption and greenhouse gas emission in buildings, any successful response to ensuring the energy and environment sustainability must consider the energy efficiency in buildings. To improve the management effectiveness and efficiency, a building automation system (BAS) is typically deployed in modern buildings to interconnect and automate the various systems and devices [2].

Building automation controls are typically fulfilled by using a two-level control structure: a supervisory level and a local level. The supervisory-level controller comprises a group of logics and typically resides in a central BAS station. It takes actions based on preset

conditions and/or rules, or the commands from the operators. The basic functions include heating/cooling switch-over, and components sequencing, event scheduling, etc. When diverse building systems are to be coordinated, the control logic becomes complicated, non-standard, and difficult to troubleshoot [3]. In addition, the rule-based responsive control, with local ON/OFF and Proportional-Integral-Differential (PID) controllers, is expensive in the long run since they operate at a non-optimal efficiency.

While predefined rules and single-loop PID controls are interpretable to and preferred by onsite engineers, obtaining optimal settings for a complex system is not an easy job. Meanwhile, neither of the two control objectives, thermal comfort and energy savings, can be explicitly expressed in conventional control laws. A building control system needs to possess some kind of predictive capability with building dynamics and deterministic and stochastic disturbances considered and can evaluate objectives (e.g. energy efficiency, thermal comfort, etc.) rather than a simple reference. One of the enabling technologies is termed model-based predictive control (MPC).

Nomenclature

Abbreviations		Δu	Rates of control inputs
BAS	Building automation system	h, f	Inverse functions
HW	Hammerstein-Wiener	z, v	Mapped variable vectors
LTi	Linear time invariant	k, i	Counter
MPC	Mode-based predictive control	T	Temperature
MIMO	Multiple input multiple output	$\dot{Q}$	Heat transfer rate
PID	Proportional integral and derivative	$\dot{m}$	Mass flow rate
RC	Resistance and capacitance	Superscripts/ Subscripts	
SISO	Single input single output	T	Transpose
Symbols		sen	Sensible heat
A, B, C	State-space matrix	lat	Latent heat
e	Reference error	ra	Room air
y	Output vector	oa	Outdoor air
x	State variable vector	og	Ground condition
L	Low limit vector	m	Material
U	Upper limit vector		

From a general point of view, MPC is not a specific control law, but a strategy which uses a combination of control methods linked by common ideas [5]. Any combination with a building system model, an objective function, a set of constraints, an optimization solver and a receding horizon algorithm can be loosely regarded as an MPC.

MPC has been utilized in the manufacturing process industry since early 1970s. Research on MPC in building systems has just begun to thrive. Models for MPC are constructed as a physical model, gray-box model, or black-box model. Typically a global search engine (e.g. genetic algorithm) is applied on the nonlinear models to identify the optimal set points [4, 6-9]. The second most popular solving method is sequential quadratic programming. For example, Wang and Jin utilized the genetic algorithms with a detailed model to optimally control a VAV air-conditioning system by identifying the proper supply air temperature, outdoor ventilation flow rate, and chilled water temperature [6]. Kolokotsa et al. used a simplified bilinear-model based predictive controller, combined with a building energy management system, to predict and control the indoor environmental conditions of a 125m² single zone laboratory [7]. Chen formulated an optimal model with a number of practical operation constraints to supervise the operation of building systems with big thermal mass [8]. Keblawi et al. developed an optimized online supervisory controller by using genetic algorithm for the chilled ceiling displacement ventilation [9].

However, most of the literatures do not consider the computational cost with the model constructed invisible to the optimization solver. The application and performance of linear classical MPC for multiple-input-multiple-output (MIMO) nonlinear building systems are not fully explored. Meanwhile, very little publication considers both thermal comfort and energy savings explicitly in the formulation of MPC. In this paper, we briefly summarized classical linear MPC in terms of the formulation, model structure, and solving method. Then

a Wiener and a Hammerstein-Weiner nonlinear MPC with classical linear MPC core are introduced and applied to a hydronic heating and cooling system to evaluate the performance. The study concludes with a brief discussion on the results and findings.

2. LINEAR QUADRATIC MPC

Linear classical MPC is one of the most mature optimal control strategies and serves as the base to the understanding and formulation of other general MPCs.

2.1 Controller formulation

A classical MPC has the following quadratic programming formulation, including cost function, linear model, and boxed constraints:

$$\min \sum_{i=1}^k \|e^y\|_Q^2 + \|\Delta u\|_s^2 + \|u\|_R^2 \quad (1)$$

$$\dot{x} = Ax + Bu \quad (2)$$

$$y = Cx$$

$$x^L \leq x \leq x^U$$

$$y^L \leq y \leq y^U \quad (3)$$

$$u^L \leq u \leq u^U$$

$$\Delta u^L \leq \Delta u \leq \Delta u^U$$

The internal model utilized in a classical MPC has linear and time-invariant features. Depending on the applications, different types of models may be used. The common model types are: impulse and step response models, auto-regressive moving average with exogenous input models, controlled auto regressive integrated moving average models, transfer function models, and more general, state-space models. All the linear dynamic models can be transformed to a state space model. A state space model provides a nice structure for MPC, and it can easily handle MIMO system. Meanwhile, it has uniform treatment of stable, integrating, and unstable processes.

2.2 Solving methods

Since the model and the constraints are linear, and the cost function is formed in quadratic, classical linear MPC can be eventually casted as a quadratic programming problem. A quadratic programming problem is a convex optimization problem with a unique optimal solution and can generally be solved in a short time, which makes it suitable for online implementation. To offset nonzero values on the variables, an integral term of manipulated variables based on augmented model is generally used in MPC. When constraints are considered in the controller, a quadratic programming problem can be casted to obtain the solution. Techniques, such as Lagrange multipliers, Kuhn-Tucker conditions, active set methods, primal-dual method, interior-points, etc., are the candidates.

3. HANDLING OF NONLINEAR ELEMENTS IN MPC

For most buildings, there are single mechanical heating or cooling sources deployed in the space and the controlled variable is typically the space air temperature. For instance, in a VAV conditioned building, the supply air temperature and the air flow delivered into the individual room are the manipulated variables. In a fan coil conditioned space, the supply water temperature and the air flow through each fan coil are the manipulated variables. In a residential building, the power input to the air-conditioner or furnace can be considered as the manipulated variable. It is beneficial to design a model-based predictive controller with the building treated as a linear time-invariant system. Fig. 1 illustrates the layout the internal system for a linear MPC structure with heat flux as the system inputs and direct measurable variables as outputs.

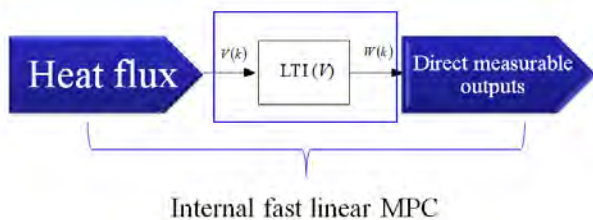


Figure 1 Pure linear classical MPC for building systems

While the building system dynamics can be reasonably simplified into a LTI model as a RC thermal network, the nonlinearity still exists. Fig. 2 illustrates the typical model structure in reality. The two ends of the linear time invariant (LTI) system are static nonlinear mapping. A system with only the static nonlinearity before the LTI is termed a Hammerstein nonlinear system. A system with only the static nonlinearity after the LTI is called a Wiener nonlinear system. A system with the nonlinearity at both ends is termed Hammerstein-Wiener (HW) nonlinear system [10]. The nonlinearity in a Hammerstein system comes

from the fact that the heat flux into the air space is usually not the direct manipulated variable.

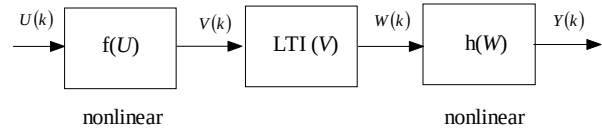


Figure 2 Hammerstein-Wiener nonlinear model

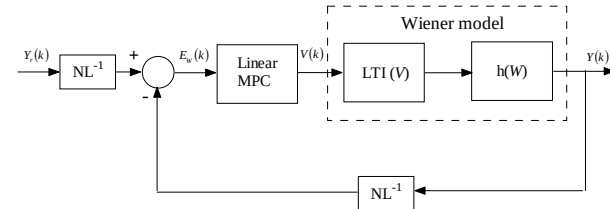


Figure 3 Control structure for Wiener model plant

To utilize the advantage of linear MPC for online implementation and solve the nonlinear system with static nonlinearity, the system inputs and outputs need to be processed accordingly. For a Wiener model, as shown in Fig. 3, the outputs from the LTI are mapped into the desired outputs by using the inverse function of the mapping. In a Hammerstein model, as illustrated in Fig. 4, the control action generated by the linear MPC first goes through a static nonlinear inverse function, e.g. from the heat flux to the water flow rate. The actual signal is then obtained and fed into the system after the inverse map.

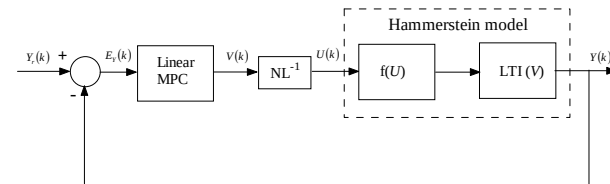


Figure 4 Control structure for Hammerstein model plant

A nonlinear Hammerstein-Wiener (HW) model consists of a linear dynamic block and two static output non-linearity functions on both sides. This structure is more realistic in representing space conditioning for thermal comfort in most building systems. The nonlinear HW system can be described by the following equations:

$$x(k+1) = Ax(k) + Bu(k) \quad (4)$$

$$y(k) = Cx(k) \quad (5)$$

$$z(k) = h(y(k)) \quad (6)$$

$$v(k) = f(u(k))$$

2. SYSTEM DESCRIPTION AND MPC FORMULATION

2.1 System description

The Intelligent Workplace north integrated building energy system is a living laboratory for the research on high performance buildings and

conditioning techniques. Approximate 600 m² area is conditioned by twenty-six sets of water mullions deployed along the facade. The other water-based terminals for local environment control include suspended radiant ceilings and ductless radiant chilled beams. When additional or individual conditioning is needed, these terminals can be controlled individually to provide extra heating and cooling. Detailed description and development of dynamic models for the building, thermal comfort, and the hydronic systems can be found in [4].

2.2 MPC formulation

Two heat fluxes from the terminals are the direct inputs to the air nodes. The outdoor air temperature, boundary surface temperature, and solar radiation heat flux are the external inputs that affect the space temperature through the construction surfaces. In addition, the sensible heat gain, latent heat gain, infiltration air flow rates, etc., are the disturbances to the system. The indoor air temperature and average air humidity are the direct outputs from the LTI space system. The mathematic expression of the LTI system for the two room layout 2R1C space can be put as:

$$x(k+1) = Ax(k) + Bu(k) \quad (7)$$

$$x = [T_{ra,i=1,2} \quad T_{m,i=1,...,11} \quad g_{ra}]_{14 \times 1}^T \quad (8)$$

$$u = [\dot{Q}_{i=1,2}]_{2 \times 1}^T \quad (9)$$

$$d = [\dot{Q}_{sen,i=1,2} \quad \dot{Q}_{lat} \quad \dot{Q}_{sol,i=1,2}]_{5 \times 1}^T \quad (10)$$

$$w = [T_{oa,i=1,...,6} \quad T_{og} \quad \dot{m}_{oa} \quad g_{oa}]_{9 \times 1}^T \quad (11)$$

The following scenarios in Table 1 are investigated in this study to evaluate the performance of classical MPC and proposed HW nonlinear MPC for the hydronic system. All scenarios are based on MPC. The simulations are conducted for five days in the heating season.

Table 1 Control scenarios for classical MPC and HW nonlinear MPC

SC	Problem description
A	Classical MPC, direct heat flux inputs, temperature oriented control.
B	Wiener nonlinear MPC, thermal comfort oriented control.
C	HW nonlinear MPC, energy consumption and thermal comfort oriented control. Thermal comfort serves as constraints, PMV = [-0.5, 0.5].
D	As case 5, with variable occupancy schedule. When unoccupied, PMV = [-3, 3]; when occupied, PMV = [-0.5, 0.5]. Band control.

3. EXPERIMENTAL RESULTS AND ANALYSIS

Fig. 5 plots the outdoor air temperature, relative humidity, solar radiation to the two rooms. The outdoor air fluctuates between 2°C to 18°C during the simulation period, which represents a typical winter.

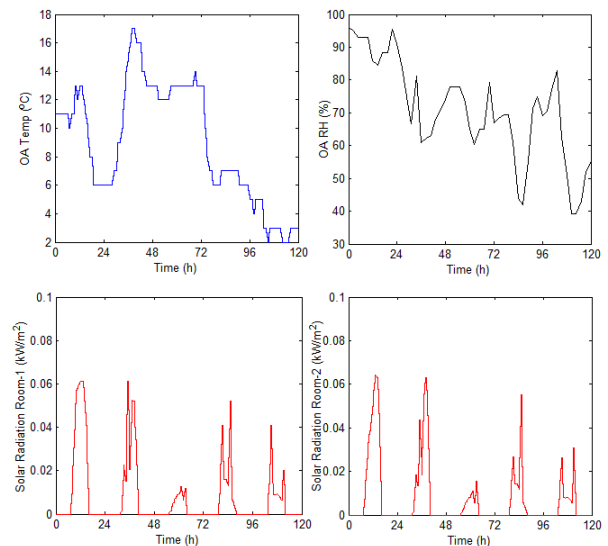


Figure 5 Outdoor air condition and solar radiation

A conventional space thermal conditioning is to track a room air temperature reference. It is typically a fixed set point, e.g. 24°C as used as a base case in this study. An office occupancy based air temperature control is also simulated for the five day period serving as an additional base case. The results are collected in Fig. 6. The system has a morning warm-up operation to ensure a smooth transition in the space from unoccupied status in the night to occupied status in the day. The total energy consumption for this case is 1941 kWh.

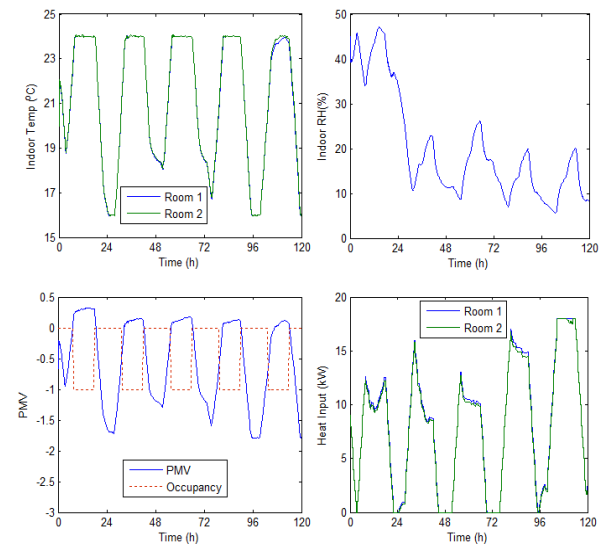


Figure 6 Simulation results, SC-A

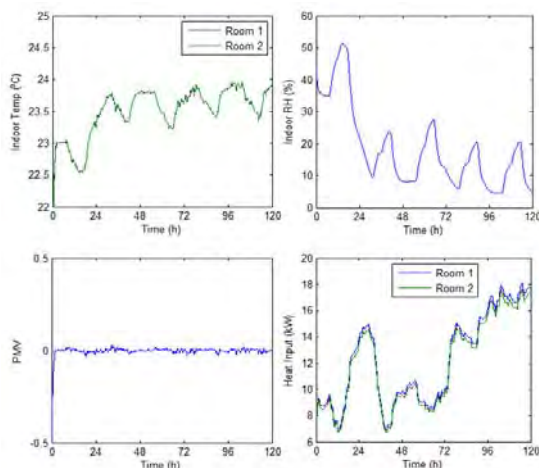


Figure 7 Simulation results, SC-B

Scenario SC-B is constructed to investigate the thermal comfort based MPC control. The nonlinearity related to the energy consumption is ignored in this control. With a Wiener structure for the thermal comfort mapping, the internal classical MPC works well with the overall hourly summation on square PMV equal to 0.07. The PMV tracking output is plotted in Fig. 7, close to the set point 0. The manipulated variables are the heat fluxes. Compared to Scenario SC-1, the room air temperature drops to be below 24°C and fluctuates along with the room air humidity as shown in Fig. 7. The overall energy consumption is reduced by around 6% to 2895 kWh.

From the study of Scenario SC-B, we see that the proposed Wiener nonlinear MPC can make full use of the advantages of classical linear MPC. It turns the room air temperature based space conditioning into a thermal comfort oriented control while achieving additional energy savings. The overall run time for 5 days control takes only around 80 seconds to identify the optimums for the 180 variables. The nonlinearity is successfully handled with this approach. In reality, a heat flux is usually not directly controllable. The corresponding nonlinearity from the manipulated variables, e.g. water flow rates, to the heat fluxes needs to be considered. A Hammerstein-Wiener structure is applied next.

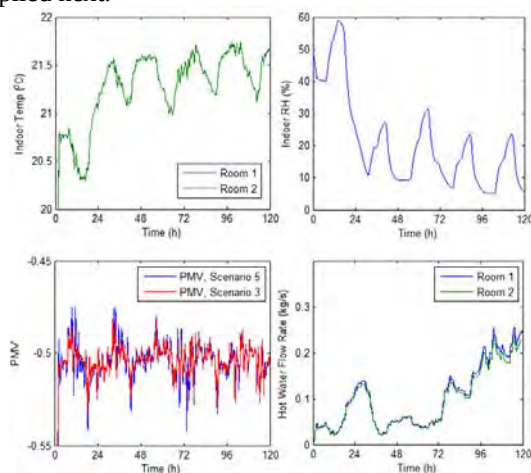


Figure 8 Simulation results, SC-C

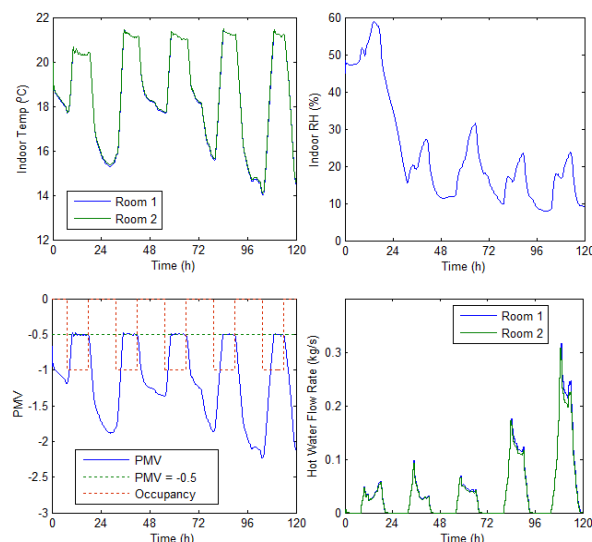


Figure 9 Simulation results, SC-D

In Scenario SC-C, the water flow rates are used as the manipulated variables in the HW nonlinear MPC. The controller is designed to minimize the discomfort from the occupants by letting PMV track -0.5. From Fig. 8, it can be seen that the HW MPC handles the system well and the PMV tracking error is close to that of a Wiener nonlinear MPC where the heat fluxes are used as the manipulated variables. With this system layout, the computational cost is increased by only 100 seconds than SC-B. The controller brings the system to the lowest acceptable PMV to minimize the energy consumption. The hot water flow rates are controlled smoothly. In the following scenarios, HW nonlinear MPC is utilized to respect the reality.

Commercial office buildings may have variable occupancy. The change of set point is a disturbance to the controller in addition to the unmeasured solar heat flux and internal heat gains. In Scenario SC-D, a variable occupancy schedule is studied and a band control is adopted. When the space is unoccupied, the constraint on PMV is loosened to [-3, 3]. They are reduced to [-0.5, 0.5] when the space is occupied. The controller is set to minimize the energy consumption during the 5 days. As plotted in Fig. 9, the controller drives the system between the two different constraints with the occupancy switched between occupied status and unoccupied status. The simulation takes 187 seconds to obtain the optimum.

The simulation results in terms of the energy consumption, thermal comfort, and simulation duration for the scenarios are collected in Table 2. In addition to the simulation duration and energy savings, an additional index based on PMV value in the occupied hours is also included. We assume -0.5 should be adopted in winter, the results show that the higher the mean PMV value, the lower the energy savings.

Table 2 Comparison of energy consumption, thermal comfort, and simulation duration

Case number	Energy Consumption (kWh)	Mean (PMV+0.5)	Simulation Duration (s)
Base	3091.7	0.72	36.1
SC-A	1941	0.69	40.6
SC-B	2895	0.56	76.9 - 78.1
SC-C	1977	0	175.8 - 198.1
SC-D	840.8	-0.25	187.4

4. DISCUSSIONS AND CONCLUSIONS

To reduce a high computational cost and ease the online implementation, Wiener and Hammerstein-Wiener nonlinear MPC for building systems are proposed and investigated in this study. With a Wiener structure-based MPC, the nonlinearity involved on the controlled variable side in a system is handled by decomposing the system into a linear system plus a static nonlinear map. With a Hammerstein-Wiener structure-based MPC, the nonlinearity on the input and output side is processed by using a linear system plus two static nonlinear maps.

The space thermal conditioning system can be controlled toward a minimum energy consumption condition, tracking a desired thermal comfort condition, and handling the nonlinearity involved in the system. Thermal comfort in a band offers more energy savings than a fixed thermal comfort reference approach. The computational cost of the simplified approach with linear MPC as the core is low which makes it suitable for online implementation in conventional building systems for space thermal conditioning. All the scenarios for one week control can be accomplished in less than eight minutes. By applying thermal comfort oriented control, in an office schedule condition, about seven to forty two percent of energy may be saved by using MPC empowered thermal comfort control.

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STUDY ON LIGHT WEIGHT BUILDING ENVELOPE WITH PCM FOR SUMMER INDOOR THERMAL ENVIRONMENT ADJUSTMENT

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ABSTRACT

Comparative experiments of indoor thermal environment of two experimental rooms were carried out. One experimental room was made by polystyrene foam board with the phase change material (PCM), and another one was made by polystyrene foam board only. The experiments included the variation of indoor air temperature and wall surface temperature. It was determined for the performance of indoor thermal environment adjustment of lightweight building envelope with PCM. The air conditioning and heating energy saving efficiency were also calculated. According to the study, for the hot summer, PCM can effectively control the fluctuation and rise of indoor air temperature of lightweight building envelope. Compared with common lightweight building envelope, PCM can reduce the indoor temperature up to 8.5°C. The average reduction is 1.5°C.

Keywords: PCM, Lightweight envelope, Indoor thermal environment, Energy saving.

1. INTRODUCTION

With the rapid development of construction industry, the traditional heavy Weight Building Envelope such as brick, concrete, cannot be satisfied with the needs of modern construction, while light weight building envelope is applied widely^[1]. Most of the light Weight Building Envelope are made by polystyrene (EPS), polyurethane (PU) foam board, extruded polystyrene (XPS), EPS foam mortar and part of the light, porous inorganic heat preservation material. Using lightweight enclosure structure, it can greatly reduce its bearing structure and also can improve the building palisade structure heat preservation effect with the good heat insulation effect. However, because of its small heat capacity, low thermal inertia and poor thermal storage ability, it would often led to increase indoor temperature fluctuation^[2]. In the summer, the effect is not obvious which attenuate and delay solar radiation heat exposure to building exterior wall. It causes the indoor temperature too high; in winter, the effect is limited which absorb and store the daytime solar radiation heat. It causes indoor temperature diurnal fluctuations. For these reasons, light weight building envelope is restricted in the practical application^[3, 4].

In view of the above problems, domestic and foreign scholars put forward using phase change material (PCM) in the light weight building envelope, and carried out a series of related research work. Feustel H E^[5] evaluated PCM wallboard store latent heat performance in architectural environment based on the finite difference approximation numerical simulation

program. They used double PCM panels to improve the building heat storage capacity, so that the room temperature can be maintained closing to comfort temperature limit without using mechanical cooling conditions. Frédéric Kuznik^[6] measured the thermal environment differences between PCM experimental room and ordinary experiment room in the artificial climate laboratory in the summer, winter and transitional season. Andre Bontemps^[7] measured the PCM on the thermal performance of passive solar houses, and numerical simulation was carried out based on the measured data. Xiao wei^[8] used an enthalpy model for a south-facing room of Beijing in winter to the effects of the thermal properties, surface heat transfer coefficients and amount of the PCM panel on the indoor temperature; the analyses are helpful for the application of the shape-stabilized PCM panel in lightweight house in winter. Yang cheng^[9] manufactured graphite foam-paraffin composite as a PCM and an experiment study was carried on the abilities of heat insulation and thermal control with the composite PCM which was applied as the building envelope. Den Anzhong^[10] established the experimental cubicle with the light weight PCM wallboard which is prepared by bonding PCM composite on polystyrene foam board and improved thermal storage capacity compared with conventional light weight walls. According to the existing researches, PCM for light weight building envelope heat environmental regulation and capacity still needs further research and quantitative.

In this paper, there were two experimental rooms. One experimental room was made by polystyrene foam board with the phase change material (PCM), and another one was made by polystyrene foam board only. It will improve light weight building envelope indoor thermal environment performance and potential through experimental contrast analysis method of PCM and provide important experimental support for regulating the indoor thermal environment of light weight building envelope.

2. EXPERIMENTAL SYSTEM AND METHOD

2.1 Experimental subject

The experimental rooms were built in the roof of Architecture and Environment Building of Sichuan University, shown in Figure 1. Two room dimensions are 800mm (L) x 1000mm (width) x 1300mm (H). The outer envelope used 0.8mm thick stainless steel +40 mm thick lightweight polystyrene (EPS) foam board + 0.8 mm thick the +8 mm stainless steel sheet of gypsum board. The structure was shown in Figure 2. The one room was installed with the tubular PCM. First, the PCM was arranged and fixed on the plasterboards match box size (Figure 3), then the plasterboards were installed in the room inner surface, the PCM constitute a composite structure between the plasterboard and the inner surface. The PCM is encapsulated in ABL and formed "TCM tube", which is a new easy-wall composite material by the Tri-Y Environmental Research Institute in cooperation with the University of British Columbia. Generally, a "TCM tube" is weight of 100 to 120 grams, a length of about 175 mm, and the width for 45mm thick as 20 to 25mm. The material is non-toxic, non-flammable, and phase transformation temperature ranging of 18 to 26°C. The physical parameters of the material are shown in Table 1.

Table 1 material parameters table

Parameter	Density kg/m ³	Thermal conductivity W/(m·K)	Enthalpy kWh/m ³
plasterboard	580	0.18	—
Stainless steel plate	7966	100	—
Polystyrene foam board	20	0.041	—
PCM	1300~1380	—	50~55



Figure 1 Experimental room

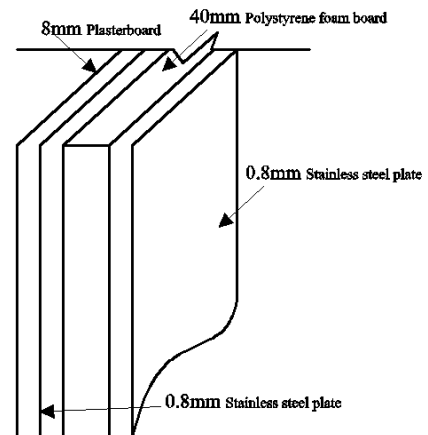


Figure 2 Experimental room wall structure



Figure 3 Composite PCM with plasterboard

2.2 Test instrument and equipment

JTRG-II is used to test thermal temperature and heat flow automatic tester which developed by a Beijing company and it was used in experiment as roving testing equipment. The device used a high-precision AD module, large-screen LCD character display. It can measure 90-way T-type thermocouple temperature and heat flux of 30 real time, and storage temperature and heat flux according to the storage time interval automatic. Its temperature measurement range is -20°C to 100°C, measurement accuracy is $\pm 0.5^\circ\text{C}$, temperature resolution is 0.1°C , the storage time interval is 1 to 60 minutes, the data storage capacity is

1000, with communication interface RS232 and operating voltage AC220V. The test used T type thermocouple with an accuracy of $\pm 0.5^{\circ}\text{C}$ before the test calibration. The outdoor temperature was recorded by JTR13C outdoor meteorological observations station, which has a stable performance, high precision, normal operation in the harsh natural environment. Its temperature measurement range is -30°C to 70°C , the measurement accuracy is $\pm 0.5^{\circ}\text{C}$, the storage time interval is 1 to 60 minutes, data storage 8000 with communication interface RS232 and operating voltage AC220V.

2.3 Measuring points arrangement

Thermocouples were arranged in the center of two room and outer wall (five surfaces including the top surface), a total of 22. It was connected with a computer via RS232 interface to achieve real-time data recording and output. The outdoor temperature was recorded by JTR13C outdoor meteorological observations station, with storage time interval of 15 min. After 10 days test from August 25, 2012 to September 03, 2012, it recorded air temperature of inside and outside and surface temperature of internal and external wall.

3. EXPERIMENTAL RESULTS AND DISCUSSIONS

3.1 The characteristics of indoor air temperature change

As shown in Figure 4, it is a temperature curve of comparative experiment case indoor air of the room in the summer. The selected analysis period is from 8:00 to 18:00 every day. During the first 5 days and last 2 days of the test, the temperature of the outdoor air fluctuated between 22.1°C to 34.5°C due to the hot weather. The middle three days of the test was the rainy weather, and outdoor air temperature fluctuated between 18.3°C to 22.8°C . At the same time, it could be found the ordinary cabinet air temperature rapidly increased when the outdoor air temperature was higher. The outdoor air temperature rose owing to small heat storage capacity and low thermal inertia index of the ordinary cabinet room wall. The composite cabinet room can be better to prevent heat transfer to the indoor. It could achieve a heat attenuation and delay the temperature rise effect as the large thermal inertia index. The delay time of the composite cabinet room and the ordinary cabinet room were up to 3.25h and 1.5h, and the average delay time were 2.5h and 1.5h, respectively. Secondly, the reduced air temperature of composite cabinet room was up to 8.5°C , and average with 1.5°C compared with the ordinary cabinet room's temperature. In addition, for rainy weather, the phase change material could release heat which was accumulated early and did not release completely. It can effectively maintain the indoor temperature basically stable.

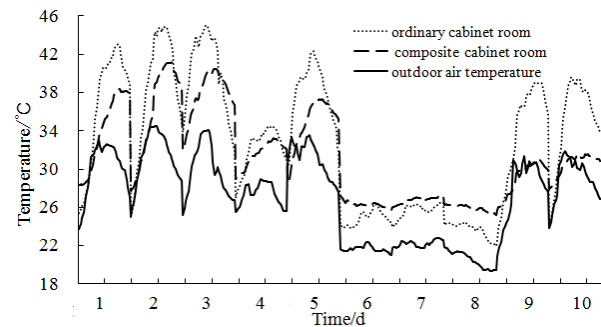


Figure 4 Indoor temperature curve of composite room and ordinary room (8.25~9.3)

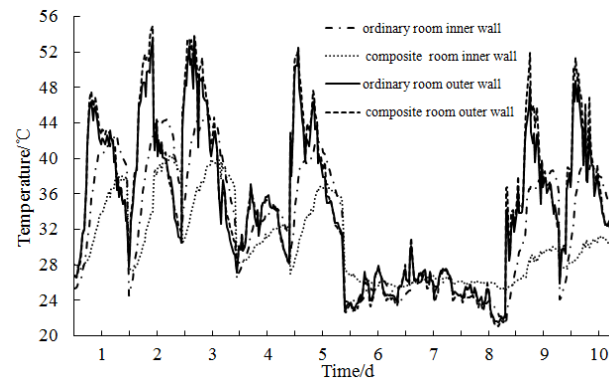


Figure 5 Inside and outside south wall surface temperature change of composite room and ordinary room (8.25~9.3)

3.2 The characteristics of temperature change of the inner and outer wall surfaces

The object of analysis was the south wall of the inner and outer surface temperature. Figure 5 is a comparative experiment cabinet room in the summer south wall of the inner and outer surface temperature curve. The selected analysis period was from 8:00 to 18:00 every day. As shown in Figure 5, it can be found that the south wall surface of the two experiments cabinet rooms were at a higher temperature range from August 25 to September 3, and the error range of both room remained fairly consistent, average temperature difference was 0.48°C from 21.1°C to 54.9°C . Secondly, the south wall surface temperature of the composite phase change material cabinet room fluctuated between 25.2°C to 40.5°C , while the ordinary cabinet room fluctuated between 21.3°C to 44.9°C . Therefore, the degree of temperature fluctuations of the ordinary cabinet room south wall inner surface were higher than that of the composite phase change material cabinet room, and surface temperature difference of two experimental box room south wall was up to 9.8°C . The reason for this characteristic is that the latent heat storage capacity of the phase change material reduces the actual indoor heat gain.

Through the above analysis, it found that the surface temperature of inner wall of the composite phase change material cabinet room can be significantly.

4. CONCLUSION

In this paper, it was tested and analyzed for the regulation performance of the composite phase change materials for the lightweight building envelope construction room temperature and the wall temperature, and carried out the following conclusions:

(1) In the hot summer weather conditions, the lightweight building envelope construction use the composite phase change material to effectively control the indoor temperature fluctuations and temperature rise, reduce the indoor temperature up to 8.5°C, average temperature with 1.5°C, and good temperature delay function, the delay time is up to 3.25h.

(2) When the south wall surface temperature of two experimental cabinet room are the same, the composite phase change material can effectively reduce the inner surface temperature of the south wall in hot climates, the inner surface temperature difference of two experimental cabinet room south wall is up to 9.8°C.

ACKNOWLEDGEMENTS

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THE THERMAL PERFORMANCE EVALUATION OF THE “TRI-LAYER” STRUCTURE WALL WITH PASSIVE PHASE CHANGE HEAT STORAGE IN SOLAR GREENHOUSES

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ABSTRACT

To improve the thermal environment of crops in non-heated greenhouses, on the basis of the phase change heat storage wall materials developed by this research team, the structural method of building a “tri-layer” structure wall with passive phase change heat storage in greenhouses is proposed by this research, and this structure wall has been applied to the north wall of the tomatoes-planted greenhouse. The results of a comparative experiment conducted from November 2011 to March 2012 show: compared with the common green house, the tomato yield per unit area in the phase change greenhouse has increased by 70%, and the fruit quality has improved significantly. According to the research results, combined with the theories of the agricultural meteorology, phase change energy storage as well as architectural thermal physics, the impact of the “tri-layer” wall structure’s thermal performance on the thermal environment of the solar greenhouse has been quantitatively evaluated and analyzed from the aspects of the internal surface temperature of the north wall, the soil temperature of the plough layer, the effective accumulated temperature, the daily effective heat storage as well as the daily effective heat supply. The analysis results show that the daily effective heat storage of the “tri-layer” wall structure in the greenhouse that applied phase change storage wall has increased by 18%, the daily effective net heat supplied has increased by 20% compared with the common greenhouse, and the time of effective heat storage has increased by 3 hours. The “tri-layer” structure wall with passive phase change thermal storage in solar greenhouses can effectively improve the thermal environment of winter crops and significantly increase the quality and yield of crops.

Keywords: solar greenhouse; “tri-layer” structure wall; passive phase change thermal storage; heat transfer performance; evaluation

1. INTRODUCTION

Temperature can affect the rate of plant’s growth and development, quantity and distribution by exerting direct and indirect influence on the metabolism of plants[0]. However, due to the scarcity and intermittence of solar energy and the cold winter in the northern area of China, the role of non-heating solar greenhouse in passively utilizing solar energy to improve the thermal environment of crops is constantly challenged [0]. The walls are instrumental in heat absorption, heat storage and heat preservation; North wall plays a significant role in controlling the thermal environment of greenhouses. Scientific and reasonable greenhouse wall structures have become the major research subjects of facility agriculture engineering in China nowadays.

Some relevant researches concerning the improvement of wall’s passive utilization of solar energy and the combination of new materials with solar greenhouses have been conducted by domestic and foreign scholars. By means of numerical modeling, Kumari N [0] has researched the effect of PCM-thermal insulation

materials-heterogeneously composite wall on the thermal environment when it is applied on the north wall of solar greenhouses. AtyahNajjara et al [0] have built a coupling model of PCM-solar greenhouse, Fang Hui [0] et al have developed a regenerative system that consists of the heat collection device of the wall, the heat storage device of the shallow layer soil and the heat circulation device. Zou Zhi-rong et al [0] have conducted relevant researches in the preparation of PCM and its application on greenhouses. On the basis of the GH-20 phase change wall material developed by our research team as well as the fundamental environment of solar greenhouses [0],[0]this study further proposes to build a “tri-layer” structure wall with new passive phase change thermal storage in the solar greenhouse, and a comparative study has been conducted in the experimental base of solar greenhouse in the suburban area of Beijing from October 11, 2011 to March 20, 2012, and based on the experimental results, the thermal performance of the “tri-layer” structure wall and its influence law on the thermal environment of solar greenhouses have been evaluated and analyzed.

NOMENCLATURE

Abbreviation

PCM	Phase change material
DEHST	Daily effective heat storage
DEHSU	Daily effective heat supply
EHS	Effective heat storage
DENHS	Daily effective net heat supplied

Symbols

T	Temperature (m^2/s)
Q	Quantity of heat ($^{\circ}C$)
q	Heat release per unit area (W/m^2)
A	Area (m^2)

Subscripts

i	Number of each day
PCM	Phase change materials
$Brick$	Brick materials
x	Heat storage
f	Heat release
$loss$	Outside wall heat dissipation
$night$	Net effective heat supplied
net	

Greek Letters

η	Efficiency (%)
τ_1	9:00 on first day
τ_2	16:00 on first day
τ_3	9:00 on second day

2. NEW “TRI-LAYER” WALL STRUCTURE

Figure 1 shows typical Chinese structural style of solar greenhouses. During the daytime in winter, the solar radiation is irradiated into the greenhouse through a fore slope roof, some of which is irradiated into a superficial layer of the greenhouse's north wall, and some part of the solar energy is absorbed, accumulated and conducted onto the internal layer of the north wall, while another part of the solar energy gets through the internal surface of a wall and reflects on the other wall surfaces via a radiation heat transfer and heat exchanges with the air via a heat convection. (Figure 1a). When the air temperature in the greenhouse decreases in the nighttime, the north wall conducts its thermal energy that accumulated during the daytime onto the superficial layer of the wall from inside to outside, then the north wall transfers heat with its surrounding via radiation and convection, by this means, the greenhouse's heat that released to the outdoor is supplemented. (Figure, 1b)

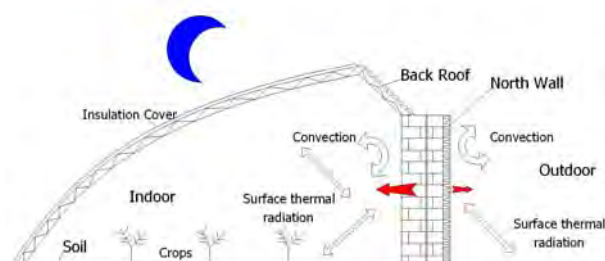
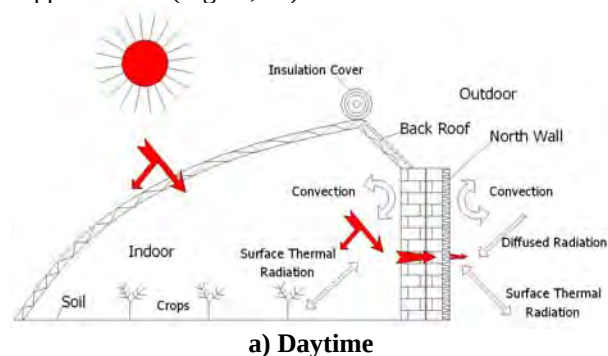


Figure 1 Greenhouse wall heat transfer process

Obviously, in order to improve the heat storage capacity and utilization rate of solar energy, the first key is maximally improving the thermal storage capacity of the material layer of the greenhouse wall's internal surface; the second key is maximally increasing the thermal resistance of the material layer of the greenhouse wall's external surface to utmostly decrease the loss of solar thermal energy that released towards the outdoor; the third key is fully strengthening the thermal storage capacity of the wall that in middle. Therefore, this paper suggests the characteristics of the material's specific heat capacity and thermal resistance in solar greenhouses are supposed to bring into full play, a “tri-layer” structure wall is suggested to be constructed. This structural mode suggests that PCM is adopted on the internal surface of the greenhouse walls; the thermal insulation material with large thermal resistance is adopted on the external surface; the heavy block brick with load-bearing structure and sensible heat as well as thermal storage is adopted on the middle wall. (Figure 2)

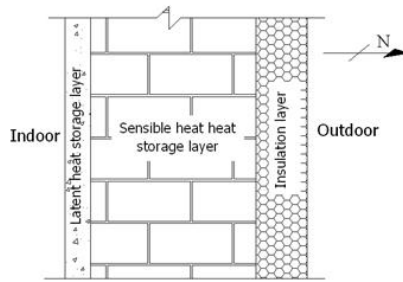


Figure 2 The structure schematic diagram of the new "tri-layer" wall structure

3.APPLICATION OF THE " TRI-LAYER " STRUCTURE WALL

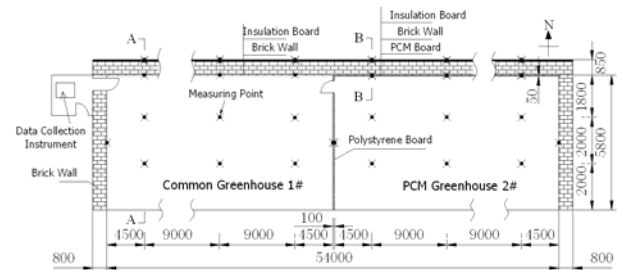
In order to validate the scientificity of the construction concept of "tri-layer" structure, a comparative test has been conducted in the experimental base of solar greenhouse in the suburban area of Beijing from October, 2011 to March, 2012 in this research.

3.1 Experimental and Methods

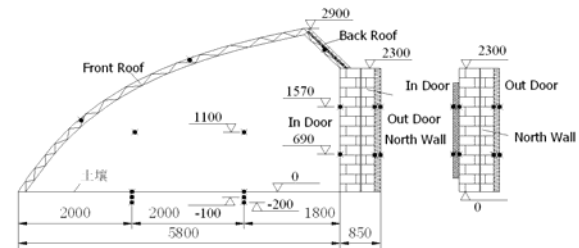
The experimental solar greenhouse is a non-heating greenhouse. To conduct a comparative study, the greenhouse is evenly divided into 2 sections, 1# is a common greenhouse and 2# is a PCM greenhouse; the north walls of the two greenhouses are pasted with a 50mm thick polystyrene foam wallboard; the internal surface of north wall of the PCM greenhouse 2# is pasted with 50mm thick GH-20 PCM developed by this research team, thus forming a "tri-layer" structure wall (The phase-transition temperature of this heat storage material is 18 to 25 ° C and the latent heat is around 80KJ/kg); The two greenhouses are separated by a 100mm thick polystyrene foam wallboard (Figure 3a)-b)).

This research focuses on observing the internal surface temperature of the north wall, the air temperature and the temperature of the tillage layer; Temperature sensor is made by copper – constantan T thermocouple, and all the datum are collected by HP34970A data acquisition instrument, the datum are collected every 10 minutes, "x" in Figure 3 represents the location of the temperature measure point. During the test, the heat preservation quilt of the fore slope roof is uncovered in 9:00,covered at 16: 00 each day.

To compare the differences of the crop's growth conditions in the two greenhouses, 10 representative seedlings with identical varieties (Xianke VI) and growth conditions are selected respectively from the two greenhouses as measuring objects after the planting of the crops in October 11, 2011. The crops in the two greenhouses are managed identically with water and fertilizer.



a) The experimental greenhouse floor plan



b) A-A and B-B profile map

Figure 3 The experimental greenhouse

Portably removable meteorological stations are adopted during the test to monitor in real time the solar irradiance, the outdoor air temperature and the wind speed, etc. The distribution frequency (from 8.0 to 11.0MJ/m²) of Beijing's total solar radiation in winter accounts for 36.3 percent, and to facilitate analysis and comparison, two representative days (January 26 and January 27) are selected to be analyzed. The two days are both sunny days, the outdoor air temperature varies from minus 6 ° to 5 ° C, the average temperature is about minus 1.3 ° C, the total solar irradiance varies from 400 W • m² to 500W • m² at noon, , the sunshine duration spans from 8: 00 to 17:00, 9 hours in all (Figure 4)

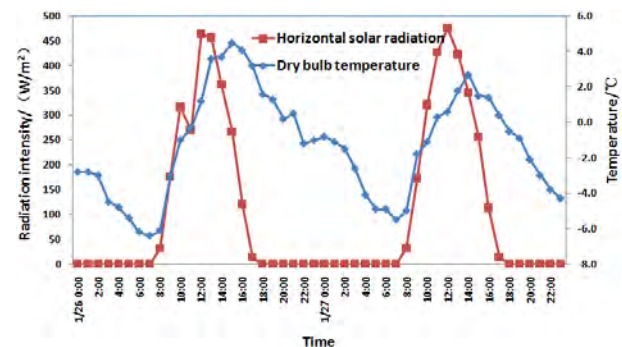


Figure 4 Outdoor air temperature and horizon solar radiation

3.2 The Planting Result

Since the planting of the experimental tomatoes on October 11, 2011, the morphological differences between the two greenhouses' crops during various growth periods have been gradually revealed. Compared with the common greenhouse 1#, the heights of the experimental crops in the PCM greenhouse 2# are all the way in the lead, having increased by 7.6cm,

22.6cm and 27.6cm on average during the periods of growing, flowering and fruiting [0].Table 1 shows the comparative results of the fruit quantity, length and the diameter as well as the output of the tomatoes.

Table 1 The tomato yields' comparison between the two greenhouses

Number	Transverse diameter t/mm	Longitudinal diameter /mm	The number	Output/(kg)	Per unit area yield/(kg/m ²)
Greenhouse 1#	60.2	42.0	7	0.45	0.7
Greenhouse 2#	45.0	31.5	25	3	1.2

3.3 Greenhouse Temperature Characteristic

3.3.1 Plough Layer Temperature

Greenhouse plough layer refers to the soil layer that extends from the soil surface to the 20cm depth of soil. Figure 5 shows the daily temperature variation of the soil surface and the 10cm and 20cm depth of soil in the two experimental greenhouses. When the heat preservation quilt is uncovered, the maximum soil surfaces temperature difference is 2.6 ° C and the average temperature difference is 1.8° C during this period. When the heat preservation quilt is covered, The maximum difference in soil surfaces temperature is 1.2 ° C and the average temperature difference is 0.8° C during this period. At 10cm-deep plough layer, the average soil temperature difference is 0.2° C when the heat preservation quilt is uncovered; the average

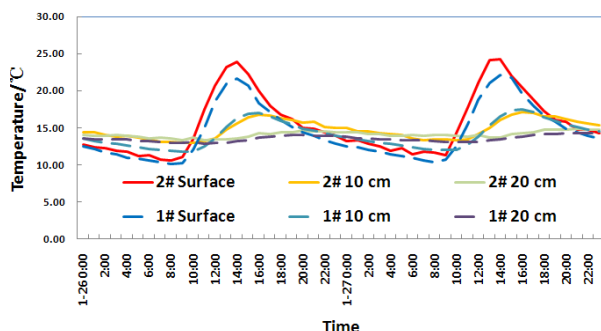


Figure 5 The soil temperature of the plough layer of the two greenhouses

soil temperature difference is 1.0° C when the heat preservation quilt is covered. At 20cm-deep plough layer, though the temperature throughout the day is fundamentally constant, the surface soil temperature in the phase change greenhouse2# is all the time higher than that of the common greenhouse 1#.The average soil temperature difference throughout the day is 0.6° C.

3.3.2 North wall inner-surface temperature

Figure 6 shows the surface temperature history of the north walls in the two experimental greenhouses. As shown in the figure, the surface temperature of the north

wall in PCM greenhouse 2# is all the time higher than that of the common greenhouse 1#.When the heat preservation quilt is uncovered (from 9:00 to 16:00), owing to the fact that the north wall of the PCM greenhouse 2# is influenced by the heat storage performance of the phase change wallboard, the maximum temperature difference is 3.4° C, and the average temperature difference is 3.2° C. When the heat preservation quilt is covered (from 16:00 to 9:00 of the next day), due to the fact that the heat that accumulated by the PCM in the daytime begins to be constantly released, the maximum temperature difference is 3.9° C, and the average temperature difference is 2.8° C during this period.

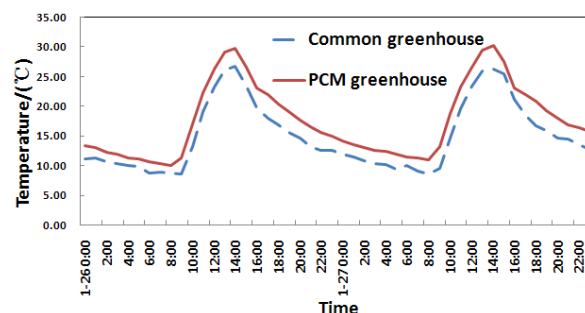


Figure 6 North wall inner surface temperature contrast

3.3.1 Air Effective Accumulative Temperature

Effective accumulative temperature of the air can reflect the crop's demand for heat during its growth and development. The method of calculating the effective accumulative temperature is heat time calculation method.[0] According to datum collected during the experiment, the statistics of the effective accumulated temperatures during various growth periods are gathered. Compared with the common greenhouse 1# (304.3° C · d), the accumulative temperature in the PCM greenhouse 2# (346.4° C · d) has a net increase of 42.1 ° C · d, thus the PCM has effectively improved the thermal environment needed for the growth of the crops.

4. "TRI-LAYER" STRUCTURE WALL THERMAL PERFORMANCE EVALUATION

4.1 Evaluation Index

4.1.1 Daily Effective Heat Storage (DEHST)

he effective heat storage capacity of the north wall is served as one of the key parameters in evaluating the thermal performance of the wall. The biological zero of tomato is 5° C. Therefore, merely the quantity of heat is counted when the wall temperature is greater than the biological zero[0].

Take the "tri-layer" structure wall as an example, when quilt is uncovered ($\tau_1, \tau_1 = 09:00$), suppose

the effective heat storage(EHS) of the wall at this point is Q_{τ_1} (Formula (1)). The wall will complete its process of heat accumulation when the heat preservation quilt is covered ($\tau_2, \tau_2=16:00$), the effective heat storage of the wall at this point is Q_{τ_2} (Formula (2)).

$$Q_{\tau_1} = Q_{PCM, \tau_1} + Q_{Brick, \tau_1} \quad (1)$$

$$Q_{\tau_2} = Q_{PCM, \tau_2} + Q_{Brick, \tau_2} \quad (2)$$

Where Q_{τ_1} and Q_{τ_2} are the effective heat storage of the wall (J) at τ_1 and τ_2 respectively; Q_{PCM, τ_1} and Q_{PCM, τ_2} are the effective heat storage of the GH-20 PCM (J) at τ_1 and τ_2 ; Q_{Brick, τ_1} and Q_{Brick, τ_2} are the effective heat storage of the brick layer (J) at τ_1 and τ_2 . The quantity of heat accumulated by the wall from 9:00 to 16:00 is the DEHST of the wall Q_x (Formula (3)).

$$Q_x = Q_{\tau_2} - Q_{\tau_1} \quad (3)$$

4.1.2 Daily Effective Net Heat Supplied(DENHS)

When the heat preservation quilt is covered, the walls in the solar greenhouse begin to release solar thermal energy to the greenhouse until the heat preservation quilt is uncovered ($\tau_3, \tau_3=09:00$). Suppose the effective heat storage of the wall at this point is Q_{τ_3} (Formula (2)), then the daily effective heat supply (SEHSU) of the wall is Formula (4)

$$Q_f = Q_{\tau_2} - Q_{\tau_3} \quad (4)$$

Suppose the cumulative time is $\Delta\tau$ (s) ($\Delta\tau = \tau_3 - \tau_2$), then the accumulative heat loss of the wall's external surface released to the outdoor in corresponding period is Q_{loss} (J) (Formula (5)), the DENHS to the greenhouse is Q_{net} (Formula (6)).

$$Q_{loss} = \int_0^{\Delta\tau} qAd\tau \quad (5)$$

$$Q_{net} = Q_f - Q_{loss} \quad (6)$$

4.1.3 GH-20 PCM's Contribution

To evaluate the GH-20 phase change heat storage material's contribution to the of the greenhouse wall's utilization rate of solar energy, three evaluation parameters (heat storage ratio, heat release ratio and the rate of heat release) have been introduced, the corresponding calculation formula is shown in formula (7).

$$\eta_x = \frac{Q_{x,PCM}}{Q_x} \quad \eta_f = \frac{Q_{f,PCM}}{Q_f} \quad \eta = \frac{Q_{f,PCM}}{Q_{x,PCM}} \quad (7)$$

Where $Q_{x,PCM}$ and $Q_{f,PCM}$ are the daily effective heat storage capacity and the net heat supply of the GH-20 PCM.

4.2 Evaluation and analysis of results

4.2.1 The North Wall's Distribution Characteristics of Temperature Field

The characteristics of the time-variable temperature field of the wall are supposed to be made certain in order to solve the various heat indexes. Therefore, a heat transfer model with "tri-layer" structure wall has been built by this research team. The measured values of the wall's internal and external surface are directly taken as the boundary conditions. Effective heat capacity method is applied to solve the problem of PCM heat transfer; On the basis of the computational analysis software, and take the actual measurements of the north wall's internal and external surface in Figure 3b) as boundary conditions to substitute.

4.2.2 The Dynamic Characteristics of the Heat Storage-release

On the basis of the calculation results in section 4.1.1 and Formula (1) , Formula (2), the hourly effective heat storage (Figure 7) of two greenhouses' north wall during the daytime can be calculated; As shown in the figure, 1) the rate of heat storage of the PCM greenhouse 2#'s north wall is all the time higher than that of the common brick greenhouse 1# from 13:00 to 15:00; 2) The EHS of the common greenhouse 1# reaches its peak (the end of the heat storage) at 15:00, while the PCM greenhouse 2# reaches its peak at 18:00, the time for heat storage of the common greenhouse 1# is 3 hours longer than that of the former.

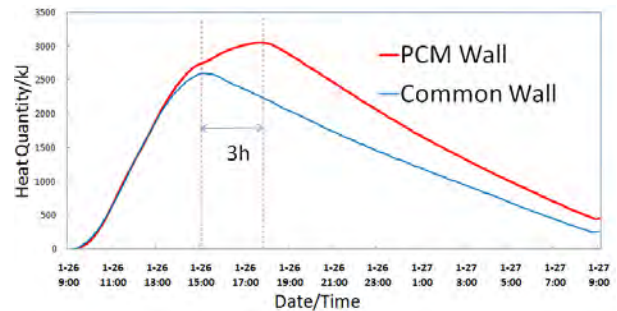


Figure 7 The dynamic change curve of the heat on the north wall

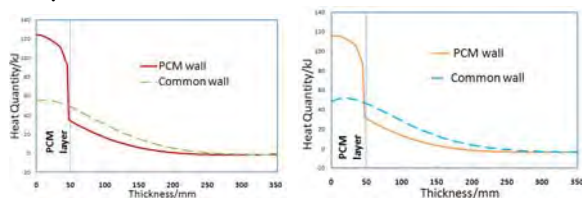
According to the analysis results(section4.2.1), the influence of the solar radiation on the internal surface of the two greenhouses are mainly at the depth of 350mm, thus a selective analysis of the daily effective heat storage- release of this area.

(1) The comparison of the DEHST of the north wall

As shown in the figure, the DEHST of the north wall in the PCM 2# (3019KJ) has increased by 18 percent compared with the common greenhouse # (2570KJ).

(2) The DENHS of the north wall

The DEHSU of the north wall in the PCM greenhouse 2# (2719KJ) has increased of 14 percent compared with that of the common greenhouse 1# (2380KJ). On the basis of Formula (5) and Formula (6), it is calculated that the DENHS of the north wall in the greenhouse 2# (1923KJ) has increased by 20 percent compared with the common greenhouse 1#.



a) daytime heat storage b) nighttime heat supply
Figure 8 DEHST and DEHSU of the wall

4.2.3 The Contribution Rate of GH-20 PCM

In accordance with the calculation results in section 4.2.2 as well as Formula (7) and Formula (1) --- (6), the heat storage capacity of the GH-20 PCM in the daytime is 2261 KJ by calculation, which accounts for 75 percent (η_x) of the wall's total heat storage; the heat supply and release of this material is 2126 KJ, which accounts for 78 percent (η_f) of the wall's total heat supply and release, the heat release rate (η) of the GH-20 PCM reaches up to 94 percent, the effective release of the heat storage within a day is well achieved.

5. CONCLUSIONS

1) The "tri-layer" structure wall can significantly improve the production quality and output of crops. Compared with the common greenhouse 1#, the crop yields per unit area in the PCM greenhouse 2# are 1.7 times as much as that of the former.

2) The "tri-layer" structure wall can effectively improve the thermal environment of the crops. Compared with the common greenhouse 1#, the effective heat storage of the PCM greenhouse 2# has increased by 42.1° C/D, the north wall inner surface temperature and plough layer temperature has also been improved significantly;

3) The analysis results show that the DEHST of the "tri-layer" wall structure in the greenhouse has increased by 18%, the DENHS has increased by 20% compared to the common greenhouse, and the time of effective heat storage has increased by 3 hours.

3) The GH-20 PCM has played a vital role in improving the heat storage capacity of the solar greenhouse's north wall. The heat storage-release quantity of the GH-20 accounts for 75 percent and 78 percent respectively of the entire "tri-layer" structure wall, and the rate of heat release has achieved 94 percent.

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RESEARCH ON THE PHASE CHANGE SOLAR ENERGY FRESH AIR THERMAL STORAGE SYSTEM

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ABSTRACT

In this paper, a new kind of solar fresh air system is designed in order to realize the improvement of thermal efficiency by the integrated application of the PCMs and heat pipe technology. Under the adequate sunshine condition, the fresh air is directly delivered into the indoor environment after being heated by the solar collector. During this time, the solar energy is served as the main heat source which isothermally afferent the surplus heat to the phase change heat storage equipment by heat pipe; When the sun radiation is reduced, the heated air temperature cannot satisfy the requirement of supply air temperature, and then the main heat source is changed to phase change heat storage equipment instead of solar energy. Heat pipe and part of the PCMs is encapsulated as an integration to finish the heat exchange in serpentine duct. The system adopt heat pipe for a high-efficiency and isothermal heat transfer which recover the shortcomings of PCMs such as: low coefficient of thermal conductivity and poor thermal efficiency. This article established the physical model of phase change solar energy fresh air thermal storage system and creates the mathematical model of its unsteady heat transfer to simulate and analysis the operation process by using fluent software. The results of the study show that, compared to normal fresh air system, the phase change solar energy fresh air thermal storage system has a significant improvement in energy saving and indoor comfort level and will play an important role in the energy sustainable development.

Keywords: Solar fresh air system, Heat pipe, Phase change thermal storage, Simulation analysis

1. INTRODUCTION

The problem of improving energy efficiency and completing the transformation of the traditional energy to green energy is the current research focus, in order to achieve the target of reducing energy consumption and improve the quality of life, a new kind of phase change solar energy fresh air thermal storage system is the ability to achieve air supply of new air after double efficient heating by simulation.

In the north cold regions, the average outdoor temperature in winter is often in minus 10 to minus 20 degrees, due to requirements of energy saving, sealing requirements of doors and windows is increasing, meanwhile, in order to ensure the requirement of indoor temperature in winter, interior doors and windows often closed, air does not flow well, although the temperature reaches the heating requirements of design, from the comfort point is not conducive to people's health. In the fenestration behavior research on cold area in winter in 2013, statistics show that accounted for 80% of residential windows open for ventilation, such as heating temperature is not enough and other reasons for choosing do not open the window accounted for 20% (see Figure 1), this shows that people still hope for a certain number of windows open for ventilation every day to improve air quality in the living room.

Per capita area of residence is usually larger currently, so the design is usually selected the number of ventilators 0.3 times one hour as winter fresh air ventilation standards, considering the requirements of energy conservation and the new air requirements, indoor fresh air ventilation is usually taken 25 m²/(h•people). But the actual situation and the design requirements are usually not the same, people will open the window for short indoor ventilation when in a period of time before leaving the room or indoor room exist peculiar smell, which is fall far short of the time to improve indoor air quality.

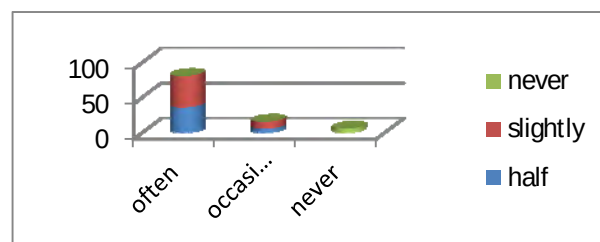


Figure 1 Statistics of research results

Solar new air technology is already very mature, run to achieve an average of more than 75%, in the sunny day of well solar radiation solar new air system

can get to from 60% to 75% of solar energy, transformed by the light and heat of the collector plate, the general air temperature will rise 15 to 40°C, which is higher than the indoor temperature, and can be directly used on the indoor ventilation. Outdoor fresh air enter from the air intake and heated by the collector plate, reach the highest temperature on the top, then pumped into the heat transfer equipment to heat up again and send into the room[1-3].

However, due to the heat extraction and storage of the system with air as the medium, this traditional convective heat transfer constrained by convection coefficient, medium flow velocity factors, the overall efficiency will decline, and will inevitably need to increase the volume of heat transfer equipment to reach the heat exchanger requirements, which will sacrificing interior space, while the low temperature of new air also seriously increases building energy consumption [4].

2. THE PHASE CHANGE SOLAR ENERGY FRESH AIR THERMAL STORAGE SYSTEM

Based on the above problems, innovation lies in the design of new air system described in this article are select heat pipe as a heat transfer means, phase change energy storage apparatus as a secondary heat source to ensure the thermal comfort of new air room. The basic working principle as shown in Figure 2.

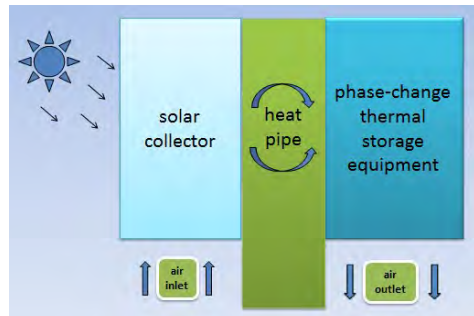


Figure 2 Working principle

Where in the heat transfer of the heat pipe portion shown in Figure 3. The fluid in the tube-shell vaporized evaporate endothermic in the case of only one end heated, expansion in the form of steam, condensate exothermic conversion to liquid after flowing through the adiabatic section, this heat absorption by wick, the performance is heat transfer to the other end of the heat pipe, liquid fluid flow back to the evaporator section in the capillary forces to complete a exothermic cycle.

The advantages of heat pipe are:

1. Fast heat conduction speed, high strength, high efficiency, heat conduction can even reach to velocity of sound
2. Good isothermal: Pass a lot of heat in a small temperature difference and heat transfer resistance is small

3. The Safety Reliability: there is no overpressure in the pipe, and not afraid to dry burning. After the vaporization of the working fluid, the inner pressure of the heat pipe will not change with temperature change.

4. Reversibility of the direction of heat flow: due to its internal cycle power is the capillary force, either end heated can be the evaporating section, and the other end outwardly heat dissipation will be the condensation section.

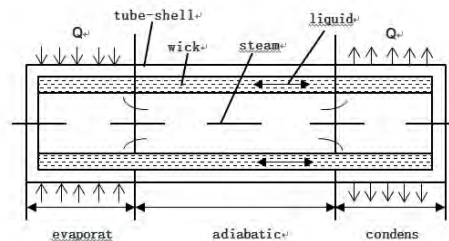


Figure 3 Heat pipe working principle

The studies show that in the case of without efficient heat transfer thermal device, even if the weather is sunny and solar radiation sufficient, fresh air only after warm-up of solar energy new air collector and then directly send into the room, only one hour time around 1:00 does not increase the indoor heat load, the other most of the time causes the indoor temperature fluctuations[5-7].

This structure of phase change solar energy fresh air thermal storage system is shown in Figure 4, the inside part of the device according to the design position of heat pipe fixed the phase change material and heat pipe to a rectangular baffle shape, divided the space into a serpentine flow channel (see Figure 5), the working principle is: With the gradual increase of the solar radiation, the temperature of the collector plate is rising, the heat gathered in the heat collecting plate when the new air is heated, at the same time the heat pipe isothermal pass for the heat accumulator of phase change material, fresh air was leaded into the heat exchanger equipment through the fan and flow in the flow channel, make the heat transfer with the phase change material and send into the interior through the outlet. When solar radiation weakened, the temperature of the heat pipes is lower than the phase change material melting point, the phase change material exothermic heat fresh air as the primary heat source. New air continued heated reservoir until the heat in phase change material is release all, this system prolonged the time of ventilation, Contrast with traditional air heating phase change materials enhance efficiency obviously; weather fluctuations will not affect the fresh air heat exchanger, guarantee the stability of the indoor temperature, maximum the human comfort[8-9].

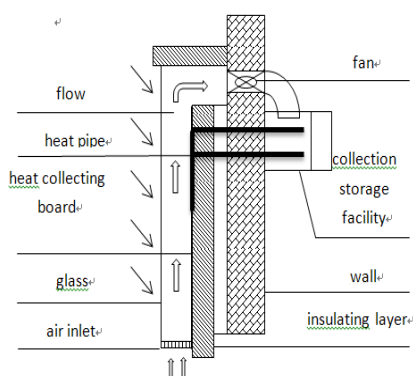


Figure 4 System structure diagram

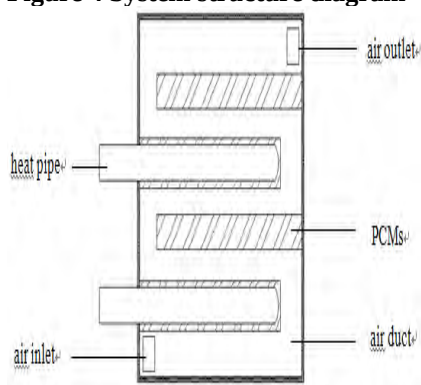


Figure 5 Equipment internal structure

3. SIMULATION CALCULATION

For the above system, using the FLUENT software simulates the following four conditions:

Condition 1: Time periods ranging from 9:00 am to 11:00 am, average solar radiation intensity is about $305.56 \text{ (W/m}^2\text{)}$ [10]. At this time the whole system is in the warm-up phase, heat of collector panels extracted from the heat pipe evaporator section unable to reach the phase transition point of the phase-change material, and does not making indoor ventilation. Outdoor air temperature increased after going through collector plates and heat transfer equipment, but did not reach the ventilation standards. The results of simulation show that heat collecting on the collector plate is less in this time period, secondary heating effect of heat transfer equipment is not obvious, new air unable to meet the minimum supply air temperature through heat exchanger, at this time heat exchanger equipment is in the warm-up phase state (see Figure 6).

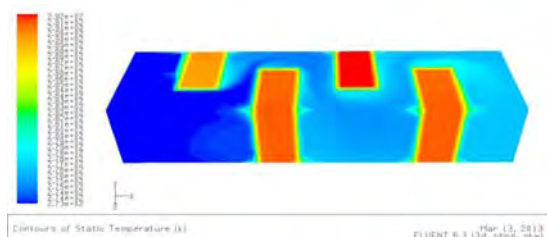


Figure 6 Warm-up phase of the heat transfer equipment

Condition 2: From 11:00 am to 14:00 pm, in the strongest period of solar radiation, solar radiation intensity reached the peak of $396.30 \text{ (W/m}^2\text{)}$, at this time the temperature of collector plate is much higher than the phase transition point temperature [11]. The heat pipe efficient spread heat of the collector plate to the phase change material, then phase change material storing heat, while excess heat continue to transfer to the new air in the flow channel through phase change materials, the temperature of secondary heating new air has reached the blast standard, then send into the interior for ventilation. The results of simulation show that heat collected from the collector plates reaches the maximum, the highest temperature of new air in the air collector is more than 50 degrees. With efficient heat transfer through the heat pipe, sufficient heat can be supplied to the heat transfer equipment. At this time new air temperature inside the device has exceeded the minimum supply air temperature, and can directly send into the interior (see Figure 7).

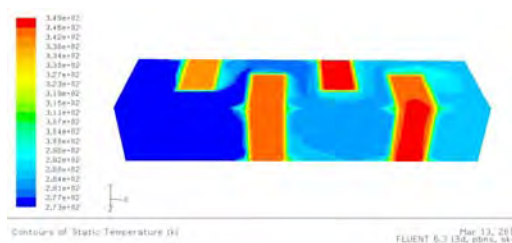


Figure 7 Strongest stage of effect of heat transfer in heat transfer equipment

Condition 3: From 14:00 pm to 16:00 pm, average solar radiation intensity is about $208.33 \text{ (W/m}^2\text{)}$. At this time solar radiation weakened stage, after efficient heat storage in the two hours at noon, phase change material has been completely fills up with the heat, with the weakening of the solar radiation, and the temperature of the collector plate has dropped. When the temperature is reduced to less than the phase transformation point, the heat pipe is no longer available to conduct heat to the heat storage device, at this time secondary heat to heat the new air is completely from the phase change material in the noon. Without this equipment, and only rely on solar air collector for new air heating, the fresh air cannot be directly sent into the room, because the new air temperature is lower than the indoor heating temperature, and after efficient heat the phase change material has overcome itself shortcomings of low heat transfer coefficient, has the ability to fills the heat in a short period of time, extended time for ventilation, improve the indoor air quality [12] (see Figure 8).

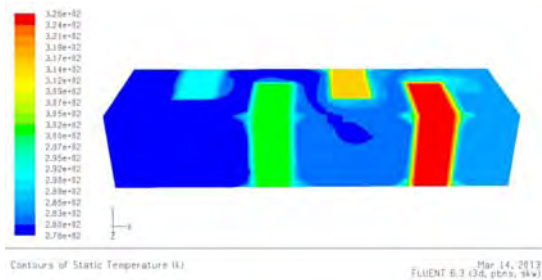


Figure 8 Exothermic initial stage of heat transfer equipment

Condition 4: From 16:00 pm to 18:00pm ,Heat storage in the phase change material will gradually release completely, and temperature of new air is gradually reduced too, the system will shut down after reduced to indoor heating temperature, at this time indoor temperature remains in the heating temperature, so it will not cause the indoor temperature fluctuations for ventilation and do not affect the indoor normal life, thus avoiding the discomfort situation caused by the windows open for ventilation[13], while improving air quality (see Figure 9).

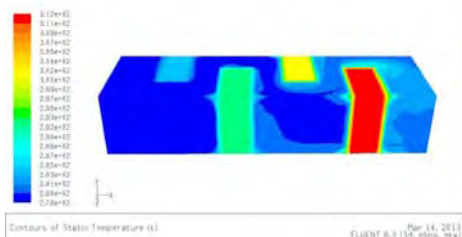


Figure 9 End stage of exothermic process

4. CONCLUSIONS

Through research statistics, we get the actual situation of the average residential indoor ventilation in cold regions in winter, learned that low outdoor ambient temperature and other reasons forced people to reduce the ventilation, and there is a widespread dissatisfaction with the low air quality of indoor which is caused by lack of air circulation. In order to improve residential air quality, improve the indoor comfort, the article designs a phase change solar energy fresh air thermal storage system. Through the initial simulation found that this system can extend ventilation time under the premise of without increasing winter residential heating load, proposed an entirely new way of thinking for fresh air ventilation in cold regions.

ACKNOWLEDGMENTS

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INDICES AND METHOD TO ASSESS PERFORMANCE OF THERMAL STORAGE DEVICES

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ABSTRACT

Thermal energy storage (TES) is becoming one of the key technologies that may reduce the energy consumption of maintaining thermal comfort and producing hot water in buildings. Different structure design of TES and different phase change material (PCM) forms are all developed and investigated for optimization of integrated system. Various methods have been investigated by some researchers to reduce the supercooling degree, which is related to the thermodynamic metastable phenomenon. In these years, there are many published paper about TES with PCM as the increasing application in building, especially focus on optimization design of TES system and enhancement of energy storage performance of PCMs. This paper presents a new set of performance indices for active latent-heat storage devices, namely the total storage capacity (kJ), and the volumetric storage capacity (kJ/m³). In this paper, the applications of PCMs are reviewed firstly and different types of thermal storage techniques and TES system are summarized as well, including different structure of TES system, different PCM forms and novel PCM materials for TES system. In order to study enhancement of energy storage performance of PCM, a computational fluid dynamics (CFD) model is developed to study coupled thermal progress of straight pipe between inside stream and outside material in TES system. Heat transfer process between fluid inside pipe and outside material is numerically studied and two presented indices are compared respectively. Comparison studies on CFD results and analyzed calculation results under different typical conditions are carried out as well. Finally, under actual operation conditions, temperature field and energy analysis are carried out for the heat charging and discharging period of straight pipe with ground heat exchanger. Temperature curves of inside and outside parts with time are drawn during charging and discharging period.

Keywords: Phase change material (PCM), Thermal energy storage, Energy saving, Performance indices.

1. INTRODUCTION

In the modern period, building energy consumption becomes one of the top sectors of total consumption of whole society. Around 40% of the total fossil energy was used for all kinds of building services in the United States of American and European Union [1]. Thus, in order to save building energy consumption, thermal energy storage is more and more becoming the hot research topic in these years. According to the energy storage media, thermal energy storage has two parts: one is use of sensible heat storage and another is latent heat storage.

Energy storage has many forms. Fossil fuels, the use of which the basis of the industrial revolution, are a chemical form of energy storage. In a pump hydro, water is pumped up a reservoir at night time and flows down to run a turbine during day time to generate electricity. However, as reviewed recently by MacCracken et al. [2], cooling and thermal storage may prove to be the least expensive, when measured using capital cost per unit energy or power, and most energy efficient. Thermal energy storage material like ice, snow, rock had been already used in living field many years ago [3]. Using latent heat storage in building applications can meet the demand for thermal comfort

inside building and energy storage purpose. As the refrigeration and energy storage technology developing, phase change material (PCM) has been studied by researchers and applied in many fields, especially in building energy saving applications. According the different energy demand and balance, the TES system based on PCM can be employed to charge, store and discharge cold or heat energy between days and nights or peak and off-peak time (i.e. weekly, monthly, seasonally). Latent energy can be stored/released through melting/solidification processes by thermal energy storage system (TES). A PCM based TES system can store heat or cold energy at a small temperature difference between the charging source and discharging user, and can potentially complete the charging in a short period.

Dincer and Rosen [4] gave a comprehensive and overall description of thermal energy storage including general introductory, thermal energy storage systems and methods, energy and exergy analysis, numerical simulation of TES systems and other research development in TES methods, technologies and applications.

The thermal energy storage (TES) system for building cooling applications is a promising technology that is continuously improving. The TES system can balance the energy demand between the peak (daytimes) and off-peak hours (nights). A well-designed TES system would effectively decrease the electricity demand with a reasonable cost. Y.H. Yau et al. [5] summarizes the findings, investigations and analysis of the TES systems for the space cooling applications. They advised that the designer must focus on the best practical and economical solution influenced by localized parameters.

Zhaolin Gu et al. [6] tried to recover the rejected heat from air conditioning systems by latent heat thermal energy storage systems which can be used to get low-temperature hot water. A recovery system using phase change materials (PCMs) to store the rejected (sensible and condensation) heat from air conditioning system has been developed and studied.

2. THERMAL STORAGE TECHNIQUES

In the ancient, people had recognized keeping hot rock to warm their body and air inside room, and storing cold ice from high mountains or frozen river during winter to use it in hot summer. Recently, some researchers carried out the investigation on common feature of modern air-conditioning system design. Jianlei Niu et al. proposed that three main functions of ventilation, dehumidification and sensible heat removal can be assigned to different components in an air-conditioning system, in order to provide better energy efficiency and a better overall indoor environment quality [7-8]. It is found that horizontal type ceiling panels can handle a space sensible cooling load of up to 80 watts per square meter, which should be adequate for most office buildings which comply with energy saving envelope design standards. The relatively high chilled water temperature required for the chilled ceilings, typically above 17 °C, can significantly raise the efficiency of a chiller which operates at the much higher evaporator temperature of 12 °C, in comparison with the 5 °C setting of the conventional design, and extend the period of using natural cooling. With such a system, fan energy is only needed to supply the outdoor ventilation air. There are many experimental and simulation studies on the PCM characteristics of melting, solidification and heat transfer processes in TES system. Pablo Dolado et al. [9] carried out the experimental studies to test thermal cycling of a real-scale PCM-air heat exchanger. The influence of inlet air temperature and air flow was studied and an empirical model was developed from the experimental results, which could be a useful designing tool for applications that use such technology: green housing, curing and drying processes, plant production, HVAC and free-cooling. Dong Zhang et al. [10] made thermal energy storage aggregates (TESAs) from porous aggregates absorbing phase changing materials (PCMs) firstly. Then thermal energy storage concrete (TESC) was produced with a normal mixing method and using TESAs. The experimental results showed that the

geometrical features of the porous structure of the aggregates had significant effect on their absorbing ability of the PCM. Aggregates with large pore connectivity factor and transport tunnel in boundary part can absorb more PCM materials and the phase changing behavior was affected by the volume fraction of PCM in concrete. Viktoria Martin et al. [11] tried to enable high power fir charging and discharging while providing a high storage capacity. They presented one concept using PCM for storing of cold, where the cold carrier (water) was in direct contact with the PCM. C.Y. Zhao et al. [12] summarized recent research works on latent thermal energy storage in building applications, covering PCMs, the impregnation methods, current building applications and their thermal performance analysis and numerical simulation of buildings with PCMs.

Ibrahim Dincer et al. [13] carried out energy and exergy analyses for the charging period of an ice-on-coil thermal energy storage system. The present model was developed using a thermal resistance network technique. The results indicated that working and design parameters of the ice-on-coil thermal storage tank should be determined by considering both energetic and exergetic behavior of the system. They reported that storage capacity and energy efficiency of the system increases with decreasing the inlet temperature of the heat transfer fluid and increasing the length of the tube and the exergy efficiency increases with increasing the inlet temperature of the heat transfer fluid and increasing the length of the tube. Yaling He et al. [14] developed a mathematical model for the overall exergetic efficiency of two phase change materials named PCM1 and PCM2 storage system with a concentrating collector for solar thermal power based on finite-time thermodynamics. It is found that melting temperatures of PCM1 and PCM2 have different influences on the overall exergetic efficiency, and overall exergetic efficiency decreasing the melting temperature of PCM1, increasing the melting temperature of PCM2. H. Huseyin Ozturk et al. [15] carried out a seasonal thermal energy storage using paraffin wax as a PCM with the latent heat storage technique for heating the greenhouse of 180 m² floor area.

PCM materials have been also applied to storage cold thermal energy from cryogenics liquid flow. Recently, liquefied natural gas (LNG) is more and more becoming important energy source and it has large cold energy when it is gasified. Yanzhong Li et al. [16] developed an experimental investigation of the cold storage with liquid/solid phase change of water based on the cold recovery of LNG refrigerated vehicles. The research work presented valuable references for the design and optimization of cold storage unit of LNGV and for the numerical simulation on the unsteady two-dimensional conjugated heat transfer with phase change. They also concerned freezing process of PCM used for recovery and storage of cryogenic gas cold energy. Experimental study and CFD simulation on the above process were carried out respectively.

3. DIFFERENT TYPES OF TES SYSTEM

Vineet Veer Tyagi et al. [17] presented a comprehensive review of various methods for gaining heat and cold in buildings. The energy performance of different types of systems like PCM wallboards, PCM shutters, PCM building blocks, air-based heating systems, floor heating, ceiling boards, etc., were also summarized. Akio Saito et al. [18] reviewed recent literatures in the field of cold thermal energy storage (CTES). Various types of the CTES are defined and compared as for their characteristics. Several pairs of cold storage systems, a cold water storage vs. an ice-making storage, a static ice-making storage vs. a dynamic ice-making storage and direct ice-making vs. indirect ice-making storage, are compared with each other respectively.

Ahmet Alper Aydin et al. [19] presented a kind of novel organic phase change materials for thermal energy storage. The thermal properties were given about phase change temperature, enthalpy, specific heat and thermal decomposition temperature with related statistical calculations. The differential scanning calorimeter (DSC) analyses indicated that the melting points of the novel organic PCMs were between 40 and 50 °C with phase change enthalpy above 200kJ/kg. The research results showed that the novel materials were suitable for low temperature heat transfer applications with superior thermal properties and reliability among the other PCMs. Another kind of novel shape-stabilized PCM was introduced by Y.P. Zhang et al. [20]. The preparation for the above material and the measurement of thermophysical properties were presented in this paper. Simulation models of analyzing the thermal performance of the systems were developed and validated with the experimental studies. A new kind of phase change material (PCM) for energy-storing wallboard was presented by Haifeng Guo et al. [21]. They established the one-dimensional non-linear mathematical model for heat conduction of PCM energy-storing wallboard and according to the “effective heat capacity method”, simulation and calculation were made using the software MATLAB to analyze and solve the heat transfer problem of the PCM room. They found that the heat storing/releasing ability of the new PCM is significantly higher than that of ordinary materials by experimental study. The result showed that applying proper PCM to the inner surface of the north wall in the ordinary room can not only enhance the indoor thermal-comfort sharply, but also increase the utilization rate of the solar radiation.

4. INDICES OF THERMAL STORAGE DEVICES PERFORMANCE

In order to enhance the performance of TES system, different method and research have been reported. Microencapsulation is also one kind of the advanced technologies for better utilization of PCMs with building parts within the building materials. V.V. Tyagi et al. [22] presented an overview of the previous

research work on microencapsulation technology for thermal energy storage incorporating the PCMs in the building applications.

Use of phase change material (PCM) capsules assembled as a packed bed is one of the important methods that have been proposed to achieve the objective of high storage density with higher efficiency.

As its effectiveness and acceptance widely, suitable indices to assess performance of TES system is one important research point as well.

4.1 Physical model and numerical algorithm

In this paper, as an example, coupled heat transfer problem with fluid flow inside pipe and conduction outside is studied by CFD simulation method. The indices to assess performance of thermal storage devices are proposed. Figure 1 shows the simulation model.

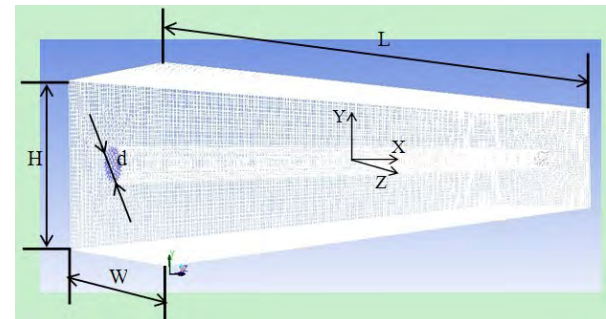


Figure 1 Simplified physical model

The pipe volume is V_f and the tank volume is V . The pipe diameter is d (0.04m) and the total length of the pipe (X) is 1 m. The outer surface of the tank is adiabatic. The width (Z) and height (Y) of tank are same and equal to 0.1, 0.2 and 0.3m respectively in this study which mean the spacing between two parallel pipes inside storage tank. The volume configurations are defined with V/V_f ratios equal to 7, 31 and 71 respectively. The fluid keeps water inside pipe and the materials outside pipe are water, soil and PCM material respectively.

The temperature field and heat transfer process were simulated as transient coupled heat transfer process by CFD method. Single-phase flow inside pipe is concerned. The standard $k-\epsilon$ model is selected when it is a turbulence flow case. Mass conservation equation and general form of governing equation are as follows:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \mathbf{u} = 0 \quad (1)$$

$$\frac{\partial (\rho \phi)}{\partial t} + \text{div}(\rho \mathbf{U} \phi) = \text{div}(\Gamma_\phi \text{grad} \phi) + S_\phi \quad (2)$$

Where: ϕ is the general variable which represents u, v, w, T, K, ϵ ; Γ_ϕ is the generalized diffusion coefficient; S_ϕ is the generalized source term.

The energy balance equation is:

$$\rho \frac{D}{Dt} \left(e + \frac{1}{2} u_j u_j \right) = \frac{\partial \sigma_{ij}}{\partial x_i} u_j + \sigma_{ij} \frac{\partial u_j}{\partial x_i} + \rho u_j f_j - \frac{\partial q_j}{\partial x_j} \quad (3)$$

Where: e is internal energy per mass flow (J/kg); q is heat flux (W/m²).

For the heat transfer process of materials outside pipe, conduction and nature convection is used for the solid and liquid. The solidification/melting model is used for cases that materials outside pipe are PCM. Enthalpy-porosity formulation is used for simulating the solidification process and The “volume of fluid” (VOF) model is also used to describe the PCM solidification and melting process in the commercial software Fluent 13.0. Thus, the governing equations for phase change process are as follows:

Continuity:

$$\frac{D\alpha_n}{Dt} = 0 \quad (4)$$

Momentum:

$$\rho \frac{DV}{Dt} = -\nabla p + \mu \nabla^2 V + \rho g + S \quad (5)$$

Thermal energy:

$$\rho \frac{DH}{Dt} = k \nabla^2 T \quad (6)$$

Where, V is the fluid velocity vector, ρ is the density, μ is the dynamics viscosity, P is the pressure, g is the gravitational acceleration, k is the thermal conductivity and H is the specific enthalpy which is defined as follows:

In this study, simplifying and assumptions are applied for the simulation study on heat transfer process of TES system. The assumptions of simulation model are following: (1). The fluid flow inside pipe is stable whatever it is laminar or turbulence; (2). The heat transfer mechanism is conduction when TES tank around pipe is solid state. Nature convection is adopt when the material of tank is liquid state; (3). The fluid inside pipe is treated as incompressible Newtonian fluid; (4). Heat radiation between fluid and solid is neglected. Thermal conductivity alone axial of pipe is neglected as well. The boundary conditions for series of cases in this paper are listed in Table 1.

Figure 1 Simulation cases and boundary conditions

Case No.	Materials	Tank size	Mesh number	Size and boundary conditions
Case 1-1	Water/Water			
Case 1-2	Water/Soil	0.1m×0.1m×1.0m	363303	
Case 1-3	Water/PCM			1, Pipe size: d=0.04m, L=1m;
Case 2-1	Water/Water			2, Inside pipe: Vin=0.183m/s; Tin=285K;
Case 2-2	Water/Soil	0.2m×0.2m×1.0m	339894	
Case 2-3	Water/PCM			3, Outside pipe: Initail T=300K; Tank wall keeps heat insulation;
Case 3-1	Water/Water			4, Interface: Copper; Coupled temprature.
Case 3-2	Water/Soil	0.3m×0.3m×1.0m	740000	
Case 3-3	Water/PCM			

4.2 Results and discussion

The total energy storage and volumetric storage capacity at time of 60 minutes under different V/Vf values are shown in Figure 2 and Figure 3 respectively.

The results show that total energy storage are increased with the increase of the ratio between V and Vf and volumetric storage capacity are decreased with volume ratio increasing as well. In addition, growth rate of total energy storage is slow down with the increase of volume ratio and the slope of volumetric storage capacity become gentle with volume ratio increasing from 7 to 71.

In this paper, review on development and optimization of thermal energy storage with PCM is carried out firstly. Simplified simulation model of TES system on heat transfer process is studied and coupled heat transfer of conduction and convection is developed as well. The results shows CFD method can solved these coupled heat transfer problem and indices of total energy storage and volumetric storage capacity can access the performance of thermal storage devices effectively.

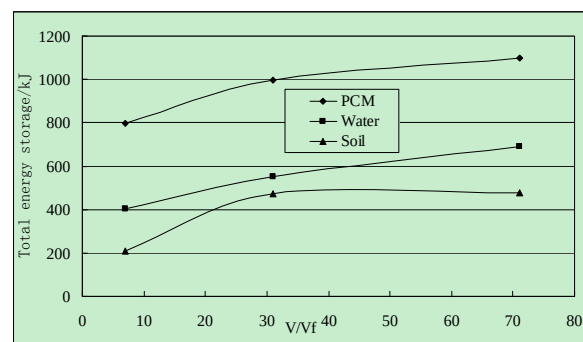


Figure 2 Total energy storage with the increase of V/Vf values

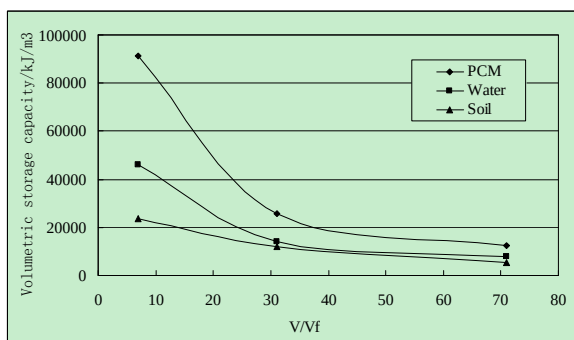


Figure 3 Volumetric storage capacity with the increase of V/Vf values

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ENERGY AND ECONOMIC ANALYSIS ON BUILDING ENVELOPE WITH PHASE CHANGE MATERIAL IN SUMMER

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ABSTRACT

Latent heat energy storage systems have received increasing attention because of the large amount of heat absorption/release during the phase change processes. The one-dimensional heat transfer of building envelop with phase change material is analyzed with moving heat-source method in this paper. And the calculation is verified by experimental data. Combined with the analysis of meteorological parameters in five cities of China, the energy and economic savings both by natural cold resource and air conditioning system are evaluated. Electricity saving ratio and economic recovery period are developed to assess the application of phase change material in buildings. As last, the selection of phase change material for different local climate is analyzed, using economic recovery period as index. It is suggested that phase transition temperature should be 3 °C higher than mean outdoor temperature, making the investment worth.

Keywords: Phase change material, Building envelop, Electricity saving ratio, Economic recovery period, Phase transition temperature

1. INTRODUCTION

Building energy consumption has continued to increase, reaching 28% of China's total energy consumption in 2000, which is close to the level of about 30% in developed countries [1-2]. Most air conditioning and refrigeration systems are sized to meet the maximum level of demand, which occurs only a few hours per year, and thus spend their operational life working at reduced capacity and low efficiency, resulting in huge energy consumption [3]. Latent heat energy storage systems have received increasing attention for cooling in buildings because of the large amount of heat absorption/release during the phase change processes, with only small temperature variations. Since phase change materials (PCMs) in buildings can store cool energy by using nighttime cheap electricity or free one from nature environment, and use it with a time delay for space cooling in summer. Therefore, they can be used for peak cooling load shifting [4] providing significant economic benefit [5].

Evola et al. [6] investigated the PCM board for summer thermal comfort with night ventilation and the intensity of thermal discomfort could be reduced by 31.2% in Chambery and 15.1% in Catania. S. Takeda [7] proposed a ventilation system utilizing PCM to reduce ventilation load. The results obtained from these studies have demonstrated that the proper use of PCMs in buildings can help reduce energy consumption and improve thermal comfort. However, it is difficult to

predict the energy conversation potential and economic feasibility.

In this work, moving heat-source method was used to evaluate the location of solid-liquid interface and heat flux during phase change process. The heat transfer process was verified by experimental data. Five cities that cover a wide range of different climatic conditions were selected in China. The energy savings and electricity savings of the application of PCM in building were investigated for three summer months from 1st June to 31st August. Energy saving ratio by natural cold resource, electricity saving ratio (ESR) by air conditioning system and economic recovery period (ERP) were developed to assess this application. With ERP as index, the selection of PCM for buildings was obtained to conduct the design.

2. MATHEMATICAL MODEL AND EXPERIMENTAL VERIFICATION

2.1 MATHEMATICAL MODEL

Moving heat-source method was first proposed by Lightfoot [8] to solve the transient phase change problem. Despite widespread applications of these types of problems, they have no general solutions [9]. The problem to be analyzed is shown in Fig.1, where the PCM board is attached on the inside surface of wall. In order to simplify the calculation, the following assumptions are introduced in this analysis:

The phase change is one-dimensional and conduction dominated.

Material properties of the solid and liquid phases are different, but are assumed to remain constant within each phase.

It is assumed the PCM changed phase at a single temperature.

The fluid in the liquid PCM is ignored.

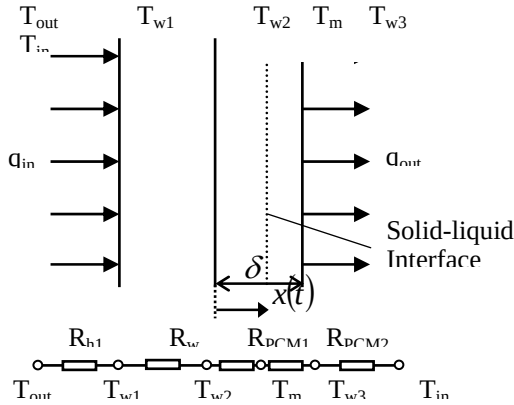


Figure.1 Mathematical model for wall with PCM board

For cold energy storage process, the heat flux is as follows:

$$q_{in} = \frac{T_{w2} - T_{out}}{R_{h1} + R_w} = \frac{T_m - T_{w2}}{R_{PCM1}} \quad (1)$$

$$R_{PCM1} = \frac{x(t)}{k_s} \quad (2)$$

Where q_{in} is the heat flux for energy storage process (W/m²); T_{w2} is temperature of the wall that PCM attaches on, used as boundary temperature for energy storage process (°C); T_{out} and T_m are the outdoor temperature and freezing temperature of PCM (°C); R_{h1} , R_w and R_{PCM1} are outdoor heat convective resistance, wall conduction resistance and PCM conduction resistance (m²°C/W); $x(t)$ is the location of solid-liquid interface (m); k_s is the heat conduction coefficient of solid PCM (W/m/°C).

For cold energy release process, the heat flux as shown:

$$q_{out} = \frac{T_{in} - T_{w3}}{R_{h2}} = \frac{T_{w3} - T_m}{R_{PCM2}} \quad (3)$$

$$R_{PCM2} = \frac{\delta - x(t)}{k_l} \quad (4)$$

Where q_{out} is the heat flux for energy release process (W/m²); T_{w3} is wall temperature, used as boundary temperature for energy release process (°C); T_{in} and T_m are the indoor temperature and melting temperature of PCM (°C); R_{h2} and R_{PCM2} are indoor heat convective resistance and PCM conduction resistance (m²°C/W); δ is the PCM board thickness (m); k_l is the heat conduction coefficient of liquid PCM (W/m/°C).

2.2 EXPERIMENTAL VERIFICATION

In order to verify the model in Fig.1, an experiment was designed. The heat transfer through walls is well-developed; so this experiment aims at verifying the heat transfer process of PCM board. The phase change material used in the test is octadecane paraffin. The details of phase change material can be found in Table 1. An environmental simulator (46in × 46in × 46in) was used in this research, shown in Fig. 2. Inside the simulator, eight 200W light bulbs were used as heat source to heat up temperature. And a portable air conditioner (2500 W) was connected with the simulator through a hole cut at the top of simulator to cool down temperature. The bulbs were all connected to dimmers to create different boundary conditions for phase change process. Two fans were installed inside the simulator to stir the air and reduce stratification effects.

A cubic box (8in×8in×8in) with a frame inside was made to store the phase change material. The frame was used to fix thermal couples. Fifty-nine type T thermal couples were installed inside and outside the box to measure PCM temperatures, box surface temperatures, insulation surface temperature and air temperatures in simulator. Thermocouples were shielded with aluminum tape to eliminate the effects of radiation. The box was insulated with R30 and R19 insulations to prevent heat transfer with environment, leaving one surface exposed. One heat flux meter was attached to each face to monitor heat fluxes. A data logger and a computer collected data every 10 seconds

Table 1 Parameters of phase change material in calculation

Melting point (°C)	Heat of fusion (kJ/kg)	Thermal conductivity (W/m/k)		Specific heat (kJ/kg/k)			Density (kg/m3)		Kinematic viscosity (m2/s×106)	
		liqui	solid	liq	so		li	s	liqui	solid
		d		uid	lid		quid	olid	d	d
27.4	243.5	0.15	0.35	2.1	2.		7	7	5.03	5.7
		2	8	8	15		75	78		2



Figure.2 Environment simulator

According to the assumptions in 2.1 section, heat transfer through one direction was tested for current research. Bottom and one side surface were used as heat transfer interface respectively. Three sets of experiments both for freezing and melting were carried on. If the temperatures of PCM change dramatically, we assume all the PCM get melted or solidified.

2.3 RESULTS COMPARISON

Fig.3 shows the time for all PCM finishing melting or freezing process in calculation and experiment. Three values in experiment should be skipped as show in open dot because the difference between red dots and calculation was larger than 30%. While for the other experiment data, it agrees well with the calculation.

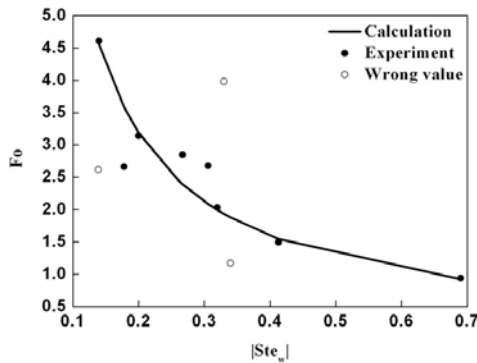


Figure. 3 Comparison on dimensionless time

Based on our calculation, the relationship between solid-liquid interface position and all related variables is shown in Fig. 4. A relationship was abstracted out through experimental data. As shown in the figure, the interface position was an exponent of boundary temperature, time and PCM parameters. This dimensionless relationship is significant for the design of energy exchanger with phase change material. The dimensionless expressions are shown as following:

$$Ste = \frac{c(T_i - T_m)}{L} \quad X = \frac{x}{x_{\max}} \quad Fo = \frac{at}{x_{\max}^2}$$

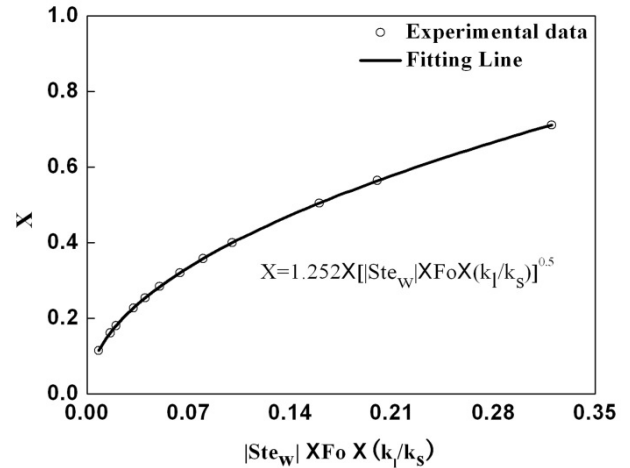


Figure.4 Solid-liquid interface vs. all variables

3. ANALYSIS OF ENERGY SAVINGS FOR BUILDING

Alfonso et al. [10] used life cycle assessment methodology to evaluate if energy savings by phase change material were large enough to balance the environmental impact caused by manufacture and installation of phase change material. Specific energy savings by PCM for building were calculated, divided into energy savings by natural cold energy and electricity savings by air conditioning system. The thickness of PCM board was designed to store daily maximum cold energy. The building information for this research is shown in Table 2.

Table 2 Details of building in calculation

Type	Rh1	Rw	Rh2
490-brick	0.04	0.801	0.11

3.1 ENERGY SAVINGS BY NATURAL COLD ENERGY

Outdoor air at night is cold that it can provide the cold energy that is demanded during day time; so PCM could be solidified by natural cold resource at night and release the cold energy in day. The energy savings, $Q_{s/n}$, can be obtained by Eq. (5).

$$Q_{s/n} = \sum_{i=1}^{91} \left(\sum_{j=20}^{24} q_{in}^{i,j} \times \tau_{ij} + \sum_{j=0}^7 q_{in}^{i+1,j} \times \tau_{i+1,j} \right) \times \eta_{i+1} \quad (5)$$

Where $q_{in}^{i,j}$ and $q_{in}^{i+1,j}$ are heat flux during energy storage process at time j in day i and time j in day i+1 (W), which can be calculated by Eq. (1); τ_{ij} and $\tau_{i+1,j}$ are time j in day i and in day i+1 (hr); $q_{out}^{i+1,j}$ is the heat flux during energy release process at time j in day i+1 (W), which can be calculated by Eq.

(3); η_{i+1} is the energy storage efficiency (%), as shown in Eq. (6).

$$\eta_{i+1} = \frac{\sum_{j=8}^{19} q_{out}^{i+1,j} \times \tau_{i+1,j}}{\sum_{j=20}^{24} q_{in}^{i,j} \times \tau_{i,j} + \sum_{j=1}^7 q_{in}^{i+1,j} \times \tau_{i+1,j}} \quad (6)$$

There is no energy cost for energy storage and release process; so the energy saving ratio is 100%.

3.2 ELECTRICITY SAVINGS BY AIR CONDITIONING SYSTEM

When outdoor environment is not cold enough, PCM should be solidified by cold energy generated by air conditioner. Part of cooling load in daytime could be shifted to nighttime. The decrease of condensing temperature will increase COP, which means that it will cost less electricity to generate same amount of cold energy. The electricity savings, $Q_{s/a}$, by air conditioner can be evaluated by Eq. (7).

$$Q_{s/a} = \rho_s \times \delta \times L \times \sum_{i=1}^{91} \left(\frac{1}{COP_i^d} - \frac{1}{COP_i^n} \right) \times \delta(T^d - T^n) \times \eta_{i+1} \quad (7)$$

$$\delta(T^d - T^n) = \begin{cases} 1 & T^d > T^n \\ 0 & T^d < T^n \end{cases} \quad (8)$$

Where L is the fusion of heat of PCM (kJ/kg); COP_i^d and COP_i^n are coefficient of performance in day i and night i , mean day and night temperature were used as condensing temperature; T^d and T^n are mean day and night temperature (K).

Electricity saving ratio (ESR) is defined as following:

$$ESR = \frac{\sum_{i=1}^{91} \left(\frac{1}{COP_i^d} - \frac{1}{COP_i^n} \right) \times \delta(T^d - T^n) \times \eta_{i+1}}{\sum_{i=1}^{91} \frac{1}{COP_{i+1}}} \quad (9)$$

Where COP_{i+1} is mean day coefficient of performance.

Electricity saving ratio is shown in Table 3. ESR has nothing to do with the amount of PCM, that is to say increasing the amount of PCM brings more electricity savings but same ESR. For the cities except for Hong Kong, all the energy saving ratio was bigger than 10%.

Table 3 Electricity saving ratio and energy storage efficiency for different locations in China (%)

Location	Energy saving ratio			$\bar{\eta}$		
	23°C	24°C	25°C	23°C	24°C	25°C
Shenyang	19.4	19.4	19.4	100	100	98.9
Zhengzhou	17.7	17.7	17.7	100	100	99.9
Changsha	11.0	11.0	11.0	100	100	100
Kunming	12.4	12.4	12.3	100	100	99.0
Hong Kong	5.0	5.0	5.0	100	100	100

4. ANALYSIS OF ECONOMIC SAVINGS

Different electricity tariffs during the night period and day period are applied in many cities in China. The electricity tariff in the nighttime is about 1/3 to 1/2 of that in the daytime. Similarly, economic savings are divided into two forms.

4.1 ECONOMIC SAVINGS BY NATURAL COLD ENERGY

Economic savings by natural energy is the economic savings that is cost by air conditioning system to generate the same amount of cold energy in Eq. (5).

$$C_{s/n} = \frac{Q_{s/n}}{COP} \times P \quad (10)$$

Where $C_{s/n}$ is economic savings by natural cold energy (USD); P is the electricity price, 16 Cents/kWh was used; 4.2 was used as COP in this paper.

4.2 ECONOMIC SAVINGS BY AIR CONDITIONING SYSTEM

Different electricity tariffs during daytime and nighttime were took into consideration in this calculation. The economic savings by air conditioning system is defined as Eq. (11).

$$C_{s/a} = \rho_s \times \delta \times L \times \sum_{i=1}^{91} \left(\frac{P^d}{COP_i^d} - \frac{P^n}{COP_i^n} \right) \times \delta(T^d - T^n) \times \eta_{i+1} \quad (11)$$

Where $C_{s/a}$ is economic savings by air conditioning (Cents), P^d and P^n are electricity price in daytime and nighttime.

4.3 ECONOMIC RECOVERY PERIOD

Economic recovery period (ERP) is an important index to measure if the investment is worth or not. The economic recovery period is defined as $ERP_{s/n}$ by natural cold energy and $ERP_{s/a}$ by air conditioning system.

$$ERP_{s/n} = \frac{C_{PCM}}{C_{s/n}} \quad (12)$$

$$ERP_{s/a} = \frac{C_{PCM}}{C_{s/a}} \quad (13)$$

Where C_{PCM} is economy cost by purchasing and installing phase change material, 1.5 dollars/kg was used in this paper.

We can see that in Table 4, the investment in Hong Kong, Changsha, Zhengzhou was not worth at all. The mean outdoor temperature among three months is 23.45, 26.2, 27.61, 19.33 and 28.41°C in Shenyang, Zhengzhou, Changsha, Kunming, and Hong Kong, respectively. Higher mean daily temperature means that there is less cold energy in air and less energy savings could be obtained; so ERP is longer. It is not suggested to use PCM with a phase transition temperature which is lower than mean outdoor temperature.

Table 4 ERP by natural cold resource for different locations by natural cold energy (Year)

Temperature	Shen Yang	Zheng Zhou	Chang Sha	Kun Ming	Hong Kong
23	23.1	59.0	174.7	12.5	582.1
24	17.7	39.7	103.6	10.2	541.1
25	14.0	28.3	63.1	8.5	447.6

ERP by air conditioner was too long, longer than 100 years. So the investment for PCM only cooled by air conditioning system is not worth for current PCM price.

5. THE SELECTION OF PHASE CHANGE MATERIAL FOR BUILDINGS

Based on the analysis above, the heat resistance of PCM is much smaller than the heat resistance of wall itself. So the heat flux for energy storage process can be simplified by the following equations. The error between two equations is less than 5%.

$$q_{in} = \frac{T_m - T_{out}}{R_{h1} + R_w + R_{PCM1}} \approx \frac{T_m - T_{out}}{R_{h1} + R_w} \quad \delta \leq 15mm \quad (14)$$

$$q_{in} = \frac{T_m - T_{out}}{R_{h1} + R_w + R_{PCM}} \quad \delta > 15mm \quad (15)$$

Where R_{PCM} is the total heat resistance of PCM ($m^2 \cdot ^\circ C / W$).

Fig.8 shows that the ERP by natural cold resource varies with different phase transition temperatures. It was obvious that the ERP decreased with the rise of phase transition temperature when it was higher than mean outdoor temperature. The shortest ERP is 7.87 year in Kunming for current PCM price and the longest ERP is 10.4 year in Zhengzhou.

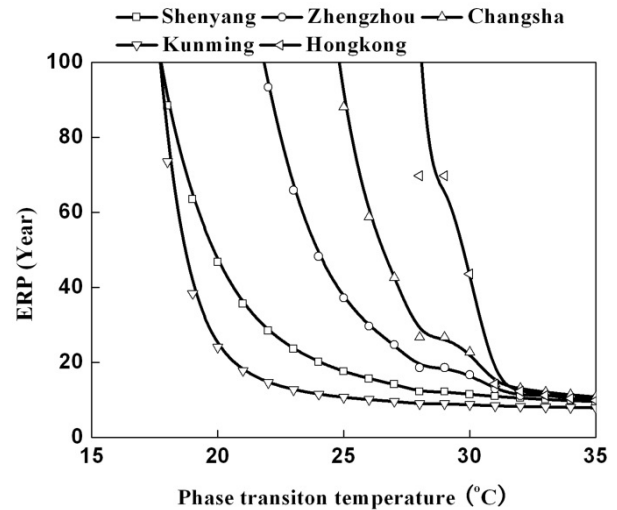


Figure.5 Economic recovery period with different phase transition temperatures in China

6. CONCLUSION

The application of phase change material with natural cold energy and air conditioning system for cooling in building was analyzed in this paper. Energy saving ratio and economic recovery period were developed to assess the application from aspect of energy and economy. The following conclusions could be drawn.

The energy saving ratio by natural cold resource is 100%. There is no energy savings but electricity savings by air conditioning system. The biggest electricity saving ratio happens in Shenyang, 19.4% and the smallest electricity saving ratio occurs in Hong Kong, 5.0%. The economic recovery period by natural cold resource decreases with the increase of phase transition temperature. And it is not recommended just using air conditioning system freeze PCM.

The optimal ERP can be obtained when phase transition temperature is 3 °C higher than mean outdoor air temperature.

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Experimental Research of Anhydrous $\text{Na}_2\text{SO}_4 + \text{H}_2\text{O}$ Phase Change Energy Storage system

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ABSTRACT

In this paper, a new kind of phase change energy storage system was prepared through a lot of experiments based on $\text{Na}_2\text{SO}_4 \cdot \text{H}_2\text{O}$ phase change energy storage system, which is used in air conditioning cooling water system as the cold source used in engineering. Ensure the quality ratio of Na_2SO_4 and H_2O is 1:1.28, adjust the ratio of different additives in $\text{Na}_2\text{SO}_4 \cdot \text{H}_2\text{O}$ phase change energy storage system, test optimized; test phase change materials super-cooling degree through cooling curve method; test phase change temperature and enthalpy value of phase change materials through DSC differential scanning calorimetry, and determine the optimal proportion of the additives at last. The new prepared phase change energy storage system, not only improve the sub-cooling and phase liquid separation problem of inorganic salt hydrate, but also reduce the phase change temperature, making the phase change temperature range appropriate. The phase change latent heat value is bigger, and the enthalpy attenuation rate is small through forty times' fusion-frozen cycle DSC test, with melting temperature of $26^\circ\text{C} \sim 28^\circ\text{C}$ (peak 32°C), phase change latent heat value of 156J/g . It can meet the requirement of air conditioning cooling water source. And the new phase change energy storage system original is more efficiency and maneuverability better than before.

Key words: $\text{Na}_2\text{SO}_4 + \text{H}_2\text{O}$; sub-cooling; phase liquid separation; phase change temperature

1. INTRODUCTION

Phase change energy storage technology is one of the attention energy storage technology in recent years, especially in building materials, storage air conditioning and materials science application[1-2]. It uses the principle of material absorbing or releasing a lot of energy in the phase change process, store up the unused energy temporarily, then release when needed [3-4]. This technology can effectively solve the mismatching contradiction of energy supply and demand in time and space, can promote the optimization of energy utilization, and improve the efficiency of energy utilization [5]. Scholars at home and abroad did a lot of research on inorganic salt hydrate phase change energy storage materials. For instance, American Dr. Maria Telkes built the world's first PCM passive solar house in Massachusetts State through long-term research on hydrated salt PCM especially $\text{Na}_2\text{SO}_4 \cdot \text{H}_2\text{O}$ [6], Japan Mitsubishi electronic company and Tokyo electric power company studied hydration nitrate, phosphate, fluoride and calcium chloride for heating and refrigeration system of phase change materials. At home, Yan Licheng, Xu Lingling, Sun Xinquan etc., studied the degree of sub-cooling, phase separation problems of $\text{Na}_2\text{SO}_4 \cdot \text{H}_2\text{O}$ through experiments, and achieved significant results [7-9].

This study is based on an air conditioning cooling pool of engineering, with the update of engineering internal equipment constantly, calorific value increased

greatly, so the heat storage capacity of the original air conditioning cooling pool can not meet the needs of air conditioning condensation heat storage. In accordance with the relevant requirements, bring the phase change material ten water sodium sulfate into air conditioning cooling pool. Make the use of melting endothermic phase change characteristics of ten water sodium sulfate to cool the air conditioning cooling pool, so as to improve the heat storage capacity of the cooling pool. It is equivalent to reduce the volume of cooling pool under the same heat storage capacity, which is of great significance for sustainable development.

2. $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ PHASE CHANGE ENERGY STORAGE SYSTEM

Through a lot of experiments, prepared phase change energy storage system of ten water sodium sulfate as the main body firstly. Adding borax, sodium carboxymethyl cellulose (CMC), gas phase silica, Sodium hexametaphosphate, sodium chloride in the pure ten water sodium sulfate reagent, through repeated test optimized the best proportion of various additives. Finally, using borax as nucleating agent to minimize the degree of sub-cooling of $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ phase change energy storage system; using sodium carboxymethyl cellulose and gas phase silica as thickener and suspending agent respectively to reduce the phase separation problem of $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ phase change energy storage system; using sodium hexametaphosphate as crystal habit modifier to inhibit

the generation of big crystal particles, in order to reduce the attenuation degree of phase transition enthalpy value after multiple melt-frozen cycle; at the same time, adding sodium chloride to reduce the melting point of

$\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ phase change energy storage system in order to meet the needs of the application. Table 2.1 lists the added reagent's name, ratio and the main function in the energy storage system.

Table.2.1 Reagent proportion and function of phase change energy storage system

Material name	Mass fraction(%)	Main function
$\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$	84	Phase change energy storage material
borax	3	Nucleating agent, decrease the degree of sub-cooling
Sodium carboxymethyl cellulose	2	Thickener, slow phase separation
The gas phase silica	6	Suspending agent, slow phase separation
Sodium hexametaphosphate	1	Crystal habit modifier, crystallization inhibition of larger particles
Sodium chloride	4	Lower melting point

The prepared $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ phase change energy storage system in accordance with the above method solves the problems such as sub-cooling and phase separation of $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$, adjusting the phase change temperature to suitable air conditioning cooling water system, and ensure the service life of the phase change energy storage system after repeated melt-frozen cycle experiments.

3. PHASE CHANGE ENERGY STORAGE SYSTEM OF ANHYDROUS Na_2SO_4 AND H_2O

3.1 Reason analysis of using anhydrous Na_2SO_4 and H_2O preparation method

Through research analysis and practical operation, comprehensively considering various factors, found that in practical applications, the use of $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ as main heat storage material has some problems:

(1) In terms of encapsulation, $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ is crystalline solid, large pieces of crystal is not easy to shatter, while the packet unit body volume is small, large pieces of crystal is not easy to load; If fill the phase change materials after melting, due to the $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ phase change system become sticky liquid after melting, with high viscosity and easy to stick wall, which is not easy to flow smoothly, and could produce bubbles in the process of flow, mixing in the phase change materials can reduce the coefficient of thermal conductivity, affect the heat transfer effect.

(2) In the process of filling, the environmental temperature influenced on the phase transformation system greatly. When the temperature is below 26°C , the phase change material begins to solidification. The solidification phase change material is not easy to load into the packet unit body, which could result in a certain amount of waste.

(3) The solubility of additive NaCl is not big, so it cannot disperse evenly in the mixed sticky liquor

even stirring, so that cannot play the role of regulating phase change temperature very well.

(4) On the economic front, the price of $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$ is 28 Yuan/kg, the project needs a lot of material, the cost is high, and with poor economy.

To solve these problems, use the method of anhydrous $\text{Na}_2\text{SO}_4 + \text{H}_2\text{O}$ with the ratio of 1:1.28 as main phase change heat storage material in experiments, the advantage of this method is:

(1) Anhydrous Na_2SO_4 is powder crystal, easy to load into packet unit body shell.

(2) The method of anhydrous $\text{Na}_2\text{SO}_4 + \text{H}_2\text{O}$ is to put various kinds of additives and anhydrous Na_2SO_4 mixture into the packet unit shell body firstly, stir well, then add the aqueous solution of NaCl, the phase change materials will not become solidification before loading into encapsulation shell and NaCl can dissolve more evenly.

(3) The price of analysis pure anhydrous Na_2SO_4 in market is 20 Yuan/kg, compared with $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$, choosing anhydrous Na_2SO_4 can greatly reduce the cost.

3.2 The preparation method of using anhydrous $\text{Na}_2\text{SO}_4 + \text{H}_2\text{O}$ as main phase change heat storage material

In $\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$, the molecular weight of Na_2SO_4 and 10 H_2O is 142:180 respectively, reduction of 1:1.26760. In order to prevent factors of water loss by evaporation and not fully combined with anhydrous sodium sulfate in reaction process and so on, the actual value of water should be slightly more than the calculated value, thus taking Na_2SO_4 and H_2O mass ratio of 1:1.28.

In addition, we adjusted the proportion of suspending agent silica and thickener CMC in the system. The silica particle size of the second purchase batch in the experiments is less than the first batch, the two suspension effect are different, so the particle size of silica has a great influence on the effect of the suspension. Through continuous debugging, the proportion of silica in the system changed from 6% to

1%, the proportion of CMC also corresponding reduced to 1.5%, and adjusted the viscosity slightly according to the actual needs.

The proportion of each material in the phase change thermal storage system after this adjustment is: anhydrous Na_2SO_4 +water 89.5% (with the ratio of 1:1.28), borax 3%, gas phase silica 1%, CMC 1.5 %,sodium hexametaphosphate 1%, sodium chloride 4%.

3.3 The performance test of using anhydrous Na_2SO_4 + H_2O as main phase change heat storage material

In experiment, test the anhydrous Na_2SO_4 + H_2O phase change energy storage system by the method of cooling curve firstly, the test equipment of the cooling curve method is shown in figure 3.1.

Make the temperature data into "temperature-time" curve, namely the cooling curve, which is shown in figure 3.2. After many tests measured, the sub-cooling degree is less than 1 °C, which can meet the using requirements of phase change materials.

To the preparation phase change heat storage system adopting the method of anhydrous Na_2SO_4 + H_2O , did melt-frozen cycling test 40 times repeated using constant temperature water bath, did a DSC test for every 5 times, then observed the enthalpy attenuation situation of the phase change energy storage system. Figure 3.3 (a) to figure 3.3 (i) show the testing DSC curve, table 3.1 shows the DSC test results and calculation attenuation rate in 40 times melt-frozen cycles.

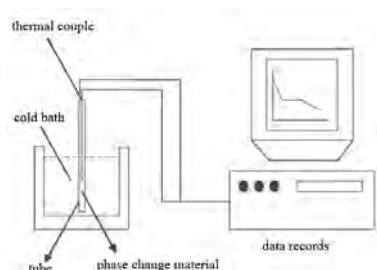


Figure 3.1 Cooling curve method test device

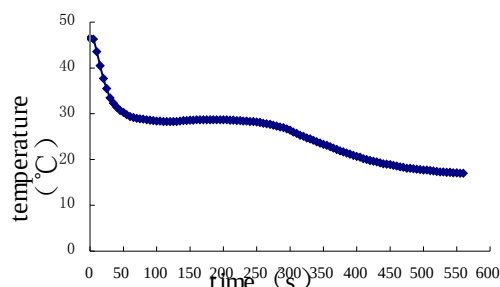


Figure 3.2 Na_2SO_4 + H_2O phase change system cooling curve

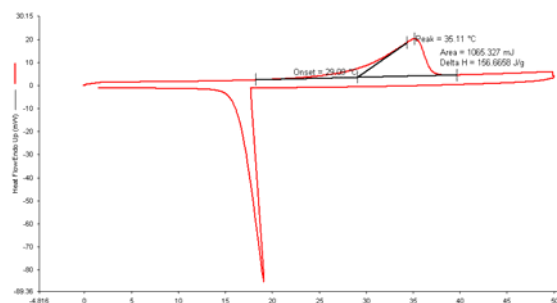


Figure 3.3 (a) 1 time

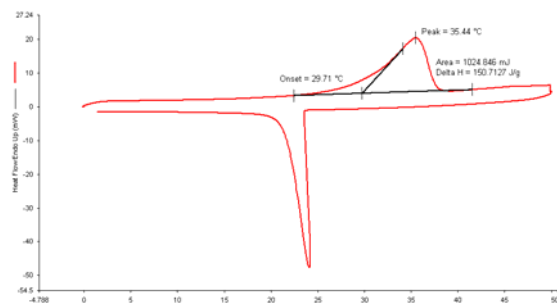


Figure 3.3 (b) 5 times

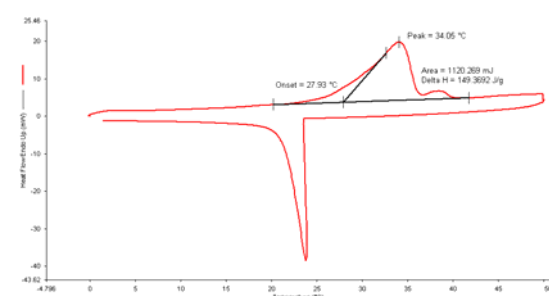


Figure 3.3 (c) 10 times

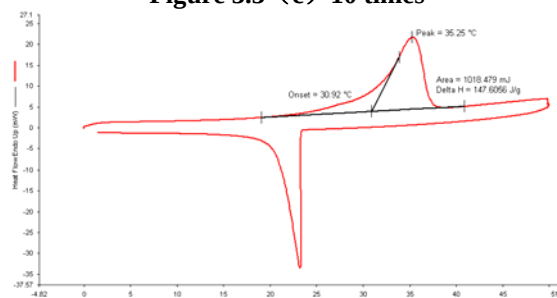


Figure 3.3 (d) 15 times

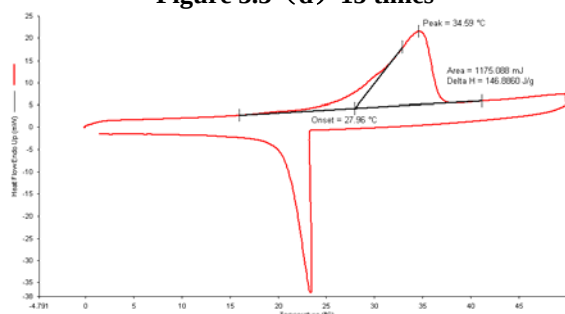


Figure 3.3 (e) 20 times

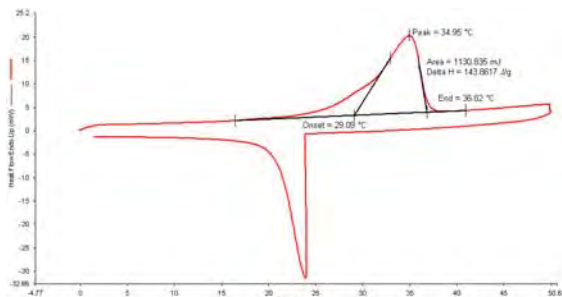


Figure 3.3 (f) 25 times

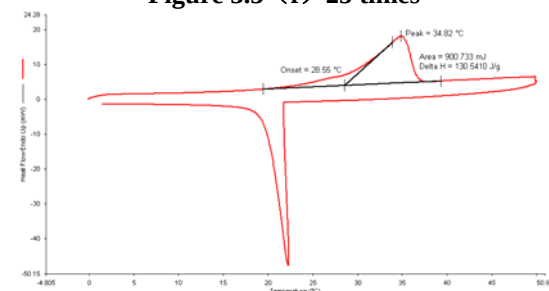


Figure 3.3 (g) 30 times

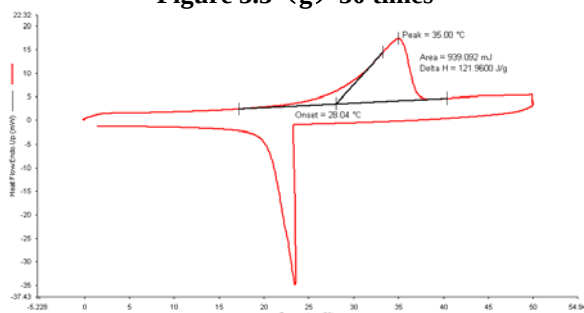


Figure 3.3 (h) 35 times

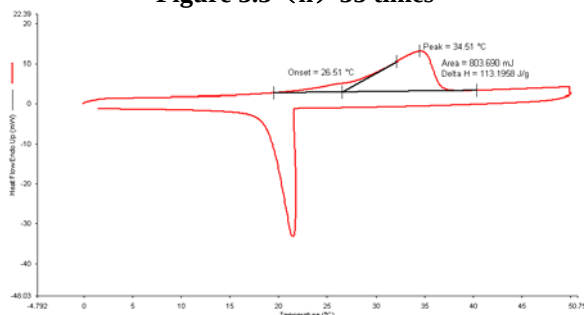


Figure 3.3 (i) 40 times

As can be seen from figure 3.1 (a), the phase change temperature of the phase change energy storage material is 27 °C, the initial phase change enthalpy value is 156 J/g.

Table.3.1 Result of DSC testing

Cycle index	Onset (°C)	Peak (°C)	H (J/g)	Attenuation rate
1	29.09	35.11	156.67	--
5	29.71	35.44	150.71	3.80%
10	27.93	34.05	149.37	4.66%
15	30.92	35.25	147.61	5.78%

20	27.96	34.59	146.89	6.24%
25	29.09	34.95	143.86	8.18%
30	28.55	34.82	130.54	16.68%
35	28.04	35.00	121.96	22.15%
40	26.51	34.51	113.20	27.75%

Analyze the DSC test results of the 40 times melt-frozen cycles, found that with the increase of cycles, the melting heat of phase change energy storage system reduced gradually, and the attenuation rate suddenly became bigger starting from the 30 times. During the experiment, also found that with the increase of cycling times, the system under solidified state appeared free water gradually, and at the same time, there were also some white hard particle, and the amount of free water and white hard particles increased with the increase of cycling times. Through analysis, the white hard particles were anhydrous sodium sulfate particles. After many times of melt-frozen cycles, formed a separation state of anhydrous sodium sulfate particles and water because of the anhydrous sodium sulfate can't all dissolved to the release crystallization water, and it is also the main cause of phase change enthalpy attenuation of the inorganic salt hydrate in the repeated use.

3. CONCLUSION

Adjusting on the basis of the sodium sulfate phase change energy storage system, we prepared phase change energy storage system for air conditioning cooling water system by the method of anhydrous sodium sulfate and water with the ratio of 1:1.28.

(1) The ratio of phase change energy storage system is anhydrous $\text{Na}_2\text{SO}_4 + \text{H}_2\text{O}$ 89.5% (with the ratio of 1:1.28), 3% borax, 1% gas phase silica, 1.5 % CMC, 1% sodium hexametaphosphate, and 4% sodium chloride.

(2) The sub-cooling degree of the phase change energy storage system is less than 1 °C; the phase change temperature is around 27 °C; and the initial phase change enthalpy value is 156 J/g, which can well meet the using requirements.

(3) The enthalpy damping rate of the phase change energy storage system in 40 times melt-frozen cycling test is 27%, which can meet the needs of using frequency.

(4) The preparation technology of the prepared phase change energy storage system adopted by the method of anhydrous sodium sulfate + water is simple and convenient for engineering application, with low cost, good efficiency, huge energy saving potential, which is a kind of new phase change energy storage materials of promising prospects.

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CONTRIBUTION TO PEAK POWER REDUCTION: NUMERICAL AND EXPERIMENTAL APPROACH FOR THE DESIGN AND CHARACTERIZATION OF AN AIR-PCM HEAT EXCHANGER COUPLED TO A BUILDING

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ABSTRACT

The need of exceptional amount of electricity during the peak power periods is becoming an important issue for energy companies nowadays. Regarding France, the peak power demand occurs in the late afternoon of winter periods (18:00-20:00), since a big part of heating systems is based on electricity.

In this context, our study investigates the potential of the use of an air-PCM heat exchanger, integrated in the ventilation system. The heat storage system is composed of a set of PCM (paraffin) plates, embedded in the exchanger and placed downstream in the ventilation duct.

The objective is to create a heat storage configuration that can stock a significant amount of heat, sufficient enough to turn off the electrical heating system during the peak power demand, without affecting the thermal comfort and the indoor air quality.

Toward this goal, two approaches were considered: experimental and numerical.

The experimental approach involved the conception and the construction of a heat exchanger coupled to an experimental cell in order to simulate real life indoor environment. In a second phase, preliminary tests were conducted in order to characterize the behavior of the exchanger.

Characterization results were used for the numerical approach in order to build a black box model that simulates the behavior of the system.

Keywords: Control strategies, Peak power reduction, PCM, thermal storage, Thermal Comfort, Indoor air quality.

1. INTRODUCTION

For ten years, France breaks records for peak electricity consumption. According to RTE (Electricity Transmission System), the electricity demand in France will experience an average annual growth between 0.8% and 0.9% in 2025. It should go from 485 TWh in 2008 to 515 TWh in 2015, 535 TWh in 2020 and 560 TWh in 2025, an increase of 15% (1).

This increased demand for electricity is due to several factors, but it is mainly driven by the residential sector. On one hand, French households are more and more equipped in electric devices. The high demand for electricity is also due to electric heating, which is also the focal point of this article. In 2010, 35% of the buildings, about 9.5 million (new and old), were heated with electricity. The French electricity consumption will therefore become increasingly sensitive to cold weather waves.

Peak demand is observed in winter period around 19:00 pm, it is due to the sum of the tertiary activities, transport activity and household activities.

The peak power will be stronger and longer, so it becomes necessary to put in place load management strategies. One of these strategies introduces special tariffs in order to influence consumer behavior to facilitate the balance between supply and demand on the network. It is not necessarily energy savings, because the client can consume the electricity at a different time, when the network is less used (2).

In practice, it is delaying some features in order to avoid global or local over-consumption.

The interest of peak power reduction is to avoid critical operation. In fact, nuclear power ($\approx 77\%$ of French electricity) covers only a part of the peak demand. This leads to an energy production that relies on oil or gas which is costly in terms of money and greenhouse gases emissions.

Another interest is to bring greater stability to the network. Widespread instability in the network can lead to damages to the generation system, the distribution system, customers or in extreme cases turn off the entire network (blackout).

The scope of peak power reduction strategies concerns industry, services and individual customers (heating/cooling, electrical devices).

1.1 OBJECTIVE

The objective of this work is to develop peak power reduction strategies that can be applied in the construction sector and more specifically in the heating systems. Thus, the potential use of an air-PCM (Phase Change Material) heat exchanger was investigated. The aim is to design and experimentally characterize a heat exchanger which allows storing heat using paraffin material and discharging it during the peak power period in order to avoid the usage of electrical heat. Two challenges have to be taken up:

1. Store enough heat in a reduced volume in order to integrate it in a HVAC system.
2. Manage storage phase and discharge phase while providing comfortable indoor condition inside the building.

1.2 APPROACH

To achieve these objectives, two approaches are considered: experimental and numerical.

The experimental approach (subject of this paper) consisted in designing and building an Air-PCM heat exchanger, taking into accounts the needs and constraints intrinsic to this case study, including the geometry of the system.

The numerical approach consists to model the system in order to solve the problems posed by the temporal and financial limitations of the experiment. Two levels of complexity are proposed for the model of the heat exchanger:

- A first level based on the experimental characterization, in order to build a simple model
- A second level, based on CFD tools, which use an approach to variable capacity (modeling the latent heat absorbed or released by the phase change by an apparent increase in the heat capacity of the PCM);

Those models will be used to develop and evaluate system control strategies. Evaluation criteria are proposed, taking into account the impact of the system on indoor environment and the robustness of the developed control strategies in terms of energy savings, thermal comfort and indoor air quality.

This study is part of a French project called StockAir 2. This project aims to achieve a configuration that will provide 4kWh of energy between 18h and 20h, during the peak power.

2. EXPERIMENTATION AND CALIBRATIONS

2.1 Experimental set up

The heat exchanger is composed of a set of boxes incorporating PCM, which are integrated into the ventilation system. The energy is supplied by means of two electrical resistances.

The objective is to switch off the electric heating system during two hours without affecting the thermal comfort and indoor air quality.

PCMs are materials that can store a large amount of energy as latent heat in a narrow temperature range and in small volumes, in comparison to other materials.

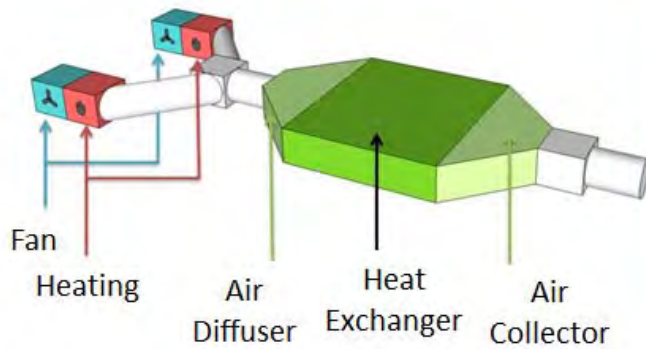
PCM are commonly used in many building applications (3) to (7), including heat exchangers (8) to (10).

In this study Mikrotek37 paraffin was selected due to its melting point (37 °C), a relatively small percentage of expansion between phases (12%) and its latent heat (227 KJ/kg).



Figure 1 Air-PCM exchanger

The heat exchanger (1m long, 0.67 m width and 0.2 m height) consists of 34 aluminum boxes filled with 32 kg of PCM and sealed with elastic glue. Fins are placed between the rules in order to augment the convective heat exchanges. Once designed and manufactured, this exchanger has been instrumented and coupled with a full scale experimental test cell called Hybcell (12). The HVAC system coupled to the Hybcell test consists of two fans, two electrical resistances and the heat exchanger (Figure 2).



HYBCELL Test cell

Figure 2 Tested HVAC system coupled to a full scale experimental building

Measurements are taken at the following locations:

- 1- Temperature, relative humidity and air flow rate before and after the heat exchanger
- 2- PCM temperatures and surface temperatures of three boxes in the beginning part, middle part and final part
- 3- Surface temperatures of the boxes in the outlet part

The acquisition system was designed in order to study the fusion and the solidification of the PCM, the heat losses in the heat exchanger and the charge / discharge phases of the heat exchanger.

A control system is integrated to the experimental setup in order to automatize tests at different air flow rate and temperatures.

2.2 Exchanger behavior characterization

Several calibration and characterization tests have been made, varying the airflow rate from $100 \text{ m}^3 \cdot \text{h}^{-1}$ to $500 \text{ m}^3 \cdot \text{h}^{-1}$.

Inlet temperature of the exchanger has been fixed to 44°C during the storage phase and 24°C during the discharge phase.

The Figure 3 shows the evolution of the temperature of the PCM inside the heat exchanger at the beginning, middle and end part of the three boxes; these results refers to an air flow rate of $200 \text{ m}^3 \cdot \text{h}^{-1}$; the inlet air temperature was controlled to 44°C in the charge phase and 24°C in the discharge phase.

The overall melting of the PCM took 6 hours and a delay of 4 hours between the beginning part and the final part of the exchanger has been noticed. Even if the desired amount of storage has been achieved, the time needed to complete the storage could lead to overheating or thermal discomfort in terms of air velocity.

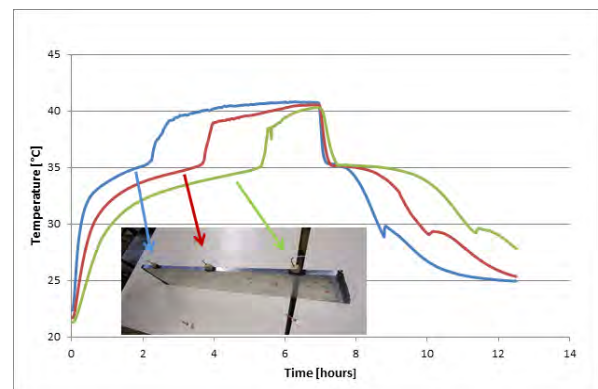


Figure 3 Downstream PCM temperatures

Characterization tests have also put in evidence that obviously the discharge phase depends on the airflow rate. In the configuration case of figure 3, the duration of the discharge phase is greater than 2.5 hours which fit to our initial goal (2 hours). The discharge phase shows also two solidification steps due to the intrinsic properties of the paraffin.

Figures 4 and 5 show the characterization results (charge and discharge).

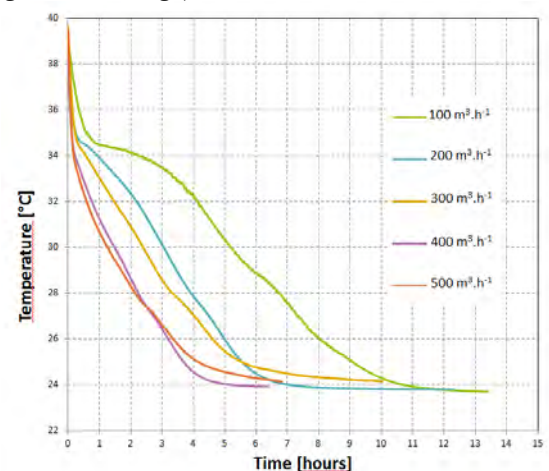


Figure 4 Variation of charge time function of airflow rate

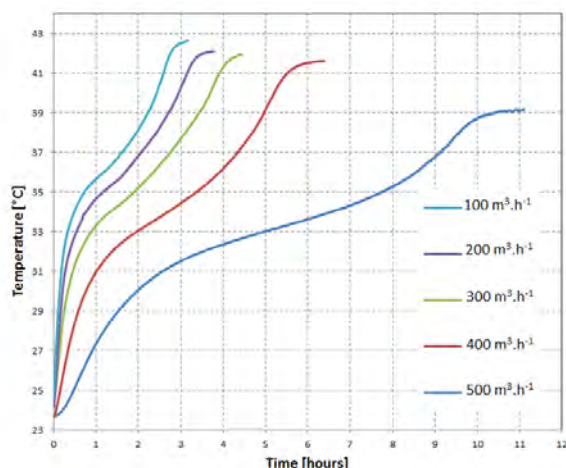


Figure 5 Variation of discharge time function of airflow rate

The characterization of charge and discharge time according to airflow rate revealed two aspects:

1-The storage time depends more on time than discharge time. However, even for high air flow rates the exchanger can provide enough heat for two hours.

2-Low air flow rate, desirable for indoor air quality, are not high enough to store latent heat in a reasonable time. Thus, in case of high insolated building, the integration of the tested heat exchanger in HVAC system has to include advanced control strategies.

3. COUPLING WITH A REAL BUILDING AND DISCUSSION

Once characterized, the air-PCM exchanger has been coupled with the test cell. As illustrated in figure 2, Hybcell test cell represents an office or meeting room (50 m³) equipped with a hybrid ventilation system, heating devices and simulated occupancy. Outdoor and indoor conditions in terms of temperature, relative humidity, CO₂ as well as energy consumptions are monitored. National Instruments based technology allows the overall control and management of Hybcell data acquisition system.

Figure 6 shows an example of developed and experimentally tested control strategies in order to manage the integration of the Air-PCM exchanger into Hybcell HVAC system. Figure 6 shows the evolution of the inner air temperature of Hybcell and the electric heating power (100% refers to 2000W);

As seen in the figure 6, a part of the heat is absorbed by the heat exchanger and the rest directly goes into the test cell. The green areas correspond to the 18h-20h period (peak power period) where the HVAC heating system was switched off. During this period the indoor air temperature is still in thermal comfort range, with a very low decrease. This means that the discharge of the AIR-PCM exchanger is enough to bring a good thermal comfort while contributing to energy peak reduction.

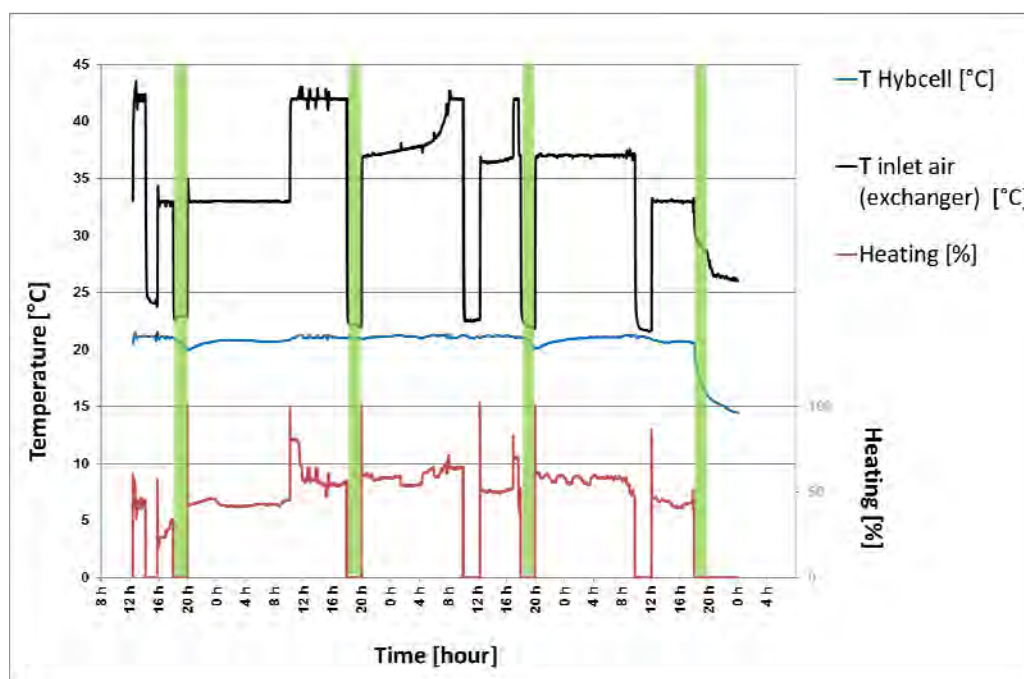


Figure 6 Integration of AIR-PCM exchanger and HVAC system of Hybcell

During the fourth day, starting from 18h, the heating system and the ventilation system were switched off, that's why we can observe that the inner temperature dropped to 14°C. This day will be taken as

reference day in order to assess the potential of the heat exchanger.

4. CONCLUSION AND OUTLOOKS

In this work an experimental Air-PCM heat exchanger coupled with a test cell has been studied in order to reduce the energy consumption peak due to electric heating.

The characterization of the exchanger showed that the complexity of its behavior needs control strategies in order to avoid overheating and thermal discomfort.

During the period of the peak power (18h-20h) the electrical heating system was turned off and the latent heat stored into the heat exchanger was enough to maintain a comfortable indoor air temperature. The PCM heat exchanger represents an efficient way to contribute to the reduction of the energy demand peak.

Further work concerns the development of a numerical model which simulates the behavior of the whole system, including building model. This approach will be useful to develop and evaluate advanced control strategies.

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THE APPLICATION OF ICE SLURRY FOR COOLING MILK

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ABSTRACT

The technique of ice slurry is firstly proposed for milk cooling process in this paper. Warm water is used to prepare 0°C ice slurry which provides 1°C cold water for milk cooling processing. A 170RT ice making equipment with 98 m³ ice storage tank is developed coupled with a 170RT main unit. The total cool storage of the ice making equipment is 1020RTH. This equipment can remove peak electricity by 222,900 kWh, saving 380,400RMB on air conditioning system annually.

Keywords: Ice slurry; Cool storage; Milk cooling process

1. Dynamic ice slurry preparation with supercooling method

The principal of this method is to produce cool water below 0°C, called supercooling water, through refrigerator. Then, the supercooling water is changed to ice slurry using supercooling lifting device. The supercooling water with temperature of 2°C can produce 2.5% concentration of ice slurry. By using this method, ordinary water can be made into ice slurry.

Characteristics of ice slurry: 1) Compared to the coil and ice hockey, refrigerant evaporation temperature is 3 °C higher, COP of refrigerating unit is improved significantly, and energy consumption of ice making is reduced by 9.0% than hockey and coil technology; 2) The specific surface area of the ice slurry is 10 times more than the puck and coil, and ice-melting speed and load response can meet any architectural change in load demand; 3) There is no hockey and coil in Ice storage tank; so ice tank volume is reduced. Various existing waste water tank can be used, basement, parking and roof sink, which have no strict requirements on the shape of the storage tank; so it has strong adaptability; 4) Low concentration of ice slurry (below 20%) is available to common pipeline pump. Since dynamic ice slurry ice melting speed, it can provide 0-2°C low temperature water continuously, especially suitable for the milk industry of low temperature cold water demand.

Through system simulation and experiments [1-4], Japanese researcher has grasped the core technology of dynamic ice storage continuous preparation and successfully applied it to the process of the milk industry and large building ice storage. Jiang Yi et al. in Tsinghai University firstly systematically studied the dynamic ice slurry preparation techniques [5-9], supercooling release technical and ice anti-communication technologies, but failed to achieve

industrial applications. Supported by National 863 Projects and Natural Science Funds, Guangzhou Energy Research Institute of the Chinese Academy of Science established a set of dynamic ice storage of central air-conditioning demonstration system, and systematically studies supercooling method ice slurry preparation technology, and developed a series of large-capacity ice slurry preparation equipment and industrialization [10-14]. Figure 1.1 is dynamic ice slurry preparation system with supercooling method, and the largest single capacity is up to 500RT.



(a) Micro ice crystals filter



(b) Ice slurry preparation unit(500RT)
Figure 1.1 High-capacity dynamic ice slurry preparation systems

2. Overview of process with cold of a milk factory

The milk production process is shown in Figure 2.1. There are three cooling steps in the production process: (1) Raw milk cooling; (2) cooling after

pasteurization; (3) milk cooling. The cold demand of the factory in production process is shown as following:

Yogurt Production: a production of 70-80 tons/day, generally produced between 4:00 to 8:00. Yogurt temperature is from 40-43°C drop to 15-18°C, and circulation quantity 10 tons/hour. Cooling water temperature is from 10-12°C to 12-17°C, and circulation quantity 30 tons/hour.

Milk beverage production: a production of 40-50 tons/day, generally produced between 13:00 to 14:00. Milk beverage temperature is from 40°C to 17-18°C, and circulation quantity 10 tons/hour. Cooling water temperature is from 10-12°C to 12-17°C, and circulation quantity 30 tons/hour.

Milk production: a production of 8-12 tons/day, generally produced at 8:00 to 13:00. Milk temperature is from 20°C drop to 2-6°C, and circulation quantity 3 tons/hour. Cooling water temperature is from 1°C to 5°C, and circulation quantity 10 tons/hour.

Milk tank insulation: back to factory milk, about 25 tons, is 6-10°C, cooling down to 4°C and generally produced between 8:00 to 12:00.

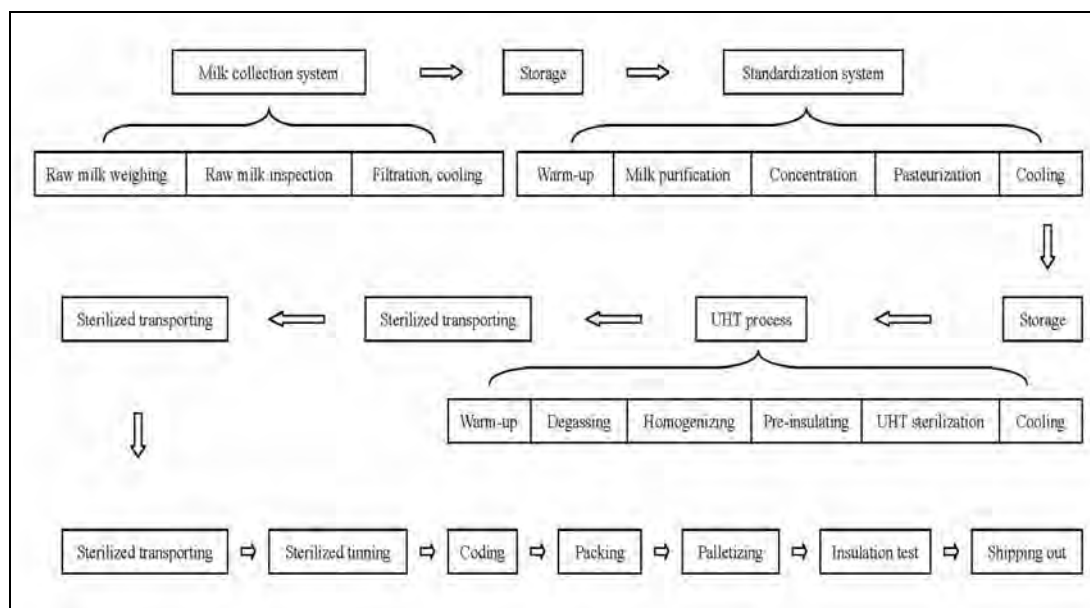


Figure 2.1 Milk production process diagram

3. SYSTEM REFORM PROJECT

Figure 3.1 is milk factory of cold process transformation program graph; original unit is 2 sets of ammonia machine with cooling capacity of 156RT and 2 sets of piston machine with cooling capacity of 50RT. The original system generated 1°C low-temperature water uniformly for cooling yogurt (10°C cold water is needed actually) and milk (1°C cold water), which

leads to low energy efficiency and big energy consumption. After transformation, low-temperature cold source and high-temperature cold source will cool yogurt and milk separately. This method significantly reduces the power consumption of the refrigeration. A set of dynamic ice storage system was used with original unit. It achieved peak load shifting and reduced product cost through ice thermal storage with dynamic ice storage system at night.

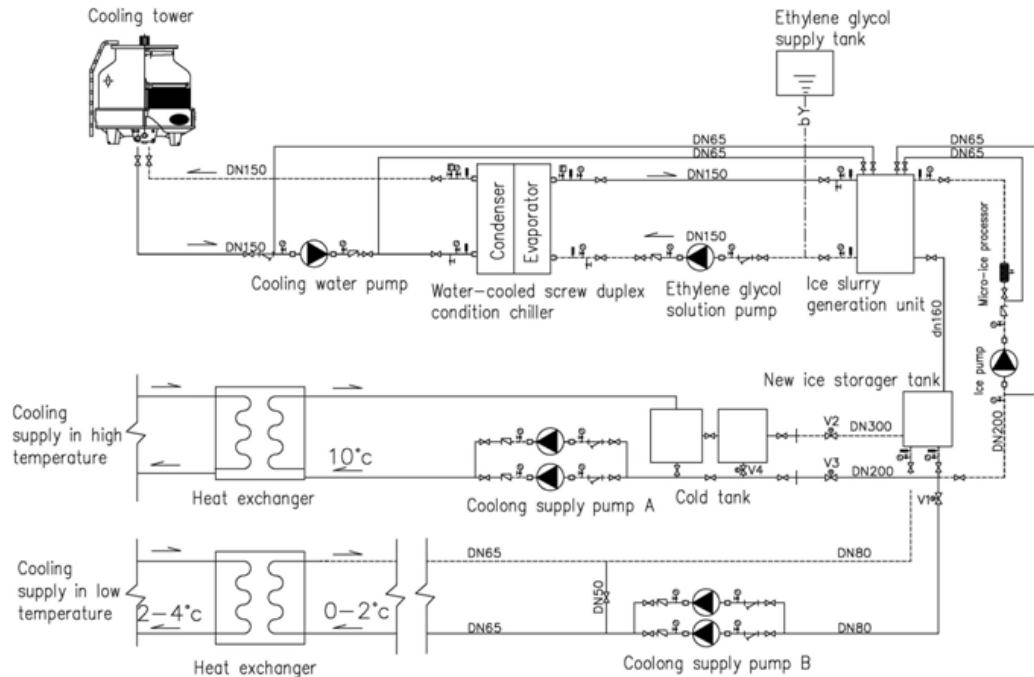


Figure 3.1 Milk factory of cold process transformation program graph

According to the current situation of dairy production conditions and existing space, retrofit scheme is as follows:

1) A 170RT duplex conditions refrigerator is added for ice storage. Ice storage system and the original system are independent of each other, mutual backup, thus improving the reliability of the production system; increasing the total cooling capacity, convenient for expansion of output. New refrigerator, ice-making equipment, pumps and control systems are installed in the chiller room. Original chilled water pump, new release cold pump and Ice pump are variable frequent. From 0:00 to 8:00, new ice maker runs 8 hours for cold storage, the total storage amount for 1020 RTH, the rest of the period the main unit also can supply cold water.

2) Ice storage tank with total volume of 98 m³ was built next to the original cold sink. The dimension (L × W × H) is 6.5 × 3.6 × 4.2m, and underground depth is 2.1m. Ice storage tank and original cold sink disconnected with each other for isolation. And a set of water treatment equipment is added.

3) The main equipment of the storage system including refrigerator, ice-making equipment, micro ice crystals processor, storage pumps, glycol pumps, cold relief pumps, water treatment equipment, inverter, storage tank and control system. Figure 3.2 is figure of modified equipment and ice slurry.



(a)170RT duplex conditions refrigerator



(c) Low temperature water pump



(c) Ice making system control interface



(d) Façade of flow of ice slurry preparation

Figure 3.2 Modified ice slurry preparation system

4. Reformed operation mode

Because the refrigeration system belongs to cooling process, the load has little to do with the environment temperature. But it has a direct relationship with the production yields. According to actual situation of the cooling load, the annual load can be classified into 3 levels: 100%, 75% and 50%. The

annual distribution of each level can calculate by the days:

- 100%load operation state 120 day/year
- 75%load operation state 120 day/year
- 50%load operation state 120 day/year

We can use 4 electrically operated valves to achieve different operational patterns after reformation. As shown in table 4.1:

Table 4.1 Operation Pattern

No.	Operation Pattern	Equipment, valves operation state								
		Main engine	Ethylene glycol pump	Ice pump	Release Ice Pump A	Release Ice Pump B	V1	V2	V3	V4
1	Cold storage pattern at night	On	On	On	Off	Off	Off	Off	Off	Off
2	Special ice storage tank designed in low temperature pattern(for the production of fresh milk and etc)	Off	Off	Off	Off	On	On	Off	Off	Off
3	Special ice storage tank designed in high temperature pattern(for the production of yoghourt ,milk beverage and etc)	Off	Off	Off	On	Off	Off	On	On	Adjustment
4	Old water tank for special pattern(for the production of yoghourt ,milk beverage and etc)	Off	Off	Off	On	Off	Off	Off	Off	Adjustment

1) 100% load

At the low period of electricity, a new 170RT refrigerator is responsible for cool storage and the previous refrigerator is used for cooling; At the peak period of electricity, the ice storage tank is responsible for cooling, and main unit, cooling pump and cooling tower stops running; at flat period of electricity, the previous refrigerator is responsible for cooling. As shown in figure 4.1.

2) 75% load

At the low period of electricity, a new 170RT refrigerator is responsible for cooling storage and the previous refrigerator is used for cooling; At the peak period of electricity and some flat period of electricity,

the ice storage tank is responsible for cooling. The main engine, cooling pump and cooling tower stop running; at other electricity periods, the previous refrigerator is responsible for cooling, as shown in figure 4.2.

3) 50% load

At the low period of electricity, a new 170RT refrigerator is responsible for cooling storage and the previous refrigerator is used for cooling; at the peak period of electricity and some flat period of the electricity, the ice storage tank is responsible for cooling. The main engine, cooling pump and cooling tower stop running; at other electricity periods, the main engine is responsible for cooling, as shown in figure 4.3.

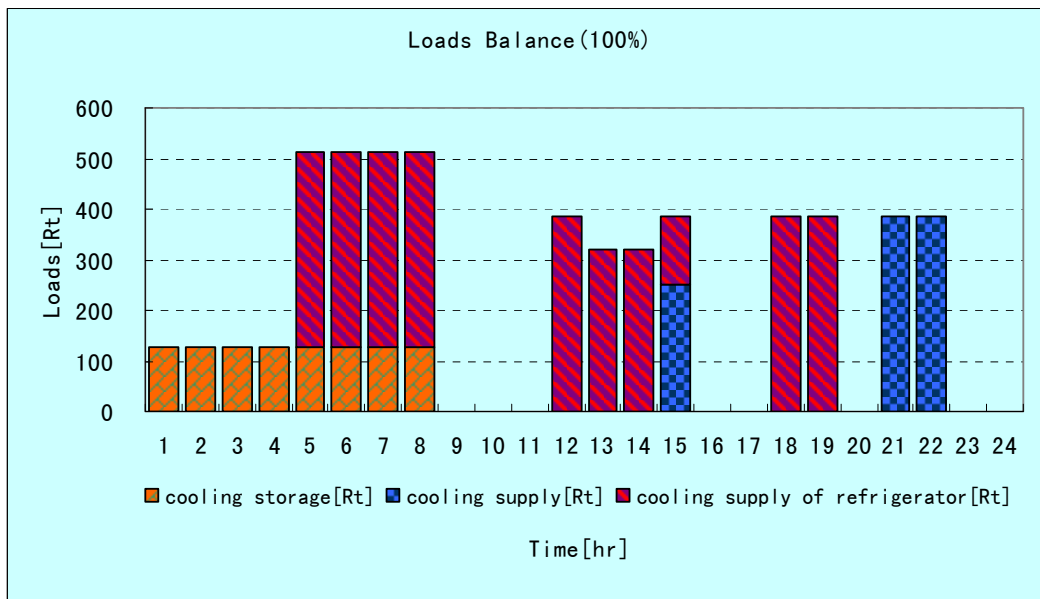


Figure 4.1 Loads Balance of 100% loads

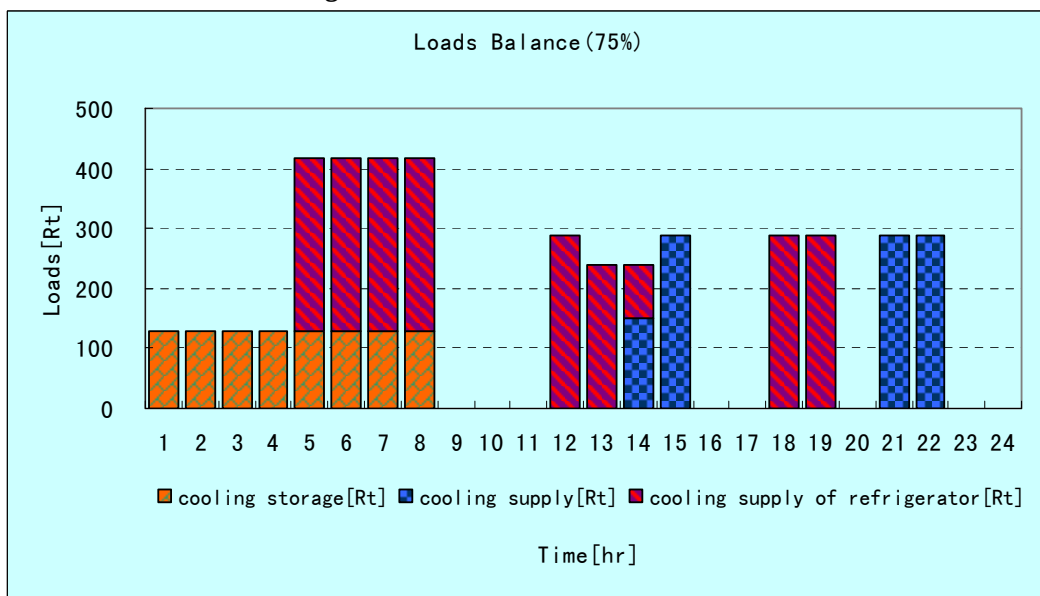


Figure 4.2 Loads Balance of 75% loads

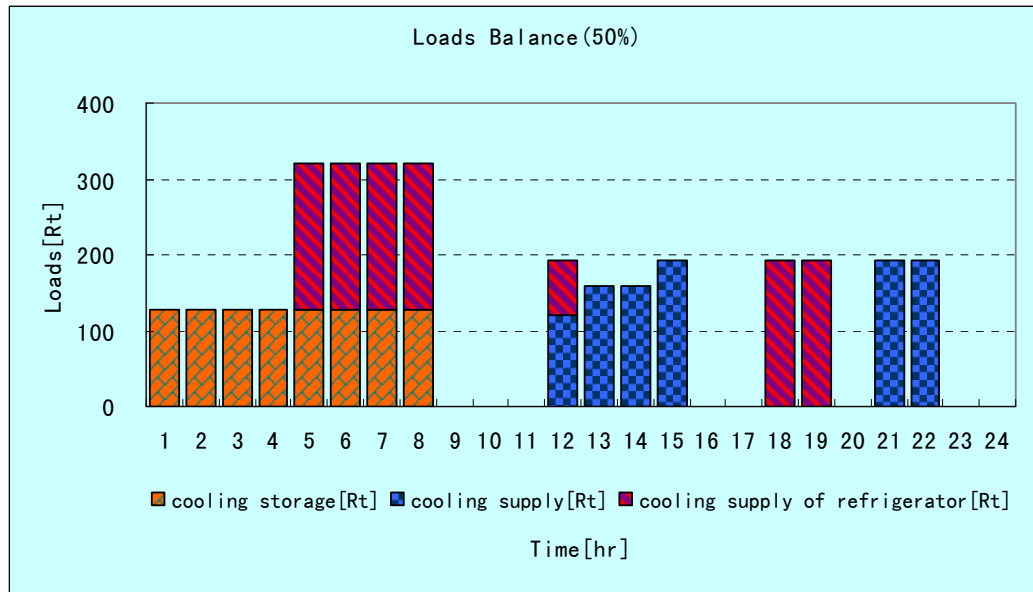


Figure 4.3 Loads Balance of 50% loads

5. ECONOMIC ANALYSIS OF THE PROJECT

According to the Off-peak tariff of electric power in Guangzhou, the large—scale industry of off-peak tariff of electric power as shown in Table5.1.

In order to study the economic operation of the ice storage system compared with conventional system, we simulated the running condition of the ice storage system. The annual power consumptions of ice storage

system and the conventional system are shown in Table 5.2.

According to the calculation and analysis, the annual power consumption of previous cooling system is 11,116, 000 kWh. It can be transferred by 23,050,000 kWh of the power consumption in peak hours with ice storage cooling technology. The annual operating electricity cost is about 794,300 yuan, saving 321,800 yuan. The annual saving rate is about 28.8%. This technology's initial investment is 1,150,000 yuan with economic payback period of 3.57year.

Table 5.1 Off-peak tariff of electric power

		Unit: (Yuan/ kW·h)
Classification of the electric power	Implementation period Time	Electricity price Yuan
Peak period	14:00-17:00	1.1811
	19:00-22:00	
Flat period	08:00-14:00	0.7362
	17:00-19:00	
	22:00-24:00	
Off period	00:00-08:00	0.3940

Table5.2 Annual power consumption in cooling period of electric power supply

		Dynamic ice storage system			Conventional refrigeration system		
Days	Load	Peak period	Flat Period	Off period	Peak period	Flat Period	Off period
Daily power consumption of different load modes (kWh)	100%	273.60	2188.00	2472.80	1087.02	2491.20	1203.40
	75%	273.60	1428.00	2261.60	945.50	1998.00	1694.00
	50%	273.60	436.00	2032.80	708.82	1504.80	1289.20
	25%	0.00	0.00	0.00	0.00	0.00	0.00

	120	100%	3.28	26.26	29.67	13.04	29.89	14.44
Annual power consumption of different load modes (10 ⁴ kWh)	120	75%	3.28	17.14	27.14	11.35	23.98	20.33
	120	50%	3.28	5.23	24.39	8.51	18.06	15.47
	0	25%	0.00	0.00	0.00	0.00	0.00	0.00
Total (10 ⁴ kWh)	360	power consumption	9.85	48.62	81.21	32.90	71.93	50.24
Annual power consumption (10 ⁴ kWh)				139.68			155.06	
electrovalence (yuan/kWh)			1.1811	0.7362	0.3940	1.1811	0.7362	0.3940
Operating electricity cost (banyan)			11.63	35.80	32.00	38.85	52.95	19.79
Total Operating electricity cost (banyan)				79.43			111.60	
Annual saved operating electricity cost						32.18		
Annual saved rat						28.8%		
20years energy saving benefit (banyan)						643.51		
Total additional initial cost (10 ⁴ yuan)						115.00		
Investment recovery period (year)						3.57		
Annual charge transfer in peak period (10 ⁴ kWh)						23.05		
Annual co2 emission reduction (t)						200.97		
Annual so2 emission reduction (t)						15.63		
Annual oxynitride emission reduction (t)						3.02		

2. THE OPERATION RESULT

Through more than one year's operation, the system works steadily and received a very good economic benefit. The following table 6.1 and 6.2 make a comparison of power consumption before and after the system's reformation.

Save a lot of electrical charge for the enterprise. After taking in the ice storage technology, it can transfer 222,900kWh electricity from peak period of electricity, saving 380,400yuan per year and the cost of

product is cut down to 9.9yuan per ton.

Increase the cooling capacity, provide guarantee for expanding production. The newly add main unit can provide 200RT cooling capacity in day time except for cooling storage.

Greatly improve the system reliability and stability. The newly added ice storage system and the previous system are independent with each other and back up for each other, so it can greatly improve the system reliability and stability. And the operation cost and maintenance cost is reduced.

Table 6.1 Power consumption, power price and Unit consumption of product before and after the system's reformation

	Time	Power Consumption (kWh)	Electric Charge (Yuan)	Output (t)	Power Consumption per ton (Yuan/t)
Before	June 2010	170628.00	117488.22	3126.00	37.58
	August 2010	152240.00	101265.97	3105.00	32.61
	April 2010	146704.00	97583.58	3099.00	31.49
	October 2010	109732.00	69509.53	2762.00	25.17
	Average	144826.00	96461.83	3023.00	31.91
	December 2010	62588.00	35237.40	1733.00	20.33
After	December 2010	78345.00	40972.93	2244.00	18.26
	January 2011	82468.00	44535.79	2313.00	19.25
	February 2011	91631.00	50040.21	2542.00	19.69
	March 2011	97480.00	53806.68	2676.00	20.11

April 2011	124356.00	68895.32	3071.30	22.43
May 2011	128048.00	69741.85	2993.70	23.30
June 2011	142256.00	77143.81	2919.00	26.43
June 2010	142928.00	74400.71	3148.00	23.63
August 2011	128540.00	71010.04	3148.00	22.56
April 2011	113004.00	63040.29	2780.00	22.68
October 2011	108326.00	59258.00	2599.00	22.80
Average	108330.83	59006.92	2680.58	22.01

Table6.2 The power consumption and electricity cost summary table before and after the System transformation

Items	Power consumption before transformation	Power consumption after transformation	Increase or decrease power consumption	Electrovalence	Increase or decrease Electric Charge
	kWh/a	kWh/a	kWh/a	Yuan/KWh	Yuan
Peak period of electricity	326993	103998	-222995	1.1811	-263379
Flat period of electricity	731842	428992	-302850	0.7362	-222958
Off period of electricity	498275	766985	268710	0.3940	105872
Total	1557110	1299975	-257135	--	-380466

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THERMAL ANALYSIS OF IN-SERIES VERTICAL GROUND HEAT EXCHANGERS FOR SEASONAL HEAT STORAGE

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ABSTRACT

The seasonal thermal energy storage offers an efficient and favorable way to integrate the industrial waste heat with ground heat exchangers which have a serious cold accumulation during a year. A novel borehole configuration of multistage-series circuits is proposed to not only satisfy the requirement of the large temperature drop by the industrial process but also make full use of industrial waste heat. Based on the heat transfer model of conventional GHE, a mathematical simulation model of multistage series circuits is developed to illustrate the heat transfer process of the underground thermal storage of industrial waste heat. The simplex method is employed to solve the outlet/inlet temperatures and the optimal fraction of the recharging heat rate of each circuit. An example system with three-level series circuits is simulated by means of the simulation model. A set of parameters that strongly affect the thermal performance of the GHE with multistage series circuits is discussed in detail. According to the trends of the recharging heat rate fractions of the circuits in series, regularly reversing the flow direction of the circulating fluid through the circuits are suggested to achieve a balanced ground temperature and a higher thermal efficiency as well. Both a larger borehole spacing and the soil with higher thermal properties can improve the thermal efficiency of the storage system. The simulation results indicate that the developed multistage series model can better describe the heat transfer characteristics of the thermal storage.

Keywords: multistage-series circuits, seasonal thermal energy storage, utilization of industrial waste heat, ground-coupled heat pump.

1. INTRODUCTION

The wide application of the ground-coupled heat pump (GCHP) technology is hindered by the imbalance of full-year cumulative heating and cooling loads in severe cold areas in north China. On an annual basis in heating-dominated buildings, the amounts of heat rejected to the ground in summer cannot balance the heat extracted from the ground in winter. Especially for the large-scale system with a long term operation, the annual imbalance in ground load will cause an irreversible temperature drop of underground soil and correspondingly a lower temperature of circulating fluid entering the heat pump. As a result, the unbalanced ground loads will significantly degrade the system performance, even causing the system collapsed during the coldest time of winter. The cooling load accumulation in the ground can be eliminated by recharging additional heat into the ground through some

mechanisms, such as the solar energy and the industrial waste heat^[1]. It is well known that a large amount of industrial waste heat, produced in the industrial process, is usually wasted by way of cooling tower emissions into the atmosphere in the non-heating season. The seasonal thermal energy storage offers an efficient and favorable way to integrate the industrial waste heat with GHEs which have a serious cold accumulation. Using the ground heat exchanger (GHE) with vertical boreholes as a thermal store region can not only solve the storage problem of industrial waste heat in the non-heating season, but also improve the operating performance of GCHPs.

The vertical GHE used for the seasonal thermal storage has the similar heat transfer mechanisms as conventional GHEs used for space heating and cooling. In the two cases, heat rejection/extraction into/from the ground is accomplished by circulating working fluid

NOMENCLATURE

Symbols

C	Heat capacity (J/kg . °C)
H	Borehole depth (m)
M	Mass flow rate (kg/s)
Q	Heat transfer rate (W)
t	temperature(°C)
R _b	Borehole thermal resistance (m ² ·K/W)

Greek Letters

α	Thermal diffusivity (m ² /s)
β	Fraction of the injection heat rate of circuit

Subscripts

-in	Inlet of the borehole/GHE
-out	Outlet of the borehole/GHE
-f1	Fluid in down tube in a borehole
-f2	Fluid in up tube in a borehole
-‘	Fluid inlet temperature of a borehole/GHE
-“	Fluid outlet temperature of a borehole/GHE
-i	The number of the circuit
ε	Thermal efficiency of a borehole
τ	Time (s)

through U-shaped plastic pipes installed in boreholes. The vertical boreholes for GCHP system are generally connected in parallel before being connected to the heat pump through the pipes, as shown in Figure 1(left). In most industrial production processes, the industrial waste heat are generally featured by delivering thermal energy of over 50°C and requiring a relatively lower return temperature, such as 20°C. Obviously, the large temperature drop required by the production process cannot be implemented through the conventional parallel connection of boreholes which can mostly provide a maximum temperature difference of 5°C. In this study, a novel GHE configuration with multistage series is proposed to satisfy the high temperature gradient up to 20°C. The multistage series can not only effectively reduce the fluid temperature to a requested low value, but also make fully use of the industrial waste heat. Figure 1 illustrates the two different borehole connection configurations, i.e. the parallel connection for GCHP systems and the multistage-series connection for the seasonal thermal storage of industrial waste heat.

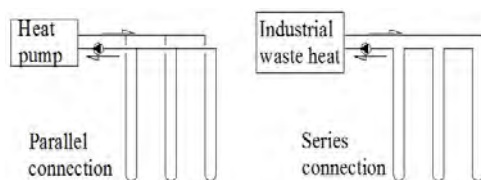


Figure 1 left: conventional GHE in parallel connection; right: multistage-series GHE for the thermal storage based on the industrial waste heat

Since the 1980s, a number of models have been developed to evaluate GCHP systems using vertical boreholes for injection and extraction of heat in the ground based on analytical, numerical and hybrid models^[2-6]. Recently, many studies have been carried out to investigate the performance of the seasonal storage of solar heat energy^[7-9]. However, most of them employed the conventional GHE configuration of in-parallel connection. Considering the high-grade thermal energy (70°C-90°C) injected into the ground and the relatively high temperature drop through the GHE, it is necessary to design a complex configuration of the GHE mixed with parallel and series connections. Based on the available quantity of the industrial waste heat, a representative

circuit with certain boreholes connected in parallel will be first designed and the numbers of the circuits connected in series will be determined according to the injection heat load and the required temperature drop. Little research is available to describe the heat transfer process of such a complex system. The main objectives of this research are to develop the heat transfer model of multistage-series GHE for the seasonal thermal storage and to further investigate the thermal performance of the system.

2. HEAT TRANSFER ANALYSIS OF VERTICAL GHE WITH MULTISTAGE-SERIES CIRCUITS

2.1 Assumptions for the heat transfer model

As for a single borehole, its heat transfer process can be directly analyzed by means of the same mathematical methods as the conventional vertical borehole in GCHP systems. The finite line source heat transfer model commonly used is employed here in this study. To keep the problem analytically manageable, the theoretical model of the multistage-series GHE is based on the following simplified assumptions:

- 1) The ground is regarded as a homogeneous medium with a uniform initial temperature t_0 , and its thermophysical properties do not change with temperature.
- 2) The heat transfer in the ground is assumed to be carried out solely by conduction with the neglect of groundwater advection, using an effective ground thermal conductivity. The moisture migration in the ground is also negligible.
- 3) Each circuit is composed of a number of individual boreholes connected in parallel with the same geometric parameters and thermal properties.

2.2 Heat transfer model of multistage series GHE

For a given GHE configuration with multistage series, the main objective of the thermal analysis is to determine the optimal heat injection/extraction rate (Q_i) of each independent circuit and to consequently obtain the inlet/outlet temperatures (t_i' , t_i'') of the circulating

fluid of each circuit. Figure 2 describes the physical model of the multistage series, where, R_{bi} and t_i are the borehole thermal resistance and the fluid temperature of the i th circuit, respectively.

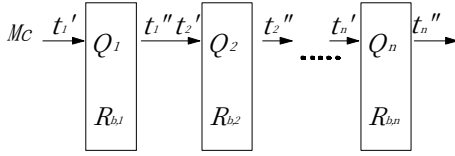


Figure 2 the physical model of multistage series

Based on the energy conservation law, the injection heat rate to the surrounding ground by i th circuit is equivalent to the heat rate transferred from the circulating fluid in the pipe to the borehole wall. The equation for the energy balance is given:

$$\begin{cases} Q_i = Mc(t_i' - t_i'') \\ Q_i = \left(\frac{t_i' + t_i''}{2} - t_{bi} \right) / R_{bi} \end{cases} \quad (1)$$

Where, t_{bi} is the representative borehole wall temperature of the i th circuit.

The inlet and outlet temperatures of the i th circuit can be easily derived according to Eq. 1:

$$\begin{cases} t_i' = t_{bi} + R_{bi}\beta_i Q + \beta_i Q / (2Mc) \\ t_i'' = t_{bi} + R_{bi}\beta_i Q - \beta_i Q / (2Mc) \end{cases} \quad (2)$$

Where, Q is the total injection heat rate by the GHE and $\beta_i = Q_i / Q$ is the recharging heat rate fraction which is defined as a ratio of the recharging heat rate occupied by the i th circuit to the total heat rate. The following two subsections presents the heat transfer models inside and outside the boreholes to achieve the thermal resistances and the borehole wall temperature as well.

2.3 Thermal resistance inside the borehole

The quasi 3-D model of the heat transfer inside the borehole is employed to calculate the total borehole thermal resistance, which takes into account the 2-D heat conduction in the transverse cross-section as well as the convective heat transfer in the axial direction by the fluid inside the U-tubes. The energy equilibrium equations can be written for up-flow and down-flow of the circulating fluid:

$$\left. \begin{aligned} -Mc \frac{dt_{f1}}{dz} &= \frac{(t_{f1} - t_b)}{R_1^\Delta} + \frac{(t_{f1} - t_{f2})}{R_{12}^\Delta} \\ Mc \frac{dt_{f2}}{dz} &= \frac{(t_{f2} - t_b)}{R_2^\Delta} + \frac{(t_{f2} - t_{f1})}{R_{12}^\Delta} \end{aligned} \right\} \quad (0 \leq z \leq H) \quad (3)$$

Two conditions are necessary to complete the solution:

$$\left. \begin{aligned} z=0, \quad t_{f1} &= t_{f1}' \\ z=H, \quad t_{f1} &= t_{f2} \end{aligned} \right\} \quad (4)$$

where

$$R_1^\Delta = \frac{R_{11}R_{22} - R_{12}^2}{R_{22} - R_{12}}, \quad R_2^\Delta = \frac{R_{11}R_{22} - R_{12}^2}{R_{11} - R_{12}}, \quad R_{12}^\Delta = \frac{R_{11}R_{22} - R_{12}^2}{R_{12}}, \quad R_{11}$$

and R_{22} are the thermal resistance between the circulating fluid and the borehole wall, and R_{12} is the resistance between the two pipes.

The general solution of this problem is derived by Laplace transformation, which is slightly complicated in form. At the instance of the symmetric placement of the U-tube inside the borehole, the temperature profiles in the two pipes were illustrated by Diao et al. (2004)^[10].

For the purpose of practical applications an alternative parameter $\varepsilon = (t_f' - t_f'') / (t_f' - t_b)$ is derived from the temperature profiles, which is named as the heat transfer efficiency of the borehole. It should be noticed that t_f' and t_f'' are the entering /exiting fluid temperatures to /from the U-tube. From the derived temperature profile the more accurate heat conduction resistance between the fluid inside the U-tube and the borehole wall can be calculated by,

$$R_b = \frac{H}{Mc} \left(\frac{1}{\varepsilon} - \frac{1}{2} \right) \quad (5)$$

2.4 Heat conduction outside the borehole

Based on the finite line source model, the temperature on the borehole wall, where $r = r_b$, was given by Zeng et al. (2002)^[11]:

$$t_b = \frac{q_i}{4\pi\kappa_g} \int_0^H \left\{ \frac{\operatorname{erfc} \left(\frac{\sqrt{r^2 + (z-h)^2}}{2\sqrt{a\tau}} \right)}{\sqrt{r^2 + (z-h)^2}} - \frac{\operatorname{erfc} \left(\frac{\sqrt{r^2 + (z+h)^2}}{2\sqrt{a\tau}} \right)}{\sqrt{r^2 + (z+h)^2}} \right\} dh + t_0 \quad (6)$$

It is noticed that the borehole wall temperature varies with time and borehole depth. The temperature at the middle of the borehole depth ($z=0.5H$) is usually chosen as its representative temperature.

2.5 Constraint conditions

For the n -stage series, following the scheme shown in Figure 2, the fluid outlet temperature of circuit 1 corresponds to the inlet temperature of circuit 2 and so on. Therefore, the coupling conditions are given as follows:

$$t_1'' = t_2', \dots, t_i'' = t_{i+1}', \dots, t_{n-1}'' = t_n'$$

The objective function is defined as:

$$\Delta = |\Delta_1| + |\Delta_2| + \dots + |\Delta_{n-1}| \quad (7)$$

$$\text{Where, } \Delta_1 = t_1'' - t_2', \quad \Delta_i = t_i'' - t_{i+1}' \quad (i=1, \dots, n-1).$$

It is noticeable that the objective function is a function of the variables β_i ($i=1, \dots, n-1$), i.e. $\Delta = f(\beta_1, \beta_2, \dots, \beta_{n-1})$ according to equation 2. The final aim of the thermal analysis for the GHE with n -stage series circuits is to seek the unique value of recharging heat rate fraction (i.e. β_i) and the inlet/outlet temperatures of each circuit for every time step with the constrain conditions. Theoretically, once the value of the objective function approaches to zero, the simulation model can determine the inlet/outlet temperatures of each circuit in the multistage series as well as the recharging heat fraction β_i .

2.6 The solution of the heat transfer model

The calculation program begins with an initial value of β_i , for example an assumption of equal distribution, $\beta_i = 1/n$ for the first time step. Then, the inlet/outlet temperatures of each circuit (t_i' and t_i'') can be estimated using the thermal resistances both inside and outside of the boreholes. The initial value of the objective function $\Delta = f(\beta_1, \beta_2 \dots \beta_{n-1})$ will be derived for the first time step. Based on the downhill simplex method, an optimal minimum value of the objective function will be found, which satisfies the constraint conditions. Finally, the recharging heat rate and the inlet/outlet temperatures of each circuit can be obtained for the first time step. For the next time steps, the thermal resistances of the ground will be calculated based on the previous calculated heat rates and the current heat rate for each circuit. The same procedure can be employed to complete the whole time steps.

3. RESULTS AND DISCUSSION

A copper processing plant is taken as an example of seasonal industrial heat storage of the GHE. The amount of the industrial waste heat rate available in the copper plant is about 10MW. To keep the storage region heat balanced during a whole year, both the injection and the extraction heat rates are set to be 10MW. The recharging heat time and the extraction time are both six months during a year. The recharging period is from April to September while the extraction period starts from October to March next year.

The GHE can be designed by using the simulation models with the given conditions and parameters. The designed GHE consists of 3300 boreholes with a depth of 100m and is arranged in a matrix of 33×100 with a spacing of 5 meters. The 3300 boreholes are divided into 3-stage series circuits each of which consists of 1100 boreholes connected in parallel configuration. Each borehole with a radius of 75 mm consists of a single U-tube. The HDPE (PE 3408) pipes are adopted with an outer diameter of 32 mm and an inner diameter of 26 mm.

According to the thermal response tests, the thermal conductivity and volumetric specific heat of the local soil are estimated to be 1.3W/m°C and 2.014×10⁶J/m³°C, respectively. The initial soil/rock temperature is measured to be 15°C.

Figure 3 illustrates the schematic diagram of the three-stage series GHE with sing U-tube boreholes. The objective function can be easily derived:

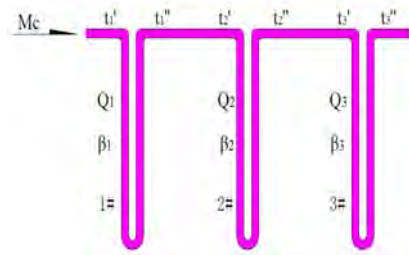


Figure 3 the flowchart of the GHE with 3-stage series

$$\begin{aligned}
 \Delta &= |t_1' - t_2'| + |t_2' - t_3'| \\
 &= \left| (\beta_1 R_1 - \beta_2 R_2) Q - \frac{(\beta_1 + \beta_2) Q}{2Mc} + t_{b1} - t_{b2} \right| + \\
 &= \left| (\beta_2 R_2 - \beta_3 R_3) Q - \frac{(\beta_2 + \beta_3) Q}{2Mc} + t_{b2} - t_{b3} \right|
 \end{aligned} \quad (8)$$

The downhill simplex method is employed here to solve the mathematical problem of three series circuits.

3.1 Discussion of the recharging heat rate fractions of different circuits

Figure 4 illustrates the variations of the heat rate fractions of three-stage series circuits against the simulation time. As Figure 4 shows, the fraction of the first circuit is significantly higher than the second one and consequently the second one is higher than the third one because the temperature gradient between the first circuit and the surrounding ground is highest compared to the other two circuits. The highest value of the fraction of the first circuit reaches 0.58 at the beginning of the operation. It is noticed that the sum of the fractions of the three circuits equals one.

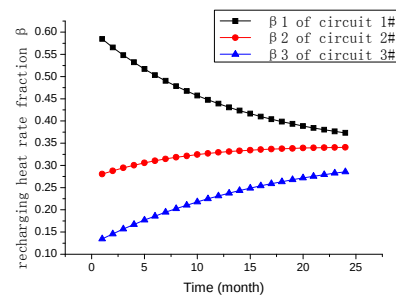


Figure 4 recharging heat rate fractions with simulation time

3.2 Effect of borehole spacing on the thermal performance

The heat losses to the surrounding ground are roughly proportional to the area exposed to the surroundings. For an industrial thermal storage with a large size of buried boreholes, the heat losses to the surrounding ground can be neglected.

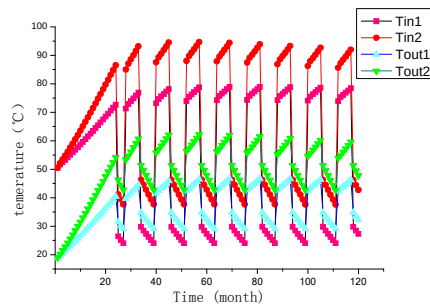


Figure 5 effect of borehole spacing on thermal performance (T_{in1} and T_{out1} are the temperatures of the fluid entering and exiting the GHE for 5 m spacing; T_{in2} and T_{out2} are the temperatures for 4 m spacing)

A smaller spacing between boreholes leads to a smaller storage volume and a significant thermal interference between boreholes. On the contrary, a larger borehole spacing can reduce the thermal shortcut among the boreholes and enhance the thermal performance of the heat injection/extraction process. The fluid inlet/outlet temperatures for two values of borehole spacing (4 meters and 5 meters) are monthly simulated for 10 years operation, as shown in Figure 5. As expected, the fluid temperatures in the case of the 5 m spacing are obviously lower than those in the case of 4m spacing. Consequently, the temperature of the storage volume with a larger spacing is lower compared to the closer spacing, which leads to a higher thermal performance. However, a large borehole spacing will increase the capital cost for a vast amount of land. Therefore, it is necessary to determine a reasonable borehole spacing considering both the capital cost and the thermal performance of the system.

3.3 Effect of ground thermal properties on the thermal performance

The ground thermal properties are the critical factors which can thermally influence to a large extent the storage performance of the GHE. In this study, three types of soils (clay, sandy clay and siltstone) are selected as storage mediums for comparisons of their individual heat transfer capability. The thermal properties of these soils are listed in table 1.

Table 1 Thermal properties of different soils

Soil type	Density kg/m^3	Heat capacity $\text{J/kg} \cdot ^\circ\text{C}$	Thermal conductivity $\text{W/m} \cdot ^\circ\text{C}$	Thermal diffusivity $\text{X}10^{-6} \text{m}^2/\text{s}$
clay	1285	1200	0.8	0.519
Sandy clay	1925	1000	2.2	1.143
siltstone	2570	1556	3.2	0.8

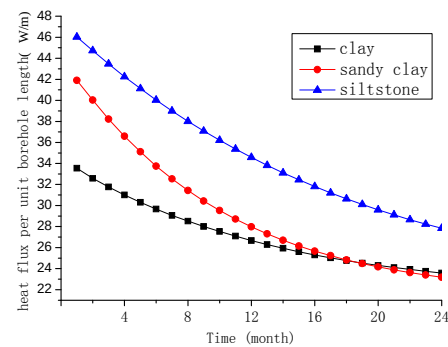


Figure 6 Effect of the soil thermal properties on recharging heat rate per unit borehole length

Figure 6 shows the sensitivity of the injection heat rate per unit length of borehole to the soil thermal properties for the three cases of soils during the two years of continuous recharging. Evidently, all the heat transfer rates of the three kinds of soils are decreased with time due to the gradually reduced temperature gradient between the fluid and the soil. The siltstone behaves the highest heat flux because of its largest density and conductivity. The heat transfer rate of the sandy clay is larger at the beginning of the heat recharge because of its larger thermal diffusivity and becomes lower compared to the clay at the end of the simulation. This is due to the lower heat capacity of the sandy clay, which results in a higher temperature rise among the surrounding soil and consequently a performance degradation. Therefore, the recharging heat performance of the GHE in the soil is comprehensively influenced by the thermal conductivity and heat capacity of the soil.

4. CONCLUSIONS

A novel borehole configuration of multistage-series circuits is proposed for the seasonal thermal storage based on industrial waste heat. A simulation model of multistage series circuits is developed to illustrate the heat transfer process of the underground thermal storage. The simplex method is employed to solve the outlet temperature and the recharging heat rate of each circuit.

A set of parameters that strongly affect the thermal performance of the GHE with multistage series circuits is discussed in detail. The recharging heat rate fraction of the first circuit is largest among all the circuits due to the largest temperature gradient between the borehole and the surrounding soil. It is noteworthy that the soil temperature rise or decrease nearby the former circuits is significant compared to the soil temperature changes nearby the latter circuits. As a result, the heat transfer efficiency of the former circuits reduces considerably. In order to equilibrate the ground temperature for multistage circuits, it is recommended to reverse the flow direction at regular time intervals to enhance the total heat transfer efficiency of the GHE system. The influence of the borehole spacing on thermal performance is considerable, especially for closer borehole spacing. A larger borehole spacing results in a larger storage volume and consequently reduce the effect of the thermal interference among boreholes.

The ground thermal properties also play an important role in determining the GHE size and the thermal efficiency of the system. It is necessary for designers and engineers to obtain dependable on-site data of the thermal properties of the ground as a basis for designing an optimum seasonal thermal storage system.

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ENERGY STORAGE FOR RESIDENTIAL BUILDINGS: REVIEW AND ADVANCES

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ABSTRACT

Residential buildings are mostly sensitive to climatic variations, where building envelopes work as the interface between indoor and outdoor environments preventing heat gain in the summer and heat loss in the winter. Proper use of energy storage technologies may largely reduce the energy needs in residential dwellings while delivering better indoor environment quality. This paper provides a brief review on several energy storage technologies, both active and passive, for residential building applications. Particular attention is paid to the usage of phase change materials (PCM), which has been studied for a few decades with a growing interest recently. Modeling methods of PCM-embedded wall systems are comparatively reviewed and a new simulation program has been developed that can simulate the thermal and energy performance of PCM-embedded walls and buildings in a much stable and fast manner. The paper further presents a case study that integrates PCM with the traditional Kang heating system for the residential dwellings in Northeastern rural China. Both technical and economic performances of the solution are explored.

Keywords: Phase change material, Energy modeling, Residential building, Kang.

1. INTRODUCTION

Residential buildings are mostly sensitive to climatic variations, where building envelopes work as the interface between indoor and outdoor environments preventing heat gain in the summer and heat loss in the winter. Proper use of energy storage technologies may largely reduce the energy needs in residential dwellings while delivering better indoor environment quality. When developing proper energy storage solutions for residential applications, several important factors are to be considered, including: energy density, efficiency, power output, toxicity, lifetime, reliability, self-discharge, duration of storage, and cost. Before these factors can be fairly assessed, it is important to analyze the potential end-uses of the stored energy. According to a survey by the U.S. Energy Information Administration, on average about 59% of residential energy use goes to space or water heating, 12% goes to air conditioning and refrigeration, and 29% goes to other electrical needs [1]. Therefore, it is understandable that most research in residential energy storage has been on thermal storage. Converting thermal energy into electricity has extremely low efficiency at realistic temperatures for residential applications. As a result, energy storage methods for generating electricity require particular attentions and research efforts. Battery and compressed air are potential energy storage methods that have been investigated for residential purposes. Although more efficient than other approaches, these two solutions may not be economically-wise. This paper provides a brief

review on various energy storage technologies, both active and passive, for residential building applications. Particular attention has paid to thermal energy storage (TES) by using phase change materials (PCM), which has been studied for a few decades with a growing interest recently. Modeling methods of PCM-embedded wall systems are comparatively reviewed and a new simulation program has been developed that can simulate the thermal and energy performance of PCM-embedded walls and buildings in a much stable and fast manner. The paper further presents a case study that integrates PCM with the traditional Kang heating system for the residential dwellings in Northeastern rural China. Both technical and economic performances of the solution are explored.

2. REVIEW OF ENERGY STORAGE TECHNOLOGIES FOR RESIDENTIAL BUILDINGS

There are many potential energy storage approaches. This paper has been focused on thermal energy storage (TES), battery, and compressed air energy storage (CAES), as these appear to be the most relevant and promising for residential applications. TES can be categorized into sensible, latent, and chemical; and chemical can be further categorized into sorption and reversible chemical reactions. Table 1 highlights some advantages and disadvantages of each energy storage technology.

Table 1: Summary of advantages and disadvantages of different energy storage methods

	Advantages	Disadvantages
Sensible	Cheap, Reliable, Long lifetime, Non-toxic materials	Low energy density, Self-discharge
Latent	More dense than sensible, Isothermal, Some materials are cheap and abundant, Non-toxic materials	Many materials are flammable or corrosive, Phase separation may occur
Chemical	Extremely high energy density, No self-discharge	Low power output, Low thermodynamic efficiency
Battery	High efficiency, High power	Short lifetime, Toxic materials
Compressed Air	Reliable, No self-discharge, Long lifetime	Mediocre energy efficiency
	Non-toxic materials	

2.1 Thermal Storages

Thermal energy storage is the most studied form of energy storage for residential applications. TES can be organized into three categories: sensible, latent, and chemical, as described in the following subsections. Figure 1 shows the comparison in energy densities of these TES systems, with the higher density systems typically becoming more difficult to engineer and/or more costly as of today. Detailed reviews of thermal energy storage were performed by Pinel et al. [2] and Li et al [3].

(1) Sensible

Sensible heat storage (SHS) is one of the simplest, cheapest, and most reliable forms of energy storage available. This method consists of heating up some material and thermally isolating it from the environment until the heat is needed. The amount of energy which sensible heat can store depends solely on the heat capacity of the material and the temperature at which it is kept. The main disadvantages of sensible heat storage are that it has one of the lowest energy densities of any energy storage system, and it will self-discharge over time. Nevertheless, sensible heat storage has been one of the longest and most widely used forms of energy storage. The most commonly used SHS material is water, because water is cheap, abundant, non-toxic, and it has a relatively high thermal capacity of about 4.2 kJ/kgK. The water for sensible heat storage can be stored in tanks, aquifers, or other bodies of water such as ponds. Other material systems for sensible heat storage include rock beds and ground soil. A major area of research for the improvement of sensible heat storage is on stratification. Stratification is used to help maintain the highest possible temperature difference between the stored heat and the heat source, and therefore increase the heat transfer to the storage, as illustrated in Figure 2. Chung et al. [5] found that the diffuser shape in a stratified thermal storage tank plays a significant role in the performance of the system. Moreover, they proposed a new design for a diffuser in a rectangular water tank which improves stratification. Other research groups have

designed segmented storage, variable position inlets, and other devices to reduce mixing and improve sensible storage performance.

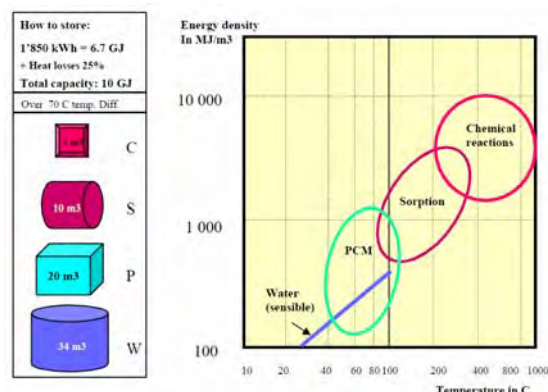


Figure 1: Energy density vs. temperature (log scale) and comparison of theoretical volume needed to store 10 GJ of heat based on four different principles (taken from J. Hadorn [4])

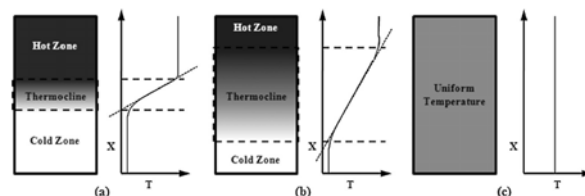


Figure 2: Differing levels of stratification within a storage tank with equivalent storage energy: (a) highly stratified, (b) moderately stratified, and (c) fully mixed (or unstratified) (taken from Pinel et al. [2])

(2) Latent

Latent heat storage (LHS) uses phase change materials (PCMs) to store thermal energy. Materials must be selected which have a phase change of liquid to gas, or more typically solid to liquid, at the working temperature. Other considerations for choosing a phase change material are its heat of fusion (how much heat it can store in its phase transformation), thermal conductivity, toxicity, corrosiveness, and phase separation characteristics. PCMs are typically classified as organic (paraffin, fatty acids, alkanes) or inorganic (salts) [2]. Table 2 presents the specific advantages and disadvantages between organic and inorganic PCMs. LHS for residential applications has been extensively researched over the past few decades, and the types of studies out there are abundant. A more thorough review of latent heat storage can be found in the studies of Zalba et al. [6] and Tyagi et al. [7].

Table 2: Advantages and disadvantages between organic and inorganic PCMs (reproduced from Zalba et al. [6])

	Advantages	Disadvantages
Organics	No corrosives, Low or no undercooling, Chemical and thermal stability	Lower phase change enthalpy, Low thermal conductivity, Flammability
Inorganics	Greater phase change enthalpy	Undercooling, Corrosion, Phase separation, Phase segregation, Lack of thermal stability

PCMs are best placed in the envelope of the building, i.e., the walls, roof, and floor [8]. This is because the envelope offers larger areas for heat transfer to and from the rooms of the building. With passive solar heating, heat is collected from the sun and stored in the PCM envelope. It is then released at night when it is cooler. Active heating is a similar concept, where the PCMs are charged (heated) by electricity during the day with either photovoltaics or cheap grid electricity. The heat is then released once again at night when it is cooler. For night cooling, the PCMs are ventilated by cool night air to remove heat from the PCMs. The PCMs can then absorb heat during the day to help cool the inside of the building. There is much research on composite materials and encapsulation of PCMs to enhance the thermal conductivity, dimensional stability, and shape retention. For example, Li et al. [9] micro-encapsulated paraffin (the PCM) and then blended it in with a high density polyethylene/wood flour composite matrix. The composite matrix acted to improve the shape retention and dimensional stability of the system. The micro-encapsulation acted to reduce reaction and evaporation of the paraffin with the outside environment, and increase heat transfer area. Their experiments and simulations agree that the PCM had proper phase transition temperature, large apparent specific heat in phase change temperature region, suitable thermal conductivity, good shape retention, and dimensional stability. They also performed a cost-benefit analysis to show that this PCM has the potential to be implemented into an electric floor heating system.

(3) Chemical

Chemical storage can be separated into chemical reactions and thermo-chemical (sorption) processes. Chemical storage in general has some great advantages, including very high storage density and no self-discharge. However, there is still a lot of research that needs to be done to find materials that have the desired operational properties, higher coefficients of performance, and that are economical. A more comprehensive review on chemical storage can be found in the study by Cot-Gores et al. [10].

(i) Sorption processes

In sorption processes a solid or liquid adsorbs or absorbs a gaseous or liquid phase component naturally. To use these processes for TES, heat is applied to charge the system and remove the adsorbed or desorbed component from the sorbent. The two are then kept

separate until energy is required. When energy is required, the system is discharged by bringing the two components back together, and heat is released. Figure 3 illustrates this charging and discharging process for a closed sorption system. So far the reactants studied for sorption energy storage have been ammoniates, hydrates, and metal hydrides [10]. Reactants are usually chosen for specific applications, depending on what the thermodynamic constraints are. The main issues with chemical sorption are poor heat and mass transfer in the reactive beds, which lead to low power systems. More detailed theory on sorption processes for TES is provided by Hauer et al. [11].

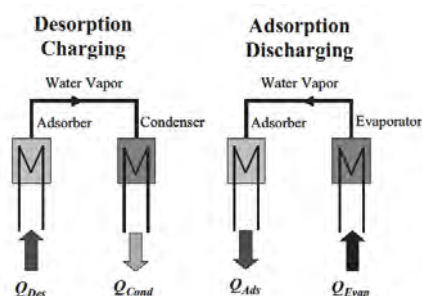


Figure 3: Closed sorption system (taken from Hauer et al. [11])

One example of a sorption energy storage system was built by Mauran et al. using SrBr_2 [12]. They chose this reactant, because it satisfied the thermodynamic constraints such as: using a solar heat collector with $<80^\circ\text{C}$ temperature generation, having a minimum ambient temperature of 7°C , and having a maximum ambient temperature of 35°C , along with other constraints. It also does not require toxic chemicals, and it has a high energy density. Their prototype had a total volume of 1m^3 , and it was able to store 60kWh for heating and 40kWh for cooling. This is an order of magnitude less than what is predicted theoretically, because this is what was needed to have somewhat acceptable heating or cooling powers. However, they still did not achieve heating and cooling powers that would be good enough for commercial use. This is because there is low heat transfer between the reactive layer and the exchanger wall, which Mauran et al. [12] plans to address with future work.

(ii) Reversible chemical reactions

The basic concept behind using reversible chemical reactions as energy storage is to heat a material so that it separates into two other components, then separate those components and store them until energy is needed. When energy needs to be generated, the components are then recombined to create heat. Figure 4 illustrates the processes for reversible chemical reaction energy storage. Some important things to consider when choosing reaction components are the reaction temperature, the reactivity of the reactant, and the heat and mass transfer of the reactor and the heat exchanger. Reversible chemical reactions provide some of the highest energy densities of any of the energy storage methods, with storage densities on the order of GJ/m^3 [2]. The research on chemical

reactions has been focused on hydration and carbonation of metal oxides. More specific theory for reversible chemical reaction energy storage can be found in the study of Ervin et al. [13].

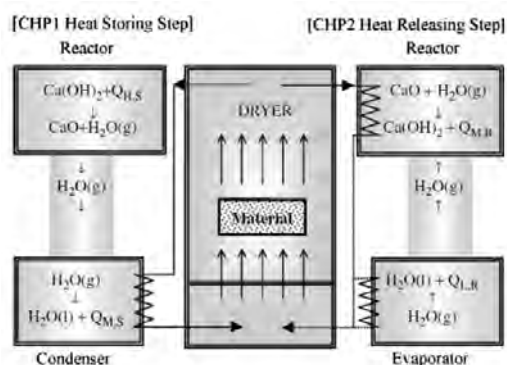


Figure 3: Schematic diagram of a CaO/H₂O/Ca(OH)₂ chemical heat storage system (taken from Cot-Gores et al. [10])

Like with sorption storage, the main problems facing chemical reactions are poor heat and mass transfer in the reactive bed, and low thermodynamic efficiency [10]. Therefore, much research has gone into improving this performance using composite materials. One method to improve the thermodynamic performance of these chemical reaction systems is to engineer materials with the desired operating conditions. For example, Kato et al. [14] made a composition of mixed metal oxides of $Mg_{0.8}Ni_{1.0}(OH)_2$ to create a reaction with a decomposition temperature between 200-300 °C. They found that this temperature range was best suited for internal combustion engine waste heat, industrial process waste heat, and, in relation to residential energy storage, many solar energy systems. Most reactions of simple metal oxides do not have these decomposition temperatures, and the ones that do are not very reactive with water. By mixing highly reactive MgO with low decomposition temperature NiO, a more suitable compound may be created for typical residential energy storage applications.

2.2 Batteries

Batteries are one of the most studied energy storage methods of any kind. However, they are mostly utilized in small-scale applications, and offer a lot of challenges when bringing them up to the residential scale. The benefits of using batteries are that they provide extremely high energy efficiency as well as power output. The main problems with batteries are: they lose their capacity over their lifetime, they have notable self-discharge, some use toxic materials, and they can be flammable. Lead-acid, Nickel cadmium (NiCd), Nickel metal hydride (NiMH), and Lithium ion (Li-ion) are the most prominent batteries available today. Table 3 summarizes the advantages and disadvantages between the four.

Table 3 Common battery comparisons (derived from Nair et al. [15])

	Advantages	Disadvantages
Lead-Acid	Mature technology, Low investment cost, Lowest self-discharge, Easy maintenance	Short cycle life, Poor performance and un- ideal temperatures, Failure due to deep and continuous cycling, Environmentally unfriendly
NiCd	Mature technology, Durable and reliable	Made of toxic materials, Severe self-discharge
NiMH	Environmentally friendly, Higher energy density, Lower cost than Li-ion	Severe self-discharge
Li-ion	Nearly 100% energy storage efficiency, Highest energy density	High investment costs, Complicated charge management

Batteries are already being used today in homes for solar power energy storage or peak demand shaving. Leadbetter et al. [16] did a study to model, simulate, and size battery energy storage systems for residential electricity peak demand shaving. They predicted that with the proper usage, which they specified, a lithium ion battery system could last 10-20 years. Moreover, they found that with a 5kWh battery system a house in certain regions of Canada could reduce the peak demand by between 42% and 49%. Other regions required an 8kWh system for the same demand shaving, while Quebec required 22kWh for only 28% demand shaving due to the massive heating load from electric heaters. There are also promising new battery technologies in the making, such as solid-state batteries. Solid-state batteries show a lot of potential as being a safe (non-flammable), highly robust battery technology, which is anticipated to be commercial within 3-5 years.

2.3 Compressed Air

Compressed air energy storage (CAES) is a simple, well-understood technology. The idea is to drive a compressor to compress air and store it in a tank or cavern. When energy is needed, the compressed air is expanded through a turbine to produce electricity. CAES has been investigated for utility scale energy storage for multiple decades. The first CAES plant was a 290MW plant built in Huntorf, Germany in 1978 [17], but only in the last few years has there been a push to move this technology forward. It turns out that at the utility scale, CAES shows a lot of promise for a relatively low cost energy storage method. However, bringing CAES to the residential scale is a whole other consideration - one that only few have investigated. The advantages of CAES are that it is reliable, it has negligible self-discharge, it does not use toxic materials, and CAES systems have very long lifetimes. The typical problem with CAES is that during compression there is a rise in temperature of the air, and heat is lost to the environment. Moreover, during expansion there is a drop in temperature, and an outside source of heat energy is required to bring the air back to room

temperature. These effects lead to a fairly low thermodynamic efficiency. Typically natural gas is used to heat the air during expansion, but this defeats the purpose of a fully renewable energy system. Moreover, when they have used this technique at plants such as the one in Huntorf, Germany, they have still only reached overall efficiencies of about 50% [18].

There have been several novel techniques developed to increase the efficiency of CAES, but the most notable one for residential purposes is CAES in combination with TES for tri-generative heating, cooling, and power generation. The idea is to use a thermal energy storage system to not only store solar thermal energy, but also the heat during compression of air that is normally wasted. The thermal energy is

then used to heat the rooms, water, and compressed air during expansion for electricity. Moreover, if cooling is required, air is rapidly expanded, and this cold air circulates the house. Li et al. [19] of UK have designed and analyzed such a system, shown in Figure 4. Their results showed a comprehensive efficiency of about 50% in winter months and 30% in summer months. Although this may seem low, they claim that these efficiencies are higher than conventional tri-generation systems which are typically based on absorption chilling technology. Moreover, absorption chillers and boilers not only require high capital costs, but many also involve the use of toxic/harmful materials such as LiBr as the working fluid [19].

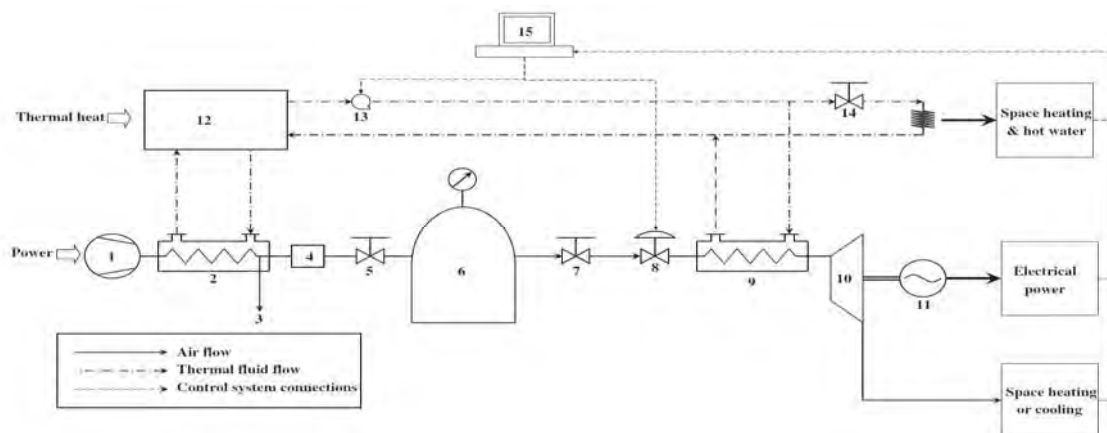


Figure 4: Schematic of the proposed micro-trigeneration system.

(1) air compressor, (2) cooler, (3) drainer, (4) dryer/moister trap, (5), (7) and (14) cut-off valve, (6) high pressure vessel, (8) regulating valve, (9) heat exchanger, (10) expander, (11) generator, (12) heat storage tank, (13) pump and (15) control system. (taken from Li et al. [19])

3. MODELING OF PCM-EMBEDDED WALL SYSTEMS

The comparison of various energy storage solutions reveals that latent heat storage (using PCMs) is probably one of the most viable solutions at the current development stage. Due to its ability of storing significant thermal energy within a small volume, PCM appears one of most promising technologies for developing energy efficient buildings, particularly useful to reduce the inherited climatic deficiency in light weight buildings. To quantify the technical and economic feasibility of using PCM for building applications, computational models of PCM-based TES are highly desired. Computational modeling offers inexpensive alternative to solve phase change heat transfer problems when compared to field studies for analyzing, optimizing and fine-tuning designs. For wall systems with constant properties, heat transfer mechanisms can be readily solved using variety of approaches. However, when PCMs are utilized, special treatment is necessary to account for dynamic heat storage and release. The following sections provide an overview of general mathematical methods, numerical schemes used to model PCMs, and modules

capable of simulating PCMs in whole building simulation tools. Furthermore, this paper introduces a new development of a design toolbox that can be used to evaluate classical and advanced facade system designs with and without PCMs.

3.1 Mathematical Methods and Numerical Schemes for Modeling Phase Change Problems

Four mathematical models are commonly used to simulate heat transfer associated with phase change: Enthalpy Method, Heat Capacity Method, Temperature Transforming Method, and Heat Source Method [20]. Different numerical schemes can be used to solve the discretized forms of the heat transport equations in the above methods. For instance, the discretized nonlinear heat transport equation of the enthalpy method with an implicit time step scheme can be resolved using the Newton's method with the aid of an auxiliary temperature-enthalpy function. Alternatively, the nonlinear terms can be linearized using the methods explained by Patankar [21] and subsequently solved using iterative algorithms. While iterative algorithms such as Gauss-Seidel are inherently slow, fast numerical schemes have been developed to improve the

computational efficiency [22-24]. Pham [22] for example proposed a method that combines the features of the enthalpy and heat capacity methods. The method consists of two steps: a prediction step followed by a correction step. The method is not iterative and therefore may not conserve energy at every time step especially with large time steps. A better conservative iterative scheme was proposed by Swaminathan and Voller [24]. The method iterates the predicted and corrected intermediate results until a convergence is achieved. The most evident advantage of both methods is their computational efficiency with the latter being conservative at each time step. Table 4 summarizes the main features, mathematical methods and potential

numerical schemes commonly used for solving phase change problems.

3.2 PCM Models Integrated into Whole Building Simulation Programs

Building simulation programs are essential design tools for architects and engineers to evaluate new concepts and test advanced technologies that may improve the energy and thermal performance of buildings. Several building simulation tools are listed at the U.S. Department of Energy (DOE) web directory [25]. The twenty prevalent whole building energy simulation programs that are considered accurate and capable of handling the dynamic behaviors of a building and its systems are reviewed by Crawley et al. [26].

Table 4 Features, mathematical methods and numerical schemes used for phase change problems

Math Method	Heat Equation Formulation	Main Features	Potential Numerical Schemes
Enthalpy Method	$\rho \times \frac{\partial h}{\partial t} = \frac{\partial}{\partial x} (k \frac{\partial T}{\partial x})$	Enthalpy, h , accounts for sensible and latent heat	1) For nonlinear equation: Iterative scheme with Non-linear solvers (e.g. Newton's methods) 2) For linearized equation: • Corrective iterative scheme • Correction non-iterative scheme
Heat Capacity Method	$\rho \times C^A(T) \times \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} (k \frac{\partial T}{\partial x})$	Heat capacity, $C^A(T)$, accounts for both sensible and latent heat	Iterative Scheme (e.g. Gauss-Seidel iterative scheme) if a proper heat capacity approximation is selected
Temperature Transforming Method	$\rho \times C_{eff}(T) \times \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) - \rho \times \frac{\partial S}{\partial t}$	Heat capacity term, $C_{eff}(T)$, accounts for sensible heat and source term, $-\rho \times \frac{\partial S}{\partial t}$, accounts for latent heat	Iterative Scheme (e.g. Gauss-Seidel iterative scheme) after linearizing the source term
Heat Source Method	$\rho \times C_{avg}(T) \times \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) - \rho \times L \times \frac{\partial f_l}{\partial t}$	Heat capacity term, $C_{avg}(T)$, account for sensible heat and Latent heat, $-\rho \times L \times \frac{\partial f_l}{\partial t}$, is treated as a source term	Iterative Scheme (e.g. Gauss-Seidel iterative scheme) after linearizing the source term

Based on a recent study, few whole building simulation programs can handle the thermal performance of building envelope with PCMs such as EnergyPlus, TRNSYS, ESP-r, and BSim [20].

Table 5 summarizes the simulation programs, the implemented modules and their limitations.

It is noted that most PCM models integrated into whole building simulation programs are based on the heat capacity method. Hence, it becomes necessary to reduce the typical one hour time step to a very small time step to achieve acceptable level of accuracy. For annual thermal performance evaluation, such building simulations become computationally inefficient since iterative methods are used in each time step. In addition, the convergence may not be attained due to numerical instability especially when PCM enters or leaves the phase change region. With all constraints and limitations above, none of whole building simulation programs are currently implementing efficient mathematical models that are quick, accurate and numerically stable at realistic time steps.

Table 5: Methods and modules for latent heat evolution in whole building simulation programs

Whole Building Simulation Tool	PCM Modules	Methods Used for Latent Heat Liberation	Limitations
EnergyPlus*	CondFD	Heat Capacity Method	- Time step < 3min - Small grids - Hysteresis in PCM is not modeled - Phase change at isothermal temperature is not modeled
TRNSYS**	TYPE204***	Heat Capacity Method	- Computationally inefficient - Small time step to avoid instability
ESP-r*	SPMCMP53-SPMCMP56	- Heat Capacity Method - Heat Source Method	Small time step to avoid instability
BSim**		Heat Capacity Method	Small time step to avoid instability

*: free public program; **: commercial program; ***: TYPE204 is the only model available for public use in TRNSYS

3.3 Development of a Standalone Toolbox in SIMULINK

Advanced façade systems are becoming important architectural elements in modern buildings. While many façade systems are installed for architectural aesthetics, they can potentially be utilized to perform multiple functions such as heat storage using enhanced-PCMs components and abundant solar energy. When this strategy is embraced, the abundant solar energy is captured by charging the PCMs, and heat transfer medium such as air or water can then be utilized to distribute the stored energy to where it is needed. The whole system acts like a living environment where fluid (e.g., air, water or both), building massive materials (e.g., PCMs, or other layers), and energy source (e.g., sun, or internal building load) or energy sink (e.g., sky or ground) are interacting with each other in a sequential or simultaneous manner to transport and utilize energy. However, there is no tool yet that helps quantify the thermal performance of these combined systems. The overall goal was then to provide a framework tool or a library of modules that can be easily used to achieve this task.

Advanced Façade Systems “AdvFacSy” tool is developed for this purpose using MATLAB/SIMULINK environment. The heat transfer mechanism of PCMs is handled using the enthalpy method with the iterative correction scheme proposed by Swaminathan and Voller [24]. The toolbox consists of several necessary components including weather file reader and processor, and solar radiation model, glazing module, generic wall module, cavity module and cavity wall module. With few blocks, different wall designs as illustrated in

Figure can be easily generated and assessed under different climatic conditions. The main advantages of the developed standalone toolbox are the quick, reliable and the trouble-free evaluation of the complex façade systems by drag, drop and link blocks in one SIMULINK screen. In addition to envelope designs, the toolbox can be used to evaluate different operational strategies to charge and discharge energy stored in PCMs using the air in the cavity as a heat transfer medium. The air is driven by either thermal buoyancy or mechanical fan. The air can be induced from the outdoor environment or indoor via air vents located at top and bottom of the façade system. The controlling mechanism facilitates the charging and discharging of PCMs in the wall unit. The recovered heat can then be directly used to meet the heat demand, or used to preheat fresh air before treated centrally in air conditioning system, or transported to other heat storage medium for later usage. Figure 6 demonstrates the simulations of three wall structures with and without PCMs. It is clear that proper design of wall layers and selection of PCMs are critical to maintain appropriate wall surface temperatures. Amount of heat that can be

collected through air cavity/channel is also heavily dependent on these two factors.

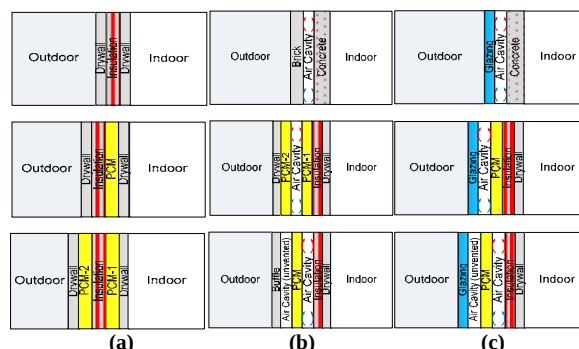


Figure 5: Classical and advanced façade designs with and without PCMs in “AdvFacSy” Toolbox
[(a) Multi-layer wall; (b) Ventilated opaque multi-layer wall; (c) Ventilated transparent multi-layers walls]

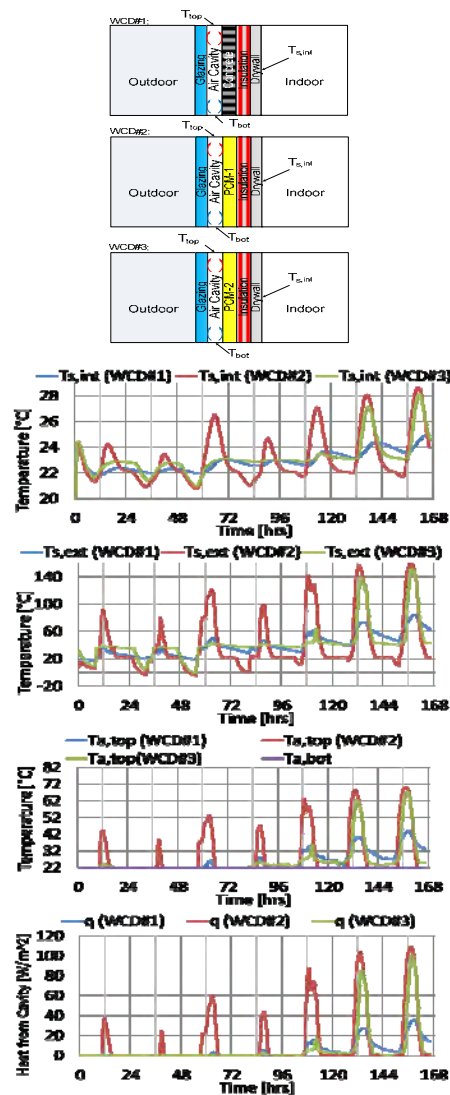


Figure 6: Simulation results of three wall structures
(T_s – wall surface temperature; T_a – air temperature in cavity; q – heat collected by air in cavity)

4. APPLICATION OF PCM FOR RESIDENTIAL DWELLINGS IN RURAL CHINA

The developed toolbox can be used to analyze various building structures with and without PCMs. Besides being integrated into building envelopes, PCMs can be placed at different locations of a space to store energy from various potential sources, such as the waste heat from cooking. This section demonstrates a special application of integrating PCM with a traditional Kang heating system for the residential dwellings in Northeastern rural China. Traditional stove and Kang-platform systems are used to heat more than 85% of homes in the northeast region of China [27]. The Kang is a traditional bed platform made from masonry, adobe and cement. An inner chamber functions as a flue for hot smoke produced from a connected stove. Energy is transferred from the smoke to the Kang as smoke travels through the Kang chamber and out a connected chimney, see Figure 7. Heat from the smoke is stored in the thermal mass of the Kang. The stored heat is then released into the air of the room containing the Kang. Residents sleep on the surface of the Kang that will stay warm several hours after the firing of the stove. Research and experience show that the current system is inefficient and requires high levels of fuel consumption. During the firing period and the time period directly afterwards, the Kang's body remains warm (sometimes too hot) and transfers energy to the room's air. However, several hours after the stove is fired, the Kang and air temperatures drop quickly and largely, sometimes below 10°C, reducing the thermal comfort of the home's residents. Many residents thus rely on coal fired boilers and radiator systems to achieve a comfortable temperature in their home. Much research has already been conducted regarding possible solutions, for instance, improved building envelope insulation, coupling the Kang with passive solar systems, and the use of PCMs.

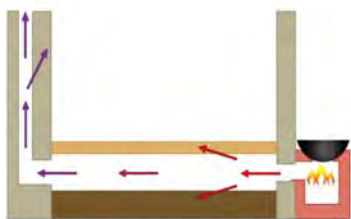


Figure 7: Kang system during stove firing

This study investigates the effects of adding a PCM to the surface of the Kang. The PCM does not add any energy to the system, functioning only as a thermal battery that can store heat that can be released at a later time. Figure 8 shows the effects of adding 70 kilograms of PCM to the surface of the Kang. The melting point of the PCM is set at 28°C and is not exceeded during the simulation. 70 kilograms of PCM are required to

achieve thermal saturation. Less than 70 kilograms results in the PCM's temperature increase above 28°C. The simulated results reveal that the PCM does not increase the temperature of the Kang or air significantly (7.73% and 2.85%, respectively). It does flatten the Kang's surface temperature curve which would make the surface more comfortable for residents. Generally, the PCM cost includes material and insulation costs. Insulation would require reconstructing the top surface of the Kang. It was assumed this job would take 12 hours and the costs were calculated similarly to installation costs for the room radiator system. The PCM's life cycle is much lower than other systems. A typical PCM can last 10,000 phase changes [28]. If the PCM melts and freezes during each firing, that is 6 phase changes each day. This makes the life span of the PCM roughly 5 years. At the end of 5 years, new material would have to be purchased and re-installed. If a PCM with longer life was implemented, the annual cost for the PCM would be less. Table 6 summarizes the performance comparison of PCM with other potential improvement measures (i.e., adding insulation to envelopes; using hot water radiator from cooking waste heat - placed in room and Kang, respectively; adding solar hot water, or simply using coal for space heating) for Northeastern rural China.

Table 6: Improvement feasibility matrix

Improvement	Improvement Feasibility				
	Technical			Economic	Cultural
	Air	Kang	Bathing		
Insulation	●	○	○	●	●
Room Radiator - 500 L Tank	○	○	○	○	○
Kang Radiator - 500 L Tank	○	●	●	○	○
SHW	○	○	●	●	●
PCM	○	●	○	○	○
Coal	●	○	○	○	○

5. CONCLUSIONS

This paper reviews several energy storage technologies with potentials for residential building applications. Although technically sound, some technologies are still not quite mature yet for wide and commercial deployment. The importance of the cost of an energy storage system cannot be stressed enough when assessing an energy storage solution. If the cost per unit energy of a given system does not compete in today's (or near future) market, it will not survive no matter how high of performance it may provide. Not many research focus on the economics of their residential energy storage concepts/designs. It is known that most of the costs of energy storage systems are upfront, and so they can be unappealing to purchase at first. Nevertheless, to determine if an energy storage system can compete economically, one must find the expected energy output of the system over its entire lifetime, and divide that by the total cost of the system over its lifetime. As of January 2013, the average cost of electricity in the U.S. was \$0.129/kWh [29]. It should also be acknowledged that natural gas prices are hovering around only \$0.013/kWh in the U.S. as of March 2013 [1]. Therefore, using natural gas as

an energy reserve to supplement renewable energy storage systems is currently very economically attractive. No energy storage systems can compete with those prices as of today. However, if one assumes increases in natural gas costs over the lifetime of an energy storage system, along with governmental incentives for sustainable energy implementation, some current sustainable residential energy storage methods may become competitive.

The paper demonstrates the effectiveness of using PCMs to manage the surface and air temperatures in

buildings by developing a generic toolbox that can predict and optimize the performance of building structures with and without PCMs. The results indicate that proper use of thermal energy storage technologies may largely reduce the energy needs in residential dwellings while delivering better indoor environment quality. It is echoed that the economic and cultural impacts need be considered when determining proper technologies for specific applications.

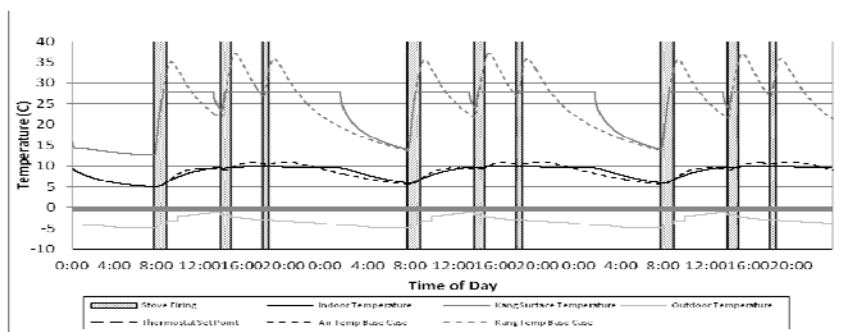


Figure 8: Predicted room air and Kang surface temperatures with and without using PCM

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STORAGE OF SOLAR ENERGY BY PHOTOCATALYTIC HYDROGEN PRODUCTION USING HIGHLY-REACTIVE-FACET PT-TIO₂ NANOSHEETS

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ABSTRACT

Hydrogen is a promising fuel for storage of renewable energy and can be produced by solar photocatalytic water splitting to store solar energy. In this study, Pt-TiO₂ nanosheets with reactive facets were prepared by hydrothermal method, followed by photo reduction of Pt(acac)₃ on doped TiO₂ nanosheets. The as-prepared materials were characterized by transmission electron microscopy (TEM) and UV-vis diffuse reflectance spectroscopy (DRS). The performance of photocatalytic hydrogen production was investigated in methanol aqueous solution under UV-visible irradiation. The results show that the Pt loading, TiO₂ nanosheet dimensions and photocatalyst loading have significant effects on photocatalytic hydrogen production. The maximum hydrogen production rate obtained was 5,086 $\mu\text{mol/h/g}$ with 1.0 wt.% Pt loading TiO₂ nanosheets. It is concluded that TiO₂ nanosheets of small size yield the best performance. Moreover, small amount of CH₄ was also detected by gas chromatographic analysis and regarded as one of the intermediates for hydrogen production.

Keywords: Photocatalytic hydrogen production; Titania; Nanosheets; Reactive facets; Water splitting.

1. INTRODUCTION

Hydrogen was proposed as the fuel of the future. The successful transformation from present fossil fuel to a hydrogen economy needs to solve many complicated scientific and technological problems [1]. Photocatalytic hydrogen production over semiconductors was suggested as one of the most promising methods, due to the simplicity of the concept and its technical feasibility [2]. So far, many photocatalysts have been examined in order to improve hydrogen production performance. Several semiconductors (e.g. TiO₂, Fe₂O₃, ZnS, CdS, WS₂ etc.) in photocatalytic hydrogen production were compared and discussed in recent review papers [3-6]. So far, titanium dioxide (TiO₂) was generally believed as the most promising semiconductor, because it is photostable, environmental-friendly and cheap [2, 6].

The main challenges in photocatalytic hydrogen production over TiO₂ include how to take advantage of solar light and reduce the recombination rate of electrons and holes. Modification of TiO₂ with co-catalysts (such as Pt, Pd, Ir and Au) has been developed to reduce the recombination of charge carriers [7-10]. The co-catalysts can play the role as electron collectors to effectively separate the photo-generated electron-hole pairs and the active sites for the proton reduction to

produce hydrogen. Among these metals, Pt showed the lowest overpotential and the best activity for hydrogen evolution [11,12]. In addition, the morphology, size and crystallinity of TiO₂ can also affect the photocatalytic performance. Previous study suggested that anatase TiO₂ showed higher activity than rutile and brookite. However, anatase TiO₂ crystals were usually dominated by thermodynamically stable (101) facets (c.a. 94%) and a minority of (001) facets [13, 14]. It is also reported that TiO₂ (001) facets were found to be the most reactive facets. The surface free energy of 0.90 Jm⁻² for (001) facets is much higher than (100) (0.53 Jm⁻²) and (101) (0.44 Jm⁻²) facets [15-18]. Therefore, it is important to take advantage of these reactive facets to further improve the photocatalytic performance.

Although many anion modified TiO₂ nanosheets with reactive facets have been synthesized and investigated, such as nitrogen-TiO₂ [19], carbon-TiO₂ [20] and sulfur-TiO₂, their corresponding photocatalytic water splitting performance has not yet systematically studied. Yu et al. recently reported that Pt-TiO₂ nanosheets with exposed (001) facets can enhance the performance of photocatalytic water splitting [21]. Modifying TiO₂ with more than one type anion is regarded as an effective strategy to improve photocatalytic performance. Xiang et al. reported that nitrogen and sulfur co-doped TiO₂ nanosheets with exposed facets showed much higher photocatalytic

activity than un-doped TiO₂ nanosheets [22]. Thus, the modification of TiO₂ nanosheets by anions is also important for hydrogen production application.

In this work, Pt decorated TiO₂ nanosheet with reactive facets was studied for photocatalytic water splitting. Two nonmetal elements, fluorine and boron, were used to modify the surface of TiO₂. The effect of Pt loading, TiO₂ size and photocatalyst loading on photocatalytic water splitting were systematically investigated.

2. EXPERIMENT

2.1 Preparation of TiO₂ nanosheet

A series of TiO₂ nanosheets were prepared by hydrothermal method. In brief, titanium butoxide (Ti(OBu)₄, 97%, Sigma Aldrich) as TiO₂ source and tetrafluoroboric acid (HBF₄, 48wt.%, Sigma Aldrich) as capping agent and dopant precursors were slowly mixed in a dry autoclave. The autoclave was sealed and treated at 160°C in an electric oven for 24 hours. After complete cooling, the sample was collected in a plastic tube and then centrifuged and washed with DI water and ethanol for three times. Finally, the as-prepared product was dried at 60°C overnight. In this study, Ti(OBu)₄ of 10 ml reacted with different HBF₄ solution 0.4 mL, 0.8 mL, 1.2 mL and 1.6 mL represented the sample abbreviated as TiO₂-NS-1, TiO₂-NS-2, TiO₂-NS-3 and TiO₂-NS-4, respectively.

2.2 Deposition of Pt on TiO₂ nanosheets

The deposition of Pt on TiO₂ nanosheets was realized by photo reduction method. In a typical synthesis, as-prepared TiO₂ material of 0.2 g was dispersed in 50 mL DI water (Millipore, Inc., 18 MΩcm⁻¹) and platinum(II) acetylacetonate (Pt(acac)₂, Sigma Aldrich, 99.99%) dissolved in ethanol (Sigma Aldrich, 99.9%) was added in the suspension under magnetic stirring. A 300-W xenon lamp (Aulight, CEL-HX UV300, 200 nm - 2000 nm) was used as the light source to initiate the photo reduction of platinum precursors. Under irradiation, ethanol in the suspension played the role as sacrificial reagent to enhance the charge separation. After irradiation for 30 min, the liquid sample was centrifuged and washed with DI water for three times. Finally, the photocatalyst sample was dried at 60°C overnight before use. The Pt loadings were varied from 0.5 wt.% to 4.0 wt.% by changing the amount of added Pt precursors. As-prepared TiO₂ samples of different sizes and morphologies (e.g. nanoparticles and nanosheets) were also used in this synthesis.

2.3 Material characterizations

The morphology of TiO₂-Pt samples were characterized by transmission electron microscope (TEM, FEI Tecnai G2 20 scanning). Optical properties of TiO₂ (TiO₂-NS-2) and TiO₂-Pt (TiO₂-NS-2-Pt) were recorded on diffusion reflectance UV-vis spectrometer (Hitachi, UV-3010).

2.4 Photocatalytic hydrogen production

Photocatalytic water splitting was performed with Pt-TiO₂ photocatalyst and UV-vis irradiation in a LabSolar H₂ production system (Trustech, Beijing) at room temperature. In a typical run, photocatalyst of 20 mg was ultrasonically suspended in 100 mL methanol (Sigma Aldrich, 99%) solution of 10 vol.% concentration. The photocatalyst suspension was bubbled with N₂ gas (HKO, 99%) for 1 hour to remove the dissolved oxygen. Then, the photocatalyst suspension was added and magnetically stirred in a photo reactor with a quartz cover. The top of the reactor was filled with DI water in order to filter the IR from the light source. In this study, a 300-W xenon lamp (Aulight, CEL-HX UV300) was used to investigate the photocatalytic hydrogen performance of the as-prepared photocatalysts. Before turning on the light source, the photocatalytic reaction system was evacuated by a vacuum pump to make the system a vacuum condition. An electromagnetic air pump was employed to accelerate the gas circulation in the system. The gas sample in the headspace of the photo-reactor was periodically analyzed by a gas chromatograph (GC-2010, Shimadzu) equipped with Msieve 5A column (Restek, 30 m × 0.53 mm × 50μm) and a thermal conductivity detector (TCD). The temperature of inlet, column and detector were 35°C, 35°C and 120°C, respectively. Helium gas was used as the carrier gas at a flow rate of 30 mL/min. Hydrogen gas was quantitatively analyzed by injecting pure H₂ gas in the reactor system. In this study, different Pt loadings, TiO₂ samples and photocatalyst loadings were investigated in the H₂ production system.

3. Results and Discussion

3.1 TEM analysis of Pt-TiO₂

Figure 1 shows the TEM analysis results of Pt-TiO₂ and Pt samples. It can be seen from Figure 1(a) that as-prepared TiO₂ was composed of nanosheets with 10 nm side length and 2.5 nm in thickness. In addition, some Pt nanoparticles of a few nanometers can be observed. This suggested that Pt-TiO₂ was successfully prepared by hydrothermal method followed by photo-deposition method. Figure 1 (b) shows the enlarged TEM image of Figure 1(a).

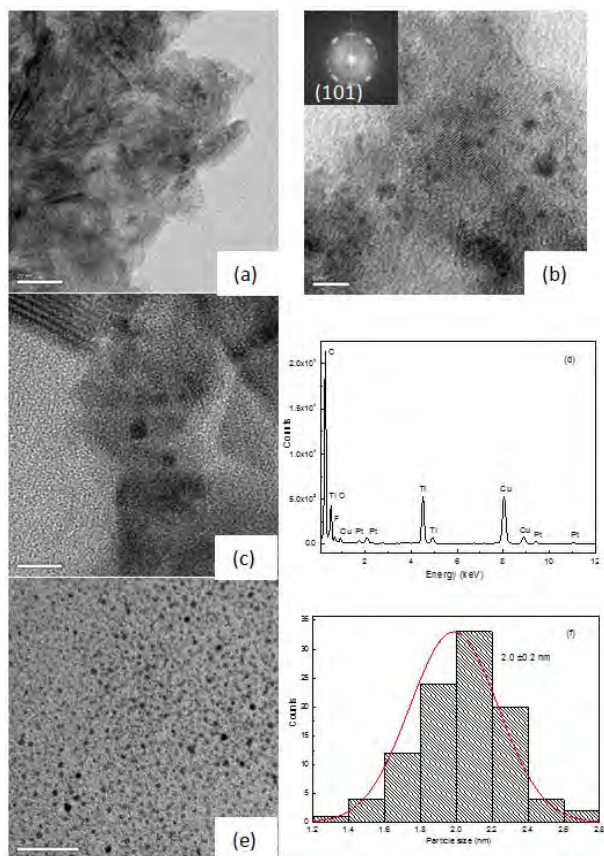


Figure 1. TEM analysis of TiO₂-Pt and Pt samples
(a): morphology of TiO₂-NS-2-Pt; (b) and (c): enlarged TEM image of (a), inset: fast fourier transform image; (d) typical EDX analysis of TiO₂-Pt sample; (e): morphology of Pt nanoparticles; (f) histogram analysis of Pt size distribution

The lattice structure can be clearly found in this image, which indicated the presence of TiO₂ nanosheets. The inset fast fourier transform image implied the existence of TiO₂ (101) facets. Also, some Pt nanoparticles can be found in this image. Figure 1(c) shows the typical morphology of TiO₂ nanosheets decorated with Pt nanoparticles. As seen in Figure 1(d), it is found that elements, such as C, Ti, O, F, Cu and Pt, can be detected by energy-dispersive X-ray (EDX) spectroscopy. Cu and C were supposed to come from the grid holder used in TEM analysis. Ti, O, F and Pt were from the as-prepared Pt-TiO₂ product. The fluorine dopant can be measured while boron was not detectable due to its low concentration. Figure 1(e) shows the typical morphology of Pt nanoparticle prepared by photo-reduction method. It can be seen that Pt nanoparticles were around a few nanometers on average. The small Pt size was ascribed to the low energy from UV light. The histogram analysis of Pt size distribution in Figure 1(f) suggested that the size of Pt nanoparticle was 2.0 ± 0.2 nm.

3.2 Hydrogen Production

3.2.1 Effect of Pt loading

Figure 2 presents photocatalytic hydrogen production over Pt-TiO₂ nanosheets synthesized with different Pt loadings and TiO₂ sizes. Figure 2(a) shows the effect of Pt loading on photocatalytic hydrogen production. TiO₂-NS-2 was chosen as the TiO₂ source due to its better photocatalytic activity in our previous study. It is obvious that TiO₂-NS-2 without Pt decoration shows very weak performance in hydrogen production. This is due to the fast electron-hole recombination and the absence of active sites for hydrogen evolution. When TiO₂-NS-2 was loaded with 0.5 wt.% Pt, photocatalytic hydrogen production performance was significantly increased. The result suggests that Pt sites are beneficial for the decrease of overpotential for hydrogen production. The highest hydrogen production of 329 μ mol within 4 hours was found for TiO₂-NS-2 loaded with 1.0 wt.% Pt. However, when the loading of Pt onto TiO₂ was further increased, the performance of photocatalytic hydrogen production was not further improved. TiO₂-NS-2 loaded with 2.0 wt.% showed the close performance with 1.0 wt.% Pt loading. Surprisingly, the performance of TiO₂-NS-2 modified with 4.0 wt.% Pt showed even worse activity than 0.5 wt.% Pt. This is possibly due to the fact that the higher Pt loading onto TiO₂ can block the light penetration into the photosensitive sites for photocatalytic reaction. In addition, overloading of Pt can increase Pt-TiO₂ contact area, which can bring more probability for the recombination of electron-hole pairs [25]. Therefore, overloading Pt can reduce the overall photocatalytic activity of TiO₂. According to literatures, the optimal Pt loading onto TiO₂ materials for water splitting was varied from 0.2 wt.% to 2.0 wt.% [21, 26-28], depending on the morphology, crystal phase, porosity and particle size of TiO₂. In this work, the optimal Pt loading was 1.0 wt.% for TiO₂ nanosheets.

3.2.2 Effect of TiO₂ size and morphology

The data presented in Figure 2(b) show the effect of TiO₂ size and morphology on photocatalytic hydrogen production over 20 mg Pt-TiO₂ photocatalysts with 1.0 wt.% Pt loading. The property details of Pt-TiO₂ used in hydrogen production can be found in Table 1. TiO₂-NP-1 was TiO₂ nanoparticles with 5 nm in size, according to TEM analysis. When it was loaded with Pt, the amount of generated H₂ was 172 μ mol during a four-hour measurement. The best performance of hydrogen production was obtained at TiO₂-NS-2, which was composed of TiO₂ nanosheets with dimension of 10 $\times$ 10 $\times$ 2.5 nm. With an increase in TiO₂ size from 10 nm to 30-45 nm, the performance of hydrogen production reduced significantly. This phenomenon suggests that the TiO₂ size can also affect photocatalytic hydrogen production performance. Small TiO₂ with nanosheet morphology is beneficial to the

enhancement of photocatalytic activity. On the basis of the above study, the best photocatalyst for hydrogen production was TiO₂-NS-2 loaded with 1.0 wt.% Pt.

3.2.3 Effect of photocatalyst loading

Figure 2(c) illustrates the effect of Pt-TiO₂ loadings on photocatalytic hydrogen production. TiO₂-NS-2-1.0Pt was selected as the model photocatalyst and the photocatalyst loadings were varied from 5 mg to 30 mg in 10 vol.% methanol solution of 100 mL. When the photocatalyst loading was less than 20 mg, the amount of generated hydrogen was increased from 102 μ mol to 329 μ mol. However, when the loading was further increased to 30 mg, the amount of hydrogen was decreased to 275 μ mol. Previous work suggested that the photocatalytic reaction was dominantly governed by the active sites which should have access to light and reactants [29]. When the concentration of photocatalyst was higher than the optimal value, the light was scattered or blocked by the excess photocatalysts, leading to the shield of the active sites.

3.2.4 Effect of Pt loading on CH₄ formation

Figure 3 demonstrates the effect of Pt loading on methane formation during photocatalytic hydrogen production over Pt-TiO₂. It can be seen from Figure 3 that with the increase of Pt loading, the formation of methane was inhibited. Previous work suggested that when methanol was used as sacrificial electron donor, the main gaseous products were CO₂ and H₂ [7]. The involved mechanism was illustrated as follows [35, 36]:

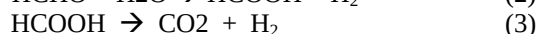
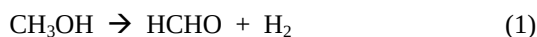


Table 1 Properties of Pt-TiO₂ and photocatalytic hydrogen production rate

Sample	TiO ₂ size (nm)	Pt loadin g (wt.%)	Photocat. loading (mg)	H ₂ prod. rate (μ mol/h/g)
TiO ₂ -NS-2	10×10×2.5	0	20	81
TiO ₂ -NS-2-0.5Pt	10×10×2.5	0.5	20	2877
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	20	4107
TiO ₂ -NS-2-2.0Pt	10×10×2.5	2.0	20	3884
TiO ₂ -NS-2-4.0Pt	10×10×2.5	4.0	20	2675
TiO ₂ -NP-1-1.0Pt	5	1.0	20	2145
TiO ₂ -NS-3-1.0Pt	30×30×6.0	1.0	20	2633
TiO ₂ -NS-4-1.0Pt	45×45×6.5	1.0	20	2499
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	5	5086
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	10	4694
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	30	2287

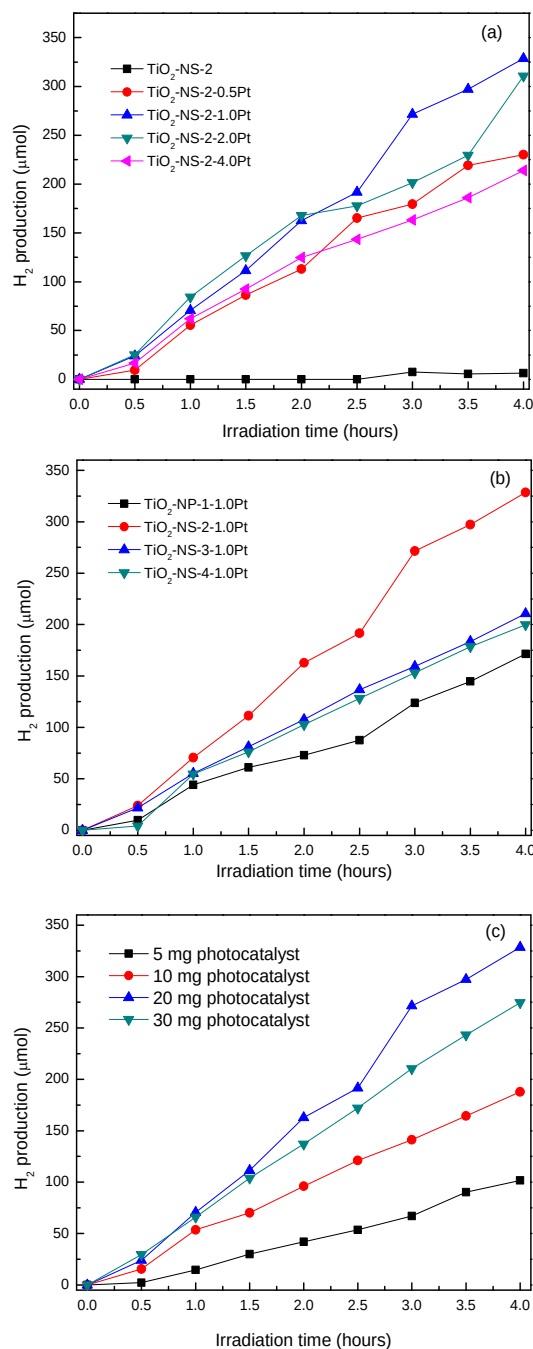


Figure 2: Photocatalytic hydrogen production over Pt-TiO₂ nanosheets (a) the effect of Pt loading; (b) the effect of TiO₂ size and morphology; (c) the effect of photocatalyst loadings.

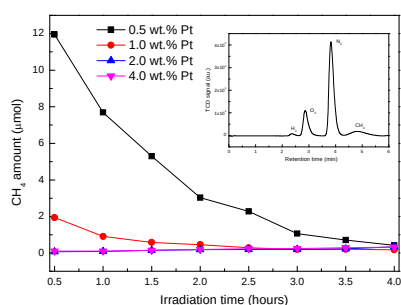


Figure 3: Effect of Pt loading on methane formation during photocatalytic hydrogen production over TiO₂-Pt

4. Conclusion

F,B-codoped TiO₂ nanosheets decorated with Pt nanoparticles have been successfully prepared by a simple one-pot hydrothermal method followed by photo-deposition method. The Pt size was around 2.0±0.2 nm. The highest hydrogen production rate of 5,086 μmol/h/g was obtained for TiO₂-NS-2 with 1.0 wt.% Pt loading. The effects of Pt loading, TiO₂ size and morphology and photocatalyst loading on photocatalytic hydrogen production were systemically investigated in this study. The results reveal that TiO₂ nanosheets possess better photocatalytic property than TiO₂ nanoparticles. Moreover, TiO₂ nanosheets of small size show higher photocatalytic hydrogen production performance than larger TiO₂. Small amount of CH₄ detected by gas chromatographic analysis suggests that CH₄ is one of the intermediates for photocatalytic hydrogen production.

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ESTABLISHMENT AND NUMERICAL SIMULATION OF COOL STORAGE AND HEAT EXCHANGE MODELS DEPENDING ON PHASE CHANGE MATERIALS

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ABSTRACT

The cubical cool storage units considered here are part of a plate and shell type heat exchanger of ternary and two-phase, accomplishing the purpose of refrigeration by means of peak and valley electric charges, then two different models have been proposed. In the first model unit, the phase change material is packed in row of tubes and the refrigerant flows parallel to it. In the second, pipe lines containing the refrigerant are embedded in the PCM. In the study, several phase change materials (CCHH, PARAFFIN, SSDH) known the thermo-physical parameters are selected, and a series of numerical simulation are undertaken in the process of cool storage by using a computational fluid dynamics (CFD), receiving the temperature distribution of the PCMs and the liquid fraction figures. In addition, the temperature distribution are researched in nature convection and without nature convection. Finally, the cool storage capacity of organic phase change materials and inorganic phase change material are compared.

Key words: phase change materials (PCMs), cooling energy storage, CFD, energy conservation

1. INTRODUCTION

With the development of the society, air conditioning power consumption increases gradually, and even holds the power load of 30% ~ 40% in particular areas. At the same time, air-conditioning peak often synchronize with industrial commercial peak, so peak power shortage is caused further. Power resources in the trough relatively surplus, which leads to the electricity peak valley load difference is too big, so that large amounts of energy losses and wastes. Using the cool storage air conditioning system, which also known as the Thermal Energy storage system (the TES system) is an important strategy to solve the peak valley power imbalance. namely the refrigeration system refrigerates and storage cold during the night time and the low load of air condition, and the system will release the cool storage to meet the needs of the air conditioning load during the day and the peak time of air conditioning load.

Phase change thermal storage has high density, small volume and high heat efficiency, which is widespread concerned in the development of phase change thermal storage air conditioning field. And the choice of phase change materials is the key in the system. Organic phase change materials become hot topic in the study of phase change heat transfer because of good formability, without of the phenomenon of too cold and phase separation, stable performance and non-toxic. But the coefficient of thermal conductivity of

organic materials is small, increasing the energy storage and release time, and reducing the efficiency of thermal control system. Using phase change materials for energy storage is one of the important means to improve the energy efficiency and environmental protection. Developing the technology of phase change thermal storage air conditioning has become an irreversible trend.

2. THE ESTABLISHMENT OF PHYSICAL MODEL

2.1 The establishment of physical model

As shown in figure 1 (a), energy storage device is rectangle, which is part of the ternary two-way phase change energy storage devices, and mainly is responsible for implementing unidirectional heat transfer between refrigerants and phase change materials. We choose the regenerative heat storage unit in the system as the research object, which the external surface of the heat storage unit is the internal interface, namely the symmetry of the circle in figure. The cross-sectional area to the heat storage unit is equal to the cross section area of the pipe. Therefore the paper puts forward two kinds of physical models, as shown in figure 1 (b), (c):

Model 1 (Fig. 1(b)): A vessel packed in the vertical direction with cylindrical tubes; the PCM is inside the tubes and the refrigerant parallel to it. For this model we assume that the refrigerant which is surrounding the cylindrical tubes is situated in theoretical cylinder

jackets all around the tubes.

Model 2(Fig. 1(c)): A vessel filled with the PCM in which cylindrical pipes for the refrigerant have been mounted. We shall embed all the PCM of the tank into cylindrical jackets surrounding the pipes, as we did with the refrigerant around the cylinders for mode 1.

In order to facilitate analysis, the establishment of the physical models is simplified assumed.

1 the temperature change range in the process of phase change is not too big, and materials' properties

can be considered as constant. The density variation is considered only in the effect of buoyancy;

2 the fluid in the liquid phase area is incompressible Newtonian fluid, two dimension, laminar, and the Boussinesq's hypothesis is developed.

3 the material of the tube is metal. The coefficient of thermal conductivity is very high. The thermal resistance is neglected;

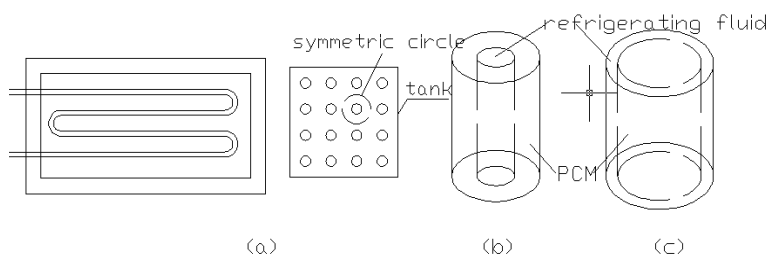


Figure 1 heat exchange and storage device and two physical models

4 heat storage tank does not leak heat loss, namely the top and bottom surface of heat storage tank is adiabatic.

5 the mass flow rate is usually very big for tube heat transfer liquid fluid, which can be thought that the import and export temperature are the same, and not having heat transfer along the length direction (namely constant wall temperature) in the small ratio of the length and diameter.

2.2 The selection of materials

This article selects three representative of phase change materials, including organic phase change materials(PARAFFIN) and inorganic phase change materials (calcium chloride, sodium sulfate,).the phase transition temperature are between 25 °~ 40 °, the physical parameters are listed in table 1.

Table 1 Physical property parameters f of phase change materials for numerical simulation

properties	CCHH	PARAFFIN	SSDH
Density (kgm ⁻³)	1710	830	1460
Specific heat (Jkg ⁻¹ k ⁻¹)	2130	3260	2000
Thermal conductivity(Wm ⁻¹ K ⁻¹)	0.539	0.224	0.3
Melting temperature(k)	303	305	312
Latent heat of fusion(Jkg ⁻¹)	187490	251000	180000

3.THE ESTABLISHMENT OF THE CONTROL EQUATIONTO THE MODELS

According of the above assumptions, using Gambit processor to model and mesh grid, and based on enthalpy and variable using the solidification/melting model in the FLUENT software, we carry on the numerical simulation. The basic equations are as follows:

The Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} = 0 \quad (1)$$

The momentum equation:

$$\frac{\partial U}{\partial t} + (U \cdot \text{grad})U = \text{div} \left(\nu \cdot \text{grad}U \right) - \frac{1}{\rho} \text{grad}P + S_U \quad (2)$$

The energy equation:

solid phase:

$$\rho_s c_s = \frac{\partial T_s}{\partial t} = \text{div}(k_s \cdot \text{grad}T_s) + S_{T,s} \quad (3)$$

liquid phase:

$$\rho_L c_l \left(\frac{\partial T_s}{\partial t} + (U \cdot \text{grad})T_l \right) = \text{div}(k_l \cdot \text{grad}T_l) + S_{T,l} \quad (4)$$

Solid-liquid interface:

$$\rho_s \lambda U_\phi = \left(k \frac{\partial T}{\partial n} \right)_s - \left(k \frac{\partial T}{\partial n} \right)_l \quad (5)$$

$$\text{Mass conservation equation: } \text{div}(U) = 0 \quad (6)$$

U is the velocity vector of phase change materials; ν is the kinematic viscosity; P is the pressure; C is the specific heat; ρ is density; K is the thermal conductivity; S_U and S_T respectively are the source term of momentum and energy conservation equation; λ is phase change latent heat.

In the Momentum equation, the density related to gravity is expressed as:

$$\rho = \rho_{ref} [1 - \beta(T - T_{ref})] \quad (7)$$

The boundary conditions:

Due to the four adiabatic surfaces of the storage unit , so the boundary conditions for:

$$\begin{aligned} \frac{\partial T}{\partial y} \Big|_{y=0} &= 0 ; \frac{\partial T}{\partial y} \Big|_{y=l} = 0 \\ \frac{\partial T}{\partial x} \Big|_{x=0} &= 0 ; \frac{\partial T}{\partial x} \Big|_{x=\frac{R}{2}} = 0 \end{aligned} \quad (8)$$

l is the length of the heat storage unit; R is the radius of heat storage unit.

The heat exchange surface wall between Refrigerants and phase change materials (inside wall surface) is coupled. there are three kinds of forms of coupling boundary:

(1)the temperature continue:

$$T_w \Big|_f = T_w \Big|_p \quad (9)$$

(2)heat flux continue: $q_w \Big|_f = q_w \Big|_p$ (10)

(3)the third boundary condition in the coupling surface:

$$-\lambda \frac{\partial T}{\partial x} \Big|_{x=0} = k(T_w - T_f) \quad (11)$$

f represents the refrigerant area; p represents phase change materials area.

The initial conditions:

For initial temperature, the refrigerant and the phase change materials are equal, namely:

$$T_f \Big|_{(x,y,t)|t=0} = T_p \Big|_{(x,y,t)|t=0} = T_0 \quad (12)$$

T_0 - initial temperature of the cold storage unit, to make use of storage capacity of the sensible heat of phase change materials, the initial temperature is below the phase change temperature in general. Due to the assumption that in the process of cool storage the surface of the cooling temperature remains the same, so at the beginning of the solidification the temperature at cooling surface keeps constant, namely:

$$T(x, y, t) \Big|_{x=0, t>0} = T_f \quad (13)$$

4 . THE RESULTS AND ANALYSIS OF NUMERICAL SIMULATION

As shown in Figure 2, choose the inorganic salt phase change material CCHH, chilled wall temperature is 283 k, the initial temperature of phase change materials is 318K.In the two kinds of modes, the qualities of the phase change materials and the refrigerant respectively are equal in according to the assumption. By the liquid phase fraction in figure we can see, in the 560 s, the solidification process of basic model 1 accomplished but yet model 2 has the rest of the liquid.it shows that under the same environmental conditions, the coagulation time of model 1 is shorter than the coagulation time of the model 2. Because with the progress of solidification, solid PCM is thicker, increasing the heat transfer resistance, slowing down the solidification rate, prolonging the condensation time. In this paper, we adopt the first model in the following research.

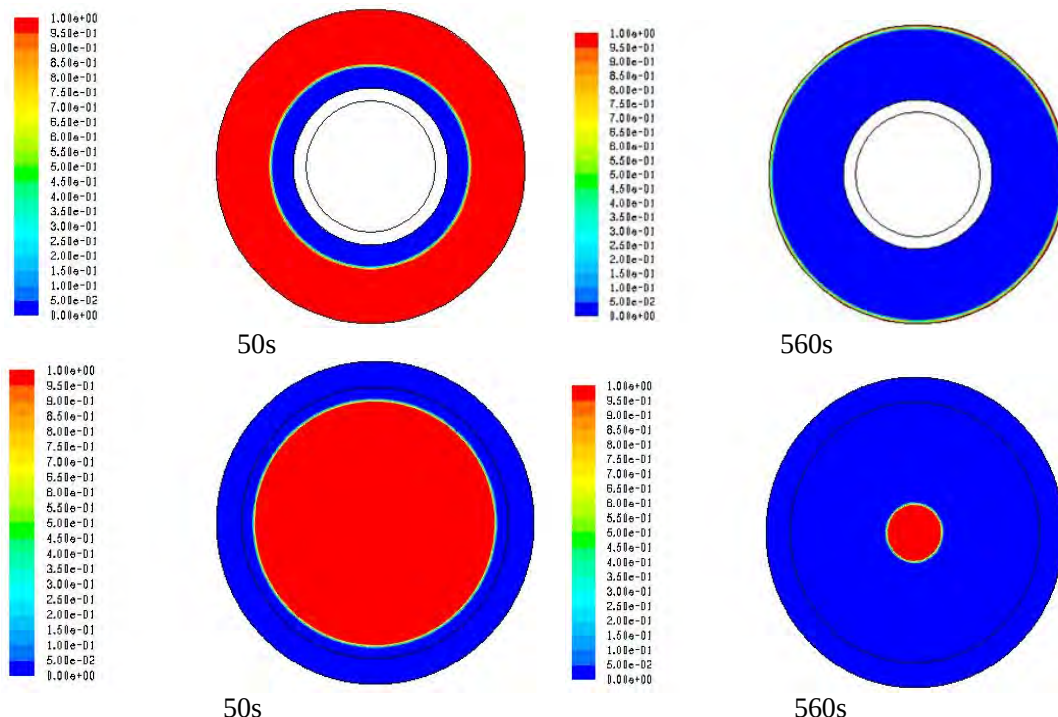


Figure 2 The liquid fraction contrast between two models respectively in the 50 s and 560 s

In previous research the density variation of phase change materials is almost not considered, and the influence of natural convection is usually also ignored. But in the actual phase change process, natural convection caused by temperature difference has always existed, and with the proportion of the liquid phase increasing gradually, the influence of natural convection also gradually increases, complicating the solid-liquid interface movement. In order to

demonstrate the effect of natural convection in the melt process. In this paper, the numerical simulation analysis was implemented about the natural convection effect on phase transition process. As shown in figure 3, the liquid phase fraction and temperature distribution of CCHH are showed respectively in the case of natural convection and without natural convection.

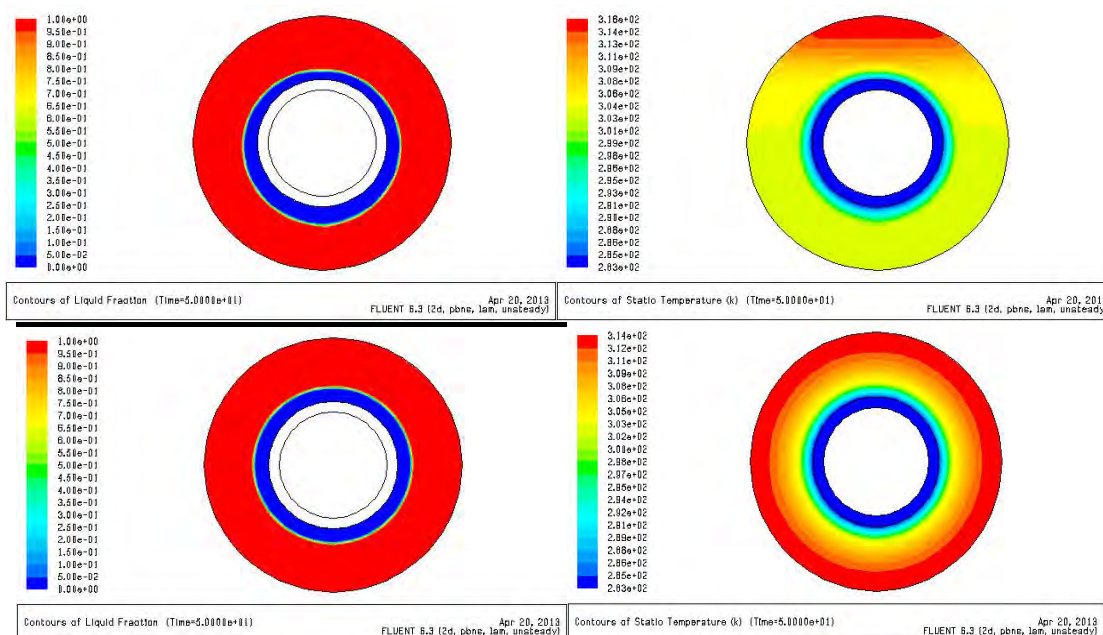


Figure 3 50 s liquid phase fraction and temperature distribution in the natural convection and without natural convection

As shown in figure 3: in the natural convection situation, solidification rate below the circular tube is faster, and basically the solid-liquid interface is round face, the shape changed small over time. By the temperature distribution we can see the obvious temperature gradient along the gravity direction. Figure 4 shows the temperature variation at (0,0.008) and (0,-0.008), the temperature is located in the negative direction of y axis have a wide range of fluctuation in the early due to the forced circulation caused by natural convection. and with the solidification, the effect of natural convection in solidification process is more and more smaller, the melting rate of phase change materials areas around the circular tube close to the

same, and solid liquid interface is almost circular surface (figure 5). On the one hand, due to in the solidification process liquid has always exists, and natural convection gradually weakened with the liquid quantity decreasing; on the other hand, the circulation flow caused by natural convection uniformed the temperature in whole transformation area, and the temperature gradient in phase change zone decreases continuously, so as to make the natural convection weak continuously. Natural convection makes the temperature of the liquid region homogenization, and slow the reduce the temperature of the cooling surface area, thus slow solidification speed, so natural convection has inhibitory effect on solidification.

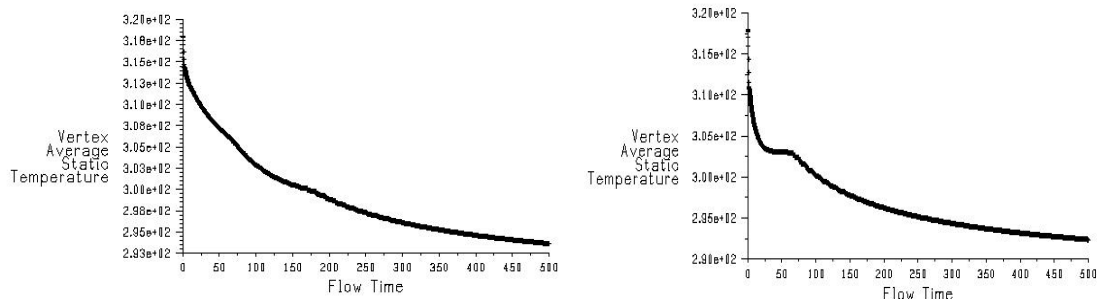


Figure 4 the contrast figure of temperature variations in the positive and negative of y axis

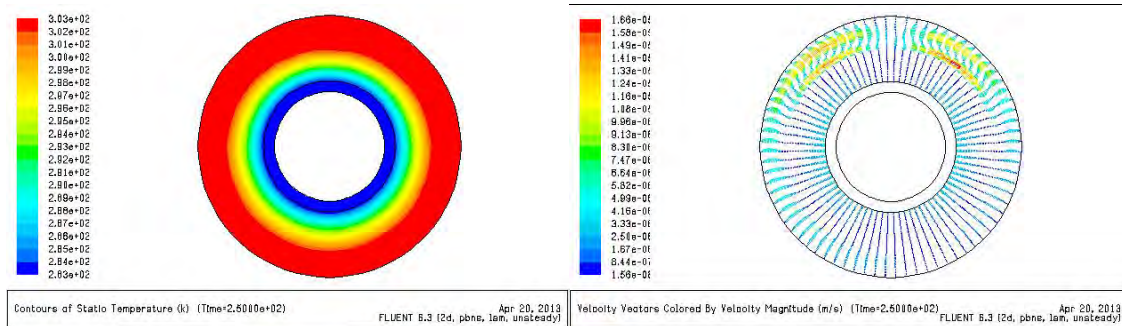


Figure 5 250 s the solidification temperature and velocity vector distribution about CCHH

In Figure 6, we choose three types of phase change materials (CCHH, SSDH, PARAFFIN), the phase change temperature of organic phase change material PARAFFIN is between inorganic salt phase change materials (CCHH and SSDH). in the temperature of the cooling wall is 283 K, and the initial temperature of phase change materials is 318 K. at 500 s, the liquid

phase fraction of PARAFFIN is less than CCHH's and SSDH's. It indicates the thermal conductivity of organic phase change materials is small, compared with inorganic salt phase change material, and also indicates it increases energy storage and release time during the process of phase change, reducing the efficiency of thermal control system.

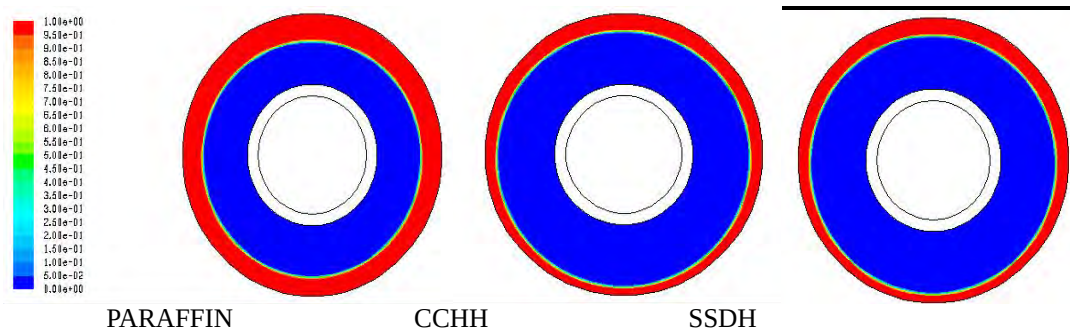


Figure 6 the distribution of liquid phase fraction of three phase change materials at 500 s

5. CONCLUSION

Through researching phase change heat transfer characteristics, understanding the effect of phase change materials thermal properties and boundary conditions on the phase change heat transfer process, and controlling the movement of the solid-liquid interface in phase change heat transfer process so as to determine the number of phase change materials and the phase change process parameters such as the time required, which can play an important role about the effective use of phase change materials and the optimization design. For the early solidification process of phase change materials, the process is controlled by natural convection and heat conduction, with the solidification over time, natural convection is abate, and solidification is gradually controlled by the thermal conductivity. Natural convection makes the temperature of the liquid region homogenize, slowing the reduce of the temperature of the cooling surface area, thus slowing solidification speed, so natural convection has inhibitory effect on solidification. At present in terms of phase change energy storage materials, the main research is based on organic materials for the engineering application of energy storage medium. And whether to develop cheaper and safer (no fire hazards), more excellent performance inorganic phase change energy storage materials, which will be the key to the phase change energy storage technology to practical application.

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ANALYSIS OF TECHNOLOGY INTEGRATED ON SOLAR ENERGY COMBINE WITH RECLAIMED WATER SOURCE HEAT PUMP AND PHASE CHANGE ENERGY STORAGE

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ABSTRACT

Energy shortage and environmental pollution problems are getting worse in today's society. The temperature of the reclaimed water which is derived from urban sewage changed little all the year-round. The quantity of the reclaimed water is stable. the temperature characteristics of it is warm in winter and cool in summer. it also be recognized as the recyclable and available clean energy. The compound energy utilization system of solar energy and reclaimed water source heat pump is comprised of the technology of reclaimed water source heat pump, Solar collector technology and the technology of Phase change energy storage. It combines the advantages of the three; the system is new environment friendly technologies which are both energy conservation and environmental protection but also play the efficacy of the load shifting electricity consumption. This article describes the meaning, operating principle and economic indicators of the system. it can be seen that the operating costs of the system is lowest and the pollutants released into the atmosphere is least by compared with other heating mode like boilers, solar energy, water source heat pump, air source heat pump. the system is of great significance to energy conservation and improve economic efficiency.

Keywords: Solar energy, Reclaimed water source heat pump, Phase change energy storage, Energy conservation, Economic benefits.

1. INTRODUCTION

With the accelerated development of urban construction, the heating energy of many cities is lacking, the problems of traditional heating method such as coal-fired boiler and oil-fired boiler, electric boiler are that there are high resource consumption, high operation cost and environmental pollution problems. The development of new energy, the use of alternative energy sources and the development and utilization of space resources will become the inevitable trend under the push of social progress and sustainable development. In this situation, solar thermal technology and heat pump technology have also entered into a rapid development stage.

Solar thermal technology is now the most successful use of field devices, but the solar system is influenced by weather and climate conditions, it can't meet the demand of user's quantity of heat all the year round.

The reclaimed water source heat pump is an air conditioning which uses the reclaimed water of sewage treatment plant for cooling and heating cycles. The reclaimed water is the secondary or tertiary effluent of sewage treatment plant, it has many advantages, the quantity of it is sufficient, the temperature of it is higher than the surface water and the quality of reclaimed water is better. It can be used as the ideal heat source of heat pump, compared with other low grade heat sources of water heat pump as water, sea water, groundwater,

etc. The reclaimed water source heat pump has the following characteristics:

(1) The quality of reclaimed water is better that it can be directly into the heat pump for heating cycle, the efficiency of the heat pump has been greatly improved.

(2) The temperature of the reclaimed which is originated in the sewage is appropriate and stable, it is warm in winter and cool in summer and changed little with the season, the temperature of it can be 10°C at winter in cold regions, it is the ideal, abundant heat source of heat pump, the temperature of it is about 20 °C in the summer, so it is the ideal cooling water of air conditioning systems

(3) The reclaimed water source heat pump has reduced the automatic filter equipment, the heat exchanger materials can choose copper which is commonly used on the HVAC air conditioning heat exchanger, the cost of the equipment is reduced greatly.

The compound energy utilization system of solar energy and reclaimed water source heat pump is combined with solar system and water source heat pump system, it not only has the characteristics of high efficiency, no pollution and high reliability, it can also make full use of renewable energy solar power through controlling under the condition of meeting the demand of heat users. The cost of the system operation can be further reduced

2. THE SYSTEM'S STRUCTURE AND THE CHARACTERISTICS OF OPERATION

Figure 2.1 is the structure of the compound energy utilization system of solar energy and reclaimed water source heat pump. It is comprised of solar collectors, heat circulation pump, cryogenic tank, heat pump units, the user circulating pump, electric auxiliary heater, high temperature phase change energy storage water tank, fan coil, controller, etc. The solar hot water system series on the heat removal section of the heat pump units through the low temperature water tank, the unit open at the time of valley electricity price at night, the evaporation obtain the heat from low-temperature water tank. The heat stored in low temperature water tank is generated by the solar hot water system in the daytime, the heat pump get the low grade heat energy from the low-temperature water tank at night, through the heating cycle of the heat pump, it can releases high grade heat. Part of the heat is stored in high temperature water tank, another part of the heat will be provided directly to fan coil heating for heating at night. Put some Phase change materials into the high temperature water tank in order to improve the heat storage and thermal latency ability of the water in the water tank, the heat stored in the reclaimed water and the phase change materials of the high temperature water tank through the heat pump at night will be provided to the fan coil units for the indoor heating at the time of peak electricity price.

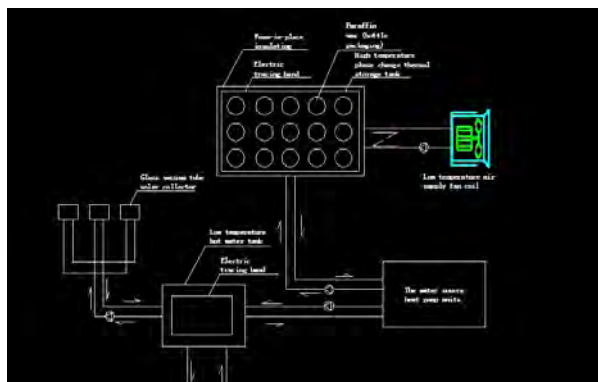


Figure 2.1 Structure diagram of the compound energy utilization system of solar energy and reclaimed water source heat pump

This system not only makes the solar system run efficiently during the sunny day, but also makes the water source heat pump run in the high efficient continuously at night, the system makes full use of the peak valley electricity policy, it has considerable economic and energy saving.

3. THE ECONOMIC ANALYSIS OF THE COMPOUND ENERGY UTILIZATION SYSTEM OF SOLAR ENERGY AND

RECLAIMED WATER SOURCE HEAT PUMP

The system test is located in Shenyang area which is between east longitude 122 ° 25 '123 ° 48' and north latitude 41 °11 ' to 43 °2' in the middle of liaohs plain of liaoning province. Shenyang region is temperate monsoon continental climate, it is cold and dry in winter but hot and humid in summer, the temperature changes rapidly in spring and autumn season, it is windy in spring and sunny in autumn, Annual temperature range in - 29 °C to 36 °C, the average annual temperature is 8 °C, the highest temperature is 35.2 °C, the extreme minimum is 30.6 °C. Years of average rainfall is 628 mm which is increasing from north to south. Uneven distribution of rainfall during the year, the summer is influenced by the southeast monsoon, so the rainfall is concentrate. The heavy rain occurs usually in July and August, which is about 47% of the total precipitation throughout the year. The average evaporation is 1444.9 mm for many years, years of average relative humidity is 63%. The average wind speed of the year is 3.1, annual average sunshine hours is 2372.5 hours, so it is belong to the areas where the sun can be used. As it is shown in figure 3.1

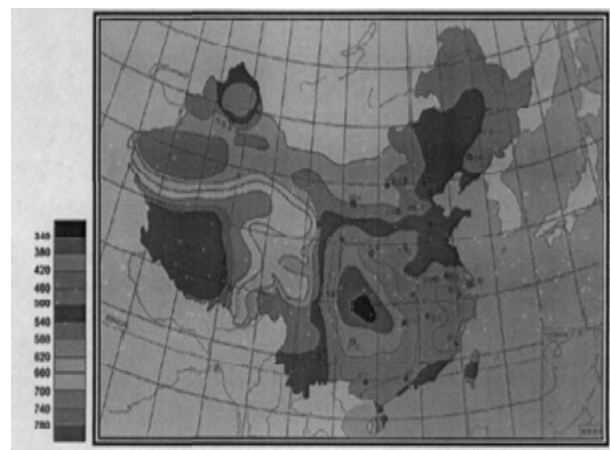


Figure 3.1 Solar energy resources distribution

3.1. The operating cost of the compound energy utilization system of solar energy and reclaimed water source heat pump

The system test is to use the compound energy utilization system of solar energy and reclaimed water source heat pump in combination with fan coil to heat for a plant of the company in Shenyang, the indoor temperature of the plant is requested at least 15 °C, the room size is 36 cubic meters, the heat load of heating in winter is about 1800 w.

3.1.1. Valley electricity consumption

The power consumption of the reclaimed water source heat pump is 15 kw h/day, the circulating pump and fan power is 2 kw, h/day.

The power of the electrical heating is 8 kw. Because there is change in the weather factor and unforeseen indoor heat load change, so the running time

of electric heater is not fixed, the thermostat will automatic control electric heater according to the corresponding temperature sensor sensing temperature change and start-stop.

3.1.2. Peak electricity consumption

The power consumption of the circulating water pump and fan power consumption is 1.8 kw, h/day

The heating period of Shenyang is from November 1 to March 31 of next year, about 150 days, the peak electricity price is 1.1898 yuan/kw.h; Normal Electricity price is 0.7932 yuan/kw. h; Valley electricity price is 0.3966 yuan/kw. h.

The year's cost of the system: Valley of electricity running cost + peak electricity running cost= (power consumption of water source heat pump+ power

consumption of the circulating water pump and fan + power consumption of electrical heating)×Number of days per year × Valley of electricity+ power consumption of the circulating water pump and fan× Number of days per year × Peak electricity= (15+2+8) *150*0.3966+1.8*150*1.1898=1799.496 yuan

Because the water source heat pump units and electric heaters are automatic controlled according to the corresponding temperature sensors measured temperature, rather than individual through time automatic stop device, so the operation time of the two devices is not stable, so there will be some volatility on the power consumption of the valley of electricity and the peak electricity.

3.2. The comparison of some heating ways

Table1 temperature and sunlight conditions of Shenyang

Month	1	2	3	4	5	6	7	8	9	10	11	12
The average temperature(℃)	-10	-7	1	10	17	22	25	24	18	10	1	-7
The average daily sunshine(hour)	5.4	6.5	7.5	8	8.6	8.2	6.9	7.2	7.8	7	5.5	4.9

Table2 the comparison on energy consumption of some heating means

The heating mode	Coal-fired boiler	Fuel oil boiler	Gas boiler	The electric boiler	The solar energy	Air source heat pump	Water source heat pump	The compound energy utilization system
Fuel type	coal	oil	natural gas	electricity	electricity	electricity	electricity	electricity
Pollution of the environment	very serious	exist	nothing serious	none	none	none	none	none
The risk	exist	more dangerous	very dangerous	more dangerous	none	none	none	none
Combustion value (kcal/kg or kcal/kw·h)	4300	10200	9000	860	860	860	860	875
The average thermal efficiency	64%	85%	75%	95%	300%	400%	500%	650%
Fuel price (yuan/kg or yuan / kw·h)	0.45	4.2	2.2	0.79	0.79	0.79	0.79	0.79
Daily need fuel (kg/day or kw·h /day)	55.39	17.58	22.58	186.57	59.08	44.31	35.45	26.80
Fuel costs (yuan/day)	24.93	73.84	49.68	147.39	46.67	35.01	28.01	12.00
fuel costs of a year (yuan)	3739.5	11076	7452	22108.5	7000.5	5251.5	4201.5	1799.496
Artificial cost	exist	exist	exist	none	none	none	none	none
duration of the equipment	5-8 years	5-8 years	5-8 years	5-8 years	12-15years	12-15years	12-15years	12-15years

4. THE APPLICATION PROSPECT OF THE COMPOUND ENERGY UTILIZATION SYSTEM OF SOLAR ENERGY AND RECLAIMED WATER SOURCE HEAT PUMP

It is can be seen from the comparison of various heating means, the daily operation cost of the compound energy utilization system is the minimum. It is 48% of coal-fired boilers, 16% of the oil burning

boiler, 24% of gas-fired boile, 8.1% of the electric boiler, 25.7% of the electric auxiliary solar energy, 34.3% of the air source heat pump, 42.8% of the water source heat pump.

This system can reduce 802 kg CO₂, SO₂ and other harmful substances which is discharged by the standard coal that is burning of low ignition point. So it is one of the effective ways to solve the environment pollution which is caused by the traditional heat source of fossil fuel combustion process, the system can reduce CO₂ emissions by 30% on average compared with coal-

fired boiler; Compared with the oil burning boiler, CO₂ emissions can be reduced by 68%, it can also reduce emissions of heat at the same time. It has a positive effect on reducing the urban environmental pollution and the greenhouse, and it has no electric potential danger during operation, the system is running without artificial cost.

At present, our country is generally in the international advanced level in the basic theory of solar energy research, the process materials, the application, the technical standard, the manufacture level, and the product quality, etc. The research of the water source heat pump in our country is still in its early stages, but the similar heat pump industry in our country is more mature. So the development of the system will be rapid under the impetus of the market.

The costs of the system will be down under the development of the solar energy and the reclaimed water source heat pump. The system has wide popularization and application value in the regions which is rich in solar energy as long as to perfect the relevant technology supporting system.

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THERMAL EXCHANGE CHARACTERISTICS OF DOUBLE DIRECTIONAL PHASE CHANGE ENERGY STORAGE DEVICE

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ABSTRACT

Air source heat pump use outdoor air as heat source and this kind of heat pump operation mode itself has the disadvantage of mismatching in the energy supply and demand as well as operation efficiency and operating time; Phase change energy storage technology is a kind of green and sustainable energy utilization which can effectively optimize the time allocation of energy and solve the mismatching within energy supply and demand in both time and space. Finally, energy utilization efficiency is improved. Air source heat pump operating time optimization is researched aiming at improving air source heat pump energy conversation and solving the inherent contradiction within the operating schedule and efficiency. Two-way phase change energy storage device which have both phase change cold storage and phase change thermal storage function is developed and the heat change performance is tested in experimental way by simulating the heat transfer mechanism of air source heat pump and phase change energy storage system.

Key words: double directional phase change energy storage device, air source heat pump, operating time optimization, experimental research

1. INTRODUCTION

Resources and environment problems is getting even worse nowadays, how to achieve energy allocation and effective energy usage while ensuring comfort and health at the same time has become the very problem we must deal with. Phase change energy storage technology is a kind of green and sustainable energy utilization which can effectively optimize the time allocation of energy and solve the mismatching within energy supply and demand in both time and space. According to related research of IEA (International Energy Agency), air source heat pump has taken 27% of the whole quantity of heat pump and over 70% heat pump in our country is air source heat pump^[1]. In order to make air source heat pump a more energy conservation way of energy use, change energy storage technology is combined with air source heat pump. Recently, the research mostly concentrates on ice storage; PCCSM and PCHSM used for defrost in winter^[2-5]. To combine phase change energy storage Technology with air source heat pump, a new way of phase change energy storage named two-way phase change energy storage is proposed. A new kind of phase change energy storage device is created and the heat transfer property is tested experimentally.

2.TWO-WAY PHASE CHANGE ENERGY STORAGE THEORY

2.1 Two-way phase change energy storage theory

Two-way phase change energy storage contains two kind of PCMs, named PCCSM and PCHSM, which can both achieve phase change cool storage and phase change heat storage. The principle of the system is shown as figure 1.

For the cool storage process, low temperature cooling medium is forced to through the heat exchanger from ① to ② by way of PCCSM and PCHSM, in this way heat exchange happens. PCCSM

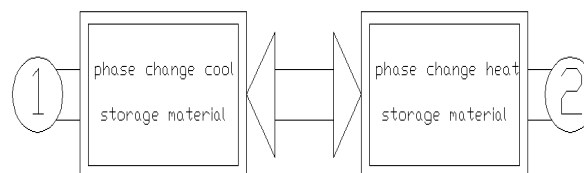


Figure1 the two-way phase change energy

NOMENCLATURE

Abbreviation		Symbols	
PCESD	Phase change energy storage device	U	Two kinds of PCM is contained in the device and certain function is achieved which is the first device domestic
PCCSM	Phase change cool storage material		
PCHSM	Phase change heat storage material		
PCM	Phase change material		

Table 1. Properties of Pt-TiO₂ and photocatalytic hydrogen production rate

Sample	TiO ₂ size (nm)	Pt loading g (wt.%)	Photocat. loading (mg)	H ₂ prod. rate (μmol/h/g)
TiO ₂ -NS-2	10×10×2.5	0	20	81
TiO ₂ -NS-2-0.5Pt	10×10×2.5	0.5	20	2877
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	20	4107
TiO ₂ -NS-2-2.0Pt	10×10×2.5	2.0	20	3884
TiO ₂ -NS-2-4.0Pt	10×10×2.5	4.0	20	2675
TiO ₂ -NP-1-1.0Pt	5	1.0	20	2145
TiO ₂ -NS-3-1.0Pt	30×30×6.0	1.0	20	2633
TiO ₂ -NS-4-1.0Pt	45×45×6.5	1.0	20	2499
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	5	5086
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	10	4694
TiO ₂ -NS-2-1.0Pt	10×10×2.5	1.0	30	2287

2.2 Storage theory

Achieve energy storage with latent heat while PCHSM stores energy with sensible heat.

For the cool release process, high temperature cool medium is forced to through the heat exchanger from② to ① by way of PCHSM and PCCSM, in this way heat exchange happens. PCCSM achieves energy release with latent heat while PCHSM releases energy with sensible heat.

In the same way heat storage and heat release process is carried on. With combing of the two kinds of PCM, the two-way phase change energy storage achieves both phase change cool storage and phase change heat storage. With this kind of function, phase change energy storage technology and air source heat pump technology are possibly to operating together.

2.3 The experimental two-way phase change energy storage system

Two-way energy storage system is comprised of phase change energy storage device, tub, and temperature control unit and data collection device.

2.3.1 Two-way phase change energy storage device

The phase change energy storage device is composed of several PCM units which is comprised of metal framework and PCM inside. Two points in the material unit is chosen to measure the very temperature in every unit, means the surface temperature and the central temperature Shown in figure 2. The energy storage device contains five phase change cool storage unit and five phase change heat energy storage units, and other three points chosen is in the heat exchanger medium layer, the position is shown in figure 3.

2.3.2 Temperature control unit

The temperature control unit is composed of temperature controller, thermal couple and a/c contactor, as is shown in figure 4. The temperature of the inlet heat transfer medium is monitored and controlled by temperature control unit.

2.3.3 Data collection device

Paperless recorder is adopted in the experiment in order to achieve experiment data automatic recording. Supported by the software, data read and curve output can be realized.

Finally, after combing of every part with the tube, the whole system is shown as figure 5.

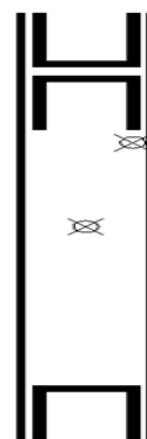


Figure2 PCM unit and measure point

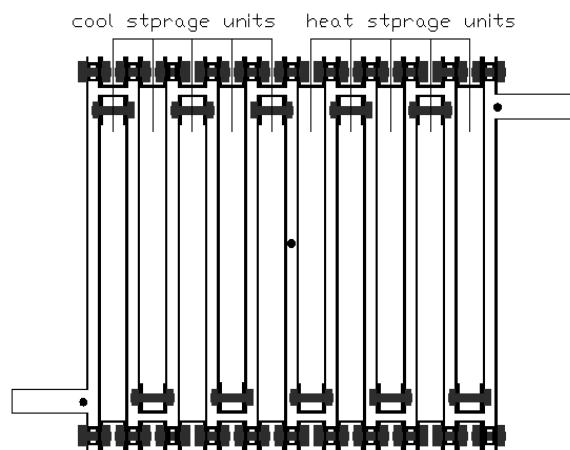


Figure 3 phase change energy storage device and measure point



Figure 4 the temperature control unit



Figure 5 The experiment system

The phase change energy storage device contains two kind of PCM. In this paper, 54# partly-sourced paraffin is adopted as the PCHSM and the Eutectic of dodecanol and decanoic acid is adopted as the PCCSM. The phase change temperature of the two materials is shown in table 1.

Table1 phase change temperature of the material

material	Phase change temperature
Paraffin	47.33℃
Eutectic content	9.05℃

2.3.4 Energy storage device heating test

In order to get heat exchange performance of the PCCSM and PCHSM in the device, experiment process firstly cooled the device with cool water which is introduced from the tank. In the end, the temperature of the PCM is 6.2℃.

The device heating experiment is carried on after the whole cooling process. The heating process is achieved with the temperature control unit and heating rod. The temperature control unit can achieve interval control of inlet water temperature. When the inlet water temperature get below the minimum temperature, heating is active by means of heating rod and heating is ceased when temperature reached the maximum temperature. Each order is sent by the temperature controller and each action is done by a/c contactor. The Whole process has mainly two phases, PCCSM melting process and PCHSM melting process. The maximum inlet water temperature is 15℃, while the minimum temperature is 10℃ in the PCCSM melting process. In the PCHSM melting process, the maximum inlet water temperature is 53℃, while the minimum temperature is 48℃.

3. RESULT AND ANALYSIS

3.1 Heating test

The heating test result as follows. Figure 6 is the eutectic content melting temperature-time curve and Figure 7 is the paraffin melting temperature-time curve. Figure 8 is the tested water temperature of the inlet, the center and the outlet water temperature. The three curves in figure 8 have the same trend.

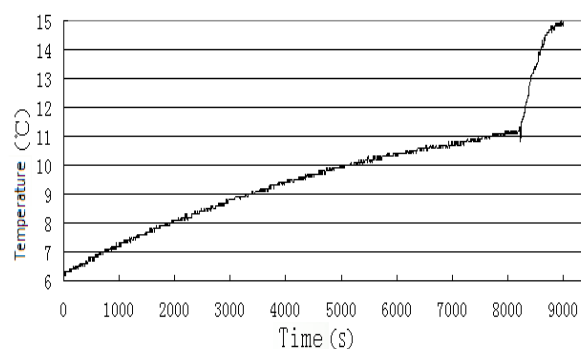


Figure 6 Eutectic content melt curve

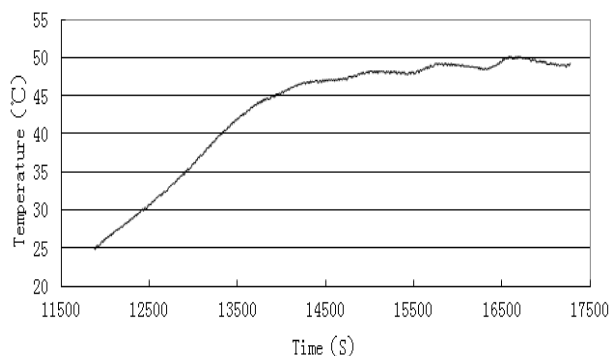


Figure 7 Paraffin melting curve

Due to the repeatedly start and stop of the air source heat pump outdoor unit when air source heat pump is running actually, the actual temperature of supply water fluctuate from 45 to 53°C. Based on the fact, the maximum temperature is set as 53°C while Hy as 5°C in order to control the supply water temperature fluctuate from 48~53°C in the simulation experiment.

The curve in figure 7 demonstrates temperature fluctuate after 14130 seconds is just the result of the very setting. The temperature range remains about 2°C all the way. At time 24018s, the temperature of the three testing points approach which means that the phase change process as well as the heat exchange process ended.

During the PCCSM melting process, the water heating adopt equipower heating method with a heating power of 3 kW. The supply water temperature ranges from 11.5 to 11.8°C, the output water temperature increases notable at time 8226 second which means that at the heating power 3kW, the two-way PCCSM totally achieved phase change within 2 hours and a temperature difference of 2.5~2.8°C between phase change cool storage, PCHSM absorbed the energy of 193.12kJ, which taken the percentage of 7.89% of the phase change cool energy storage by means of latent heat.

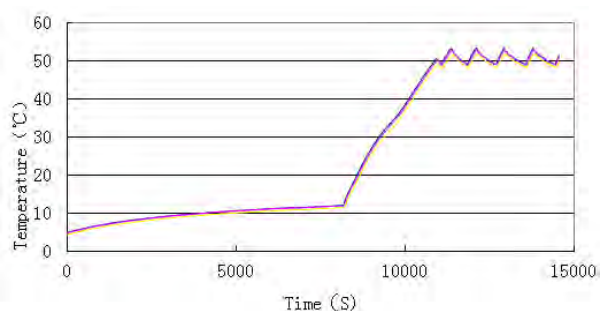


Figure 8 The water temperature changes during the process of heating

Water temperature of the three testing point variation trend is shown in figure 8 and the temperature changes rather slow in the phase change temperature of both kinds of PCMs.

The temperature change and corresponding time taken during the eutectic temperature melting progress is shown in table 2. According to table 2, during the eutectic content melting process, time taken between 9~11°C take up the most part and material get a rapidly temperature increase when the material temperature increase up to 11°C, which means that the phase change process of eutectic content of dodecanol and decanoic acid mostly happens in 9~11°C and the process ends in the temperature of 11°C.

Table 2 Eutectic temperature change and time consuming statistics

Tart temperatur e(°C)	Ending temperatur e(°C)	Start time(S)	Ending time(S)	Whole time(S)
8	9	1908	3270	1362
9	10	3270	5034	1764
10	11	5034	7536	2502
11	12	7536	8332	796
12	13	8332	8418	86

3.2 Contrastive analysis of phase change energy storage device in different operating conditions

In order to get known of the heat exchange performance of the device, working condition testing is carried out. Two energy storage working conditions and two energy release working conditions are tested in this paper. Working conditions with 3°C and 5°C temperature difference are tested in the two energy storage working conditions and the different working temperature is shown in table 3.

Table 3 supply water temperature in different working conditions

Working condition	supply water temperature
1	51°C
2	53°C
3	44°C
4	42°C

3.3 Heat storage test

3.3.1. Working condition 1

The temperature of the PCMI is equal before starting the experiment and the very temperature must be lower than the minimum phase change temperature. The temperature of supply water remains 51°C and the experiment should be ceased when the center temperature of PCM is 2°C higher than the phase change temperature. Figure 9 gives the PCM temperature range with time in working condition 1.

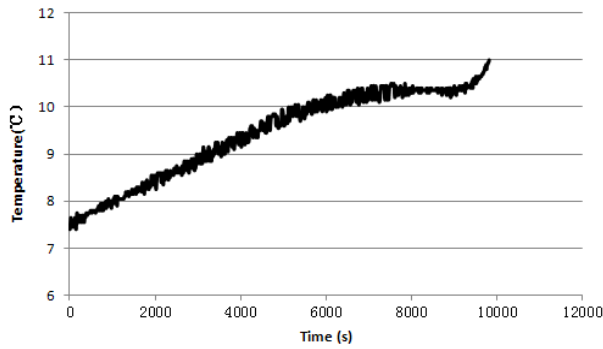


Figure 9 The PCM temperature change curve

Figure 9 gives the temperature variation with time which indicates that the material melting process mainly happened in 5000~9000 seconds, temperature range from 45 to 47°C. With a temperature difference of 3°C between the water and material, the phase change melting process takes about 3 hours.

3.3.2. Working condition 2

The temperature of supply water remains 53°C and the experiment should be ceased when the center temperature of PCM is 3°C higher than the phase change temperature. A temperature difference of 5°C exists between the PCM and water. Figure 10 gives the PCM temperature range with time in working condition 2.

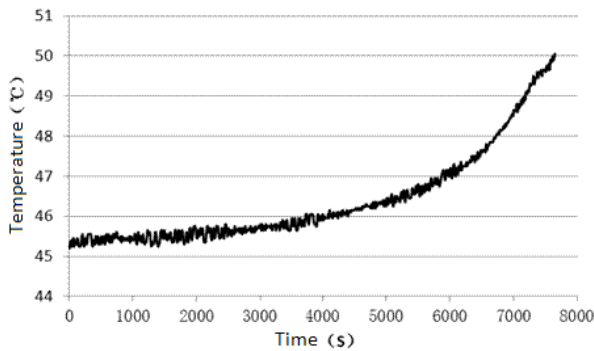


Figure 10 the PCM temperature change curve

Compare figure 9 with Figure 10, conclusion is arrived at that the heat exchange curve of 5°C temperature difference become relaxed than 3°C which means that 5°C temperature difference improved the heat exchange performance compared with 3°C temperature difference. The heat exchange performance is improved obviously and the heat exchange time is less than 2 hours, which is short by about 1/3. Finally, conclusion is drawn that the temperature of supply water should be higher in order to shorten operating time.

3.4 Heat release test

3.4.1 Working condition 3

Figure 11 gives the PCM temperature- time curve when supply water temperature remains 44°C.

3.4.2 Working condition 4

Figure 11 gives the PCM temperature- time curve when supply water temperature remains 42°C.

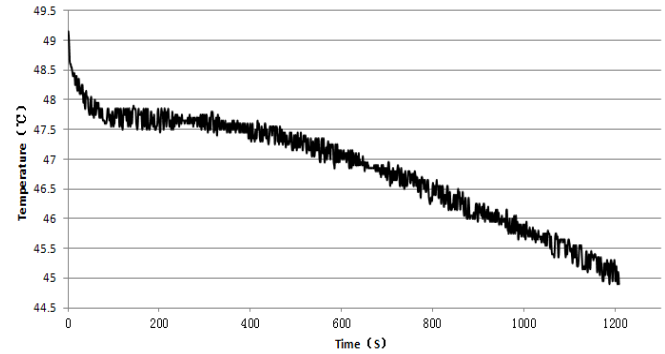


Figure 11 The PCM temperature change curve

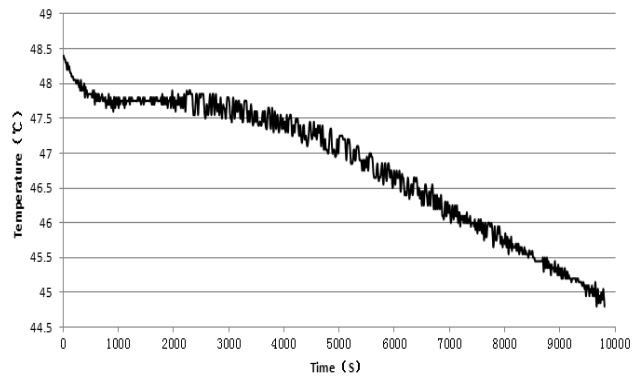


Figure 12 the PCM temperature change curve

Compare figure 10 with Figure 11, conclusion is arrived at that the temperature variation trend of the two working conditions approaches, bigger temperature difference can only shorten operating time by about 0.5 hour, which means that enlarge temperature difference is not an effective way to enhance heat exchange.

3.5 Quantitative analysis

In order to get the heat exchange ratio of the two kinds of PCMs, quantitative analysis is done in this paper. Here is the major parameters related.

- (1) Phase change cool storage marked constantly as $Q_{冷1}, Q_{冷2}, Q_{冷3} \dots Q_{冷n}$;
- (2) Phase change heat storage marked constantly as $Q_{热1}, Q_{热2}, Q_{热3} \dots Q_{热n}$;
- (3) Specific heat of water marked as C ;

- (4) Mass flow rate of water marked as M_w ;
 (5) Time of heat exchange marked as t ;
 (6) Supply water temperature marked constantly as $T_{a1}, T_{a2}, T_{a3} \dots T_{an}$;
 (7) Center water temperature marked constantly as $T_{b1}, T_{b2}, T_{b3} \dots T_{bn}$;
 (8) Output water temperature marked constantly as $T_{c1}, T_{c2}, T_{c3} \dots T_{cn}$;

Water flow over PCCSM and then PCHSM, the center temperature of water represent the water changes heat with one material but not the other. The quantity ratio of exchanged energy can be obtained through calculation. For moment k :

$$Q_{冷k} = C \cdot M_w \cdot t \cdot (T_{ck} - T_{bk}) \quad (1)$$

$$Q_{热k} = C \cdot M_w \cdot t \cdot (T_{bk} - T_{ak}) \quad (2)$$

$$\frac{Q_{冷k}}{Q_{热k}} = \frac{C \cdot M_w \cdot t \cdot (T_{ck} - T_{bk})}{C \cdot M_w \cdot t \cdot (T_{bk} - T_{ak})} = \frac{T_{ck} - T_{bk}}{T_{bk} - T_{ak}} \quad (3)$$

during the whole period:

$$\frac{Q_{显}}{Q_{潜}} = \frac{C \cdot M_w \cdot t \cdot \sum_{k=1}^n (T_{ck} - T_{bk})}{C \cdot M_w \cdot t \cdot \sum_{k=1}^n (T_{bk} - T_{ak})} = \frac{\sum T_c - \sum T_b}{\sum T_b - \sum T_a} \quad (4)$$

$$\frac{Q_{显}}{Q_{总}} = \frac{C \cdot M_w \cdot t \cdot \sum_{k=1}^n (T_{ck} - T_{bk})}{C \cdot M_w \cdot t \cdot \sum_{k=1}^n (T_{ck} - T_{ak})} = \frac{\sum T_c - \sum T_b}{\sum T_c - \sum T_a} \quad (5)$$

3.6. Result

The four working conditions are analysis and the final calculation result is given in table 3.

The specific heat takes a small part in the whole energy quantity but as high as 20% in the energy release peak point which obviously improved the performance of the device. Data indicates that two-way phase change energy storage can take advantage both latent heat and specific heat to improve heat exchange performance, which has certain superiority.

Table 3 Latent heat and sensible heat ratio

Working condition	Energy ratio	
	Specific heat:	Specific heat:
	latent heat	latent heat
1	8.26%	7.63%

2	7.67%	7.12%
3	26.18%	20.75%
4	32.33%	24.43%

4. CONCLUSION

1. The phase change process of the eutectic content happens within the temperature of 9~11℃ and heat exchange should be ceased at 11℃. The specific heat takes up 7.89% of the amount of latent heat.
2. Enlarging the temperature difference of material and water is an effective way to improve heat exchange performance in the heat storage process. 2℃ temperature difference can shorten operating time by 1/3.
3. For the energy release process, enlarging the temperature difference of material and water can not effectively improve heat exchange performance.
4. Specific heat being a relatively small part of the whole energy quantity but take up 20% of the peak energy release. Two kinds of PCM combining can effectively improve the performance of heat exchange and two-way phase change energy storage has certain superiority.

ACKNOWLEDGMENTS

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OPTIMIZATION ANALYSIS OF MAXIMUM UTILIZED QUANTITY OF PHASE CHANGE MATERIAL FOR HEAT STORAGE TYPE AIR CONDITIONING COOLING WATER SYSTEM

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ABSTRACT

As the increasing of equipment and staff, the heat capacity of existing air conditioning cooling water reservoir of protective engineering becomes shortage. Yet the heat storage capacity can be enhanced by use of heat storage type air conditioning cooling water reservoir (HSACCR), by added phase change heat storage modules (PCHSMs) into the reservoir. The model experiment of heat storage type air conditioning cooling water system, in this paper, is established to study the heat storage performance of HSACCR and optimize the maximum utilized quantity of phase change material (PCM). The results from model experiment show that the heat storage capacity increases with the increasing number of PCHSMs. When the addition of PCHSMs account for 13.5% of the rated water volume of reservoir, the heat storage capacity is enhanced by 42.25% by compared with the previous reservoir. Then, the analysis of the maximum utilized quantity of PCM is carried out based on the testing results, and it indicates that the maximum utilized quantities of PCHSMs is that the total volume of PCHSMs account for 24%~27% of the rated water volume of reservoir.

Keywords: Phase change material, Air conditioning cooling water reservoir, Heat storage capacity, Phase change heat storage module, Maximum utilized quantity.

1. INTRODUCTION

Civil air defense engineering projects are built to protect people's air defense command, communications and masking protection building. The normal operation of the air conditioning system is an important guarantee for the civil air defense engineering with good thermal environment [1]. In order to avoid the exposure of civil air defense engineering due to the application of the cooling tower, the air conditioning cooling water is provided by air conditioning cooling reservoir constructed in the underground or mountain. However, with the increasing in heating equipment and people, the internal condensation heat of civil air defense projects significant increased. Because of limited initial design capacity of the existing air conditioning cooling reservoir, it is definitely difficult to run the air conditioning system continuously. Furthermore, for existing air conditioning cooling reservoir, it is hard to add the cooling tower and unrealistic to expand air conditioning cooling reservoir to increase the water

capacity of the air conditioning cooling reservoir [2]. Therefore, to ensure the normal operation of the air conditioning system of the civil air defense projects, it is necessary to find out effective ways for the air conditioning cooling reservoir to enhance the heat storage capacity. For this purpose, phase change energy storage technology [3-5] could be used to solve that problem. As a result, an efficient solution had been worked out by our research group [6], and the specific measure of this solution is that placing phase change heat storage modules (PCHSMs) into the air conditioning cooling reservoir and thus formed a new type air conditioning cooling reservoir, which is defined as heat storage type air conditioning cooling water reservoir (HSACCR). The study emphasis in this paper, in order to ensure the utilization effectiveness, depending on the testing results of model experiment of heat storage type air conditioning cooling water system, is laid on analyzing and suggesting the maximum utilized quantity of phase change material (PCM).

Nomenclature		Symbols	
Abbreviation			
HSACCR	Heat storage type air conditioning cooling water reservoir	Y	Volume percentage of total PCHSMs volume accounting for the rated water volume of reservoir (%)
PCHSM	Phase change heat storage module	X	Maximum temperature of PCHSM which located at the most unfavorable position of the reservoir, recorded by measuring point 3# (°C)
PCM	Phase change material		
PET	Polyethylene terephthalate		

2. MODEL EXPERIMENT OF HSACCR

2.1 Apparatus and running process

The apparatus of HSACCR model experiment consist of one reservoir used as air conditioning cooling reservoir, one heating water tank, one circulating pump, several hot water pipes, copper ball valves and heat preservation material. The rated water volume of reservoir during test is 2.5m³. Heating water tank with an electric heater of a power of 3kW is used to simulate the condensation heat generated by air conditioning system. The insulated HSACCR model experiment is shown in Figure 1.



Figure 1 Thermal insulation of model experiment

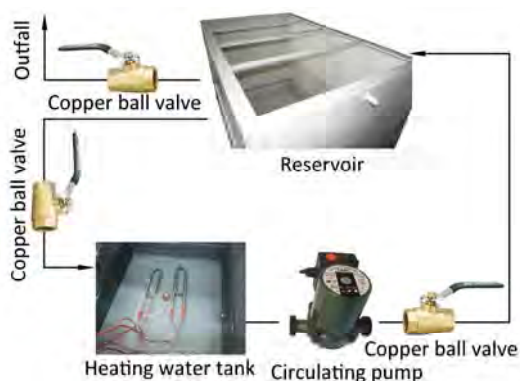


Figure 2 Running process of HSACCR model experiment

The running process of HSACCR model experiment (Figure 2): the circulating pump as a power source, hot water in the heating water tank is pumped into the reservoir, then the heat exchange will be

occurred between the hot and cold water in the reservoir.

Relying on water pressure, water in the reservoir is transported to the heating water tank via the outlet of the reservoir. The heated water in the heating water tank is pumped into the reservoir again. This process will be not stopped until outlet water temperature of the reservoir reaches at 37°C. The flow and velocity of the circulating water are set to 1.13m³/h and 1m/s respectively.

2.2 PCHSMs and placement

Homemade Na₂SO₄·10H₂O composite phase change material which with the phase change temperature of 26~28°C and the latent heat of 150J/g is used as PCM in HSACCR model experiment. PCM is encapsulated into a cylindrical polyethylene terephthalate (PET) plastic bottle, and we call “one PET bottle with PCM” is a phase change heat storage module. The diameter and net volume of PCHSMs are 65mm and 5.5×10⁻⁴ m³ respectively.

PCHSMs are placed in the reservoir and should give full consideration to the heat exchange performance between PCHSMs and water. PCHSMs are equidistantly strung by rope and evenly placed in the reservoir with staggered form (Figure 3). And in HSACCR model experiment, testing are divided into seven cases that the different number of PCHSMs that 0 (no bottle placed in the reservoir), 104, 208, 312, 412, 512 and 614 are respectively placed into the reservoir. When 0 PCHSM is placed into the reservoir, heat is stored merely by water sensible heat. When different number of PCHSMs (104, 208, 312, 412, 512 and 614 PCHSMs) are respectively placed into the reservoir, heat is stored relying on the latent heat (primary) and sensible heat (secondary) of PCM and sensible heat of the rest water that excludes the total volume of PCHSMs from the rated water volume of reservoir (2.5m³).

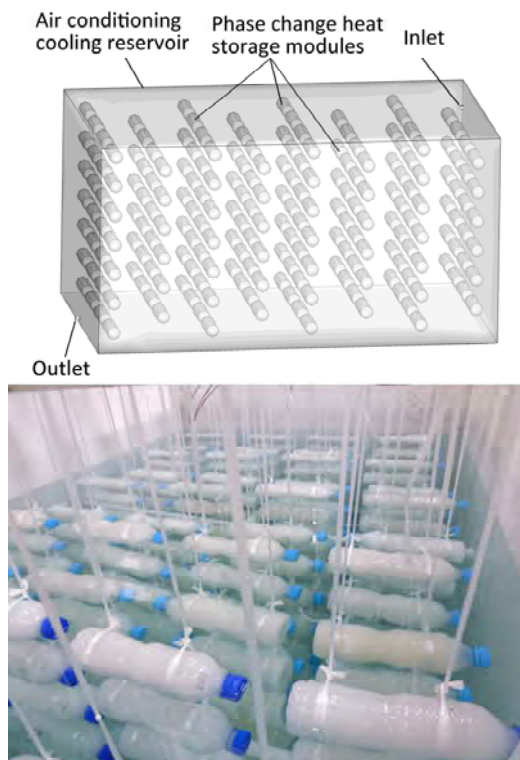


Figure 3 PCHSMs in reservoir with staggered form (Shown above is a schematic diagram of staggered form, shown below is the actual placement state.)

2.3 Data collector and measuring points

The paperless recorder is used for collecting the testing data, setting the data save time interval of 1s, temperature accuracy is 0.1°C . Platinum resistance thermometers (PT100) which the temperature range of $-50\sim 450^{\circ}\text{C}$ are used as temperature sensors. The testing instruments are calibrated before the test.

Heat storage capacity of HSACCR is reflected in the time when the outlet water temperature of the reservoir reaches at 37°C (This temperature is usually a upper limit temperature of air conditioning cooling water system). The measuring points of inlet and outlet of the reservoir are arranged in the heating water tank, and respectively marked as 1# and 2#. Another measuring point is put into a PCHSM which located at the most unfavorable position of the reservoir, and marked this measuring point as 3#.

3. DISCUSSION OF TESTING RESULTS

To ensure the test data of each experimental group has a better contrast, the test data are all in a temperature interval which limits the reservoir outlet water temperature rise from 19°C to 37°C . The analysis and research of the testing results will be in this temperature interval.

3.1 Temperature trends of inlet and outlet

When 0 PCHSM (no PCHSM) is placed into the reservoir, the temperature trends of inlet and outlet are shown in Figure 4. It shows the temperature of inlet and outlet rise linearly. The average temperature difference between inlet and outlet is 2.9°C , and it takes 17.37h to rise the temperature of outlet water from 19°C to 37°C .

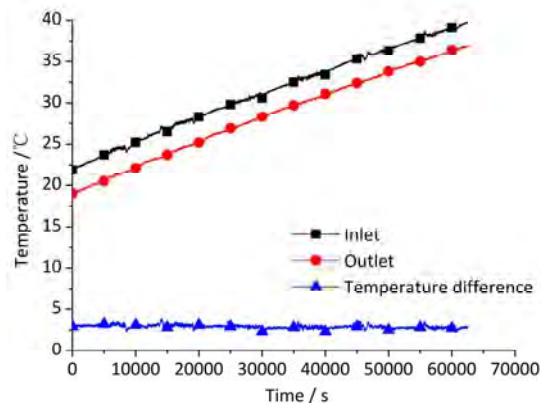


Figure 4 Temperature trends of inlet and outlet when 0 PCHSM in the reservoir

Figure 5 shows the inlet and outlet temperature trends when 512 PCHSMs are placed. The results indicate that the temperature of inlet and outlet rise stable, the mean temperature difference between inlet and outlet is 3°C . It takes 23.13h for temperature of outlet water to go up from 19°C to 37°C . Compared with the results of which 0 PCHSM is placed into the reservoir, we can see that inlet and outlet temperature rises linearly in former 25000s, then the heating rate gets slower during 25000~68000s, and then the temperature of inlet and outlet linearly rise again from 68000s. Temperature increasing trends of inlet and outlet of placing 104, 208, 312, 412 and 614 PCHSMs into reservoir are similar to that of 512 PCHSMs, and the mean temperature difference between inlet and outlet is approximately 3°C , the only different is the time they taken.

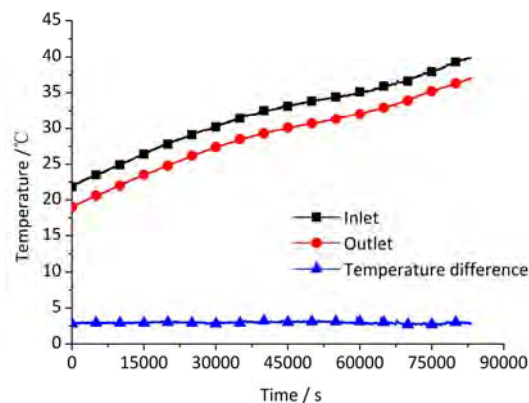


Figure 5 Temperature trends with 512 PCHSMs in reservoir

3.2 Heat storage capacity enhancing with increasing PCHSMs

The heat storage capacity of HSACCR is reflected in the change of outlet water temperature over time. For this purpose, outlet water temperatures which changing with time are obtained by analyzing test data from placement of different number of PCHSMs in the reservoir, and the results are shown in Figure 6. Obviously, time when outlet water temperature rises from 19°C to 37°C gets longer with the increase in number of PCHSMs. It indicates heat storage capacity of reservoir has a significant strengthening after the addition of PCHSMs.

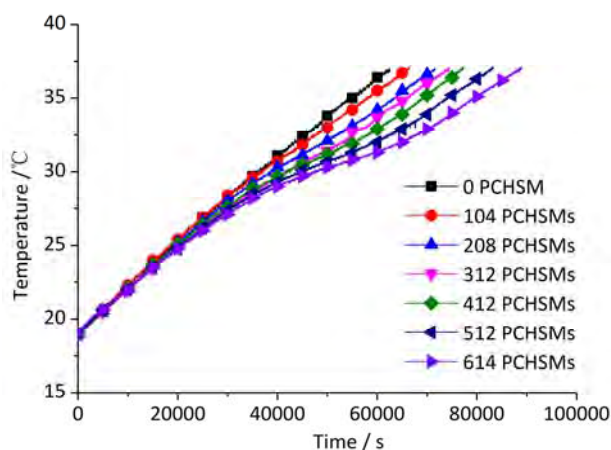


Figure 6 Outlet water temperature trends

Heat storage capacity of HSACCR is shown in Table 1. From Table 1, it shows that as the more number of PCHSMs put into the reservoir, the longer time it takes when outlet water temperature rising from 19°C to 37°C, the larger heat storage capacity the reservoir has. Volume of 614 PCHSMs accounting for 13.51% of the rated water volume of reservoir, and comparing with the addition of 0 PCHSMs, the heat storage capacity of reservoir is enhanced to 42.25%. Consequently, PCHSMs can significantly improve the heat storage capacity of air conditioning cooling reservoir.

Table 1 Heat storage capacity of HSACCR

Nu mber of PCHSMs	PCHS Ms volume percentage (%)	He at storage capacity (h)	Perenta ge of increased heat storage capacity (%)
0	—	17.37	—
104	2.288	18.47	6.33
208	4.576	19.88	14.45
312	6.864	20.69	19.11
412	9.064	21.47	23.60
512	11.264	23.13	33.16
614	13.508	24.71	42.25

3.3 Heat storage three stages of PCHSMs

Due to the PCHSMs placed in the reservoir, the heating rate slows down that occurred in the temperature rising process. Heat storage processes of PCHSMs are divided into three stages (shown in Figure 7): First stage, temperature of reservoir water is lower than the phase change temperature of PCM. So PCM is solid and heat is stored by sensible heat of water and solid-phase PCM. Second stage, water temperature of the reservoir increases from 26°C to 35°C. When the water temperature reaches 26~28°C, PCM begins to melt and uses latent heat to store heat. The higher water temperature rising, the more PCM melting, and a solid-liquid coexisting state is appeared. When the water temperature goes up to 35°C, PCM is completely melted. Heat storage in the second stage of a combination of sensible heat of water, phase change latent heat and parts of PCM sensible heat in solid-liquid coexisting state. Because of the latent heat is much larger than sensible heat, as a result, the temperature of outlet becomes slower rising. The third stage, water temperature of reservoir is higher than 35°C. PCM is completely melted to liquid, heat is stored by sensible heat of water and liquid PCM. The temperature, in the central of PCHSM, changes over time recorded by 3# measuring point is shown in Figure 7.

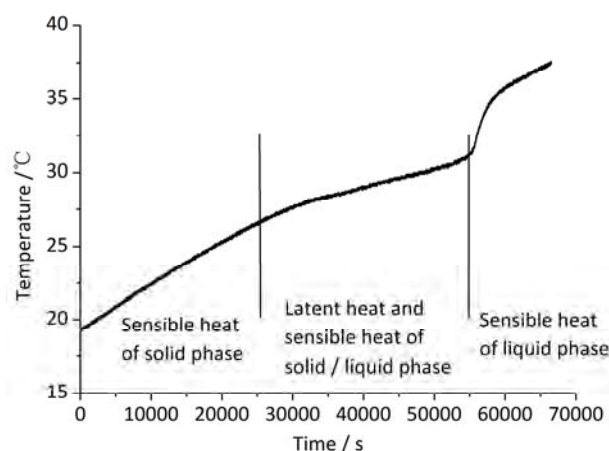


Figure 7 Temperature trend of PCHSM center

4. OPTIMIZATION ANALYSIS OF MAXIMUM UTILIZED QUANTITY OF PCM

Heat storage capacity of HSACCR increases as number of PCHSM increasing, however, it is obvious that the heat storage capacity will not limitless increasing. The heat storage processes of PCHSMs will be taken a long period of time and not instantaneous or at a short time, and with the increasing of number of PCHSMs, the water in the reservoir will be reduced. Therefore, there is necessarily a state that the outlet water temperature has already reached at 37°C (a upper limit temperature of air conditioning cooling water

system) yet parts of PCM still be solid-phase. That not only cannot exert the strongest advantage of heat storage of PCM, but also may reduce the heat storage capacity of reservoir. Hence the maximum utilized quantity of PCM should be researched and optimized. The following discussion assumes that the volumes of PCM and PCHSMs are equal.

An evaluated method to optimize the maximum utilized quantity of PCM is when the outlet temperature reaches at 37°C, the maximum temperature of PCHSM which located at the most unfavorable position of the reservoir reaches at the completely melting temperature (35°C). The temperature changes of PCHSM located at the most unfavorable position of the reservoir are recorded by measuring point 3#, and when the outlet temperature reaches at 37 °C , the maximum temperatures of measuring point 3# that with different number of 104, 208, 312, 412, 512 and 614 PCHSMs are given respectively. The result that relationship between the maximum temperatures of PCHSM located at the most unfavorable position of the reservoir and their corresponding total PCHSMs volume percentage is shown in Figure 8. Figure 8 shows that the maximum temperatures are reduces with the increasing of total PCHSMs volume percentage. From linear fitting, PCHSMs volume percentages Y and maximum temperatures of PCHSM located at the most unfavorable position of the reservoir X may be developed the following relationship:

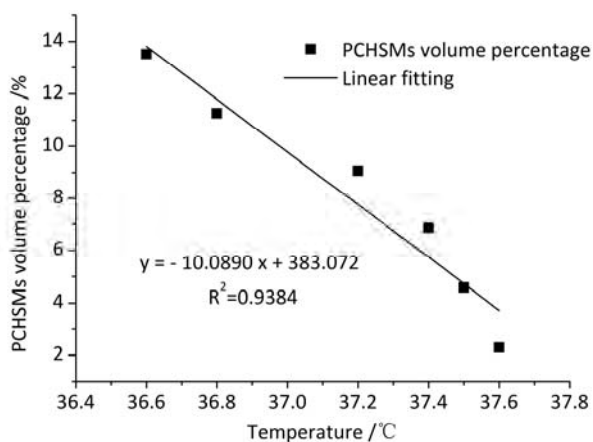


Figure 8 Maximum temperatures and volume ratio

$$Y = -10.089X + 383.072$$

where Y is the volume percentage of total PCHSMs volume accounting for the rated water volume of reservoir, %; X is the maximum temperature of PCHSM which located at the most unfavorable position of the reservoir, recorded by measuring point 3#, °C. The completely melting temperature of PCM is $X = 35\text{ °C}$, then $Y = 29.955\%$. However, in practical projects, to ensure the utilization effectiveness, safe utilized quality of 10%~20% should be considered. Consequently, the maximum utilized quantities of PCHSMs is that the total volume of PCHSMs account for 24%~27% of the rated water volume of reservoir.

5. CONCLUSIONS

It is an efficient approach to enhance the heat storage capacity of air conditioning cooling water reservoir by placing PCHSMs into reservoir. The results from HSACCR model experiment show that the heat storage capacity increases with the increasing number of PCHSMs. When the addition of PCHSMs account for 13.5% of the rated water volume of reservoir, the heat storage capacity is enhanced by 42.25% by compared with the previous reservoir. Depending on the testing results, the analysis of the maximum utilized quantity of PCM is carried out, and it indicates that the maximum utilized quantities is that the total volume of PCHSMs account for 24%~27% of the rated water volume of reservoir. Therefore, in new HSACCR and renovation of existing air conditioning cooling water reservoir, the proper utilized quantities of PCHSMs may be placed into the reservoir depending on the requirement of design. However, in order to ensure the utilization effectiveness, the maximum total volume of PCHSMs should not exceed 27% of the rated water volume of reservoir.

ACKNOWLEDGMENTS

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THE SIMULATION AND ANALYSIS OF STORAGE BALL BASED ON ANSYS SOFTWARE

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ABSTRACT

Ice ball tank is the most important equipment of the ice storage system, the ice ball accumulation mode owe the merit such as large heat transfer area per unit volume, simple structure, high reliability, low water resistance. The charging characteristics of single ball in the ice ball tank directly impact the overall system heat transfer efficiency and cold storage capacity. Numerical analysis is used to simulate and analyze the phase change process of storage ball, simulation results show that:

The lower temperature of secondary refrigerant, during cool storage process the greater heat transfer temperature difference, the shorter solidification time of ice storage ball and the faster phase interface moving; during cool release process the smaller temperature gradient, the longer time of release cold. When the temperature drops to certain value, the solidification time is not shortened in proportion with the temperature decreasing, and the efficiency of the refrigerating unit drops greatly; so when determining the temperature of secondary refrigerant, we should comprehensively consider valley power time, characteristic of storage ball and the efficiency of refrigerating unit.

The required solidification time of storage ball is inversely proportional to the ball diameter, so the design of ball diameter should be appropriate to control the cool- storage time in the valley power, then achieve the purpose of energy and money saving.

Keywords: Ice Storage ball, Phase change, Finite element.

Cool storage tank is one key part of cool storage system, which its cold storage capacity and heat exchange efficiency have a very important influence on air conditioning system. The charging characteristics of single ball in the storage tank is the foundation of the whole system about the thermal efficiency^[1]. Based on the theory of finite element method, ANSYS software is used to simulate and analyze the phase change process of storage ball in this paper.

1. INTRODUCTION OF PHASE CHANGE ANALYSIS USING ANSYS

One of the most powerful features of ANSYS software is the ability to analyze phase change process which is a nonlinear transient problem. ANSYS deal with molten latent heat problem as shown in figure 1, the latent heat is considered during the phase change process by defining the enthalpy changing with temperature. The unit of Enthalpy is J/m³, which is the temperature integral of the product of the density and specific heat:

$$h = \int \rho c(T) dT \quad (1)$$

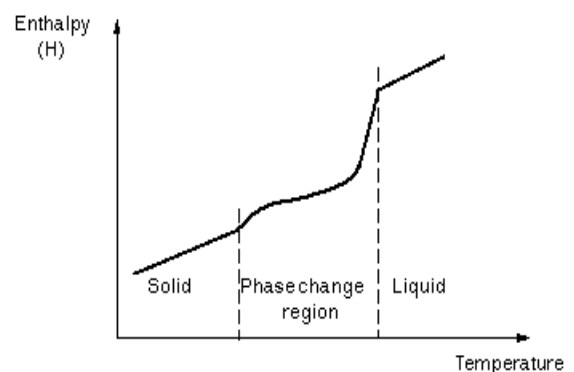


Figure 1 Curvilinear transformation of phase enthalpy from solid to liquid

The low-order heat unit is selected in ANSYS, such as PLANE55 or SOLID70, PLANE55 is plane unit and axisymmetric element, which has four nodes, each node has only one degree of freedom – temperature and used for 2-d heat conduction analysis.

PLANE55 Energy (temperature) diffusion of matrix function as follows:

$$T = \frac{1}{4} T_i (1 - s) \quad (2)$$

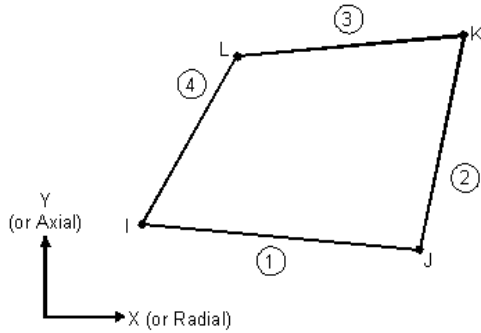


Figure2 PLANE55 Finite Element

Solid70 is the three-dimensional entity used for 3-d heat conduction, which has eight nodes, each node only one degree of freedom---temperature.

Its Energy (temperature) diffusion of matrix function is as follows:

$$T = \frac{1}{8} T_i (1-s) \quad (3)$$

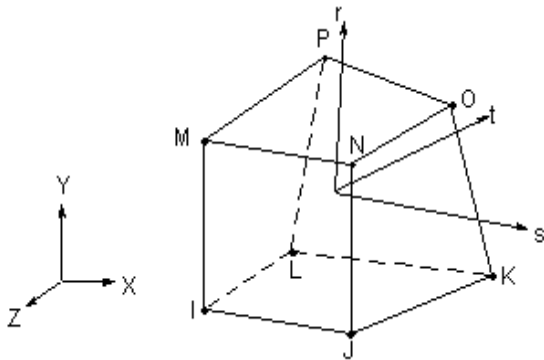


Figure3 SOLIDE70 Finite Element

2. THE PHYSICS AND MATHEMATICAL MODELS OF STORAGE BALL

The phase change process of cool storage material in the ball is with phase change heat conduction and complex natural convection heat transfer. Considering the ball diameter is smaller and liquid temperature difference is not big, therefore, the temperature difference caused by the natural convection heat conduction is relatively weaker compare to heat conduction. Following assumption is made for analyze conveniently ,

Assumption of the physical model:

1) Phase change heat transfer in the ball is given priority to natural convection which is ignored here during thermal conductivity.

2) Phase change temperature is constant;

3) Phase change material solid, liquid specific heat, thermal conductivity, density is constant;

4) The occurrence of phase transition from the outer to the inner;

5) The temperature of secondary refrigerant is constant;

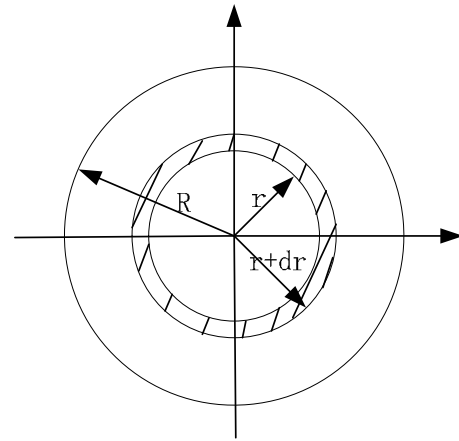


Figure4 Physical Model for Cool Storage Ball

According to assumption, the mathematical model of storage ball is as follow:

$$\rho_s C_{ps} \frac{\partial T_s}{\partial t} = k_s \frac{1}{r} \frac{\partial^2 (r T_s)}{\partial r^2} \quad (\text{solid zone}) \quad (4)$$

$$\rho_l C_{pl} \frac{\partial T_l}{\partial t} = k_l \frac{1}{r} \frac{\partial^2 (r T_l)}{\partial r^2} \quad (\text{liquid zone}) \quad (5)$$

$$k_s \frac{\partial T_s}{\partial r} - k_l \frac{\partial T_l}{\partial r} = L \rho_l \frac{\partial r}{\partial t} \quad (\text{Liquid-Solid interface}) \quad (6)$$

The initial conditions:

$$T(r, 0) = T_0 \quad (0 \leq r \leq R) \quad (7)$$

The boundary conditions:

$$\left. \frac{\partial T}{\partial r} \right|_{r=0} = 0; \quad T|_{r=R} = T_f \quad (8)$$

Which ρ_s, ρ_l is the density of phase change material when solid phase and liquid phase respectively, (kg/m^3) ; T_f is the temperature of secondary refrigerant.

3. THE COOL STORAGE SIMULATION AND ANALYSIS OF SINGLE ICE STORAGE BALL

3.1 Meshing and Parameter Set

The finite element model of the center section was established for storage ball with different radius, the finite element unit adopts plane 55, free meshing. Selected, -10°C chosen as the reference temperature, the water enthalpy change Δh relative to the -10°C is shown in figure 5.

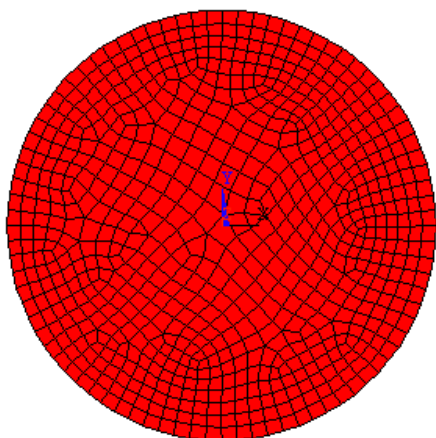


Figure4 Meshing of Center Section R80mm

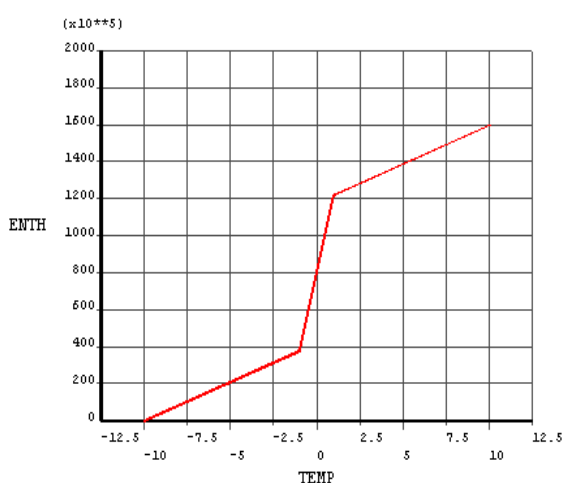


Figure5 Curve Diagram of Enthalpy

The density of water and ice is set as 1000kg/m^3 (assuming constant), specific heat, coefficient of thermal conductivity vary with temperature as shown in figure 6 and figure 7, then the initial parameter for calculation is completed. The boundary conditions of cool storage ball set as the first kind of boundary conditions.

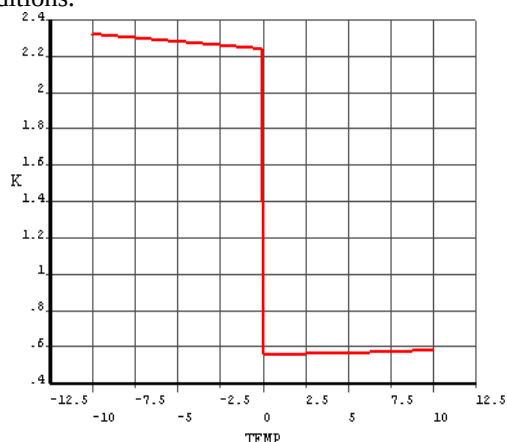


Figure6 Coefficient of Heat Conductivity Change with Temperature

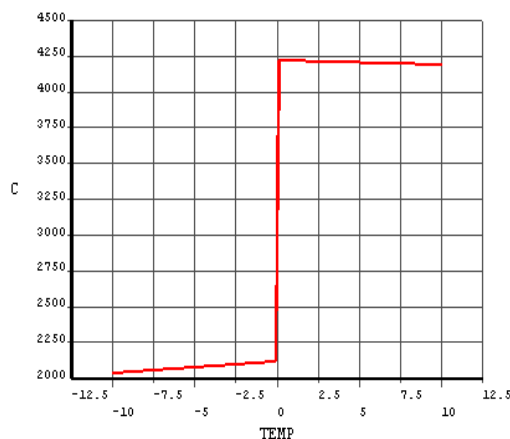
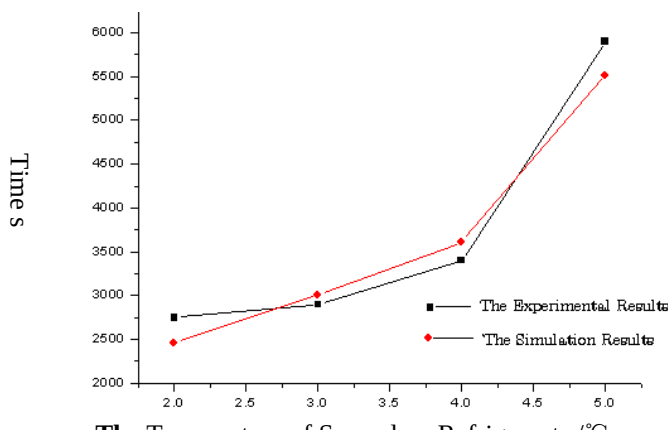


Figure7 Special heat change with temperature

3.2 The reliability test of simulation results

In order to test and verify the reliability of the calculation results, only change the physical parameters of phase change materials, the one dimensional solidification process was simulated for cool storage ball on the literature [2] using ANSYS software ($R = 35\text{ mm}$, phase transition temperature is 8°C , phase change latent heat is 66 kJ/kg , the density is 1615.3 kg/m^3), calculation result was compared with the literature [2] experimental results as shown in figure 8.

The graph shown that the theoretical simulation and experimental results are in good agreement, which means that the method is feasible, and can be used to simulate the solidification process of ice storage.



The Temperature of Secondary Refrigerants / $^\circ\text{C}$
Figure8 The comparison of simulating result with the experiment of document[59]

3.3 The temperature distribution of Ice-ball

Figure 3.15 shows the temperature distribution of Ice-ball at time 1500s when $R = 80\text{ mm}$, cooling load temperature is -5°C ,

NODAL SOLUTION
TIME=1500
TEMP (AVG)
RSYS=0
SMN=-5
SMD=-.799E-07

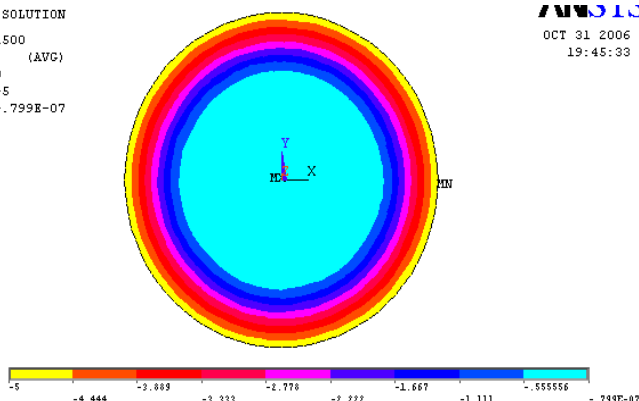


Figure9 The temperature distribution of Ice-ball at time 1500s

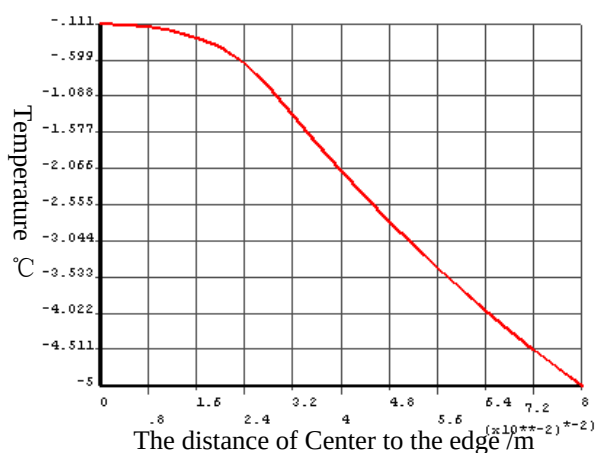


Figure 10 Temperature variation curve center to boundary at 5005s

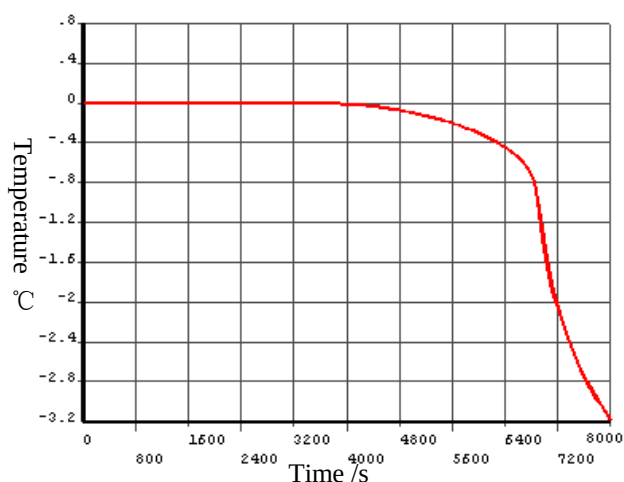


Figure 11 Temperature variation curve from of center node changed with time

Simulation analysis not only can get equivalent temperature nephogram of different time, the curve of temperature changes over time of all nodes can also get. as shown in figure 10, which is helpful to analyze the phase change process of temperature change; meanwhile the distribution of temperature on

ANSYS
OCT 31 2006
19:45:33

geometric scale can also be got, as shown in figure 11, the temperature distribution can be obtained accurately, so it is is very convenient to analyze the phase change.

3.4 The radius of ice storage ball and the condensing temperature effects on freezing time

when the temperature of secondary refrigerants is -5°C the freezing time of vary ice ball radius can be got by simulating, as show in figure12. Fitting curve show that when the temperature of secondary refrigerants is constant, the solidification time is basically a proportional to the ball diameter, the smaller ball diameter is, the faster the transformation, the shorter the time needed for phase change.

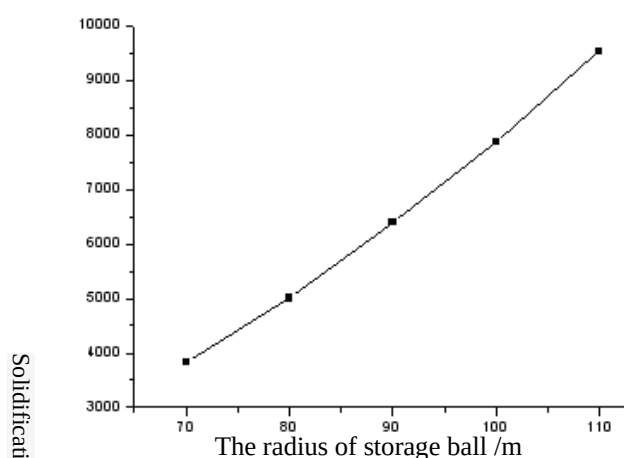
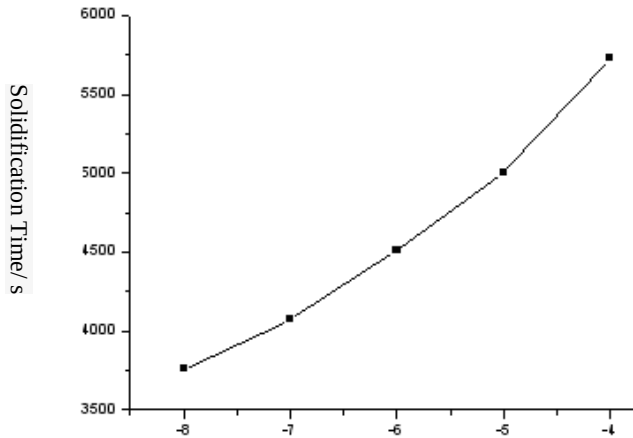


Figure12 The freezing time of vary radius ice all when the temperature of secondary refrigerants is -5°C

When cold storage ball diameter is constant ($R = 80 \text{ mm}$), the solidification time of cool storage ball is shown in figure 13 with cooling load temperature, respectively -4°C , -5°C , -6°C , -7°C , -8°C , the larger heat transfer temperature difference of cold storage ball surface is, the greater the heat transfer increases, the shorter coagulation time is, meaning that the coagulation time is inversely proportional to heat transfer temperature difference. The difference of solidification time needed between -4°C and -5°C is much longer than between -7°C and -8°C . It means that when the temperature of the cooling load is low to a certain value it is not obvious to shorten solidification time by reduce the temperature of secondary refrigerants. By contrast, the refrigeration unit efficiency is greatly reduced. The heat transfer temperature difference should be comprehensive consideration of system load, cold storage and refrigeration unit of the size of the economy



The Temperature of Secondary Refrigerants/ °C

Figure13The freezing time varied with
 temperature of different secondary refrigerants
 when radius is 80mm

4. CONCLUSION:

Using ANSYS software can be very convenient for heat transfer analysis of phase change problem, the influence of different radius and different refrigerant temperature in cool storage are simulated, the simulation results in cool storage ball has reference value for design and improvement.

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APPLICABILITY ANALYSIS ON PCM-BASED FREE COOLING TECHNOLOGY FOR BUILDINGS IN FIVE TYPICAL CITIES IN CHINA

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ABSTRACT

Phase change material (PCM)-based free cooling is understood as a means to store outdoors cold in a thermal energy storage unit using PCM during the night, to supply indoors cooling during the day. The evaluation of applicability of PCM-based free cooling technology is crucial to its application. In this paper, five typical cities locating in different climate zones in China were selected. Based on the hourly outdoor air dry bulb temperature in the typical meteorological year, the air conditioning cooling load and the potential of PCM for cold storage in summer for the five typical cities were evaluated. Then, combined with the two aspects, an applicability analysis on PCM-based free cooling technology for buildings was presented.

Keywords: PCM-based free cooling technology, Air conditioning cooling load, Potential of PCM for cold storage, Applicability.

1. INTRODUCTION

The use of thermal energy storage (TES) system using phase change material (PCM) is an effective way to compensate the mismatches that occur between the times of energy supply and demand, and has the advantages of high energy storage density during phase change at an extremely low temperature difference or a relatively constant temperature. For those reasons, TES system using PCM has gained increasing attention [1-5]. With the constant raising of people's living standard in China, air conditioning is becoming a necessity for most families, and the energy consumption will keep rising. So, it is necessary to promote the application of TES technology using PCM for heating and cooling of buildings, using renewable energy (such as solar energy and natural cold resource). PCM-based free cooling is understood as a

means to store outdoors cold in a TES unit using PCM during the night, to supply indoors cooling during the day.

The evaluation of applicability of PCM-based free cooling technology is crucial to its application. In this paper, based on the hourly outdoor air dry bulb temperature in the typical meteorological year [6], an applicability analysis on PCM-based free cooling technology for buildings in five typical cities in China was presented. The five typical cities are Guangzhou, Kunming, Changsha, Zhengzhou and Shenyang, locating in five different climate zones, respectively. Their geographical positions and climatic zones are shown in Table 1. Due to the great differences between climatic conditions among all the cities, an equal time period of the three summer months (from the beginning of June to the end of August) was selected for all the cities to make a direct comparison of the results.

Table 1 Selected cities and their geographical positions and climatic zones

City	Guangzhou	Kunming	Changsha	Zhengzhou	Shenyang
Latitude / °	23.17	25.02	28.22	34.72	41.73
Longitude / °	113.33	102.68	112.92	113.65	123.45
Climatic zone	Hot summer and warm winter zone	Mild zone	Hot summer and cold winter zone	Cold zone	Severe cold zone

Table 2 Total numbers of CDD26 for whole year in the five typical cities

Cities	Guangzhou	Kunming	Changsha	Zhengzhou	Shenyang
CDD26/°C·d	282	0	179	114	15

Table 3 Monthly average temperature from June to August in the five typical cities

Cities	Guangzhou	Kunming	Changsha	Zhengzhou	Shenyang
Monthly average temperature in June / °C	27.2	20.1	25.6	25.9	21.5
Monthly average temperature in July / °C	28.8	20.0	28.5	27.0	25.7
Monthly average temperature in August / °C	28.0	19.7	27.3	25.7	23.2

Table 4 Monthly average CDH26 from June to August in the five typical cities

Cities	Guangzhou	Kunming	Changsha	Zhengzhou	Shenyang
Monthly average CDH26 in June / °C·d	41	1	30	50	7
Monthly average CDH26 in July / °C·d	76	0	71	52	28
Monthly average CDH26 in August / °C·d	55	0	50	29	14

2. AIR CONDITIONING COOLING LOAD

Cooling degree days (CDD) and cooling degree hours (CDH) can provide a simple estimate of energy consumption by cooling systems during the summer season, for the reason that both of them contain the hot degree and duration [7-9]. In this report, the air conditioning cooling load was evaluated by cooling degree days base on 26 °C (CDD26, °C·d) and cooling degree hours based on 26 °C (CDH26, °C·h).

The total numbers of CDD26 for the whole year can be calculated by:

$$CDD26 = \sum_{n=1}^{365} (T_n - 26)D \quad (1)$$

Where n is the number of days of the year; T_n is the daily outdoor average dry bulb air temperature, °C; D is one day, d. If T_n is lower than 26 °C, the temperature difference should be taken as 0 °C.

The total numbers of CDH26 for a day can be calculated by:

$$CDH26 = \sum_{i=1}^{24} (T_i - 26)h \quad (2)$$

Where i is the number of hours of the day; T_i is the hourly outdoor dry bulb air temperature, °C; h is one hour, h. If T_i is lower than 26 °C, the temperature difference should be taken as 0 °C.

Table 2 shows the total numbers of CDD26 for the whole year in the five typical cities. Table 3 and Table 4 show the monthly average temperature and the monthly average CDH26 from June to August in the five typical cities, respectively. The results show that the temperature and the air conditioning cooling load in summer are high for Guangzhou, Changsha and Zhengzhou, while the temperature and the air conditioning cooling load in summer are lower for Shenyang and Kunming.

3. POTENTIAL OF PCM FOR COLD STORAGE

The monthly average diurnal temperature from June to August in the five typical cities is shown in Figure 1. It is found that the diurnal temperature in all cities has a good value. The average diurnal temperature in the three summer months is 7.0, 6.5, 7.2,

9.0 and 9.2 °C, for Guangzhou, Kunming, Changsha, Zhengzhou and Shenyang, respectively.

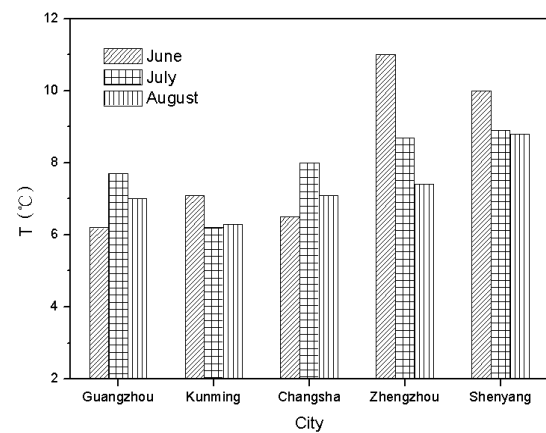


Figure.1. Monthly average diurnal temperature from June to August in the five typical cities

Artmann et al. [10] proposed climate cooling potential (CCP) to evaluate the potential of night ventilation. Based on the concept of CCP and CDH, cold storage degree hours (CSDH, °C·d) was developed to evaluate the potential of PCM for cold storage.

The total numbers of CSDH during the time from h_i to h_f in one day can be calculated by:

$$CSDH = \sum_{h=h_i}^{h_f} (T_m - T_i)h \quad (3)$$

Where i is the number of hours of the day; T_i is the hourly outdoor air temperature, °C; T_m is the melting temperature of PCM, °C; h is one hour, h. If T_m is lower than T_i , the temperature difference should be taken as 0 °C. In this report, the phase change process started at 19:00 pm and ended at 7:00 am in the next day morning. For example, to calculate the value of CSDH on the tenth day, the charging period was from 19:00 pm on the ninth day until 7:00 on the tenth.

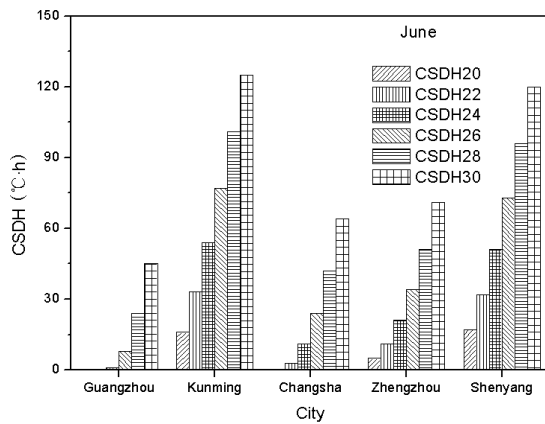


Figure 2 Monthly average CSDH in June in the five typical cities

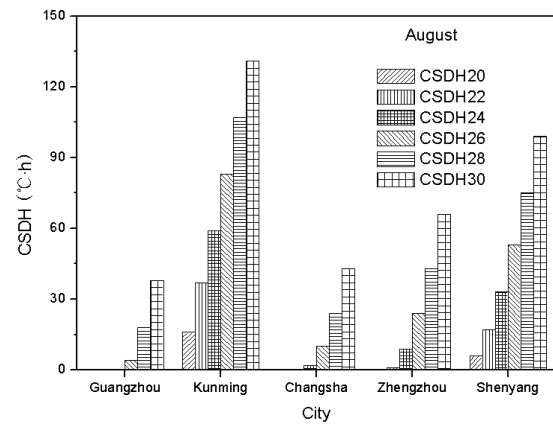


Figure 4 Monthly average CSDH in August in the five typical cities

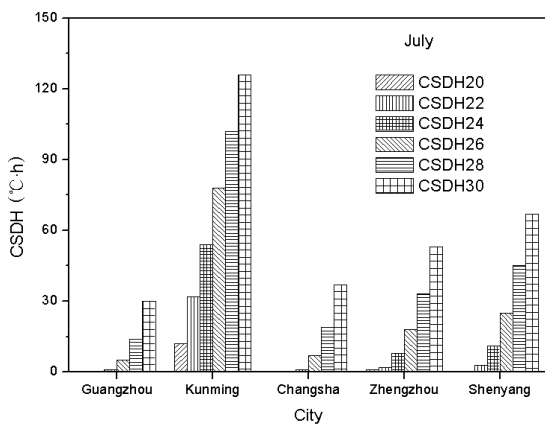


Figure 3 Monthly average CSDH in July in the five typical cities

The monthly average CSDH from June to August in the five typical cities are shown in Figure 2 - Figure 4. CSDH20 to CSDH30 is the value of CSDH for melting temperature of PCM from 20 to 30 °C. The results show that the CSDH increases along with the increment of the melting temperature of PCM, that is, the potential of PCM for cold storage increases. The order of decreasing potential of PCM for cold storage is as follows: Kunming, Shenyang, Zhengzhou, Changsha and Guangzhou. In addition, the lowest melting temperature that the PCM can store cold in all the three summer months is 26, 20, 26, 24 and 24 °C for Guangzhou, Kunming, Changsha, Zhengzhou and Shenyang, respectively.

4. APPLICABILITY ANALYSIS

With the melting temperature of PCM close to the outdoor ambient temperature at night, the cold to be stored will be limited due to the low temperature difference between PCM and outdoor air. While, with the melting temperature of PCM close to the indoor design temperature during the day, the cold to be stored will improve, but the cold to supply indoors cooling during the day is reduced. So, it is important to select an appropriate melting temperature of PCM to maintain indoor thermal comfort, and the selection of PCM must base on local climatic condition and air conditioning cooling load. Numerical simulations by Arkar and Medved [11] showed that a PCM with a melting temperature between 20 and 22 °C is the most suitable for free cooling in the case of a continental climate. Medved and Arkar [12] found that the optimum PCM has a melting temperature that is approximately equal to the average ambient air temperature in the hottest month, and that the free cooling potential is proportional to the average daily amplitude of the ambient air's temperature swings. Butala and Strith [13] considered that the human comfort zone in summer time is between 23 °C and 27 °C. In order to achieve sufficient heat transfer, the temperature difference between the air temperature and the melting temperature of the PCM should be within the range of 3-5 °C. So, for a summer cooling system, the melting temperature of the PCM should be from 19 to 24 °C. Mosaffa et al. [14] considered that in building applications, PCMs with a phase change temperature of 18-30 °C are preferred to meet the need for thermal comfort.

According to "Code for design of heating ventilation and air conditioning (GB50736-2012)" in China [15], the thermal comfort design temperature in summer for the region that people stay for a long time is about 24-28 °C, and the corresponding value for the region that people do not stay for a long time can increase 1-2 °C. Take into consideration about the

temperature difference of 3-5 °C for heat transfer, the melting temperature of PCM for the former can be 20-24 °C, while the value of the latter can be 22-26 °C. Combined with the air conditioning cooling load and the potential of PCM for cold storage in summer, an applicability analysis on PCM-based free cooling technology for buildings was presented as follows.

In Guangzhou and Changsha, when the melting temperature is 26 °C, the monthly average CSDH in all the three summer months are very small. When the melting temperature is 28 °C, the monthly average CSDH from June to August in Guangzhou are 24, 14 and 18 °C·h respectively, and the corresponding values in Changsha are 42, 19 and 24 °C·h respectively. PCM-based free cooling technology in Guangzhou and Changsha has relatively poor applicability. However, due to the high temperature and the high air conditioning cooling load, a TES system using PCM with melting temperature of 28 °C can be selected for cooling or precooling hot fresh air in the two cities.

When the melting temperature is 24 °C, the monthly average CSDH both in July and August in Zhengzhou is small and the value in July in Shenyang is small. When the melting temperature is 26 °C, the monthly average CSDH from June to August in Zhengzhou are 34, 18 and 24 °C·h respectively, and the corresponding values in Shenyang are 73, 25 and 53 °C·h respectively. PCM-based free cooling technology in Zhengzhou and Shenyang has good applicability. Therefore, a TES using PCM with melting temperature of 26 °C can be selected for the thermal comfort design for the region that people do not stay for a long time, along with for the cooling of hot fresh air in the two cities. In addition, a TES using PCM with melting temperature of 24 °C can be selected for the thermal comfort design for the region that people stay for a long time in Shenyang, due to the lower air conditioning cooling load, and the corresponding monthly average values of CSDH from June to August are 51, 11 and 33 °C·h, respectively.

In Kunming, the PCM can store cold in all the three summer months even the melting temperature is 20 °C, and the monthly average CSDH from June to August are 16, 12 and 16 °C·h, respectively. When the melting temperature is 24 °C, the monthly average CSDH from June to August are 54, 54 and 59 °C·h, respectively. Therefore, the potential of PCM for cold storage in Kunming is great. However, due to the low temperature and the low air conditioning cooling load, the thermal comfort environment can be reached by fresh air supply directly without cold storage.

5. CONCLUSIONS

The evaluation of applicability of PCM-based free cooling technology is crucial to its application. In this paper, combined with the air conditioning cooling load and the potential of PCM for cold storage in summer, an applicability analysis on PCM-based free cooling technology for buildings in five typical cities in China

was presented. The conclusions were obtained as follows.

(1) The diurnal temperature in all the five cities has a good value. The order of decreasing potential of PCM for cold storage is as follows: Kunming, Shenyang, Zhengzhou, Changsha and Guangzhou. In addition, the lowest melting temperature that the PCM can store cold in all the three summer months is 26, 20, 26, 24 and 24 °C for Guangzhou, Kunming, Changsha, Zhengzhou and Shenyang, respectively.

(2) In Guangzhou and Changsha, PCM-based free cooling technology has relatively poor applicability. However, due to the high temperature and the high air conditioning cooling load, a TES system using PCM with melting temperature of 28 °C can be selected for cooling or precooling hot fresh air in the two cities.

(3) In Zhengzhou and Shenyang, PCM-based free cooling technology has good applicability. Therefore, a TES using PCM with melting temperature of 26 °C can be selected for the thermal comfort design for the region that people do not stay for a long time, along with for the cooling of hot fresh air in the two cities. In addition, a TES using PCM with melting temperature of 24 °C can be selected for the thermal comfort design for the region that people stay for a long time in Shenyang, due to the lower air conditioning cooling load, and the corresponding monthly average values of CSDH from June to August are 51, 11 and 33 °C·h, respectively.

(4) In Kunming, the potential of PCM for cold storage is great. However, due to the low temperature and the low air conditioning cooling load, the thermal comfort environment can be reached by fresh air supply directly without cold storage.

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EXPERIMENTAL STUDY OF SOLAR LOW-TEMPERATURE WATER HEATING SYSTEM WITH SHAPE-STABILIZED PCM PANELS

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ABSTRACT

Shape-stabilized phase change material (PCM) has large apparent specific heat for phase change temperature region, it can keep shape stabilized in the phase change process and no need for containers. Taking solar low-temperature hot water as the heat source, we set up the floor heating system with shape-stabilized PCM panels and dry-pipe floor heating system. Two systems performance of storing and releasing thermal energy and its influence on thermal comfort was compared. Phase change energy storage heating room minimum temperature increased 2-3°C while temperature fluctuation reduced 3.5°C compared dry-pipe floor heating system room. The use of phase change materials expands new ideas on how to improve the utilization efficiency of solar thermal

Keywords: Shape-stabilized PCM, floor heating system, thermal energy heat storage, temperature fluctuation.

1. INTRODUCTION

In buildings, a major portion of the energy is used by heating and cooling applications and it is estimated that 10–20% of this energy is used by HVAC equipment [1]. Global concern of the environmental impacts of the fossil fuel usage has prompted the interest to search and use passive techniques for heating and cooling of the buildings [2]. Increases in building energy demands and energy costs have caused people to seek alternate cheaper, renewable energy sources for the operation of buildings. As one of renewable energy sources, solar energy as heat can be used as the energy source for building cooling, heating, and ventilation to conserve energy, as well as to protect the environment by avoiding pollutant and CO₂ emissions associated with the generation of electric power and the burning of natural gas.

Abundant solar energy resources are widely distributed, clean, non-polluting. It is difficult to use due to its instability and intermittent. It is available only during the day, and hence, its application requires an efficient thermal energy storage so that the excess heat collected during sunshine hours may be stored for later use during the night. Latent heat storage method provides much higher storage density, with a smaller temperature difference between storing and releasing heat. Phase change materials (PCM) can provide large latent heat storage over the narrow range of temperature typically encountered in buildings. Application of the phase change energy storage technology is able to

achieve the purpose of energy saving or the transfer of load.

Helmut conducted a research on PCM-facade-panel for daylighting and room heating, compared to a double glazing without PCM, a facade panel with PCM shows about 30% less heat losses in south oriented facades. Solar heat gains are also reduced by about 50% [3].

K.Nagano proposed a new floor supply air conditioning system, using phase change material to augment building mass thermal storage, results from measurements simulating an air conditioning schedule in office buildings indicate that 89% of daily cooling load could be stored each night in a system that used a 30 mm thick packed bed of the granular PCM [4].

Xiao Wei think up a system that air solar collector combined with shape-stabilized PCM, air is heated by solar collector and transported to PCES floor mezzanine through pipe insulation, complete process of heating and PCM heat storage, room air temperature can be increased due to the exothermic of PCM, reduce heating energy consumption [5].

There are several combination forms of PCM and building: (1) Composite PCM to the building envelope directly, such as PCM wallboard, PCM ceiling etc. (2) Combined with the radiant floor heating system laying on the floor layer. (3) Used in regenerative heating, air conditioning system in the building.

NOMENCLATURE

Abbreviation		Subscripts	
PCM	phase change material	-s	supply
DN	Nominal diameter	-b	backwater
Symbols			
C_p	Specific heat ($\text{kJ kg}^{-1}\text{°C}^{-1}$)	T	Temperature(°C)
M	Mass flow(kg s^{-1})		

In comparison, radiant heating has a number of advantages over convective air heating systems. It is possible to achieve comfortable conditions with a low water temperature by a large floor-heating surface, which is resulting in reduced infiltration heat losses. Floor radiant systems are cleaner than more conventional ones because movement of dust is diminished.

Taking solar hot water as the heat source, we set up the phase change energy storage floor heating system and dry-pipe floor heating system. Characteristics of storing and releasing thermal energy and its influence on thermal comfort of the room were tested.

2. EXPERIMENTAL SYSTEM

2.1 Thermal performance of PCM

Shape-stabilized phase change material is a kind of novel PCM. It has the following salient features: large apparent specific heat for phase change temperature region, suitable thermal conductivity, keeping shape stabilized in the phase change process and no need for containers [6].

A kind of shape-stabilized PCM panel designed by DuPont was utilized in floor heating system. The panel is a fine mixture of ethylene based polymer (40%) and paraffin wax (60%) laminated on both sides with a 100 μm aluminium sheet. The edges are closed with a 75 μm aluminium tape. Fig.1 shows the appearance of a shape-stabilized PCM panel. Its thermal physical properties were measured and presented in Table 1. Shape-stabilized PCM panel is laid on the floor layers directly, so that it is in touch with heat transfer medium, reduce heat losses. We calculated the released heat of the floor layer according to thermal physical properties of PCM (Tab.1). Heat of per unit area of phase change material is about 450KJ, if we assume that in the stable heating cycles of heating for 12h and releasing for 12h, we can estimate exothermic capacity of phase change energy storage floor layers, 10.42W/m².



Figure 1 Appearance of the PCM panel

Table 1 Thermal physical properties of PCM

Properties	Test method	Unit	Value
Thickness		mm	5.2
Area weight		Kg/m ²	4.5
Melt point(paraffin)	DSC method	°C	21.7
Latent heat storage capacity	DSC method	KJ/kg	>70
Total heat storage capacity	DSC method	KJ/kg	140
Conductivity (solid/liquid)	BS EN 12667-2001	W/(m.k)	0.18/0.14

2.2 Experimental investigation of system

2.2.1 Description of the heating system

In order to investigate the thermal performance of two different types of floor heating system, an experimental house with solar low-temperature water system was setup in Beijing, China.

The experimental house was equipped with phase change energy storage floor heating system and dry-pipe floor heating system respectively. The dimensions of the experimental house were 4.6 m (depth) ×4 m (width) ×3 m (height). The roof and walls were made of 150mm-thick polystyrene foam board wrapped by metal board. It had a triple glazing windows facing south, covered by light curtain.

The under-floor heating system included 150 mm-thick polystyrene insulation, reflective film, DN15 pipes embedded in the extruded plate, aluminum, single layer of PCM and the ceramic tile layer. The experimental room area is relatively small, therefore, heating pipe is arranged into S with 200mm tube pitch. Fig. 2 shows photos of the hot water heating system. Fig. 3 shows shape-stabilized PCM panels. Fig. 4 illustrates the structure of the floor heating system.



Figure 2 Hot water heating system



Figure 3 Shape-stabilized PCM panels

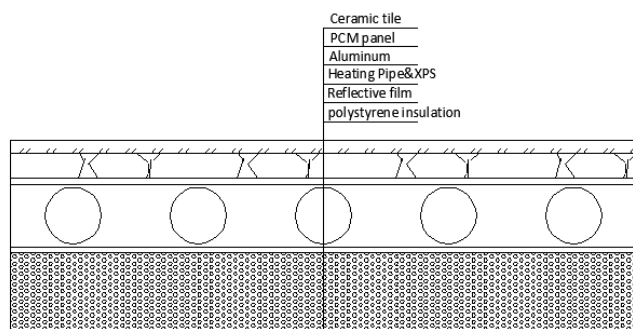


Figure 4 Schematic of floor heating system

2.2.3 Control of system

There are two major aspects for the system control.

1. Indoor temperature control. Measure room temperature, stop heat pump operation as well as heating cycle when temperature is higher than 22 °C. If the room temperature is below 18 °C, the heating cycle restart, ensure reasonable room temperature to prevent overheating of the indoor.

2. Control system for building solar collecting. When temperature difference of the collector endpoint temperature and tank is more than 5°C, the temperature difference controller gives the signal to start the pump. Solar heat is collected into the water tank. When the intensity of sunshine weakens, temperature difference is less than 3°C, system stops running.

Water level sensors are installed into the hot water tank, the water level sensor is disconnected when the tank water level is lower than the predefined value. The total circuit is cut off to prevent pump cavitation. Gas

left in the water system is drained away through the exhaust valve, electric wire and insulation are attached on pipes external surfaces to prevent freezing of water system under low temperature.

2.2.2 Calculation of heating load and Test instrumentation

The experiments aimed at determining indoor temperature, supply-return water temperature and flow, temperature and heat flux of floor surface, and surrounding wall temperature. The following apparatus were applied for measuring.

Table 2 Instruments and their functions

Instrument	Test Object
HP-34970A dataloggers	Record data
Heat flow meter	Measure heat flow
Temperature controller	Control on-off of pump
Glass rotor flow meter	Test flow of water system
Thermocouple temperature transducer	Measure temperature

3. RESULTS AND DISCUSSIONS

Beijing climate is typical warm temperate semi-humid continental monsoon, hot and rainy in summer, cold and dry in winter. The heating time in Beijing is from 15 November to 15 March. The test of the experimental system was carried out in February.

The key to testing system works is the all-day indoor air temperature fluctuation. Operation of pumps was controlled by the automatic control system according to the set control strategy. System debugging had been done in the first day of test. Analyze data in 14-16 February. Fig.5 shows average indoor temperature both phase change energy storage floor heating system and dry-pipe floor heating system. Temperature fluctuation in dry-pipe floor heating system room was 11.5°C. Average indoor temperature in phase-change heating room was 8°C, minimum temperature increased 2-3°C while temperature fluctuation reduced 3.5°C compared dry-pipe floor heating system room.

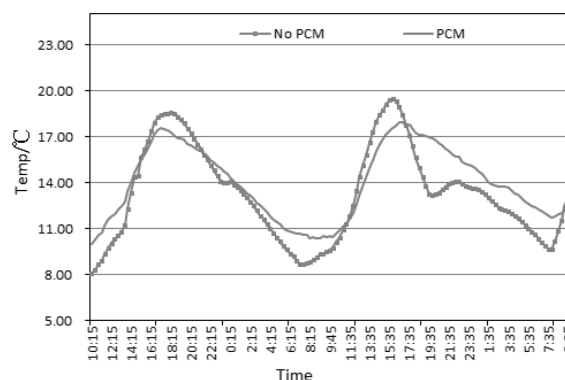


Figure 5 Contrast curve of indoor temperature

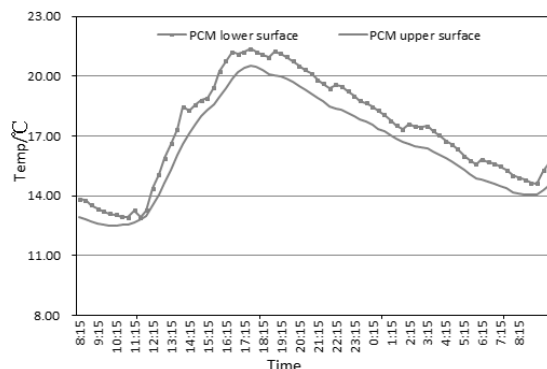


Figure 6 Temperature change of the PCM upper and lower surfaces

Fig.6 reflected temperature change of the phase change material upper and lower surfaces. Temperature of the phase change material lower surface reached the melting temperature and began to store energy with the rise of the supply water temperature at 14:00. Now, phase-change heating room temperature fluctuation is more smoothly than dry-pipe floor heating system room as heat stored. As the decreased of supply water temperature, temperature of the phase change material surface dropped to melting temperature, phase change materials solidified and temperature changes slowly. Cooling rate of phase-change heating room is significantly smaller than the other room. Transformation temperature of shape-stabilized PCM is relatively low as well as supply water. It led to that temperature of phase change material only reach the melting temperature in a short time, This shows there is still a part of the phase change material was not fully utilized, material utilization needed to be improved.

Supply and return water temperature difference of floor heating system remained between 1.5°C to 3.0°C basically. Flow of supply and return water was 370L/h, we can calculate the heat supply of the day based Formula (1).

$$Q = \sum C_p M (T_s - T_b) \quad (1)$$

Heat supply of floor heating system is 93240 kJ, average value of the day is 1080W.

For testing the process of phase change material stored and released thermal energy. Heat flow meter were affixed to the material surface, calculate the thermal energy of PCM (Fig.7). During 10:05 on 14th to 10:05 on 16th, stored heat of phase change energy storage floor heating system was 22134kJ, released heat was 11203kj. If we can further improve the insulation of floor layer, it may achieve better results. We can estimate utilization across comparing actual cumulative value of heat supply and calculated. The ratio is 73.2%. This result is consistent with previous (Fig.7) results, shows that the phase change material has not been fully functional.

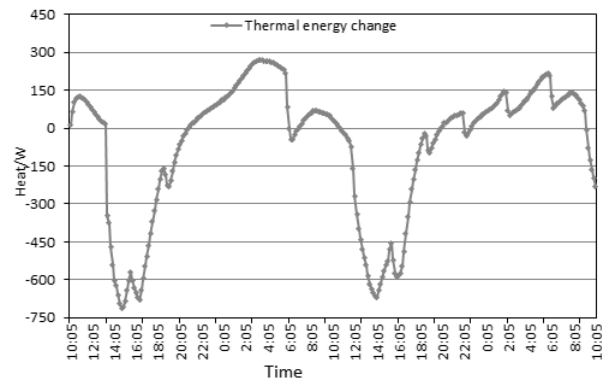


Figure 7 Measured PCM thermal energy change

3. CONCLUSIONS

Since manufacturers supplied the phase change material consumption could not achieved the desired requirements, phase change thermal storage floor heating system was covered with a layer of phase change material, we also set up the tank to storage heat auxiliary, this combination also achieved good results. The following conclusions can be made:

1. We put forward a new structure of solar low-temperature water heating system with shape-stabilized PCM panels in this paper. The experimental results testified the feasibility and good performance for the heating system under adding any auxiliary energy.

2. Phase change energy storage floor heating system minimum room temperature increased 2-3°C while temperature fluctuation reduced 3.5°C compared dry-pipe floor heating system room.

3. Shape-stabilized PCM can keep shape stabilized in the phase change process and no need for containers, therefore, it is easy to combine with building envelope, if we choose PCM has proper phase transition temperature, this system maybe transform the radiant floor cooling system in the summer.

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MICROBIAL CONTAMINATION RESEARCH AND ANALYSIS OF CENTRAL AIR CONDITIONING TREATMENT UNIT

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Abstract:

With more and more people's requirements increase in building environment comfort, the more use of central air conditioning system in public buildings. Improper operation and management of the central air conditioning system will not only make the blooms of microbial contaminants inside the air-conditioning system, air handling units and recycling of delivery, return air pollute the air-conditioned environment, but also affects the quality of people's lives and health. In this paper, three typical public buildings air handling units in Shenyang are tested, analysis of the types of microbial growth in air handling units; Research and analysis of the relationship between the environment temperature, humidity and microbial contaminants in air handling units and correlation analysis of three test air processing unit fungal and bacterial pollution is given. According to the air handling unit internal structure and test results, putting forward relevant microbial contamination treatment technology and control method, the microbial source of air conditioning system is effectively prevented and controlled, thus providing clean and safe air quality for air conditioning environment.

Key words: air handling units; public buildings; microbe; pollution

1. INTRODUCTION

With society's progress and development, central air conditioning has become indispensable equipment in modern public buildings. Bring comfortable environment also produced many negative effects. Microbial contamination is an important factor that influences the quality of indoor air. Due to the human body, room, air handling units in the room forming a closed circulation system, air handling unit microbial contamination occurs will greatly influence on indoor environment and human health^[1]. And micro-organisms in the appropriate temperature and humidity environment easy blooms^[2], which is an important factor to cause various diseases. Pasanen^[4] found that fungi can be multiplied in the wet interior surface in a week. In recent years, some unexpected events (such as the 2003, SARS, etc.) raise awareness of the seriousness and urgency of microbial contamination detection and control in the central air conditioning system.

2. BASIC CONCEPTS OF THE MICROORGANISMS

2.1 CLASSIFICATION OF MICROORGANISMS

A wide variety of microorganisms do mostly harmless to humans and even benefit. But there are also some do affect people's lives. The microbial biological characteristics can be divided into: virus, chlamydia, rickettsia, mycoplasma, bacteria, actinomycetes, fungi, algae and protozoa. According to their living habits,

chemical composition and structure, they can be divided into three categories. First, the higher the degree of differentiation of the nucleus of eukaryotic microbial cells, it has nuclear membrane, nucleolus and chromosome; complete organelles in the cytoplasm. Fungi belong to this type of microorganisms. Second, the low degree of differentiation of prokaryotic microbial cells nucleus, only the original nucleus and cytoplasm, nuclear membrane and nucleolus; organelles is not very perfect. Such a wide variety of microorganisms include bacteria, spirochetes and mycoplasma, rickettsia, chlamydia and actinomycetes bacteria. Third, the non-microbial cells neither have typical cell structure nor enzyme system to produce energy, growth and reproduction only in living cells. The virus belongs to this type of microorganisms.

2.2 MICROBIAL CHARACTERISTICS

Microbial individual is small, wide variety, widely distributed and a huge number. But microorganisms have mutated, strong ability to adapt. Most exist in the way of Single-cell which leads an independent life, and metabolism is strong, grow faster, quantity as well as direct contact and the external environment, even if the mutation frequency is very low, in a short period of time also can produce a large number of variants of offspring. Microorganisms are ubiquitous in nature which can spread by wind and water and be invisible to the naked eye. Microbial small size, simple structure, but still has a common feature of the metabolism, growth, reproduction and genetic variation of life activities. Harmful microbes can cause a range of diseases of the people. Common diseases are mainly the

following four: tuberculosis, Legionella pneumonia, allergic reactions, hypersensitivity pneumonitis.

3. MICROBIAL SOURCES AND TRANSMISSION IN AIR HANDLING UNIT

3.1 MICROBIAL SOURCES

Microbial sources in the air handling unit has the characteristics of diversity, which can be divided into internal sources and external sources of air handling units^[3].

(1) External sources

A large number of industrial emissions and automobile exhaust emissions caused outdoor atmospheric pollution, the destruction on the ecological environment, which leads to the emergence of global air warming, dust storms and other inclement weather. Providing a suitable living environment for micro-organisms so there are a large number of non-pathogenic spoilage microorganisms in the air. Unclean outdoor air is used as a new wind in unit, containing a large number of micro-organisms particles into the unit which becomes one of the main sources of the pollutants. Modern architecture generally do not use full fresh air system, and directly use the return air in the room after handling units processing then sent, so that the individual room is high air pollution levels through the return air which is polluted caused by the

introduction of the unit.

(2) internal sources

Surface cooler and condensate tray in air handling units provide a moist environment, the local dust contained in the microbes in this environment rapid propagation caused microbial contamination inside the unit. Imperfect Operation and management also contributed to the pollution of the important reasons. If the lack of air conditioning duct cleaning and maintenance, leading to the central air conditioning system provide channel and power for the dissemination of pollutants.

3.2 ROUTE OF TRANSMISSION

People's production activities may make the air in the presence of pathogenic micro-organisms, including microorganisms, mainly in the air as a medium for the spread of the disease. Some microorganisms from the human body, especially in poorly ventilated crowded environment, there will be more microorganisms present. The microorganisms need to be attached to the dust, larger dust can fall quickly, as people's activities or dissemination of the air flow. In one's speech, cough, sneeze droplets formed, which contains a large number of micro-organisms. Droplet transmission has important significance in public health place.

4. DETECTION OF POLLUTION IN AIR HANDLING UNITS

4.1 PROFILE OF DETECTION OBJECT IN THE AREA

Shenyang is located in the southern part of Northeast China and is a cold region. Detection on three air handling units was a university office in Shenyang, a large museum in Shenyang and heating or cooling of a commercial center in Shenyang. Office building adopts water source heat pump air conditioning system to cool, office space most adopt fan-coil unit plus fresh air; Meeting rooms and multi-function hall adopts full fresh air system. Heat and Moisture environment in the museum have stringent requirements, showrooms and painting area showcase use two different sets of air-conditioning system to control the indoor environment. Business center, six floors above integrated shopping mall uses constant air volume air systems with a return air, air heating is used in underground garage and equipment room.

4.2 AIR HANDLING EQUIPMENT AND MEASURING POINTS

The order of air handling units' common combinations is new wind segment, early effect, return air section, table cold section, fan section. The air-conditioned environment cleanliness requirements are relatively high unit also creates a high efficiency filter section, sub-high efficiency filter section or high efficiency filter section. In order to comprehensive understand air handling units microbial contamination situation and grasp the source of microbial contamination, types, composition, sampling measuring point 1 is disposed in the air handling units in new wind and back wind hybrid filter segment which detect concentration of microorganisms which is introduced by air handling units; sampling points 2 arranged in the air through the table after the cold period, the detection processing unit internal breeding.

4.3 Microbial detection

Detection of microorganisms in the air is generally natural sedimentation and air impact. Natural sedimentation method is the most economic, most simple, the most effective way. This experiment uses the impact method for air sampling; The bacteria and fungi medium melt in the sterile glass plate with a diameter of 90 mm which is made into tablets, tablets will do a good job number in impactor sampler, put impactor sampling point selected, the export of the impactor with rubber hose is connected to a host of air inlet. After sampling, remove the cover plate and mark, culture counts.

4.4 Test results and analysis

Dust in the air handling unit by wiping method for acquisition, 50 cm² area around the point set as sample area, sampling completed back to laboratory testing. Of dust detection should be carried out in air conditioning unit equipment surface, and the central air conditioning system is to stop state. Table 4.1 for summer conditions, of each measuring point measured the amount of dust.

Test results show that the monitoring dust amount does not exceed national standards (20 g/m² or less dust quantity) minimum amount of dust in the air handling

unit is the office building, followed by museums, commercial center is the largest.

After investigation and analysis form the reason, we get the following three situations: (1) building the geographical position is different. (2) the building use function is different. (3) the level of clean air conditioning system.

Take the sample for testing analysis, we can get the distribution of bacterial and fungal particles in the air handling unit graph, Accompanying the bacteria staphylococcus aureus showed that dust and air superiority strains (figure 4.1 in figure 4.2); Dust is penicillium genus (figure 4.3), the advantage of fungi in the air bacteria branch spore mildew genera (figure 4.4).

Table 4.1 Dust sampling of three air processing unit sampling point (g/m²)

		A ₁	B ₁	B ₂	B ₃	C ₁	C ₂
Point 1	first time	3.82	9.03	13.84	12.21	13.35	16.64
	second time	4.33	9.49	14.57	12.95	14.92	17.38
	third time	4.76	10.11	15.68	13.39	15.87	17.87
Point 2	first time	5.62	10.62	16.54	14.03	17.07	18.53
	second time	6.19	11.16	17.46	14.62	18.81	19.14
	third time	7.07	11.70	18.38	15.21	19.13	19.76

Note: A Represents office B On behalf of the Museum C Represents the business center

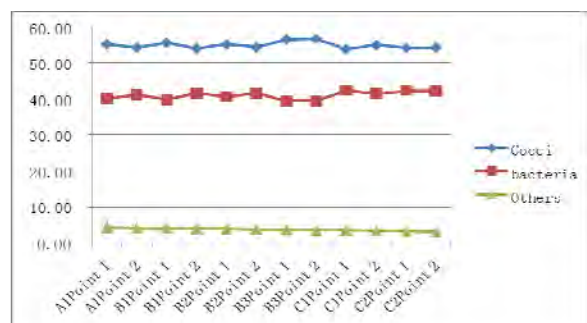


Figure 4.1 Air handling unit of air bacteria particles distribution curve

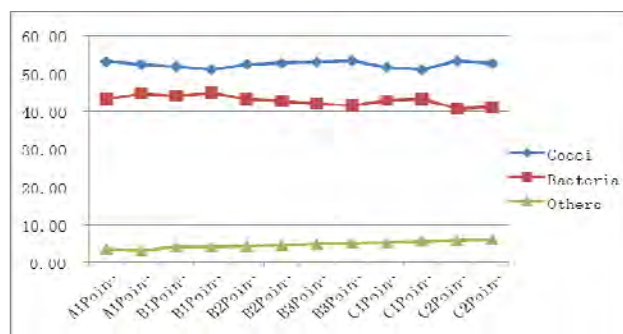


Figure 4.2 Bacteria in air handling units dust particle distribution curve

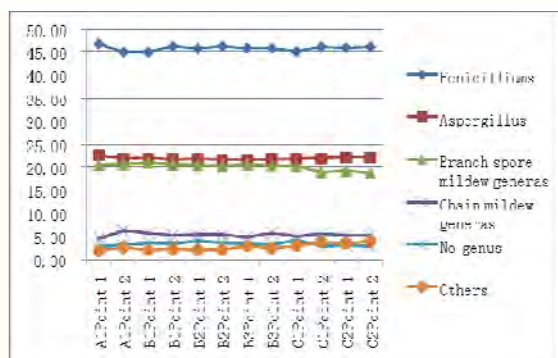


Figure 4.3 Air handling unit dust fungal particles distribution curve

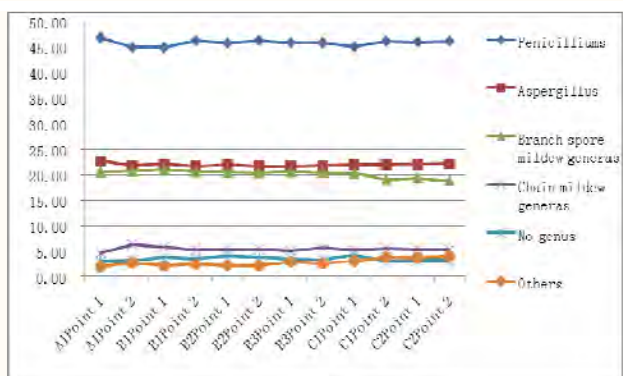


Figure 4.4 Air handling unit air fungal particles distribution curve

4.5 The effect of temperature and humidity on the microorganism concentration

Test analysis of the layout of sampling points table 4.5 and table 4.6 data are obtained.

Through analysis, microorganism concentration and temperature and humidity basic relation follows the logarithmic equation. By the fitting get the relation between temperature and bacteria, the relation between fungi; Humidity, fungi and bacteria.

Temperature and the concentration of bacteria fitting relationship:

$$y = 419.19 \ln(x_1) - 1218.9 \quad R^2 = 0.8785 \quad (3.1)$$

Temperature relationship with fungi concentration fitting:

$$y = 537.96 \ln(x_1) - 1539 \quad R^2 = 0.78 \quad (3.2)$$

Humidity and fungi simulated concentration close relationship:

$$y = 220.77 \ln(x_2) - 876.16 \quad R^2 = 0.8625 \quad (3.3)$$

Humidity and fungus in the concentration of fitting relationship:

$$y = 161.21 \ln(x_2) - 613.52 \quad R^2 = 0.95 \quad (3.4)$$

In the formula:

y-Microbe concentration

x_1 -The temperature

x_2 -The humidity

R^2 -The correlation coefficient

According to curve with the increase of temperature, humidity, microbe concentration is larger

4.6 Air handling units in microbial control

We have developed a lot of methods about the control of Harmful microbial in air conditioning systems, To do the best control effect, we should take prevention first to prevent microbial contamination in the air handling units, infested by sterilization method, the principle of comprehensive and effective in microbial control of air handling unit processing. Now Tianjin university Tianshuyuan [5] has drawn: the effect of air filtration material filtration sterilization capacity is more than 90%, the high efficiency filter is more effective and stronger in controlling pollutants ability, ShenJinming [6] [7] a professor of Tongji university has proposed the humidity control priority. The above method is prevention of microbial, but short of sterilization effect. Catalyst is a light in the reaction of the catalyst. Using photocatalytic sterilization is an effective way of sterilizing, In order to reach the goal of prevention and cure is put forward after the first select

the filtering material and control the humidity by using the method of photocatalytic sterilization. With efficient filter material (or control humidity by photocatalytic sterilization, prevention first, then sterilization

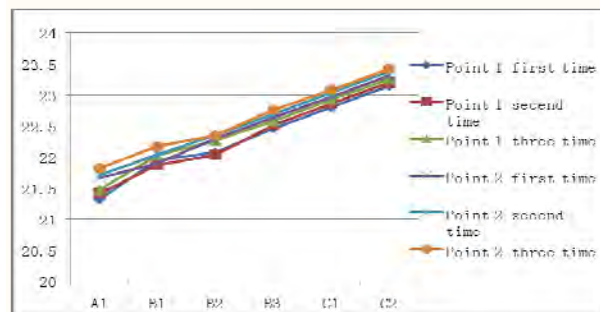


Table 4.5 Sampling point temperature change curve

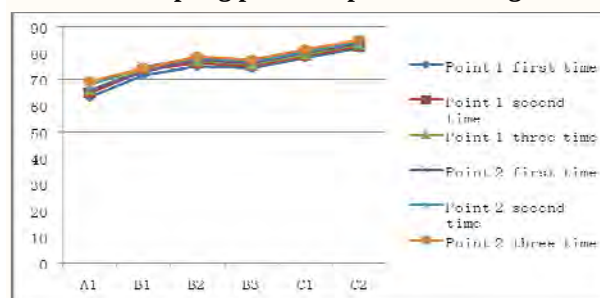


Table 4.6 Sample point humidity change curve

5. CONCLUSIONS

Tested analysis that the amount of dust in business center processing unit is the largest, because the personnel density and liquidity within the mall is very large, Airhandling unit dust accumulated inside so fast that it can't be clear in time. Office personnel density and liquidity is small and the timely processing of the clean-up crew so the amount of dust is the least, museum is between the two, but three processing units found in bacteria, fungi strains contained content has the same rule. And air handling units contained within different microorganisms exist obvious logarithmic correlation between concentration and temperature and humidity.

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EXPERIMENTAL TESTING OF A SCROLL EXPANDER IN AN ORGANIC RANKINE CYCLE

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ABSTRACT

This paper presents the results of an experimental study of a scroll expander integrated into an Organic Rankine cycle (ORC), whose working fluid was R123. The built test rig mainly consisted of an evaporator, a scroll expander, a pump, and a condenser. The maximum heat source temperature was 250°C. And the scroll expander was modified from a scroll compressor used in automotive air-conditioning. The output electric power and isentropic efficiency of the expander under different operating conditions were measured and analyzed. Maximum output electric power of 2558W was measured, and maximum isentropic efficiency achieved 21.6%.

Keywords: scroll expander, ORC, Isentropic efficiency, heat recovery

1. INTRODUCTION

Organic Rankine Cycle (ORC) system as one of the solution to utilize moderate-to-low grade heat for power generation has gained more and more attention against the background of energy crisis and global warming. ORC system shows dramatic potential in utilization of industrial waste heat, solar energy, geothermal energy and biomass energy[1].

ORC system which mainly consists of evaporator, condenser, expander (turbine) and pump converts heat to power with low-boiling organic working fluids. The expander as one of the critical components plays a key role in overall efficiency and stability of the system. Output power and overall efficiency exhibit linear variation along with isentropic efficiency of the expander[2].

Scroll expanders as a positive displacement device have definite advantages such as the reduced number of parts, the ability to cope with two-phase working fluids, low rotation speed, the stability and the inherent reliability proven in compressor[3]. With above characteristics scroll expanders are much suitable for small scale ORC system. However there are few scroll expanders specially designed for ORC systems to date. Most scroll expanders are derived from scroll compressors used in air compression or HVAC system. Experiment is the most popular method in the investigation of scroll expanders which employs organic refrigerants, steam or compressed air as working fluids[3][4][5]. Among them organic refrigerants are dominant.

Kim et al.[4] designed a double-side scroll expander and conducted an experimental testing in a low-temperature Rankine Cycle, recording its maximum output power of 12kW and maximum isentropic efficiency of 33.8%. Mathias et al.[6]

investigated experimentally the performance of scroll expanders in an ORC system with R123 as working fluid. They got the conclusion that the scroll expander was a good candidate for ORC system. The maximum output power of 3kW and isentropic efficiency of 83% were achieved. Lermort et al.[7] built a gas cycle test rig for a hermetic scroll expander with working fluid of R245fa. Maximum overall isentropic efficiency of 71.03% was measured. And they also investigated the impact of the lubricating oil quantity on isentropic efficiency.

This paper presents an experimental investigation of a scroll expander in an ORC system with working fluid of R123. The evolution of the performance of the machine under different operating conditions was presented, which could provide reference for selection of scroll expanders in ORC system design and operation.

2. SCROLL EXPANDER

2.1 structure and principle

The scroll expander is made up of two involutes curves in opposing directions which called orbiting and fixing scrolls. The orbiting scroll revolves around a fixed shaft center without any rotation around itself. Such assembly of the scrolls form several separate chambers. As the orbiting scroll moves, the chambers increase in size and the fluid in the chambers expands. Figure 1 illustrates the expansion process of a scroll expander. In Figure 1, frames 1-3 show the charging process. The high pressure vapor enters the suction port. The second expansion chamber forms in frame 4 and then the working fluid expands with the volume increasing from frame 5 to 8. At the end of the expansion the low pressure is exhausted as shown in frame 9.

NOMENCLATURE

Abbreviation		T	Temperature (°C)
ORC	Organic Rankine cycle	W	Output electric power (W)
Symbols		Subscripts	
V	Volume of scroll chamber (ml)	-i	inlet
P	Pitch of scroll wrap (mm)	-o	outlet
H	Wrap height (mm)	-s	isentropic
N	Number of spiral revolutions	Greek Letters	
t	Thickness of the scroll wraps (mm)	θ	Rotation angle (rad)
s	Entropy (kJ/(kg • K))	θ^*	Initial expansion angle (rad)
h	Enthalpy (kJ/kg)	η	Efficiency
p	Pressure (MPa)	α	Initial angle of involutes (rad)

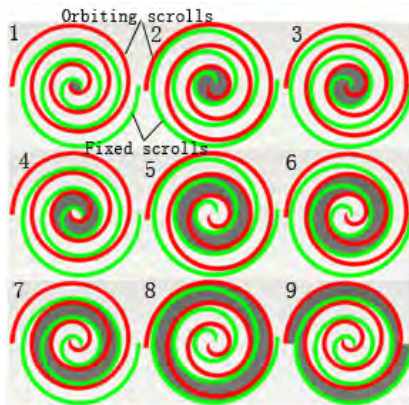


Figure 1 Expansion process of a scroll expander

2.2 Parameters

The main geometry parameters of the scroll wraps are presented in Table 1.

Table 1 Geometry parameters of the scroll wraps

Base circle radius	a
Pitches of scroll wraps	$P=2\pi a$
Wrap height	H
Initial angle of involutes	α
Thickness of the scroll wraps	$t=2a\alpha$
Initial expansion angle	θ^*
Number of spiral revolutions	$N+1/4$

In the case of the involutes of circle, the volume of separate chambers can be described as a function of rotation angle. From the first expansion chamber to the discharge chamber, the volume can be calculated by equation (1), in which “i” is the order number of the chambers with a range from 2 to N and “ θ ” is the rotation angle with a range from 0 to 2π .

$$V_i = \pi P(P - 2t)h \left[2(i-1) - 1 + \frac{\theta}{\pi} \right] \quad (1)$$

The volume of initial expansion chamber can be calculated by equation (2).

$$V_2 = \pi P(P - 2t)h \left(1 + \frac{\theta^*}{\pi} \right) \quad (2)$$

The volume of the outer chamber at the end of the expansion can be calculated by equation (3).

$$V_N = \pi P(P - 2t)h(2N - 1) \quad (3)$$

Then the built-volume ratio of the scroll expander can be described by equation (4).

$$\nu = \frac{V_s}{V_0} = \frac{2N - 1}{1 + \theta^* / \pi} \quad (4)$$

3. EXPERIMENTAL INVESTIGATION

3.1 Description of the ORC test rig

A 3kW ORC test rig was built to investigate the performance of the scroll expander, which mainly consisted of a fin and tube evaporator, a scroll expander, a plunger pump, a shell and tube condenser, a cooling water pump, a cooling tower, a natural gas burner and an asynchronous generator. The test rig is schematically described in Figure 2. The natural gas burner was used to simulate the low-temperature heat source whose temperature was controlled around 250 °C. And the cooling water from a cooling tower was employed as the cooling source whose temperature depends on the ambient temperature. R123 was selected as working fluid for the test rig which is dry fluid and shows good cycle efficiency with the temperature range[3][11].

The heat source for the experiment was the hot gas from the natural gas burner. A finned-tube heat exchanger with 0.952cm diameter copper tubes and aluminum was selected as the evaporator.

Scroll expander

The scroll expander was modified from an automobile air conditioning scroll compressor with a built-in volume ratio of 1.91.

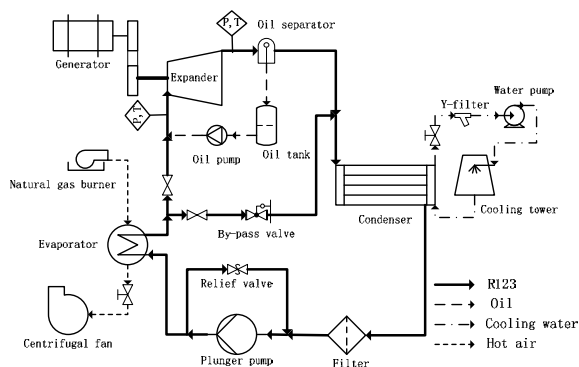


Figure 2 Schematic representation of the ORC test rig
Evaporator

Working fluid pump

The working fluid pump was an axial plunger pump. The range of flow rate of the pump was 0-126L/h, and the maximum output pressure was 2MPa. The plunger contacts with the working fluid directly under high pressure which may lead to leakage. The initial test rig encountered this problem which was resolved by replacing a proper plunger.

Condenser

The condenser was a shell and tube heat exchanger. The vapor condensed in the shell side. And cooling water flowed through the tube side. The volume of shell side was large, which can store some liquid working as a reservoir to maintain a small degree of subcooling.

3.2 Measurement devices

The measurement devices included temperature transducers, pressure transducers and output electric power meter. Those measured data were collected once per second.

The temperatures were measured by K-thermocouples which have an uncertainty of $\pm 0.1K$ and a range of 300~1280K. The pressures were measured with silicon absolute pressure transducers which have uncertainty of $\pm 0.2\%$ of the span. And the pressure transducers have a range of 0.3~1.0MPa. The electric power was measured with a watt transducer, which was installed between the electric generator and power consuming devices, i.e. lighting bulbs.

4. DATA REDUCTION

The collected data including temperatures and pressures were used to determine the state properties. NIST RefProp 8 software was used to calculate the enthalpy and entropy from the temperatures and the pressures measured at the inlet and outlet of the scroll expander as following.

$$h_i = \text{enthalpy}(p_i, T_i) \quad (5)$$

$$s_i = \text{entropy}(p_i, T_i) \quad (6)$$

$$h_o = \text{enthalpy}(p_o, T_o) \quad (7)$$

$$h_{os} = \text{enthalpy}(p_i, s_i) \quad (8)$$

These properties were used to determine the isentropic efficiency of the expander as following.

$$\eta_s = \frac{h_i - h_o}{h_i - h_{o,s}} \quad (9)$$

5. RESULTS AND DISCUSSION

In order to evaluate the performance of the scroll expander under a wide range of conditions, the stroke of the plunger pump was changed to control the circulation flow rate of the working fluid. The flow rate of the exhaust hot gas from the burner was changed to control the inlet pressure and temperature of the expander. During the experiment, load resistance was kept constant.

Different inlet and outlet thermodynamic quantities and measured output electric power were obtained under 30% and 50% of the plunger stroke respectively. The ranges of the measured variables are shown in Table 2.

The outlet pressure of the expander is stable with a maximum variation of 0.02MPa. Meanwhile, the suction and discharge process of the scroll expander is not continuous. So the test results focused on the output electric power and isentropic efficiency as a function inlet pressure.

The evolution of output electric power and the isentropic efficiency with the supply pressure under 30% of the plunger stroke is given in Figure 3 and Figure 4. The range of the supply pressure was 0.710-0.839MPa, and the supply temperature was 88.5-125.1 °C. Under such conditions, maximum output electric power of 1857W was achieved. And the output electric power increased with the supply pressure. It can be explained by that the torque and the rotation speed increase with the pressure ratio which led to the increasing output electric power. The isentropic efficiency decreased with the supply pressure due to the under-expansion losses. The maximum isentropic efficiency of 0.216 was achieved.

Figure 5 and Figure 6 show the evolution of output electric power and the isentropic efficiency with the supply pressure of 0.710-0.839MPa under 50% of the plunger stroke. The maximum output electric power of 2558W was measured. And the maximum isentropic efficiency was 0.168. The isentropic efficiency increased with the rotation speed (for leakage decreasing), but decreased with the supply pressure (for under-expansion losses). The combination of the two effects led to the results in Figure 6.

Table 2 Range of the main measured variables

Plunger stroke	Range	p_i	T_i	p_o	T_o	W	η_s
30%	min	0.515	88.5	0.07	48.8	1014	0.076
	max	0.620	125.1	0.09	110.9	1857	0.216
50%	min	0.710	102.4	0.11	68.4	2003	0.107
	max	0.839	128.6	0.13	109.9	2558	0.168

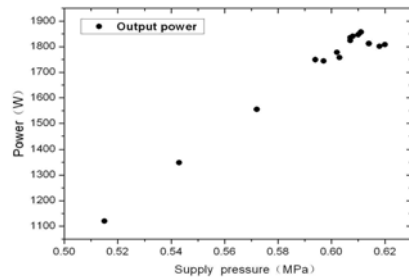


Figure 3 Output electric power versus supply pressure under 30% of the plunger stroke

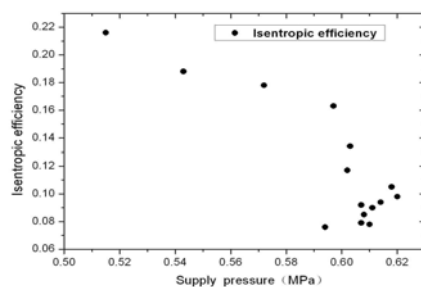


Figure 4 Isentropic efficiency versus supply pressure under 30% of the plunger stroke

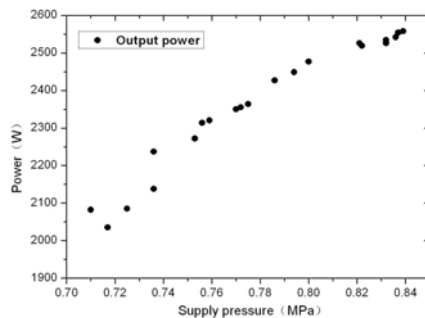


Figure 5 Output electric power versus supply pressure under 50% of the plunger stroke

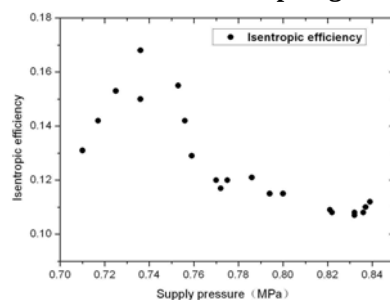


Figure 6 Isentropic efficiency versus supply pressure under 50% of the plunger stroke

The isentropic efficiency measured was relatively lower than the results of other papers. One reason might be that the pressure ratio of from inlet to outlet was too high, which lead to the huge under-expansion losses. Another reason might be the excessive lubricating oil, which has been confirmed having negative effect on isentropic efficiency [7]. So the test rig should be modified in the future investigation.

6. CONCLUSIONS

An ORC system was built to test the performance of the scroll expander with working fluid R123. The maximum output electric power of the generator was 2558W. The maximum isentropic efficiency was 21.6%, which is lower than the results of other papers. Two possible reasons were analyzed. Therefore, the test rig needed to be modified in further investigation. The experimental results showed that the output electric power increased with the supply pressure, but the isentropic efficiency tended to decrease with the supply pressure under the operating conditions.

ACKNOWLEDGEMENTS

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TEST AND ANALYSIS OF GROUNDWATER SOURCE HEAT PUMP SYSTEMS IN SHENYANG

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ABSTRACT

APEC conference

In order to grasp the application status of ground source heat pump technology in Shenyang and analyze the feasibility and energy-saving of heat pump technology in residential building heating in Shenyang, this paper research and measured the groundwater source heat pump project during winter heating period for a typical residential area, and then analyze test data. The analysis showed that the household indoor temperature throughout the heating season is able to meet the minimum temperature requirements, the coefficient of performance of the heat pump unit is 2.84, and the energy efficiency of the heat pump system is 2.55. Heat pump system runs stable, which conducive to the long-term operation of the system; As the outdoor temperature decreases and the increase of the heat load, changing the flow rate and temperature difference to meet the requirements of the heating indoor temperature; Compared with other heating mode, the heat pump system is both energy-efficient and economical.

Keywords: heat pump; heating coefficient; COP; heating system

1. INTRODUCTION

Shenyang is one of cities chosen for an experiment with promoting the application of renewable energy by the National department of Housing Construction. In Shenyang, the ground source heat pump is widely used for cooling and heating.

In this paper, test the groundwater source heat pump systems in a subdistrict. According to the data and the investigation from the tests, analyze the coefficient of performance and the energy consumption and the operating conditions of the heat pump system.

2. TEST PROJECT OVERVIEW

The test project is 69,000m² for four residential buildings. The area of groundwater source heat pump application reached about 60,000m². The heating period lasts for 152 days, cooling period lasts for 92 days. Domestic hot water is available all year round. There are 3 screw heat-pump units, such as Table 1 below. Each unit is equipped with two compressors. It has 3 NP150/315V-55/4 circulating pumps, used 2, non-variable frequency. There are also 2 MV1805-1/16/E/50-2 complement pumps, 2 L100/145-11/2 domestic hot water circulation pumps, non-variable frequency, used 1. The system totally has eight wells, either as pumping wells or recharge wells.

Table 1 Types of screw heat-pump units.

Types	RHSBW33 0H	RHSBW270 HJ
number	1	2
power rating(Refrigerating)	275.8 KW	166.3 KW
Refrigerating output	1289 KW	1047 KW
power rating(heating)	275.8 KW	225.2 KW
Heating output	1365KW	1097KW

3. TEST PROJECT OVERVIEW

3.1 Test contents and methods

3.1.1 Indoor temperature and humidity

Choose the typical room or area for indoor temperature and humidity testing by a temperature and humidity measuring instrument.

3.1.2 Water temperature of the units

Paste the temperature recorder probe on the bare tube, test import and export water temperature, 10 minutes at a step. Choose the major one to test when test point has a number of units. Test at the same time when 2 units work in rotation.

Subscript

COP_H the coefficient of performance of the heat pump unit

Symbols

Q_H Average heating capacity of the unit during the test,kW

N_i The average input power of the unit during the test,kW

V The average flow of the user-side of the unit,m³/h

Δt_w The differences in temperature between user-side of the water , °C

ρ The average density of hot water,kg/m³

c The constant pressure specific heat of the hot water,4.18kJ/(kg.°C)

Subscripts

COP_{SH} the energy efficiency ratio of the heating heat-pump

Symbols

Q_{SH} The total heating capacity of the system during the test,kW·h

N_i The power consumed by the unit during the test,kW·h

N_j The power consumed by the pump during the test,kW·h

V The average flow rate of the system on the user side,m³/h;

Δt_w The temperature difference at the inlet and outlet of the system on the user side, °C ;

ρ The average density of hot water,kg/m³

c The average specific heat capacity of the hot water,kJ/(kg.°C)

3.1.3 Flow rate

Use the Fushi portable type ultrasonic flowmeter to test unit heat source-side,customer-side and system heat source-side flow rate,one unit interval per 20 seconds,continued for 20 minutes.

3.1.4 Input power

Record unit compressor work on the control panel display,lively.

3.1.5 Outdoor temperature

Use the temperature recorder to test the outdoor temperature continuously,one unit interval per 10 minutes.

3.2 Theory

3.2.1 Performance coefficient of the heating heat pump unit

The equation:

$$COP_H = \frac{Q_H}{N_i} \quad (1)$$

The average heating capacity of the unit:

$$Q = V\rho c\Delta t_w / 3600 \quad (2)$$

3.2.2 Energy efficiency ratio of the heat pump in heating season

The equation:

$$COP_{SH} = \frac{Q_{SH}}{N_i + \sum N_j} \quad (3)$$

The total heating capacity of the system during the test:

$$Q = \sum_i^n q_i \quad (4)$$

$$q = V\rho c\Delta t_w \quad (5)$$

4. Results and analysis

4.1 Results and analysis of the temperature test

4.1.1 The temperature outside

The changes of outdoor temperature from 15th to 16th,January 2012 is shown in Fig 4.1.1.

It can be seen that the outdoor temperature reached the top at 2:00pm,reached the foot at 2:00pm.

4.1.2 The temperature inside

Choose the heating relatively unfavorable position to do a 24-hour continuous monitoring.

The temperature of typical room indoor changes from 15th to 16th,January 2012, Fig 4.1.2. The Fig 4.1.2 shows that the household room temperature as the outdoor temperature fluctuations, volatility is not. Most of the time the indoor temperature is above 18°C which has reached the heating requirements.

4.1.3 Temperature of the heat pump unit

Choose the NO.1 heat pump unit as a test object.Placement of the temperature probe in the out of the condenser and the evaporator.Constantly monitor the temperature of the water out of the heat source-side and the user-side. The temperature of the heat source-side and the user-side inside the unit changes from 15th to 16th,January 2012, Fig 4.1.3.

From the Fig 4.1.3,the highest temperature of the heat source-side inlet is 14.63 °C ,the minimum temperature is 14.49 °C ,the average temperature is 14.56°C.The highest temperature of the heat source-side outlet is 11.14 °C ,the minimum temperature is 9.16 °C ,the average temperature is 10.53 °C .The minimum temperature of the user-side inlet is 42.29°C,the highest temperature is 45.42°C,the average temperature is 44.11°C.The minimum temperature of the user-side outlet is 44.08°C,the highest temperature is 45.42°C,the average temperature is 44.11°C.Based on the data above,temperature of the heat source-side and the user-side inside the unit changes little along with the fluctuation of the outdoor temperature.The difference in temperature of the heat source-side and the user-side is relatively stable.The difference in temperature of heat source-side is almost 4°C,the uer-side is almost 2°C. It means the operation of the heat pump unit is relatively stable.

4.1.4 Temperature of the heat pump system

Placement of the temperature probe in the heat source-side and the user-side.Constantly monitor the

out of the water temperature.The temperature of the heat source-side and the user-side in the system changes from 15th to 16th,January 2012, Fig 4.1.4.

From the Fig 4.1.4,the minimum temperature of the heat source-side inlet is 14.3 °C ,the highest temperature is 14.4 °C ,the average temperature is 14.4 °C .The minimum temperature of the user-side outlet is 44.74°C,the highest temperature is 47.65°C,the average temperature is 46.64°C.

Temperature of the heat source-side and the user-side inside the system changes little along with the fluctuation of the outdoor temperature.The difference in temperature between the heat source-side and the user-side is relatively stable.The temperature of heat source-side is almost 4.6°C,the others is almost 2.1°C. It means the operation of the heat pump system is relatively stable,too.

4.2 Analysis of the Heat pump system energy efficiency

According to the equation (1)and (3),compute coefficient of performance of the heat pump units and the energy efficiency ratio of the heating heat-pump system.Its variation with time is shown in Fig 4.2.1 -2.

It can be seen in Fig 4.2.1-2,in the heat-pump units the lowest COP is 2.79,the highest COP is 2.91,the average COP is 2.84.In the heating heat-pump system,the lowest COP is 2.14,the highest COP is 2.96,the average COP is 2.55.

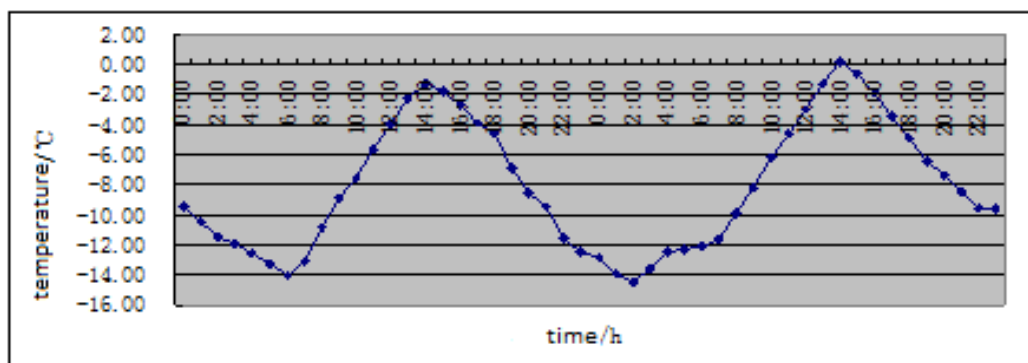


Figure 4.1.1 Outside temperature curve from 15th to 16th, January 2012.

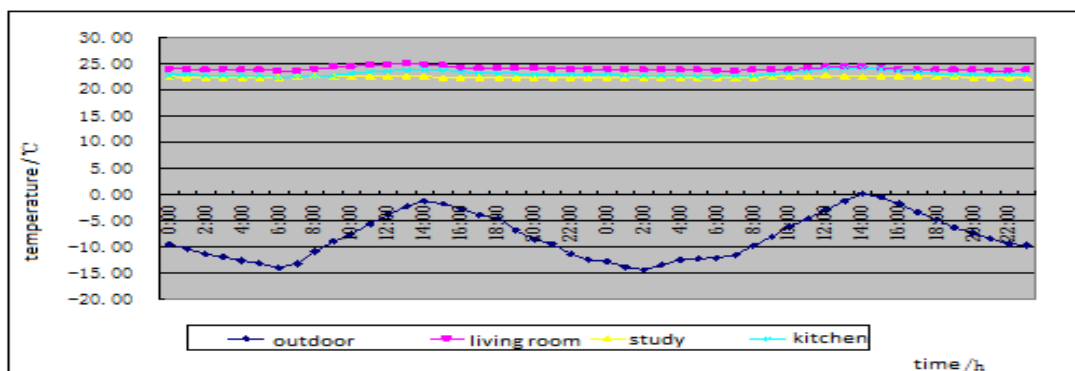


Figure 4.1.2 The temperature of typical room indoor changes from 15th to 16th,January 2012.

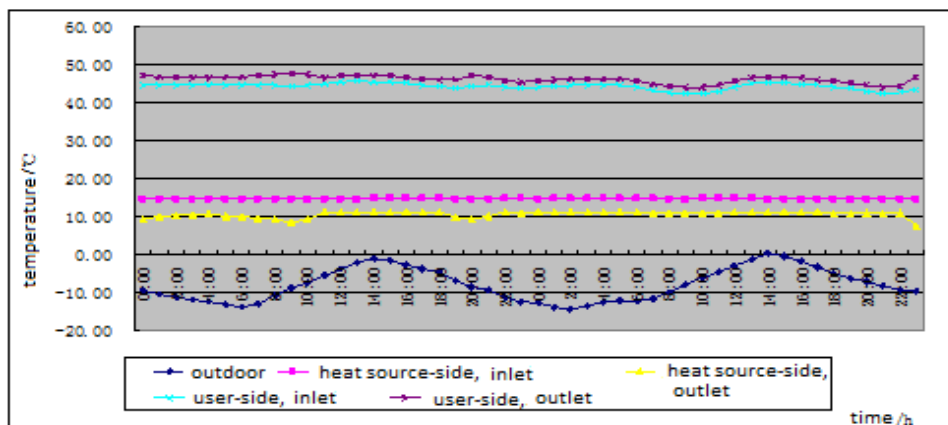


Figure 4.1.3 The temperature of heat source-side and the user-side inside the unit changes from 15th to 16th, January 2012.

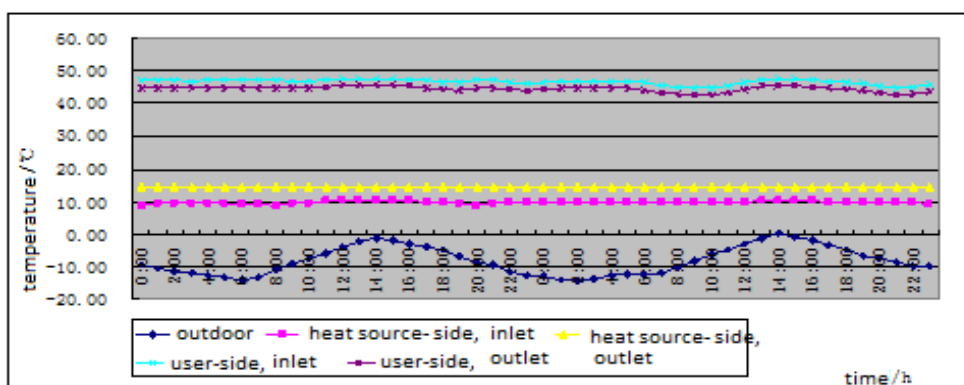


Figure 4.1.4 The temperature of heat source-side and the user-side inside the system changes from 15th to 16th, January 2012.

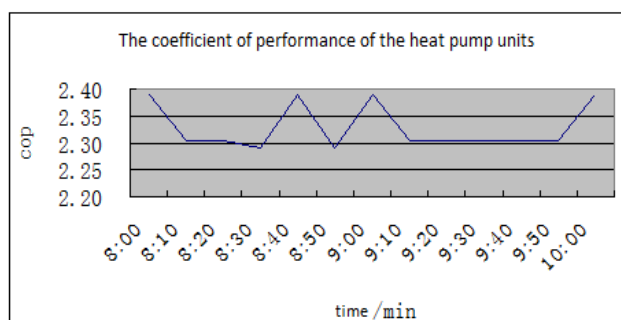


Figure 4.2.1 The coefficient of performance of the heat pump unit changes along with variation of time.

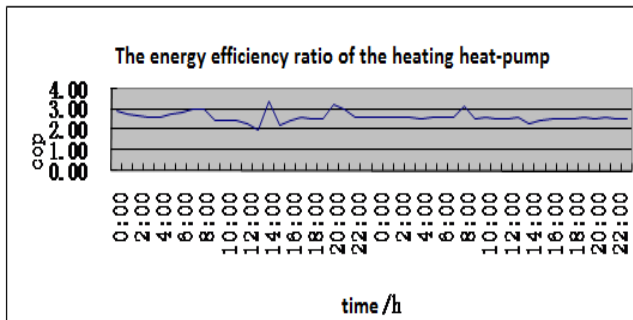


Figure 4.2.2 The energy efficiency ratio of the heat pump system changes along with variation of time.

5 . Conclusions

Based on the data above, indoor temperature and humidity are basically achieve the thermal comfort requirements of the user. The stable operation of the heat-pump units and the heat pump system are conducive to the long-term and efficient operation of system.

As the outdoor temperature decreases and the required load increases, the well water inlet temperature of the heat source-side slightly reduced, the temperature of the inlet increases slightly. at the same time, take the

inverter pump to increase the well water flows into the unit to meet the increased load. In the user-side, owing to increasing of the required load, the temperature of the user-side decreases, temperature of difference between supply and return water increases slightly. The engines appropriate to open the unit and circulating pump valve larger to increase the flow of the user side to meet the increasing demand of the load. The engines said it need the changes in the user side of the flow by about 40 % - 50 % of the rate to meet the changing needs of the load. According to the analysis of energy efficiency above, the coefficient of performance of the heat pump

unit reaches is about 2.84, the energy efficiency of the heat pump system is about 2.55.

Compared with other heating mode, the heat pump system is both energy-efficient and economical.

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MEASUREMENT OF ENERGY EFFICIENCY: A NEW ECONOMIC INDICATOR VS. ENERGY PRODUCTIVITY

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ABSTRACT

Energy productivity, defined as the ratio of GDP to energy used, is pervasively used as an indicator of energy efficiency at national level, although it is not an appropriate one. We point out that the traditional energy productivity could measure only the cost effectiveness of energy to achieve a given level of real GDP, but not the efficiency of energy per se. After critically reviewing a number of existing indicators, we propose a new economic indicator to measure energy inefficiency that could be decomposed into four components, with each representing the inefficiency source. Meanwhile, the new indicator also allows us to compare energy inefficiency under alternative environmental regims. Actual data is used to demonstrate the trend of energy efficiency in Chinese Taipei, and the underlying driving forces contributing to inefficiency are also identified.

Keywords: indicator, energy efficiency, energy productivity, technical efficiency, real profit loss, demand efficiency

1. INTRODUCTION

Energy was one of the main themes at the 2002 World Summit on Sustainable Development in Johannesburg where the importance of energy efficiency to sustainable development was emphasized. The significance of energy efficiency in response to climate change is also addressed in both UNFCCC and Kyoto Protocol meetings. Nowadays, energy efficiency is so crucial that many countries have initiated relevant innovative policies and programs.

Energy efficiency improvement is one of the six national policy goals in Chinese Taipei. Since the government is extraordinarily keen in dealing with challenges of climate change and energy crisis, the National Energy Conference, organized by the central government was held in 1998, 2005 and 2009, had concluded hundreds of 'energy-saving and carbon-reduction' programs to be implemented by responsible authorities. Official documents such as 'Framework on Sustainable Development Strategy in ROC', 'Implementation Plan for Sustainable Development, Executive Yuan', and the newly approved 'Framework on Sustainable Energy Policy' (FoSEP) consistently call for industrial transformation in accordance of the 'Two highs and two lows' principle.¹ Furthermore, specific targets for energy efficiency improvement have been claimed in several important national conferences such as Chinese Taipei Sustainable Development Conference

(April 2006), Chinese Taipei Economic Sustainable Development Conference (July 2006) and the 8th National Conference on Science and Technology (January 2009). FoSEP, for instance, calls for an annual increase in energy efficiency of no less than 2% in the coming eight years, and an energy intensity reduction to a level as low as 20% of the 2005 level in 2015 and 50% in 2025 by means of technological and institutional innovation.

The question, although not new, is how to make energy efficiency measurable, verifiable and reportable. Recently this question has received worldwide attention particularly soon after the UN Secretary-General of urged all countries in COP 14 to conduct adequate evaluation.

At the national level, energy productivity and energy intensity, reciprocal to each other, are commonly used to measure energy efficiency. As shown in Figure 1, energy productivity, defined as the ratio of real GDP to energy used, fluctuates dramatically over time and could hardly characterize the actual status of energy efficiency performance for several reasons. Firstly, technological progress of energy production and consumption occurs progressively without sudden cutback. Secondly, the energy user's rationality in energy allocation is normally consistent and rarely unstable over time. Thirdly, industrial structural change toward cleaner and more efficient manner is unlikely to happen suddenly.

Finally, the numerator of the energy productivity, GDP, is the sum of value added of all sectors and attributable to numerous production factors (e.g., capital

stock, human capital), as well as technological, structural and institutional changes. Hence, energy

productivity cannot ideally measure energy efficiency and, therefore, more justifiable indicator is warranted.

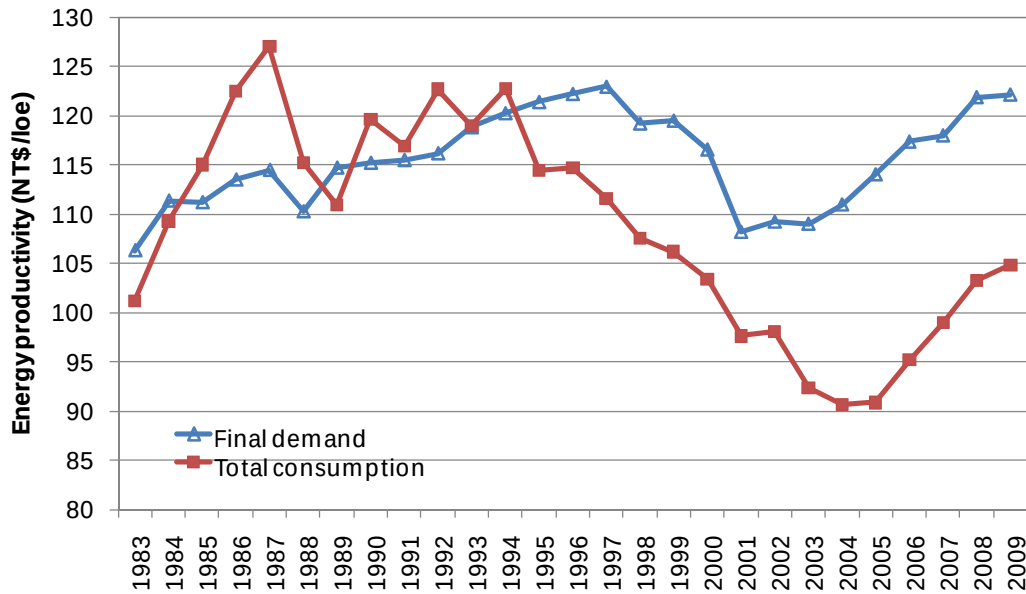


Figure 1. Changes of energy productivity in Chinese Taipei

The paper is organized as follows. The second section highlights the genuine economic meanings and the drawback of energy productivity as an energy efficiency indicator. In the third section, we critically review a number of existing efficiency indicators, including technical efficiency and allocative efficiency, as well as the efficiency indicator for individual input as proposed by Feijoo et al. (2002). A new indicator is introduced and its properties examined in the fourth section, with particular emphasis on its decomposition and responsiveness to alternative environmental regims. In the fifth section, we use actual data to estimate the performance trend of the new indicator in Chinese Taipei and identify the driving forces contributing to energy inefficiency. The result, as compared to energy productivity, exhibits striking difference.

2. PROBLEMS WITH ENERGY PRODUCTIVITY AS AN INDICATOR

As energy efficiency measurement is not a new issue, numerous indicators were proposed (Turvey and Norbay 1965; Berndt 1978). Patterson (1996) classified efficiency indicators into four categories, namely, thermodynamic indicator, physical-thermodynamic indicator, economic-thermodynamic indicator, and economic indicator. Energy productivity belongs to the third category.

As mentioned earlier, energy productivity could measure neither technical efficiency nor allocative efficiency of individual inputs such as energy. Suppose the production function of real GDP is concave and represented by $GDP = f(E)$, where E denotes energy use. Given the energy price w , the efficient energy use

is E^* , that maximizes the net benefit and is characterized by the tangent point (i.e., A in Figure 2) between GDP production curve and the iso-expenditure line of energy (IEE). Energy productivity at this input level is given by the slope of line ①, identical to G^*/E^* . Now consider two technically efficient alternatives, say E' and E'' , that produce G' and G'' , respectively. The associated energy productivity is given by the slope of line ② (G'/E') and line ③ (G''/E''), respectively. Although E^* is both technically and allocative efficient, its productivity is less than E' while greater than E'' . As long as the production is concave, the energy productivity will decrease with energy use, despite all energy uses are technically efficient.

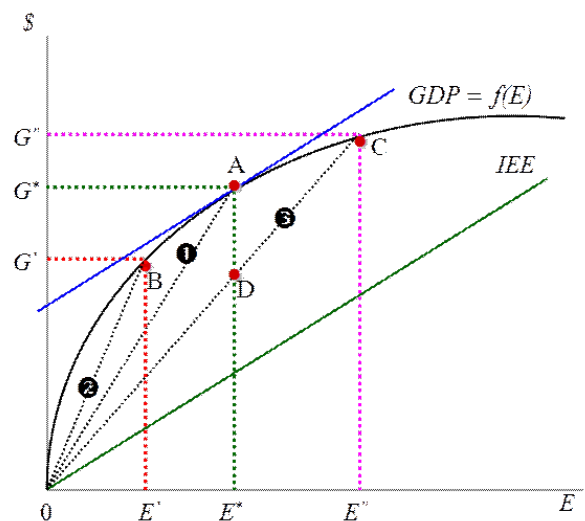


Figure 2 Measurement problems with energy productivity as an efficiency indicator

Energy productivity cannot be used to measure energy efficiency at the firm level either. For example, a technically inefficient firm using E^* but producing only DE^* (characterized by point D in Figure 2) has the same energy productivity as the firm C, that is technically efficient.

3. ALTERNATIVE ENERGY EFFICIENCY INDICATORS

Technical efficiency (TE) and allocative efficiency (AE) are the efficiency indicators widely used in literature to evaluate individual sector or firm's overall efficiency in resource allocation (Schmidt and Lovell, 1979; Kopp and Diewert, 1982). They could be estimated using data envelopment analysis (DEA) and the stochastic production frontier (SPF).

Both TE and AE, estimated using either DEA or SPF, could not reflect the efficiency level of individual inputs such as energy. This could be illustrated by Figure 4, where $Q = f(E, K)$ is the production frontier of a group of firms (represented by points A, B, H, M, G and F) employing energy (E) and capital (K) to produce the same level of output Q^0 . Given constant input prices, the iso-cost line are represented by C_i ($i = A, B, E$)).

In Figure 3, for example, firm H is less efficient than firm A both in terms of TA and AE as both produce the same level of output. Nevertheless, firm H is employing an efficient level of energy, while A consumes less energy. Hence, the inefficiency of energy use could be attributable to the fact that the firm is likely to use both inputs inefficiently, too much capital and inadequate energy. On the other hand, firm G is presumably more efficient than firm H, even though both are using the same optimal level of energy. The relative efficiency for firm G may originate from adopting capital-saving technology. In sum, estimates of TE and AE could hardly infer whether a firm is using energy more efficiently than another.

To better measure energy efficiency, Feijoo et al. (2002) suggested using the conditional input demand (i.e., E^* in Figure 3) as a benchmark. The energy efficiency of firm H, for example, is measured by the ratio of the optimal level (E^*) to the realized level (E^0); i.e., $E^*/E^0 = 1$). In other words, the firm will be more energy efficient as long as E^*/E^0 is larger. While such an indicator allows for energy efficiency comparison among firms with heterogeneous technology, it also suffers certain limitation. Consider the firm A and firm B, both using energy less than the optimal level. Firm A will be more energy efficient than firm B, although firm B is more cost efficient as well as technically efficient. Similarly, firm M is more energy efficient than firm A and firm B, but less cost effective. Accordingly, the benchmark approach will be inconsistent with cost effectiveness if the firm is using less energy than the optimal level.

In practice, an energy inefficient firm may overuse as well as underuse energy. A proper energy efficiency indicator should be capable of reflecting the energy inefficiency resulting from both cases.

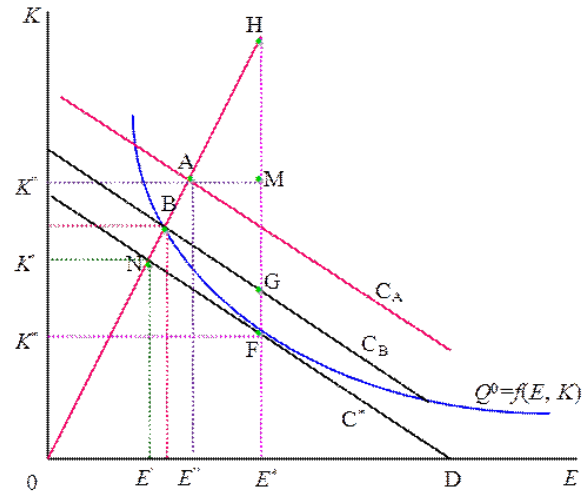


Figure 3 Measurement of TE and AE

4. A NEW INDICATOR: ENERGY DEMAND EFFICIENCY

In light of the shortcomings mentioned above, we propose a new indicator based on the average real profit loss and named the 'energy demand efficiency'. A firm, facing a given energy price (w_E), is considered inefficient if its energy use diverges from the efficient level (E^*), characterized by the first-order necessary condition for profit maximization, i.e., $VMPE = w_E$, where VMPE is the value of marginal product of energy.

Suppose π_j^* and π_j^o are the demand-efficient profit and the realized profit for firm j . We define the average real profit loss (RPL $_j$) as follows:

$$RPL_j = \pi_j^* - \pi_j^o = (\pi_j^* - \pi_j^e) + (\pi_j^e - \pi_j^o), \quad (1)$$

where

$$\pi^* = f(E^*; K^*) - (w_E / P)E^* - (r / P)K^*,$$

the maximal profit when all inputs are demand efficient;

$$\pi_j^e = f(E_j^e; K_j^*) - (w_E / P)E_j^e - (r / P)K_j^*,$$

the potential profit when energy use is technical efficient (but demand inefficient); and

$$\pi_j^o = f(E_j^o; K_j^o) - (w_E / P)E_j^o - (r / P)K_j^o,$$

the observed profit, under which energy use is technical as well as demand inefficient.

$\pi_j^* - \pi_j^e$ in Eq. (1) is the real profit loss resulting from technical inefficiency (LTI) in energy

use, while $\pi_j^e - \pi_j^o$ the loss from demand inefficiency (LDI). Measurement of both LTI and LDI can be illustrated by Figure 4.

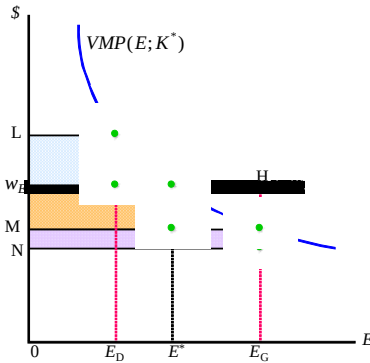


Figure 4. Real profit loss of energy inefficiency and its components

Let's consider energy use of six firms ($j = A, B, C, D, F$ and G) in Figure 4, where A, B and C are technical efficient, while D, F and G are demand inefficient as well as technical inefficient. Among others, firm B is most efficient and receives maximal profit π_B^* , given the energy price w_E . LTI for firm A and firm C is measured by ΔBDA and ΔBHC , respectively. For the firm such as D, F and G , profit loss consists of both LTI and LDI. For firm D, F and G , LDI is measured, respectively, by the shaded areas, $\Delta ADw_E L$, $\Delta BFMw_E$ and $\Delta CGNM$. Accordingly, we define a new energy indicator as follows:

$$EEI_j = RPL_j / E_j^o \quad (2)$$

Note that EEI_j is nothing but the real profit loss per unit of energy use. The firm's energy efficiency is, therefore, decreasing with EEI_j ; the smaller the EEI_j , the more efficient the firm.

The decomposition of the real profit loss in Eq. (1) is based on the implicit assumption that inputs other than energy are all demand efficient (i.e., $K = K^*$), that is not realistic in practice. As long as demand of energy is not independent of K , misallocation of other inputs may introduce additional profit loss for energy use. To identify such factors, one may modify the profit loss identity as follows:

$$RPL_j = \pi_j^* - \pi_j^0 = (\pi_j^* - \pi_{jE}^e) + (\pi_{jE}^e - \pi_{jK}^e) + (\pi_{jK}^e - \pi_{jE}^0) + (\pi_{jE}^0 - \pi_j^0)$$

$$= (\text{AEL}_E + \text{AEL}_K + \text{TEL}_E + \text{TEL}_K) \quad (3)$$

where

AEL_E = profit loss due to demand inefficiency with energy;

AEL_K = profit loss due to demand inefficiency with non-energy inputs;

TEL_E = profit loss due to technical inefficiency with energy inputs; and

TEL_K = profit loss due to technical inefficiency with non-energy inputs.

In general, the new indicator has some advantages over the benchmark approach.

(a) The new indicator incorporates both technical and demand inefficiency of individual input;

(b) The estimates are always consistent with cost effectiveness and responsive to environmental regimes;

(c) The decomposition clarifies the relative contribution of factors to inefficiency;

(d) Access to data base and estimation are easy in practice and free from specification bias associated with SFP;

(e) As it will be shown later, the indicator estimates exhibit smooth trend, that differs from energy productivity; and

(f) Inter-firm comparison of energy efficiency is plausible.

5. AN EXAMPLE

For illustration, a GDP production function of Cobb-Douglas is specified for Chinese Taipei to benchmark the optimal levels of energy and non-energy as inputs. The trend of the observed levels (E_o and K_o) and the optimal levels (E^* and K^*) is depicted, respectively, in Figure 5 and Figure 6. It's interesting to find that both energy and non-energy were overused. The divergence between the observed and the optimal is significantly magnified in recent years. The optimal level of energy began to decrease after 2004, reflecting partly the volatile energy prices and partly the government's effort in encouraging renewable energy as substitute.

The real profit loss (RPL) originating from inefficient energy use is estimated and presented in Figure 7. The total RPL exceeds a trillion NT\$ and has been increasing with energy use since last decade. Decomposition analysis of RPL reveals that the major contribution comes from the loss due to technical inefficiency with non-energy inputs. In comparison, the profit loss due to demand inefficiency with energy accounts for limited shares (below NT\$200 billion and about one-sixth of the total RPL). The implications are twofold.

While much attention is paid to energy efficiency improvement, more attention should be directed to the improvement of technical inefficiency of non-energy. Accelerating technology innovation of non-energy inputs might be more effective.

The estimates of RPL for energy exhibit smooth upward trend, implying energy use is moving away from its optimal level. Energy efficiency seems not sensitively respond to the energy price variation in the world market. This, to a great extend, could be attributable to the domestic price stabilization measures

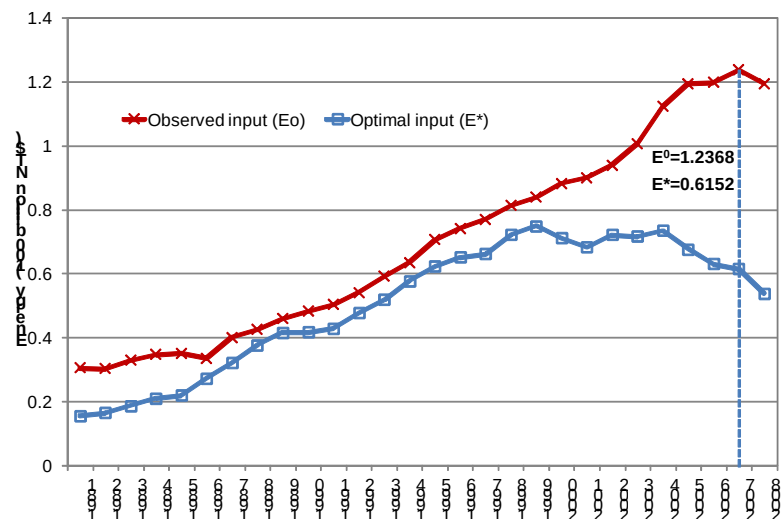


Figure 5. Energy use: observed vs. optimal

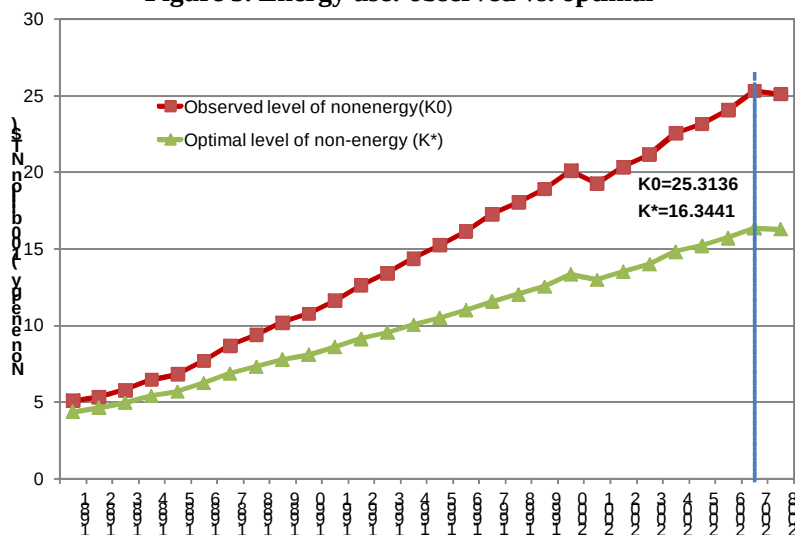


Figure 6. Non-energy use: observed vs. optimal

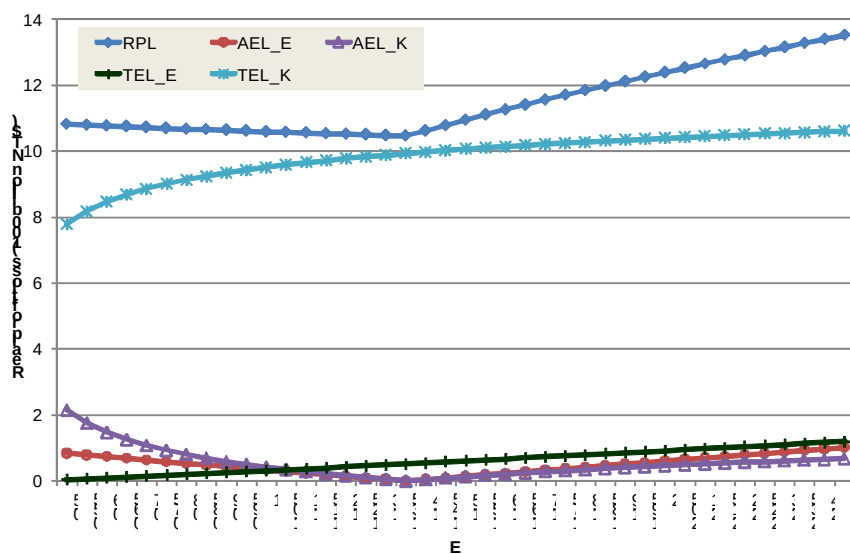


Figure 7. Decomposition of the real profit loss of inefficient energy use EP-Modified = Real total value produced / Real value produced of energy sector

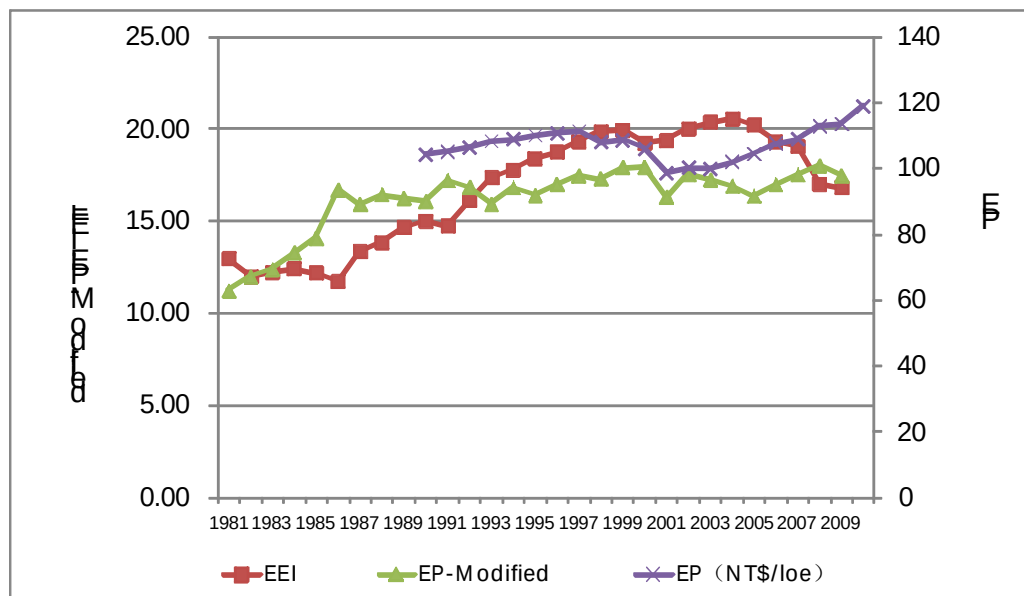


Figure 8. Energy efficiency performance: EEI vs. energy productivity

The estimates of the new energy efficiency indicator (EEI) are depicted in Figure 8, together with the trend of energy productivity. Obviously, energy efficiency inferred by both indicators is not completely consistent. While energy productivity indicates an increasing improvement in energy efficiency, the results of EEI infer a more conservative performance. According to EEI, energy efficiency in Chinese Taipei had been stably improving until 2005. Since then, energy efficiency began to decline, mainly due to government intervention in stabilizing energy price while the international market price rocketing. Another reason for the worsening of energy efficiency arises from the fact the dominant petrochemical firm failed to pass the environmental impact assessment after years of effort and, as a result, was unable to modernize its aging production system.

6. CONCLUDING REMARKS

Energy efficiency, typically measured as the ratio of GDP to energy consumption, is one of the key factors for greenhouse gas emission reduction. Its role and potential in addressing CO₂ decoupling from GDP growth has been overwhelmingly emphasized without rigorous empirical evidence. Statistical data indicates that energy efficiency in Chinese Taipei had been steadily improved until the turn of the century. Since then, dramatic decline with significant fluctuation was observed. Various hypotheses were proposed to explain the peculiar phenomenon, but none of these were statistically tested. Given uninterrupted technological progress in energy use and rationality in decision making, we contend that the traditional measurement of energy efficiency is not appropriate and, therefore, a new definition is warranted. This was echoed by the UNFCCC's 14th COP Meeting in Poland since such issue, for the first time, was raised and called for global attention.

The paper begins with the driving-force identification of CO₂ emission in Chinese Taipei by applying different approaches for contrasting purpose. A decomposition approach was used that factorizes CO₂ emission in Chinese Taipei into four components, namely, population growth, GDP growth, industrial structural change and energy intensity. Just as what happened in other industrialized countries, so does in Chinese Taipei energy intensity dominate others. The results were further supported by a fixed effect model using panel data consisting of six energy intensive sectors. Energy efficiency was then redefined and measured based on the empirical results.

The underlying implications from the empirical results are multifold. The most profound one is that CO₂ decoupling from GDP growth is not as pessimistic as expected. Energy efficiency, measured in its new terms, shows continuous and steady improvement overtime. Its turbulence fluctuation is not attributable to energy inefficiency, but to the deterioration of such factors as capital accumulation, foreign direct investment, competitiveness, human-induced inefficiency, etc.

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EXPERIMENTAL INVESTIGATION ON HEAT TRANSFER OF ASYMMETRIC MICROCHANNEL SEPARATE HEAT PIPE

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ABSTRACT

In this paper, the telecommunication base station (TBS) indoor and outdoor environment simulated by the triad air condition comprehensive laboratory. The experiment model using R22 as working fluid, visibility Experimental study on the separation of asymmetric microchannel heat pipe take consideration of different filling rate, temperature and so on. Visualization experiments to increase the experimental data are not only convinced, but also conducive to the detection of the separate type heat pipe. The main measured data of this experiment consists of the temperature of the air inlet and outlet of the evaporator, the condenser section and the wind speed of the wind out of the evaporator. The results showed that the average heat transfer coefficient and the EER achieved maximum at filling rate 136%, the maximum of the average heat transfer coefficient for 1750W / (m²·°C), the maximum EER is about 7.3. When the Indoor and outdoor temperature is changing from 10 °C to 20 °C, the EER would increased 100%. According to the study the asymmetric microchannel separate heat pipe studied the performance of heat transfer higher 20% than the symmetric microchannel separate heat pipe studied in reference[3].

Keywords: filling ratio, asymmetric, microchannel, heat transfer

1. INTRODUCTION

In order to ensure the telecommunications base station (TBS) temperature, relative humidity and air cleanliness, the separate type heat pipe unit its smaller compact structure, ensure room cleanliness requirements advantages to a wide range of application. The appropriate range of filling ratio, which is defined as the volume ratio of charged liquid to the evaporator. Chen Lan [2] proposed that filling ratio should be 70%-114% through studied a special separate type heat pipe filling rate changed between 32% and 179%. Through a comprehensive analysis of the separate type heat pipe heat transfer and heat transfer coefficient, the phase change heat transfer dominates the heat pipe working fluid, the heat pipe heat exchanger achieve the best results at the filling rate 70%-114%. Jinxin [3] showed that the appropriate range of filling ratio was 113%-140% that studied the separation microchannel heat pipe (evaporator and condenser sections symmetrical) heat transfer performance. Qian Xiao Dong [4] studied the brass string fins separate type heat pipe, by contrast R22 and R134a at different filling rate system heat transfer performance, the result showed these two working fluids the best filling rate are 80%. Zhang hong [1] calculated the best statement of separate type heat pipe by the minimum filling rate theory. As the advantages of compact structure, small size, low wind resistance, and the effect of heat transfer, the microchannel heat exchanger has been widely used in

the air conditioning industry. This paper presents the asymmetric heat pipe. The so-called asymmetry is that the structure of the means used in the evaporating section and the condensing section is different, i.e. the symmetrical type refers to have the same structure of the traditional design of the evaporating section and the condensing section [1]. Asymmetry microchannel heat pipe has been little researched, this asymmetry the microchannel heat pipe unit filling rate and the performance of heat transfer studied less. The use of the asymmetric heat pipe unit has the obvious advantage is the small the start-up temperature difference of the heat pipe, thus greatly improving the performance of heat pipe heat exchanger, and energy conservation. In this study, used different area microchannel heat exchanger as a separate type heat pipe evaporator and condenser section (the condensing section area is 10% larger than the evaporator section area), and used the triple comprehensive laboratory to simulate the environment of telecommunications base station (TBS) indoor and outdoor, the visibility experimental study on the separation of asymmetric microchannel heat pipe for different filling rate, indoor and outdoor temperature difference and other factors. The main purpose of this experiment is to solve the problem of low heat transfer performance of the symmetric heat pipe unit, low start-up temperature difference, the use of natural cold source short time and so on.

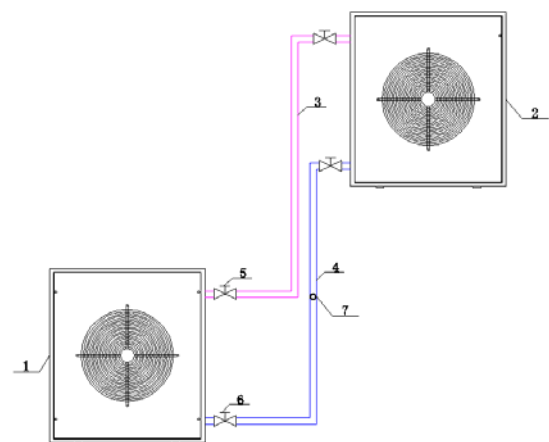
NOMENCLATURE

Abbreviation		Subscripts	
EER	Energy efficiency regulation	e	evaporator section
TBS	Telecommunications base station	1	the front of evaporator section
		2	the behind of evaporator section
Symbols		Greek Letters	
K	Average heat transfer coefficient (W/(m ² °C))	ρ	Density (kg/m ³)
Q	Heat transfer rate (W)		
A	Heat transfer surface (m ²)		
V	Air volume (m ³ /s)		
c _p	Specific heat (J/kg K)		
P	Fan power (W)		
t	Temperature (K)		

2. EXPERIMENTAL DEVICE AND DATA ANALYSIS

2.1 Experimental device

The asymmetric microchannel separate type heat pipe can transport heat from the indoor air to the outdoor air when the outdoor air temperature is cooler enough than the indoor air. It constitutes of an evaporator, a condenser, vapor ascending tube, condensate descending tube, air valves and liquid valve, as shown in Fig.1. The evaporator (indoor unit) is located on the bottom with a fan in the indoor air and the condenser (outdoor unit) is upper with another fan in the outdoor air, which is connected by the vapor ascending tube and the condensate descending tube. When the asymmetric microchannel separate type heat pipe working, heat can be absorbed form the indoor air and its working fluid can be vaporized on the evaporator that produces a pressure rise to make the vapor of working fluid flow to the condenser through the vapor ascending tube. On the condenser, the vapor condenses into the liquid releasing the heat to the outdoor air, and then the liquid goes back to the evaporator because of its gravity. In this cycle, the heat can be transported from the indoor air to the outdoor air, thus the indoor air is cooled down. Using the more asymmetric is the use of increasing the area of the condensation section, while the source of the refrigerant and natural cold heat exchanger.



1. Evaporator section; 2. Condenser section; 3. Vapor ascending tube; 4. Condensate descending tube; 5. Air valve; 6. Liquid valve; 7. Moisture and Liquid indicator

Figure 1 System of the asymmetric microchannel separate heat pipe

The parameters of the Asymmetric microchannel separate heat pipe showed in Table 1, where the height difference of the two is 1.2m. It's enough to overcome the loss of resistance within the heat exchanger. At the condensate descending tube install with the exhaust valve, however, the exhaust valve as the working fluid filling port. The working fluid is R22. This asymmetric microchannel separate type heat pipe system is characterized by the use of all-aluminum microchannel heat exchanger as the condenser section and the evaporator section and not the area of the evaporator and the condenser section. Due to the advantage of compact structure, small size, low wind resistance, good effect of heat transfer, that the all-aluminum microchannel heat exchanger has a wide range of applications. The asymmetric separate type heat pipe can greatly to enhance the performance of heat transfer and to extend the use of natural cold source.

Table 1 Parameters of the asymmetric microchannel separate heat pipe

Type	Evaporation section	Condenser section
Height×Length(mm)	790×840	830×880
The outer diameter of the upper current(mm)	19	19
The outer diameter of the bottom current(mm)	16	16
The size of the microchannel(mm)	1.2×16	
Fin height(mm)	8.02	
Fin spacing(mm)	1	
Louver angle	27°	
The number of flat tubes	85	89
Channel configuration	Single-channel	

The experiment used the triad air condition comprehensive laboratory to simulate the telecommunications base station (TBS) indoor and outdoor environment. The air handling units which in the triad air condition comprehensive laboratory can provide warming and cooling, humidification and dehumidification and so on. The way to simulate the telecommunication base stations is that using automatic control functions and high-precision control instruments, and testing the upper orifice plate uniform air supply and the lower part of the way back to the wind. This approach can be more good airflow assurance simulation environment uniformity. The condenser section of the asymmetric microchannel separate heat pipe on the outdoor side (the dry-bulb temperature of 18 ° C, the wet bulb temperature of 10 ° C), it's the evaporation section on indoor side (the dry-bulb temperature of 28 ° C. the wet bulb temperature of 20 ° C). Temperature control through the triad air condition comprehensive laboratory automatically controls the error of ± 0.2 °C. During experiment, six different working fluid fill ratios are tested in order to determine the optimal value. In the inlet and outlet of the evaporator and condenser sections, respectively, in accordance with Qiebeiqiefu way (GB10178-88) set measuring point position, the measuring point is arranged as shown in Figure 2. The position of the measuring point is respectively arranged thermocouples and in the outlet of the evaporator section of the measuring point carried out the measurement of the amount of wind. Install sight glass at the condensate descending tube that experimental effects of visualization and also install pressure gauges at the condensate descending tube and the vapor ascending tube.

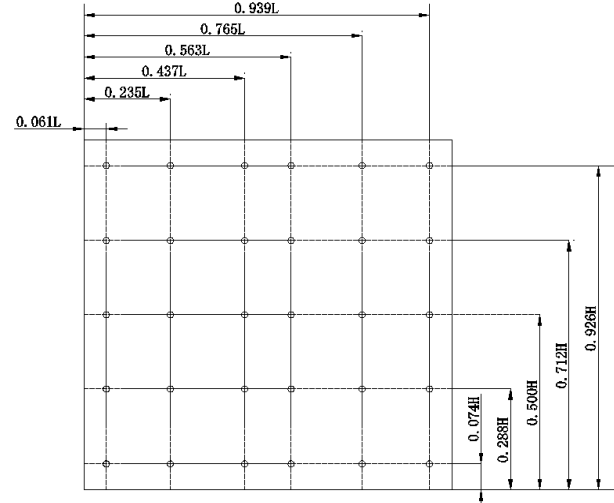


Figure 2 Position distribution of the thermocouple

2.2 Data Analysis

The average temperature of the inlet and outlet is the arithmetic mean of the 30 experiments thermocouple measured temperature.

The filling rate in Eq. (1) is defined as

$$\Omega = \frac{G / \rho_l}{V_e} \quad (1)$$

The commonly used method of engineering data analysis is indirect measurement: (1) Calculate the heat transfer coefficient method (2) Calculate the energy efficiency regulation (EER). Through these two methods can more comprehensive consideration heat transfer performance.

Calculate the heat transfer coefficient method, the heat transfer coefficient in Eq. (2),

$$K_e = \frac{Q_e}{A_e \Delta t_m} \quad (2)$$

The logarithmic mean temperature difference Δt_m in Eq. (3)

$$\Delta t_m = \frac{\Delta t_{\max} - \Delta t_{\min}}{\ln \frac{\Delta t_{\max}}{\Delta t_{\min}}} \quad (3)$$

The heat transfer of evaporation section Q_e in Eq. (4)

$$Q_e = \rho V c_p |t_1 - t_2| \quad (4)$$

Calculate the energy efficiency regulation (EER), the EER in Eq. (5),

$$EER = \frac{Q_e}{p} \quad (5)$$

3. RESULTS AND DISCUSSINS

3.1 The effect of heat transfer of the asymmetric heat pipe under different filling rate

Working fluid fill ratio in the separate type heat pipe heat transfer performance plays a significant role. When the filling rate is too high, the gas - liquid mixture enters the vapor ascending pipe, or even into the condensate section, so that reducing the heat transfer performance of the system; When the filling rate is too low, then make the inner wall surface of the liquid film covering on the tube in the evaporating section, so that evaporation section wall temperature is rising rapidly which leading heat transfer deterioration. Only find a reasonable the filling fate in the best condition that could have a good performance of separate heat pipe. To determine the optimal fluid fill radio, experiments are conducted under different fill radios of the working fluid in the asymmetric heat pipe. For each fluid fill radios Table2 show the basic data was measured in the experiment. We can see from the Table 2 that the different filling rate will cause the inlet and outlet of the evaporator section temperature change, which reflects the evaporator section cooling capacity changed. As can be seen from Figure 3, the average heat transfer coefficient achieved maximum at filling rate 136%, then the EER reaches the maximum.

Table 2 The basic tests data

Test time	Changer(Filling radio) (Kg)	Indoor		Outdoor	
		Inlet (°C)	Outlet (°C)	Inlet (°C)	Outlet (°C)
14:00	2.8(119%)	28.3	24.5	18.5	21.8
15:00	2.9(123%)	27.8	24	18.3	21.5
15:30	3.0(128%)	27.7	23.7	17.9	21.4
16:00	3.1(132%)	28.1	23.9	18.1	21.8
16:30	3.2(136%)	28.5	24.2	18.6	21.8
17:00	3.3(142%)	28.3	24.1	18.5	22.2

We can see from the Figure 3, when the filling rate change from 118% to 142%, the heat transfer coefficient is increased to achieve local maximum at first, and then is decreased increases again to the maximum. Reaches a local maximum, and the average temperature difference of the previous two point indoor can be clearly seen from the experimental data of Table 2 is different with the temperature difference between the outdoor. As can be seen from this experiment outdoor temperature difference is smaller, more able to reflect the separate type heat pipe evaporator section of the heat transfer performance improves. From this figure also can be seen when the filling rate of about 136%, the heat transfer coefficient reached the maximum. Reference [3] data in figure 3 shown the relationship of the symmetric split heat pipe filling rate and average heat transfer coefficient. Reference [3] in addition to using R134a as working fluid and using the

same area of the condensing section and the evaporating section , other experiment conditions keep constant(R22 and R134a filling rate consistent, but the working fluid R22 when the average heat transfer capacity than R134a high 19.2%. [4]). Derived from the comparison of the two curves in Figure 3, separate type heat pipe used in this experiment symmetrical than the reference [3] studied symmetric separated heat pipe heat transfer performance on average about 20%.

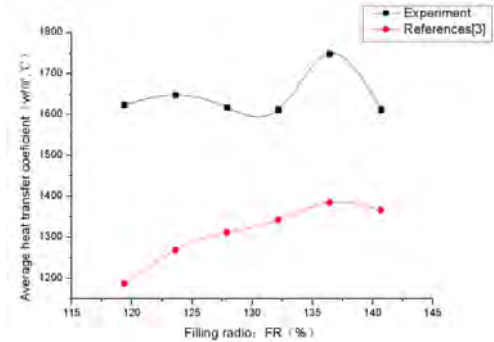


Figure 3 Average heat transfer coefficient at diverse filling rate

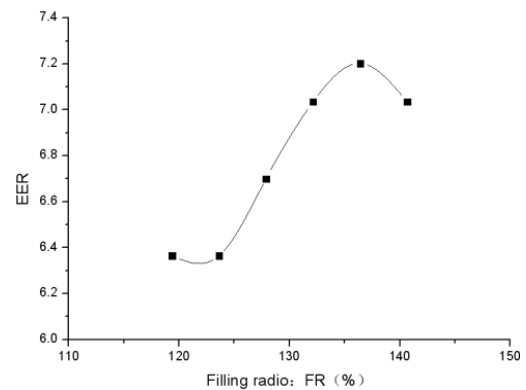


Figure 4 EER under different filling rate

What's more, this experiment install the moisture and liquid indicator (the moisture and liquid indicator can know the condition of the refrigerant in the system, which can advance to know whether refrigerants and system problems.), that the entire experiment can be measured visualization. The data in Table3 under the condition that the indoor and outdoor temperature of 10 °C. Table 3 can be seen in the fluid-filled rate of 107% from the sight glass when seen in the flow a little bit of a long time before, after the inspection found that there is a refrigerant leak. When the filling rate is about 136%, the condensate descending tube almost filled with refrigerant. In conclusion, from the above three ways that the experimental results are at the filling rate about 136%, the heat pipe has best performance.

Table 3 The state of Flow liquid at diverse filling rate

Filling ratio (%)	The state of flow liquid depending on the Moisture and Liquid indicator
107	Flow a little for a long time

119	1/3 pipe flow
124	1/3 pipe flow
128	1/3 pipe flow
132	1/2 pipe flow
136	Pipe almost filled with refrigerant.
141	2/3 pipe flow

3.2 The effect of heat transfer of the asymmetric heat pipe under the different between indoor and outdoor temperature difference

Different indoor and outdoor temperature difference is one of the important factors that affect the separate type heat pipe heat transfer performance. As we can see from Figure 5, the EER increasing rapidly when the indoor and outdoor temperature difference increase and when the indoor and outdoor temperature difference change from 10 °C to 20 °C, the EER increase 100%, which fully showed the separate type heat pipe the energy saving effect of a wide range of applications the prospects [7,8].

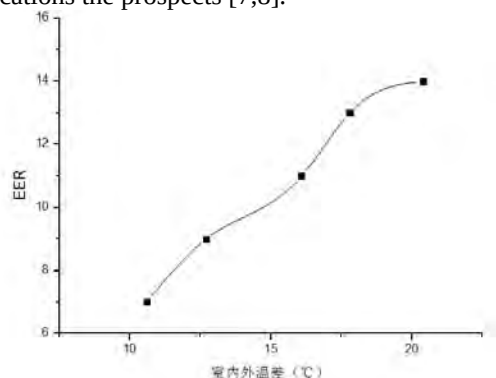


Figure 5 EER under the different between indoor and outdoor temperature difference

4. CONCLUSIONS

In this paper, the telecommunication base station (TBS) indoor and outdoor environment simulated by the triad air condition comprehensive laboratory. Visibility Experimental studies on the separation of asymmetric microchannel heat pipe are taken into consideration of different filling rate, temperature and so on. The following conclusions can be made:

(1) The appropriate range of filling ratio, which can keep the asymmetric microchannel heat pipe steady and effective. Through the comparison of different evaluation index, and also uses visualization experiment, reach a conclusion that the asymmetric microchannel has best performance at pipe filling rate of about 136%. At this filling rate the heat transfer coefficient and the EER reached maximum, and also can be observed through the sight glass to the considerate descending tube is almost full pipe flow.

(2) Through the experimental comparison, the asymmetric heat pipe model used in this experiment is coefficient about average 20% more than the symmetric heat pipe model [3].

(3) In the case of different indoor and outdoor temperature difference, the asymmetric microchannel separation of the heat pipe with the greater the temperature difference between the indoor and outdoor, the greater the EER, and when the indoor and outdoor temperature is changed from 10 °C to 20 °C, the EER increase 100%.

This paper is mainly about the asymmetric separate type heat pipe experimental study, and our further works include:

(1) This paper is only for a asymmetrical structure of the experimental study, and we will need further experimental and theoretical studies of this asymmetrical relationship of the size match.

(2) This paper is mainly for the asymmetric separate heat pipe heat transfer performance of the one-to-one, and we will further study on the asymmetric separate type heat pipe heat transfer performance of the combination in the form of one-to-many, in order to reduce heat pipe start temperature difference and extend the time of using natural cold source.

(3) We will study one-to-many combinations asymmetric separate type heat pipe through the automatic control to improve the heat transfer performance of heat pipe.

ACKNOWLEDGEMENTS

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LIVING WALL MATERIALS AND SYSTEMS FOR AUTOMATIC BUILDING THERMO-REGULATION IN RESIDENTIAL BUILDINGS

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ABSTRACT

The objective of this National Science Foundation, Emerging Frontiers in Research and Innovation NSF EFRI project by the University of Colorado Boulder's College of Engineering and Applied Science is to develop a "living wall" concept that follows the principles of biomimicry of the body's thermo-regulation systems, thereby changing the way buildings efficiently adapt to the surrounding environment through sophisticated heat transfer processes and artificial metabolic adjustments. The living wall system embeds two sets of optimized micro vascular fluid channel (MVFC) networks and a distributed phase change medium (ePCM) into an innovative polymer-based building wall allowing the rapid movement of heat via integral liquid flow, the charge/discharge of ePCM's, the focusing of heat via architectural structures, the movement of air within solar cavity walls, the collection of PV energy, all embedded within the living wall. The walls respond to real-time indoor and outdoor temperature variations so as to dynamically regulate the thermal behavior of the building envelope and the entire dwelling. While natural convection and thermal uplift drives air upwards through interstitial walls, novel hydrogels with tailored temperature sensitivity move liquid into the wall's depth. Mathematical models and computational design approaches are being developed to optimize the layout of the holding material, the MVFCs and ePCM components. A prototype of the living wall will be fabricated and studied under a series of thermal and structural loading scenarios. The performance of the entire living wall system under dynamic conditions is being investigated through multiple simulations suggesting the development of the living wall concept universally applied in future building constructions.

While our NSF funded work will develop both residential and commercial living wall and Zero Net Energy ZNE strategies, this paper will focus specifically on net-zero-energy residential buildings. A significant reduction or in some cases the elimination of a residential building's energy footprint is possible with the living wall's heating, cooling and ventilation strategies, via smart building envelope designs. Based on preliminary results, the new living wall concept is expected to reduce 90-95% of the current residential building energy needs. Heat collected within the concentrating structures of the living wall is drawn off as hot air and/or water and stored, re-circulated, redistributed, seeded, concentrated and mixed throughout the building's solar interstitial walls and interior structures.

Solar heat collected is used to:

- create stack effects, allowing for the rapid upward buoyancy of air within interstitial walls
- draws air summer or winter via the temperature differential within the living walls at a predictable and quantified rate
- draws cool ground coupled or nighttime supply air through the building with negative pressure
- conditions air to the specified temperatures via stack effect ventilated heat exchangers
- combines to provide sufficient ventilation to exceed Air Changes/Hour ACH requirements
- creates sufficient cfm evacuation of buoyant hot air to draw stored cool air through the building's interior on the very hottest days
- supplies the entire structure's heating, cooling and ventilation needs without furnaces, boilers, chillers, air conditioners, ventilators or fans.

Interstitial stack effect walls are not new. The living wall innovation however allows the expansion of the performance and structure of a building's exterior skin, turning it from just a wall to an "actively passive conditioning system". The living wall facilitates the concentration of heat, the rapid collection of heat, the rapid infusion of heat into and out from the exterior skin turning it into a heat exchanger system that allows for the rapid and efficient collection, distribution and circulation of that heat. This innovation creates thermal air movement opportunities never before seen, allowing these walls to supplant typical HVAC equipment based systems for a totally passive approach. Living walls thereby provide the opportunity for complete and full ZNE building strategies. With the integration of PV or micro-wind technologies within these systems, true Positive Net Energy PNE buildings could become a reality. Our work continues to develop and test these systems and theories.

The success of this research will greatly stimulate the application of biomimetic design principles for developing sustainable energy efficient engineering designs and reshape the current building design and construction philosophy and practice.

The proposed research will ultimately lead to the development of a brand new interdisciplinary academic program in Bio-Architectural Science and Engineering (BASE). This will include the development of a horizontally and vertically integrated cross-disciplinary research and education program for pre-college, undergraduate and graduate students from broad fields of interest, as well as developing a rich, multi-faceted outreach program. The PIs will team up with the NSF-funded Colorado Alliance for Graduate Education and the Professoriate and College of Engineering's BOLD Center to attract and integrate undergraduate and graduate students from underrepresented groups into this research project. A formal assessment plan will be implemented to measure the success of the education and outreach activities.

1. INTRODUCTION

The Living Wall System is developed as an integral part of an interstitial wall system for residential building solutions. The Living Wall reacts to the building's external environment and climatic opportunities, driven mostly by insolation, and secondarily by temperature and wind. A multiple layer interstitial wall system with an exterior layer consisting of "smart" or tuned glazing with multiple high tech coatings forms the foundation of the Living Wall System's integration into building skins. This outer curtain wall/skin acts like a tall and narrow vertical greenhouse collecting and concentrating solar energy inside the thermal interstitial wall. The inner wall consists of the Living Wall panels mounted in a curtain wall system and forming the interior of the interstitial wall's greenhouse. The rising heat developed within the interstitial wall creates not only excess heat collected within the wall, but also a powerful ventilating negative pressure, dually utilizing the heat to fuel the thermal uplift as well as collected heat for the building's central heat collection and redistribution system controlled by the Building Automated System (BAS). Our work focuses on developing and testing these concepts.

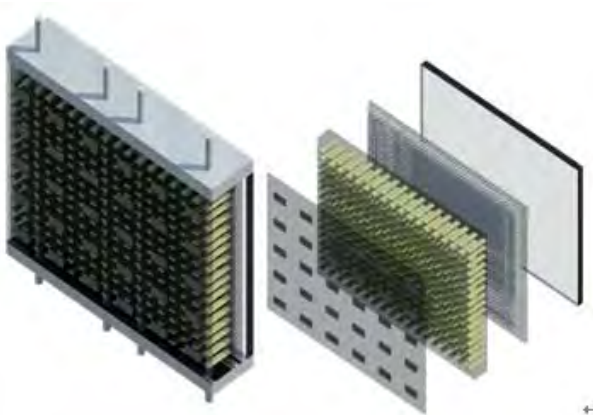


Figure 1.1 Living Wall Construction Composite Panels. Smart glass, PCM Capsules, Translucent Solar Panel, Kalwall Panel

1.1. Residential Strategies: Heating, Cooling & Ventilation

Our research focuses on developing new passive energy profile strategies addressing the typical energy use patterns and supplanting the typical fossil fuel approach to energizing and conditioning buildings. The Living Wall acts as a building envelope system or skin, auto regulating the interior environment by utilizing the heat generated by solar gain around the building's perimeter. Other architectural strategies build on the Living Wall to enhance the macro-scale and system-level functions of the entire building. The main Living Wall architectural concepts and prototypes focus on an interstitial wall system as the outer skin of a building. This outer skin enhances the conditioning of interior systems and even supplants traditional HVAC systems entirely through passive heating, cooling and ventilation through vertical stack-effect within the envelope. Through whole system integration we've developed an approach incorporating the Living Wall within the whole building system thereby producing heat and ventilation. That in turn produces cooling opportunities with ground coupling and thermal uplift within these walls. From this whole building integrated system, the heating, cooling and ventilation needs of the building will be conditioned within and distributed from the Living Wall. Our design and modeling tools include typical programs such as: AutoCAD, Revit, Rhino, Sketch up and animation tools such as 3d Studio Max, Maya with preliminary analysis of the developed designs analyzed in Energy 10, Ecotect, Open Studio, Vasari and e-Quest, focusing on typical "architectural" programs that architects readily employ.

Our models have been analyzed on 4 seasonal day and night variations summer, winter, spring, fall. Redistribution of heat gains from within the Living Wall and ventilation from negative pressure generated in an interstitial Living Wall System are controlled by readily available building automated systems (BAS). Conceptually, the Living Wall is being developed, modeled and tested as an efficient means for heat/energy regulation and transport through the combination of hydrogels within micro vascular fluid channels, phase change material, fluid heat collection matrix and insulation layer. These interstitial Living Wall assemblies could surround any side or portion of a building creating effective building ventilation by producing a quantifiable heat source and resultant thermal uplift within the double walls. Overall width

of the interstitial walls would vary depending on the air flow efficiency of the spaces between the curtain wall and Living Wall. Those widths in residential applications could range between 6" to 1.5 feet and be tuned to enhance the stack effect and air speed within the interstitial wall system. The Living Wall as the inboard component of a glass wall produced in panels, would span from floor to floor with between 2 – 3 inches in thickness and be set into an integrated curtain wall frame. Our team will include details of such an interstitial curtain wall system, some of which accompany this report.

The Living Wall itself comprises three layers, an outboard heat conducting layer, an inboard insulation layer and an intermediate collection layer consisting PCM intersected with the hydrogels and the fluid matrix. Mathematical analysis and spreadsheets illustrated in this report quantify insolation, material, and ventilation in stack effect applications and allow for schematic analysis of the Living Wall system as well as the quantification of the heat generated. The multiple layers of the Living Wall act as a heat pump which swell during hot solar days allowing the hydrogels to rapidly transmit heat into the Living Wall's interior phase change material and fluid filled capillary matrix. Heat rapidly transports into the Living Wall acting as a building skin heat exchanger panel, collecting the heat within a liquid filled matrix. The heat is then drawn off to a centralized storage and redistribution system all controlled by the BAS.

This insolation produces the necessary thermal gradient within the interstitial Living Wall to produce the necessary air changes per hour (ACH) of the building's volume required for naturally conditioned buildings. The air column speed within the interstitial walls, analyzed by mathematical calculations produced expected wind speeds of just under 4 mph. This high ventilation rate would supplant electric fans for daily use though PV powered 5,000 cfm back-up fans would be required for non-sunny days. The collected heat is then concentrated, stored and utilized within the building for heating loads as well as redistributed to cold non-solar walls, artificially "seeding" the upper portion of those walls, thereby generating a full building perimeter solar thermal gradient and uplift across the entire interstitial wall's cross sectional area. Delta T from the top of the wall to the bottom remains the key variable in the speed and volume of this air movement, though height remains also an important variable, discussed in the residential section.

This passive approach allows sufficient air movement through 4 pipe (H & C supply and return) heat exchangers to effectively condition the building's interior. Underground air channels supply cool micro-climate conditioned supply filtered air to heat exchangers connected to ground water at 58 degrees F on the cold side and collected/stored hot water at 120 – 140 degrees F sourced from the Living Wall System. Cold air drawn through these ground coupled water to air heat exchangers supplies the cool air to a plenum or chase in a basement or crawl space incorporated within the building's design. These elements could

include an interior and occupied atrium though code issues restrict the use of such elements. This main cold air supply connects into each level's raised floor system, diffusing the cool air from the floor through the interior spaces and exhausting at ceiling level into the building's perimeter interstitial wall. Challenges for this strategy have included getting the cold air supplied from below to the center of floor plates of multiple building configurations as illustrated in Figure 2.5.3. Figure 2.5.4 illustrates the mathematical predictions for varying geometries of an interstitial wall system in variable conditions.

Considerations from our Living Wall research:

- Heat recovery strategies should be considered for exhausted hot air from interstitial wall systems augmenting the collected heat stored within the building's central system.
- The Living Wall panels would act as re-chargers to a water or ground based heat sink, producing an annual net zero energy by collecting cool/cold energy at night or seasonally thereby facilitating a "slinky effect" stabilizing the ground coupled heat sink.
- The inboard oriented Living Wall panel must accommodate effective insulation against unwanted solar heat gain on the building's skin in the summer and conversely heat loss on cold nights in the winter.

2. ARCHITECTURAL INTEGRATION

2.1 Whole Residential Building Integration and Energy Modeling

Our Living Wall research and computer modeling focuses on creating an architectural "kit of parts" approach to high performance true Zero Net Energy (ZNE) strategies. Our work remains focused on the use of the Living Wall System for totally passive true Zero Net Energy Building (ZNEB) thermal and ventilation strategies. Day-lighting strategies within the projects will affect the thermal and electrical loads of a ZNE profile. Once developed, the Living Wall System's heat exchanging building skin properties will also be secondarily tested for minimum Tvis and light transmitting properties, thereby creating a hi-tech translucent, heat exchanging building skin.

2.2 Material and methods Residential Building Strategies: Single and Multi-Family Heating and Cooling

Residential projects considered envelope or skin dominated buildings require heating in the winter and cooling and/or dehumidification in the summer. Passive strategies for residential projects were developed by the original solar movement in the 20th century and as such have many well vetted approaches mostly focused on passive heating, the dominant residential energy need in many parts of the US. 3 Passive System Stages

The residential approach consists of three stages of passive heating and cooling strategies for summer, winter, spring, fall, day and night. The design prototype contains a thermal labyrinth with three major damper controls to instigate the passive ventilation and heat movement. The damper 1 and 3 supplies the h/c air to the plenum space and damper 2 supplies the air movement to the thermal labyrinth.

Stage 1 is a direct heating and cooling method where damper 1 would be open and outside temperature air would move through a heat exchanger and supply air through the plenum space to the interior. Exhausted hot air moves into the Living Wall or chimney system.

Stage 2, the storage mode, requires the closure of 2 dampers to store the hot or cold within the thermal phase change ePCM labyrinth for later use. This Stage requires supplying the stored heat or cooling to the plenum space by simplified BAS controlling damper operation. In the summer night mode the thermal labyrinth is used for night flush and space cooling.

Stage 3 supplies outside or preconditioned labyrinth air to the heat exchanger which conditions the supply air with stored hot water from the Living Wall or cold water through heat exchanged night flush cool air or through ground coupling. Our goals are for 58 degree F cold water storage and 140 degree F hot water storage.

2.2.1 living wall for residential passive heating

The Living Wall components allow for increased thermal capacity without increasing the mass and thickness of the wall. Additionally, heat from the hi-tech Living Wall similar to a high tech Trombe wall system, will heat adjacent interior space in the winter by supplying the heat by conduction/convection from a one or two story wall system directly to interior spaces. Heating controls would be handled by centrally controlled vents and windows within the interstitial wall. Heat is stored within the ePCM thermal labyrinth. In a similar fashion to a Trombe wall, solar heat is rapidly transported directly into the Living Wall's fluid matrix, circulated, stored and supplied to either radiant floor heating coils or the heat exchanger for the Stage 3 heating cycle.



Figure 2.1: Residential building section with integrated interstitial Living Wall Passive Heating System during winter

2.2.2 residential living wall passive cooling: thermal uplift and ground coupling

On hot summer days, super solar heat from the Living Wall couples with cool air from below causing a thermal uplift and predictable air volume exhaust, drawing cool air through the house via Stage 1, 2 or 3 strategies. The cooled air is supplied through the below floor plenum space and exhausted through the house and into the Living Wall.

Summer nighttime cooling is accomplished by night flush, powered by the stored heat of the Living Wall's thermal components with interior heat exhausted from the ceilings into the exterior double wall system. While the Living Wall utilizes a fluid based interior matrix and PCM to concentrate and collect the solar heat, the convective heat gains provide direct solar heat in the winter and thermal gradient exhaust in the summer. This system requires digital modeling and energy/CFD analysis as one key component of thermal uplift remains the height of the space. Residential project are typically 1 – 3 stories high and therefore produce a very weak thermal uplift. The model will be tested utilizing hot, dry, temperate, and humid climates.

This concept utilizes buoyancy ventilation, through temperature induced stack effect, to draw ground coupled cool air into the living space. Traditional solar walls and chimneys capture daytime solar energy that heats the chimney and the air within it. Our research is focusing on creating a hi-tech wall system where suction created at the base minimally produces enough draw to ventilate with cool air from below. The living wall and ePCM's could allow for storage of this daytime solar energy for release during the night to allow for a continuous stack effect through the night. Utilizing Living Wall technology to produce additional heat from the attic or roof could increase the temperature differential, thus increasing air speed and cold air draw. Wind catcher technologies could also be incorporated to maximize air exchanges through the Venturi Effect as illustrated in Figure 2.5.7. Cold air intake utilizing underground constant temperatures can be further cooled by passing over water.

2.2.3 living wall & evaporative cooling: dry climates

This concept utilizes buoyancy ventilation, or humidity induced ventilation. Hot dry air passes through a moisture membrane at the top of the tower. The air humidifies and cools, becoming more dense, and falls downward to be distributed into the living space. As illustrated in Figure 2.5.8 the heat captured by the Living Wall and associated interstitial wall thermal uplift would provide coupled air flow exhaust creating negative pressure and drawing the cold air down through the cool tower to maximize the cool air supply to the interior. Evaporative cooling technology is effective in dry climates only and not in hot, humid climates.

2.2.4 living wall with desiccant enhanced evaporative cooling: humid climates

Researchers at the National Renewable Energy Laboratory (NREL) are currently working on a breakthrough technology creating cooling with heat. NREL combined desiccant materials, which remove moisture from the air using heat, and advanced evaporative technologies to develop a cooling unit that uses 90% less electricity and up to 80% less total energy than traditional air conditioning (AC). This solution illustrated in Figure 2.5.8 dubbed the Desiccant Enhanced Evaporator (DEVap), also controls humidity effectively to improve the comfort zone. Desiccant materials absorb water from the air and are then dried by thermal heat potentially sourced from the Living Wall's PCM and liquid matrix layers. This technology is still under development, but may couple well with the Living Wall concept. Interfacing with NREL through RASEI at UCB will allow testing of these two new technologies for further DEVap development. Our team is currently researching these available strategies and will choose the best fit to couple technologies with the capabilities of the Living Wall System.

In residential cases, the thermal uplift speed and volume produced by a large temperature gradient (up to 100 degrees F) created by ground coupling with 58 degree air at the bottom and solar heated air column through the height of the wall, creates a quantifiable air movement. Our preliminary tests indicate these Living Wall passive heating and cooling systems produce enough solar heat, move sufficient air, supply sufficient cool air, exhaust enough heat to create true residential Zero Net Energy Buildings (ZNEB) by eliminating mechanical heating, cooling and ventilation systems.

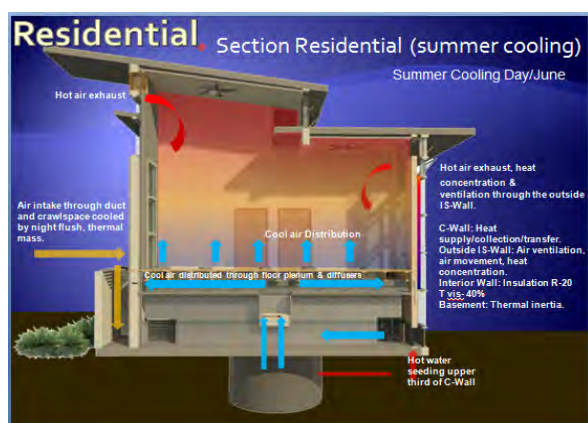


Figure 2.2: Residential building section with integrated interstitial Living Wall Summer Cooling
Thermal uplift and ground coupling

3. THEORY/CALCULATION

Quantification of these strategies and schematics is ongoing. Detailed 3D models in AutoCAD, Revit, Sketch-up and Rhino include the architectural and developmental elements discussed. Animations to illustrate the air and heat movement within the building were generated in Maya utilizing Rhino and detailed

Revit models. Air flow patterns will be analyzed by CFD programs as well as predicted with mathematical models and spreadsheets. Variables within the "kit of parts" will allow future designers to model the workings of their projects with auto-parametric spreadsheets and CFD models.

The impact to commercial building energy use is potentially significant due to the ongoing retrofit of existing buildings and the construction of new ones into the 21st Century. The development of a retrofit and new construction Living Wall System that fundamentally alters the methodology of building conditioning is affordable and interchangeable would have a significant impact on the overall baseline energy footprint of commercial buildings.

4. RESULTS

Energy-10 was used to calculate the heat generated inside the residential interstitial wall. This data was used to calculate air exchanges per hour (ACH) and air speed. The heat generated was compared to a baseline building energy demand. Early results show that the energy generated inside the interstitial wall would meet and exceed energy demand.

4.1 Conclusions

4.2 References

4.2.1 citation in text

4.2.2 web references

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UndergroundLabyrinth:
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Wind Turbine:

Magnetic Levitation Wind Turbine

http://www.enviromission.com.au/EVM/content/technology_technologyover.html

5. TABLES

Table 1 High Efficiency Residential Annual Heating and Cooling Loads – E10

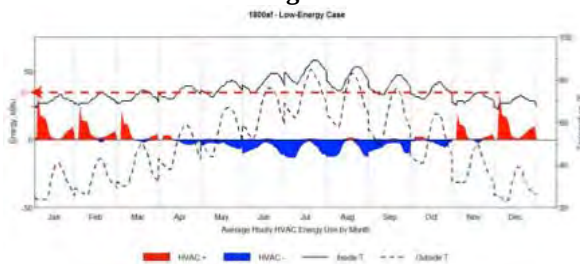


Table 2 Heat Gain Calculations from Residential Living Wall (30' x 12' South Facing)

Solar Glass	A	360 ft ²
Solar Heat Gain on Glass	SG	250 BTU/hr/ft ²
Solar Heat Gain Coefficient	SHGC	0.8
Sunshine for Climate	S	85%
Dust for Climate	D	90%

Net Solar Heat Gain on Glass		
E = SG x A x SHGC x S x D		
55,080	E	Net Energy (Heat Gain, BTU/hr)
360	A	Area of Solar Glass (ft ²)
250	SG	Solar Heat Gain on Glass (BTU/hr/ft ²)
0.80	SHGC	Solar Heat Gain Coefficient
85%	S	% Sunshine for Climate
90%	D	% Dust for Climate

Required	Produced	
30,000 Btu	55,000 Btu	= 183%

Table 3 Annual Hourly Temperatures Inside Interstitial Wall (E-10 winter range: 11 -119 F, summer range: 83-107 F, fall range: 23-132 F)

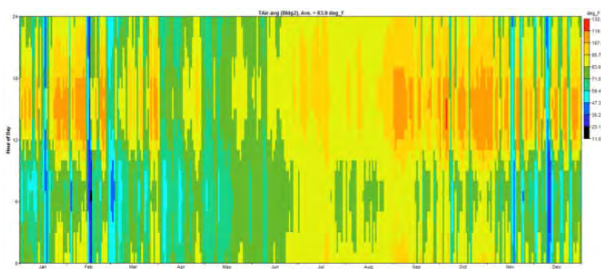


Table 4 Residential Cooling & Ventilation; General Air Distribution Formulas and Calculations

Required Ventilation Rate		
VR = V x AC _H		
1,800	VR	Ventilation Rate (ft ³ /min)
21,600	V	Building Volume (ft ³)
5	AC _H	Air Changes per Hour (AC/hr)

Interior Air Supply for spec'd ACH		
v _A = VR / A _{CS}		
0.8	v _A	Interior Air Speed (miles/hr)
72	v _A	Interior Air Speed (ft/min)
1,800	VR	Ventilation Rate (ft ³ / min)
25	A _{CS}	Stack Cross-Sectional Area (ft ²)

Stack Effect		
Q _{Stack} = [C _d x A _{CS} x (2 x g x h (T _t - T _b) / T _t) ^{1/2}] * e		
2,901	Q _{Stack}	Volume of Ventilation Rate (ft ³ / min)
1.3	v _{Stack}	Air Speed in Stack (miles / hr)
0.65	C _d	Discharge Coefficient
25	A _{CS}	Stack Cross-Sectional Area (ft ²)
32.15	g	Acceleration due to Gravity (ft / s ²)
12	h	Height of Stack (ft)
85	T _t	Temperature Top of Stack (°F)
60	T _b	Temperature Bottom of Stack (°F)
50	e	Efficiency Factor (%)

Building Volume	V	21,600 ft ³
Air Changes per Hour	AC _H	5 AC/hr
Fan Capacity	FC	5,000 cfm
Wind Speed	v _w	8 mph
Coefficient of Effectiveness	K	60 %
Stack Cross-Sectional Area	A _{CS}	25 ft ²
Stack Height	h	12 ft
Temperature Top of Stack	T _t	85 ° F
Temperature Bottom of Stack	T _b	60 ° F
Efficiency Factor	e	50 %

Table 5 Insolation Calculations: Residential Wall – Net Energy Generated at Top of Wall, 12/21 & 6/21.

Net Solar Heat Gain on Glass		
E = SG x A x SHGC x S x D		
24,166	E	Net Energy (Heat Gain, BTU/hr)
360	A	Area of Solar Glass (ft ²)
135	SG	Solar Heat Gain on Glass (BTU/hr/ft ²)
0.85	SHGC	Solar Heat Gain Coefficient
85%	S	% Sunshine for Climate
90%	D	% Dust for Climate

Variables		
Solar Glass	A	360ft ²
Solar Heat Gain on Glass	SG	135BTU/hr/ft ²
Solar Heat Gain Coefficient	SHGC	0.85
Sunshine for Climate	S	85%
Dust for Climate	D	90%

Net Solar Heat Gain on Glass		
E = SG x A x SHGC x S x D		
47,438	E	Net Energy (Heat Gain, BTU/hr)
360	A	Area of Solar Glass (ft ²)
265	SG	Solar Heat Gain on Glass (BTU/hr/ft ²)
0.65	SHGC	Solar Heat Gain Coefficient
85%	S	% Sunshine for Climate
90%	D	% Dust for Climate

Variables		
Solar Glass	A	360ft ²
Solar Heat Gain on Glass	SG	265BTU/hr/ft ²
Solar Heat Gain Coefficient	SHGC	0.65
Sunshine for Climate	S	85%
Dust for Climate	D	90%

1.2. Formula

Net Solar Heat Gain Calculation:

$$E = SG \times A \times SHGC \times S \times D$$

Area of Solar Glass vs. Heat Loss:

$$A = E / (SG \times SHGC \times S \times D)$$

Required Ventilation Rate

$$VR = V \times AC_H$$

Number of Fans:

$$F = VR / FC$$

Air Intake:

$$Ai = VR / vW \times R$$

Air Exhaust

$$Ae = Ai \times 110\%$$

Interior Air Supply for spec'd ACH

$$vA = VR / ACS$$

Stack Effect:

$$Q_{Stack} = [Cd \times ACS \times (2 \times g \times h (T_t - T_b) / T_t)^{1/2}] \times e$$
