J. Cent. South Univ. (2015) 22: 3168−3179
DOI: 10.1007/s11771-015-2854-6

Springer

Energy performance assessment on central air-conditioning system of commercial building: A case study in China

ZHOU Xuan(周璇)[1, 2], LIAN Si-zhen(练斯甄)[1, 2], YAN Jun-wei(闫军威)[1, 2], KANG Ying-zi(康英姿)[1, 2]

1. School of Mechanical & Automotive Engineering, South China University of Technology, Guangzhou 510640, China;
2. City Air-conditioning Energy Conservation and Control Project Technology Research Exploitation Center of Guangdong, Guangzhou 510640, China

Abstract: Energy performance assessment on central air-conditioning system is essential to optimize operating, reduce operating costs, improve indoor environmental quality, and determine whether the retrofitting of the equipment is necessary. But it is difficult to evaluate it reasonably and comprehensively due to its complexity. A "holistic" approach was discussed to evaluate the energy performance of central air-conditioning system for an extra-large commercial building in a subtropical city. All procedures were described in detail, including field investigation method, field measurement instruments, data processing and data analyzing. The main factors affecting energy consumption of air-conditioning system were analyzed and the annual cooling-energy use intensity of this building was calculated and also compared with other shopping malls and other types of buildings in Guangzhou. And COP (coefficient of performance) of chiller, water transfer factor of chilled water system and cooling water system were taken into consideration. At last, the thermal comfort and indoor air quality issues were addressed. The results show that the chilled water pumps are over-sized and the indoor environmental quality should be improved. The purpose of this work is to provide reference for energy performance assessment method for air-conditioning system.

Key words: energy performance assessment; thermal comfort; indoor air quality; central air-conditioning system; energy saving

1 Introduction

Nowadays, large-scale and complex air-conditioning systems are almost ubiquitous in modern buildings. Public concern over rising energy consumption of this system is on the up. Energy performance assessment on central air-conditioning system is the key to optimizing operating system, reducing operating costs, improving indoor environmental quality, and determining whether the retrofitting of the equipment is necessary. But it is difficult to evaluate it reasonably and comprehensively due to its complexity. Therefore, more and more assessment issues involving design, energy consumption and operation efficiency of air-conditioning specific to certain type of building, have been researched by many scholars.

SANTOLI et al [1] assessed the quality of public schools in Rome in order to define possible intervention strategies to reduce energy consumption. QUANG et al [2] developed a multi-component model that can be used to maximize indoor environmental quality to assess the potential improvement of indoor air quality and energy saving under different ventilation conditions in typical air-conditioned office buildings in the subtropical city of Brisbane. HANSEN and RASUL [3] discussed performance assessment and improvement of an existing air-conditioning system of a supermarket. FUNAMI and NISHI [4] proposed a system that could control an air conditioner by three control methods for evaluating and comparing a balance between power saving and comfort. ZHANG et al [5] discussed the energy saving possibility of digital variable multiple air conditioning system in three office buildings in Shanghai and used eQUEST to simulate the annual building performance. Based on the previous works, YANG [6] put forward that the air-conditioning system operation energy efficiency rate (SOEER) could be used to dynamically evaluate the advantage and disadvantage of the optimal operation of programs and a comprehensive energy-saving operation method of the primary pump system with variable or constant water flux was also given. FAN et al [7] proposed a coupled simulation of BES-CFD and performance assessment method for energy recovery ventilation system for office model.

Foundation item: Project(2011B061200043) supported by the Guangdong Provincial Department of Science and Technology, China
Received date: 2014−07−23; **Accepted date:** 2015−01−31
Corresponding author: ZHOU Xuan, PhD, Associate Professor; Tel: +86−20−22236946; E-mail: zhouxuan@scut.edu.cn

Several widespread design and operational issues associated with air-conditioning system, such as improper equipment sizing, low efficiency equipment, imperfect management and imprecise manual control system, have been shown to contribute to electricity load and energy consumption. But so far, few scholars are researching on a comprehensive assessment method for air-conditioning system from an operational point of view.

Located to the south of China, Guangzhou has a humid subtropical climate characterized by hot summer and warm winter. Therefore, the air-conditioning system of commercial buildings in Guangzhou needs to run throughout the year which is the main energy consumer and accounts for about 40% of total building energy consumption, higher than that in other cities [8−9].

In this work, an air-conditioning system of an extra-large shopping mall in Guangzhou was analyzed as a case with energy consumption, energy efficiency, thermal comfort and indoor air quality assessment proposed in the following sections.

2 Background

This shopping mall whose building area is 1.7×10^5 m^2 and air-conditioning area is 1.25×10^5 m^2, is a seven-story building built in 1996, with average daily passenger flow up to 3×10^5 people. Located at city center, the mall is comprised of over 100 international brand-name shops, one large department store, movie theaters, restaurants and cafes. Most of the major brands stores are located on the ground and second floors. There is a department store on the 3rd−5th floors while there are many full-service restaurants and movie theaters on the 6th−7th floors. This extra-large commercial building is equipped with central air-conditioning system which works for more than 10 h per day (h/d), more than 350 d per year (d/a). Electricity is the greatest energy cost item for this commercial building, of which the cost is over US $\$6\times10^6$ per year. And the air-conditioning system is among the highest energy users in this building. A thorough assessment of air-conditioning system was entrusted to City Air-conditioning Energy Conservation and Control Project Technology Research Exploitation Center of Guangdong before the energy retrofits.

After hard work of detailed planning, a full field investigation was carried out in July 21st, 2011, including equipment information, energy consumption of air-conditioning system, specified operating parameters, indoor thermal comfort, indoor air quality, and etc.

The indoor environment issue was taken into account when evaluating the energy performance of air-conditioning system because the thermal comfort and indoor air quality cannot be sacrificed for the sake of energy efficiency. In this case, the main factors affecting thermal comfort which are air temperature and humidity could only be measured manually throughout the entire building during the field investigation.

25 participants attending the field investigation were divided into five groups shown in Fig. 1. Each

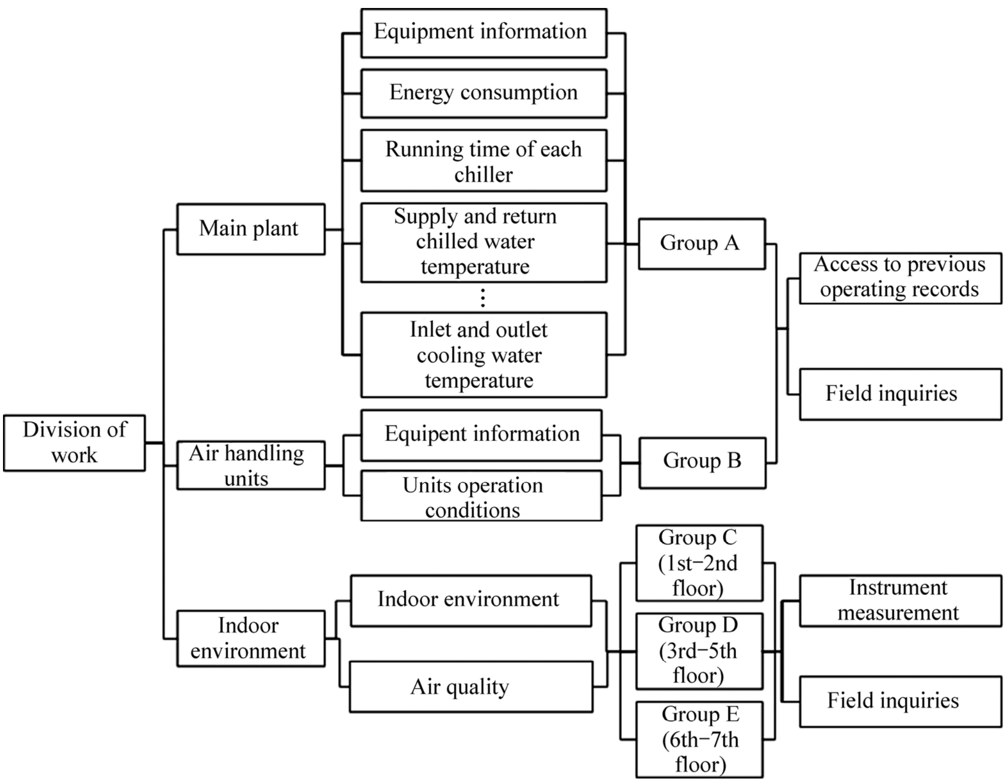

Fig. 1 Task division

group was led by an experienced engineer. Group A undertook field investigation of chiller plant while Group B undertook terminal equipment associated with air-conditioning system. The other three Groups assumed responsibility for the indoor environment parameters measurement in ground and 2nd floors, 3rd and 5th floors, 6th and 7th floors, respectively.

Equipment information and operating status of most AHUs were obtained by field inquiries.

Operating parameters were obtained from previous operating records which were hourly recorded by facilities managers, including the operating hours of each chiller, supply and return chilled water temperature, inlet and outlet cooling water temperatures, part-load factor of chillers and electricity consumption of chillers. In addition, the electricity consumption of cooling water pumps and chilled water pumps were also recorded manually once per day. Furthermore, carbon dioxide data were collected by Building Automation System (BAS) in this building once per hour. In this building, the BAS can only monitor and control AHUs.

3 Configuration of central air-conditioning system

The central air-conditioning system consists of a main plant located on the basement and intermittently spaced AHUs. The main plant is equipped with five water-cooled centrifugal chillers, five chilled water primary pumps, sixteen chilled water secondary pumps, seven cold water pumps and eight cooling towers. The configuration of main equipment is listed in Table 1 and shown in Fig. 2.

As listed in Table 1, the total rated power of the chillers accounts for 52.9% of the total installed capacity, far greater than the rated power of other equipment, such as chilled water pumps, cooling water pumps, cooling tower fans, air handling units, which hold the proportions of 18.7%, 12.8%, 4.4%, 11.1%, respectively.

As shown in Fig. 2, the primary−secondary chilled water system is used in this building. The chilled water primary pumps adopt constant-frequency control strategy on a four-duty/one-standby basis to keep the chiller with constant flow rate while the chilled water secondary pumps are under frequency conversion of constant pressure difference control to coordinate with the size of the load to regulate the flow rate. The chilled water secondary pumps are divided into four groups which are supplying chilled water to the southeast, northeast, southwest and northwest areas of the building respectively.

The cooling water system adopts frequency conversion control mode and these five cooling pumps provide cooling water for four chillers of 2000*RT* with rated flow of 1814 m^3/h each (operated on a four-duty/one-standby basis) and the other two smaller pumps serve the chiller of 1000*RT* with rated flow of 663 m^3/h. In addition, there are eight cooling towers which are also divided into four groups (two units a group), same as the chilled water secondary pumps. Each cooling tower is equipped with four cooling fans.

On each floor, there are a certain number of AHUs supplying the building interior and core areas with cold air.

4 Energy consumption

4.1 Energy consumption of air-conditioning system

A statistics table of energy consumption for main equipment per month was listed systematically, as listed in Table 2.

According to field investigation, all of the AHUs were in normal operating, but none of them was installed with energy metering devices, so their energy

Table 1 Main equipment list

Device	Quantity	Rated power/kW	Cooling capacity, *RT*	Head/m	Flow/($m^3 \cdot h^{-1}$)
Chiller	4	1300	2000	—	—
	1	570	1000	—	—
Chilled water primary pump	5	132	—	21	1450
Chilled water secondary pump	4	110	—	33	943
	4	90	—	30	781
	4	90	—	30	781
	4	55	—	33	493
Cooling water pump	5	250	—	35	1814
	2	75	—	22	663
Cooling tower fan	32	15	—	—	—
AHU	127	1215	—	—	—

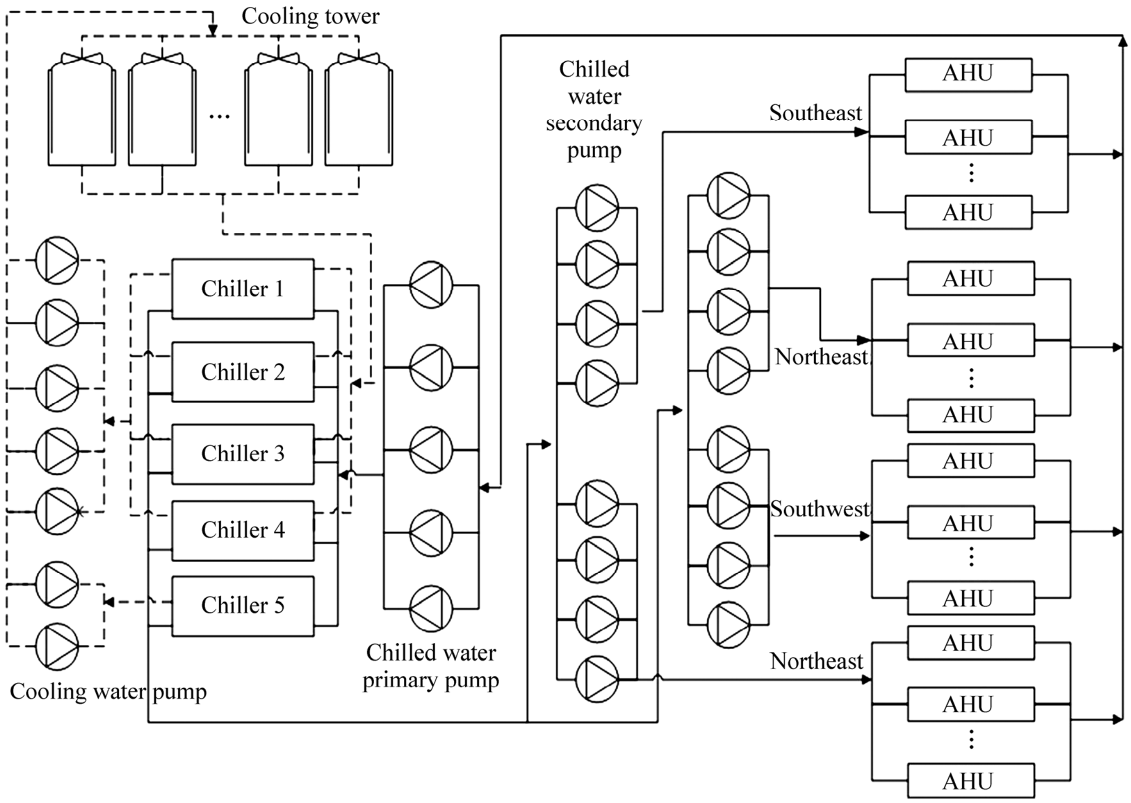

Fig. 2 Schematic of central air-conditioning plant

Table 2 Statistics on energy consumption for main equipment in 2010

Month	Operating time/h	Energy consumption/(10^3 kW·h)					
		Chiller	Cooling water pump	Chilled water pump	Cooling tower fans	AHU	Total
Jan.	209	176.7	45.8	90.2	25.1	253.9	591.7
Feb.	195	147.9	30.0	59.0	34.8	236.9	508.6
Mar.	349	403.4	58.7	123.9	35.4	424.0	1045.5
Apr.	361	490.8	76.8	147.4	56.4	438.6	1210.0
May.	380	628.1	95.2	187.4	74.4	461.7	1446.8
Jun.	401	789.8	97.1	217.9	90.2	487.2	1682.3
Jul.	434	973.4	114.4	233.4	103.6	527.3	1952.1
Aug.	419	998.7	121.5	232.3	104.4	509.1	1965.9
Sep.	417	1005.1	116.2	245.4	111.7	506.7	1985.1
Oct.	410	770.9	117.4	189.6	87.4	498.2	1663.5
Nov.	397	509.4	67.0	148.2	56.9	482.4	1263.9
Dec.	381	363.5	75.7	109.1	45.7	462.9	1057.0
Total	4353	7257.8	1015.9	1983.7	826.0	5289.0	16376.7

consumption cannot be obtained directly. The energy consumption of AHUs was estimated by multiplying the rated power of AHUs in kW by the operating time in Table 2. Regarding the structure of energy consumption in 2010 (Fig. 3), total actual energy consumption of chillers accounted for the highest proportion followed by the cooling tower fans and cooling water pumps consume lower energy consumption than other devices. The energy consumption of chilled water system was higher than the cooling water system. Energy consumption of primary chilled water pumps with fixed-frequency current mode was higher than the secondary pumps with frequency conversion control mode.

As a result, more energy consumption statistics are listed in Table 3.

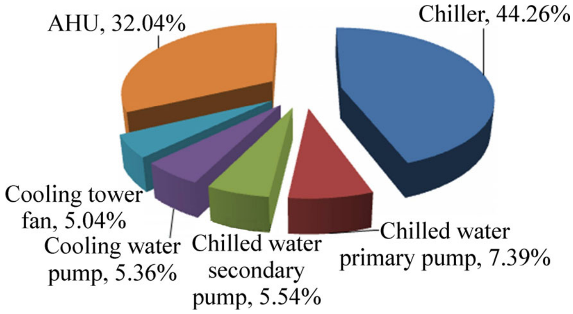

Fig. 3 Energy consumption structure of equipment

Table 3 Total energy consumption statistics

Item	Value
Construction area/m²	1.7×10^5
Air conditioning area/m²	1.25×10^5
Total energy consumption of building/(10^6 kW·h)	4.19×10^4
Air conditioning energy consumption/(10^6 kW·h)	1.64×10^5
Building energy consumption per unit area/ (GJ·m^{-2}·a^{-1})	2.83
Air conditioning energy consumption per unit area/ (GJ·(m²·a)$^{-1}$)	1.59
Proportion of air conditioning energy consumption/%	39.07

4.2 Key factors affecting energy consumption of air-conditioning system

There are numerous energy factors which could influence the energy usage of air-conditioning system, for instance, outside air temperature & humidity, indoor air temperature & humidity and passenger flow. Based on the outdoor monthly average maximum temperatures, the maximum temperature, the minimum temperature, average relative humidity and the minimum relative humidity, which were provided by Guangdong Climate Center. Four broken line graphs are plotted as shown in Fig. 4. It can be seen that the proportion of monthly average energy consumption and outdoor maximum temperature are most similar and nearly followed the same trend. However, a direct relationship between the average relative humidity and energy consumption can not be found in Fig. 5. So, it can be concluded that energy consumption of air-conditioning system is much more strongly influenced by outdoor temperature than by average relative humidity.

As shown in Fig. 4, it can also be noticed that in February, all of these four curves reached the bottom. In this month, the operating time of air-conditioning system was shorter than January mainly because the passenger flow was lower than usual in Guangzhou and most people went back to their hometowns for family reunion during the Chinese Spring Festival holidays. Therefore, energy consumption of air-conditioning system in February was less than that in January. In July, August

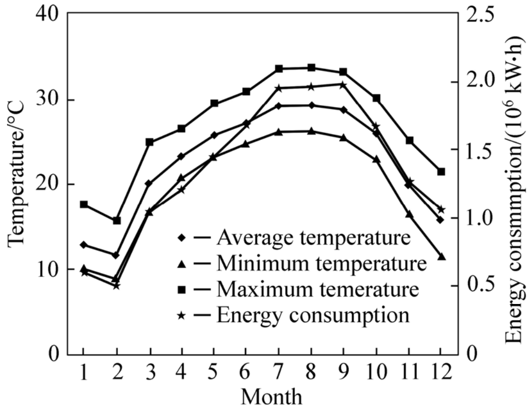

Fig. 4 Monthly average energy consumption of air-conditioning system and monthly outdoor temperature

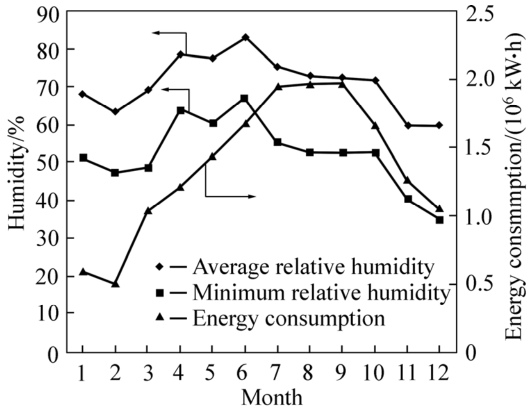

Fig. 5 Monthly average energy consumption of air-conditioning system and outdoor relative humidity

and September, these four curves reached the peak due to the high outdoor temperature.

4.3 Comparison of energy consumption

In this section, total energy and cooling-energy use intensity, and the proportion of air-conditioning energy consumption in total energy consumption were compared with the same and other types of commercial buildings in Guangzhou to assess the energy consumption level of this building.

The total energy use intensity and the cooling-energy use intensity of this case study was the No. 10 building listed in the last line of Table 4 which was on the medium level for all the three malls, with 1.59 GJ/(m²·a) and 2.83 GJ/(m²·a).

As can be seen from Table 4, all the malls including the No. 6, No. 9 and No. 10 buildings had a high level of energy consumption as nearly all the energy consumption indicators were higher than the other types of commercial buildings because of their longer operating hours and higher passenger flow [9]. So, the building location, type and running time should be taken into consideration when the energy consumption evaluation indexes system of the building is established.

Table 4 Comparison of energy consumption to other buildings of different types in Guangzhou, China

No.	Construction area/m²	Air condition area/m²	Building energy consumption per unit area/(GJ·m^{-2}·a^{-1})	Energy consumption of Air-conditioning system per unit area/(GJ·m^{-2}·a^{-1})	Proportion/%	Building type
1	80000	63400	1.209	0.473	31.02	Hotel
2	80500	79000	1.61	0.533	32.46	Office
3	94905	71445	1.852	0.628	25.55	Office
4	55000	40000	1.399	0.854	44.08	Office
5	101026	80000	2.108	0.867	32.55	Office+mall
6	420000	296000	1.49	0.939	44.18	Mall
7	158433	110000	2.017	1.433	49.32	Hotel+Office+Mall
8	117755	106332	3.379	1.836	49.06	Hotel+Office+Mall
9	19491	16200	3.921	1.884	39.93	Mall
Average for other malls			2.7055	1.4115	42.06	
10	168875	125000	2.83	1.59	39.07	

5 Energy efficiency

China is expected to surpass the U.S.A as the world's biggest user of energy consumption of air-conditioning by 2020. Therefore, China released "Economic Operation of Air Conditioning Systems (GBT17981—2007)" in 2007. It defined several indices to evaluate air-conditioning operation, such as air-conditioning energy consumption per unit of conditional floor area, cooling energy use per unit of conditional floor area, EERs, EERr, WTFchw, EERt, COP, WTFcw [10]. Furthermore, some provinces enacted local design standards for energy efficiency of public buildings, such as DBJ 15-51—2007 in which the design transfer factor of energy efficiency rate of chilled water system was specifically defined.

5.1 Water chillers

Average annual COP of water-cooled centrifugal water chiller whose cooling capacity is more than 1163 kW·h should not be less than 4.8 under normal operating conditions while COP should not be less than 5.1 under typical operating conditions [11]. The value of COP, C^{k}_{chl,i_1}, for the i_1-th chiller at the k-th moment and average annual C_{chl,i_1} under normal operating conditions for the i_1-th chiller can be calculated by

$$C^{k}_{\text{chl},i_1}=\frac{Q^{k}_{\text{chl},i_1}}{W^{k}_{\text{chl},i_1}} \tag{1}$$

$$C_{\text{chl},i_1}=\frac{\sum_{k=0}^{H_{\text{T,chl},i_1}} Q^{k}_{\text{chl},i_1}}{\sum_{k=0}^{H_{\text{T,chl},i_1}} W^{k}_{\text{chl},i_1}} \tag{2}$$

Afterwards, the average annual $C_{\text{chl,avg}}$ of chiller system can also be calculated by

$$C_{\text{chl,avg}}=\frac{Q_{\text{chl,total}}}{W_{\text{chl,total}}}=\frac{\sum_{i_1=0}^{N_{\text{chl}}}\sum_{k=0}^{H_{\text{T,chl},i_1}} Q^{k}_{\text{chl},i_1}}{\sum_{i_1=0}^{N_{\text{chl}}}\sum_{k=0}^{H_{\text{T,chl},i_1}} W^{k}_{\text{chl},i_1}} \tag{3}$$

As shown in Fig. 6, in general, the daily COPs of chiller 1 and chiller 2 were higher than those of the other two chillers and the COP of each chiller under the operating mode of multiple chillers was lower than the COP under the operating mode of single chiller in 2010. The average annual C_{chl,i_1} values of these four large chillers were 5.18, 5.08, 4.97 and 5.06, respectively, and the average annual $C_{\text{chl,avg}}$ of chiller system was 5.07. All the chillers can meet the requirements of the standard [11] in which the lower limit was 4.8 under normal operating conditions. Chillers 1 and 2 were more energy efficient and run for more time than chillers 3 and 4.

However, it's not enough to assess energy

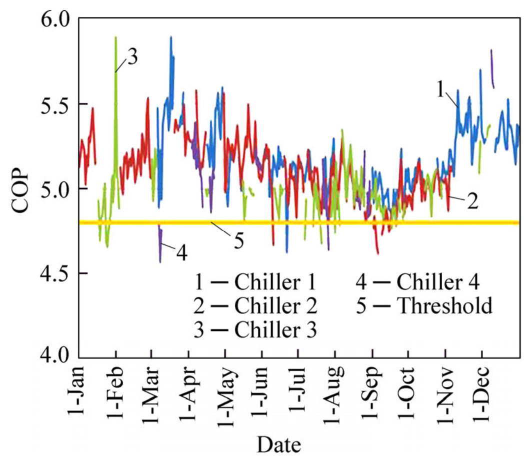

Fig. 6 Daily variation of COP of each chiller in 2010

performance for chiller system only by COP of each chiller. As listed in Tables 5 and 6, running time under different cooling load factor conditions for a single chiller or chiller system as added up.

Table 5 Operating hours and frequency under different cooling load factor conditions for single chiller (Unit: h)

Chiller	Chiller						Total
	40−50	50−60	60−70	70−80	80−90	Over 90	
1	0	52	249	994	925	298	2518
2	8	204	555	1059	761	189	2776
3	23	60	351	400	291	34	1159
4	0	33	82	191	104	13	423
5	0	0	0	0	0	0	0

Note: Cooling load factor, is the ratio of actual cooling capacity to rated cooling capacity of single chiller.

Table 6 Operating hours and frequency under different cooling load factor for chiller system

Cooling load factor/%	Operating hours	Frequency/%
10−20	0	0
20−30	1559	36
30−40	934	21
40−50	1264	29
50−60	542	12
60−70	69	2
Over 70	0	0

According to Table 5, the average annual operating hours of these four 2000*RT* chillers was 1719 h in 2010, with different average annual load factors of 77.19%, 74.69%, 71.36% and 75.20%, respectively. As a result, it can be concluded that the facility managers are very familiar with the chillers' performance, and they can make the right energy efficient decisions.

As listed in Table 6, the operating hours of chiller system with load factor ranging from 20% to 50% accounted for 86% of the total operating hours throughout the whole year while load factor above 50% accounted for 14%.

However, it was worth mentioning that even under maximum cooling load conditions, only three chillers were required to put into operation while another 2000*RT* unit was idled and the 1000*RT* unit was idle too. And the maximum value of actual cooling load was 5460*RT*, only taking up 61% of the design capacity, mainly because the air-conditioning system was designed under the extreme conditions. The oversized design capacity not only reduced the utilization of the equipment, but also increased the initial investment.

In fact, at a low part load, the chiller system achieved higher energy efficiencies by using unequally sized chiller combinations than by equally sized chiller combinations. So, there still existed potential for energy saving through chiller combined optimization of small size chiller and large size.

5.2 Chilled water system

Water transfer factor of chilled water system (W_{chw}) was regulated to evaluate the efficiency of the chilled water system which can be defined as [11]

$$W_{\text{chw}}=\frac{\sum_{i_2=0}^{N_{\text{chw}}}\sum_{k=0}^{H_{\text{T,chw},i_2}}Q_{\text{chw},i_2}^{k}}{\sum_{i_2=0}^{N_{\text{chw}}}\sum_{k=0}^{H_{\text{T,chw},i_2}}W_{\text{chw},i_2}^{k}}\cong\frac{Q_{\text{chw,total}}}{\sum_{i_2=0}^{N_{\text{chw}}}\sum_{k=0}^{H_{\text{T,chw},i_2}}W_{\text{chw},i_2}^{k}}\cong\frac{Q_{\text{chl,total}}}{\sum_{i_2=0}^{N_{\text{chw}}}\sum_{k=0}^{H_{\text{T,chw},i_2}}W_{\text{chw},i_2}^{k}} \tag{4}$$

The total cooling capacity transferred by chiller water pumps was considered the same as the actual cooling capacity produced by chiller system and the actual electricity consumption of chiller water pumps was measured by electrical meter. In the Chinese national standard GBT17981—2007, the lower limit of W_{chw} was 30 for normal operating conditions, and 35 for typical operating conditions.

As shown in Fig. 7, there were large fluctuations in daily average value of W_{chw} between 7.7 and 22.8 and the operating time when W_{chw} fluctuated between 15 and 25 made up 74% of the total operating time, much lower than 30 which was the lower limit regulated in the standard.

There are many factors affecting W_{chw}, such as low supply and return temperature difference of chilled water loop, over-design capacity and control mode. The curve of monthly supply and return temperature difference of the chilled water loop was drawn as shown in Fig. 8.

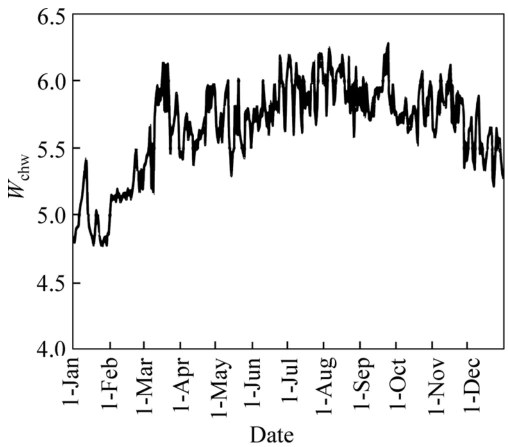

Fig. 7 Daily average of W_{chw} in 2010

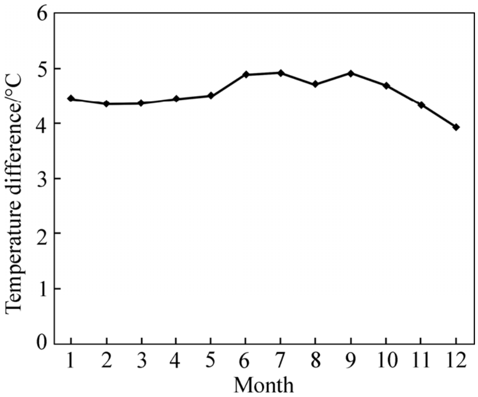

Fig. 8 Monthly average temperature difference between the supply and return chilled water

As shown in Fig. 8, the temperature difference between the supply and return chilled water was relatively high in summer, from June to October, with value close to 5 °C; while in the transition seasons, such as March, April, May and November, the temperature differences were close to 4.5 °C. In December, the temperature difference was even less than 4 °C. The average chilled water supply and return temperature difference was 4.53 which cannot satisfy the design requirement.

Additionally, according to the design standard [10], under standard conditions, energy efficiency ratio can be defined as

$$E_{\mathrm{chw}} = 0.002342(H_{\mathrm{chwp,avg}} + H_{\mathrm{chws,avg}}) / \left(\Delta T \cdot \frac{\eta_{\mathrm{chwp,avg}} + \eta_{\mathrm{chwp,avg}}}{2}\right) \tag{5}$$

where $H_{\mathrm{chwp,avg}}$, $H_{\mathrm{chws,avg}}$, $\eta_{\mathrm{chwp,avg}}$ and $\eta_{\mathrm{chws,avg}}$ can be calculated as the design standard defines [10].

And for the primary−secondary pumping water system in this building, the calculated value was 0.0315, much higher than the upper limit of energy efficiency ratio which was 0.02722 [10].

Therefore, it can be concluded that the low efficiency of chiller water system was mainly caused by the constant-frequency control mode and the oversized design. For energy saving, it is necessary to change the chilled water pumps into smaller ones.

5.3 Cooling water system

Water transfer factor of cooling water system (W_{cw}) is regulated to evaluate the efficiency of the cooling water transfer system defined as.

$$W_{\mathrm{cw}} = \frac{\sum_{i_3=0}^{N_{\mathrm{cw}}} \sum_{k=0}^{H_{\mathrm{T,cw},i_3}} Q_{\mathrm{cw},i_3}^{k}}{\sum_{i_3=0}^{N_{\mathrm{cw}}} \sum_{k=0}^{H_{\mathrm{T,cw},i_3}} W_{\mathrm{cw},i_3}^{k}} \cong \frac{Q_{\mathrm{cw,total}}}{\sum_{i_3=0}^{N_{\mathrm{cw}}} \sum_{k=0}^{H_{\mathrm{T,cw},i_3}} W_{\mathrm{cw},i_3}^{k}}$$

$$\cong \frac{Q_{\mathrm{chl,total}}}{\sum_{i_3=0}^{N_{\mathrm{cw}}} \sum_{k=0}^{H_{\mathrm{T,cw},i_3}} W_{\mathrm{cw},i_3}^{k}} \tag{6}$$

In the standard GBT17981—2007, the lower limit of W_{cw} is 25 for normal operating conditions, and 30 for typical operating conditions.

Figure 9 shows that there were large fluctuations of daily average value of W_{cw} between 15.3 and 59.4. The cooling water pumps were controlled by the mode of load-adaptive variable frequency control, based on supply and return temperature difference of the cooling water loop. In the whole year, the average chilled water supply and return temperature difference was 4.84, close to 5 °C, and the curve of monthly supply and return temperature difference of the cooling water loop is shown in Fig. 10.

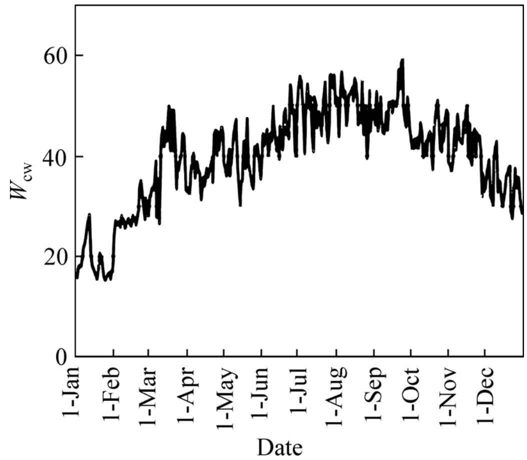

Fig. 9 Daily average of W_{cw} in 2010

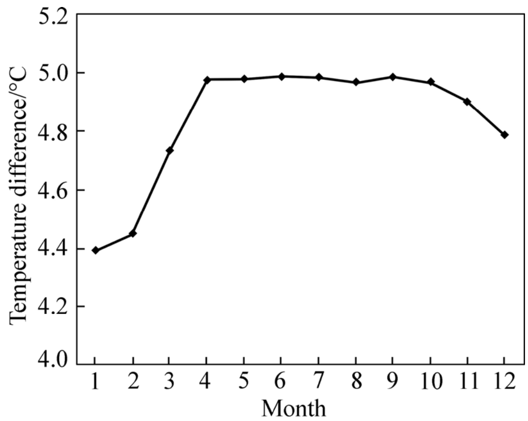

Fig. 10 Monthly average cooling water supply and return temperature difference

The average cooling water transfer factor for normal operating condition was 42.1, much higher than the standard threshold. In summary, it can be included that the cooling water system is much more energy efficiency than chiller water system due to its reasonable design and frequency conversion control mode.

6 Thermal comfort and indoor air quality

6.1 Thermal comfort

In China, ISO 7730 standard is also widely used for indoor thermal environment assessment which describes the PMV (predicted mean vote) and PPD (predicted percentage dissatisfied) indices. PMV and PPD indices which are elaborated by Professor FANGER can specify acceptable conditions for thermal comfort [12]. To calculate PMV value, it is necessary to measure the indoor air temperature and relative humidity whose measurement must cover different separate areas in the shop, such as special stores, restaurant, corridor. Several issues need to be discussed, e.g., measuring points for different separate areas, measure time for each point, processing of measurement data at different time.

6.1.1 Measuring points

Measuring points were selected to measure indoor temperature and relative humidity inside the shopping mall based on the selection principle of measuring points presented in GB/T 18204.13 [13]. The details were as follows.

If separated area<16 m², then set up 1 measuring point;

Else if 16 m²≤ separated area ≤ 30 m², then set up 2 measuring points;

Else if 30 m²≤ separated area ≤ 60 m², then set up 3 measuring points;

Else if separated area>60 m², then set up 5 measuring points.

There were totally 123 stores in this building and 265 measuring points were set up.

6.1.2 Measuring instruments

The indoor temperature and relative humidity were measured by Fluke-971 temperature humidity meter whose temperature accuracy was typically ± 0.5 °C from 0 °C to 45 °C and the relativity humidity accuracy was ±2.5% (RH) from 10% to 90%.

6.1.3 Measuring time

The indoor environment measurement process was divided into four periods due to variation of population flow and outdoor temperature, which is from 10:00 to 12:00, 12:00 to 14:00, 15:00 to 17:00 and 19:30 to 21:30, respectively. That is to say, each measuring point had one measurement value during each period and had four measurement values totally.

6.1.4 Data processing

The average temperature and relative humidity could be illustrated as follows:

$$T^m_{l_1,l_2}=\frac{\sum_{l_3=1}^{P_{l_1,l_2}} T^m_{l_1,l_2,l_3}}{P_{l_1,l_2}} \tag{7}$$

$$T^m_{l_1}=\frac{\sum_{l_2=1}^{S_{l_1}} (T^m_{l_1,l_2})\cdot A_{l_1,l_2}}{\sum_{l_2=1}^{S_{l_1}} A_{l_1,l_2}} \tag{8}$$

$$T_{l_1,\text{avg}}=\frac{\sum_{m=1}^{4} T^m_{l_1}}{4} \tag{9}$$

$$\varphi^m_{l_1}=\frac{\sum_{l_3=1}^{P_{l_1,l_2}} \varphi^m_{l_1,l_2,l_3}}{P_{l_1,l_2}} \tag{10}$$

$$\varphi^m_{l_1}=\frac{\sum_{l_2=1}^{S_{l_1}} (\varphi^m_{l_1,l_2})\cdot A_{l_1,l_2}}{\sum_{l_2=1}^{S_{l_1}} A_{l_1,l_2}} \tag{11}$$

$$\varphi_{l_1,\text{avg}}=\frac{\sum_{m=1}^{4} \varphi^m_{l_1}}{4} \tag{12}$$

Based on the Chinese National Code GB 50019—2003 [12], in summer, the thermal comfort range varied from 23 °C to 26 °C with corresponding relative humidity (RH) levels from 65% to 40%. According to statistics, the ranges of the temperature and relative humidity on each floor are listed in Tables 7 and 8, respectively. In addition, the average indoor temperature and relative humidity on each floor are shown in Fig. 11. The total average temperature and relative humidity were 23.66 °C and 62.95%.

As listed in Table 7, the average indoor temperature mainly varied from 23 °C to 26 °C. There was 24.2% of the total area whose temperature was lower than lower limit of indoor temperature while 1.37% higher than limit of indoor temperature. That is to say, over 25% area was beyond the thermal comfort.

Table 7 Ratio (%) of different temperature ranges on each floor

Floor	Ratio/%				
	T≤23 °C	T∈(23 °C, 24 °C]	T∈(24 °C, 25 °C]	T∈(25 °C, 26 °C]	T≥26 °C
First	2.40	41.83	45.19	9.62	0.96
Second	17.92	64.36	15.06	2.66	0.00
Third	25.88	59.85	12.24	2.03	0.00
Fourth	22.44	32.27	35.76	8.89	0.64
Fifth	42.98	34.48	18.03	3.88	0.63
Sixth	39.35	38.00	19.83	1.39	1.43
Seventh	18.40	22.82	39.19	13.69	5.90
Average	24.20	41.95	26.47	6.02	1.37

Table 8 Ratio of different relative humidity ranges on each floor

Floor	Ratio/%			
	≤40%	(40%, 65%]	(65%, 80%]	>80%
First	0.00	66.35	31.73	1.92
Second	0.00	89.22	10.78	0.00
Third	0.00	85.70	14.30	0.00
Fourth	0.00	77.37	22.63	0.00
Fifth	0.00	76.38	23.62	0.00
Sixth	0.00	54.05	45.95	0.00
Seventh	0.00	56.35	43.65	0.00
Average	0.00	72.20	27.52	0.27

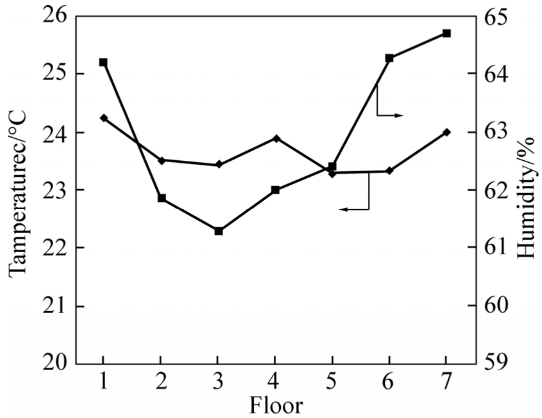

Fig. 11 Average temperature and relative humidity on each floor

As listed in Table 8, the relative humidity values mainly fluctuated between 40% and 80%. Moreover, the relative humidity values at 27.79% measuring points exceed 65% which was also beyond the thermal comfort. Specially, the relative humidity values on the 1st, 6th and 7th floors were far beyond the thermal comfort zone.

As shown in Fig. 11, the average temperature and relative humidity on the 1st floor were higher than those of other floors because the four entrances on the first floor enhanced heat transfer between indoor and outdoor.

The average temperature of the 7th floor was the second highest because there were lots of restaurants which produced a large amount of heat on this floor. The measuring points, of which the temperatures reached up to 26 °C accounted for 5.9% of total while the relative humidity value above 65% accounts for 43.65% of total. Meanwhile, on the 4th floor, there were lots of jewelry counters whose nature had high light requirements. As a result, the average temperature of this floor reached the third-highest because the lights dissipate large amount of heat.

In the 5th and 6th floors, the area of which the average temperature was lower than 23 °C took up 42.98% of total area. By the detailed investigation, there were two main reasons: one was that the air supply outlets are installed too densely; the other was that the population density was lower than other floors, but the air flow was not regulated with the population density.

Furthermore, it was obvious that the temperature and relative humidity with high population density, such as discount store, boutique and restaurant, were generally higher than other areas.

6.1.5 Distribution of PMV

According to ISO 7730, a PMV value between +0.5 and −0.5 provided thermal comfort conditions. But in China, since energy conservation in relation to thermal comfort became a large issue, some researchers proposed that the range of PMV could be adjusted to −0.75<PMV< 0.75 [13].

Before calculating PMV, several assumptions were made to calculate PMV by Fanger's equation [14]: M=69.78 W/m^2, I_{cl}=0.5 clo (1clo=0.155 m^2·°C/W), W=0, t_a=t_r, v=0.2 m/s.

The four values of PMV at each measuring point for different periods were calculated which were within the range of [2, 0] and displayed in different colors as shown in Fig. 12. And there were a large number of PMV values within the range from −2 to −0.75. The PMV values of 66.79% measuring point in the morning were within this range, 62.45% at noon, 58.11% in the afternoon and 59.62% at night which were beyond the thermal comfort range.

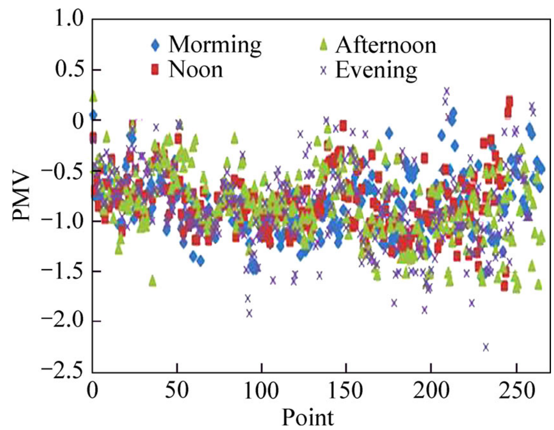

Fig. 12 Distribution of PMV for each period

In light of these facts, the indoor temperature was too low to make people feel comfortable. In addition, lower indoor temperature meant higher energy consumption. So, the indoor temperature could be set higher for energy saving and thermal comfort. For example, it was concluded that the mean energy consumption reduction corresponding to a 1 °C increase of the set-point at 1 °C(from 22 °C to 28 °C) was about 6.14% [15]. From an energy point of view, if the indoor temperature set-point of the field investigation day was increased by 2.34 °C, which meant the total average temperature increased from 23.66 to 26 °C, the energy consumption of air-conditioning system of that day would be decreased by 14.36%.

6.2 Carbon dioxide concentration

Thermal comfort and CO_2 are two important aspects of indoor air quality that receives considerable attention. To a certain extent, the concentration of carbon dioxide is an indicator of the indoor air quality in public buildings. In this mall, several CO_2 concentration sensors were installed in return air ductwork of the southeast, northeast, southwest and northwest areas on each floor and the data of CO_2 concentration were collected by BAS. Furthermore, the minimum opening position of fresh air valve were determined by the CO_2 measurements. From the previous records recorded by BAS, the statistics of actual average indoor CO_2 level was up to 0.326%, little higher than 0.3% which was the upper limit set by facilities managers for energy-saving, and it is far higher than 0.07% which was suggested by standard ANSI/ASHRAE 62.1—2013.

Although in summer, fresh air reduction can lead to energy-saving, it cannot sacrifice indoor air quality. Therefore, the setting of CO_2 concentration level should be adjust to a reasonable value and the fresh air valve opening should be increased in order to improve the indoor air quality.

7 Conclusions

1) The energy use intensity of the building and the cooling-energy use intensity of the air-conditioning system are on the medium level for this type of building, but much higher than the other types of buildings in Guangzhou.

2) As far equipment performance of the chiller plant, the chilled water transfer factor is much lower than the lower limit in the standard because of its oversized design.

3) Additionally, more than 50% PMV values of the measuring points are below the thermal comfort range, which means that the temperature could increase, and the CO_2 concentration level should decrease. As for the field investigation day, if the total average temperature increased from 23.66 °C to 26 °C, the energy consumption of air-conditioning system of that day would decrease by 14.36%.

4) There is still space for further improvement in replacing chilled water pumps, increasing thermal comfort and indoor air quality and optimizing operating systems of the air-conditioning system for this mall.

Acknowledgements

The authors would like to thank the contribution and hard work of other members of the project "Operational energy efficiency monitoring and optimal control of air conditioning system" for permission to use information from the field investigation.

Symbol

EERs	Energy efficiency ratio of air-conditioning system
EERr	Energy efficiency ratio of refrigeration system
W_{chw}	Chilled water transfer factor
EERt	Energy efficiency ratio of air-conditioning terminal
COP	Coefficient of performance
W_{cw}	Cooling water transfer factor
H	Pump head (m)
F	Rated flow (m^3/h)
ΔT	Supply and return temperature difference (°C)
η	Pump efficiency under standard conditions (%)
Q	Cooling capacity (kwh)
W	Energy consumption (kwh)
H_T	Total operating hours
N	Total number
T	Measurement value of temperature (°C)
φ	Measurement value of relative humidity (%)
P	Number of measuring points
A	Separate area
S	Number of separate area

Subscripts

chl	Chiller
chw	Chiller water pump
chwp	Primary water pump
chws	Secondary water pump
cw	Cooling water pump
avg	Average
i_1	i_1^{th} chiller
i_2	i_2^{th} chiller water pump
i_3	i_3^{th} cooling water pump
i_4	i_4^{th} primary chiller water pump
i_5	i_5^{th} secondary chiller water pump
l_1, l_2, l_3	l_3^{th} point of the l_2^{th} separate area on the l_1^{th} floor

Superscripts

k	k-th operating moment of air- conditioning

plant

m — m-th measuring moment of indoor environment

References

[1] de SANTOLI L, FRATICELLI F, FORNARI F, CALICE C. Energy performance assessment and a retrofit strategies in public school buildings in Rome[J]. Energy and Buildings, 2014, 68(1): 196−202.

[2] QUANG T N, HE Cong-rong, KNIBBS L D, DEAR R D. Co-optimisation of indoor environmental quality and energy consumption within urban office buildings [J]. Energy and Buildings, 2014, 85(12): 225−234.

[3] HANSEN M, RASUL M. Performance assessment and improvement of an existing air conditioning system of a supermarket: A case study on Bi-Lo supermarket [C]// Energy and Environment III: Proceedings of the 3rd IASME/WSEAS International Conference on Energy and Environment, Cambridge: University of Cambridge, 2008: 55−60.

[4] FUNAMI K, NISHI H. Evaluation of power consumption and comfort using inverter control of air-conditioning [C]// IECON 2011-37th Annual Conference on IEEE Industrial Electronics Society Melbourne: IEEE, 2011: 3236−3241.

[5] ZHANG Dong-liang, ZHANG Xu, CAI Ning. Study on energy saving possibility of digital variable multiple air conditioning system in three office buildings in Shanghai [J]. Energy and Buildings, 2014, 75(6): 23−28.

[6] YANG Wen-hui. Study on comprehensive energy-saving operation model of public buildings air-conditioning system [D]. Chongqing: College of Power Engineering, Chongqing University, 2008. (in chinese)

[7] FAN Yun-qing, HAYASHI T, ITO K. Coupled simulation of BES-CFD and performance assessment of energy recovery ventilation system for office model [J]. Journal of Central South University, 2012, 19(3): 633−638.

[8] YANG Chang-zhi, WU Xiao-yan, LI Wen-jing. Energy consumption status and energy saving potentials of air conditioning systems in public buildings of Changsha [J]. HV&AC, 2005, 35(12): 39−43.

[9] ZHOU, Xiao-qing, LIU Fang, CHEN Wei-qing, YANG Shu-rong, DING Yun-fei. Investigation and study on energy consumption of public buildings in Guangzhou [J]. Building Science, 2007, 23(12): 76−80.

[10] DBJ 15-51—2007. Design standard for energy efficiency of public buildings[S].

[11] GB/T 17981—2007. Economic Operation of Air Conditioning Systems [S]. (in Chinese)

[12] GB 50019—2003. Code for Design of heating, ventilation and air-conditioning [S]. (in Chinese)

[13] GB/T 18204.13—2003, Methods for determination of air temperature in public places [S]. (in Chinese)

[14] ISO International Standard 7730—2005, Third edition. Ergonomics of the thermal environment — Analytical determination [S].

[15] KONGKIATUMPAI P. Study of impact of indoor set point temperature on energy consumption of air-conditioner and green-house gases emission [R]. Thailand: School of Energy Technology Program, King Mongkut's University of Technology Thonburi, 1999.

(Edited by DENG Lü-xiang)