Indoor Constant Temperature Control Method of Intelligent Building Based on Bi-linear Control Algorithm

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***Abstract*—With the continuous improvement of technological innovation in construction industry, intelligent building management on the internal environment of the building is also realized. Through the simulation analysis of the building thermal environment, comprehensive reflection on the influence characteristics of the various disturbances under the thermal inertia of the envelope structure is shown in this paper. Based on the bi-linear simulation analysis method, optimal construction method for the temperature regulation of the heat supply end in heating system is explored for the first time, and the application and effect of the optimized construction method is analyzed through specific cases. Therefore, research in this paper has strong engineering application value for optimizing the operation regulation and promoting energy saving in heating system.**

Keywords-Bi-linear Control; Intelligent Building; Constant Temperature Control

I. INTRODUCTION

It has practical effects for assisting energy-saving diagnosis and measuring energy-saving transformation that benchmark evaluation method for smart building energy consumption is established, which is of great significance to establish a perfect system for building energy consumption quota management in China, and it is also one of the important indicators to measure the effectiveness of building energy conservation in China.

Nowadays reducing the energy consumption of building heating in northern cities has become the focus on building energy conservation work. In the past 20 years, China has carried out a lot of theoretical and practical work to enhance the thermal insulation performance of the envelope structure, which has played a role in reducing the actual heating demand of the building. However, after the insulation performance of the envelope structure is enhanced so that the actual heat demand of the building is reduced, if the heating system is not well matched to the actual heat demand of the building, there will still be excessive heating in the system. In heating season, the weighted average external temperature changes from -8.9℃ to 9.1℃. For northward room, the daily average base room temperature increases from 12.5℃ to 23.2℃, with a increase of 10.7 ℃, while average daily heating temperature reduces from 5.8℃ to 1.6℃, with a reduction of 4.2℃. Besides it, for the southward room, the daily average room temperature rise rate and the average heating temperature reduced rate is 11℃ and 4.2℃ , while for the north direction room without internal disturbance, the two variations are 14.4℃ and 8.6℃. The inconsistency between the basic room temperature and the temperature rise of the heating which is mentioned above leads to instability in room temperature of the heating room[1-3].

Based on the management of the internal environment of smart buildings, indoor constant temperature control mode of wind condensation and dehumidification optimization mode is studied to propose bi-linear control method according to indoor ventilation, air conditioning and integrated heating management in this paper. What's more, the optimization of energy consumption in intelligent management system is achieved by optimizing the temperature regulation of the heat supply end in heating system.

II. INDOOR AIR CONDENSATION DEHUMIDIFICATION TECHNOLOGY OF SMART BUILDING

A. *Condensation Dehumidification Technology Application*

The fresh air of the project adopts the condensing and dehumidifying method. Due to the large amount of exhaust air in the intelligent building, the indoor exhaust air is used to take away the condensation heat, so no additional outdoor unit or cooling water system is needed. In order to further reduce the exhaust air temperature, humid film humidification is set before the condenser, and the exhaust air temperature is lowered by the equal humidification process so that the condensation temperature is further reduced. Besides it, the device exchanges energy through the refrigeration system where the fresh air has no contact with the exhaust, which avoids the risk of cross-infection[4].

Since all the latent heat in the room is borne by the fresh air, it is necessary to check whether the air supply moisture content calculated by the indoor fresh air volume exceeds the capacity of the dehumidification equipment. So according to the formula:

$$D_n - D_o = \frac{w_n}{\rho G_w} \tag{1}$$

In formula (1) D_n is the indoor moisture content (g/kg), W_n represents the indoor wet load (g/h), and G_w refers to the indoor fresh air volume. High temperature varies from 1℃ to 2℃. At 25℃, the relative humidity (1) is outdoor humidity.

Calculated by formula (1), the moisture content of the fresh air after being pre-cooled by the condensing and dehumidifying equipment is within the processing capacity of units, and the results are basically the same, which indicates that the amount of fresh air required for a smart building can be well matched to the amount of fresh air required for dehumidification. Besides it, cooling load of the condensing and dehumidifying equipment in new wind turbine unit is 2005.82 kW. Since the condensing and dehumidifying equipment is an air-cooled condenser where COP=3.2W/W, power consumption is also a linear relationship between fan power and total pressure in the case of constant air volume and efficiency. In addition to the condensing and dehumidifying equipment, new fan unit in THIC system has the same functions as the conventional new fan unit. There are generally three rows of coils in condensing and dehumidifying equipment, and according to the formula, total pressure loss gauge in new fan unit is fully pressurized to 800 Pa[5].

$$N = K\frac{G \cdot P}{3.6\eta_m} \quad (2)$$

In formula (2) N is the unit power consumption (kW), G refers to the unit full pressure (Pa), η represents transmission efficiency of the machine (%), and K indicates the added power of the new fan. In addition, due to the small change in the function of exhaust duct, the exhaust fan power is negligibly compared to the conventional unit.

B. Dry Fan Coil

The dry fan coil in THIC system mainly undertakes indoor sensible heat, and the dry coil adopts the countercurrent high efficiency heat exchange copper tube where tube layout and tube design are different from wet coils and cannot be replaced by ordinary wet coils.

According to the comparison between the dry fan coil and the wet fan coil, cooling capacity of the dry fan coil unit is half of the wet fan coil under the same working condition and air volume, and the calculation results are shown in Table 1 and Table 2.

TABLE I. UNIT COOLING CAPACITY POWER CONSUMPTION OF DRY FAN COIL

model	Air supply volume (mid-range)/m3/h	Cooling capacity (mid-range)/kW	power/W	unit cooling capacity power consumption W/kW
FP-02	340	2.13	40	0.05325
FP-03	520	3.26	58	0.05621
FP-04	680	4.17	71	0.05873
FP-05	850	4.84	83	0.05831
FP-06	1020	5.81	108	0.0538

TABLE II. UNIT COOLING CAPACITY POWER CONSUMPTION OF WET FAN COIL

device ID	Equipment type and specification	equipment power/kW	amount	Remarks
L-B1-1~3	Water-cooled variable frequency centrifugal chiller，Cooling capacity：2532kW（720USRT）	329.2	3	COP：7.69
L-B1-4	Water-cooled variable frequency screw chiller，Cooling capacity：1231kW（350USRT）	170.0	1	COP：7.24
L-B1-5	Water-cooled variable frequency screw type cold water heat recovery unit,Cooling capacity：1319kW（375USRT）	210.0	1	COP：6.28
B-B1-1~4	Chilled water pumps，flow：480m3/h Head：40m	75.0	4	Three-in-one
B-B1-5~6	Chilled water pumps，flow：230m3/h Head：40m	37.0	2	one-in-one

The dry fan coil mainly bears the indoor sensible heat load, and the wet fan coil takes up all the indoor load. In addition, dry fan coil power consumption is N1=Qs1×0.02559=84.75kW, and wet fan coil power consumption is N2=Qs×0.05516=231.26kW.

C. Energy Saving Analysis

1) Main equipment power conventional air conditioning system is adopted.

The total installed capacity of the equipment room and the power consumption of the pump have decreased when THIC is used. Compared with the conventional air conditioning system, the total power distribution of the refrigeration system equipment (including condensing dehumidification equipment) is saved by 8.1%, while the total power distribution of the end fan is 46%, which obviously shows the total power distribution in the THIC system is saved by 10.80%[6-7].

2) Operational adjustment

The changes in climate parameters throughout the year are shown in Figures 1 and 2. During the transitional season, the dry bulb temperature varies from16.5℃ to 25℃ and the wet bulb temperature varies from 12.6℃ to 19.2℃. What's more, the number of hours between the regions is around 3000h, which accounts for about 1/3 of the annual hours. Additionally, outdoor fresh air enthalpy in this interval is less than indoor devaluation, and the moisture content is less than 12g/kg, which shows that fresh air has a certain cooling and dehumidifying capacity so that the new wind turbine group does not need to be pre-cooled or dehumidified. Moreover, it is possible to eliminate the residual heat and humidity in the room only through turning on the condensing and dehumidifying equipment or increasing the amount of fresh air, where the energy saving rate of the system will be further improved. Meanwhile, the new fan group is dispersed and arranged, which will be easier to be controlled.

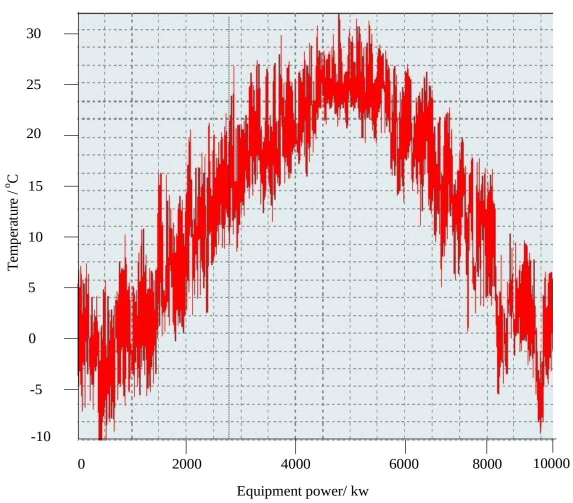

Figure 1. Annual Dry Bulb Temperature Change Chart

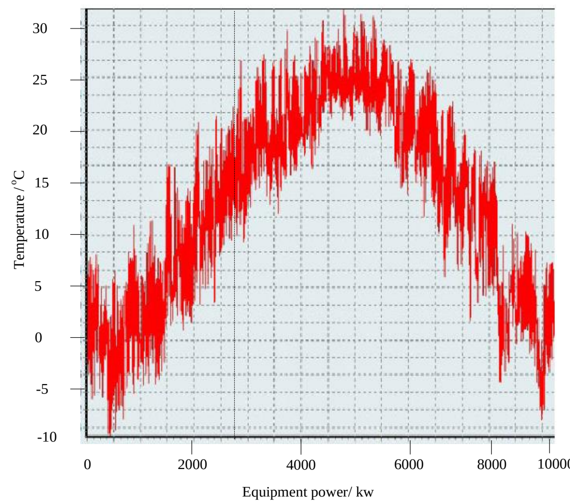

Figure 2. Annual Wet Bulb Temperature Change Chart

III. OPTIMIZATION ADJUSTMENT ON MODE CONSTRUCTION

For a given building form, the room temperature of the room and its variation with the external temperature should be fixed under the conditions of external disturbance and internal disturbance. Therefore, in order to maintain a constant room temperature in the heating room, the range of change in heating temperature rise should be adjusted to make it consistent with the change in the base room temperature[8-9].

Taking the northward room in the middle layer as an example, the variation of the heating temperature rise of 4.6℃ is much smaller than the variation of the basic room temperature of 10.7℃ under the water supply temperature mode existing at the end of the heat dissipation. Thus, the influence of the heating system on the room temperature fluctuations cannot offset the effects of external disturbances and internal disturbances on the room temperature fluctuations, which in turn leads to instability of the room temperature in the heating room. In this regard, it is necessary to increase the range of changes in room heating temperature rise. From the perspective of formation reasons, for a certain architectural form, the change in room heating temperature rise directly depends on the water supply temperature at the end of the heat dissipation, and daily average water supply temperature T_g and the weighted average external temperature T_w exhibit a linear relationship with $T_g = -aT_w + b$. Therefore, the key to maintaining a constant room temperature is to adjust the water supply temperature mode, and the key to optimizing the configuration of the water supply temperature mode in heating system lies in the reasonable determination of the coefficients *a* and *b*.

A. Analysis on Water Supply Temperature Range

For the linear relationship of $T_g = -aT_w + b$, in order to determine a reasonable coefficient a, it is first necessary to determine the water supply temperature range corresponding to the weighted average external temperature variation range. Therefore, under the condition of stable water flow, the relationship between the daily average heating temperature rise of the room and the daily average water supply temperature at the end of heat dissipation is analyzed in terms of the northward facing room in the middle layer, which is shown in Fig. 3.

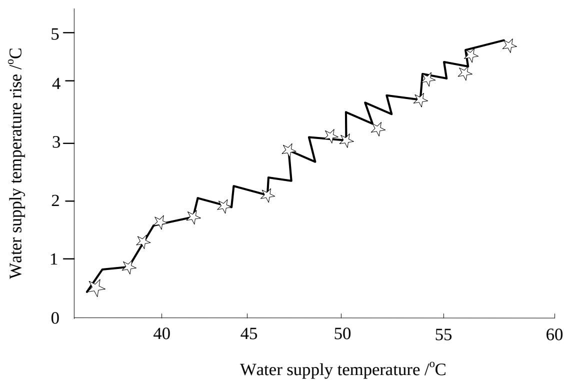

Figure 3. Daily Average Heating Temperature Rise with Daily Average Water Supply Temperature Change

It can be seen from Fig. 3 that for the heating system with quality regulation operation, the room average heating temperature rise and the daily average water supply temperature also show a good linear relationship, which is consistent with the linear relationship between heating temperature rise and weighted average external temperature as well as water supply temperature and weighted average external temperature in above analysis. Formula (3) can be got:

$$\Delta T_{up} = 0.23\Delta T_g \quad (3)$$

In formula (3) ΔT_{up} is the fluctuation range of the heating temperature rise, and ΔT_g refers to the variation range of the daily average water supply temperature. Therefore, for the northward facing room of the middle layer, if the fluctuation range of the room heating temperature rise in the optimized mode is consistent with the fluctuation range of the base room temperature of 10.7℃, the corresponding water supply temperature variation range will be shown in formula (4).

$$\Delta T_g = \frac{\Delta T_{up}}{0.23} = 46.5\ ^{o}C \quad (4)$$

B. Analysis on Coefficient a

According to the above calculation results, it is necessary to ensure that the water supply temperature changes ΔT_g is 46.5 ℃ to maintain the room temperature stability of the northward room in the middle layer and make base room temperature decrease the same as the heating temperature rise under the condition where the weighted average external temperature changes from -8.9 ℃ to 9.1 ℃. Formula (5) will be obtained through further derivation.

$$a = \Delta T_g / \Delta T_w \quad (5)$$

Therefore, the outdoor weighted average temperature variation range ΔT_w of $9.1\text{-}(\text{-}8.9) = 18\ ^{o}C$ and the daily average water supply temperature change width ΔT_w of 46.5℃ are brought in, and the coefficient a=2.6 required for the water supply temperature optimization adjustment mode will be obtained.

C. Analysis on Coefficient b

The reasonable determination of the coefficient a ensures that the room temperature fluctuation range of the room is basically the same as the fluctuation range of the room heating temperature rise so that the daily average room temperature of the heating room will be basically stable, and the next question is how to determine the coefficient b so that the daily average room temperature of the heating room can be maintained within the design range of 18 ± 1 ℃. In view of the base room temperature and external temperature as well as the room temperature rise and water supply temperature at cooling end have a good linear relationship, the external temperature and temperature of the water supply are adopted as two important parameters to characterize the room temperature of the heating room. Therefore, according to the existing water supply temperature mode, the relationship among the daily average room temperature of the room at 18℃ and the weighted average external temperature along with the daily average water supply temperature is analyzed. Figure 8 shows the relationship between the daily average room temperature of the room and the weighted average external temperature under the conditions of the existing water supply temperature mode $T_g = \text{-}T_w + 47.3\ ^{o}C$. What's more, the external temperature Tw value corresponding to room temperature at 18℃ is -11.2℃, and the value is brought into the existing water supply temperature mode -11.2 ℃ to obtain a corresponding daily average water supply temperature of 58.5℃, which indicates that at an external temperature of -11.2℃, water supply temperature of 58.5℃ is required to achieve room temperature of 18℃ in the room. Therefore, the external temperature of -11.2℃ and the water supply temperature of 58.5℃ are brought into the formula, and in the case where the coefficient a is 2.6, the coefficient b is calculated to be 29.4. Moreover, the time-dependent change of the water supply temperature is obtained in accordance with the principle that the hourly water supply temperature is equal to the average daily water supply temperature. Therefore, the analysis determines the optimal adjustment mode of the water supply temperature at the end of north facing room in middle layer will be determined through analyzing, which can be seen in equation (6).

$$T_g = -2.6T_w + 29.4 \qquad (6)$$

For the case that there are existing mode and the optimized adjustment mode in water supply temperature at the end of heat dissipation, daily average room temperature of northbound room is obtained through simulation calculation and the two are compared with the room average daily room temperature. What's more, compared with the existing mode of the water supply temperature at the end of the heat dissipation, the optimized adjustment mode makes the room temperature rise significantly and improves the stability of the room temperature. However, the room temperature under water supply temperature optimization mode is still lower than 17℃ in most heating period, which does not meet the room temperature setting requirement.

IV. CONCLUSION

The THIC system has a very good energy saving effect, and it can improve the healthy and comfortable indoor environment at the same time. In summary, through the application of thermostatic control in smart buildings, it is proved that the system studied in this paper reduces the energy consumption of chillers. Meanwhile, the transmission system and the energy consumption at the end of the air conditioner are reduced, and the lower COP is reduced as well so that the energy saving efficiency in the entire refrigeration system can be reduced. Besides it, it offsets the energy-saving advantages of high-temperature cold source. At the same time, there is a large amount of indoor exhaust, which uses the indoor exhaust to remove excess condensation heat and save the outdoor unit and the cooling water system. In addition, air supply temperature processed by new fan unit condensing and dehumidifying equipment is low, and the heat is reheated by the heat of condensation to avoid direct cooling and save heat.

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