INDOOR THERMAL COMFORT AND ENERGY EFFICIENCY OF VARIOUS AIR-CONDITIONING SCHEMES FOR MUSEUM BUILDINGS

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ABSTRACT

In museum buildings, air conditioning systems with precise thermal-hygrometric control of indoor environment are necessary for preventing degradation of artworks and providing a pleasant thermal environment for visitors. In traditional constant air volume (CAV) systems widely used in museums, the supply air is often cooled and then reheated for dehumidification without over-cooling conditioned spaces during warm and humid seasons. The cooling and reheating processes are energy-intensive. This paper presents a new air conditioning scheme for indoor thermal environment control in museums. The scheme integrates a dedicated outdoor air system (DOAS) with a CAV system. Control strategies are also developed for independent indoor temperature and humidity controls. Simulation tests are conducted to compare performances of a CAV system and the integrated system on the platform of TRNSYS. These two schemes are: (1) a CAV system with separate temperature and humidity controls by cooling-reheating; (2) a DOAS-integrated CAV system with independent indoor temperature and humidity controls. Simulation results show that the DOAS-integrated system is more energy efficient. It consumes about 60% less electricity than the CAV system with cooling-reheating on the test day. The DOAS-integrated scheme can also achieve satisfactory indoor thermal comfort considering the space temperature and relative humidity (RH). Both schemes can provide good indoor air quality measured by CO_2 concentration.

Keywords: Air-conditioning; Museum; Humidity control; Temperature control; Building energy efficiency

INTRODUCTION

There are a large number of museum buildings in the world. The functions of museums are to exhibit artworks to visitors and to preserve them as long as possible. The indoor thermal environment, mainly indoor temperature and humidity, and ventilation effectiveness in exhibition rooms are usually significant concerns. It should meet the requirements on artwork preservation as well as provide a comfortable environment for visitors. Temperature and RH in a museum is crucial for preservation of artworks. Fluctuating RH and temperature may lead to mechanical damage of artifacts because of expansion and contraction of materials. For those artifacts which are made from hygroscopic materials, effective control of indoor RH in exhibition rooms is quite important. Lower humidity can stiffen organic materials, making them more vulnerable to fracture, while higher humidity increases the risks of microbial growth and erosion (ASHRAE 2007). ASHRAE Handbook (2007) recommended a RH below 60% and temperature range between 15 and 25℃ for general museums. Bellia (2007) and Ascione (2009) proposed indoor air temperature of 22±1℃ and RH of 50±5% for exhibition rooms. Plenderleith and Werner (1974) reported that the room temperature should be pre-determined to suit the comfort of visitors in the museums, say within the limits of 16-25℃, and the RH could be determined by the collections. Thomson (1978) suggested that the RH upper and lower limits for moisture-absorbent materials preferably were 65% and 45%.

In order to provide comfortable indoor environment to visitors, besides indoor air temperature and RH, effective outdoor air ventilation is also important. The total fresh air system is preferable for the sake of human's comfort and healthy. However, it will result in unreasonably high latent loads on humid days and consequently unacceptably high energy consumption. In addition, it may cause indoor humidity fluctuations in museums. The demand-controlled ventilation (DCV) strategy, which can meet the space occupant ventilation requirement while minimizing energy (Kusuda 1976; Elovitz 1995), is recommended to control fresh air ventilation for occupants in exhibition spaces (ASHRAE 2007).

In CAV systems widely used in museum buildings, cooling and reheating is the most common way to prevent indoor air RH from exceeding the upper limit. The cooling and reheating processes are inherently energy-intensive, and are not encouraged in the HVAC applications. In addition, simple local controllers are usually adopted to regulate the supply water flow rates in cooling coil and heating coil. It will cause larger fluctuations on indoor RH and temperature, because the indoor humidity and temperature change slowly due to building thermal and moisture capacities but the humidity and temperature of the off-coil air change rapidly in the air handling unit with the change of the supply water flow rate.

To obtain better performance of temperature and humidity control and improve energy efficiency of air-conditioning systems, many researches have been carried out. Xu et al. (2006) presented a generalized predictive control strategy based proportional plus integral plus derivative (PID) controller for a cooling coil unit in air-conditioning system, which can deal with a wide range of operating conditions and achieve better performance than conventional PID controllers. Huh and Brandemuehl (2008) presented an application of direct expansion (DX) air conditioning system to simultaneously control indoor air temperature and humidity. Cascade control (Shin et al., 2002; Song et al. 2003; Wang et al., 2008) was proposed and adopted to enhance system robustness and control stability. Some intelligent control strategies (Dounis et al. 1994; Ben-Nakhi and Mahmoud 2001; Dounis and Caraiscos 2008) were used for energy and comfort management in building environments. These strategies can improve control accuracy and stability, but they can not solve the problem of coupled indoor temperature and RH controls in conventional central air conditioning systems.

An integrated air-conditioning scheme which integrates the DOAS with parallel sensible load treatment equipments, such as ceiling radiant cooling panels, dry fan-coil units and constant or variable volume all-air systems, was proposed to balance sensible load and latent load of conditioned space separately (Mumma and Jeong 2005; Shank and Mumma 2001). In this kind of systems, the dedicated outdoor air subsystem, which delivers the required amount of outdoor air into conditioned space, is responsible for the treatment of the total latent load and perhaps a part of sensible load. The parallel subsystem removes the remainder part of the space sensible load. Therefore, the DOAS-integrated air conditioning system can decouple the treatment of sensible load and latent load and consequently realize independent indoor temperature control and humid control.

In this paper, an integrated air conditioning system combining the dedicated outdoor air system with a CAV system is presented to control the temperature and humidity in an exhibition room. Energy consumption and indoor thermal comfort of the space, adopting this integrated system and a CAV system with reheating and cascade control, are evaluated and compared on the platform of TRNSYS (Klein et al. 1990).

AIR-CONDITIONING SCHEMES AND CONTROL STRATEGIES FOR MUSEUMS

The investigated modern museum is located in Hong Kong, which covers a total area of 17,500 m^2. The total exhibition area is 7,000 m^2, which can be divided into 8 exhibition rooms. Each air conditioning system serves one exhibition room. The simulated exhibition hall has a total area of 758 m^2 and a height of 6 m. The indoor design temperature is 24±1℃; indoor RH is 55±5%. The design occupant density in the exhibition room is 0.25 person per square meter. The moisture, sensible heat and CO_2 generation rates per person are 2.83×10^{-5} kg/s, 65 W and 5×10^{-6} m^3/s, respectively. The occupant's outdoor air requirement is 3.8 L/s per person and the outdoor air rate requirement per area is 0.3L/s per square meter, according to the ANSI/ASHRAE Standard 62.1-2007 (ASHRAE 2007). The outdoor air CO_2 concentration is 360ppm as a constant. The lighting thermal load is 30W per square meter. The museum opens from 9:00 am to 20:00 pm.

Because cooling is almost demanded year-round in buildings in Hong Kong, the control strategies for heating and humidification are ignored in this study. The energy consumptions and performances of the two types of all-air air conditioning schemes for cooling and dehumidification are investigated and compared. In both types of systems, enthalpy recovery devices (Zhang and Niu 2002) are adopted to save energy consumption of the air-conditioning system by recovering the sensible and latent heat of the exhaust air. Occupancy-based demand-controlled ventilation strategy (Wang and Jin 1998; Xu et al. 2009) is used to regulate the fresh air flow rate. The strategy is presented in detail in later section.

Scheme 1: The CAV system with separate temperature and humidity controls by cooling-reheating

In this system, the supply air is firstly cooled by a cooling coil and then reheated by a heating coil for precise indoor air temperature and RH controls in the exhibition hall, as shown in Fig.1. The cascade control approach is adopted for fast response and robust control performance (Wang et al., 2008). For the cascade control of the space RH, the actual indoor air RH in space is measured and compared with the RH set point. The supply air humidity set point is determined by the deviation. Then, the chilled water flow rate passing the cooling coil is modulated by controlling the opening of the chilled water valve to maintain the off-coil air humidity ratio at its pre-determined set point. The cascade control for the indoor temperature is similar to the humidity cascade control.

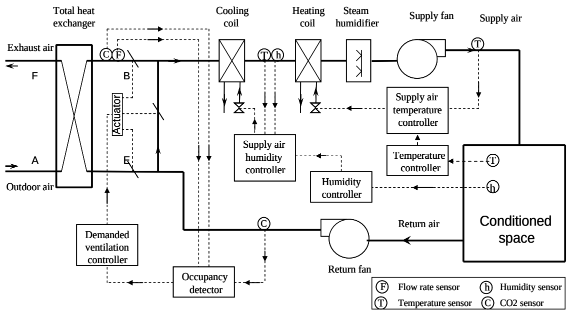

Fig.1 The CAV system with separate temperature and humidity controls by cooling-reheating

Scheme 2: The DOAS-integrated system with independent temperature and humidity controls

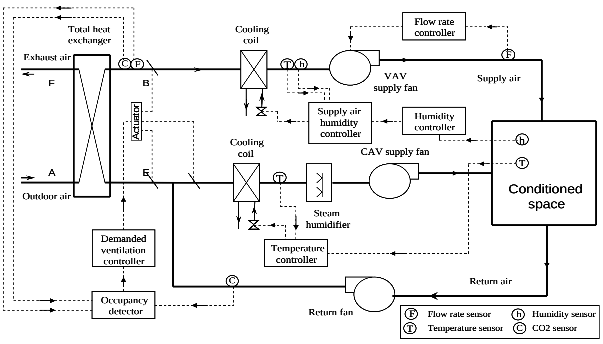

Fig.2 The DOAS-integrated system with decoupling control strategies for museums

In this system, the fresh air in the DOAS is processed and distributed to the exhibition hall separately, which is responsible for removing the space moisture load and a portion of the sensible load of the hall. The parallel CAV system is responsible for treating the remainder of the sensible load, as shown in Fig.2. The temperature of the supply chilled water in the CAV system is 15℃, slightly higher than the indoor air dew point temperature 14.6℃. The cooling coil in the parallel CAV system works on dry condition. The reheating process is not required in this integrated air conditioning system.

SIMULATION TESTS AND RESULT ANALYSIS

Overview of the simulator

A dynamic simulator of all-air air-conditioning systems serving the exhibition room was developed on the platform of TRNSYS. In this study, the building model and most of the component models are the same as those used in the VAV system developed by Wang (Wang 1999) with some modification on model parameters.

A simplified node model is used to simulate the exhibition space in the simulator. It is characterized by three state variables: the space air temperature (T), the space air humidity ratio (G) and the space CO_2 concentration (C). The air in the exhibition hall is assumed to be fully mixed so that the temperature, humidity and CO_2 concentration distribution are uniform in the space. The energy, moisture and CO_2 concentration balance governing equations of the zone air are shown in equation (1)-(3).

$$Mc_p \frac{dT}{d\tau} = Q + m_{CAV}c_p(T_{CAV}-T) + m_{DOAS}c_p(T_{DOAS}-T) + \frac{T_w-T}{R_{wi}} + \frac{T_{fut}-T}{R_{fut}} + \frac{T_{sa}-T}{R_{win}} + m_{\inf}c_p(T_{amb}-T) - m_{exf}c_pT_i \quad (1)$$

$$M\frac{dG}{d\tau} = GS + m_{CAV}(G_{CAV}-G) + m_{DOAS}(G_{DOAS}-G) + m_{\inf}(G_{amb}-G) - m_{exf}G \quad (2)$$

$$V\frac{dC}{dt} = CS + v_{CAV}(C_{CAV}-C) + v_{DOAS}(C_{DOAS}-C) + v_{\inf}(C_{amb}-C) - v_{exf}C \quad (3)$$

Where, T_{sa} is the solar air temperature; R_{wi} is the inside thermal resistance of external wall; R_{fut} is the thermal resistance of internal structure and furniture; R_{win} is the thermal resistance of external window.

A first order differential equation is used to represent the dynamics of a cooling/heating coil with lumped thermal mass as shown in equation (4). The dynamic equation on the basis of energy balance ensures that the energy is conserved.

$$C_c\frac{dt_c}{d\tau} = \frac{t_{a,in}-t_c}{R_1} - \frac{t_c - t_{w,in}}{R_2} \quad (4)$$

Where, t_c is the mean temperature of the coil; $t_{a,in}$ and $t_{w,in}$ are the inlet air and water temperatures; C_c is the overall thermal capacity of the coil; R_1 and R_2 are the overall heat transfer resistances at the air and water sides, respectively. The heat transfer calculation applies the classical Number of Transfer Unit (NTU) and heat transfer effectiveness methods. The coil model can simulate both dry coil and wet coil conditions.

A centrifugal fan with variable speed is used in the DOAS subsystem. The state of the centrifugal fan is characterized by two normalized variables representing the air volume flow rate and fan total pressure rise, respectively (Brandemuehl 1993). Constant speed fans are used in CAV systems, and fan energy consumptions are fixed once fans operate.

The total heat recovery exchanger is effective to decrease the transferred heat of the cooling coil, consequently minimize the energy consumption, particularly in the hot and humid weather. The enthalpy recovery device is simulated by a steady model with two effectiveness, sensible effectiveness (ε_S) and latent effectiveness (ε_L) as shown in equation (5) and (6).

$$\varepsilon_S = \frac{T_A - T_B}{T_A - T_E} \tag{5}$$

$$\varepsilon_L = \frac{G_A - G_B}{G_A - G_E} \tag{6}$$

Where, A, B, E and F, are four air streams, as shown in Fig. 1. Zhou et al (Zhou et al. 2007) adopted a sensible effectiveness of 0.79 and latent effectiveness 0.62 for the plate air-to-air heat exchanger in the cooling operating mode. The sensible and latent effectiveness of enthalpy wheel in the study by Ascione (2009) are 0.75 and 0.7, respectively. In this study, the sensible effectiveness is selected as 0.75, and the latent effectiveness 0.7. Dynamic models of other components, such as air duct, sensor, digital controller and actuator, are not described in detail here.

The occupancy-based demand-controlled ventilation strategy is adopted to control fresh air ventilation. According to the ASHRAE Standard 62.1-2007, the ventilation rate of fresh air for occupied space shall be determined as follows:

$$DVR = R_P P + R_B A \tag{7}$$

Where, R_P and R_B are the outdoor air requirements per person and per unit area, respectively. A is the occupied area and P is the actual space occupancy.

An on-line dynamic strategy is used to detect the actual number of occupant in the space (Wang and Jin 1998). In this strategy, CO_2 concentration is used as an indirect parameter for ventilation control, where is used for occupancy detection for further determining the fresh air flow rate set point.

$$P^i = \frac{E_{ac}(m_{OA}^i + m_{OA}^{i-1})(C_R^i - C_{OA}^i)}{2S} + V\frac{C_R^i - C_R^{i-1}}{S\Delta t} \tag{8}$$

Where, m_{OA} is the mass of outdoor air flow; C_R and C_{OA} are the CO_2 concentrations of the return air and outdoor air, respectively. V is the total indoor air volume; S is the average CO_2 generation rate per person. E_{ac} is the air change effectiveness. Superscript i and i-1 represent the current and previous sampling instants, respectively.

In this study, the energy consumptions and indoor thermal comfort of the exhibition space adopting the above-mentioned two air conditioning schemes under a typical summer day are investigated and compared by simulation. Figure 3 shows the outdoor air dry bulb temperature and humidity ratio on the test day.

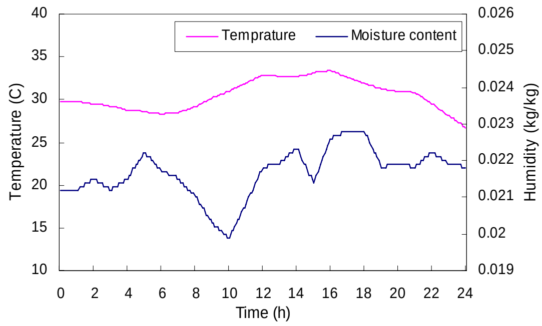

Fig.3 Temperature and humidity of the test day

Indoor thermal comfort in conditioned space

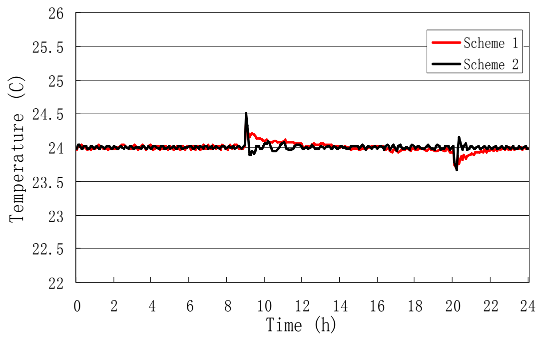

Fig.4 Temperature in exhibition room of two schemes on the test day

The indoor air temperatures of conditioned exhibition space of those two schemes are shown in Fig.4. There are overshoots and fluctuations at the start-up time (9:00 am) and the stop time (20:00 pm) of the museum in both air conditioning systems. It clearly shows that the response and robustness of the DOAS-integrated air-conditioning system (Scheme 2) are better than that of the CAV scheme (Scheme 1).

The RH of conditioned space of two schemes is shown in Fig.5. The RH of these two schemes is not as stable as the temperature during the whole operation period (from 9:00am to 20:00pm). However, it still can guarantee the required RH range for the museum, i.e. 55±5%. Scheme 1 achieves better humidity control performance. The DOAS-integrated system (Scheme 2) has larger overshoot and fluctuations compared with the CAV scheme (Scheme 1). The major reason for larger fluctuations and lag is the reduced supply air flow rate from the DOAS, which causes the indoor RH more sensitive to the latent load disturbance in the space.

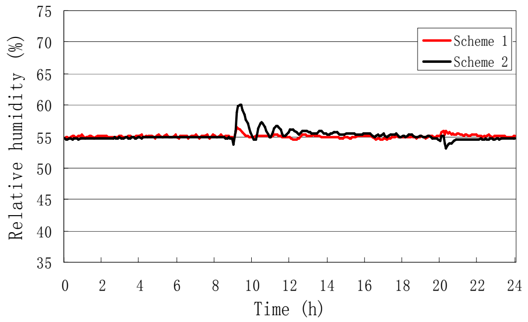

Fig.5 Relative humidity of exhibition for two schemes on the test day

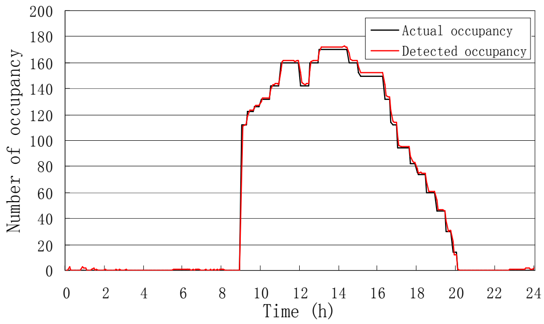

Fig.6 Comparison between actual occupancy and detected occupancy

Figure 6 presents the occupancies detected by the occupancy detector when the occupancy-based DCV strategy is used in air-conditioning systems. The difference between the detected occupancy and the actual occupancy can be observed and is small, which will not affect the DCV to set the outdoor air flow rate set-point.

In this simulation study, it is found that the CO_2 concentration in the exhibition room is much higher than the prescribed level (i.e. about 700 ppm above the fresh air CO_2 concentration) when according to the ASHRAE ventilation standard 62.1-2007. Therefore, in this study, the following rule was used to determine the fresh air requirement of the exhibition space. The outdoor air requirement is computed according to the number of occupants, i.e. 6.0 l/s per person while the minimum fresh air requirement is taken according to the conditioned area, i.e. 0.3 l/s per square meter for the museum space. This rule considers the minimum fresh air rate for occupant-generated pollutants and the minimum fresh air rate for diluting non-occupant-generated pollutants simultaneously.

The CO_2 concentrations of conditioned space of these two schemes are shown in Fig.7. The ventilation performances of two schemes are similar, because the same ventilation control strategy is adopted. The minimum CO_2 concentration is about 360ppm at the start-up time (9:00 am), because there is minimum outdoor air ventilation in the night time determined by the area of conditioned hall. The maximum CO_2 concentration is about 1000ppm which satisfies the requirement specified in ASHRAE Standard 62.1-2007. Since both ventilation requirements for occupants and area are considered, the CO_2 concentration in the single-zone space is changing.

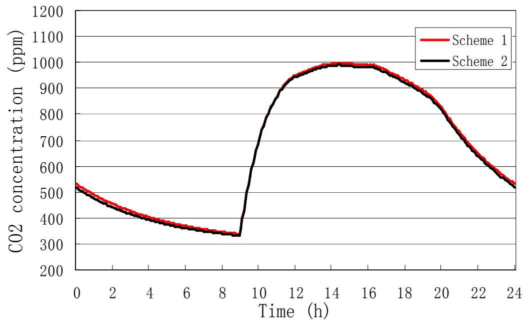

Fig.7 CO_2 concentrations of exhibition for two schemes on the test day

Comparisons on energy consumption

Energy performances of the air-side systems adopting two air conditioning schemes on the test day are tabulated in table 1. Energy consumption of the CAV system with reheating is used as benchmark. In this study, the design supply flow rates of CAV fans for these two schemes are 6.5m^3/s and 7.0m^3/s, respectively. The total efficiency of CAV fans is assumed as a constant value of 0.6. An overall coefficient of performance (COP) of the chilling systems is assumed to be 3.0 when evaporating at 5℃, and 4.45 at 15℃ in converting cooling load to electricity use (Zhang et al. 2005). The efficiency of electric hot-water boilers is 0.95 (ASHRAE 2007).

As can be seen, the fan consumptions of scheme 2 is just slightly larger than that of scheme 1, because the CAV fan in scheme 2 do not need to take charge of the resistances of reheat coil and total heat recovery device, although the supply fan flow rate in scheme 2 is larger. The total coil consumption of scheme 2 is about 73% less than that of scheme 1. It demonstrates that the overcooling and reheating process is low energy efficiency and consumes intensive energy. Correspondingly, the overall power consumption of DOAS-integrated air conditioning scheme is about 60% less than that of the CAV system.

DISCUSSION AND CONCLUSIONS

For museum buildings, CAV systems with reheating are widely used to meet the artwork preservation requirements as well as provide a comfortable environment for visitors. However, the over-cooling and reheating process which prevents the indoor air RH from exceeding a high limit is energy-intensive. A new air conditioning scheme which combines a DOAS with a parallel sensible system (i.e. CAV system) can independently control the space temperature and RH without the reheating requirement, which is promising to realize the independent and precise temperature and humidity controls.

In this study, the indoor thermal comfort of the space and energy consumptions, adopting a CAV system with reheating and a DOAS-integrated system, are evaluated and compared on the platform of TRNSYS. The results show that both these two air conditioning schemes can obtain good temperature control, particularly the DOAS-integrated scheme. The DOAS-integrated system can guarantee the required RH control range, although it is less stable than the CAV scheme. The major reason causing the instability is the reduced supply air flow rate from the DOAS subsystems. Therefore, it is recommended to adopt high aspiration ceiling diffusers, or adopt underfloor air distribution in DOAS-integrated systems.

Comparison of energy consumption illustrates that the DOAS-integrated scheme is more energy-efficient compared with the CAV scheme with reheating, which consumes about 60% less overall power than that of the CAV system when electrical hot-water boiler is utilized. Heat pump or waste heat is recommended for reheating which can save a lot of reheating energy. However, the energy saving potential of the DOAS-integrated system is still significant.

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Tables

Table 1. Energy consumptions of two air conditioning schemes on the test day

Energy performance	Scheme 1 CAV+ reheat (cascade control)	Scheme 2 DOAS+ CAV
DOAS fan (MJ)	-	36.61
Supply fan (MJ)	780.02	630.02
Return fan (MJ)	287.97	419.99
Total fan consumption (MJ)	**1067.99**	**1086.62**
Saving (%)	**--**	***-1.74***
DOAS cooling coil demand (MJ)	-	1835.47
AHU cooling coil demand (MJ)	7719.84	3656.07
Total cooling demand (MJ)	7719.84	5491.54
AHU heating demand (MJ)	2657.49	-
Total coil consumption (MJ)	**5370.64**	**1433.41**
Saving (%)	**--**	***73.31***
Overall electricity consumption (kWh)	**1788.51**	**700.01**
Saving (%)	**--**	***60.86***