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Enhanced Space Temperature Control of an Air-conditioning System with Small-scale ON/OFF Chiller

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Abstract

Small-scale on/off chiller integrated air-conditioning (AC) system is wildly used for cooling buildings. This paper presents a bilinear control algorithm to improve the robustness of the space temperature control of the AC system with on/off controlled chiller. The bilinear control algorithm was validated in a simulation A/C system as well as on an experimental platform. A water tank is designed in the water system for further enhancing the system performance. The validation results show that the bilinear control achieves better space temperature control performance than conventional PID control does when the chiller is switched on/off frequently both in the virtual AC system and the real experiment system. The control performance is also tested in the simulation AC system when the water tank is introduced. The results show that the optimal water tank size can guarantee the allowed on/off frequency of the chiller very well and simultaneously enhance the space temperature control performance efficiently.

Keyword - small-scale on/off chiller; bilinear control; water tank; space temperature control

1. Introduction

According to the studies [1,2], more than 40% of the building energy consumptions are mainly used to provide for occupancy a good indoor environment, such as satisfactory thermal comfort and indoor air quality. However, the indoor environment is not optimal in many buildings, which results in comfort complaints and/or reduced productivity of employees [3]. One of the difficulties in optimizing the indoor environment is due to the real controls, which are far from ideal in terms of their ability to react and anticipate to disturbances or uncertainties in the indoor environment.

It is well-known that in a variable-air-volume (VAV) air-conditioning (AC) system, the space temperature variation is a bilinear process suffering from load uncertainties [4]. In order to improve the robustness of the space temperature control, a number of control strategies were developed for the VAV AC system, such as model-based predictive control [5,6], robust infinite control [7] and many other controls listed in Ref. [8]. Huang [4] developed a bilinear control strategy, and its application in a typical VAV A/C system has shown that this control can significantly improve the robustness of the space temperature control [3]. However, this control strategy was developed based on the assumption that the supply air temperature is relatively constant.

In this paper, the small-scale chiller integrated central AC systems are considered, which are wildly used for cooling the buildings [9]. Different from the large capacity chiller plants, the most common control method of the small-scale chiller is on/off control [10]. Normally, the chiller compressor is switched on/off according to the chilled water return (CHWR) temperature. When the CHWR temperature is higher than a pre-defined threshold, the compressor will be switched on; while it will be switched off when the CHWR temperature is lower than a pre-defined threshold [11].

In this kind system, the supply air temperature is normally not under control and its value may vary significantly with the variation of the chilled water supply temperature, which oscillates following the on/off operation of the chiller. In this case, the conventional control methods may not achieve a robust space temperature control, and the bilinear control strategy proposed by Huang [4] cannot be used directly. Another important issue of the on/off control is that frequent switch on/off will speed up the wear and aging of the device and hence the chiller start-ups number per hour is always concerned in practice.

Gao et al. [12] proposed a control method which combines a bilinear control with a set-point reset scheme for a ground source heat pump (GSHP) integrated AC system. In this method, a simplified bilinear control was developed for the supply air temperature significantly varied VAV system. It can track the space temperature set-point well. The set-point reset technology can reduce the on/off frequency of the GSHP efficiently. However, the space temperature is changed markedly when the compressor is switched on and off.

This paper proposes a control method for the existing small-scale chiller integrated central AC system, in which the simplified bilinear control algorithm is used to improve the robustness of the space temperature control, and a water tank is designed for guaranteeing the maximum start-ups number of the chiller per hour and further enhancing the robustness of the space temperature control.

2. System description

2.1 System description and control method

The small-scale on/off chiller integrated central A/C system considered in this study is illustrated in Fig. 1. The on/off of the chiller is controlled according to the CHWR temperature. The water flow rate in the chilled water side is constant. The supply air is conditioned by an air handle unit (AHU), which consists of the fresh air from outdoor and the returned air from the conditioned space. When the chiller is on, the water is chilled down and delivered to the AHU to cool down the supply air. The supply air temperature is not under control. The supply air flow rate is controlled by a variable frequency drive (VFD) fan according to the room temperature measurement and its set point. The room is a common A/C zone, and its temperature is controlled by the room temperature controller.

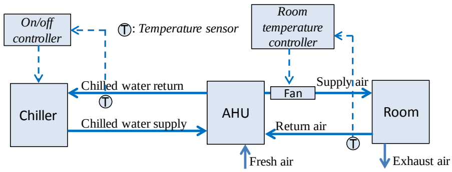

Fig. 1 Schematic diagram of the chiller integrated VAV A/C system

The proposed control method for this system aims to improve the space temperature control robustness and guarantee the maximum start-ups number of the chiller. It consists of the bilinear control algorithm and a water tank with the optimal size. The basic idea is that using the bilinear control algorithm to track the space temperature set-point and simultaneously judging whether the water volume of the chilled water system is enough to satisfy the designed maximum start-ups number of the chiller per hour. If not, a water tank is used to add the water volume to the required minimum volume.

The bilinear control algorithm is developed based on the dynamic model of (1) and (2). Equation (1) shows the dynamic process of the space temperature, and (2) describes the relationship between the fan frequency F and the supply air flow rate v_s [13,4].

$$C_z \frac{dT_z}{dt} = c_a \rho_a v_s \left(T_s - T_z\right) + q_z, C_z = c_a \rho_a V_z \tag{1}$$

$$v_s = \bar{v}_s \left(a_1 \frac{F}{\bar{F}} + a_0\right) \tag{2}$$

where T_z is the space temperature ℃, T_s is the supply air temperature ℃, C_z is the thermal capacitance of the air inside the zone kJ/K, c_a is the air specific heat kJ/kg·K, ρ_a is the air density kg/m^3, V_z is the space volume m^3, and q_z is the sensible cooling load of the zone kW. $\bar{F}$ is the rated frequency of the fan (usually 50Hz), $\bar{v}_s$ is the maximum flow rate corresponding to the rated frequency. a_0 and a_1 are coefficients which can be identified by fitting the data of the measured air flow rate at different frequency.

Equation (3) is the discretion of (1) with a sampling interval h, where k indicates the current time instant, and α_1 and α_2 are given by (4). Define the tracking error at the time instant k as $e_k = T_{z,k} - T_{z,set}$, $T_{z,set}$ is the space temperature set-point, then the supply air flow rate is given by (5), where $\zeta_{z,k}$ is a parameter on the cooling load, and γ is a user-defined parameter, satisfying (6). With this control law, the closed-loop dynamics becomes (7). When the required flow rate is calculated by (5), the frequency of the supply air fan at time k is given by (8).

$$T_{z,k+1} = T_{z,k} + \alpha_1 v_{s,k}\left(T_{s,k} - T_{z,k}\right) + \alpha_2 q_{z,k} \tag{3}$$

$$\alpha_1 = c_a \rho_a h / C_z, \quad \alpha_2 = h / C_z \tag{4}$$

$$v_{s,k} = \frac{(1-\gamma)e_{z,k} + \alpha_2 \zeta_{z,k}}{\alpha_1\left(e_{z,k} + T_{z,set} - T_{s,k}\right)} \tag{5}$$

$$0 < \gamma < 1 \tag{6}$$

$$e_{k+1} = \gamma e_k + \alpha_2\left(q_{z,k} - \zeta_{z,k}\right) \tag{7}$$

$$F_k = \left(\frac{v_{s,k}}{\bar{v}_s} - a_0\right)\frac{\bar{F}}{a_1} \tag{8}$$

When the room temperature is relative stable, *i.e.* $T_{z,k+1} \approx T_{z,k}$, the cooling load $q_{z,k}$ can be calculated by (9). Considering the dynamics of the measurement tools, the average value defined by (10) is used instead of (9) to estimate the current cooling load. In this case, $\zeta_{z,k}$ is set as (11).

$$q_{z,k} = c_a \rho_a v_{s,k}\left(T_{z,k} - T_{s,k}\right) \tag{9}$$

$$\hat{q}_{z,k} = \frac{1}{N_w}\left(\sum_{l=0}^{N_w-1} q_{z,k-l}\right) \tag{10}$$

$$\zeta_{z,k} = \hat{q}_{z,k} \tag{11}$$

2.2 Water tanks design and optimization concept

In the central AC system, the enough water volume is necessary for the chilled water loop to guarantee the safe operation of the chiller. Normally, this volume range from three to six gallons per ton in a typical HVAC system or up to six to ten gallons per ton where temperature control accuracy is critical [14]. However, this design does not consider the allowed maximum start-ups number of compressor per hour that may be specified by customers (e.g. three or two times per hour) to protect their device. In order to guarantee the allowed maximum start-ups number of the chiller, a minimum water volume inside the chilled water loop should be satisfied. If the water volume in the existing system is smaller than the required minimum water volume, a water tank is needed for increasing the water volume. The optimal size of the water tank is the minimum size to satisfy the required minimum water volume. The water tank with a larger size may be more benefit for reducing the start-ups number of the chiller, but the cost may increase and the maintenance may be more difficult.

For a single chiller with the on/off capacity control, the maximum number of the chiller start-ups occurs when the cooling load is 50% of the chiller rated cooling capacity [9], based on which the minimum chilled water volume is given by (12), where $t_{cyc,min}$ is the minimal on/off cycle corresponding to the allowed chiller maximum start-ups per hour, $\Delta T_{w,des}$ is the designed width of the CHWR temperature control band, $Q_{HP,rated}$ is the rated cooling capacity of the chiller kW, ρ_w is the water density kg/m^3, c_{pw} is the water specific heat kJ/(kg·K).

$$V_{w,sys,min} = \frac{t_{cyc,min} Q_{HP,rated}}{4\Delta T_{w,des} c_{pw} \rho_w} \tag{12}$$

Equation (12) is developed as follows. Firstly, according the energy balance of the chilled water, the CHWR temperature satisfies (13) [9], where C_w is the thermal capacitance of the water inside the chilled water system kJ/K, $V_{w,sys}$ is the water volume of the chilled water system m^3, q_l is the total cooling load of the zone kW, and Q_{HP} is the cooling amount of the heat pump kW. Using a sampling interval h_1, (13) is discretized as (14), where k indicates the time instant.

$$C_w \frac{dT_{w,re}}{dt} = q_l - Q_{HP}, C_w = c_{pw} \rho_w V_{w,sy} \tag{13}$$

$$\frac{T_{w,re}^{k+1} - T_{w,re}^{k}}{h_1} = \frac{q_l^k - Q_{HP}^k}{C_w} \tag{14}$$

When the CHWR temperature set-point and width of the CHWR temperature control band are at their designed values, the average cooling capacity of the chiller over the on time during on/off cycling period approximates to the rated cooling capacity. For a given total cooling load q_l, the on half-cycle t_{on}, *i.e.* the time used for the CHWR temperature dropping from its upper threshold to its lower threshold, is calculated by (15), where PLR is the partial load ratio. When the GSHP is switched off, Q_{HP}=0 and the off half-cycle t_{off} is calculated by (16). When PLR=0.5, the minimum full cycle is given by (17), based on which the minimum water volume inside chilled water loop can be calculated by (12).

$$t_{on} = \frac{\Delta T_{w,des} C_w}{Q_{HP,rated} - Q_{HP,rated} PLR}, PLR = \frac{q_l}{Q_{HP,rated}} \tag{15}$$

$$t_{off} = \frac{\Delta T_{w,des} C_w}{Q_{HP,rated} PLR} \tag{16}$$

$$t_{cyc,min} = t_{on} + t_{off} = \frac{4\Delta T_{w,des} C_w}{Q_{HP,rated}} \tag{17}$$

If $V_{w,sys,min} > V_{pip}$, a water tank is needed. The minimum size of the water tank for the system can be computed by (18). When the CHWR temperature set-point and width of the CHWR temperature control band are at their designed values, this size of the water tank can satisfy the allowed maximum start-ups number of the chiller at any cooling load condition.

$$V_{w,tank} = V_{w,sys,min} - V_{pip} \tag{18}$$

3. Simulation platform and Experiment Setup

In order to test the performance of the control method, a simulation platform was constructed by using TRNSYS software as shown in Fig. 2. The models used in this simulation platform include a building model, a heat pump (HP) model, an AHU model, water pump models, a water tank model, a ground source heat exchanger (GSHE) model, a cooling coil control valve model, temperature sensor models and an actuator model etc. Detailed models description can be found in Ref [12]. These models were developed based on the real facilities of an experimental platform as shown in Fig. 3.

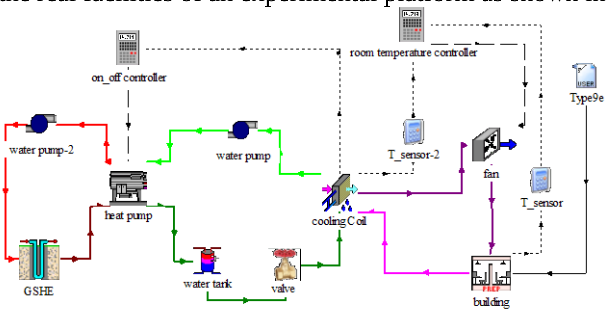

Fig. 2 Simulation platform constructed using TRNSYS

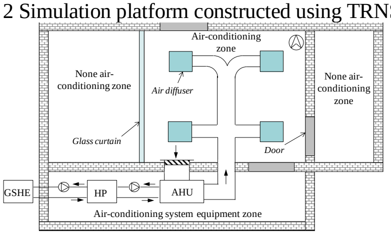

Fig. 3 Layout of the GSHP integrated AC system

In the experimental platform, the air-conditioning room has the area of 50m^2, which represents a common AC zone. The GSHP configures a scroll compressor and operates with a constant speed. The rated cooling capacity of the GSHP is 17.2kW and the input power is 3.91kW. The CHWR temperature set-point is preset of 14 ℃. The bandwidth of CHWR temperature control is considered to be ±3℃. The water volume inside the chilled water loop is about 0.09m^3. Both the chilled water flow rate and cooling water flow rate are constants, and the designed values are 0.822kg/s and 1.006kg/s respectively. The rated air flow rate of the AHU is 3000m^3/h, and the rated operation frequency of the fan is 50Hz. The fan is equipped with a variable frequency drive, and the operation frequency can be adjusted as required from 15Hz to 50Hz (*i.e.* 30% to 100% of the rated speed). The conditioned air is supplied to the room by four diffusers through the single duct system, and the room air is returned through the louver on the wall to the mixing box of the AHU.

4. Simulation and results analysis

4.1 Boundary conditions of the case studies

In the simulation tests, occupancy, lighting and equipment loads of 2kW and weather conditions were specified in the input data files. The outdoor air temperature and humidity in a typical summer day was used for the ambient air condition as shown in Fig. 4. The outside air flow rate was designed based on the occupancy and occupied area according to the standard [15] while the area-related source is ignored. The designed outside air flow rate for each occupancy was $30m^3/h$, and the occupancy number was 20. The cooling load due to the external heat gain was determined by outdoor air temperature and solar radiation gain. The set-point of the indoor air temperature is 25℃. The air-conditioning system operated from 8:00am to 18:00pm. The load condition during the system operation period was presented in Fig. 5. The sampling interval for the control algorithm was 15 seconds. The boundary conditions for each test are the same.

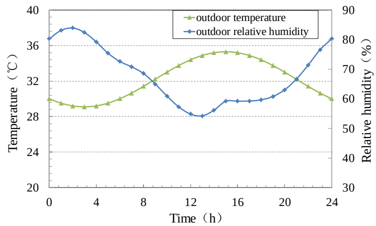

Fig. 4 Weather profiles of a typical summer day

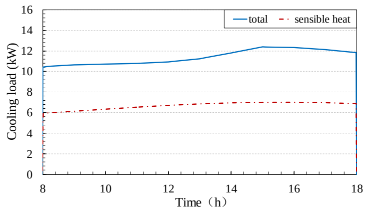

Fig. 5 Load condition of the room during the system operation period

For the bilinear control algorithm, there are two parameters that need to be preset, i.e., γ and N_w. The selection of γ is limited in its feasible range described by (6). Normally, a larger γ leads to a slow response of the controller; while a smaller one leads to a fast response of the controller. in this study, a slightly larger value γ=0.6 was used because a less aggressive closed-loop response can be obtained. N_w will not significantly affect the performance of the control, which is only used to smooth the estimation of the load. A suggested value for N_w is 5 as used in the case study.

4.2 Performance evaluation of the bilinear control

In the existing system (i.e., the water tank is not used), the control performance of the bilinear control was compared with that of the conventional PI control as shown in Fig. 6, which plotted the space temperature from 10am to 11am. The control parameters of PI control were K_p=5 and T_i=20, which were tuned at the full load condition. It can be seen that the bilinear control achieves a much better space temperature control performance than PI control. The average absolute tracking error from 10am to 11am was 0.079℃, while it was 0.292 ℃ when PI control was used.

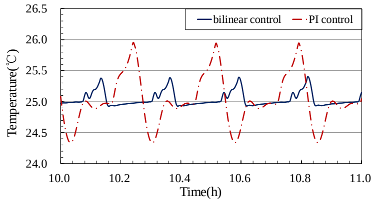

Fig. 6 Control performance of the space temperature on simulation platform

4.3 Performance evaluation with water tank

When the allowed maximum start-ups number of the chiller per hour is two times specified by customers, the required water volume is about 0.31m^3. A water tank is needed, and the minimum size is about 0.22 m^3. The performance of the space temperature control with the water tank was presented by Fig. 7. It can be seen that the space temperature control is enhanced significantly when the water tank is used. The average absolute tracking error from 10am to 11am was 0.016℃, while it was 0.079℃ without the water tank. The start-ups number of the chiller from 8:00am to 18:00pm with the water tank and without the water tank were 20 and 43 respectively. The average start-ups number per hour was 2, while it was 4.3 without the water tank. The water tank with optimal size added to the existing system can guarantee the maximum start-ups number of the chiller efficiently and enhance the space temperature control significantly.

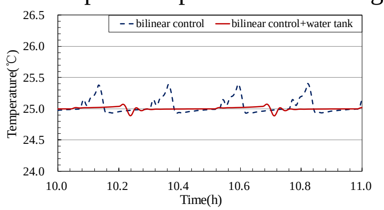

Fig. 7 Control performance of the space temperature with water tank

5. Experimental validation of the bilinear control algorithm

The bilinear control algorithm was validated on the experimental platform as shown in Fig. 3. The boundary condition for the experiment was the cooling load, as shown in Fig. 8, which was calculated by the building model [16]. The space temperature set-point was 24℃, and the sampling interval for the control algorithm was 15 seconds. The parameters of the bilinear control algorithm were preset to γ=0.6 and N_w=5. The control performance of the space temperature was presented in Fig. 8. It can be seen that the space temperature was maintained close to the set-point well, and the average absolute tracking error was 0.096℃ in one hour.

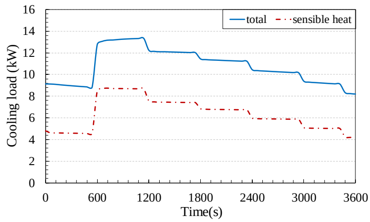

Fig. 8 Boundary condition of the experiment

The control performance was also compared with the PI control. When the PI control was used, the boundary condition approximated to that when the bilinear control was used. The control parameters of the PI control were K_p=0.8 and T_i=2, which were well tuned at the load condition of about 12 kW. The control performance of the PI control was also presented in Fig. 9. The average absolute tracking error was 0.178℃ in one hour. The comparison showed that the space temperature control performance was better when the bilinear control was used.

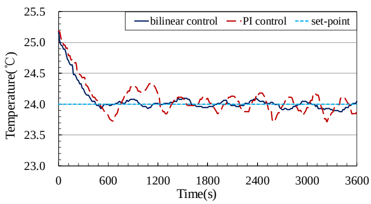

Fig. 9 Control performance of the space temperature on the experimental platform

6. Conclusion

This paper presents a control strategy for the small-scale on/off chiller integrated AC system to improve the robustness of the space temperature control and reduce the on/off frequency of the chiller. In this control strategy, the bilinear control algorithm is used to track the set-point of the space temperature, and the water tank is introduced to guarantee the allowed maximum start-ups number of the chiller per hour and enhance the space temperature control. Both the simulation result and the experiment result show that the simplified bilinear control can achieve a better space temperature control than PI control. The proposed control strategy is tested on the simulation platform. The results show that the water tank with optimal size can efficiently guarantee the maximum start-ups number of the chiller per hour and significantly enhance the space temperature control. The proposed control method has the potential for the real on/off chiller integrated AC systems to improve the robustness of space temperature control and reduce the on/off frequency of chillers simultaneously.

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