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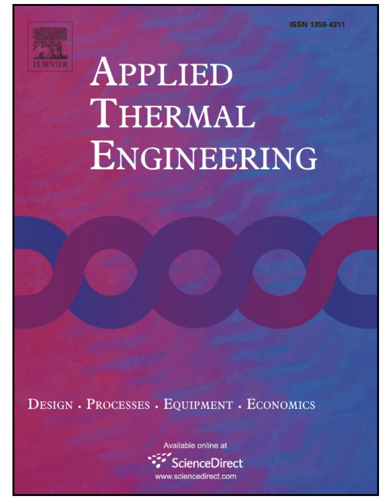

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An investigation of air and water dual adjustment decoupling control of surface heat exchanger

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Abstract: The terminal equipment of central cooling system accounts for a significant proportion of the total system's energy consumption. Therefore, it is important to reduce the terminal equipment energy consumption in central air conditioning system. In this study, the difference of the effect of the chilled water flow rate and air supply rate on the surface cooler during the heat transfer process is taken into full account. Matlab/Simulink simulation software is used to model and simulate the heat transfer of surface cooler of the main terminal equipment of air conditioning system. Simulation tests and experimental validations are conducted by using variable chilled water flow rate and variable air supply rate control mode separately. The experiment results show that the simulation model can effectively predict the heat transfer performance of heat exchanger. Further, the study introduced a dual feedback control mode, which synchronously regulates the chilled water flow rate and air supply rate. Also, under certain conditions, the complex heat transfer process of the surface cooler can be decoupled, and single variable control pattern is used to separately regulate the chilled water flow rate and air supply rate. This can effectively shorten the system regulation time, reduce overshoot and improve control performance.

Keywords: Central air-conditioning; heat exchanger; Simulation; Decoupling Control; Energy saving

Nomenclature

HVAC	Heating, ventilation and air conditioning	d	Humidity rate, g/kg
AC	Air conditioning	φ	Relative humidity %
VAV	Variable air volume	P_v	Partial vapor pressure
VRF	Variable refrigerant flow	E_w	Efficiency of water side
DDC	Direct Digital Control	a, b, c	Experimental exponents

$\dot{m}$	Flow rate, kg/s	ρ	Density, kg/m^3
$\dot{M}$	Quality storing up in heat exchanger, kg	δ	error
c	Constant-pressure specific heat, J/(kg. ºC)	q	Heat transfer rate of heat exchanger, W/m^2
t	Temperature, ºC	G	Flow rate of valve, m^3/s
Q	Total Heat transfer rate of heat exchanger, W	L	Open degree of valve
F	Face area of heat exchanger, m^2	K_1	Constant coefficient of valve
Q_{in}	Total heat rate in room, W	Subscript	
Q_w	heat transfer quality of chilled water side, W	a	Air
Q_{out}	Heat transfer rate from outdoor, W	w	water
ξ	moisture absorption coefficient	i	Inlet
ω	Water velocity in tube, m/s	o	Outlet
k	Heat transfer coefficient of heat exchanger, W/(m^2.ºC)	f	Water film
Δt_m	Logarithmic mean temperature difference, ºC	min	minimum
A	Efficiency of experiments	max	maximum
v_y	Face velocity of heat exchanger, m/s	s	Supply
B	Efficiency of experiments	r	Return
i	Enthalpy, J/kg		

1. Introduction

An enormous amount of world energy demand is associated with built environment [1], which motivated many researchers to focus on using energy efficiently in buildings. In particular, the application of heating, ventilation and air conditioning (HVAC) system in modern buildings in consistently expanding leading to significant energy consumption. In China, HVAC systems in commercial, industrial, and residential buildings consume 35% of the total energy in these buildings [2]. Therefore, reducing energy use for space heating and cooling in buildings is a key energy-saving measure.

For building air conditioning, two air conditioning (AC) systems [3], variable air volume (VAV) [4-6] and variable refrigerant flow (VRF) [7-9] are used. Both systems have been widely studied experimentally [9] and numerically [4, 5]. The VRF system was reported that it was popular in commercial buildings [3]. It can be achieved energy savings of 17% in buildings [5]. VRV system also can reduce the energy consumption. Aynur et al. [10] found that the ventilation-assisted multisplit VRV system operated in “synchronized indoor fan operation with economized” case consume around 17%-28% less energy than that of without economizer.To obtain a higher energy efficiency by considering the system with more detail factors, researchers propose some multi parameters optimization methods [11-15]. On the other hand, extensive research works were performed by Qi and Deng [16], Chen and Deng [17], Chen et al. [18], Georges et al. [19] and Tolga et al.[20] on variable refrigerant flow/volume (VRF/VRV) systems and their significance explained in modern HVAC applications regarding thermal comfort, IAQ, and energy efficiency metrics. Nassif and Moujaes [21] proposed a new method of operating the dampers in a variable air volume (VAV) air conditioning (AC) system under the economizer ventilation scheme to conserve energy. Also, the approach of variable water volume is also extensively applied to optimize the performance of the HVAC system, especially in chiller water system [22]. Jian et al. [23] developed the on-off switch control strategies based on a dynamic model of chilled water cooling system. Braun et al. [24] studied the condenser water temperature optimal control strategies by varying the changes of load and operating conditions. Ahmed et al. [25] focused on the variable water volume (VWV) systems and some of their (Direct Digital Control) DDC applications. However, most of the above previous investigations were focused on the whole system. Few investigation considered the effects of both VAV and VWV of heat exchanger performance and/or control in HVAC systems.

The surface heat exchanger (including condensers or evaporators) is one of the most essential components of the central air conditioning system terminal equipment. There had been considerable emphasis on these components due to their strong effects on the performance of central air conditioning systems [26, 27]. Evidence from theoretical analysis shows that the heat transfer process has nonlinear characteristics [28]. Accordingly, if the supply air rates are constant, the water flow rate has minimal effects on the air supply temperature with increase in the water flow rate up to a certain extent. Similarly, at constant water flow rate, when the supply air rate increases to a certain extent, the outlet air temperature remains unaffected. Also, there is a strong coupling correlation among all the parameters during the heat transfer process. To decouple this effect during the heating and/or cooling process, at present, most of the terminal equipment of air conditioning system usually adopt the operation mode where supply air rate remains constant, and air supply parameters change with the chilled water flow rate. Therefore, if the proper regulation mode is adopted, when regulating the chilled water flow rate, the air supply rate can also be adjusted. As such, the system stability can be maintained. Meanwhile, the energy efficiency of chilled water flow rate and supply air rate can both be reduced. Hence, the regulation time can be shortened thereby improving the occupants' comfort conditions.

The heat and moisture transfer process of the surface heat exchanger is complicated. There is strong coupling between the water flow and the air supply during heat transfer process, and it's challenging to de-couple. At present, many studies on the performance of surface heat exchanger are based on mathematical modelling and simulation techniques [29, 30]. A host of earlier studies focuses on the effect of parameter variation on the performance of heat exchanger [31, 32].

However, there were two points need to be analyzed in heat transfer process of surface heat exchanger. At present, fewer studies on the coupling effects of variation of water and air flow on heat exchanger during heat transfer process were conducted. In this investigation, the objective is to develop the method of decoupling by using single variable negative feedback control mode on the complex heat transfer process of the surface heat exchanger. At first, a surface heat exchanger coupling control model was established by using the effects of water flow and supply air rate on the surface exchanger performance together with their adjustment time difference. Secondly, by adjusting the chilled water flow rate and supply air rate, the experimental data from some certain experimental conditions were conducted to validate the simulation model. Thirdly, based on the control strategies of adjusting the chilled water flow rate and supply air rate at the same time, the stable time step of the indoor air temperature was analyzed. The operation energy consumption of the terminal equipment of air conditioning system can be further reduced thereby enhancing the system operation stability and human comfort.

2. Simulation model of heat exchanger

2.1 Establishment of physical model

To study the effect of inlet air temperature, relative humidity, air flow rate, chilled water flow rate, and chilled water inlet temperature changes on the outlet air temperature and chilled water outlet temperature of the surface cooler, a physical model are established based on the heat transfer process. The modeling process is shown as follow.

Heat exchange differential equation of air side.

$$\xi\, \dot{m}_a c_a (t_{ai} - t_{ao}) - Q = c_a \dot{M}_a \frac{d(t_{ao})}{d\tau} \tag{1}$$

Heat exchange differential equation of chilled water side.

$$Q - \dot{m}_w c_w (t_{wo} - t_{wi}) = c_w \dot{M}_w \frac{d(t_{wo})}{d\tau} \tag{2}$$

The heat transfer quality between air and chilled water is calculated as follow.

$$Q = Fq \tag{3}$$

In Equation (3), the heat transfer rate (q) can be calculated by the logarithmic mean temperature difference method, as follow.

$$q = k \vartriangle t_m = k \frac{(t_{ai} - t_{wo}) - (t_{ao} - t_{wi})}{ln \frac{t_{ai} - t_{wo}}{t_{ao} - t_{wi}}} \tag{4}$$

When the $\frac{t_{ai} - t_{wo}}{t_{ao} - t_{wi}}$ is lower than 2, the logarithmic mean temperature can be calculated by the simplified equation (5), with some error lower than 4% [33].

$$\vartriangle t_m = \frac{(t_{ai} - t_{wo}) - (t_{ao} - t_{wi})}{2} \tag{5}$$

Under cooling and dehumidification process, the heat transfer rate of the heat exchanger with same structure can be calculated by an empirical equation, as follow.

$$k = \frac{1}{\left(\frac{1}{Av_y^a \xi^b} + \frac{1}{B\omega^c}\right)} \tag{6}$$

The dimensionless of moisture absorption coefficient can be calculated by Equation (7) [34]

$$\xi = \frac{i_i - i_o}{c_a(t_{ao} - t_{w0})} \tag{7}$$

However, in order to enhance the accuracy of ξ, we adopt the water side efficiency method [45] to calculate the ξ, as follows.

$$t_b = t_{wi} + \frac{t_{wo} - t_{wi}}{E_w} \tag{8}$$

When the water film temperature is in the range of 10 °C to 20 °C, the enthalpy of water film can be calculated by Equation (9).

$$i_b = 0.0707 t_b^2 + 0.6452 t_b + 16.18 \tag{9}$$

If the area of surface heat exchanger was sufficiently large, and the heat is fully exchanged, the outlet water temperature t_{w0} equals t_b. Therefore, the ξ can be calculated by Equation (10).

$$\xi = \frac{i_i - i_b}{c_a(t_{ai} - t_b)} \tag{10}$$

Based on the empirical equations of air temperature and humidity, the enthalpy of inlet and outlet air can be obtained [46, 47].

$$i = 1.01t + (2500 + 1.84t)d \tag{11}$$

$$d = 0.6219 \frac{0.01 * \varphi * P_v}{101326 - 0.01 * \varphi * P_v} \tag{12}$$

$$P_v = 611.2 * \exp\left[\frac{\left(18.678 - \frac{t}{234.5}\right)t}{t + 257.14}\right] \tag{13}$$

2.2 Establishment of simulation model

According to the mathematical model of the surface heat exchanger, this paper uses Matlab/Simulink simulation toolbox to build a simulation model of a surface heat exchanger under the wet condition as shown in **Fig. 1**.

Based on the above analysis, by using the ramp module of Simulink toolbox to substitute the chilled water flow rate and supply air rate of the model, the effect of the chilled water flow and air supply rate on the heat exchanger performance can be simulated. Likewise, the quantity of heat transfer, dehumidifying coefficient, outlet air temperature and outlet chilled water temperature, can be observed in the simulation model.

3. Model validation

3.1 Experimental system

In order to validate the simulation model, a simple heat exchanger is used to assess the heat exchanger performance, which is separately carried out under variable water volume control mode and variable air volume control mode in enthalpy difference laboratory. The enthalpy difference laboratory experiment system is shown in **Fig 2**. The experimental system consists of a control and test systems. In the system, the chilled water side is connected to the chiller unit, and the supply and return water temperature of the chilled water can be obtained from the temperature measuring points which are set at the chilled water inlet and outlet of the tested fan coil. The chilled water flow volume can be adjusted by changing the opening degree of the electronic valve. The air side is connected with the enthalpy difference laboratory device. The air coming out from the fan coil has to pass through flow equalizing plate after passing through air supply sampling and measuring instrument. The air supply sampling and measuring instrument are used to measure the supply air state parameters. Also, the flow equalizing plate ensures uniform airflow mixing before the air enters the air volume measuring device via the nozzle. After passing through the induced drag fan, the cold air enters the static pressure chamber. The return air sampling device is set at the return air inlet to measure the return air state parameters. The water-cooled surface cooler, dry stream heater and dry stream humidifier which is identified on the left side of the test system diagram are used to maintain the indoor environment to meet the laboratory requirements. In the experiment system, the water-cooled surface cooler is connected with the water chilling unit, and the dry stream heater and dry stream humidifier are linked to the boiler. The experiment system monitoring platform is shown in **Fig 2**.

3.2 Experiment scheme and process

The laboratory equipment can provide and maintain the required indoor environmental conditions, and the chilled water flow amount, chilled water temperature, air rate and indoor environmental parameters can be set as needed. In this investigation, in the beginning of the experiment, the evaporator, heater, and humidifier run to control the indoor environment parameters (see **Fig. 2**). And then, the air side system and the water side system run. The indoor air was taken into the test section by the fan and cooled by the surface heat exchanger. In order to measurement the air flow rate, the air flow via the nozzle, and the difference of the pressure between in and out nozzle were measured. In the water side, the variable frequency water pump, close water store, and heater run to control the supply water temperature and water volume into the surface heat exchanger (see **Fig. 2**). During the experiment, the control system and test system operated at the same time.

In this investigation, the experimental cases were set up in term of the standard cooling quality nominal cases of fan coils. The detail information of experimental cases is as shown in Table 2. During the experiment, the level of indoor air temperature was set at the point of 27.0 ^{o}C with the relative humidity of 19.5%. Inlet water temperature was kept near 7.0 ^{o}C to meet the demands of the standard condition.

The objective of this experiment is to verify the heat transfer performance of the surface heat exchanger. The experimental procedure is as follows:

1) Keep the indoor air under standard condition. Running the fan coil until the indoor temperature and humidity are stable, and maintain the inlet water temperature of fan coil at 7°C constantly.

2) In variable water volume working condition tests. Keep the return air inlet air condition of fan coil unchanged, and the keep the air flow rate at 2670 m³/h constantly. Then, the test is carried out under three working conditions: 60%, 80%, 100% of the rate water volume.

3) After the operations become stable under each working condition for 30 minutes, we record the following data, including the air dry bulb and wet bulb temperature of the surface cooler inlet and outlet, the water temperature and water flow rate of the chilled water inlet and outlet, and refrigerating capacity of the fan coil water side and air side.

In the variable water volume working conditions, the levels of the rate of water volume, including 60%, 80%, and 100% of the rate water volume, shown in **Table 3**. All the parameters were set near the demands of the standard conditions. The levels of inlet air temperature were near 27.0 ^{o}C. The levels of inlet chilled water temperature were near 7.0 ^{o}C. During the experiments, the data are recorded every 20 seconds. A total of 30 groups of data is measured and averaged over the measurement period.

3.3 Data processing

Fan coil heat exchange data includes chilled water side and air side heat exchange parameters. The chilled water side heat exchange quantity, Q_w, can be calculated with the water temperature of inlet and outlet chilled water and the chilled water flow rate which is recorded in the experiment. The air side heat exchange quantity, Q_a, can be calculated through enthalpy difference with the air dry bulb and wet bulb temperature of inlet and outlet air and air rate. Meanwhile, considering the effect of the heat leakage rate of the test device, the modification is made to get the air side cooling quality [35].

The heat transfer quality of chilled water side Q_w can be calculated by Equation (14)

$$Q_w = G\dot{m}_w c_w (t_{wo} - t_{wi}) \tag{14}$$

In Equation (14), the specific heat at constant pressure of chilled water can be assumed to be 4.19 kJ/(kg.°C).

The heat exchange quality of air side (Q_a) can be calculated by Equation (15) [35].

$$Q_a = Q_{a1} + \Delta Q_1 \tag{15}$$

In Equation (15), the ΔQ_1 is the heat loss from the enthalpy difference, calculated by the Equation (16).

$$\Delta Q_1 = k_{as}(t_r - t_s) \tag{16}$$

In Equation (16), the enthalpy difference leakage coefficient was determined by the experimental data. Therefore, the actual heat exchange quality of fan coil can be calculated by Equation (17), as follow:

$$Q = \frac{1}{2}(Q_w + Q_a) \tag{17}$$

In this study, heat balance deviation has to be calculated between chilled water side heat exchange quantity and air side heat exchange quantity. Subsequently, we revise the chilled water side heat exchange quantity and air side heat exchange quantity. A value difference of less than 5%, therefore, implies that the experimental data is valid. The heat balance deviation is calculated by equation (18).

$$\delta = \frac{(Q_w - Q_a)}{Q} \times 100 \tag{18}$$

4. Optimized control

4.1 Optimizing control theory

In order to achieve a suitable operation control method of the surface cooler, this study employs the influence of supply air temperature on chilled water flow rate, a variation of supply air rate and their speed difference. Further, the study adopts single variable feedback mode to reduce the coupling effect of the chilled water flow rate and air supply rate on the surface heat exchanger. In this regard, indoor load regulation is divided into two phases: at first, chilled water flow rate is adjusted in accordance with the indoor air temperature to keep the indoor air temperature within the permitted target temperature range; then since the heat exchange quantity has quick response to the variation of air supply rate, by making use of this characteristic, air supply rate is adjusted to make the indoor air temperature reach target temperature. That is within a certain range of the single variable feedback mode which is adopted to adjust the chilled water flow rate and air supply rate separately to achieve better comfort and energy efficiency. The functional block diagram of the optimal control strategy is shown in **Fig. 3**.

4.2 Optimized control logic

The purpose of changing chilled water flow rate is to meet the demand of the indoor cooling load. Based on the above analysis, as long as the indoor temperature is within the comfortable zone, chilled water flow rate requires no adjustment. By changing air flow rate, on condition of meeting the demand of new air flow, achieving small disturbance to indoor temperature, a minimum new air flow can be used by detecting indoor air quality (e.g., CO_2). If indoor air temperature varies out of allowed range, the conventional control method can be utilised, based on real time and target temperature difference, changing chilled water flow rate at first to meet indoor cooling load.

Fig. 4 is the control flow rate chart of above-mentioned purpose, where Δt is the comfortable temperature tolerance. If indoor air temperature is within comfortable tolerance, then air flow rate control dominates to meet small load disturbance; if indoor air temperature is out of comfortable tolerance range, then chilled water control dominates; if indoor air temperature is higher than set temperature and air fan reaches minimum speed, then it is necessary to change the target, reduce water valve openness to stabilize the system.

Compared with conventional control method, under part cooling load, if chilled water rate can meet indoor loading demand, this strategy will keep chilled water flow rate constant, by changing air flow rate to meet loading demands. This approach avoids not only the hydraulic imbalance of chilled water network due to constant adjustment of chilled water flow rate, but also enjoys the benefit of less transportation loss of chilled water network. Using a variable-frequency fan to control the air flow rate on the surface cooler, the significant energy efficiency of air flow rate is achieved. Consequently, due to the fast response of air flow rate control, occupants will have a much more comfortable experience.

4.3 Optimized control simulation model

This study uses **Fig. 1** surface heat exchanger as a basic simulation model, incorporating strategy illustrated above, we construct an optimized surface cooler simulation model.

In **Fig. 5**, objects such as air conditioning room, fan controller, water valve, etc. are modeled according to their physical and mathematical processes. Air conditioning room model assumes the room air is completely mixed, treat air as lumped parameter and treat the heat capacity of inner room structure as a constant. According to energy conservation, the physical air temperature model in the room is illustrated as below:

$$\rho_a c_a V \frac{d(t_r)}{d\tau} = Q_{in} + Q_{out} - \rho_a c_a v_s (t_r - t_s) \quad (19)$$

Water valve model uses fast open flow characteristic valve. Its mathematical model is:

$$\frac{d(G/G_{max})}{d(\frac{L}{L_{max}})} = K_1 (\frac{G}{G_{max}})^{-1} \quad (20)$$

Fan physical model can be established using Similarity Law:

$$\frac{n}{n_o} = \frac{Q}{Q_O} \quad (22)$$

Both fan and water valve use single variable feedback control, per PID control, using the physical model as above, a respective simulation model can be established. Detailed procedure can be found in reference [36].

4.4 Optimized Simulation cases

Using simulation model and optimized control strategy shown in Fig. 5, we conducted several cases under similar working conditions. In the first case, we experimented with double variable feedback by simultaneously controlling chilled water and air flow rates. As for the second case, we examined single variable feedback by controlling air flow rate, while the third case involves single variable feedback by controlling chilled water rate. The performance of these three cases was compared and analyzed. Detailed simulation parameters are listed as follows:

p_a=101325 Pa, $t_{ai} = 35$ °C, $\varphi = 60\%$,$t_{wi} = 7$ °C, $v_{wi-max} = 1.0$kg/s, $v_{ai-max} = 1.5$kg/s, $Q = 12500$ W, $t_o = 35$ °C, $t_n = 26$ °C.

In simulation cases, at 10000th second, the cooling load of 11000 W is stressed and step up to 14000 W. Surface cooler, fan and chilled water valve parameters and other parameters take the same value as in **Fig. 1**. Total simulation time is 7200s. Detailed simulation information is presented in **Table 4**.

5. Results and Discussion

5.1 Validating the numerical simulation model against the experimental data

To ensure the accuracy of heat exchanger simulation model, this paper uses the experimental data for the model verification. **Fig. 6** shows the relation between air temperature and chilled water difference percentage at the inlet and outlet of the surface heat exchanger. It can be seen from **Fig. 6** that when there is a change of chilled water flow rate, there is a change of temperature difference at the inlet and outlet of the heat exchanger. Simulation results fit quite well with experimental results, especially the variation trend of temperature difference at heat exchanger inlet and outlet. As the chilled water flow rate increases, the surface heat exchanger has a significantly smaller temperature difference between the inlet and outlet. The main reason is that when air parameters of surface heat exchanger air side are stable, heat transfer quality with chilled water stays almost constant. Hence, the temperature difference of chilled water decreases with increase in chilled water flow rate. This analysis proves the accuracy of the model.

5.2 Variation of outlet air temperature

Based on the analysis of simulation data, the variation of outlet air temperature of the surface heat exchanger was shown in **Fig. 7 (a)**. Keeping chilled water flow rate as 2.0 kg/s, giving a step change at 30th seconds at surface heat exchanger outlet, a clear temperature rise from 9.5 °C to near 12.0 °C is observed, thereby resulting in nearly 2.5 °C increase. This indicates the condition of constant chilled water, with the increasing air flow rate to the surface heat exchanger, the outlet air temperature will increase, which accords with a typical heat exchanger performance. From **Fig. 7 (b)**, when air temperature of outlet reaches a stable value, the value is approximately 10 s. Keeping air flow rate at 1.5 kg/s, giving a step change from 2.0 kg/s to 2.5 kg/s of surface heat exchanger outlet chilled water at 30^{th} seconds, the outlet temperature of surface heat exchanger has a significant variation, from around 12 °C to decreasing rapidly, settling at a final value of about 11 °C. The settling time is around 10 s. This shows that on the condition of keeping constant air flow rate, with the increase of chilled water in the surface heat exchanger, outlet air temperature and outlet chilled water temperature drop accordingly.

Fig. 8(a) shows the variation dependence of surface heat exchanger air temperature with chilled water relative flow rate. From **Fig. 8(a)**, it can be found that at constant air flow rate, with the increase of chilled water flow rate, surface heat exchanger increases its heat transfer step by step; outlet air temperature of surface heat exchanger drops accordingly, showing the basic rule of heat transfer. Moreover, it can be seen from **Fig. (8)** that when chilled water flow rate is small, outlet air temperature changes greatly with chilled water flow rate, when chilled water flow rate reaches 65% of quota. Even with the decrease of chilled water flow rate from 100% to 62%, outlet air temperature rises from 7.5 °C to 9.1 °C. The flow rate decreases 38%, and temperature rises 20.6%. This means when chilled water flow rate reaches a certain amount, air processing unit heat transfer capability is less sensitive to chilled water flow rate.

Fig. 8(b) shows the relation between surface heat exchanger air outlet temperature and relative air flow rate. From **Fig. 8(b)**, it can be seen that when chilled water flow rate kept constant, with the increase of air flow rate, surface air heat exchanger air outlet temperature rises. Furthermore, when air flow rate varies within a small range, a rapid change in air outlet temperature changes ensued. However, with a further increase of air flow rate, when air flow rate reaches 65% of quota, even with a rapid growth in air flow rate, surface heat exchanger air outlet temperature remains with marginal variation. By analyzing heat transfer data, when air flow rate drops from 100% to 70% of quota, air outlet temperature drops from 11.4 °C to 9.4 °C, i.e., the flow rate decreases by 30%, air outlet flow rate temperature drops 17.8%. On the other hand, when air flow rate drops to 62%, air outlet temperature drops from 11.4 °C to 8.7 °C, a drop of 24.1%, which is bigger than the increase of 20.6% by chilled water flow rate. This means that with the same amount of flow rate change, air flow rate has a stronger impact than water flow rate on outlet air temperature.

5.3 Variation of indoor air temperature with different control approaches

Fig. 9 is the indoor air temperature variation with running time under three control modes. From **Fig. 9**, when the temperature is set in air conditioning room, with cooling load changing, three control modes can all stabilize to set air temperature. At a step change of cooling load disturbance of 10000 s, indoor air temperature changes accordingly. Afterward, with increasing air flow and water flow rate, room temperature resumes to the 26.0°C set point. Comparison between the chilled water single variable feedback with air flow rate single variable feedback, air/water double feedback control has a shorter response time, small overshoot, better indoor comfortable, and no chaos with surface heat exchanger during heat transfer process due to strong coupling.

Based on simulation data, incorporating both air/water double variable feedback, air conditioning room temperature reaches stability. The energy consumption of fan is reduced by up to 5%. The chilled water flow rate decreased by 15% of quota. The energy saving can be seen in chilled water pump flow rate control.

Simulation results show that by using air/water double variable feedback control method, when cooling load changes, response time is shortened, and indoor thermal comfort is improved. Meanwhile, under partial cooling load condition, it is effective to reduce the energy consumption of air conditioning terminals. Chilled water flow rate is in non-saturation region, so heat exchanger has better performance. In addition to that, due to small cooling load varied, by changing air flow rate to meet air temperature, it is not necessary to modify chilled water flow rate frequently, which is useful for the operation of entire air conditioning system. If all the air conditioning terminals adopt this strategy, it is potentially viable to reduce energy and improve system stability greatly.

6. Conclusion

It is of great importance to reduce energy consumption by refined operation control of terminals in central air conditioning system. This paper utilizes Matlab/Simulink as software to model and simulate the main terminal equipment of air conditioner. A series of tests to verify the numerical models using variable chilled water and variable air flow rate parameters were conducted. Some conclusions were found as follow.

(1) The model is capable of predicting the heat exchanger's performance. When chilled water flow rate reached a certain threshold, the sensitivity of the air processing unit to chilled water flow rate diminished.

(2) With the same amount of flow change, air flow has a bigger impact on conditioner outlet temperature than chilled water flow rates.

(3) A double feedback of both variable air flow rate and chilled water control method was proposed, which is shortened and performed better than single feedback control method of variable chilled water or air flow rate.

The heat transfer model considering latent heat and sensible is important for simulation and performance. In this investigation, the objective focuses on the decoupling control of the surface heat exchanger. The variations of the air flow rate and chilled water volume on heat transfer performance of the surface heat exchanger is as one of the emphasis. The stable time step of indoor air temperature is the optimized objective. However, in the future work, the effects of several important factors on heat transfer performance of surface heat exchanger, including the inlet air temperature, the inlet air relative humidity, and supply chilled water temperature, would be experimented and analyzed.

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Figure

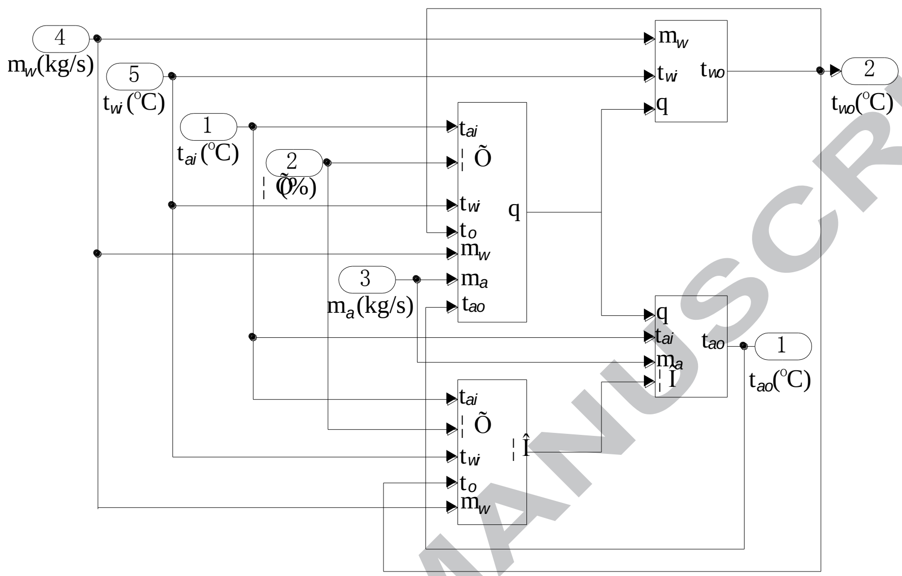

Fig. 1. Simulation in the dehumidifying conditions of surface heat exchanger

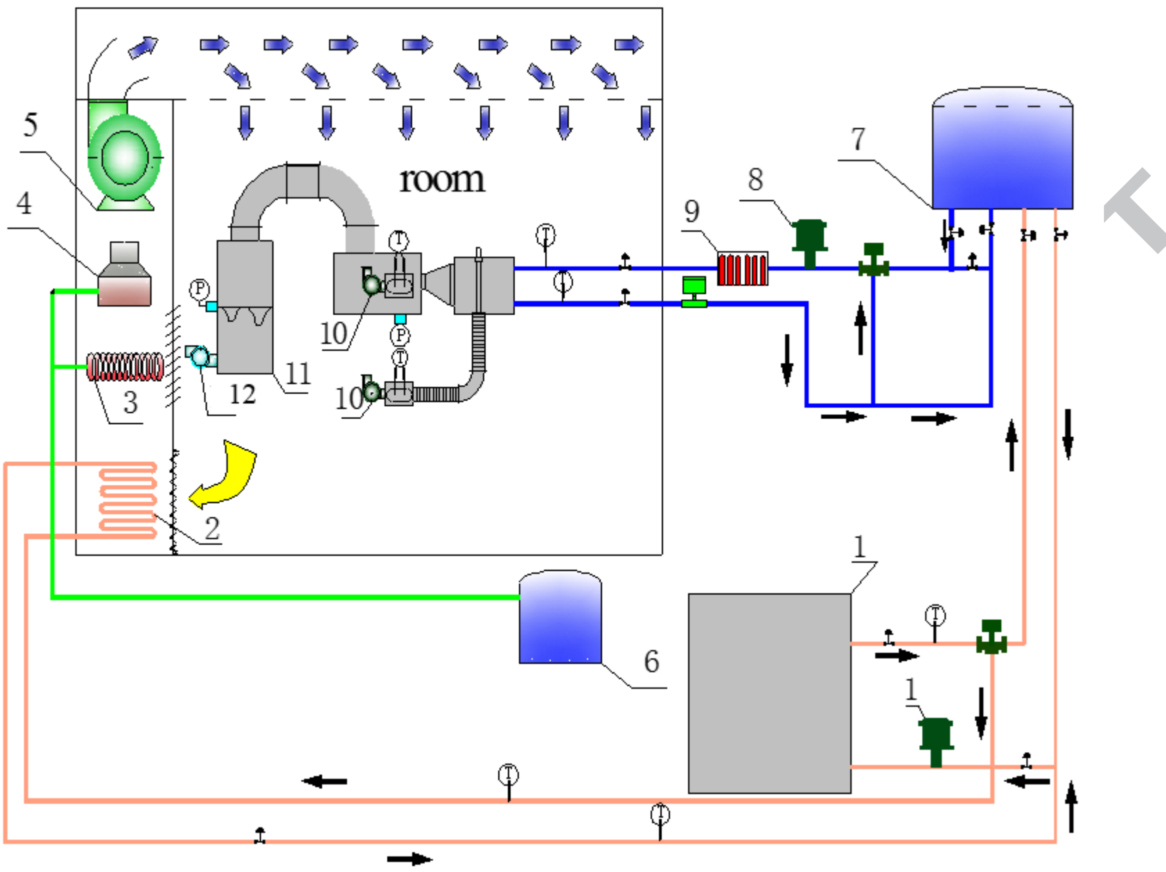

Fig. 2. Monitoring platform diagram of experiment system: 1.Chiller, 2.Evporaor, 3.Heater, 4.humidifier, 5.Fan, 6.Water store, 7.Mixing water store, 8.Variable frequency water pump, 9.Closed water store + heater, 10. Fan for taking sample, 11.mixing cabin, 12. Fan for exhaust

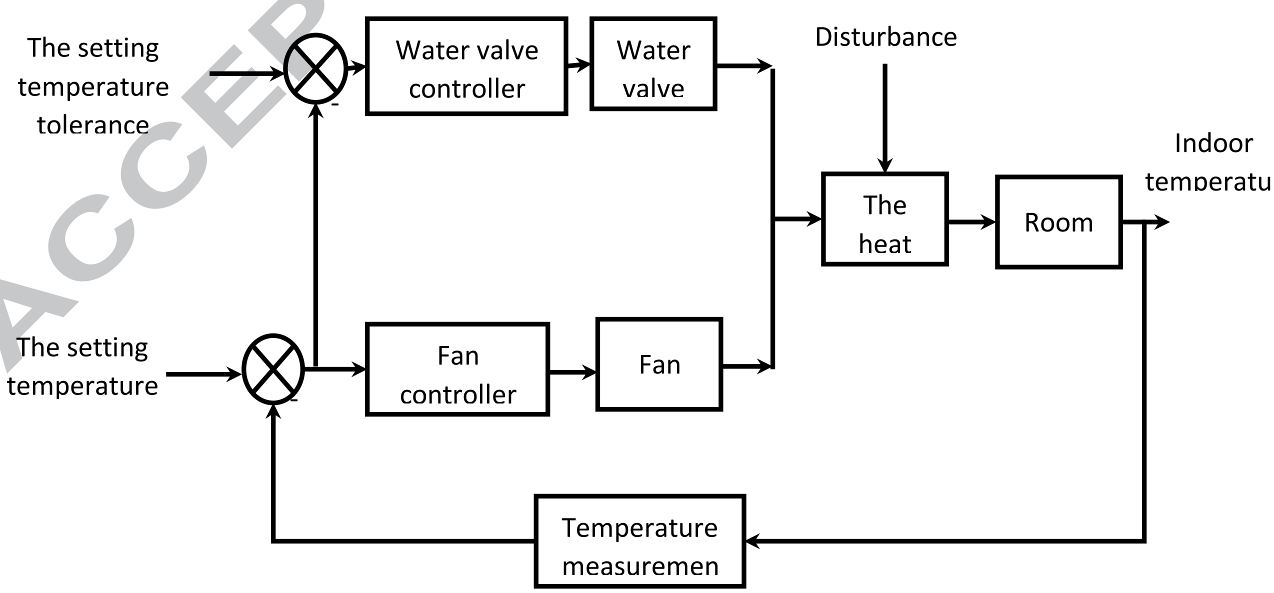

Fig. 3. Functional block diagram of optimal control strategy for the heat exchanger

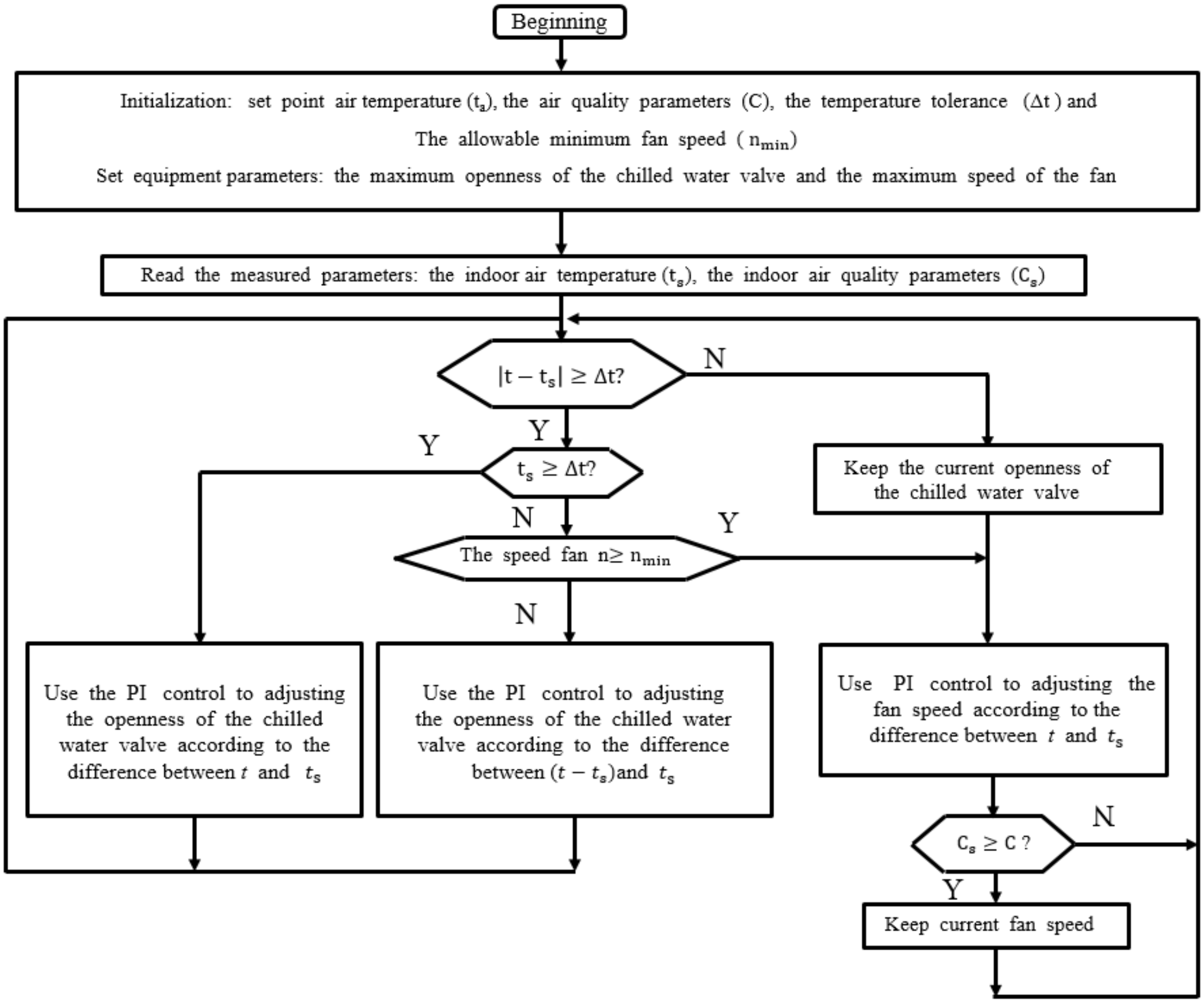

Fig. 4. The control flowchart of the optimal control strategy for the cooler

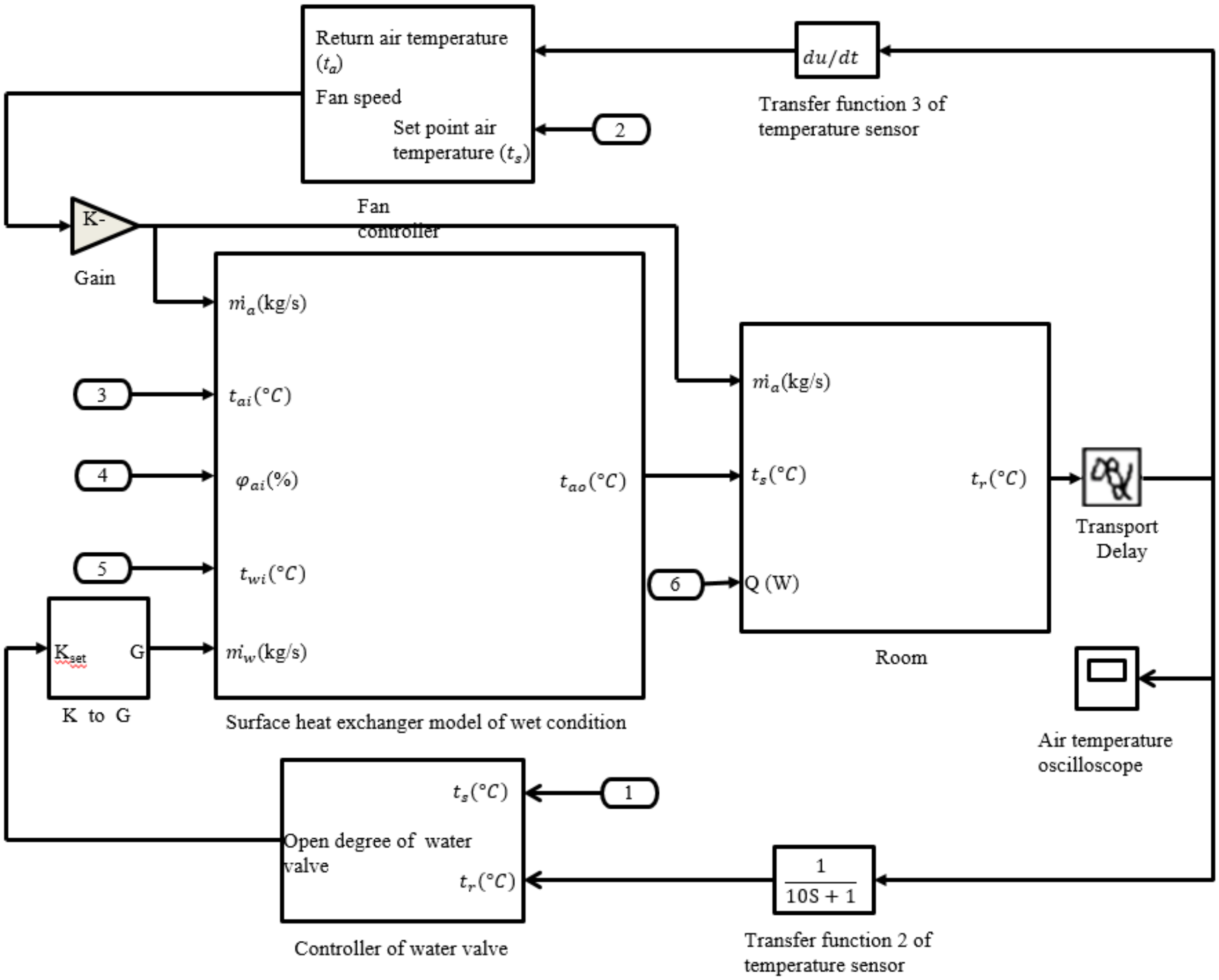

Fig.5. Simulation model of optimal control method

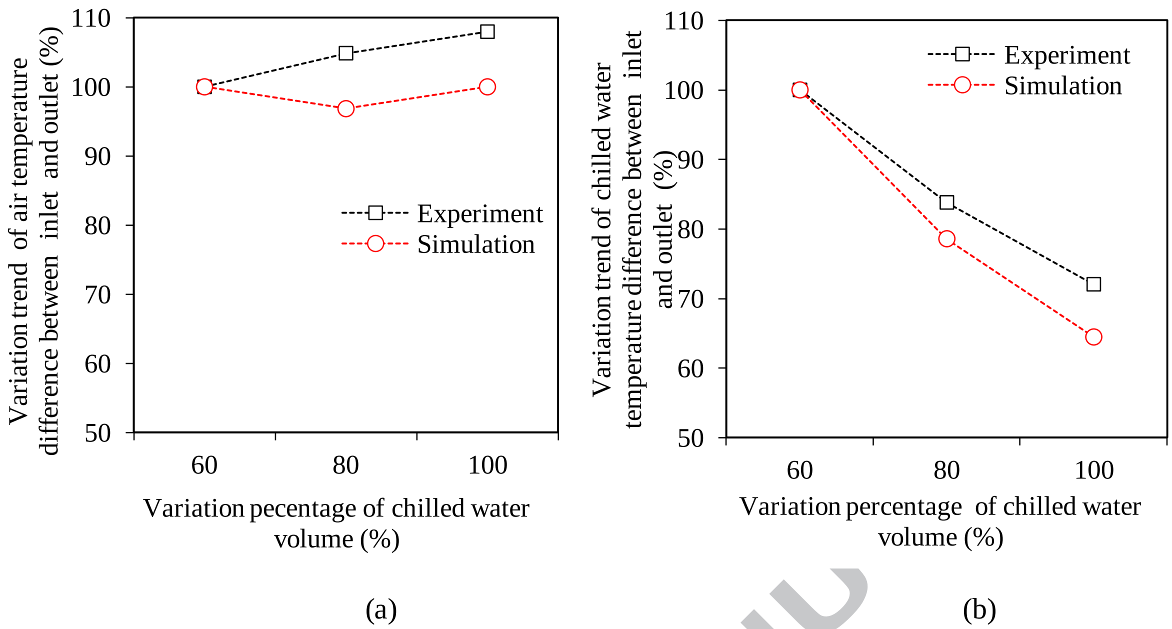

Fig. 6. Variation trend of model against experimental data, (1) air temperature difference, (2) chilled water temperature different

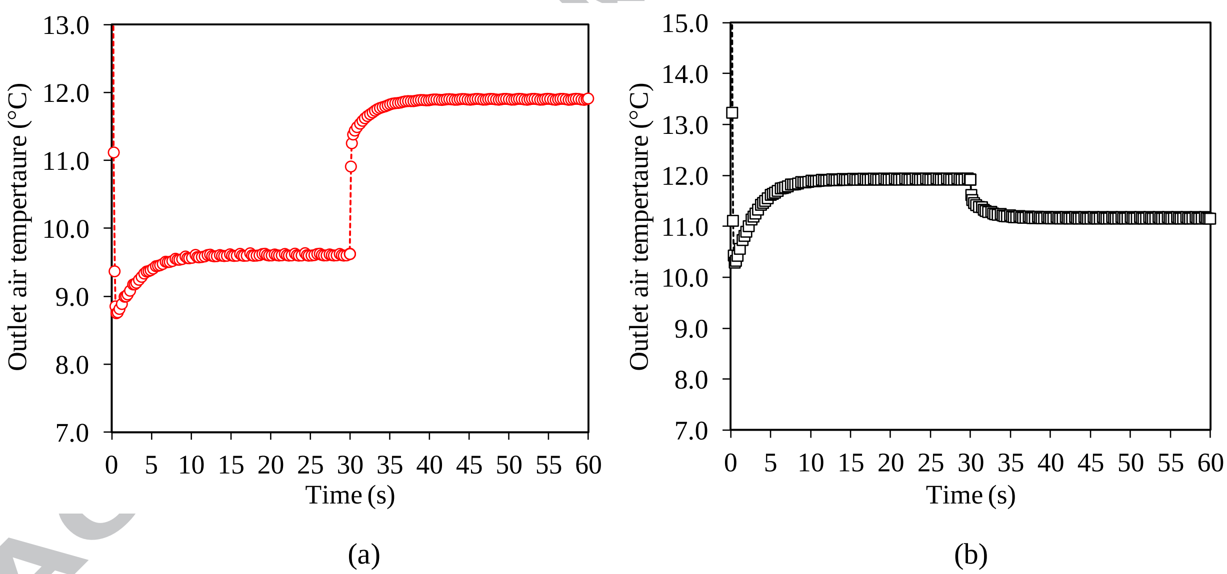

Fig.7. The curve of output air temperature of surface air heat exchanger: (a) after air volume step change, (b) after water volume step change

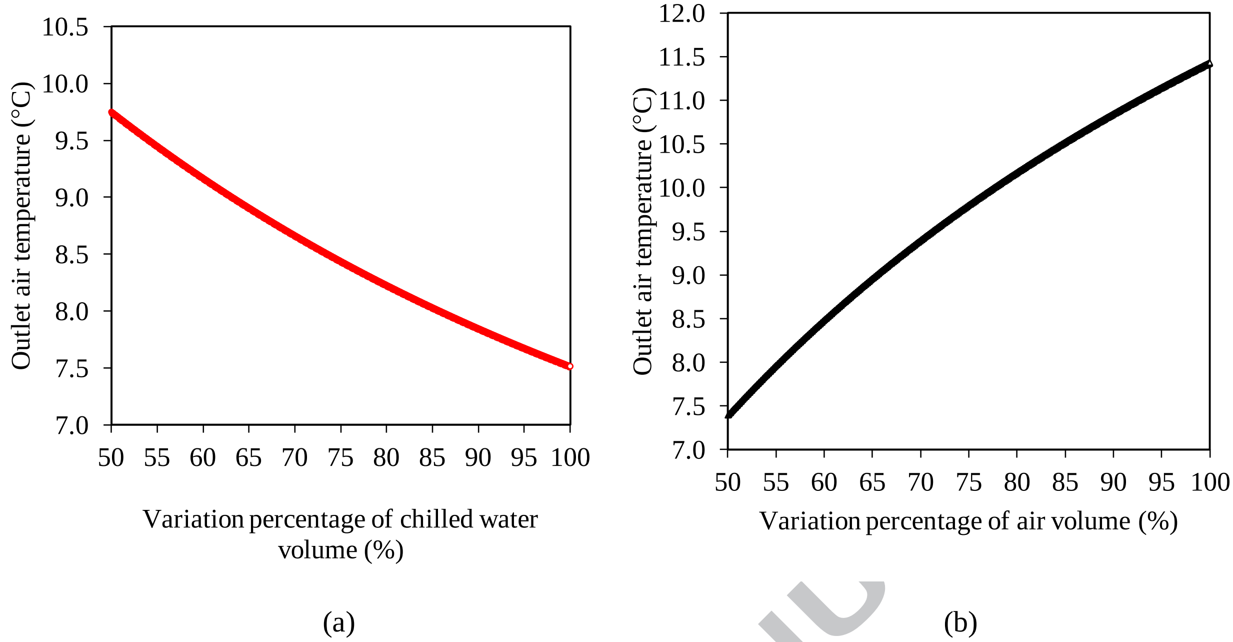

Fig. 8. Variation of outlet air temperature against: (a) variation percentage of chiller water volume, (b) variation percentage of air volume

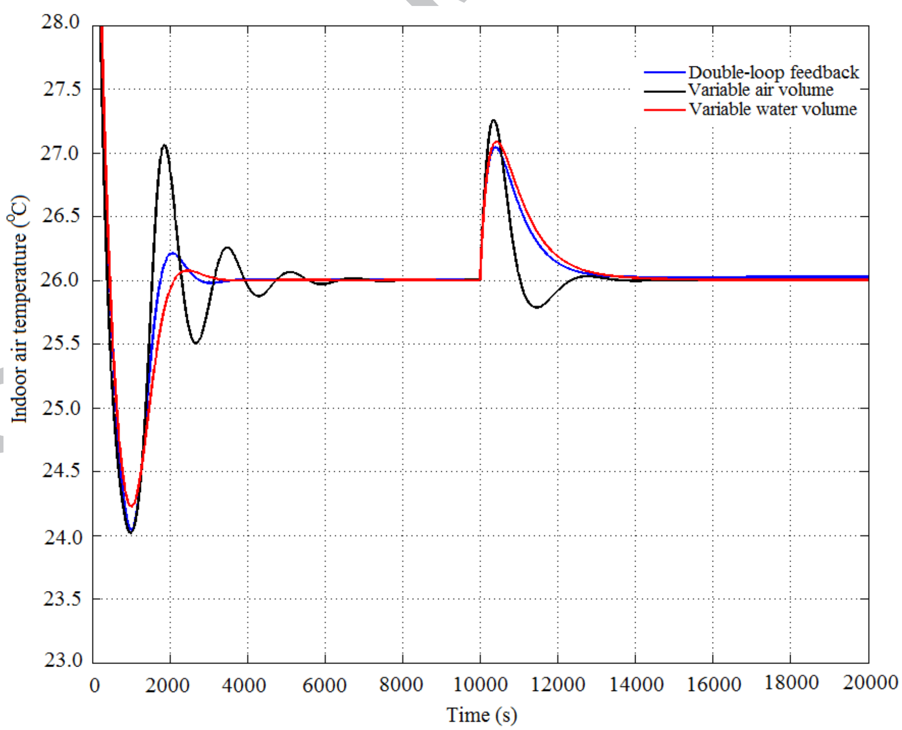

Fig. 9. The variation of indoor air temperature effected on three control methods

Table

Table 1 The performance parameters of surface heat exchanger

name	value	unit
Single discharge pipe cooling area	24.06	m^2
Frontal area	1.87	m^2
The cross-sectional area of water	0.00407	m^2
Water storage	6	kg
Air storage	1	kg
The calculation of heat-transfer coefficients	$k=\left(\frac{1}{41.5v_y^{a0.52}\xi^{1.02}}+\frac{1}{325.6\omega^{0.8}}\right)^{-1}$	$W/(m^2.^oC)$

Table 2 Nominal working condition of standard refrigerating capacity

Indoor air temperature (°C)	Relative humidity (%)	Inlet water temperature (°C)	Water temperature difference of fan coil (°C)	Steady pressure difference of fan coil (pa)
27.0	19.5	7.0	5.0	0

Table 3 The information of the variable water volume working conditions

Rate of water volume (%)	60%	80%	100%
Air flow rate (m^3/h)	2670	2668	2673
Inlet air temperature (^{o}C)	27.4	26.8	27.0
Inlet air wet bulb temperature (^{o}C)	18.8	19.3	19.2
Inlet chilled water temperature (^{o}C)	6.5	6.8	6.7

Table 4 Information of simulation cases

Parameters / Cases	Temperature gradient (°C)	Air flow rate (kg/s)	Chilled water rate (kg/s)
Case 1: Double-loop feedback	0.005	Varied	Varied
Case 2: Variable air volume	\	0.6	Varied
Case 3: Variable water volume	\	Varied	1.04

Highlights

- The model of heat transfer of surface cooler was established by Matlab/Simulink.
- A serial of experiments were conducted to variation of the model of heat transfer of surface cooler.
- The decoupled control pattern can shorten the system regulation time and improve performance.