SIMULATION ANALYSIS ON THE FRESH AIR HANDLING UNIT WITH LIQUID DESICCANT TOTAL HEAT RECOVERY

Xiaoyun Xie, Yidan Tang, Xiaoqin Yi, Shuanqiang Liu,Yi Jiang

Department of Building Science and Technology, Tsinghua University,
Beijing 100084, China

ABSTRACT

A liquid desiccant fresh air processor is presented whose driven force is low-grade heat (65~70℃ hot water). Inside the processor, the desired cooling source for air's dehumidification is indoor exhaust air's evaporative cooling energy. Multi-stage structure is used to get higher total heat recovery efficiency of indoor exhaust air. The mathematical model of the fresh air processor is set up and realized by SIMULINK procedure. As the practical liquid desiccant fresh air processor is developed, its performance is tested with average COP (cooling load divided by heat source) 1.3. Besides, the inside detailed parameters of practical facility for each heat transfer unit are acquired through testing. By comparing the tested and simulated data, the mathematical model is corrected by changing the main heat transfer units' characteristic coefficients, such as the volumetric mass transfer coefficient of air-water evaporative cooling module and the KA of sensible heat exchangers. Finally, using the corrected simulation model, the control strategy of the fresh air-handling unit is discussed, which indicates the imported strong solution's flowrate and the number of working stages are two effective regulated parameters and can be controlled singly or together at different outdoor conditions and for different indoor demands.

KEY WORDS

Liquid desiccant, fresh air handling unit, testing, simulation

INTRODUCTION

As a new potential humidity control method, liquid desiccant air conditioning system has aroused considered public attention. Compared with ordinary systems, due to humidity independent control, it can avoid condensed water and wet surface that is the breeding ground of mildew and bacteria. Also it can realize total heat recovery of indoor exhaust air using liquid desiccant and water as the medium. Besides, it can be driven by low grade heat, for instance the 65~80℃ hot water. According to these benefits, many researchers have carried on study around liquid desiccant systems. H.M.Factor et al.(1980) studied a packed bed dehumidifier/regenerator with liquid desiccants for solar air conditioning. P. Gandhidasan et al.(2004) set up a simplified model for air dehumidification with liquid desiccant. Liu Xiaohua et al.(2005) establish a performance test-bed for cross-flow dehumidifier/regenerator and get the mass transfer correlation from experimental results. Zhang Xiaosong et al.(2004) set up a liquid desiccant cooling system and carry experimental studies.

While for practical liquid desiccant facilities' development, Jiang Yi et al.(2004) developed a basic module for air and liquid desiccant's heat and mass transfer and used this basic module to construct multi-stage configurations to reduce the process' irreversibility and then constructed multiform fresh air handling units. Chen Xiaoyang et al.(2005) accomplished the first practical project of humidity independent control air-conditioning system based on hot-water powered liquid desiccant system.

When there are different sources for use, the grade of heat source for solution regenerating and cooling source for dehumidifying must be matched. In the present work, low grade heat of 65~70℃ hot water is used as the driven force and a newly developed fresh air handling unit with liquid desiccant total heat recovery is introduced, which uses indoor exhaust air' s evaporative cooling energy as dehumidifying cooling source. This fresh air handling unit can supply dry air with proper temperature just using evaporative cooling but no use of cold water. Its performance is analyzed first by simulation with established SIMULINK model. Then as the practical facility is developed, the testing performance is obtained and used to verify the simulation model. The control strategy of the fresh air handling unit is discussed by simulation finally.

PRINCIPLES OF THE HOT WATER POWERED LIQUID DESICCANT SYSTEM

A hot-water powered liquid desiccant air conditioning system consists of outdoor air processors, solution's regenerator and solution tanks. Generally one regenerator works for several outdoor air processors. An illustrational system is shown in Fig. 1, whose driven source is the waste heat of BCHP system. We just study the liquid desiccant system here. In summer work condition, the strong solution is pumped to fresh air handling units for dehumidification and it is diluted. The diluted solution, which has absorbed moisture from fresh air, is pumped to the regenerator to be concentrated. In the solution circuit, there are two solution tanks to keep strong and weak solution separately, which work as energy storage tanks and for peak load shifting.

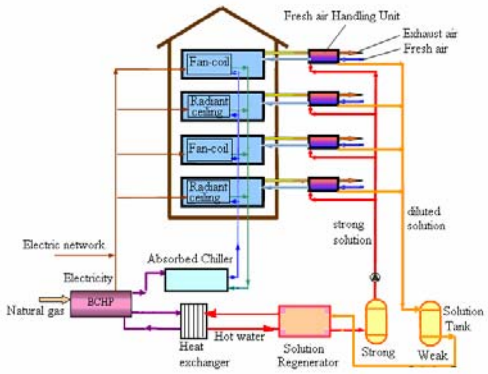


Fig. 1 Schematic diagram of liquid desiccant dehumidification system

STRUCTURE OF FRESH AIR-HANDLING UNIT WITH LIQUID DESICCANT TOTAL HEAT RECOVERY

Heat and mass transfer unit

The heat and mass transfer unit (Jiang Yi et al 2004) is the basis of the outdoor air processor, as shown in Fig. 2. The solution is driven by a solution pump from the base trough to the top of padding for spraying. Then inside the padding, air and solution are directly contacted to transfer moisture and heat with each other. In the solution circuit, there is a heat exchanger taking away or supplying heat of vaporization released from or absorbed by the heat and mass transfer process. Besides the inside solution circuit, there is another strand of solution enters into the base solution trough from one side and flows out from the other in opposite direction with the processed air. The flowrate of the inside circulating solution is one order bigger than the outside strand, while the former is to ensure heat and mass transfer process and the other is for air-solution matching of multi-stage process (shown in Fig. 3).

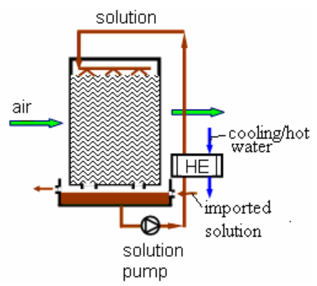


Fig. 2 Schematic diagram of basic heat and mass transfer unit (cooling water)

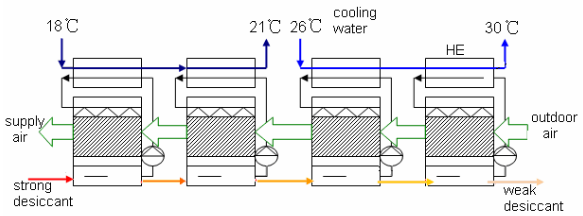


Fig. 3 Schematic diagram of processor of four stages

The way of “Dehumidification/humidification while cooling/heating” and multi-stage process both can greatly increase the unit’s performance. Using the heat and mass transfer unit, the processed air’s temperature can be controlled by regulating the temperature and flowrate of cooling or hot water of the heat exchanger, and the air’s humidity can be controlled by changing the imported solution’s concentration. Thus the processed air’s temperature and humidity can be controlled independently.

The basic unit can realize a variety of air handling processes such as dehumidifying and cooling, humidifying and heating, and accordingly, the fresh air handling unit can work for annual conditions and it becomes the basic module of both fresh air handling unit and solution’s regenerator.

The structure of fresh air handling unit

Using the basic heat and mass transfer unit, we construct the outdoor air processor shown in Fig. 4. The processor is made up of three total heat recovery stages for dehumidification process and one stage with air-cooling coil merely for supply air’s cooling. The total heat recovery stage consists of two heat and mass transfer units and one plate heat exchanger in the middle. In the upper heat and mass transfer unit, return air directly contacts with the spraying water for evaporative cooling then the produced cooling water is pumped into the middle heat exchanger to cool solution in the other side. While inside the lower unit, fresh air directly contacts with the cooled spraying solution for dehumidifying process. We use multi-stages to get higher total heat recovery efficiency. In winter mode, the upper-layer is shut down, the 35~40℃ hot water is supplied into the middle heat exchangers to heat the spraying solution and further to heat and humidify the dry and cold outdoor air.

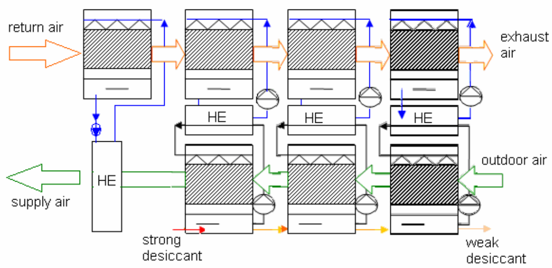


Fig. 4 Outdoor air processor with return air‘s evaporative cooling for total heat recovery

THE FOUNDATION OF MATHEMATICAL MODEL AND SIMULATION

The mathematical model of the fresh air-handling unit is established and the corresponding simulation model is set up using SIMULINK procedure. By the SIMULINK model, we get the preliminary performance of the fresh air processor.

Mathematical model of the fresh air handling unit

In the fresh air-handling unit, there are mainly two kinds of heat transfer units, which are the heat and mass transfer modules and sensible heat exchangers. The heat and mass transfer modules include the solution-air heat and mass transfer module and water-air evaporative cooling module. And the sensible heat exchangers include the water-solution plate heat exchanger and air's cooler.

Firstly, the solution-air heat and mass transfer module are adiabatic cross-flow heat and mass transfer unit. The desiccant, flowing across the direction of the airflow, is distributed over the packing by gravity and absorbs (or releases) moisture as it comes into contact with the air. We use the established air-solution cross flow model (Liu Xiaohua et al 2005) to describe the heat and mass transfer process. In this model, the three-dimensional module is simplified into a two-dimensional problem considering in the module the surfaces parallel to the air flow uniform, so just the air flow-parallel surface's heat and mass transfer process is studied. The numerical method is used and the surface is divided into infinite number of differential elements and in each element the volumetric mass transfer coefficient is obtained by experimental data (Liu Xiaohua et al 2005) and shown in formula (1).

$$ka = 5.088 \cdot F_a^{0.7329} \cdot F_z^{0.4203} \cdot \left(1 - \frac{\xi}{100}\right)^{1.5328} \qquad (1)$$

Fa and Fz represents cross sectional mass flow rate of air and solution (kg/m^2•s)

The numerical model is solved by setting up the simulation procedure using SIMULINK tool, by which the calculated average enthalpy efficiency (0.4~0.45) and average humidity ratio efficiency (0.45~0.5) is obtained and the air and desiccant's average outlet parameters of the whole module is obtained as well.

Then, for water-air evaporative cooling modules, the same simplified and numerical method is used as the solution-air heat and mass transfer unit and just the volumetric heat and mass transfer coefficient is different. However, as we don't find the experimental data for air and water's heat and mass transfer process in the chosen kind of padding, an estimated value (5 kg/(m^3.s)) by experience is used in the simulation model which is expected to be corrected by the after testing of the practical installation. For water-air heat and mass transfer unit, the volumetric mass transfer coefficient mainly depends on the mass flowrate (per section area) of air and water, a supposed constant value is reasonable when the mass flowrate of air and water are both constant.

Finally, for the sensible heat exchangers, we use $\varepsilon - NTU$ method for both air cooler and plate heat exchangers. The exchangers' KA , which is product of heat transfer coefficient and heat transfer area, represents the size of a heat exchanger and as its designed characteristic coefficient in the SIMULINK model.

Performance analyzed by simulation

By using the established mathematical model introduced above, the preliminary performance of the fresh air-handling unit is analyzed.

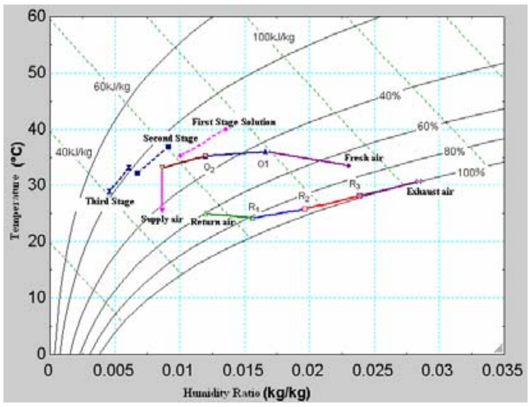
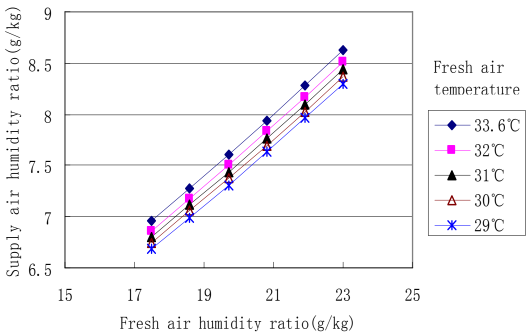


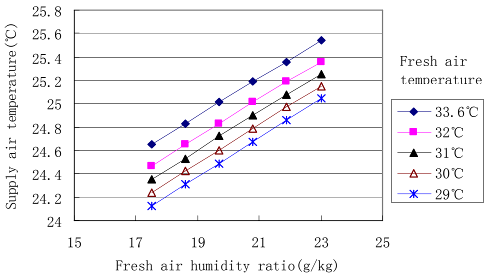
Fig. 5 Air process of fresh air handling unit

Fig. 5 shows the handling process of fresh air, return air and spraying solution inside each dehumidification stage in the psychometric chart. As can be seen, the fresh air (33.6℃, 23g/kg) is dehumidified stage-by-stage to reach a proper supply air' s state of both humidity and temperature (25.5℃, 8.6g/kg). Both the exhaust air's humidity ratio and enthalpy can be higher than fresh air which indicates that the total heat recovery efficiency is fairly high.

Fig. 6 shows supply air's state varying relation with fresh air' s state changing without any control action. As can be seen from Fig. 6(a), the outdoor air's humidity ratio is the main factor influencing supply air's humidity ratio and as the outdoor air's temperature is constant, the humidity ratio can be changed linearly with fresh air's humidity ratio. Fig. 6(b) shows the supply air's temperature influenced both by fresh air' s humidity ratio and temperature, while the maximum varying range is less than 1.5℃ which means outdoor air's state influences supply air's temperature not obviously.



(a)Simulated supply air humidity ratio



(b) Simulated supply air temperature

Fig. 6 Supply air state influenced by fresh air state.

TESTING PERFORMANCE AND VALIDATION OF THE SIMULATION MODEL

Testing performance of the fresh air handling unit

The fresh air-handling unit is developed and put into use in Tsinghua Low-Energy Building. We tested its performance under outdoor air conditions with temperature 28~34℃ and humidity ratio 12~17g/kg. The testing results are shown in Fig. 7~Fig. 8, from which we can see, supply air's temperature is 23~24℃ and humidity ratio is 6.5~8.5g/kg, satisfying the handling demand.

We define the fresh air's coefficient of performance (COP_f) as the enthalpy difference of fresh air and supply air divided by the removal moisture's latent heat of vaporization, which is expressed by formula (2).

$$COP_f=(h_o-h_s)/[(d_o-d_s)\bullet r_0] \qquad (2)$$

Testing COP_f is shown in Fig. 9 and it changes from 1.3~1.8 and more or less linearly with the outdoor air's relative humidity.

If we consider solution's regenerated efficiency COP_r, then the efficiency of the whole hot water powered liquid desiccant dehumidification system (COP_{sys}) is the product of COP_f and COP_r, as expressed by formula (3).

$$COP_{sys}=COP_f\bullet COP_r \qquad (3)$$

The efficiency of regenerator was also tested and some typical work conditions are shown in Table 1. The average COP_r is about 0.77 when hot water' s inlet temperature is about 65℃, thus the average COP_{sys} is 1.3.

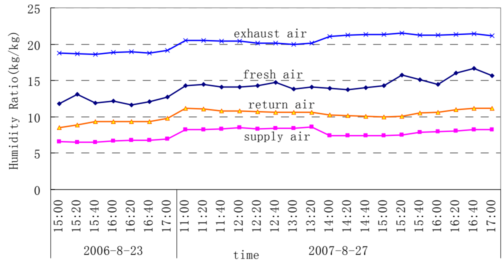


Fig. 7. Testing humidity ratio of fresh air, supply air, return air and exhaust air.

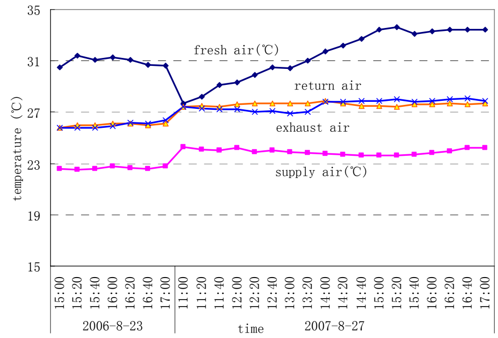


Fig. 8. Testing temperature of fresh air, supply air, return air and exhaust air.

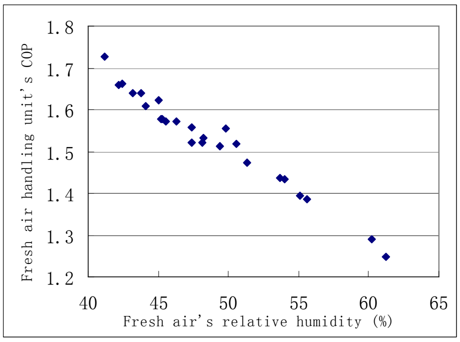


Fig. 9. Testing COP_f of the fresh air-handling unit.

Validation of the simulation model

For testing of this fresh air-handling unit, we not only tested the air's state but also water and solution state of every total heat recovery stage. So we can validate and correct the characteristic coefficient of each heat transfer device such as the mass transfer coefficient and KA of sensible heat exchangers.

First, for water-air evaporative cooling module, as we have no experimental relations to express its mass transfer performance, the supposed volumetric mass transfer coefficient β_v is not exact. So we use the testing data to deduce the real mass transfer coefficient β_v by choosing typical work conditions that is shown in Table 2. The NTU is expressed by formula (4).

$$NTU=\beta_v \cdot V/M_a \qquad (4)$$

Where M_a is return air's mass flowrate.

From Table 2 , we use ka -4.2 kg/(m^3.s) for every stage' s simulation, as the spraying water' s temperature is given by tested value, the difference of out water' s simulated and testing temperature can be very small (0.01~0.15℃) and within permitted error range. So we correct the volumetric mass transfer coefficient for air-water heat and mass transfer units to 4.2 kg/(m^3.s).

Secondly, for solution-air heat and mass transfer unit, as stated before, we use the experimental relation to describe the volumetric mass transfer coefficient for simulation. The tested fresh air's state and spraying solution's temperature and density of each module are as given conditions for the simulation model to get simulated supply air's state and out solution's concentration. The comparisons of testing and simulation results are shown in Fig. 10~12.

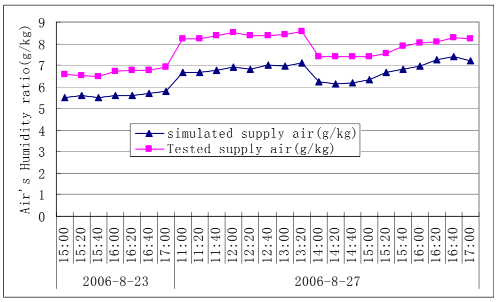


Fig. 10 Comparison of supply air's humidity ratio

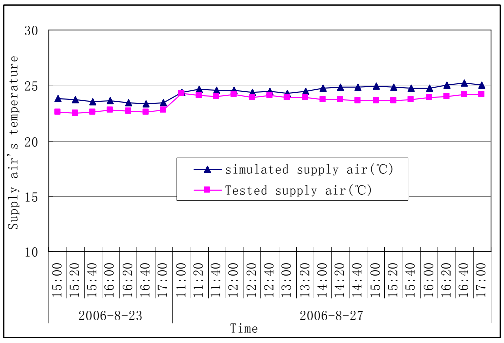


Fig. 11 Comparison of supply air's temperature

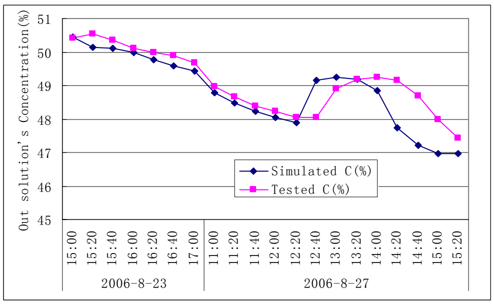


Fig. 12 Comparison of out solution's concentration

From the above three figures, we can see that, for average difference, the simulated supply air's humidity ratio is 1g/(kg•air) lower than the tested value, the simulated supply air's temperature is 0.8℃ higher than tested value and the simulated out solution's concentration is 0.28% lower than tested value. There are several reasons to cause the difference of simulated and tested data. One is testing apparatus' error. We use self-recording temperature and relative humidity sensor to test air's condition, with apparatus' precision is 0.3℃ for temperature and 3% for relative humidity, corresponding humidity ratio's error is 0.75 g/kg. Second reason is that supply air's temperature and humidity on the outlet surface isn't uniform and the testing data may not disclosure the real inhomogeneity. Besides, the simulation model is simplified to an ideal model that the padding can be fully used and only two-dimensional heat and mass transfer is considered, while for practical equipment, there must be some inefficiency area as the solution's spraying can't be infinitely uniform. If the inefficiency area reach to 10% of the total heat and mass transfer area, there will be a 0.2~0.4 g/kg increase for supply air's humidity ratio. Thus, considering testing errors and simplification of the mathematical model, the SIMULINK model can be used to simulate solution-air heat and mass transfer process.

Finally, for sensible heat exchangers (the plate heat exchanger of solution and water and the air cooling coil), from tested data we can get the heat exchangers' practical KA to correct the given KA at designing step. Take one of the three plate heat exchangers for example, as is shown in Table 3. The corrected KA of the plate heat exchanger is 4.0 while the designed value is 3.7. For the air cooling coil, the same method is used and will not be discussed here.

Till now, all the main components' characteristic coefficients have been corrected for simulation model by comparing with tested data. This corrected model can be used to describe the fresh air handling unit's performance under different inlet conditions more accurately.

CONTROL STRATEGY OF THE FRESH AIR HANDLING UNIT

We design the fresh air handling unit at outdoor worst-condition, which is the hottest and wettest weather state. While generally, outdoor air's condition is moderate and in these conditions, how to control the fresh air handling unit to save energy is very important. Fig. 6 shows supply air's humidity ratio increases linearly with outdoor air humidity ratio increasing. The goal of control strategy is to keep supply air's humidity ratio constant. One way is to change the imported strong solution's flowrate, which can control the supply air's humidity ratio continuously. However, if outdoor air is very dry, the strong solution's flowrate must be adjusted to be very low which can not be easily realized for practical regulation, and at these conditions, we turn off one or two stages, that is to say, the way of stages regulation.

Fig. 13 shows relations of supply air humidity ratio with outdoor air humidity ratio and imported solute's mass flowrate, for different number of stages working, while using the corrected SIMULINK model. For different stages working, the possible handling range can be seen clearly in Fig. 13 and we can find the control method at any condition. For example, if outdoor humidity ratio is 20 g/kg, and the desired humidity ratio of supply air is 8g/kg, so we must use three stages and the imported solute's flowrate needs to be about 0.05kg/s, if outdoor air changes to be 16g/kg, and desired supply air is 8g/kg as well, we can only use two stages and the imported solute's flowrate needs to be about 0.15kg/s. For this control strategy, the imported solution's concentration is assumed constant to be 54%, that is to say, the corresponding regenerator's control target is to keep out solution's concentration constant. And for work conditions in Fig. 13， the outdoor air's temperature is constant 33.6℃ and when it changes the correction can be simulated with the changing trend like Fig. 6 shown.

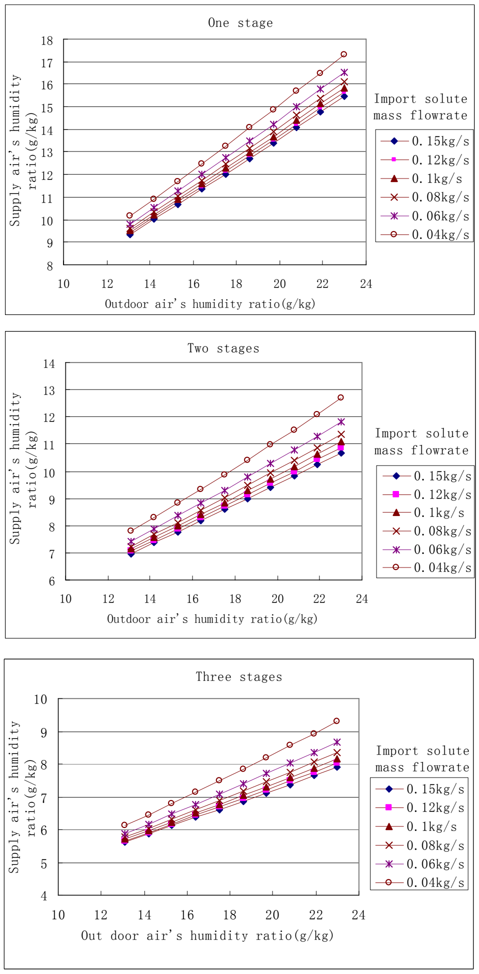


Fig. 13 Control strategy of fresh-air handling unit

CONCLUSIONS

A newly developed fresh air handling unit with liquid desiccant total heat recovery has been introduced, which uses indoor exhaust air's evaporative cooling energy to cool the dehumidification process and also cool the supply air. This fresh air handling unit is driven by imported strong solution which is produced by using low grade heat (65~70℃ hot water). The mathematical model is set up and the corresponding SIMULINK model is erected for simulation.

As the fresh air handling unit has been developed and put into use, the practical facility's performance is tested with fresh air handling unit's COP_f is 1.3~1.8, and the whole hot water powered liquid desiccant system's average COP_{sys} is 1.3. The tested performance is then used to validate and correct the simulation model. By comparing every main heat and mass transfer unit's designed and real performance, their characteristic coefficients are corrected. And by using the corrected simulation model, the control strategy is found at different outdoor humidity ratios, taking the imported strong solution's flowrate and the number of working stages as two effective regulated parameters to meet indoor demanded conditions at different outdoor states.

REFERENCE

Chen Xiaoyang, Jiang Yi, et al. 2005. Field study on independent dehumidification air-conditioning system - I: Performance of liquid desiccant dehumidification system. ASHRAE Transactions, v 111, PART 2:p 271-276

H.M.Factor, G.Grossman. 1980. A packed bed dehumidifier/regenerator for solar air conditioning with liquid desiccants. Solar energy, 1980, 24(6)：541-550

Jiang Yi, Li Zhen, et al.2004. Liquid desiccant air-conditioning system and its applications, Heating Ventilating &Air Conditioning, 34(11): 88-97

Liu Xiaohua, Zhang Yan, et al.2005. Heat and mass transfer performance analysis on liquid desiccant dehumidification process. Heating Ventilating &Air Conditioning.

P. Gandhidasan. 2004. A simplified model for air dehumidification with liquid desiccant, Solar Energy, vol76.

Zhang Xiaosong， Yin Yonggao, et al. 2004. Experimental study on the performance of liquid desiccant air cooling with energy storage. Journal Of Engineering Thermophysics.

Table 1. Testing performance of regenerator

WORK CONDITION	FRESH AIR			HOT WATER		SOLUTION IN		SOLUTION OUT		MOISTURE REMOVAL RATE	REGENERATOR EFFICIENCY
	Dry bulb	Humidity ratio	Flow rate	Inlet T	Outlet T	Mass Flow rate	Density	Mass Flow rate	Density		
	℃	g/kg	m3/h	℃	℃	kg/h	g/ml	kg/h	g/ml	kg/h	
1	34.1	15.05	2010	62.5	53.4	0.729	1.463	0.7	1.481	30.9	0.77
2	34.5	14.55	2010	63	54.2	0.713	1.474	0.682	1.494	32.3	0.8
3	34.2	14.2	2010	63.9	55	0.68	1.498	0.649	1.52	31.9	0.76
4	33.9	13.69	2010	63.9	55.2	0.677	1.5	0.645	1.523	32.2	0.74
5	32.8	12.34	2010	64.8	56	0.663	1.511	0.631	1.535	33.1	0.79
6	32.9	13.22	2010	64.8	56.3	0.659	1.514	0.628	1.538	32.3	0.77
7	32.9	13.63	2010	64.8	57.2	0.649	1.522	0.623	1.542	27.3	0.85

Table 2. Air-water evaporative cooling module 's volumetric mass transfer coefficient validation

RETURN AIR MODULE	SIMULATED PARAMETER		WORK CONDITION 1			WORK CONDITION 2			WORK CONDITION 3		
	ka	NTU	Tested spraying water	Tested out water	Simulated out water	Tested spraying water	Tested out water	Simulated out water	Tested spraying water	Tested out water	Simulated out water
	(kg/m3/s)		℃	℃	℃	℃	℃	℃	℃	℃	℃
1	4.2	0.78	26.3	24.8	24.79	27.1	25.6	25.66	28.6	27	26.79
2	4.2	0.78	24.1	22.9	22.91	25.2	24.1	23.98	26.5	24.7	24.83
3	4.2	0.78	23.9	21.8	21.84	24.4	22.8	22.7	24.4	22.8	22.8
4	4.2	0.78	18.7	18.4	18.29	20.2	19.9	19.9	20.4	20.3	20.15

Table 3 Correction of sensible heat exchanger's KA using testing data

HEAT EXCHANGER 3	WATER			SOLUTION		LOG MEAN TEMPERATURE DIFFERENCE	TESTED KA
	Inlet Temperature	Outlet Temperature	Mass Flowrate	Inlet Temperature	Outlet Temperature		
Work condition	℃	℃	kg/s	℃	℃	℃	kW/℃
1	21.8	23.9	1.11	27.4	23.4	2.4	4.0
2	22.8	24.4	1.11	27.7	24	2.1	3.6
3	23	24.7	1.11	27.5	24.2	1.9	4.2
4	23.2	24.8	1.11	27.1	24.9	2.0	3.7

NOMENCLATURE

Roman letters

COP	coefficient of performance	
d	air's humidity ratio	g/kg
F	cross-sectional mass flow rate	kg/(m2• s)
ka, β_v	volumetric mass transfer coefficient	kg/(m3• s)
h	air's enthalpy	kJ/kg
M	Mass flowrate	kg/s
NTU	number of heat transfer unit	
r_0	heat of vaporization	kJ/kg
V	volume	m^3
ξ	concentration of solution	

Subscripts

a	air
f	fresh air handling unit
o	outdoor air
r	regenerator
s	supply air
sys	system
z	solution