Application of heat pump combined two-stage desiccant wheel fresh air system of residential buildings in mixed climate zone

Shaochen Tian^{1,2}, Xing Su^{1,2,*}, and Xu Zhang^{1,2}

¹ School of mechanical engineering, Tongji University, 1239 Siping Road, 200092 Shanghai, P. R. China

² Key Laboratory of Performance Evolution and Control for Engineering Structures of Ministry of Education, Tongji University, 1239 Siping Road, 200092 Shanghai, P. R. China

Abstract. The building requires dehumidification for a long period of time in mixed climate zone of China. As a conventional method for dehumidification, vapor compression systems remove the water vapor by cooling the process air below dew point. This system consumes a lot of energy for reheating the air to meet the requirement of supply air temperature. A heat pump combined with two-stage desiccant wheel (TSDW&HP) is proposed as an air conditioning and dehumidification system in this study. The operation performance of proposed system applied in a hypothetical residence with 3 residents was investigated and simulated by using TRNSYS software. The operation modes of the system are discussed for different scenarios of season and outdoor air humidity ratio. In dehumidification season, fresh air deals with all of the latent load. In air conditioning season, fresh air deals with all of the moisture load with part of the cooling load. When evaporation temperature of HP is reduced and more moisture load is processed by evaporator in air conditioning season, there is a balance point between the performance of DWs and heat pump. The energy consumption of TSDW&HP fresh air system was compared with a conventional fresh air conditioner during dehumidification season and air conditioning season. It was found that the energy-saving potential of this system is 27.3% compared with conventional air conditioner.

1 Introduction

Nowadays, along with the development of construction industry, building energy consumption problems are mentioned constantly and energy consumed by buildings is increasing. Building energy consumption accounts for more than 40% of the world energy demand and more than half of the building energy is consumed by heating, ventilation and air-conditioning systems. Building energy consumption of China may increase to about 35% of total national energy consumption in 2020 [1].

In hot and humid climate regions of China, outdoor air temperature is usually about 30-35°C and Outdoor air relative humidity is approximately 70-80% in summer. During this period, both cooling and dehumidification are necessary. In transition season, dehumidification is much more important than cooling because outdoor air temperature is about 25°C and outdoor relative humidity is still higher than 70%. For conventional air conditioners, water vapor is removed by cooling the process air below dew point. When applying this dehumidification method, low evaporation temperature, about 5°C, is required and more energy is consumed by air conditioner. Sometimes, reheating is necessary to meet the requirement of supply air temperature, which results in energy waste. In contrast, air-conditioning systems that temperature and humidity are controlled

independently are more suitable for this climate zone. In this system, cooling panel only deals with indoor sensible load and indoor moisture gain is removed by dehumidified fresh air. This system is more energy efficient than conventional air conditioner due to high evaporation temperature could meet the requirement of cooling. This system could also provide a more comfortable thermal environment for residents. When using this system, a low energy consumption dehumidification method is important.

Desiccant wheel (DW) systems have been broadly applied in the fields of production but are seldom used in civil buildings especially in residential buildings. As a matter of fact, DW system is more suitable for residential buildings than liquid dehumidification systems because it's more compact and does not have many accessory equipment [2]. The main problem is that the regeneration temperature of traditional DW system is usually higher than 120°C when dehumidification rate is large and a huge amount of energy is consumed by regeneration air heating devices. Air source heat pumps (ASHP) are common in residential buildings in hot and humid climate regions of China. When the condensation heat of ASHP is applied to heat regeneration air of DW, energy consumption of DW system could be reduced. Different combination methods of DW and ASHP have been proposed by many scholars. The combination

^{*} Corresponding author: <u>suxing@tongji.edu.cn</u>

methods could be separated majorly into two types: increasing the condensing temperature of ASHP and reducing the regeneration temperature requirement of DW. Song et al. [3] combined DW with two-stage heat pump system and the regeneration air could be heated up to 70°C. This system is not suitable for residential buildings due to the complexity of the system. Hao et al. [4] applying the condensation heat of high temperature ASHP to regenerate DW. About 40% energy could be saved compared with conventional system. Zhang et al. [5] developed an ASHP with refrigerant R142b and regeneration air temperature could reach 79.2°C after heated by condenser. The COP of the ASHP is only 2.26. Sheng et al. [6-9] did much experiment and simulation of high temperature heat pump and desiccant wheel system. The results show that this system is much more energy efficient than conventional vapor compression system and hybrid desiccant cooling system. Ge et al. [10] combined DW with conventional heat pump and the evaporator before the DW deals with part of moisture load. Tu et al. [11-13] combined heat pump with twostage desiccant wheel and intercooling is achieved by adding an additional evaporator between the two DWs. In this system, the two DWs could be regenerated at a relatively temperature. low However, when dehumidification rate of building is large, condensing temperature of HP may not high enough to regenerate the two DWs effectively.

In this paper, a TSDW&HP fresh air system is applied in a hypothetical residence with 3 people in Shanghai. The operation modes of the system are discussed for different scenarios of season and outdoor air humidity ratio. After simulated by using TRNSYS software, the optimal operation mode under different conditions is proposed. At last, the total energy consumption of TSDW&HP fresh air system during dehumidification season and air conditioning season is calculated and compared with a conventional air conditioner.

2 TSDW&HP fresh air system

The schematic diagram of TSDW&HP system is shown in Fig.1. There are three evaporators (7,8 and 9) and two DWs (10 and 11) in this system. The function of evaporator1 and evaporator2 is to increasing process air relative humidity and reducing process air temperature. Besides, evaporator1 could remove part of the water vapor in the fresh air and the amount is linked with the evaporation temperature. As indicated in the figure, the system could be operated at two modes. When the system is operated at two-stage mode, outdoor fresh air is cooled down by evaporator1 before flowing into DW1. After being dehumidified by DW1, it is cooled by evaporator2 and then be dehumidified by DW2. Finally, fresh air is cooled by evaporator3 to the designed supply air temperature. If fresh air is bypassed from evaporator1 and DW1 and cooled by evaporator2 directly, singlestage operation mode is achieved. Regeneration air is heated by the condenser and separated into two ways to regenerate DW1 and DW2. When regeneration air

temperature is not high enough after heated by the condenser to remove all of the water adsorbed by DWs, the two electric heaters before two DWs are in-operation to further improve the regeneration air temperature.



Fig. 1. Schematic of TSDW&HP system. (1) Process fan, (2) regeneration fan, (3) compressor, (4) four-way valve, (5) condenser, (6) expansion valve, (7) evaporator1, (8) evaporator2, (9) evaporator3, (10) desiccant wheel (DW1), (11) desiccant wheel (DW2), (12) electric heater (EH1), (13) electric heater (EH2)

3 Modelling

The model of DW is from the component in TRNSYS. The model of compressor, evaporator, condenser and expansion valve is according to Sheng et al. [8].

3.1 Desiccant wheel

The model of DW in TRNSYS is based on the two characteristic potential functions F1 and F2 proposed by Hove [14]and Jurink[15]. The potential functions of silica gel are indicated in Eq. (1) and (2).

$$F_1 = \frac{-2865}{T^{1.49}} + 4.344 d^{0.8624} \tag{1}$$

$$F_2 = \frac{T^{1.49}}{6360} - 1.127 d^{0.07969}$$
(2)

In the Eqs. (1)-(2), T is the air temperature. d is the air humidity ratio.

And the potential functions should be corrected by two coefficients \mathcal{E}_{F1} and \mathcal{E}_{F2} proposed by Schultz[16].

$$\varepsilon_{F_1} = \frac{F_{1D} - F_{1P}}{F_{1R} - F_{1P}}$$
(3)

$$\varepsilon_{F_2} = \frac{F_{2D} - F_{2P}}{F_{2R} - F_{2P}}$$
(4)

In the Eqs. (3)-(4), subscript D is the actual outlet process air status points. Subscript P is the inlet process air status point. Subscript R is the inlet regeneration air status point.

The required regeneration air temperature could be calculated when process air inlet temperature and humidity ratio, regeneration air inlet humidity ratio and process air outlet humidity ratio (humidity ratio set point) is known.

3.2 Compressor

A simplified model is adopted to predict the process of compressor. The pressure loss of suction and exhaust is neglected. The lubrication and the effect of heat absorption of the shell are also ignored.

Mass flow of compressor is expressed as:

$$m_{com} = \frac{\lambda \cdot V_{th}}{v_{suc}}$$
(5)

where λ is volumetric efficiency, V_{th} is theoretical volumetric gas transmission of compressor, and v_{suc} is refrigerant specific volume at suction inlet of compressor.

Theoretical power of compressor is:

$$W_{i\hbar} = V_{i\hbar} \cdot \lambda \cdot p_e \frac{\kappa}{\kappa - 1} \left[\left(\frac{p_e}{p_e} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right]$$
(6)

where p_e is the evaporation pressure, p_c is the condensing pressure, and κ is the adiabatic index.

Input power of compressor is:

$$W_{in} = \frac{W_{ih}}{\eta_a \eta_a \eta_m} \tag{7}$$

where η_a is adiabatic efficiency, η_e is electric efficiency, and η_m is mechanical efficiency.

The temperature at compressor outlet is given as:

$$T_{ro,com} = T_{suc} \left(\frac{P_c}{P_e} \right)^{\frac{m-1}{m}}$$
(8)

where T_{suc} is the refrigerant temperature at compressor inlet.

3.3 Expansion valve

The thermal process in expansion valve is isenthalpic and the mass flow rate depends on the open degree of the expansion valve.

$$m_{r,exp} = C_D \cdot A \cdot \left[2 \rho_l \left(p_e - p_e \right) \right]^{0.5}$$
(9)

where C_D is the expansion valve coefficient, A is the flow area of expansion valve, and ρ_l is the inlet liquid density of expansion valve.

3.4 Evaporator and condenser

Tube fin exchanger is chosen as the type of evaporator and condenser. The condenser can operate at three zones (superheat zone, two-phase zone and subcooled zone) and the evaporator can operate at two zones (two-phase zone and superheat zone). The model is simplified and some assumptions are made:

- Refrigerant and air is one-dimensional counter flow.

- Heat conduction of axial is negligible and heat exchange with other equipment is ignored.

- The pressure drop of refrigerant in the pipe is ignored.

In each infinitesimal, the energy conservation equations are as follow:

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$$Q_{r} = m_{r} \left(h_{ri,i} - h_{ro,i} \right)$$
(10)

where m_r is the refrigerant flow rate, $h_{ri,i}$ is the inlet refrigerant enthalpy of i zone, and $h_{ro,i}$ is the outlet refrigerant enthalpy of i zone.

$$Q_a = m_a \left(h_{ao,i} - h_{ai,i} \right) \tag{11}$$

where m_a is the air flow rate, $h_{ai,i}$ is the inlet air enthalpy of i zone, and $h_{ao,i}$ is the outlet air enthalpy of i zone.

$$Q_{\mu} = \xi Q_{\mu} \tag{12}$$

where ξ is heat transfer loss coefficient between air side and refrigerant side.

$$Q_r = UA_i \left(T_{rm} - T_{am} \right) \tag{13}$$

where U is the heat transfer coefficient, A_i is the inner heat transfer area, T_{rm} and T_{am} are average temperature of refrigerant and air of i zone.

U is expressed as:

$$U = \left(\frac{1}{\alpha_r} + \frac{A_i}{\alpha_a A_o}\right)^{-1}$$
(14)

where α_r is heat transfer coefficient of refrigerant, α_a is heat transfer coefficient of air, and A_o is the external heat transfer area.

For single-phase zone, α_r :

$$\alpha_{r} = 0.023 R e_{r}^{0.8} P r^{0.3} \cdot \frac{\lambda_{r}}{d_{r}}$$
(15)

where λ_r is thermal conductivity of refrigerant. For two-phase zone, α_r :

$$\alpha_{r} = 0.023 R e_{r}^{0.8} P r^{0.3} \cdot \frac{\lambda_{r}}{d_{i}} \left[\left(1 - x \right)^{0.8} + \frac{3.8 x^{0.76} \left(1 - x \right)^{0.04}}{P r^{0.38}} \right]$$
(16)

where x is dryness of refrigerant.

For air side, α_a :

$$\alpha_{a} = 0.982 R e_{a}^{0.424} \left(\frac{S_{1}}{d_{f}}\right)^{-0.0887} \left(\frac{NS_{2}}{d_{f}}\right)^{-0.1590}$$
(17)

where S_1 is the space of adjacent fins, N is the number of tube rows, S_2 is the tube spacing along the air flow direction, and df is the diameter of fin root.

4 Validation

4.1 Validation for DW

The simulation results of DW are validated with the experimental results of Antonellis et al. [17] at different process air and regeneration air states (Tab. A.1). The validation results are shown in Fig.2.



Fig.2. Regeneration temperature and process temperature of DW.

Compared with the experimental results of Antonellis et al., the maximum relative error of simulation results is 7.13% for regeneration inlet air temperature (Tri) and 7.16% for process outlet air temperature (Tpo). This model is available to predict the performance of DW.

4.2 Validation for HP

The parameters of condenser and evaporator is according to [18] (Tab.3) and the simulated results of modified HP model are validated with [18]. The results are shown in Fig.3. and Fig.4.



Fig.3. Cooling capacity of HP at different outside temperature.





Compared with the experimental results of Chen et al., the maximum relative error of simulation results is 2.86% for cooling capacity and 1.05% for COP. The model can be used to analyse the performance of HP of the TSDW&HP system.

5 Operation optimization of TSDW&HP fresh air system

5.1 TRNSYS model and design information

In each operation mode, fresh air flow rate, regeneration air flow rate and dehumidification rate of the system is kept the same. The comparison of different operation modes is based on the energy consumption of the whole system, including energy consumed by compressor, the two fans, the two electric heaters and the driving motor of each DW. The hourly energy consumption of TSDW&HP fresh air system is simulated by using TRNSYS software. The model of TSDW&HP fresh air system is shown in Fig.5.

In the TSDW&HP fresh air system, fresh air deals with the whole latent load. In residential buildings, latent load mainly consists of moisture load of occupants and fresh air dehumidification load. A typical residence in Shanghai is hypothesized to calculate and discuss. The information of the residence is listed in Table 1.

According to the building information in Tab.1, fresh air flow rate is 135m3/h and indoor moisture load is 327g/h. In this system, supply air flow rate equals to fresh air flow rate and the humidity ratio of supply air could be calculated as 10.58g/kg. According to the work and rest time schedule of typical residence with 3 people, the systems is out-of-operation from 8:00 to 18:00 at weekday and in-operation at other times.



Fig.5. TSDV	V&HP	fresh	air	system	in	TRNSY
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Table 1. Design information of apartment

Parameter	Value
Area (m2)	90
Height (m)	3
Occupants	3
Design temperature in summer (°C)	26
Design relative humidity	60%
Air exchange rate	0.5
Moisture load per human (g/h)	109

5.2 Determination of operation modes under different weather conditions

Season of Shanghai could be divided majorly into four seasons according to Zhang [19]: namely air conditioning, dehumidification, ventilation and heating. In this paper, only air conditioning season and dehumidification season is discussed due to dehumidification only appears in these two seasons.

Firstly, the two operation modes of the system under different outdoor air humidity ratio are compared. Then the optimum supply air temperature in air conditioning season is proposed.

Table 2. Season division of Shanghai according to Zhang[1	9]
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Season	Start and end date		
Dehumidification	6.5-6.24 and 8.24-9.22		
Air conditioning	6.25-8.23		

5.2.1 Comparison of single-stage mode and twostage mode

As discussed in Chapter 2, the system could operate in either two-stage mode or single-stage mode. The main difference between the two operation modes is the dehumidification rate. When outdoor air humidity ratio is high, regeneration energy of single-stage mode is higher than two-stage mode. On the contrary, when outdoor air humidity ratio is low, single-stage mode can meet the requirement of dehumidification and the pressure drop of the system could be decreased.

In air conditioning season and dehumidification season, the minimum humidity ratio of outdoor air is 10.42g/kg and the maximum humidity ratio is 26.88g/kg. The comparison of the two operation modes will be conducted according to the outdoor air humidity ratio between 10.42g/kg and 26.88g/kg.

As fresh air only needs to deal with the whole latent load, supply air temperature of the two modes is kept around 26°C. On the other hand, outdoor air temperature has little influence on the energy consumption of the two modes due to energy conservation. For the sake of simplicity, outdoor air temperature is fixed at 26°C and outdoor air humidity ratio varies from 10.42g/kg to 26.88g/kg.

It is apparent that when the outdoor air humidity ratio is less than 10.58g/kg and temperature is lower than 26°C, outdoor air could be supplied directly without being processed. When the outdoor air humidity ratio is between 10.58g/kg and 26.88g/kg, single-stage mode or two-stage mode should be discussed. Energy consumption of the two operation modes according to outdoor air humidity ratio is shown in Fig.6.



Fig.6. Energy consumption comparison of single stage and two stage operation mode.

When outdoor air humidity ratio is less than 14.58g/kg, energy consumption of single-stage mode is less than that of two-stage mode. When operating at single-stage mode, only one driving motor of DW is in operation and the pressure drop of the system is small. Thus, the energy consumed by fans and driving motor could be saved at single-stage mode.

However, when outdoor air humidity ratio is higher than 14.58g/kg, energy saving potential of two-stage mode is remarkable compared with single-stage mode. The reason is that high regeneration temperature is required to remove all of the moisture content of DW in single-stage mode. Outlet air temperature of condenser is lower than the required regeneration temperature and electric heater have to operate at high load rate. As for two-stage operation mode, dehumidification rate of each DW is reduced compared with single-stage mode and the regeneration temperature requirement of each DW could be decreased.

In conclusion, the operation mode of TSDW&HP is: when outdoor air relative humidity is lower than 10.58g/kg, fresh air is supplied into the room directly. When outdoor air humidity ration is between 10.58g/kg and 14.58g/kg, the system operates at single-stage mode. When outdoor air humidity ratio is higher than 14.58g/kg, the system should operate at two-stage mode.

5.2.2 Supply air temperature in air conditioning season

Different from dehumidification season, cooing is necessary in air conditioning season. By reducing the supply air temperature of the system, fresh air can not only deal with the whole latent load but also part of the cooling load. The supply air temperature of fresh air could be reduced by decreasing evaporation temperature of HP. When evaporation temperature decreases, more moisture is condensed in evaporator1, which contributes to the reducing of dehumidification rate and regeneration temperature of the two DWs. The total energy consumption in air conditioning season of TSDW&HP fresh air system at different supply air temperature is calculated by using TRNSYS software. The cut-off humidity ratio point of single-stage and two-stage operation mode is 14.58g/kg. The results are shown in Fig.7.



Fig.7. Energy consumption of TSDW&HP system at different supply air temperature.

Energy consumption of the system is reduced when supply air temperature is lower than 26°C. When supply air temperature is higher than 22°C, energy consumption of the system decreases with the decreasing of supply air temperature. When supply air temperature is further the energy consumption increases. decreased, Apparently, fresh air should deal with part of the cooling load and there is an optimum setting point of supply air temperature. The reason is that with the increasing of evaporation temperature, energy consumed by HP decreases but the more energy is required to regenerate the two DWs. On the contrary, low regeneration temperature leads to high energy consumption of HP and low regeneration energy consumption of the two DWs. As a matter of fact, there is a balance point of HP and the two DWs. From the simulation results shown in Fig.8, when supply air temperature is controlled around 22°C in air conditioning season, energy consumption of the system is the lowest.

Thus, the optimized operation mode is: in dehumidification season, fresh air supply temperature is fixed around 26°C and in air conditioning season, fresh air supply temperature is controlled around 22°C by setting evaporation temperature of HP. Besides, single-stage mode is adopted when outdoor air humidity ratio is less than 14.58g/kg and two-stage mode is adopted when outdoor air humidity ratio is higher than 14.58g/kg.

6 Results and discussion

When TSDW&HP system is compared with a conventional air conditioner, the air conditioner only

processes fresh air and all of the latent load should be dealt by fresh air, too.

6.1 Hourly energy consumption of two fresh air systems

The hourly energy consumption of TSDW&HP fresh air system and conventional air conditioner during dehumidification season and air conditioning season could be calculated. The result is shown in Fig.8.



Fig.8. Hourly energy consumption of conventional system and TSDW&HP system.

It's apparent that except for the period of high humidity ratio time in dehumidification season, the energy consumption of TSDW&HP fresh air system is lower compared with conventional air conditioner. At high humidity ratio time period, supply air temperature is kept around 26°C and the evaporation temperature of HP is high. Dehumidification rate of the two DWs is large and more energy is consumed to regenerate DWs during this period.

The time period that supply air temperature is kept around 22°C should be extended. According to the hourly energy consumption results in Fig.8, the time period is from June 19 to September 5.

6.2 Energy saving potential of optimized TSDW&HP fresh air system

In air conditioning season, fresh air deals not only the whole humidity load but also part of the cooling load for both TSDW&HP fresh air system and conventional air conditioner. However, supply air temperature of conventional air conditioner is about 15°C and the supply air temperature of TSDW&HP system is about 22°C in the optimized operation mode. As a matter of fact, cooling is not necessary in dehumidification season. The purpose of low supply air temperature of conventional air conditioner is only to dehumidify the fresh air. Different supply air temperature deals with different cooling load in air conditioning season. The supply air of TSDW&HP fresh air system should be

processed by an additional air conditioner further to 15°C before compare with conventional air conditioner.

After the supply air of TSDW&HP fresh air system is cooled by an auxiliary air conditioner (assumed), energy consumption of the compressor is added to that of the TSDW&HP system. The energy consumption of the two systems in dehumidification season and air conditioning season is shown in Fig.9. The time period that supply air temperature of TSDW&HP system is kept around 22°C is from June 19 to September 5. Energy consumption of TSDW&HP fresh air system is lower than that of conventional air conditioner and the energysaving potential of TSDW&HP system is 27.3% compared with conventional air conditioner.



Fig.9. Total energy consumption of TSDW&HP system and conventional air conditioner.

7 Conclusion

In this paper, different operation modes of TSDW&HP fresh air system have been proposed for different seasons and outdoor air humidity ratio by calculating a typical residence in Shanghai. In dehumidification season, the supply air temperature is controlled at 26°C and fresh air only deals with all of the latent load of the residence. When outside air humidity ratio is between 10.58g/kg and 14.58g/kg, single-stage operation mode is better and when outside air humidity ratio is higher than 14.58g/kg, two-stage operation mode is applied. When fresh air supply temperature is controlled around 26°C in air conditioning season, little moisture load is processed by evaporator1 and energy consumption of DWs is high. There is a balance point between the performance of HP and DWs, which could be reflected by fresh air supply temperature. The best fresh air supply temperature in air conditioning season is 22°C in this case. The total energy consumption of TSDW&HP (operated at the optimized mode) fresh air system and conventional air conditioner in air conditioning season and dehumidification season has been calculated. TSDW&HP fresh air system is more energy efficient and the energy-saving potential of TSDW&HP fresh air system is 27.3% compared with conventional air conditioner.

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