Trans. Tianjin Univ. 2014, 20: 182−188
DOI 10.1007/s12209-014-2191-0

Optimization Analysis of High Temperature Heat Pump Coupling to Desiccant Wheel Air Conditioning System*

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Abstract: The high temperature heat pump and desiccant wheel (HTHP&DW) system can make full use of heat released from the condenser of heat pump for DW regeneration without additional heat. In this study, DW operation in the HTHP&DW system was investigated experimentally, and the optimization analysis of HTHP&DW system was carried out. The performance of DW had influence on the dehumidification (evaluated by dehumidification and regeneration effectiveness) and cooling load (evaluated by thermal and adiabatic effectiveness). The results show that the enthalpy increase occurred in all the experiments. Compared to the isosteric heat, heat accumulation in the desiccant and matrix material and heat leakage from regeneration side to process side have greater influence on the adiabatic effectiveness. Higher regeneration temperature leads to lower adiabatic effectiveness that increases more cooling load of the system. When the regeneration temperature is 63℃, the maximal dehumidification effectiveness is 35.4% and the satisfied adiabatic effectiveness is 88%, which contributes to the optimal balance between dehumidification and cooling.

Keywords: desiccant wheel; dehumidification; adiabatic effectiveness; performance optimization

The high temperature heat pump and desiccant wheel (HTHP&DW) system can make full use of both heating and cooling from the condenser and evaporator of a heat pump that provides thermal energy for desiccant regeneration without any additional heat. In this system, the sensible and latent loads are handled independently and the self-regeneration is achieved, which can reduce the environmental impact and save energy. Therefore, HTHP&DW system can be a new approach to the development of air conditioning system[1-3].

Desiccant wheel is the main component in the HTHP&DW system. Many experiments and numerical simulation analyses indicated that the operation parameters, structure parameters and physical parameters of desiccant wheel have influences on the rotary dehumidification[4-8]. Moreover, many investigations demonstrated that the deviation in the enthalpy during the actual adsorption process can influence the system cooling load. Kodama *et al*[9] put forward the adsorption process of DW on the psychrometric chart. The results show that the ideal performance is adiabatic operation. Zhang *et al*[10] analyzed the deviation of enthalpy based on the psychrometric variation at the inlet and outlet of process air. Mandegari and Pahlavanzadeh[11] proposed a novel index of adiabatic effectiveness to evaluate the effect on the cooling load of system by the deviation of enthalpy. Based on these researches, it has been concluded that the performance of DW affects not only dehumidification, but also cooling load. Therefore, the optimal operating condition of desiccant wheel is important to the development and optimization of HTHP&DW system.

Considering that the heat and mass transfer of DW is so complex that any factors, such as different desiccant materials, will lead to different results of quantitative performance, the silica gel rotor with the feature of low regeneration temperature was chosen to be analyzed in this study. The influences of operating parameters on the DW performance indexes were discussed and the optimization analysis of HTHP&DW system was also carried out.

Accepted date: 2013-08-05.
*Supported by the Danish International DSF Project (No. 09-71598) and Chinese International Collaboration Project (No. 2010DFA62410).
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1 Experimental

1.1 System description

Fig. 1 shows the schematic diagram of HTHP & DW system for summer or winter use. The mode conversion can be realized by opening and closing certain air dampers. Damper No.1 opens fully in summer and opens partly in winter to adjust the air flow. The refrigerant in the heat pump is reversed through 4-way valve when the mode is switched.

In summer mode, the ventilation system supplies minimum level of outdoor airflow (state W) and recirculates a large amount of indoor air (state N) on the process side. The return air and outdoor fresh air are mixed to become the process air (state C), which flows through the desiccant wheel where a large amount of moisture is removed by the desiccant (state E). In this process, the process air is dehumidified and cleaned as well as its temperature increases, which is the result of the dehumidification effect and the heat transfer from the regeneration airstream via DW. The process air is then cooled by the evaporator of heat pump that is controlled for conditioning the room air temperature to a comfortable level (state O). The state W is used to regenerate DW. The regeneration air is heated up by the condenser of the heat pump (state A) and is expelled after regenerating DW (state B). Fig. 2 gives an example of air processes on the psychrometric chart at the condition of $T_W = 28$ ℃, $\omega_W = 18$ g/kg, fresh air ratio (FAR) = 20%.

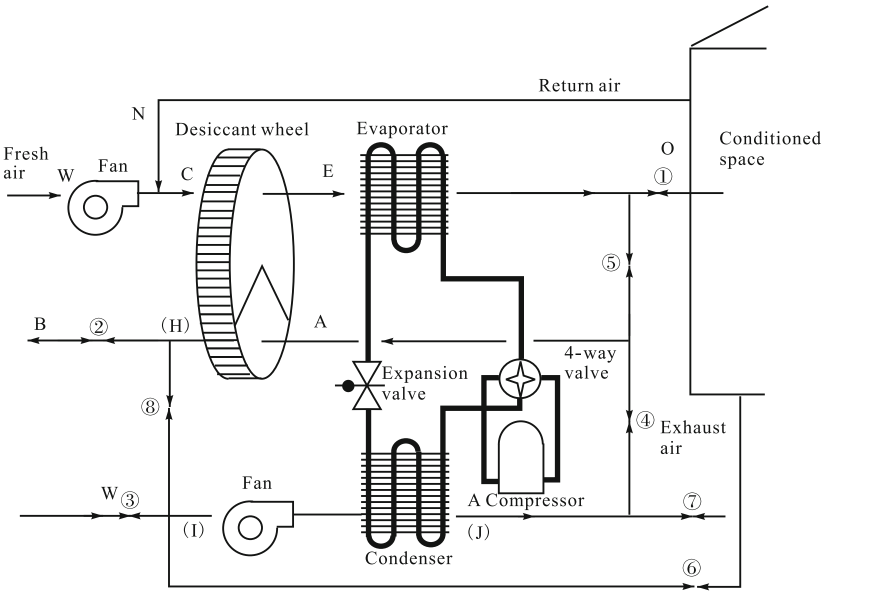

Fig. 1 Schematic of HTHP & DW system

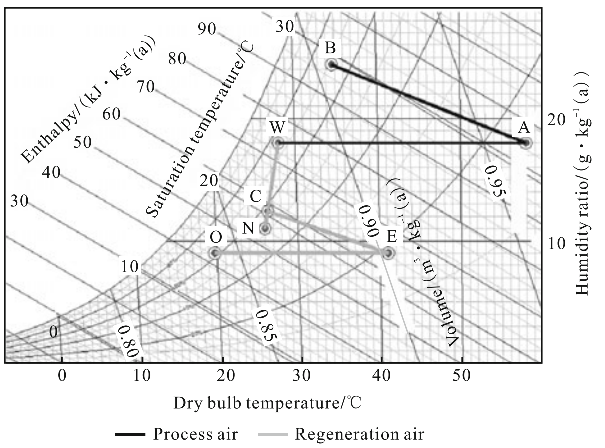

Fig. 2 Air processes on the psychrometric chart

In winter mode, the mixed process air (state C) also flows through the desiccant wheel where it is warmed (state E) and then heated by the condenser of heat pump and finally sent back into the room (state O). A small amount of the process air (state O) is used to regenerate DW. After regenerating the DW, the warm regeneration air (state H) is mixed with the exhaust indoor air (state N) and expelled through the evaporator of heat pump (state I to J).

The demands of dehumidification and cooling are large in summer, which makes the system face greater challenges. Thus, the study of the system performance in summer mode is more important and will be focused on in this paper.

1.2 Test facility and measuring devices

Fig. 3 shows the layout of test facility and the locations of measuring points. The test facility consists of desiccant wheel, heat pump, fans and pretreatment tanks. The desiccant wheel is filled with silica gel and it is characterized by the following layout: process air passes 75% of the flow area and regeneration air passes the remaining 25%. The diameter of the wheel is 45 cm, with a length of 20 cm. The closed piston compressor (Copeland, USA) is used with nominal input power of 2 kW. Electrical heaters and electrode humidifier are used for simulating different climates at the inlet of the desiccant wheel.

The parameters measured include dry bulb temperature (DBT) and wet bulb temperature (WBT) in the inlet and outlet air for the adsorption and regeneration processes, air flow rate (AFR), and the input power of the compressor. The specifications of the measuring devices are listed in Tab. 1.

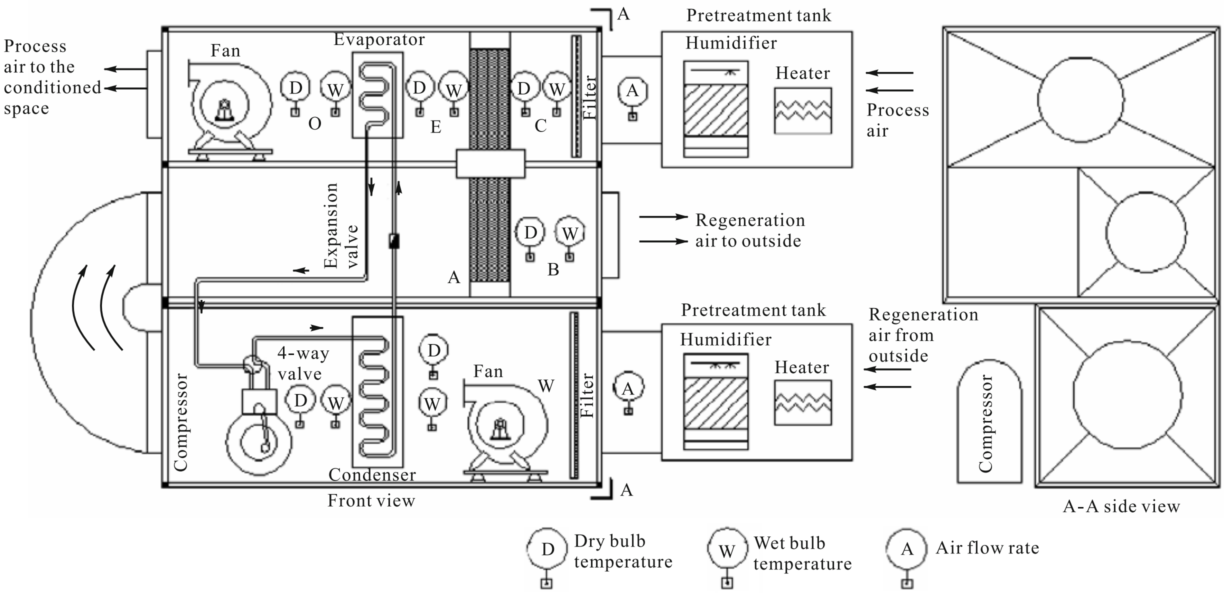

Fig. 3 Layout of test facility

Tab. 1 Specification of measuring devices

Measured parameter	Device	Type	Accuracy	Measuring range
DBT/WBT	Platinum resistance	WZP-Pt100	Class A, ±(0.15 + 0.002|t|) ℃	−200—500 ℃
AFR	Hotwire anemometer	KANOMAX KA22	±2% m/s	Low velocity: 0—4.99 m/s High velocity: 5—50 m/s
Power of compressor	Dynamometer	YN194P3-1K1	±1% kW	0—2.5 kW

1.3 Experimental procedure

Before the system operating as the procedure indicated in Fig. 2, it will be running in hot dry and hot humid climates to test the effect of single operating parameter on the wheel performance. In hot dry and hot humid climates, the absolute humidity ratio is held fixed and the inlet air temperature (state C) is changed. The process and regeneration AFRs are chosen to be 540 and 270 m^3/h, respectively. In the actual operation, the indoor air conditions are assumed as $T_N = 26$ ℃, $\omega_N = 11$ g/kg. Outdoor air temperature (T_W) varies from 30 ℃ to 38 ℃, with the incremental change of 4 ℃. At each T_W, outdoor air humidity ratio varies from 10 g/kg to 20 g/kg with the incremental change of 2 g/kg. The supply air properties for the room are temperature $T_O = 20$ ℃, humidity ratio $\omega_O = 7$—9 g/kg. The thermal energy for regeneration is recovered from the condenser of heat pump, so that the regeneration temperature ($T_r = T_A$) is derived within the range of 60 ℃ to 65 ℃ approximately, and the flow ratio of regeneration air to process air varies from 1.0 to 1.3. Tab. 2 lists the operating condition of each experiment and the variation of the parameters.

Tab. 2 Experimental conditions

No.	Experiment	T_N/℃	ω_N/(g·kg^{-1})	T_W/℃	ω_W/(g·kg^{-1})	T_C/℃	ω_C/(g·kg^{-1})	T_r=T_A/℃
1	Hot dry	—	—	—	—	28—32	8	60
2	Hot humid (1#)	—	—	—	—	28—34	16	63
3	Hot humid (2#)	—	—	—	—	28—34	16	67
4	Actual operation	26	11	30, 34, 38	10—20	26.8—28	10.7—12.5	60—65

2 Performance indexes

(1) Dehumidification effectiveness[11,12]

Dehumidification effectiveness denotes the change rate of air absolute moisture when the process air passes through the desiccant wheel.

$$\eta_{DW,1} = \frac{\omega_C - \omega_E}{\omega_C - \omega_{ideal}} \quad (1)$$

where ω_{ideal} is the ideal specific of the air stream at the outlet of desiccant wheel. If its value is taken as zero, the desiccant wheel will be ideal in which the air is completely dehumidified. According to the system presented in Fig. 1, ω_C and ω_E are the humidity ratios of inlet and outlet process air respectively, g/kg.

(2) Regeneration effectiveness[11]

Eq. (2) shows the desiccant wheel capability of reducing the air moisture content passing it based on the amount of thermal energy introduced for the regeneration of desiccant wheel.

$$\eta_{DW,2} = \frac{M_p(\omega_E - \omega_C)h_{fg}}{M_r(h_A - h_W)} = \frac{Q_{latent}}{Q_{regeneration}} \quad (2)$$

where h_A and h_W are the enthalpy values of the inlet regeneration air and fresh air respectively, kJ/kg; M_p and M_r are process and regeneration air flow rates, kg/h; and h_{fg} is the latent heat of vaporization of water, kJ/kg.

(3) Thermal effectiveness[13]

Desiccant wheel performance is illustrated in terms of calorific. In fact, desiccant wheel is assumed as a heat exchanger and the above relation is derived from the definition of heat exchanger effectiveness.

$$\eta_{DW,3} = \frac{T_E - T_C}{T_A - T_C} \quad (3)$$

where T_C, T_E, T_A are temperatures of inlet and outlet process air and inlet regeneration air, respectively, ℃.

(4) Adiabatic effectiveness[11]

$$\eta_{DW,4} = 1 - \frac{h_E - h_C}{h_C} \quad (4)$$

where h_C and h_E are the inlet and outlet enthalpy of process air respectively, kJ/kg. Based on this relation, the ideal state of desiccant wheel would be gained by completely adiabatic operation and it is just in the condition that the effectiveness value reaches 100%. This deviation in the enthalpy can be caused by isosteric heat (heat of adsorption), heat accumulation in the desiccant and matrix material, and heat leakage from regeneration to process[11].

(5) Coefficient of performance (COP)[14]

COP for the cooling of heat pump is represented as

$$\mathrm{COP} = \frac{Q_c}{E} \quad (5)$$

where Q_c is the cooling load of system, kW; and E is the electrical power used to drive the compressor, kW.

According to the uncertainty analysis theory[15], the propagation of uncertainties based on the root of error sum of squares is presented. The performance parameters described are obtained through calculation based on measured variables. The maximum values of relative uncertainty obtained for the calculated parameters are as follows: 7.07% for $\eta_{DW,1}$, 8.08% for $\eta_{DW,2}$, 3.33% for $\eta_{DW,3}$, 4.03% for $\eta_{DW,4}$, and 3.6% for COP.

3 Results and discussion

In order to demonstrate the effect of single operating parameter on the performance of DW, the performance indexes are compared and evaluated when the system is operating in hot dry and hot humid climates.

Fig. 4 and Fig. 5 show the variations of DW effectiveness versus air temperature (T_C) in hot dry and hot humid climates, respectively. It is observed that the dehumidification effectiveness and regeneration effectiveness pass through a declining trend. This is because the low process air temperature can be in favor of adsorption[16] that determines the decrease of dehumidification effectiveness. $\Delta\omega$ ($\Delta\omega = \omega_E - \omega_C$) of the process air decreases while the regeneration thermal power remains constant, thus, regeneration effectiveness declines as expected on the basis of Eq. (2).

It is also found that the adiabatic effectiveness decreases slightly with the rise of T_C, because the adsorption decreases with the increase of T_C, which accordingly causes the reduction in the isosteric heat and the enhancement of the adiabatic effectiveness. With respect to the energy balance of regeneration side, more heat exits in the regeneration process and causes the heating of the desiccant wheel and matrix when the regeneration effectiveness decreases, which determines the decline of adiabatic effectiveness. As a result, the experimental investigation shows that the influence of heat accumulation on the adiabatic effectiveness is greater than that of isosteric heat.

If the regeneration air flow remains constant, the regeneration temperature (T_r) will decide the heat accumulation. In order to assess the influence of T_r, other operating parameters must be fixed. The comparison of indexes in hot humid climate at different regeneration temperatures is shown in Fig. 6. These data were obtained by the average effectiveness. It is found that the dehumidification and regeneration effectiveness in-crease as the adiabatic effectiveness decreases with the increase of T_r. Higher T_r causes more heat accumulation and heat leakage when the regeneration air flow is constant. Therefore, the decrease of adiabatic effectiveness indicates the significant influence of T_r on the deviation of enthalpy. Meanwhile, it is also found that there is a balance between the dehumidification and adiabatic effectiveness.

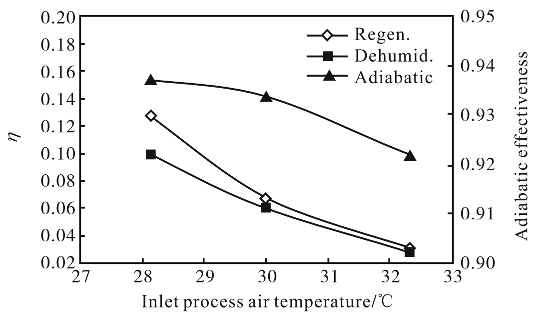

Fig. 4 Variation of desiccant wheel effectiveness in hot dry climate

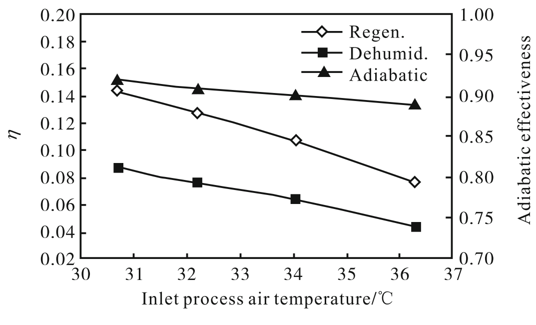

Fig. 5 Variation of desiccant wheel effectiveness in hot humid climate

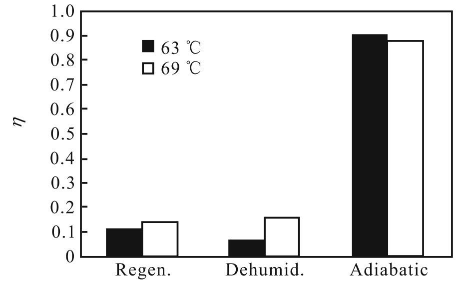

Fig. 6 Comparison of desiccant wheel effectiveness in hot humid climate at different regeneration temperatures

Based on the analysis above, the optimization analysis of HTHP&DW system is carried out when its running complies with the design schematic as indicated in Fig. 2. The cooling COP of heat pump is also evaluated to prove the effect of adiabatic effectiveness on the system performance.

In Fig. 7, the desiccant wheel effectiveness is reported as a function of the process and regeneration air thermal-hygrometric properties under the experimental conditions presented in Tab. 2 (No. 4). With the rise of inlet humidity ratio (ω_C), the dehumidification is enhanced so that dehumidification and regeneration effectiveness show slightly ascending trend.

It is observed that the thermal effectiveness increases when ω_C increases. The adsorption increases, accordingly causing the rise of adsorption heat and the increase of the outlet process air temperature (T_E). As expected on the basis of Eq. (3), the thermal effectiveness increases. The increase of the outlet enthalpy of process air also causes the decrease of adiabatic effectiveness.

In order to ensure the supply air temperature (T_O), the cooling load (Q_c) of heat pump increases with the rise of T_E. Thus the mass flow of refrigerant increases and eventually the input power of compressor (E) increases. Hence, COP decreases when the rise of E is more than that of Q_c.

Fig. 8 shows the comparisons between the desiccant wheel effectiveness and COP of heat pump (average values). The regeneration temperatures are 60.3 ℃, 63.1 ℃ and 64.6 ℃ respectively when the outdoor air temperatures are 30 ℃, 34 ℃ and 38 ℃ accordingly.

As represented in Fig. 8, COP decreases with the rise of T_r, because the cooling load of the system increases with the increase of enthalpy (i.e., adiabatic effectiveness decreases) and the rise of T_E (higher T_r determines higher dehumidification). In order to meet the supply air temperature (T_O = 20 ℃), more input power E is consumed and E rises more highly than cooling load. It is also found that the trend of COP is similar to that of adiabatic effectiveness.

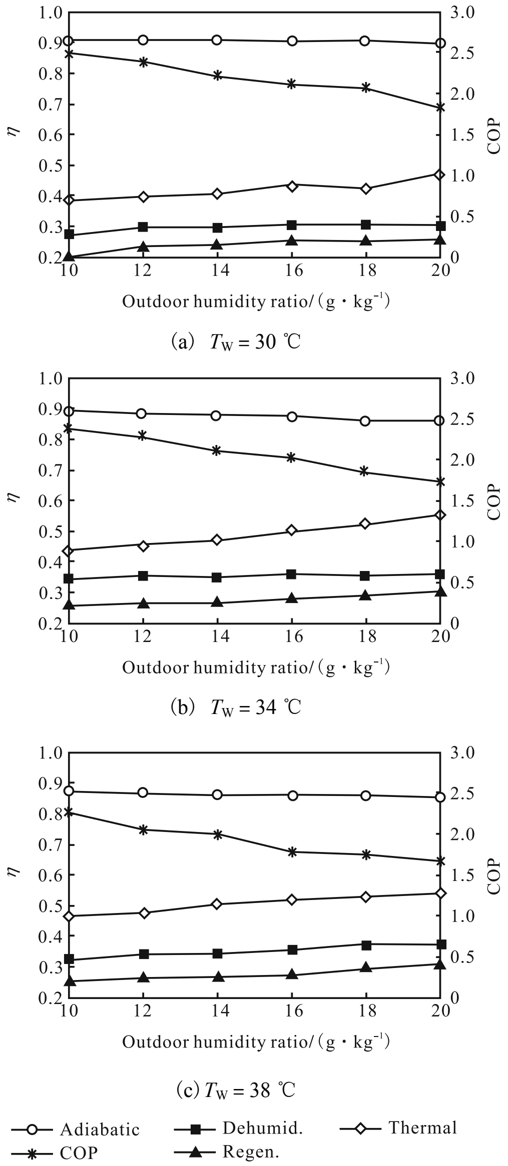

Fig. 7 Variations of desiccant wheel effectiveness and cooling COP in different outdoor thermal-hygrometric environments

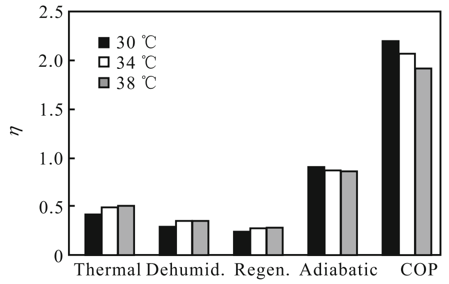

Fig. 8 Comparisons of desiccant wheel effectiveness and cooling COP in different outdoor thermal-hygrometric environment

When the regeneration temperature T_r is 60.3 ℃, the maximum value of adiabatic effectiveness is 91% and the minimum value of dehumidification effectiveness is 29.8%. When T_r is 63.1 ℃, the maximum value of dehumidification effectiveness is 35.4%, and the adiabatic effectiveness is 88%, which is a relatively high one. When T_r is up to 64.6 ℃, the dehumidification and adiabatic effectiveness are nearly the same as those at the temperature of 63.1 ℃, and the minimum value of adiabatic effectiveness is 86%. The dehumidification effectiveness can achieve the best with the acceptable adiabatic effectiveness when T_r is 63 ℃. Higher T_r determines better dehumidification, but it requires higher condensing temperature of heat pump that causes more E consumed and then leads to lower COP. Therefore, it is recommended that the regeneration temperature should be controlled at 63 ℃ when the HTHP&DW system is running.

4 Conclusions

In this study, the performance of desiccant wheel in HTHP&DW system in different climates and operating conditions was investigated by evaluating different effectiveness. According to the analysis above, several conclusions are drawn as follows.

The enthalpy increase occurs when the process air passes through the desiccant wheel. The principal effect factors are the heat accumulation in the desiccant and matrix material and the heat leakage from regeneration to process side instead of isosteric heat. When the regeneration air flow remains constant, the regeneration temperature has a significant influence on the enthalpy increase.

It is found that the optimal regeneration temperature is 63 ℃, at which the maximal dehumidification effectiveness is 35.4% and the satisfied adiabatic effectiveness is 88%.

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(Editor: Zhao Yang)