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Influence of the number of stages on the heat source temperature of desiccant wheel dehumidification systems using exergy analysis

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ABSTRACT

The influence of the number of stages on the heat source temperature (t_{hs}) of desiccant wheel dehumidification systems that adopt water or refrigerant as cooling/heating media is examined in this paper. Under the same working conditions, when the exergy provided by the hot water (E_h) or the power input of the compressor (P) are reduced, t_{hs} can also be reduced. To reduce E_h and P, the exergy destruction of the desiccant wheels and heat exchangers should be reduced. The influence of stage number on the exergy destruction of the desiccant wheels is not significant when A_r and F_r are equal to 1. However, stage number has a significant influence on the exergy destruction of the heat exchangers. For water-driven systems, as stage number increases, t_{hs} (inlet temperature of the hot water) does not always decrease. The optimal stage number varies according to the water's total mass flow rate. A $(m_c c_{pw})/(m_a c_{pa})$ value of around 4 and the number of stages being 2–4 are preferable, with t_{hs} being around 55 °C and t_{cs} being higher than 18 °C. When refrigerant is used, as the number of stage increases, t_{hs} (condensing temperature) decreases. In this scenario, a 4-stage system with *COP* around 4.5 is preferable.

1. Introduction

Effective dehumidification methods are very important for reducing the energy consumption of air-conditioning systems, especially in humid climates. Solid desiccant dehumidification methods such as rotary desiccant wheels are effective air dehumidification approaches [1]. After absorbing water from the processed air, the desiccant material has to be regenerated by another stream of air. The air is treated near the isenthalpic line in the desiccant wheel [2]. Thus, the regeneration temperature (t_{reg}; the temperature of the regeneration air entering the wheel) has to be around 110–130 °C [1–3] to meet the dehumidification demand, leading to a high heat source temperature (t_{hs}; inlet temperature of the heating medium used to heat the regeneration air). This hinders the use of high-efficiency and low-temperature heat sources, such as solar energy, waste heat from heat pump systems, etc. As a result, electric or gas burners [4] have to be used.

In many studies, researchers have tried to reduce t_{reg} and adopted high-efficiency renewable heat sources such as solar energy [5] and heat pump systems [6,7] in place of electrical heating. t_{reg} can be reduced through pre-cooling [2,8], adopting two-stage desiccant wheel systems [2,6–11], or using desiccant wheels with a facial area ratio and an air flow rate ratio of the two streams of air both equaling to 1 [2]. Multi-stage desiccant wheel systems have gained considerable attention in recent years because of the possibility of low-temperature regeneration. Tu et al. [2] explained the influence of the number of stages on t_{reg} using exergy analysis; studies showed that t_{reg} could be reduced to about 50–60 °C [2,6,7,9–11]. When a heat pump system was adopted, *COP* of the two-stage system increased to 7.29 and 4.26 under *ARI* and Beijing summer conditions, respectively, when the supplied air humidity ratio was 10 g/kg [6].

Although increasing the number of stages can effectively reduce t_{reg}, the influence on t_{hs} is much more complex and has yet to be researched. When water is used to heat the regeneration air, t_{hs} refers to the inlet temperature of the hot water; when refrigerant is used to heat the regeneration air, t_{hs} refers to the condensing temperature. Tu et al. [12] examined the influence of the number of stages on the condensing temperature of heat pump-driven multi-stage desiccant wheel systems. The results showed that as the number of stages increased, both condensing temperature and t_{reg} decreased; however, the rate of this decrease slowed down as the

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increase of the number of stages. The specific heat capacity of water is finite, while that of refrigerant is infinite. Does the number of stages have the same influence on the t_{hs} of water-based cooling/heating systems and that of refrigerant-based cooling/heating systems? What are the main factors that influence t_{hs}? Does there exist an optimal number of stages with the lowest t_{hs}?

In this paper, multi-stage desiccant wheel dehumidification and cooling systems are designed, and the influence of the number of stages on t_{hs} is examined from the perspective of exergy. First, the thermodynamic method is used to determine the key factors that influence t_{hs}. Next, a simulation method is used to validate the results. Finally, the optimal number of stages for both water- and refrigerant-based heating/cooling systems is recommended.

2. System description and exergy analysis

2.1. Description of water- and refrigerant-driven systems

Water and refrigerant represent two types of heating/cooling media and have finite and infinite specific heat capacities, respectively; Figs. 1 and 2 illustrate the corresponding multi-stage desiccant wheel dehumidification and cooling systems, taking two-stage systems as examples.

In Fig. 1, water is used as the heating and cooling medium. For the two-stage system, there are two desiccant wheels (DW1 and DW2), two counter-flow air–water coolers, and two counter-flow air–water heaters. The first stage is composed of DW1, Cooler 1, and Heater 1, and the second stage is composed of DW2, Cooler 2, and Heater 2. The processed air flows in sequence through the first stage and the second stage. After being dehumidified by DW1 (A_{pin} to A_{p1}) and DW2 (A_{p2} to A_{p3}), the air is cooled down by the chilled water in Cooler 1 (A_{p1} to A_{p2}) and Cooler 2 (A_{p3} to A_{pout}), before finally being introduced into the occupied room (A_{pout}). The regeneration air flows through the second stage and the first stage in sequence. After being heated by the hot water to the required t_{reg} in Heater 2 (A_{rin} to A_{r1}) and Heater 1 (A_{r2} to A_{r3}), the regeneration air is used to regenerate DW2 (A_{r1} to A_{r2}) and DW1 (A_{r3} to A_{rout}), before finally being exhausted (A_{rout}). The chilled water (W_{cin}) and the hot water (W_{hin}) flow in parallel into all the coolers and heaters, respectively. Chilled water from all the coolers (W_{c1} and W_{c2}) mixes together and proceeds to the chilled water outlet (W_{cout}); similarly, hot water from all the heaters (W_{h1} and W_{h2}) mixes together and proceeds to the heated water outlet (W_{hout}).

In Fig. 2, refrigerant is used as the cooling and heating medium. The heat pump system, with one compressor and one expansion valve, is integrated with the desiccant wheels. For the two-stage system, there are two evaporators used as coolers and three condensers used as heaters. The first stage is composed of DW1, Evaporator 1, and Condenser 1, and the second stage is composed of DW2, Evaporator 2, and Condenser 2. The air handling processes are the same as those of Fig. 1, except that the regeneration air leaving DW1 is heated by Condenser 0 (A_{r4} to A_{rout}) to dissipate the extra heat of the heat pump system. All evaporators and condensers are linked in parallel. Thus, all the evaporators are of the same evaporating temperature, and all the condensers are of the same condensing temperature.

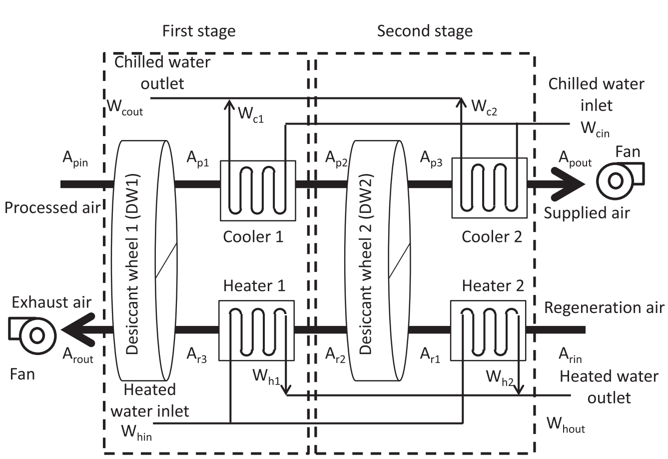

Fig. 1. Schematic of a desiccant dehumidification and cooling system using water as the cooling and heating medium.

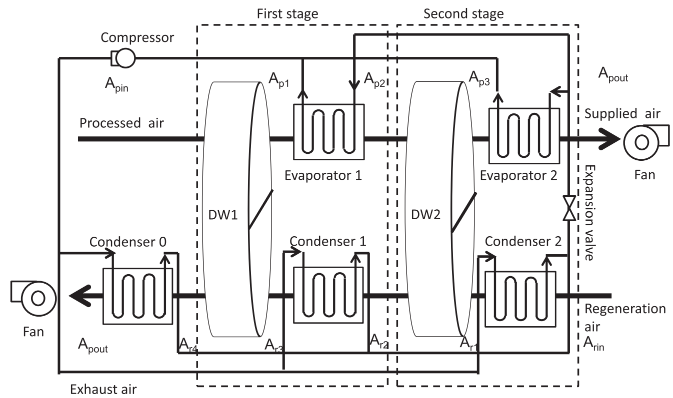

Fig. 2. Schematic of a heat pump-driven desiccant dehumidification and cooling system using refrigerant as the cooling and heating medium.

2.2. Performance indicators of the systems

For the system shown in Fig. 1, the thermal energy from water is used to heat or cool the air. The reduction of t_{hs} (temperature of the hot water entering the heaters) is beneficial for the adoption of solar energy and low-temperature waste heat, etc. For the system shown in Fig. 2, the reduction of t_{hs} (condensing temperature) is beneficial for the reduction of the power of the compressor (P).

COP can be expressed as COP_W for the system in Fig. 1 and as COP_R for the system in Fig. 2, shown as Eqs. (1) and (2), respectively:

$$COP_W = \frac{m_p(i_{pin} - i_{pout})}{Q_c + Q_h} = \frac{m_p(i_{pin} - i_{pout})}{m_c c_{pw}(t_{cout} - t_{cin}) + m_h c_{pw}(t_{hin} - t_{hout})} \quad (1)$$

$$COP_R = \frac{m_p(i_{pin} - i_{pout})}{P} = \frac{m_p(i_{pin} - i_{pout})}{Q_c} \frac{T_{evap}}{T_{cond} - T_{evap}} \varepsilon \quad (2)$$

where Q_c is the cooling capacity provided by the chilled water or refrigerant in Figs. 1 or 2, respectively; Q_h is the heat capacity provided by the hot water in Fig. 1; T_{evap} and T_{cond} are evaporating and condensing temperature in Kelvin, respectively; and ε is the thermodynamic perfectness of the compressor, which is lower than 1.

Exergy efficiency (η_e) can be expressed as $\eta_{e,W}$ for the system in Fig. 1 and as $\eta_{e,R}$ for the system in Fig. 2, shown as Eqs. (3) and (4), respectively:

$$\eta_{e,W} = \frac{m_p(e_{pout} - e_{pin})}{E_c + E_h} = \frac{m_p(e_{pout} - e_{pin})}{m_c(e_{cin} - e_{cout}) + m_h(e_{hin} - e_{hout})} \quad (3)$$