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Performance investigation and exergy analysis of two-stage desiccant wheel systems

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ABSTRACT

Two-stage desiccant wheel systems are an effective way to improve the dehumidification performance. In the present study, the performances of a one-stage system and a two-stage system with identical heat transfer areas are compared, with particular emphasis on the required heating source temperature. The exergy and unmatched coefficient (ξ) are applied to analyze the destruction of the heat and mass transfer processes. Compared to the one-stage system, the regeneration temperature (T_r) of the two-stage system is lower. The required hot water temperature (T_h) depends on the supplied water flow rate and A_p/A_r of the desiccant wheel. When $A_p/A_r = 1$, T_h of the two-stage system is lower only when the supplied water flow rate is relatively high. And due to different heat transfer area distribution demand, the exergy destruction of two-stage system is higher than one-stage system. When $A_p/A_r = 3$, the two-stage system has greater advantages. ξ of the desiccant wheel decreases from 2.9 to 2.2 when the number of stages increases from 1 to 2, leading to lower exergy destruction of the desiccant wheels and higher exergy efficiency.

1. Introduction

Air-conditioning systems that utilize rotary desiccant wheels have drawn more and more attention as an effective dehumidification method due to their compact construction, resistance to corrosion, and ability to work continuously [1]. In contrast to conventional condensing dehumidification methods, desiccant wheel approaches show significant advantages in utilizing different energy sources. However, the required regeneration temperature T_r (i.e., the air temperature entering into the regeneration section) is relatively high, restricting a wider application in desiccant wheels. As a result, many studies have focused on reducing the T_r of desiccant wheels to improve system performance and make it possible to use low-grade or renewable energy sources, e.g., solar energy, heat supplied from a cogeneration plant, waste heat, and bioenergy [2–11].

Two-stage desiccant wheel systems are one kind of effective ways to reduce T_r [12–16]. For the desiccant wheel handling process, dehumidification and regeneration are both approximately along the isenthalpic line. To remove a certain amount of moisture from the regeneration side, interstage cooling can bring the processed air and desiccant material to a relatively low temperature level, thereby reducing the required T_r. While for a single stage process, there is no interstage cooling. Temperature of inlet regeneration air should be high enough to provide a driving force for removing moisture. Thus T_r required for the single stage process will be much higher than that for a two-stage process [12]. The two-stage desiccant wheel system can be realized either by two wheels or one four-partition wheel type, as summarized in Table 1 [12–16]. Tu et al. [12] investigated the performance of a heat pump-driven two-stage desiccant wheel system; T_r was 44 °C, which was significantly lower than that of the one-stage system (80 °C). Jeong et al. [13] proposed a system combining a heat pump with two desiccant wheels or one four-portion desiccant wheel; the required T_r was 43.2 °C, which was considerably lower than that of a conventional single-stage desiccant wheel system (62.2 °C). La et al. [14] Li et al. [15] and Ge et al. [16] examined a series of two-stage desiccant wheel systems in which hot water, solar energy, and electric heaters, respectively, were applied to supply heat to the regeneration air, and T_r ranged from 50 to 90 °C. In all these studies, the T_r of the two-stage systems varied from 40 to 90 °C, while that

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Table 1
Analysis of two-stage desiccant wheel systems from the literature.

	Heat source	A_p/A_r	m_r/m_a	m_w/m_a	Dehumidification capacity	Regeneration temperature	Processed air inlet	Regeneration air inlet
Tu et al. [12]	Heat pump	1	1	–	8.5 g/kg	44 °C	33 °C, 19 g/kg	12.6 g/kg
Jeong et al. [13]	Hot water	1	1	–	4 g/kg	43.2 °C	28 °C, 11.9 g/kg	19.5 g/kg
La et al. [14]	Hot water	3	2.6	0.4–1.7	6–9 g/kg	50–90 °C	35 °C, 14.1 g/kg	14.1 g/kg
Li et al. [15]	Solar energy	3	2.5	–	9–12 g/kg	60–80 °C	26 °C, 16 g/kg	18 g/kg
Ge et al. [16]	Electricity	3	2.4	–	3–15 g/kg	60–90 °C	24.8–37.1 °C, 10.0–30.3 g/kg	3.6–16.3 g/kg

of the one-stage systems was far higher (60–140 °C). Thermodynamic analysis is also carried out for performance optimization of desiccant wheel systems [17–22], including proposing approaches to reduce T_r. It has found that the required T_r is influenced by the exergy destruction of all the components in the desiccant wheel dehumidification and cooling system [17–19].

In the above examinations, the heat sources used to provide regeneration heat included hot water from a solar collector, refrigerant from a heat pump system, and electricity, which can be classified as either constant temperature sources or varying temperature sources. These studies mainly focused on ways to reduce the required regeneration temperature. However, few studies have analyzed the influence of the number of stages on the required temperature of heating and cooling sources.

This paper aims to analyze the changes in heating source temperature (T_h) according to the number of stages when water with different flow rates is used as a heating and cooling source. The relationship between the exergy destruction of the system and the required T_h is investigated. In exergy analysis, the exergy destruction can be divided into two parts: one caused by the limited heat and moisture transfer area and the other caused by the unmatched coefficient. The unmatched coefficient [23], which has a significant impact on exergy destruction, is used to analyze the influence of the number of stages on exergy destruction. This approach should allow us to determine how the required heat source temperature can be reduced.

2. Operating principle of the desiccant wheel system

2.1. Operating principle

The operating principle of the one-stage desiccant wheel system is depicted in Fig. 1(a); it is composed of one desiccant wheel, one heat recovery unit (HR), one heater, and one cooler. For the two-stage system shown in Fig. 1(b), there are two desiccant wheels, two heat recovery units, two heaters, and two coolers. It can be seen that the desiccant wheel and the sensible heat exchangers are two key components of desiccant wheel air-conditioning systems. Heat and mass transfer between the air and the solid desiccant occurs in the desiccant wheel. High-temperature regeneration air makes the solid desiccant release moisture, and the processed air can be dehumidified by the dried solid desiccant. The dehumidification and regeneration processes are almost isenthalpic. The processed air with high temperature out from the desiccant wheel is used to preheat the regeneration air. Heating and cooling sources (hot and chilled water, respectively) are required to heat the regeneration air further before regeneration and to cool down the processed air after heat recovery unit, respectively.

The mathematical model of a silica gel desiccant wheel, which has been introduced in detail in previous studies [12], is used in the present study to simulate the performance of the desiccant wheel. Both ordinary diffusion and surface diffusion are taken into consideration in the model, and it's validated by experimental results of a dehumidification wheel (as shown in Fig. 2). The heat transfer capacity of the heat exchangers is calculated by the ε–NTU model. Two desiccant wheel designs are examined in the present study: one in which $A_p/A_r = 1$ and one in which $A_p/A_r = 3$, as depicted in Fig. 3.

The thickness of the desiccant wheel, the heat transfer area of all the heat exchangers, and the water flow rate of the one-stage system are evenly divided in the two-stage system. The operating conditions and component information are shown in Table 2. For both one-stage and two-stage systems, the inlet parameters of the processed air are 33 °C and 19 g/kg, the required supplied air humidity ratio is 9 g/kg, and the chilled water temperature is 20 °C; these conditions are set as invariant. For all systems and cases, the desiccant wheels are operated at the optimum rotation speeds. In this way, the required heat source temperatures can be evaluated and compared.

2.2. Simulation results

Fig. 4(a) shows the air handling process in psychrometric charts when $A_p/A_r = 1$ for the desiccant wheel (Fig. 3(a)), and hot and chilled water with a flow rate of 0.16 kg/s for each heat exchanger is applied in the one-stage system (for heater and coolers, $m_w/m_a = 0.16$). To reach the 9 g/kg outlet humidity ratio, T_r is 72.3 °C and the required T_h is 81.4 °C for the one-stage system. For the two-stage system, the desiccant wheel and heat exchangers are equally split, with a breakeven water flow rate of 0.08 kg/s for each heat exchanger, which maintains the same total water flow rate as the one-stage system. The air handling processes and desiccant material states of the two-stage system are shown in Fig. 4(b). T_r values of the two-stage desiccant wheels are 65.7 °C and 57.7 °C, which are both lower than those of the one-stage system. However, T_h increases to 89.5 °C, which is about 3 °C higher than that of the one-stage system.

When the water flow rate for every heat exchanger in the one-stage system is 0.72 kg/s (for heaters and coolers, $m_w/m_a = 0.72$), the required T_r is 72.4 °C and T_h is 73.2 °C. For the two-stage system, the water flow rate for every heat exchanger is 0.36 kg/s; the required T_r values of the two stages are 56.3 °C and 54.8 °C; and the required T_h is 60.0 °C. It can be seen that both T_r and T_h decrease as the number of stages increases.

These two sets of results illustrate the important influence of the water flow rate. When the applied water flow rate is low (0.16 kg/s), the T_h of the two-stage system (89.5 °C) is higher than that of the one-stage system (81.4 °C). However, when the applied water flow rate is high (0.72 kg/s), the T_h of the two-stage system (60.0 °C) is lower than that of the one-stage system (73.2 °C). This indicates that the reduction of T_r does not necessarily lead to the reduction of T_h because the results are also influenced by the water flow rate.

When the water flow rate is low, the heat exchange amount of HR, heater and cooler in one-stage system are 28 kW, 17 kW and 8 kW respectively, and 21 kW, 28 kW, 14 kW in two-stage system. It shows similar results when more water flow rate is supplied. Thus, in two-stage system, less heat recovery amount in HR results in more heat exchange demand in coolers and heaters.

This paper will analyze the influencing factors of T_h and exergy efficiency comparison when the two types of desiccant wheels shown in Fig. 2 are used.