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Performance analysis of a two-stage desiccant cooling system

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HIGHLIGHTS

- Performance of a heat pump driven two-stage desiccant wheel system is analyzed.
- Effects of heat pump system, desiccant wheel and working conditions are evaluated.
- System COP_t is 5.5 under Beijing summer condition with supplied air humidity of 10 g/kg.
- Cold–heat matching between heat pump and desiccant is the key to improve performance.
- New system using an indirect cooler has better performance with COP_t of 15% higher.

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ABSTRACT

Multi-stage desiccant systems are an effective way to improve the performance of desiccant dehumidification systems, which can greatly decrease the required regeneration temperature and make possible the utilization of exhaust heat from the heat pump. The performance of a heat pump-driven two-stage desiccant wheel system is analyzed in this paper. Models of the desiccant wheel and heat pump systems are utilized to predict system performance. The effects on system performance of the compressor power input, the heat exchange area distribution between evaporators and condensers, the wheel's rotation speed, and the inlet parameters of the processed air are investigated. When the supplied air humidity ratio is 10 g/kg, COP_t of the desiccant system is 5.5 under Beijing summer condition. The key to improving system performance is to match the cooling capacity and exhaust heat provided by the heat pump with the requirements of dehumidification and regeneration. An improved system utilizing an indirect cooler to recover the cooling capacity from the indoor exhaust air is then proposed, with COP_t improving by 15% compared to the original system.

1. Introduction

Effective dehumidification methods are very important for reducing the energy consumption of air-conditioning systems, especially in humid climates. Condensation dehumidification is one common method. However, if the cooling source temperature is lower than the dew point of the processed air, reheat is sometimes needed to adjust the temperature of the supplied air, leading to a large cold–heat offset loss. Solid desiccant dehumidification methods such as rotary desiccant wheels are effective approaches to air dehumidification that can avoid the cold–heat offset loss and be regenerated with low-grade heat sources [1]. During desiccant dehumidification, air is treated near the isenthalpic line, and the desiccant material has to be very dry to satisfy the dehumidification requirements. Thus, a high regeneration temperature (usually higher than 80 °C) is required. Many researchers have tried to reduce the regeneration temperature [2–4], by implementing high-efficiency cooling sources [3,5–7], and by utilizing high-efficiency and renewable heating sources [7–12].

Two-stage desiccant wheel systems have gained considerable attention from researchers in recent years because of the possibility of low-temperature regeneration. In these types of systems, two desiccant wheels are utilized to dehumidify the processed air stage-by-stage, and either cooling sources or heating sources are placed between the two desiccant wheels. In the dehumidification process, the processed air is dehumidified (isenthalpic line) by the first wheel, cooled down by cooling sources, and then dehumidified (isenthalpic line) by the second wheel. In this way, the processed air can maintain a low temperature, and the desiccant will remain in a higher water content range than it will in single-stage desiccant wheels. In addition, the corresponding regeneration temperature can be reduced. A two-stage desiccant wheel system can be realized with either one wheel [3,5,10,13] or two wheels [4,6,8,14]. In one-wheel systems, the wheel is split into four parts: two parts at opposite angles for dehumidification and two for regeneration. In two-wheel systems, each wheel is split into two parts like a traditional wheel.

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Nomenclature

A	area, m^2
a	pore radius, m
C	shape factor
COP	coefficient of performance
c_p	specific heat capacity, J/kg
D_A	gas diffusion coefficient, m^2/s
D_0	surface diffusion constant, m^2/s
D_S	surface diffusion coefficient, m^2/s
d_h	hydraulic diameter, m
E_c	input power of compressor, W
f	area ratio, dimensionless
G	air volume flow rate, m^3/h
h	heat transfer coefficient, $W/(m^2\,°C)$
h_m	mass transfer coefficient, $kg/(m^2\,s)$
i	enthalpy, J/kg
k_d	heat conductivity, $W/(m^2\,°C)$
M	molar mass, kg/mol
$\dot{M}$	indoor moisture generation rate, g/h
$\dot{m}$	mass flow rate, kg/s
NTU	number of heat transfer units
Nu	Nusselt number
P	wetted perimeter, m
P_a	atmosphere pressure, Pa
Q	Heat/cold capacity, W
r_s	adsorption heat, J/kg
T	temperature, °C
τ	time
u	velocity, m/s
W	water content, kg water/kg dry desiccant
x	volume ratio of desiccant material in the desiccant plate, dimensionless
Y	humidity ratio, g/kg
z	wheel thickness direction, m

Greek symbols

ρ	density, kg/m^3
ϕ	relative humidity, %
σ	porosity, dimensionless
ξ	tortuosity factor, dimensionless
η	compressor thermodynamic perfectness
ε	heat exchange efficiency

Subscripts

a	air
ad	adsorption material
$carnot$	carnot cycle
$cond$	condenser
d	solid
$evap$	evaporator
HP	heat pump system
m	matrix
max	maximum
in	inlet
out	outlet
P	processed air
R	regeneration air
r	recovery
t	total heat
w	water

Ge [14] concluded that under ARI summer conditions at the same moisture removal capacity (4.85 g/kg), the regeneration temperature of a one-stage desiccant cooling system is 90 °C, and that of a two-stage desiccant cooling system is 60 °C. Zhang [3] introduced a two-stage desiccant cooling system with evaporative cooling and a heat recovery wheel; the regeneration temperature was reduced to 60 °C when the moisture removal capacity was 10 g/kg. These studies showed that the regeneration temperature could be reduced to about 60 °C in two-stage desiccant cooling systems, which makes possible the use of a heat pump system.

Jeong [8] proposed a system that combines a heat pump system with two desiccant wheels or one four-part desiccant wheel. The simulated *COP* of this hybrid system was significantly improved compared to that of a conventional vapor compression-type refrigerator. Both evaporators and condensers are of the same heat transfer area in Jeong's research [8]. However, the heat that dissipated from the condensers was greater than the cooling capacity provided by the evaporators, and the latent heat required for dehumidification and regeneration were the same. Therefore, the key issue for increasing a hybrid system's performance is to match the cooling/heating capacity requirements of the desiccant wheel and that provided by the heat pump. In Jeong's [8] research, the hybrid desiccant system is utilized to dehumidify the indoor return air with inlet humidity ratio lower than 12 g/kg. What's the system performance when outdoor humid air is to be handled? The summer outdoor conditions in most cities in China, such as Beijing, Shanghai, and Guangzhou, are very hot and humid (around 33–35 °C and 19–22 g/kg, respectively), so examining system performance when humid outdoor air must be handled is extremely important.

In this article, the performance of a heat pump-driven two-stage desiccant wheel system used to handle humid outdoor air is evaluated. Performance-influencing factors, such as the compressor power input and the distribution of the heat exchange area between evaporators and condensers, are examined. An improved desiccant air handling process is then provided, which has a greater ability to match the cooling/heating capacity provided by the heat pump with the requirements of the desiccant wheel.

2. Operating principle of the two-stage desiccant handling system

The schematic of the heat pump-driven two-stage desiccant wheel system is shown in Fig. 1, in which evaporators are used as coolers and condensers are used as heaters. There are three evaporators located at the processed air duct before and after each wheel, and three condensers are positioned at the regeneration air duct before and after each wheel. As indicated by the figure, the processed air is cooled down by evaporator 1 (A_{Pin}–A_{P1}) before flowing into desiccant wheel 1 (A_{P1}–A_{P2}). After being dehumidified by desiccant wheel 1, it is cooled by evaporator 2 (A_{P2}–A_{P3}), and then dehumidified by desiccant wheel 2 (A_{P3}–A_{P4}). Finally, it is cooled by evaporator 3 (A_{P4}–A_{Pout}) before being introduced into occupied spaces. The regeneration air is heated by condenser 3 (A_{Rin}–A_{R1}), humidified by desiccant wheel 2 (A_{R1}–A_{R2}), heated by condenser 2 (A_{R2}–A_{R3}), humidified by desiccant wheel 1 (A_{R3}–A_{R4}), and finally heated by condenser 1 (A_{R4}–A_{Rout}) before being exhausted to the outdoor environment. The air handling processes of the heat pump-driven two-stage desiccant dehumidification system are shown in psychrometric chart in Fig. 2. During this stage-by-stage dehumidification process, the processed air is