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Performance analysis of ventilation systems with desiccant wheel cooling based on exergy destruction

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ABSTRACT

This paper investigates the performances of ventilation systems with desiccant wheel cooling from the perspective of exergy destructions. Based on the inherent influencing factors for exergy destructions of heat and mass transfer and heat sources, provide guidelines for efficient system design. First, performances of a basic ventilation system are simulated, which is operated at high regeneration temperature and low coefficient of performance (*COP*). Then, exergy analysis of the basic ventilation system shows that exergy destructions mainly exist in the heat and mass transfer components and the heat source. The inherent influencing factors for the heat and mass transfer exergy destruction are heat and mass transfer capacities, which are related to over dehumidification of the desiccant wheel, and unmatched coefficients, which represent the uniformity of the temperature or humidity ratio differences fields for heat and mass transfer components. Based on these findings, two improved ventilation systems are suggested. For the first system, over dehumidification is avoided and unmatched coefficients for each component are reduced. With lower heat and mass transfer exergy destructions and lower regeneration temperature, *COP* and exergy efficiency of the first system are increased compared with the basic ventilation system. For the second system, a heat pump, which recovers heat from the process air to heat the regeneration air, is adopted to replace the electrical heater and cooling devices. The exergy destruction of the heat pump is considerably reduced as compared with heat source exergy destruction of the basic ventilation system and the first system, leading to a great enhancement of *COP* and exergy efficiency.

1. Introduction

Efficient dehumidification methods are very important for reducing the energy consumption of air-conditioning systems, especially in humid climates. Desiccant wheel cooling is an effective air dehumidification approach, which has the potential use of low-grade heating sources. When the required regeneration temperature (t_{reg}: temperature of the regeneration air entering the desiccant wheel) is reduced, low-grade and high efficient heat sources, such as renewable energy [1–3] and waste heat from heat pumps [4,5] can be used. Compared with the condensation dehumidification, reheat is avoided for desiccant wheel systems [6]. The process air after the desiccant wheel can be cooled directly to the supply air temperature using free cooling [7–9], high evaporating temperature vapor compression cycle [2,4–6] or absorption chiller [10].

A typical desiccant wheel cooling system is comprised of desiccant wheels, cooling devices and heating devices. t_{reg} has large impacts on the selection of heating sources. Heating devices can be electrical heater [11,12], gas burner [13], solar energy [1,2,10] or heat pump systems [4,5,14]. High efficient heating sources, such as solar energy and heat pump systems can be used when t_{reg} is reduced. t_{reg} is influenced simultaneously by the desiccant wheel and the cooling device. The influences of desiccant wheels on t_{reg} are determined by a variety of factors, such as the wheel's dimensions [15], the structure of air channels [16], rotation speed [15,17,18], inlet states of the air [4,19], mass flow rate of the regeneration air [20], area ratio of regeneration section [20,21], purge section [22] and the adsorption material [23]. Tu et al. [20] found that the lowest t_{reg} can be achieved for equally divided desiccant wheels with mass flow rates of the two streams of air being the same when the wheel's dimensions and working conditions are fixed. Direct evaporative coolers and indirect evaporative coolers are two free cooling devices. Direct evaporative coolers have been widely adopted in the study of air handling cycles [23,24]. The

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Nomenclature

A	air
A_r	facial area ratio of the process air to the regeneration air
a	pore radius, m
BVS	basic ventilation system
COP	coefficient of performance
C	shape factor of desiccant material
c_p	specific heat capacity, kJ kg^{-1} K^{-1}
d_h	hydraulic diameter, m
D_A	ordinary diffusion coefficient, m^2 s^{-1}
D_S	surface diffusion coefficient, m^2 s^{-1}
D_0	surface diffusion constant, m^2 s^{-1}
e	specific exergy, kJ kg^{-1}
E	exergy rate, kW
ΔE	exergy destruction rate, kW
EA	exhaust air
f	area ratio, dimensionless
F	area, m^2
F_r	air flow rate ratio of the process air to the regeneration air
h	heat transfer coefficient, kW m^{-2} K^{-1}
h_m	mass transfer coefficient, kg m^{-2} s^{-1}
i	specific enthalpy, kJ kg^{-1}
IVS	improved ventilation system
k	coefficient of thermal conductivity, kW m^{-1} K^{-1}
Le	Lewis number
$\dot{m}$	mass flow rate, kg s^{-1}
M	mass transfer capacity, kg s^{-1}
Mol	molar mass, kg mol^{-1}
Nu	Nusselt number
NTU	number of heat transfer units
ODR	over dehumidification ratio, dimensionless
Pa	standard atmospheric pressure, Pa
P	power consumption, kW
PA	process air
Q	heat transfer capacity, kW
r	vaporization heat, kJ kg^{-1}
r_s	adsorption and desorption heat, kJ kg^{-1}
R_a	gas constant for air, kJ mol^{-1} K^{-1}
RA	return air
SA	supply air
t	Celsius temperature, °C
T	Kelvin temperature, K
u	velocity, m^2 s^{-1}
W	water content, dimensionless
x	volume ratio of the desiccant material in the solid, dimensionless
x^*	mass ratio of the desiccant material in the solid, dimensionless
z	wheel's thickness direction, m

Greek symbols

ω	humidity ratio, g kg^{-1}
η_e	exergy efficiency
ς	unmatched coefficient
ρ	density, kg m^{-3}
φ	relative humidity ratio of the humid air
σ	porosity, dimensionless
ξ	tortuosity factor, dimensionless
τ	time, s
ε	thermodynamic efficiency of the heat pump system
θ	heat or mass transfer efficiency

Subscripts

a	air
ad	adsorbent
C	condenser
E	evaporator
s	solid
w	water
p	process air
r	regeneration air
in	inlet
out	outlet
t	temperature
ω	humidity ratio
dew	dew point
reg	regeneration
HE	heat exchanger
DW	desiccant wheel
HR	heat recovery unit
HP	heat pump
ST	spray tower
H	heater
htr	heat transfer
mtr	mass transfer
sub	substrate of desiccant wheel
obt	obtain
max	maximum
$ambi$	ambient air
$space$	air conditioning space
0	dead state

main penalty for cooling effects of direct evaporative coolers is the increase of humidity ratio of the air. Experiment results showed that humidity ratio added to the process air could be higher than that removed by the desiccant wheel [23]. Performances can be improved using indirect evaporative coolers [25–28]. Jain and Dhar [7] found that coefficient of performance (COP) of systems with indirect evaporative coolers was significantly increased to over 1 from below 0.2 of the systems with direct evaporative coolers under constant t_{reg} (130 °C).

Exergy analysis has be conducted for desiccant wheel cooling systems [8,9,11–14,29–32]. La et al. [8,29] investigated the performances of a ventilation system from the perspective of exergy destructions. It was found that exergy destructions of the desiccant wheel and the heat source accounted for around 50% the total exergy destruction. The influences of the effectiveness of each component on the exergy efficiency of the ventilation system were investigated afterwards. Similar study has been carried out based on a novel desiccant wheel cooling system by Uçkan et al. [11]. The results show that exergy destructions of the electrical heater and the desiccant wheel were the top two highest, to which 65% and 18% of the total exergy destruction were attributed, respectively. The work of Kanoglu et al. [32] shows that the desiccant wheel and the heating source took 33.8% and 31.2% of the total exergy destruction, respectively. Similar results can be found in the experiment study of Hürdogan et al. [12], where the electrical heater took almost 70% of the total exergy destruction. Sheng et al. [14] found that when the electrical heater was replaced by the heat pump, exergy efficiency was increased by nearly 50%.

However, in these studies, the exergy destruction of each part was calculated using the inlet and outlet states of the air, which was not based on heat and mass transfer processes. Therefore, inherent influencing factors for exergy destructions were not