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Performance analysis of desiccant dehumidification systems driven by low-grade heat source

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ABSTRACT

If a desiccant dehumidification system can be driven by a heat source whose temperature is below 50 °C, exhaust heat from devices such as fuel cells or air conditioners can be used as its heat source, thereby saving energy. Therefore, in this study, we used a previously validated simulation model to determine the minimum heat source temperature for driving a desiccant dehumidification system. We considered four desiccant dehumidification systems that can be driven by waste heat—conventional desiccant-type systems (wheel type and batch type with only desiccant), a system with a precooler, double-stage-type systems (a type with two desiccant wheels and a four-partition desiccant wheel type), and a batch-type system with an internal heat exchanger. We found that among these systems, the last system can be driven by the lowest heated air temperature—approximately 33 °C—which is considerably lower than that of the conventional system.

Analyse de la performance des systèmes à déshumidification à déshydratant avec une source de chaleur à basse température

Motsclés : Roue déshydratante ; Optimisation ; Déshumidification ; Déshydratant ; Conditionnement d'air ; Gel de silice

1. Introduction

In order to achieve energy savings with the use of a compression-type room air conditioner, its evaporation temperature must be increased by increasing the setting temperature. However, due to this increase in the evaporation temperature, not only the temperature but also the humidity of the room increases, leading to a degradation of the indoor environment. Hence, to maintain a comfortable indoor environment, a dehumidification system is required.

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Nomenclature

$A_{al\text{-}in}$	area of aluminium surface in contact with water (m^2)
A_b	area of desiccant surface except for curve-shaped desiccant surface in contact with air (m^2)
A_f	area of curve-shaped desiccant surface in contact with air (m^2)
D	diffusion coefficient ($m^2\,s^{-1}$)
d	diameter (m)
d_h	hydraulic diameter (m)
h	specific enthalpy ($kJ\,kg^{-1}$)
j_m	mass flux between air and desiccant wall surface ($kg\,m^{-2}\,s^{-1}$)
K_h	overall heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
K_m	overall mass transfer coefficient ($kg\,m^{-2}\,s^{-1}$)
Le	Lewis number (–)
L	desiccant length along air flow path (m)
l_e	entrance region length (m)
l_h	pitch distance between flat walls (m)
l_p	wavelength of corrugation (m)
l_r	tube length along water flow path (m)
m_a	mass fraction of water vapour in moist air ($kg\,kg^{-1}$)
m_b	mass fraction of water vapour of desiccant wall at equilibrium ($kg\,kg^{-1}$)
N	rotational speed (rph)
Nu	Nusselt number (–)
Nu_t	nondimensional overall heat transfer coefficient (–)
Pr	Prandtl number (–)
Q	heat transfer rate (kW)
q_s	heat flux between air and desiccant wall surface ($kW\,m^{-2}$)
Re	Reynolds number (–)
Sc	Schmidt number (–)
Sh	Sherwood number (–)
Sh_t	nondimensional overall mass transfer coefficient (–)
S_t	switching time (s)
T	temperature (°C)
t	time (s)
t_b	thickness of corrugated sheet (m)
u	velocity ($m\,s^{-1}$)
V	volume (m^3)
X	mass fraction of water in desiccant ($kg\,kg^{-1}$)
x	humidity ratio ($g\,kg^{-1}$ (DA))
y	y axis (–)
z	z axis (–)
α_a	heat transfer coefficient between air and desiccant surface ($W\,m^{-2}\,K^{-1}$)
α_{in}	heat transfer coefficient between tube surface and water ($W\,m^{-2}\,K^{-1}$)
β	mass transfer coefficient between air and desiccant wall surface ($m\,s^{-1}$)
θ	angle of desiccant wheel (rad)
λ	thermal conductivity ($W\,m^{-1}\,K^{-1}$)
η	fin efficiency (–)
ρ	density ($kg\,m^{-3}$)
ω	angular speed of desiccant wheel ($rad\,s^{-1}$)

Subscripts

a	moist air in air channel
ads	adsorption
al	aluminium
b	desiccant bed
co	cooled
coi	cooling water inlet
he	heated
hei	heating water inlet
in	inside
ini	initial
out	outside
pi	process air inlet
po	process air outlet
r	water
ri	regeneration air inlet
vap	vaporization
w	water

A desiccant dehumidification method can dehumidify air by converting latent heat into sensible heat, making it unnecessary to supercool and then reheat the air in a mechanical dehumidification system such as a compression-type air conditioning system (ASHRAE, 2001).

Recently, solid desiccant air conditioning systems have been attracting attention because they can be driven by solar energy, waste heat, etc. In the case of solar energy as the driving heat source, Mavroudaki et al. (2002) and Halliday et al. (2002) independently conducted two feasibility studies of solar driven desiccant cooling in diverse European cities representing different climatic zones. Normally, a heat source with a temperature of at least 60–80 °C is required to drive a desiccant air conditioning system (Harriman, 1994; Meckler, 1994). However, this heat level cannot always be easily obtained. For a hybrid air conditioning system with desiccant, Yadav (1995), Dhar and Singh (2001) and Jia et al. (2006) each investigated the performance of a hybrid desiccant cooling system comprising a conventional vapour compression-type system coupled with a desiccant dehumidifier. However, because a conventional single stage desiccant was used for this system, the condensation temperature of the vapour compression-type refrigerator increased greatly. Moreover, sometimes, an electric heater is also used to compensate for ;the shortage of driving heat. Such a system cannot increase the system performance. To achieve a high performance desiccant air conditioning system that can utilize various types of heat sources, it is necessary to decrease the driving heat source temperature for the desiccant regeneration. Exhaust heat such as that of a compression-type refrigerator, whose temperature is about 40–50 °C, commonly exists everywhere. Exhaust heat is generally considered to be waste heat; the utilization of all this waste heat to drive a desiccant dehumidification system would lead to large energy savings.

With this background, we previously investigated approaches to reduce the temperature of the heat source of