Applied Energy 224 (2018) 382–397

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Performance investigation of an internally cooled desiccant wheel

Xingchao Zhou[a,*], Mark Goldsworthy[b], Alistair Sproul[a]

[a] School of Photovoltaic and Renewable Energy Engineering, University of New South Wales, Sydney, NSW 2052, Australia
[b] The Commonwealth Scientific and Industrial Research Organisation (CSIRO), Energy Centre, Mayfield West, NSW 2304, Australia

HIGHLIGHTS

- An isothermal process improves the dehumidification performance around 48%.
- A mathematic model could be used to predict the performance of the new wheel.
- The overlap of desiccant layers restricts heat exchange performance of the wheel.
- Desiccant cooling system with the new wheel has an energy efficiency ratio of 9.3.

ARTICLE INFO

Keywords:
Desiccant wheel
Non-adiabatic process
Dehumidification
Air conditioning

ABSTRACT

Desiccant wheels are commonly used to dehumidify air in air-conditioning system to reduce the energy consumption. The objective of this study was to design, test and analyse the performance of a novel tube-shell, internally water-cooled desiccant wheel. Cooling water was used in the supply section of the new wheel to shift the dehumidification process from nearly adiabatic to nearly isothermal. We conducted a series of experiments to investigate the wheel's performance and used the experimental results to validate a mathematical model. The model was then applied to investigate the influence of modifications to the base design, including changing the number of desiccant layers inside the heat-exchange tubes, and the size and number of tubes. We also analysed the influence of water temperature and flow rate on both dehumidification and temperature rise across the wheel. Our results show that isothermal dehumidification performance can be achieved, but only with no more than two desiccant layers inside the heat-exchange tubes. More layers reduced heat transfer between the air in the innermost layers and the cooling water. Lower cooling water inlet temperatures led to lower air outlet humidity and temperature. A cooling water temperature of approximately 24 °C was required to achieve isothermal dehumidification for process air with inlet temperature of 30 °C and absolute humidity of 16.3 g/kg and regeneration air with the inlet temperature of 50 °C and absolute humidity of 16.3 g/kg. This corresponds to a 48% improvement in dehumidification performance (the maximum absolute humidity change between inlet and outlet process air) compared with a conventional adiabatic desiccant wheel while using super-adsorbent polymer as the desiccant material.

1. Introduction

Heating, ventilation and air conditioning systems are used to create a comfortable environment inside buildings. However, they contribute to a large portion of overall final energy consumption: for example, around 50% of building final energy consumption and 20% of total energy consumption in the United States [1]. This places a strain on the electricity network as well as indirectly contributing to environmental problems such as ozone-layer depletion, air pollution and climate change. Desiccant-based air-conditioning systems can reduce these negative effects, because they can be driven by waste heat, such as the heat discharged from distributed power generation, various cogenerators and solar thermal collectors [2]. As such, they require a much smaller amount of electricity to operate. In addition, they do not use chlorofluorocarbon refrigerants, and so do not harm the environment. Henning [3] estimated that approximately 70 solar-assisted air-conditioning systems were installed in Europe in 2007. However, he also points out that the technology is still in the early stage of development, since many of the installed systems did not achieve the expected energy savings. Hence, further effort is required to improve overall efficiency.

The desiccant component of desiccant-based air-conditioning systems may be based around a liquid-desiccant heat exchanger or a solid-

* Corresponding author.
E-mail addresses: xingchao.zhou@csiro.au (X. Zhou), mark.goldsworthy@csiro.au (M. Goldsworthy), a.sproul@unsw.edu.au (A. Sproul).

https://doi.org/10.1016/j.apenergy.2018.05.011
Received 25 January 2018; Received in revised form 1 May 2018; Accepted 3 May 2018
0306-2619/

Nomenclature

A	area (m^2)
A_{face}	direct faced area of desiccant layer to air flow (m^2)
c	thickness (m)
c_p	specific heat capacity at constant pressure ($J\,kg^{-1}\,K^{-1}$)
COP	electrical coefficient of performance
CP	cooling capacity of desiccant wheel (W)
dP	drop pressure for air stream (Pa)
D	diameter (m)
D_{eff}	effective moisture diffusivity ($m^2\,s^{-1}$)
D_h	hydraulic diameter (m)
E	enthalpy of air (kj/kg)
EER	energy efficiency ratio
f	fraction of active desiccant
f_{air}	contact factor for air stream
f_{desi}	contact factor for desiccant matrix
h	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
h_{adsor}	heat of adsorption ($J\,kg^{-1}$)
$h_{channel}$	height of desiccant channel (m)
h_m	mass transfer coefficient ($kg\,m^{-2}\,s^{-1}$)
k	thermal conductivity ($Wm^{-1}\,K^{-1}$)
Le	Lewis number
$\dot{m}_a$	mass of air streams (kg/s)
n	number of desiccant layers
Nu	Nusselt number
P	power/electricity consumption (W)
P_a	contacted length between air flow and desiccant layers (m)
P_{atm}	atmospheric pressure (Pa)
P_{vs}	saturated water vapour pressure (Pa)
Q	cooling load (W)
T	temperature (K)
t	time (s)
u	velocity ($m\,s^{-1}$)
V	volume (m^3)
V_{face}	air volume directly facing desiccant layers (m^3)
$V_{channel}$	air velocity inside desiccant channel (m/s)
W	extent of adsorption (kg adsorbate per kg adsorbent)
Y	absolute humidity ($kg\,kg^{-1}$)
z	axial coordinate (m)

Greek symbols

ρ	density (m^{-3})
$\boldsymbol{\eta}$	electrical efficiency of fan or pump
$\varnothing$	relative humidity

Subscripts

1	state of inlet supply air to the system
2	state of outlet supply air from the system
a	air
channel	air channel
d	desiccant matrix
desi	desiccant layer owned by one channel
d−t	heat transfer between desiccant and heat exchange tubes
fan	electrical fan used to pump air
in	inlet air stream
inner	inner diameter of heat exchange tube
IEC	indirect evaporative cooler
l	liquid water
Layer	surface of desiccant matrix away from air stream
n	number of desiccant layer
out	outlet air stream
outer	outer diameter of heat exchange tube
pump	water pump
r	inlet regeneration air
reg	regeneration air stream
s	inlet supply air
sup	supply air stream
surface	surface of desiccant matrix close to air stream
t	heat exchange tube
tower	cooling tower
tube	heat exchange tube
t−w	heat transfer between cooling water and heat exchange tubes
w	water
wheel	internally cooled desiccant wheel
v	water vapour

desiccant rotary wheel. Solid-desiccant systems have several advantages over liquid-desiccant systems. These include avoiding potential desiccant loss and contamination of the building supply air due to the carryover of desiccant sorbent into the air streams, and avoiding corrosion issues that arise when using liquid desiccant. Disadvantages of solid-desiccant systems include the need for rotating air seals and the higher temperature of regeneration air. In an air-conditioning system, the desiccant component performs latent cooling, while different combinations of direct evaporative coolers, indirect evaporative coolers, chiller coils or heat recovery devices perform the sensible cooling. Additional components may be present and several different operating cycles are possible [4].

A review of selected literature reveals that many investigations into improving the performance of desiccant based air-conditioning systems have been conducted since the 1970s. These include three different desiccant cooling cycles: the ventilation cycle, the recirculation cycle and the Dunkle cycle, which were analysed in [5]. The results revealed that the coefficient of performance (COP) could be improved further by strengthening the performance of the dehumidifier. Two more desiccant cooling cycles were investigated in [6], which were a simplified, advanced, solid desiccant cycle and a direct-indirect, evaporative cooling cycle with a COP higher than 2.0. The feasibility of combining desiccant cooling systems with conventional air-conditioning systems was analysed in [7], which revealed that a desiccant driven by low-temperature regeneration air was the key to improving the system's energy performance.

In addition to system studies, researchers have also investigated individual components, especially desiccant dehumidifiers. For example, modelling studies of the desiccant wheel used the effectiveness-NTU (number of transfer units) method to predict the moisture removal performance of the desiccant wheel with a high accuracy [8]. Structural studies of the wheel [9] reduced the energy consumed by fans and blowers due to the decrease of drop pressure achieved by redesigning the traditional honeycomb matrix structure; the temperature of the regeneration air for desorption was also reduced. Operating condition studies [10] revealed that wheel rotation speed could significantly influence the performance of the wheel, and that the optimal rotational speed to maximise dehumidification effectiveness depends on the operating conditions. Optimisation studies to improve the dehumidification performance of the desiccant wheel [11] adopted a model predictive control strategy to set a maximum moisture removal capacity for the desiccant wheel. Studies of the heat source used to make regeneration air for the desiccant wheel [12] showed that regeneration air could be made using thermal energy from a micro-cogenerator with a temperature of less than 70 °C, while [13] involved heating regeneration air using a solar hot water system.