Applied Energy 212 (2018) 401–415

Contents lists available at ScienceDirect

Applied Energy

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

Mathematical modeling and performance evaluation of a desiccant coated fin-tube heat exchanger

Mrinal Jagirdar[a], Poh Seng Lee[b,*]

[a] Multi-Disciplinary Laboratory Blk E3, #03-01, 2 Engineering Drive 3, Singapore 117581, Singapore
[b] Department of Mechanical Engineering, National University of Singapore, Singapore 117575, Singapore

HIGHLIGHTS

- A mathematical model is developed to simulate a desiccant coated fin tube HX.
- Model considers solid mass transfer resistance, fin efficiency and parasitic power.
- Parametric study assesses the effects of geometrical parameters and flow rates.
- Component and system performance are shown for available waste heat at 50 °C.
- Integration with conventional HVAC system results in energy savings of up to 31%.

ARTICLE INFO

Keywords:
Desiccant dehumidification
Fin tube heat exchanger
Desiccant coated heat exchanger
Mathematical modeling
Air conditioning
HVAC

ABSTRACT

A solid-desiccant system that utilizes low grade heat is potentially a viable add-on to conventional HVAC systems since it can help reduce power consumption significantly, for achieving indoor thermal comfort conditions. In contrast to desiccant wheels which carry out adiabatic dehumidification, isothermal dehumidification process that may be realized by a cross-flow heat exchanger is much more efficient. In this paper, a novel mathematical model is developed to simulate heat and mass exchange phenomena of a desiccant coated fin tube heat exchanger (DCFTHX). This model takes solid side mass transfer resistance as well as fin efficiency into consideration. The model is validated using experimental results in the literature. It is also compared against simplified models to establish its utility. A parametric study is then conducted to investigate the effects of geometrical parameters as well as mass flow rate of water and air velocity on dehumidification and adsorption heat removal performance of the DCFTHX as well as the performance of the augmented air-conditioning system under warm and humid ambient conditions. Under the range of parameters and conditions simulated, if low grade waste heat (50 °C hot water) is available for regeneration, integration of DCFTHX sub-system with a conventional air conditioning system can yield as high as 31% energy savings (even when the additional power consumed by pumps and blower fans is accounted for).

1. Introduction

Building energy consumption in developed countries accounts for about 50% of the total energy use [1]. Especially for places experiencing hot and humid climates, HVAC (Heating, ventilation and air-conditioning) equipment is responsible for the largest fraction of building energy consumption [2]. Thus, air-conditioning contributes significantly to climate-change. A significant proportion of the cooling load on air-conditioners is due to the latent heat of moisture in air that needs to be removed for achieving a comfortable indoor environment. Desiccant wheels, presented by Pennington [3] and Dunkle [4] have sometimes been used for removal of humidity along with conventional air-conditioners, thereby removing/reducing the latent-heat load on the latter. However, desiccant wheels have two major limitations that stringently restrict their applications: (i) they require a heat source at a reasonably high temperature (typically > 80 °C [5]) for desiccant regeneration (ii) heat of adsorption released during dehumidification increases the temperature of the desiccant, thereby reducing its dehumidification capacity.

Several novel approaches have dealt with these limitations. Tu et al. [5] used a two-stage desiccant cooling system consisting of two desiccant wheels and three evaporators as well as three condensers. Rady et al. [6] used embedded phase change material into the adsorbent bed to take up adsorption heat released during dehumidification and

* Corresponding author.
E-mail addresses: mrinalkj@nus.edu.sg (M. Jagirdar), mpelps@nus.edu.sg (P.S. Lee).

https://doi.org/10.1016/j.apenergy.2017.12.038
Received 29 September 2017; Received in revised form 4 December 2017; Accepted 5 December 2017

Nomenclature

A	area (m^2)
A_o	minimum free flow area (m^2)
Bi	Biot number
C_p	specific heat (J/(kg-K))
d	diameter (m)
D	mass diffusivity (m^2/s)
D_h	hydraulic diameter (m)
E	enthalpy (J/(kg-K))
f	friction factor
f_d	mass fraction of sorbent in felt
G	mass flux (kg/(m^2 s))
h	heat transfer coefficient (W/(m^2 K))
H_a	height of the air channel (m)
H_d	desiccant layer thickness (m)
H_f	fin thickness (m)
h_m	mass transfer coefficient (m/s)
I_0	modified zero-order Bessel function of the first kind
I_1	modified first-order Bessel function of the first kind
j	Colburn's factor
K_0	modified zero-order Bessel function of the second kind
K_1	modified first-order Bessel function of the second kind
k_c	contraction coefficient
k_d	thermal conductivity of the desiccant (W/(m-K))
k_e	expansion coefficient
k_f	thermal conductivity of the fin (W/(m-K))
L_x	length of the fin (along direction of air flow) (m)
L_y	width of the fin (m)
L_z	tube length (height of HX) (m)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
N_f	number of fins
N_r	number of tube rows
N_t	total number of tubes
Nu	Nusselt number
P	pressure (Pa)
P_f	fin pitch (m)
Pr	Prandtl number
Q	volumetric flow rate (m^3/s)
q''_{gen}	heat generation rate per unit area (W/m^2)
q_{ads}	heat of adsorption (J/kg)
q_{eva}	latent heat of evaporation (J/kg)
q_t	heat transfer rate at the tube-fin interface (W)
$r_{1,i}$	inner radius of tube (m)
$r_{1,o}$	outer radius of tube (m)
r_2	radius of the sub-sector of the fin (m)
Re_{dc}	Reynolds number based on collar diameter and maximum velocity
St	Stanton number
T	temperature (°C)
t	time (s)
t_1	dehumidification time period (s)
t_2	regeneration time period (s)
U	velocity (m/s)
V	effective volume of DCTFHX (m^3)
W	sorbate uptake (kg water/kg sorbent)
W_{eq}	width of equivalent air channel without tubes (m)
W_{gen}	rate of phase change (1/s)
X_l	longitudinal tube pitch (m)
X_t	transverse tube pitch (m)
Y	specific humidity (kg moisture/kg dry air)
ΔV	volume of fin and desiccant in a unit cell (m^3)
Δx	distance from leading/trailing fin edge to the first/last tube centre (m)
ø	relative humidity
α	ratio of heat transfer area on air side to volume of a HX
β	factor to account for smaller tube area near fin ends
ε	pipe roughness (m)
ε_d	porosity of the desiccant
$\eta_{f,app}$	apparent fin efficiency
μ	dynamic viscosity (Pa·s)
ρ	density (kg/m^3)
σ	ratio of minimum free flow area to frontal area
υ_r	pore radius (m)

Sub/super-scripts

a	air
amb	ambient
atm	atmospheric
aux	auxiliary components (blowers and pumps)
avg	time average
b	base (fin-tube interface)
c	tube collar
com	compressor
con	conventional
d	desiccant
de	dehumidification
dry	dry part of air
F	fanning
f	fin
fr	frontal
i	inner
in	inlet
int	desiccant-air interface
M	macroscopic
m	modified
max	maximum
m_{eq}	matrix equivalent
o	out/outer
r	room
re	regeneration
s	surface
s-avg	spatial average
t	tube
v	vapour
vs	vapour saturation
w	water
wet	wetted
'	value at leading edge
κ	knudsen

Abbreviations

APC	auxiliary power (blowers and pumps) consumption due to DCFTHX
c.v.	control volume for the discretized equations
CC	cooling capacity of DCFTHX
CL	cooling load
COP	coefficient of performance
DCFTHX	desiccant coated fin tube heat exchanger
EE	energy efficiency
ES	energy saving
HVAC	heating, ventilation and air-conditioning
HX	heat exchanger
TPC	total power consumption
u.c.	unit cell