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A Mathematical Model for Predicting the Performance of Liquid Desiccant Wheel

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Abstract

The liquid desiccant cooling system is found to be a good alternative of conventional air conditioning system for better control of both latent and sensible loads. The major component of a liquid desiccant cooling system is desiccant dehumidifier which controls the latent cooling load. In this paper a mathematical model for rotary type liquid desiccant dehumidifier commonly known as desiccant wheel has been presented. The desiccant wheel has a cylindrical shape with a number of identical narrow circular slots distributed uniformly over the rotor cross section. The slots are filled with a porous medium carrying the solution of liquid desiccant, to make the absorbing surface. The absorption and regeneration performance of the desiccant dehumidifier is discussed in this paper for different operating conditions. The wheel performance curves which help to determine the air outlet conditions and coefficient of performance (COP) of the system are drawn for a wide range of wheel thickness (0.06-0.6m), air mass flux (1-8 kg/m^2.s), and regeneration temperature (60-85ºC). A reduction of about 30% in outlet humidity ratio is observed with an increase in the wheel thickness from 0.06 to 0.2m. The computed results show that better supply air conditions can be obtained to provide human thermal comfort in the hot and humid climate with effectiveness of the system largely dependent on air flow rate, wheel thickness and humidity ratio of process air.

Keywords: *Liquid desiccant wheel, heat and mass transfer coefficients, air conditioning, mathematical model.*

1. Introduction

Some alternative low energy consuming cooling systems which directly utilizes the thermal energy and reduces the emission of greenhouse gases are desiccant and absorption cooling. Among these systems, desiccant cooling is the main focus of many researcher for past few years [1]. Several studies describe the basic operation of desiccant cooling system. System with rotary dehumidifier commonly known as desiccant wheel are most widely used and studied [2, 3]. Although, absorption chillers are used in European market as solar cooling systems but trend is now changing towards desiccant cooling technology. It has been proved by different researchers that desiccant cooling system has better performance as compared to the conventional air conditioning system [4, 5]. Different advancements have been made to the basic desiccant cooling cycle to improve the system efficiency. The basic advantages of desiccant cooling can be summarized as:

- The energy cost is significantly reduced.
- Free, low grade thermal energy sources such as solar, waste heat, etc. can be used to provide regeneration heat.
- Environmental friendly because of no emission of greenhouse gases since no chlorofluorocarbon based refrigerant is used.
- The sensible and latent cooling loads are independently and efficiently controlled.
- System operates at atmospheric pressure which reduces the maintenance cost.

In this paper a mathematical model is developed to study and discuss the performance of a liquid desiccant dehumidifier in terms of its performance and effectiveness considering different parameters such as the air flow rates, regeneration temperature,

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and wheel speed. Also, the effect of ambient conditions on dehumidifier performance has been examined.

2. System description

The proposed liquid desiccant dehumidifier is a rotary wheel of radius R and width L. The wheel consists of number of identical narrow slots uniformly distributed over its cross-section as shown in Fig. 1. The slots are covered with porous media impregnated with solution of liquid desiccant. The wheel has separate sections for the flow of process and regeneration air. These two streams of air flows in counter arrangement as shown in Fig. 1.

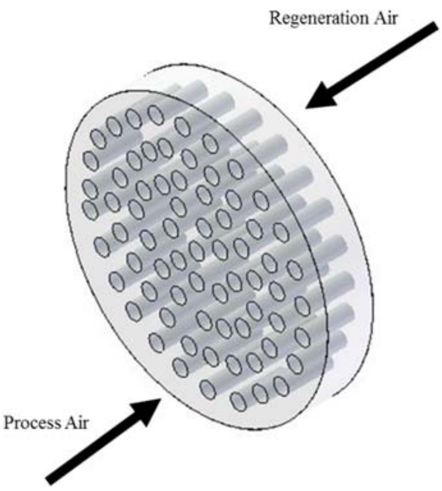

Figure 1: Schematic of liquid desiccant wheel.

3. Mathematical modeling

For modeling purpose, one slot is divided into N number of nodes (i=1, 2, 3…N) as illustrated in Fig. 2. One dimensional mass and energy balance equations are developed for air stream and desiccant surface to determine the outlet conditions of air with the following assumptions:

- The flow of air through each slot is considered as plug flow in which model variations exists only in axial direction.
- The air is assumed to be distributed uniformly across dehumidification and regeneration sections.
- The axial heat conduction and mass diffusion are neglected.
- It is assumed that there is no carry-over of the desiccant solution.
- The thermodynamic properties are assumed as constant.
- Each slot is assumed to be adiabatic.

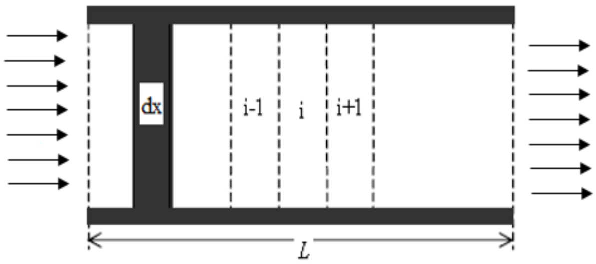

Figure 2: Pattern of flow in a slot.

The difference of vapor pressure between air and desiccant surface acts as driving force for mass transfer. For a control volume, air stream and desiccant surface moisture balance can be written in terms of moisture variations and convective mass transfer between humid air and desiccant.

$$A_c \rho_a L\left(\frac{\partial \omega_a}{\partial t} + u\frac{\partial \omega_a}{\partial x}\right) = kA_h(\omega_s - \omega_a) \quad \textbf{(1)}$$

The first and second terms on the left hand side of equation (1) are air moisture storage and rate of moisture variation due to the axial flow of air, respectively. The right hand side of equation (1) represents transfer of mass between desiccant bed and air stream due to convection.

$$\rho_d A_c L\left(\frac{\partial \omega_s}{\partial t} + \frac{\partial w}{\partial t}\right) = k A_h(\omega_s - \omega_a) \quad \textbf{(2)}$$

The left and right hand side of the equation (2) represents desiccant moisture storage and transfer of mass between desiccant bed and air stream due to convection, respectively.

The sensible and latent heat is transferred between air and desiccant due to convection and absorption phenomenon. Therefore, the energy balance for air stream and desiccant surface can be written as:

$$A_h C_{pa} \rho_a L\left(\frac{\partial T_a}{\partial t} + u\frac{\partial T_a}{\partial x}\right) = h A_c(T_s - T_a) + C_{pv} k A_c(\omega_s - \omega_a)(T_s - T_a) \quad \textbf{(3)}$$

The first and second terms of the equation (3) on the left hand side are air energy storage and energy variation due to flow of air in axial direction, respectively. While the right hand side of the equation illustrate convention and sensible transfer of heat between desiccant bed and air stream transfer, respectively.

$$C_{pd} \rho_d A_h L\left(\frac{\partial T_s}{\partial t}\right) = h A_c(T_a - T_s) + k A_c(\omega_a - \omega_s) h_{fg} + C_{pv} A_c(\omega_a - \omega_s)(T_a - T_s) \quad \textbf{(4)}$$

The term on the left hand side of the equation (4) is desiccant bed energy storage. The terms on the right hand side of equation (4) represents convection heat transfer, absorption and sensible heat transfer between air and desiccant solution respectively.

Coefficient of heat and mass transfer for rotary type liquid desiccant dehumidifiers can be expressed as [6]:

$$k = 0.704.m_a.\mathrm{Re}^{-0.51} \quad \textbf{(5)}$$

$$h = 0.671.m_a.\mathrm{Re}^{-0.51}.C_{pa} \quad \textbf{(6)}$$

Equations (1) through (6) will be solved to determine the temperature and humidity ratio of the process and regeneration air leaving the desiccant wheel.

4. Boundary and Initial Conditions

The flow of process and regeneration air are in counter flow arrangements. The boundary conditions for temperature and humidity ratio can be written as:

Process air enters the channel at axial distance L = 0:

$$T_a(0,t) = T_{p,\,in} \tag{7}$$

$$\omega_a(0,t) = \omega_{p,\,in} \tag{8}$$

Regeneration air enters the channel at axial distance L:

$$T_a(\mathrm{L},t) = T_{r,\,in} \tag{9}$$

$$\omega_a(\mathrm{L},t) = \omega_{r,\,in} \tag{10}$$

The initial conditions for temperature and humidity ratio applied to the governing equations, regardless of which air flow is being analyzed first are:

$$T_a(\mathrm{x},0) = T_{p,\,in} \tag{11}$$

$$T_s(\mathrm{x},0) = T_{r,\,in} \tag{12}$$

$$\omega_a(\mathrm{x},0) = \omega_{p,\,in} \tag{13}$$

$$\mathrm{w}(\mathrm{x},0) = \mathrm{w}_{in} \tag{14}$$

To allow independent air conditions of the process and regeneration air streams, different boundary conditions must be imposed. For most realistic applications the exiting process and inlet regeneration air conditions would be dependent in some manner. However, since such dependence is highly application specific, independent boundary conditions were used.

5. Pressure Drop

Across the desiccant dehumidifier the pressure drop can be written as [7]:

$$\Delta P = \frac{2 f \rho u^2 L}{D_h} + \frac{K_o \rho u^2}{2} \tag{15}$$

The Friction factor given is as [8]:

$$f = \frac{13}{\mathrm{Re}} \tag{16}$$

For sudden contraction loss,

$$K_o = 0.5 \tag{17}$$

6. Results and Discussion

A rotary liquid desiccant wheel is proposed in this paper to have lower regeneration temperature, no carryover of desiccant solution and better supply air conditions. A code has been developed in Engineering Equation solver (EES) using mathematical model and performance of the system is observed using initial and boundary conditions for temperature and humidity ratio.

The effect of process air mass flux on exit air humidity ratio for different widths of the rotary dehumidifier is illustrated in Fig. 3. The exit air humidity ratio increases with the increase in air mass flux which shows a decrease in air dehumidification with the increase in mass flux of air. This is due to the fact that at high flow rates, the air has less contact time with the desiccant surface and therefore, mass transfer is smaller than that at low flow rates. The system performance for dehumidification as well as for cooling process are largely affected by the width of the wheel. Figure 3 also shows that, for any given value of air mass flux, exit humidity ratio decreases with the increase of wheel width. This decrease in humidity ratio is due to increase in surface contact area and residual time between air and desiccant. This means for any value of air mass flux there should be an optimum value of wheel width for required exit conditions of air. The optimization of wheel width is favorable for the purpose of air conditioning system design because it reduces the system overall manufacturing cost.

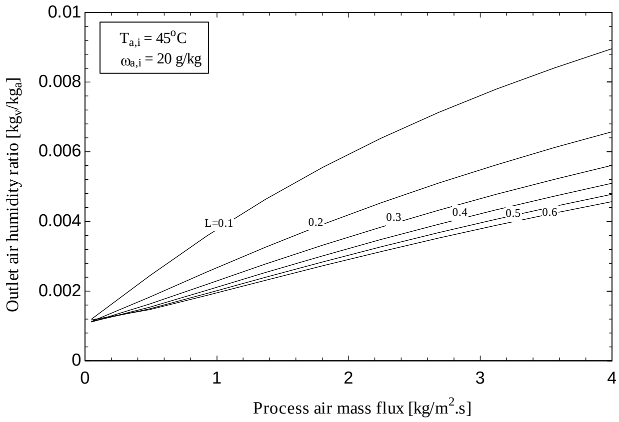

Figure 3: Effect of process air mass flux on outlet air humidity ratio.

The variation of dehumidifier effectiveness and dehumidification performance with humidity ratio of process air has been illustrated in the Fig. 4. As the humidity ratio of process air at the inlet increases the dehumidifier effectiveness and DCOP increase because of better mass transfer between two streams. The variation of DCOP for latent cooling with rotational speed of the wheel at different regeneration temperatures is presented in Fig. 5. The DCOP decreases with the increase in rotational speed because of less contact time between humid air and desiccant surface. The occurrence of the maximum DCOP at a low rotational speed is expected. If the rotational speed is too high the desiccant cannot fully absorb or desorb water molecules because of the less process time.

The pressure drop across the dehumidifier increases and so does the required fan power with the increase in air flux passing through the dehumidifier and a similar effect on pressure drop is observed with the increase in dehumidifier thickness as illustrated in Fig. 6. Figure 7 shows the air flux effect on the effectiveness of the dehumidifier and moisture removal from the humid air. At high value of air flux the wheel effectiveness and removed moisture from the moist air decreases. This happens because of increase in heat and mass transfer coefficients.

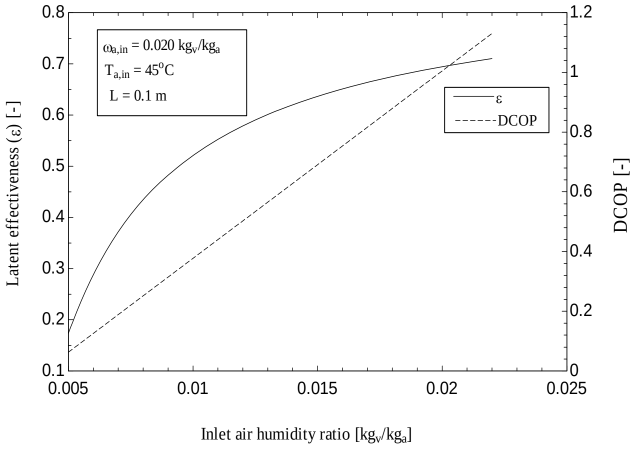

Figure 4: Effect of inlet air humidity ratio on the effectiveness and DCOP.

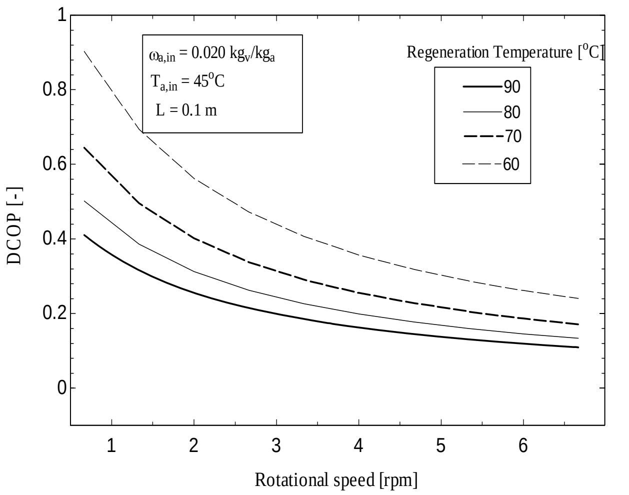

Figure 5: Effect of rotational speed on DCOP.

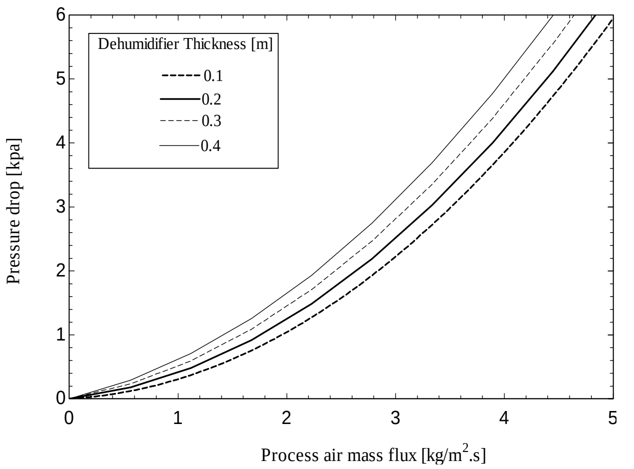

Figure. 6: Effect of air flux on pressure drop.

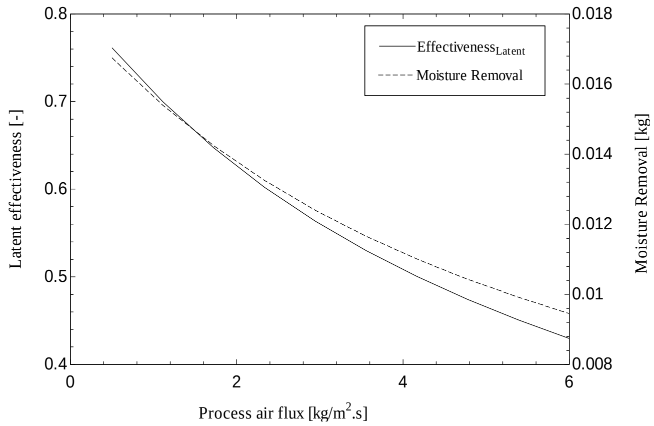

Figure 7: Effect of process air flux on effectiveness and moisture removal.

7. Conclusions

A theoretical model for the liquid desiccant wheel has been developed to observe the heat and mass transfer phenomenon for absorption and desorption process under different operating parameters like process air flow rate, humidity ratio of process air, and width of the dehumidifier. Effect of these parameters also been observed on dehumidifier effectiveness, moisture removal rate and DCOP. The computed results show that supply air condition can be obtained to provide human comfort in the hot and humid climate with effectiveness of the system largely dependent on air flow rate, dehumidifier width and humidity ratio of process air. For better performance of the dehumidification system the regeneration heat required should be as minimum as possible. The regeneration heat depends on the regeneration air flow rate. An optimum and feasible value of the regeneration air flow rate can be obtained by the parametric analysis in the required zone of operation.

Different performance curves of the desiccant dehumidifier have been drawn in order to determine optimum values of different parameters. The result shows that there is an optimum value of each design parameter under each operating condition and above that value, there is very little change in the performance of desiccant dehumidifier. The main conclusions emerging from the present work can be summarized as:

- The dehumidification performance of the desiccant wheel increase with the increase in process air inlet humidity ratio because of better mass transfer between two streams.
- Increasing the wheel width leads to better dehumidification of process air; however, the rate is decreased because of decrease in moisture difference between humid air and desiccant surface.
- Lower air flow rates produces better dehumidification of process air.
- The DCOP of the system is high at lower rotational speed because of the more process time for absorption and desorption of moisture.
- As the regeneration temperature or flow rate of regeneration air increases, coefficient of performance for latent cooling decreases because of high required input heat.

For better performance of the dehumidification system the desiccant material should have lower temperature of regeneration which is a key factor for future research.

Nomenclature

A	channel area (m^2)
C_p	specific heat (kJ/kg.K)
h	heat transfer coefficient ($W/m^2.K$)
H	specific enthalpy (kJ/kg)
h_{fg}	latent heat of vaporization (kJ/kg)
k	mass transfer coefficient($kg/m^2.s$)
L	dehumidifier width (m)
$\dot{m}$	air mass flux ($kg/m^2.s$)
N	number of cells in each slot (-)
T	temperature (K)
t	time (s)
u	face velocity (m/s)
V_H	air specific volume (m^3/kg)
w	desiccant moisture content (kg_v/kg_a)
Greek letters	
ρ	density (kg/m^3)
ω	absolute humidity (kg_v/kg_a)
Subscripts	
a	air
c	cross sectional
d	desiccant material
f	wall material
h	internal surface
p	process
r	regeneration
s	desiccant surface
v	water vapor
w	liquid water

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