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Heat and Mass Transfer between Humid Air and Desiccant Channels — A Theoretical Investigation

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Abstract: Due to the direct utilization of thermal energy and possibility of using renewable energy resources, the interest of using desiccant wheel for air conditioning application is increasing rapidly. The thermally driven desiccant cooling systems are environmental friendly and have a great potential to reduce the peak electricity demand. The high initial cost of these system as compared to the conventional units can be reduced at designing stage through selection of suitable cycle, size reduction, and flow optimization. The aim of this paper is to develop a mathematical model for the desiccant wheel to predict its dynamic performance. The model is numerically stable and easy to simulate. The desiccant wheel performance has been discussed under different operating parameters. The effect of mass flow rate on heat and mass transfer coefficient has also been studied. Results showed that, an optimum value of operating parameters like mass flow rate and regeneration temperature should be selected for better performance of the system. In addition, transient variation of vapor adsorption rate has also been discussed.

Key words: desiccant wheel, modeling, mass transfer, solar energy, latent load, air conditioning

1. Introduction

Desiccant wheels which are composed of some sorption material are used to dehumidify the air. A typical desiccant wheel is shown in Fig. 1, which consists of number of channels emended with porous desiccant material. The enlarged view of the channels is shown in Fig. 2. The desiccant wheel is divided into two sections that is adsorption and desorption as illustrated in Fig. 1. Generally, the adsorption zone shares the major part of the wheel i.e. about 2/3. The wheel rotates at very low speed to obtain the better outlet conditions. For the regeneration/reactivation of the desiccant wheel heated air is passed through the regeneration zone of the wheel. Some low grade energy resources such as solar, waste, biomass etc. can be used to heat the air up to the regeneration temperature.

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The major application of the desiccant wheel is to control the humidity of the indoor air [3]. Dehumidification or drying of air is a mass transfer process which results in water removal from the air [4]. The dehumidification of air becomes more important for the application such as, food storage, pharmaceuticals, fine chemicals industries, etc. [5]. Also, drying is required to obtain the required material in some areas of synthetic chemistry [6].

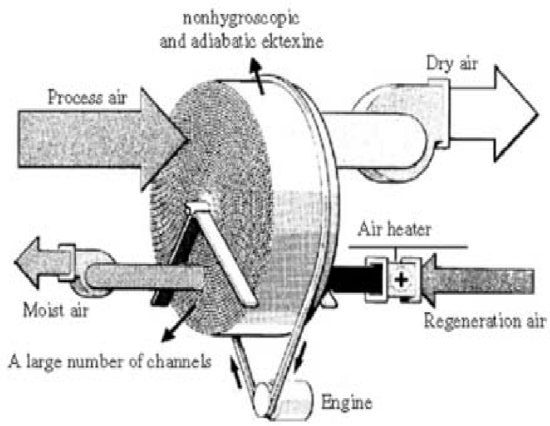

Fig. 1 Solid desiccant wheel [1].

Pesaran and Mills [7] studied the behavior of moisture transport in a packed bed system using silica gel as the desiccant material. San and Jiang [8] modeled and tested a desiccant dehumidification system compromising of two column packed bed dehumidifier. Chen and Pei [9] also developed a drying model which can be used for both non hygroscopic as well as hygroscopic materials. In all these models mentioned above, thin layer drying was not considered. The thin layer drying model describes the mechanisms of combined mass and heat transfer phenomena and enables to investigate the effects of certain parameters on process of moisture removal.

The adsorption-type rotary dehumidifier is the main focus of past all investigation and there are very few work on the rotary absorption type. Kodama et al. [10] experimentally investigated the optimal operation of a honeycomb rotor adsorber. Maclaine-cross and Banks [11] and Banks [12] utilized linear and nonlinear analogy method to study simultaneous heat and mass transfer performance of the rotary dehumidifier, whereas Mathiprakasam and Lavan [13] used linear solution to predict the performance of adiabatic desiccant dehumidifier.

In this paper a mathematical model for solid desiccant wheel has been developed using heat and mass transfer equations to predict its dehumidification performance for various operating conditions. The performance of the system has been analyzed considering different parameters such as the air flow rates and regeneration temperature.

2. Mathematical Modeling of Solid Desiccant Wheel

A rotary solid desiccant wheel is a mass and heat exchanger having low rotational speed. The desiccant material is embed into the wheel using a matrix of supporting material forming a number of flow channels as shown in Fig. 2. These flow channels can triangular, sinusoidal, honeycomb, triangular, etc. in shape. The channel shape considered for this model is sinusoidal as shown in Fig. 3.

The wheel has been divided into two sections, i.e., process and regeneration. On process side moisture is adsorbed from the air on the desiccant surface (air dehumidification), while in the regeneration section, adsorbed moisture is removed using high temperature regeneration air stream (air humidification).

One channel of length “dx” of sinusoidal shape has been considered as the differential control volume as shown in Fig. 4. Note that two channels are sharing one layer of desiccant and matrix material.

Where for one channel,

Flow passage cross sectional area = $A_{c=}$ 2ab

Channel wall thickness = δ

Flow passage height = 2a

Flow passage pitch = 2b

Total cross sectional area = A_h

$$A_h = \frac{1}{2} \times (2a + \delta) \times (2b + \delta) \tag{1}$$

Flow channel perimeter is given as:

$$P_e = 2b + 2\sqrt{b^2 + (a\pi)^2}\, \frac{3 + \left(\frac{2b}{a\pi}\right)^2}{4 + \left(\frac{2b}{a\pi}\right)^2} \tag{2}$$

Flow channel hydraulic diameter:

$$D_h = \frac{4A_h}{P_e} \tag{3}$$

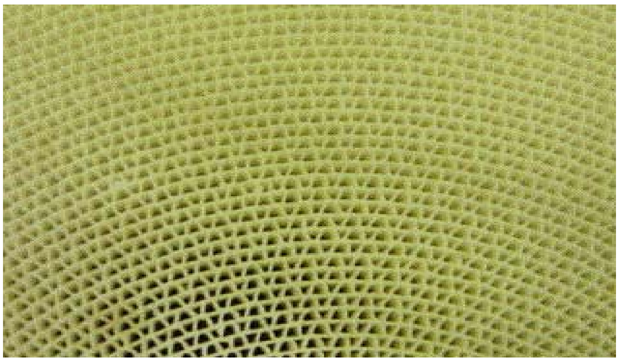

Fig. 2 Enlarged view of air channels inside the desiccant wheel [2].

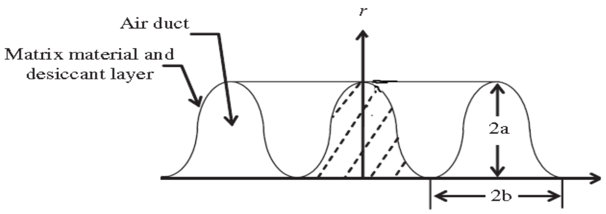

Fig. 3 Dimensions of the flow channel.

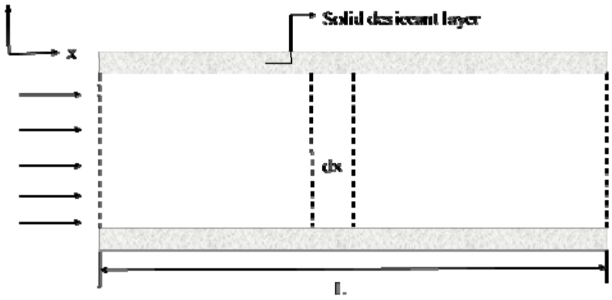

Fig. 4 Differential control volume for one channel.

2.1 Assumptions

One slot is divided into N nodes (i = 1, 2, 3…N) to develop the energy and mass balance equations for air and desiccant stream considering one dimensional flow. The following assumptions are made for this theoretical analysis are:

(1) Laminar flow.
(2) Adiabatic model.
(3) Unsteady-state model.
(4) Piston flow, i.e., model variations exists only in axial direction.
(5) Solid desiccant is uniform in size during drying.

2.2 Governing Equations

Four governing equations for two different control volumes in the channel, one for the moist air and other for the desiccant material layer are derived using fundamental laws of mass and heat transfer.

The humidity difference acts as the driving force for mass transfer coefficient k. The mass conservation equation for a control volume of air stream is:

$$\frac{(\omega_{a,i}-\omega_{a,i-1})}{\Delta t}=u\frac{(\omega_{a,i-1}-\omega_{a,i})}{\Delta x}+\frac{k.A_h}{A_c.\rho_a.L}(\omega_{s,i}-\omega_{a,i}) \tag{4}$$

For air stream, energy balance can be written as:

$$\mathrm{C_{pa}}.\frac{(T_{a,i}-T_{a,i-1})}{\Delta t}=\mathrm{C_{pa}}.\mathrm{u}\frac{(T_{a,i-1}-T_{a,i})}{\Delta x}+\frac{h.A_c}{A_h.\rho_a.L}(\mathrm{T}_{s,i}-T_{a,i}) \tag{5}$$
$$+\frac{\mathrm{C_{pv}}.\mathrm{k}.A_c}{A_h.\rho_a.L}(\omega_{\mathrm{s,i}}-\omega_{a.i}).(\mathrm{T}_{s,i}-T_{a,i})$$

Similarly, mass and energy balance for desiccant bed is represented in Eqs. (6) and (7), respectively.

$$(\frac{\omega_{s,i}-\omega_{s,i-1}}{\Delta t})=\frac{k.A_h}{\rho_d.\mathrm{A_c}.L}(\omega_{s,i}-\omega_{a,i}) \tag{6}$$

$$(\frac{T_{s,i}-T_{s,i-1}}{\Delta t})=\frac{h.\mathrm{A_c}}{(C_{pd})_{\mathrm{effective}}.\mathrm{A_h}.\rho_d.\mathrm{L}}(T_{a,i}-T_{s,i})+\frac{k.\mathrm{A_c}}{(C_{pd})_{\mathrm{effective}}.\mathrm{A_h}.\rho_d.\mathrm{L}}(\omega_{a,i}-\omega_{s,i}).\mathrm{Q} \tag{7}$$
$$+\frac{\mathrm{C_{pv}}.\mathrm{A_c}}{(C_{pd})_{\mathrm{effective}}.\mathrm{A_h}.\rho_d.\mathrm{L}}(\omega_{a,i}-\omega_{s,i}).(T_{a,i}-T_{s,i})$$

The saturated humidity ratio of the air can be written in terms of saturated vapor pressure of water as:

$$\omega_{si}=\frac{0.622P_{vi}}{1.0133\times10^5-P_{vi}} \tag{8}$$

P_{vi} is water vapor saturation pressure which is found as [14]:

$$P_{vi}=e^{\left(23.196-\frac{3816.44}{Ts,i-46.14}\right)} \tag{9}$$

2.3 Initial and Boundary Conditions

The initial conditions for temperature and humidity ratio are:

$$T_a(\mathrm{x},0)=T_{process,\,in} \tag{10a}$$
$$\omega_a(\mathrm{x},0)=\omega_{process,\,in} \tag{10b}$$
$$T_s(\mathrm{x},0)=T_{regeneration,\,in} \tag{10c}$$
$$\omega_s(\mathrm{x},0)=\omega_{process,\,in} \tag{10d}$$

The boundary conditions are:

$$T_a(0,t)=T_{process,\,in} \tag{11a}$$
$$\omega_a(0,t)=\omega_{process,\,in} \tag{11b}$$
$$T_a(\mathrm{L_w},t)=T_{regeneration,\,in} \tag{11c}$$
$$\omega_a(\mathrm{L_w},t)=\omega_{regeneration,\,in} \tag{11d}$$

2.4 Local Heat Transfer and Mass Transfer Coefficients

The Nusselt number for a fully developed flow is given by Ref. [15]:

$$Nu,_{FD}=1.1791\times\begin{bmatrix}1+2.7701(\alpha)-3.1901(\alpha)^2\\+1.9975(\alpha)^3-0.4966(\alpha)^4\end{bmatrix} \tag{12}$$

The local Nusselt number is found as [15]:

$$Nu_{local} = Nu,_{FD} + \frac{0.0841}{0.002907 + G_z^{-0.6504}} \quad (13)$$

The coefficient of local heat transfer is given as:

$$h_{local} = \frac{Nu_{local}.k_a}{D_h} \quad (14)$$

Local mass transfer coefficient is given as:

$$k_{Local} = \frac{\rho.Sh_{Local}.D_m}{D_h} \quad (15)$$

3. Performance Index

Desiccant wheel dehumidification capacity is represented by moisture removal capacity:

$$\mathrm{MRC} = \dot{m}_p(\omega_{a,i} - \omega_{a,o}) \quad (16)$$

$\omega_{a,i}$ and $\omega_{a,o}$ represents specific humidity of process air at inlet and exit of the wheel, respectively.

Latent effectiveness of the wheel is given as:

$$\varepsilon = \frac{\omega_{a,i} - \omega_{a,o}}{\omega_{a,i}} \quad (17)$$

The dehumidification coefficient of performance (DCOP) can be represented as:

$$\mathrm{DCOP} = \frac{h_{fg}(\omega_{a,i} - \omega_{a,o})}{C_{pa}(T_{reg} - T_{amb})} \quad (18)$$

4. Results and Discussions

The desiccant wheel operating and structural parameters varies a lot with area of application and different climatic conditions. For better understanding these dependencies, a mathematical model is developed to investigate the influences of the main parameters on performance of the system. The performance of the solid desiccant wheel with dimensions shown in Table 1 has been observed under different operating parameters.

The influence of humidity ratio ($\omega_{a,in}$) on dehumidification performance of the desiccant wheel and moisture removal is shown in Figs. 5 and 6, respectively. It can be observed that, both moisture removal and DCOP increases significantly with the increase of $\omega_{a,in}$. Higher $\omega_{a,in}$ causes an increase in mass transfer between desiccant and humid air which leads to increase in moisture removal capacity. DCOP has the same trend as moisture removal. As, $\omega_{a,in}$ varies from 10 to 30 g/kg, the DCOP increases from 0.1 to 0.6 at a regeneration temperature of 120°C. Fig. 5, also illustrates the effect of regeneration temperature on DCOP. It can be observed that DCOP decreases with the increase in required regeneration temperature. This happens because regeneration temperature directly affects the required input heat.

Table 1 Wheel dimensions and thermodynamic properties.

Parameter	Value	Parameter	Value
L	0.1m	ρ_d	350 kg/m^3
D_w	0.38m	ρ_f	300 kg/m^3
C_{pf}	1640j/kg.k	A_h	0.00628 m^2
C_{pd}	615 j/kg.k	A_c	0.000314 m^2
C_{pa}	1005 j/kg.k	N	10
C_{pw}	4180 j/kg.k	Q	2500 kj/kg
C_{pv}	2000 j/kg.k		

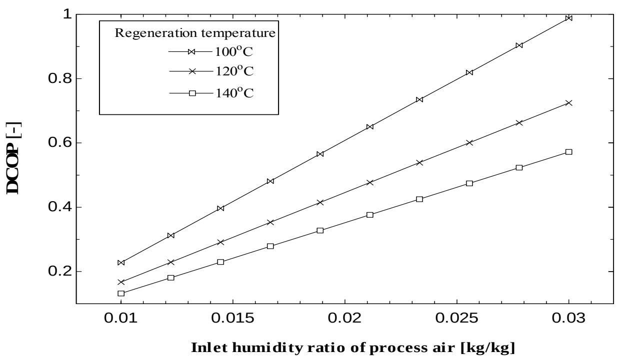

Fig. 5 Variations of DCOP w.r.t Inlet humidity of process air.

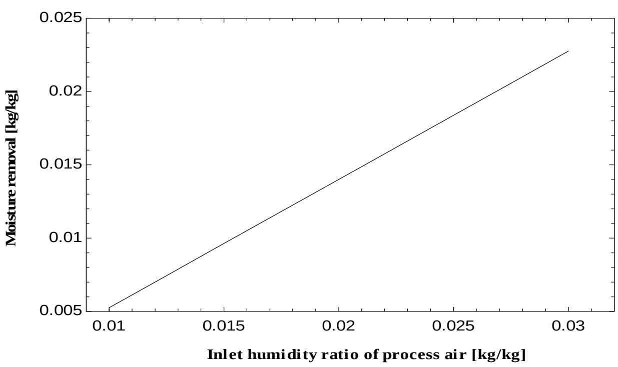

Fig. 6 Variations of moisture removal w.r.t Inlet humidity of process air.

The effect of process air flow rate on DCOP and latent effectiveness of wheel is shown in Fig. 7. Fig. 8 presents the variations of moisture removal. DCOP, effectiveness and moisture removal decreases with the increase in process air flow rate. There is about 38% decrease of DCOP when mass flow rate increases from 0.5 to 5 kg/m^2.s. Similarly, effectiveness and moisture removal decreases about 40% under the same condition. In practical application, mass flow rate should be selected carefully to trade between dehumidification performance and DCOP.

The effect of process air flow rate on outlet air humidity ratio and temperature is shown in the Fig. 9. As the air flow rate increases both the outlet humidity and temperature of the air increases which means as the mass flow rate increase the dehumidification of the air decreases so, there should be an optimum value of air flow rate.

The rate of adsorption and desorption of water vapour have a significant effect on the performance of the desiccant wheel. For the present operating conditions, the transient behaviour of the water vapour adsorption rate for desiccant wheel is presented in Fig. 10. The adsorption rate decreases rapidly at the start and then becomes almost constant. It is important to select the time of each mode to control the adsorption and desorption rate. However, air flow rate and regeneration temperature affect the rate of desorption. The desorption rate could be reduced by decrease air flow rate or regeneration temperature.

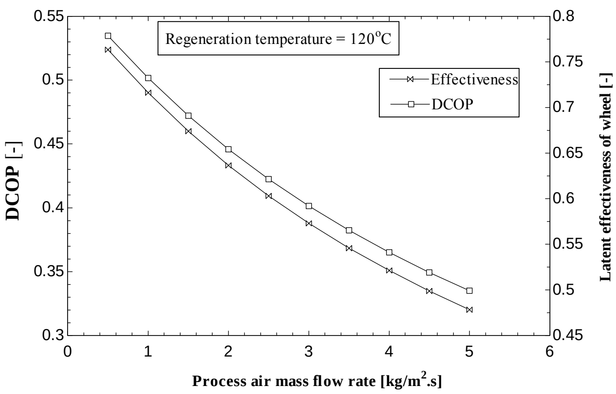

Fig. 7 Variations of DCOP and latent effectiveness of wheel w.r.t process air flow rate.

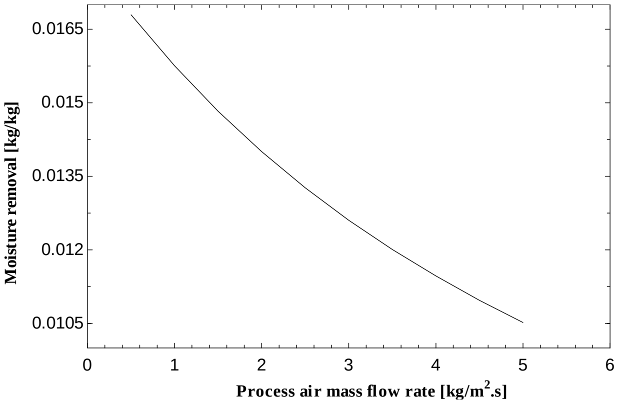

Fig. 8 Variations of moisture removal w.r.t process air flow rate.

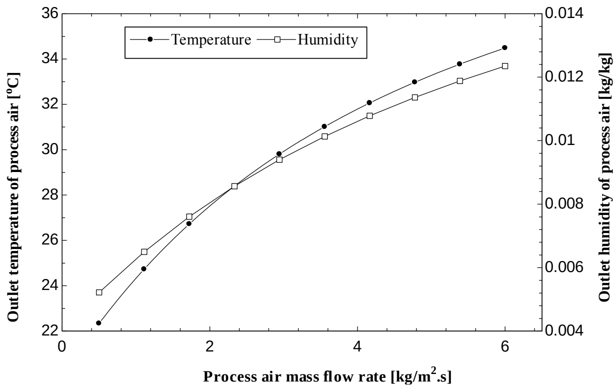

Fig. 9 Variations of outlet temperature and humidity ratio w.r.t Process air flow rate.

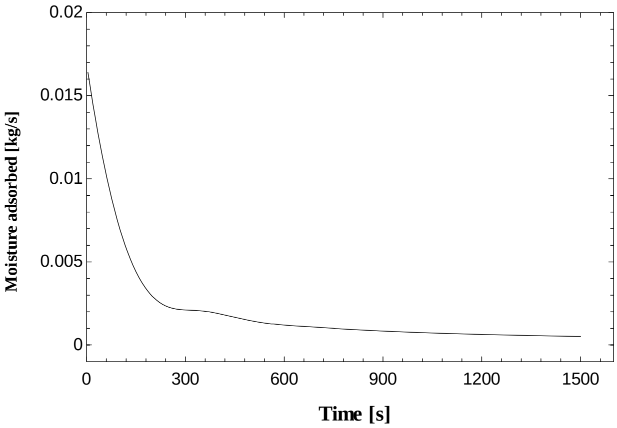

Fig. 10 Variations of moisture adsorbed from the process air w.r.t time.

5. Conclusions

A mathematical model through the mass and heat transfer equations is presented in this paper which can be used to predict the performance of rotary desiccant wheel. Effect of ambient conditions have been observed on dehumidification performance of the wheel. The results showed that as the inlet air humidity ratio is increased, the moisture removal rate increases. The temperature and humidity of the outlet processed air has been predicted with the variations of different parameters.

In addition, it has been demonstrated that the process air flow rate and regeneration temperature has a significant effect on the dehumidification performance in a way that decreasing the mass flow rate and regeneration temperature leads to better dehumidification but this improvement in the performance tends to be less marked with subsequent more decrease in these parameters. Hence, after a certain stage, little improvement will be elicited. Factors contributing to increase in the heat and mass transfer coefficient are also evaluated. Results showed that the operational conditions are very important for better performance of the system. Other method for increasing the dehumidification performance and to get the better outlet conditions is to increase the thickness of the wheel but this enlargement leads to additional financial costs and decreases the system flexibility.

Acknowledgments

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Nomenclatures

ρ_v -water vapor density (kg/m^3)
ρ_v -density of air (kg/m^3)
ω_a -air absolute humidity (kg/kg);
ω_s-saturation humidity of air (kg/kg);
N -number of cells in each slot
m_a -humid air mass flow rate (kg/m^2 s)
V_H -humid air specific volume (m^3/kg)
L -wheel thickness (m)
k -mass transfer coefficient (kg/m^2 s)
u -face velocity (m/s)
D_w -diameter of the wheel (m)
D_h-hydraulic diameter (m)
t -time (s)
A_h -internal surface area of channel (m^2)
A_c-cross sectional area of channel (m^2)
C_{pa}-dry air specific heat (kJ/kg K)
C_{pv} -water vapor specific heat (kJ/kg K)
C_{pw} -liquid water specific heat (kJ/kg K)
T_a-air temperature of air (K)
T_s-desiccant solution temperature (K)
H -heat transfer coefficient (kW/m^2 K)
ρ_f,ρ_d -wall material and desiccant density (kg/m3)
C_{pf}, C_{pd} -wall material and desiccant specific heat (kJ/kg K)
Q -latent heat of vaporization (KJ/kg)
k-thermal conductivity (W/m^2.K)
α -aspect ratio
Pe -perimeter (m)
Pr -Prandtl number
Re -Reynolds number
Nu -nusselt number
D_m-mass diffusion coefficient
f -friction factor
K_o-velocity head loss at the entry and exit of the desiccant wheel.
T_{reg} -regeneration air temperature
T_{amb}-ambient air temperature
h_{fg}-latent heat of vaporization (kJ/kg)

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