Theoretical and Experimental Investigation of the Key Components for a Rotary Desiccant Wheel

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ABSTRACT: *This paper describes the theoretical and experimental investigations of the key components and also the performance analysis of a rotary wheel for using as a solid desiccant dehumidifier and also indirect evaporator cooling system. Solid desiccants have long been used in dehumidification and cooling systems for energy efficiency or reduce electricity. Although many mathematical models on the rotary desiccant wheel have been proposed, the effect of air speed on wheel performance as a momentum equation combined with heat and mass transfer has not been studied. In this study, for the first time the two dimensional mathematical modeling of a desiccant wheel and its numerical simulation using an explicit method considering momentum equation and Ackermann correction factor were described. The results indicated that Ackermann correction factor had a significant effect on performance efficiency. Air stream velocity was one of the most effective parameter on performance and dehumidification rate of wheel. The performance was increased when process air stream velocity decreased. The model suggested the optimum air stream velocity and rotational speed of wheel as 1.86 m s^{-1} and 10 rad h^{-1}, respectively, obtaining maximum efficiency.*

KEY WORDS: *Desiccant wheel, Explicit program, Heat transfer, Mathematical model, Mass transfer.*

INTRODUCTION

In the desiccant cooling process, which is a new kind of refrigeration method, the fresh air is dehumidified and then sensibly and evaporatively cooled before being sent to the conditioned space (*Staton*, 1998) [1]. The developed psychometric model is based on the correlations between the relative humidity and enthalpy of supply and regeneration air streams (*Beccali et al.,* 2003)[2-3]. The mathematical model of an rotary desiccant wheel that can be used to calculate the performance of stationary or rotary bed and transient or steady state operation is founded by considering some of the key components [4-6]. The continuity and energy conservation equations for the transient coupled heat and mass transfer were established and solved using a finite differential model (*Dai et al.*, 2001)[7-8]. A mathematical model for explanation of the rotary desiccant wheel has been presented, in which the optimum rotational speed for achieving the maximum performance offered (*Pahlavanzadeh et al.*, 2003-2006) [9-11].

TWO DIMENSIONAL ROTARY WHEEL MODEL

In rotary heat exchangers, heat is transferred from the hot fluid to a solid energy carrier (the matrix) during

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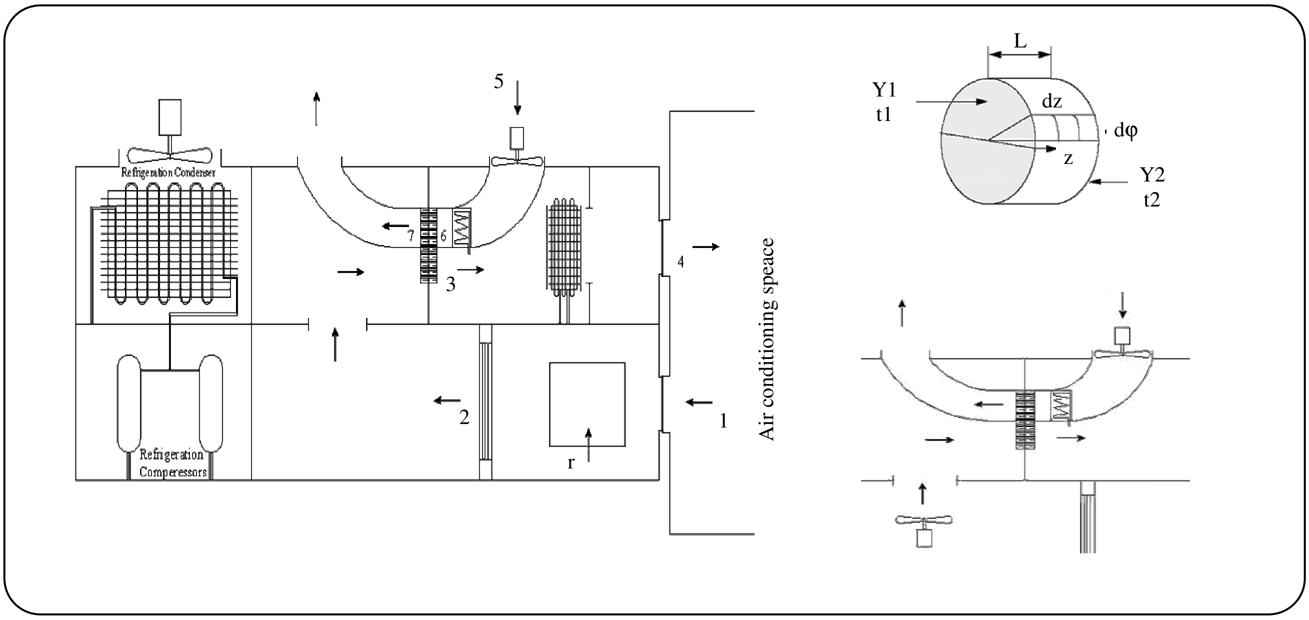

Fig. 1: Schematic diagram of the Desiccant cooling system equipment and wheel for doing experimentsf.

the first period, and, during the second period, from the solid to the cold stream. Continuous operation is permitted by rotating the matrix cyclically from one air stream to the other. Rotary heat exchangers also allow mass (water) transfer between the two air streams if the matrix contains a water adsorbing desiccant. In this case the heat exchangers exchange both sensible and latent energy, and at operating conditions where the total amount of transferred energy is at its maximum, they are referred to as enthalpy exchangers. Fig. 1 shows experimental setup system that we have applied in the present research work and also schematic coordination diagram of desiccant wheel.

Four equations concerning water content balance and energy conservation are used to describe the complicated heat and mass transfer occurring in moisture adsorption and regeneration [12-13]. Assumptions to obtain Eqs. (1) to (4) are as follows: i. effect of centrifugal force is neglected due to low rotation speed of the rotary dehumidifier; ii. no leakage takes place between dehumidification and regeneration sections; iii. shell of the rotary dehumidifier satisfies the insulated condition; iv. velocity profile is 2 dimensional and considerable along z direction; v. heat and mass transfer in radius direction is not taken into consideration; vi. desiccant is uniformly distributed in the matrix, f_V, f_S are constant; vii. thermal conductivity and diffusivity are isentropic.

Conservation of moisture for the processed air:

$$\frac{\partial(\rho_{da}Y)}{\partial\tau}+\omega\frac{\partial(\rho_{da}Y)}{\partial\varphi}+\frac{\partial(\rho_{da}Yu)}{\partial z}= K_Y\cdot f_v(Y_w-Y) \quad (1)$$

Conservation of energy for the process air:

$$\frac{\partial(\rho_{da}C_{Pe}t)}{\partial\tau}+\omega\frac{\partial(\rho_{da}C_{Pe}t)}{\partial\varphi}+\frac{\partial(\rho_{da}u)C_{Pe}t}{\partial z}= A_f\cdot\alpha f_v(t_w-t)+K_Y\cdot f_v(Y_w-Y)C_{pv}\cdot t \quad (2)$$

Conservation of water content for the absorbent:

$$\frac{\partial W}{\partial\tau}+\omega\frac{\partial W}{\partial\varphi}-D_{eff}\left[\frac{1}{r}\frac{\partial^2 W}{\partial\varphi^2}+\frac{\partial^2 W}{\partial Z^2}\right]= \frac{K_Y\cdot f_v}{\rho_W}(Y-Y_W) \quad (3)$$

Conservation of energy for the absorbent:

$$\frac{\partial t_w}{\partial\tau}+\omega\frac{\partial t_w}{\partial\varphi}-\frac{\lambda}{\rho_w(C_{pw}+WC_{pl})}\times\left[\frac{1}{r}\frac{\partial^2 t_w}{\partial\varphi^2}+\frac{\partial^2 t_w}{\partial Z^2}\right]= \frac{1}{\left[\rho_w(C_{pw}+WC_{pl})\right]}\left[A_f\cdot\alpha\cdot f_v(t-t_w)+K_Y f_v(Y-Y_w)Q\right] \quad (4)$$

Momentum equation for the process air:

$$\frac{\partial\left(\rho_g \cdot u\right)}{\partial \tau}+\frac{\partial\left(\rho_g \cdot u^2\right)}{\partial z}=\frac{\Delta P}{L} \tag{5}$$

RESULTS AND DISCUSSION

Discussion of the effects of some parameters

At zero angle, for all length of desiccant it is assumed that the humidity ratio is equal to the inlet humidity ratio to the adsorption section it is equal to 0.0141 kg/kg dry air.

At the φ>0, the humidity ratio decreases when the length of desiccant increases. The moisture content of air in earlier angels decreases because the difference of the moisture potential between air and desiccant decreases. As a result, the mass transfer from the air to the desiccant decreases in earlier angles as well. Therefore, at the last angles, the humidity ratio decreases less than that at the first angles. The humidity ratio of the air is increased with angle, this is because of the increasing contact time of the bed and the hot air (the regeneration air), therefore the bed becomes warmer and the evaporation rate increased.

Performance analysis

The performance of a solid desiccant regenerative enthalpy exchanger is not only a function of the geometric matrix design but also of the thermo-physical equilibrium properties of the desiccant. The dehumidification efficiency or dehumidifier performance (η) is defined as the ration of the humidity difference between the inlet and outlet of the adsorption process to that at the optimum rotational speed as it shows on Eq. (6).

$$\eta=\frac{Y_{in}-\int_0^{180} Y(\varphi)\,d\varphi}{Y_{in}-\int_0^{180} Y_{opt}(\varphi)\,d\varphi} \tag{6}$$

Fig. 2 shows the dehumidification efficiency versus rotational speed. This figure shows that at the optimum rational speed η is equal to 100% and the half of the optimum rotational speed (over adsorption speed), η = 98.64% and at twice of the optimum rotational speed (under adsorption case), η = 99.25%.

Air stream velocity is a parameter that is referenced in thermal process and we obtain it in this model. Any fluid that enters a channel undergoes quite a change on its way downstream. It is evident that the only exact way to characterize such a flow is as velocity field. That type of

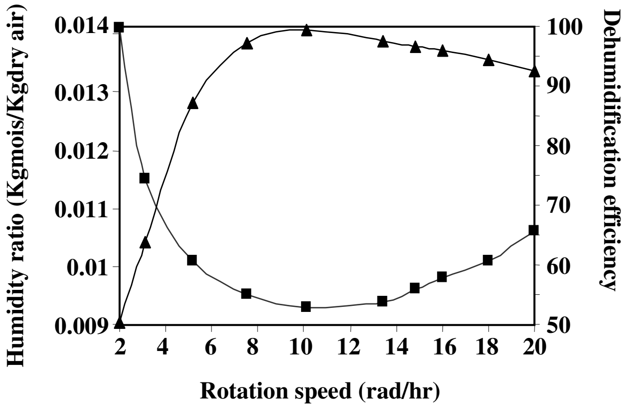

Fig. 2: The average outlet humidity ratio of the adsorption section (■) and dehumidification efficiency (▲) versus rotational speed.

specification is, of course, complex but computers can easily mange this problem. According to Fig. 3-A, the more resident time the air is given, the more heat and mass transfer is allowed to take place. Therefore, it was observed that as the velocity of the air flow increases, sensible, latent, and total efficiencies declines. Ackermann heat transfer correction factor is another parameter that is changes with humidity ratio. Fig. 3-C shows the average outlet humidity ratio of the adsorption section versus Ackermann correction factor. More humidity ratio, bigger Ackermann correction factor was obtained at the optimal rotational speed (ω=10), Ackermann correction factor has significant effect on performance efficiency.

Fig. 3-B shows the changes of the air stream velocity are measured by the measurement devices in the output section of the wheel. In this figure, it is clear that the air outlet velocity of the wheel is increased towards the entrance and the exit section in the wheel has decreased with time. Because of the air density is a function of absolute humidity and air temperature and also the process air velocity is a function of air flow density, temperature and pressure drop along the channel depth so that in low pressure drops, the process air velocity will be only a function of air flow density and temperature.

Humidity ratio or absolute humidity of process air flow channels in each level increase during the time and so because at that level and after a certain time or period the bed is become saturated so that the bed does not absorb moisture and finally the humidity ratio is become too high during the time.

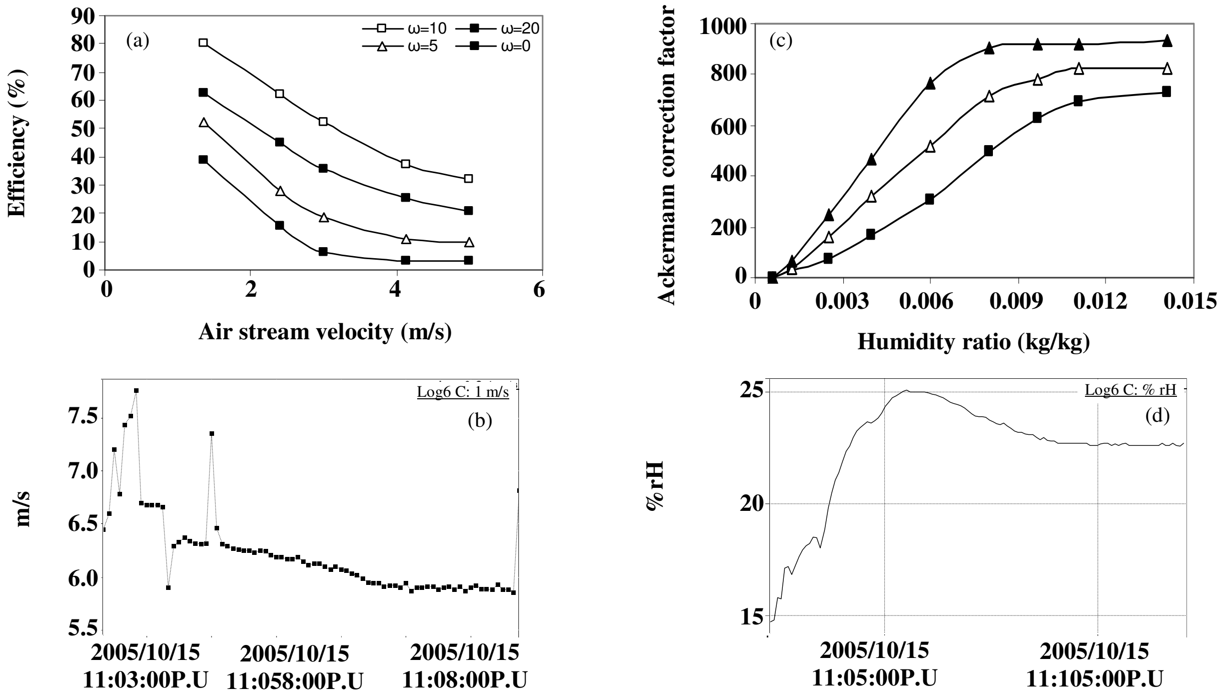

Fig. 3: (a) Rotary wheel total enthalpy exchanger efficiency versus air stream velocity. (b) Experimental results of the average outlet air stream velocity of the adsorption section versus time. (c) The average outlet humidity ratio of the adsorption section versus Ackermann correction factor (series: ▲, ω=4 rad h^{-1}; Δ, ω=10 rad h^{-1}; ■, ω=20 rad h^{-1}). (d) Experimental results of the average outlet air relative humidity of the adsorption section versus time.

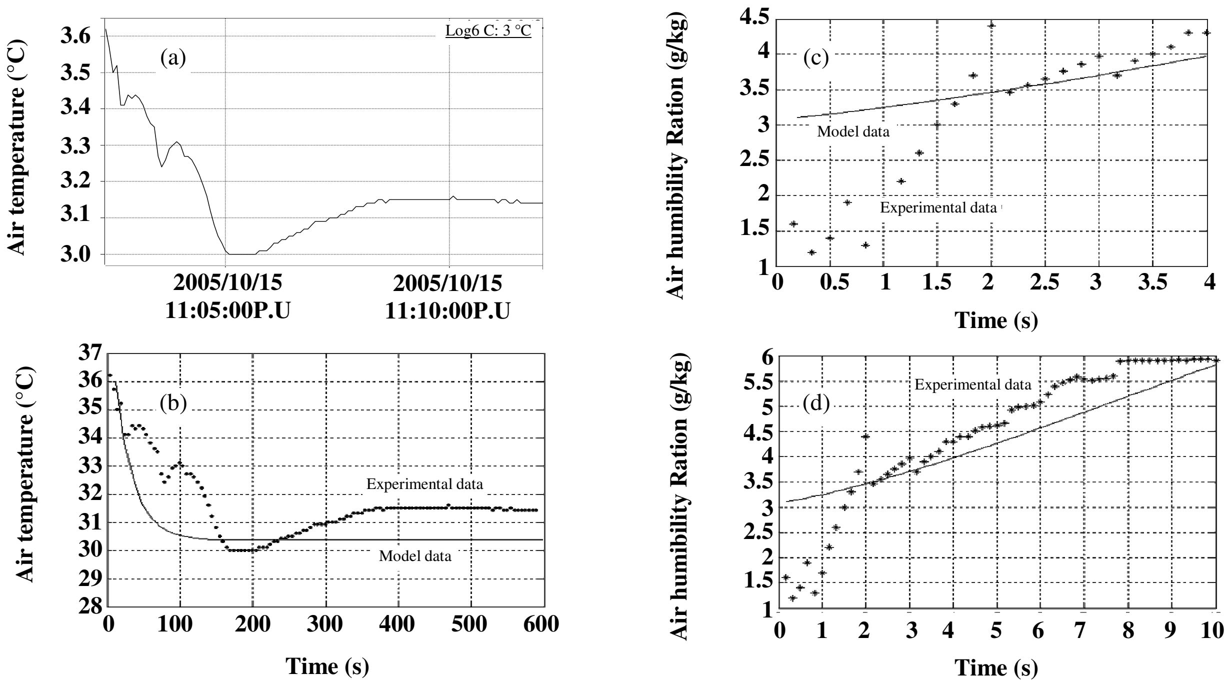

Fig. 4: (a) Experimental results of the average outlet air temperature of the adsorption section versus time. (b) Experimental and theoretical model results of the average outlet air temperature of the adsorption section versus time. (c) Experimental and theoretical model results of the average outlet air humidity ratio of the adsorption section versus time. (d) Experimental and theoretical model results of the average outlet air relative humidity of the adsorption section versus time until 10 minutes.

Table 2. Rotary Wheel Total Enthalpy Exchanger Model Summary.

Parameter	Change in parameter	Change in total efficiency
Channel Depth	0.250 m	39.3
Desiccant Thickness	0.175 mm	31.5
Substrate Thickness	0.254 mm	13.7
Air Stream velocity	4.0 m/s	98.5

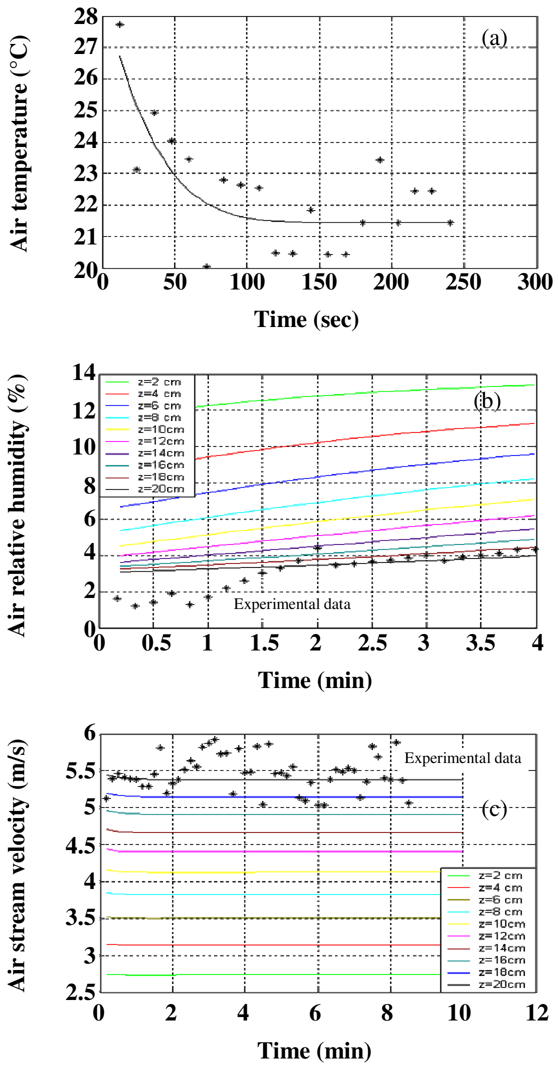

Fig. 5: (a) Experimental and theoretical model results of the average outlet air temperature of the adsorption section versus time. (b) Experimental and theoretical model results of the average outlet air relative humidity in different parts of the adsorption section versus time until 4 minutes (c) Experimental and theoretical model results of the average outlet air stream velosity in different parts of the adsorption section versus time until 12 minutes.

The results of this model runs are summarized in Table 2. Overall, the program results produced expected trends. In fact, the latent efficiency was consistently an order of magnitude lower than the sensible efficiency. This issue, along with the indication that increased surface area increased the wheel performance, led to the selection of a porous desiccant for the rotary wheel application.

Finally, the rotary wheel total enthalpy exchanger (desiccant wheel) response to changing air stream velocities was determined. Changes were made to the performance of the wheel by increasing and decreasing the speed at which both the supply and exhaust air passed through the channels of the dehumidifier. The operating air speed reflects the amount of time the air is allowed to be conditioned within the rotary wheel.

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