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Optimization and performance analysis of dehumidification rotating wheel using solid desiccant

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Abstract

In the present work, theoretical and experimental evaluation of the effect of solid desiccant wheel and operation conditions on the performance of desiccant dehumidification system has been carried out. a new model for desiccant wheel has been presented. In the theoretical part of this study, a mathematical model has been developed where its output results are compared with the experimental data. The effect of different design parameters and operating conditions on the adsorption and the regeneration processes is discussed. The effect of regeneration air temperature, the process air and regeneration air inlet humidity, the rotational speed, the process and regeneration air velocity (or flow rate), the bed length, performance analysis, etc. on the amount of water adsorbed/desorbed in a cycle is investigated.

NOMENCLATURE

A_f	Ackermann heat transfer correction factor
C_p	specific heat ($J \cdot kg^{-1} \cdot K^{-1}$)
D_{eff}	effective diffusivity of desiccant ($m^2 \cdot s^{-1}$)
D_0	effective diffusivity of desiccant at standard condition ($m^2 \cdot s^{-1}$)
f_V	ratio of desiccant surface area to volume ($m^2 \cdot m^{-3}$)
f_S	ratio of free flow area to section area of rotary wheel
J	mass flux ($kg \cdot m^{-2} \cdot s^{-1}$)
K_Y	coefficient of mass convection($kg \cdot m^{-2} \cdot s^{-1}$)
L	thickness of the desiccant matrix (m)
m_i	mass flow of air per unit section area of rotary wheel ($kg \cdot m^{-2} \cdot s^{-1}$)
P_{atm}	atmospheric pressure (Pa)
Q	adsorption heat ($J \cdot kg^{-1}_{water}$)
r	radius of the rotary wheel (m)

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t temperature of air (°C)
t_w temperature of desiccant (°C)
u Air stream velocity($m.s^{-1}$)
W water content of desiccant ($kg_{water} \cdot kg^{-1}_{adsorbent}$)
Y humidity ratio ($kg_{moisture} \cdot kg^{-1}_{dry\ air}$)
Y_w humidity ratio near the wall of desiccant ($kg_{moisture} \cdot kg^{-1}_{dry\ air}$)

Greek symbols
α coefficient of heat transfer ($W \cdot m^{-2} \cdot K^{-1}$)
ρ_{da} dry air density ($kg \cdot m^{-3}$)
ρ_g wet air density ($kg \cdot m^{-3}$)
ρ_w water density ($kg \cdot m^{-3}$)
η dehumidifier performance
λ thermal conductivity of desiccant ($W \cdot m^{-1} \cdot K^{-1}$)
τ time (sec)
ω rotation speed (sec^{-1})
r, φ, z polar coordinates
φ_R angle of regeneration section

Introduction

Many mathematical models on the rotary desiccant dehumidifier have been proposed. The effect of the rotational speed on the performance of an adiabatic rotary dehumidifier has been parametrically studied, and the optimal rotational speed determined by examining the outlet adsorption side humidity profiles and humidity wave front inside the desiccant dehumidifier (Zheng and Worek 1993; Zheng *et al.*, 1995). The mathematical model of a rotary desiccant wheel has been applied to calculate the performance of stationary or rotary bed and transient or steady state operation by considering some of the key components. Predicting the performance and evaluating the benefits of rotary desiccant wheels concerning the complicated heat and mass transfer in the rotary desiccant matrix has been also suggested (Zhang *et al.*, 1996). Recently, 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). A simple 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).

Input wet air or process air is dried in a rotary dehumidification using solid desiccant particles that is a means of dehumidification of indoor air. The humidity amount on the surface of desiccant is increased during dehumidification process and is adsorbed by regeneration air stream which is ultimately exhausted to ambient.

In this study, also the effect of Ackermann heat transfer correction factor was investigated by using mathematical modeling of solid desiccant wheel as well as mass, energy and momentum balances on air and wet solid particles of desiccant for process and regeneration air stream. The results indicated that dehumidification rate along with desiccant wheel is depended on humidity ratio, air velocity, mass and heat transfer from air stream to desiccant bed and Ackermann correction factor. This mathematical model is also capable to depict the details of superficial humidity, temperature of air stream into desiccant wheel channels in both adsorption and regeneration parts as periodic profiles.

TWO DIMENSIONAL ROTARY WHEEL MODEL

In a rotary wheel, heat is transferred from the hot fluid to a solid energy carrier (the matrix) during 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. Figure 1 shows schematic coordination diagram of desiccant wheel.

The physics of this mechanism dictate a close look into the surface phenomena taking place as the wheel rotates between contrasting conditions transferring heat and moisture. With regenerative surface effects of this kind, a lumped capacitance assumption can be used. This assumption allows for the rotary wheel total enthalpy exchanger model to be described by a two-dimensional model solution. With lumped capacitance effects occurring at the desiccant material surface, a two-dimensional solution can be formulated considering temperature and moisture effects along the length of the wheel channels.

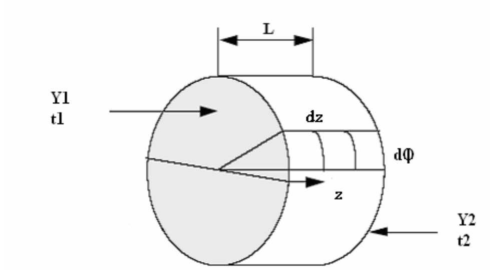

Figure 1.matrix of a rotary desiccant wheel

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. Assumptions to obtain Equations 1-4 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:

$$\frac{\partial(r_{da}.Y)}{\partial t} + w\frac{\partial(r_{da}.Y)}{\partial j} + \frac{\partial(r_{da}.Y.u)}{\partial z} = K_Y.f_v(Y_w - Y) \tag{1}$$

Conservation of energy for the process air:

$$\frac{\partial(r_{da}C_{Pe}t)}{\partial t} + w\frac{\partial(r_{da}C_{Pe}t)}{\partial j} + \frac{\partial(r_{da}uC_{Pe}t)}{\partial z} = A_f.a.f_v(t_w - t) + K_Y.f_v(Y_w - Y)C_{pv}.t \tag{2}$$

Conservation of water content for the absorbent:

$$\frac{\partial W}{\partial t} + w\frac{\partial W}{\partial j} - D_{eff}\left[\frac{1}{r}\frac{\partial^2 W}{\partial j^2} + \frac{\partial^2 W}{\partial Z^2}\right] = \frac{K_Y.f_v}{r_W}(Y - Y_W) \tag{3}$$

Conservation of energy for the absorbent:

$$\frac{\partial t_w}{\partial t} + w\frac{\partial t_w}{\partial j} - \frac{l}{r_w(C_{pw} + WC_{pl})}\cdot\left[\frac{1}{r}\frac{\partial^2 t_w}{\partial j^2} + \frac{\partial^2 t_w}{\partial z^2}\right] = \frac{1}{[r_w(C_{pw} + WC_{pl})]}[A_f.a.f_v(t - t_w) + K_Y f_v(Y - Y_w)Q] \tag{4}$$

Momentum equation for the process air:

$$\frac{\partial(r_g.u)}{\partial t}+\frac{\partial(r_g.u^2)}{\partial z}=\frac{\Delta P}{L} \quad (5)$$

The Ackerman heat transfer correction factor for mass transfer fluxes in phase j (Winkelman *et al.*, 1992) :

$$A_f=\frac{C_f}{e^{C_f}-1} \quad (6)$$

$$C_f=\frac{\sum_i J_{i,j}C_{p_{i,j}}}{h_j} \quad \text{(j=g or l)} \quad (7)$$

For air stream process :

$$C_f=\frac{J.C_{pa}}{h} \quad (8)$$

$$J=K_Y\,(Y_w-Y) \quad (9)$$

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 $\varphi>0$, 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. This figure also indicates that the air humidity increased with bed depth with a decreasing slope, because the air get closer to the saturation state as passes through the bed, hence the rate of evaporation decreases.

In adsorption section, in earlier angles, the air temperature cannot be predicted because the air from previous section (regeneration section) is leaving out if the desiccant's pores and it has its temperature. The air temperature with length of the bed increases ,the slope of this increasing in earlier angels is higher than that in later angles .This difference is because of the decreasing of the desiccant temperature and then the temperature driving force between desiccant and air decreases , therefore the heat transfer rate diminished.

Optimum rotational speed of desiccant wheel

Optimum rotational speed of wheel occurs when that average humidity ratio in outlet air of adsorption section is minimum, because of dehumidification. Optimum rotational speed will be calculated with the variation of the outlet humidity profile, for different rotational speed. Figure 2 shows the average outlet humidity ratio of the adsorption section versus rotational speed.

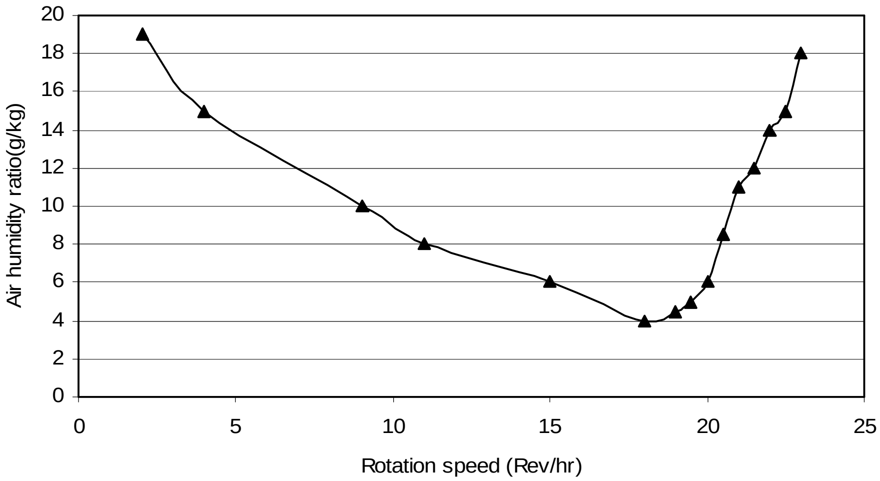

Figure 2: average outlet humidity ratio of the adsorption section versus rotational speed

Optimum rotational speed is related also to regeneration section. Therefore, the average outlet humidity ratio of the regeneration section should be calculated with the variation of the rotational speed. This is shown in figure 3.

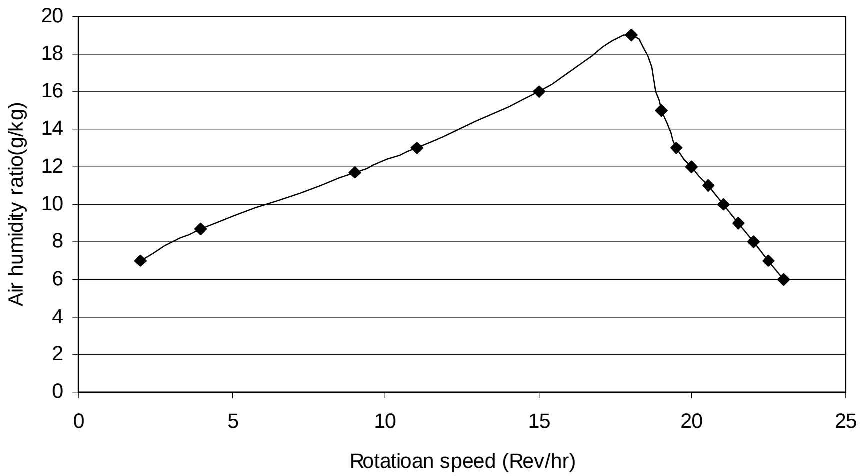

Figure 3: average outlet humidity ratio of the regeneration section versus rotational speed

For optimization of the desiccant wheel, first of all these two curves in figure 1 and figure 2 should be described as correlations with use of curve fitting. According to this method, equations are defined by the following relations:

For regeneration section:

$$Y = -0.008\omega^3 + 0.2197\omega^2 - 0.9718\omega + 8.642 \qquad (10)$$

For dehumidification section:

$$Y = 0.0102\omega^3 - 0.275\omega^2 + 0.9278\omega + 16.88 \qquad (11)$$

In a desiccant wheel rotational speed has inverse relationship with time of adsorption or regeneration:

$$j = 2pwt \qquad (12)$$

If φ_{id} be the angle of adsorption section, so wheel rotational speed is follow:

$$w = \frac{j_{id}}{2pt_{ad}} \qquad (13)$$

When wheel is rotating in a different rotational speed the performance of dehumidifier is become low. Adsorption and regeneration in a desiccant wheel with the same and optimum rotational speed are maximum and minimum respectively. Humidity ratio and temperature of outlet air in every section of desiccant wheel will be calculated as follows:

If $0 \le j \prec 2pa$:

$$t_{out} = \frac{1}{2pa}\int_0^{2pa} t(t, j, L)\,dj \qquad (14)$$

$$Y_{out} = \frac{1}{2pa}\int_0^{2pa} Y(t, j, L)\,dj \qquad (15)$$

If $2pa \le j \prec 2p$:

$$t_{out} = \frac{1}{2p(1-a)}\int_{2pa}^{2p} t(t, j, 0)\,dj \qquad (16)$$

$$Y_{out} = \frac{1}{2p(1-a)}\int_{2pa}^{2p} Y(t, j, 0)\,dj \qquad (17)$$

α is the angle fraction of adsorption section so 1-α will be the angle fraction of regeneration section. Coefficient of performance or COP of a desiccant wheel is the ratio of difference between humidity ratio in inlet and outlet for adsorption process in optimum rotational speed per inlet humidity ratio my be written as follow:

$$COP = \frac{Y_{ad,in} - \int_0^{2pa} Y_{out,opt}(t,j,L)\,dj}{Y_{ad,in}} \tag{18}$$

With replace of equation 11 in 18, it will be written as follow:

$$COP = \frac{Y_1 - (0.0102 w^3 - 0.275 w^2 + 0.9278 w + 16.88)}{Y_1} \tag{19}$$

In this study, silica gel is chosen as desiccant and equations 1-5 as followed are basic relations and also several auxiliary equations are appended meanwhile. FORTRAN version 90 is used for solving the simulation method. The calculation conditions of the base case are listed in the table 1.
Figure 4 shows the variation of coefficient of performance versus rotation speed and in figure 5, COP versus length of wheel in different rotation speed.
According to figures 4 recognizes that in optimum rotational speed COP is equal to %90 and in the half* of that COP is %68 and in more than rotational speed† for example in ω= 20 Rev/hr COP= %80.

Table 1: input parameters for simulation program

parameter	unit	amount
L	Cm	20
f_v=4/de	m^2/m^3	1333
$_{ad}\varphi$	Rad	π
$_{re}\varphi$	Rad	π
P_a	kPa	101.33
$u_{0,ad}$	m/s	1.512
$u_{in,ad}$	m/s	1.512
$u_{0,re}$	m/s	1.512
$u_{in,re}$	m/s	1.512
RH_a	%	50
t_a	C°	25
$RH_{in,ad}$	%	90
$t_{in,ad}$	C°	35
$RH_{in,re}$	%	3.5
$t_{in,re}$	C°	88
$_w\rho$	kg/m^3	720.06

* . Over adsorption

† . Under adsorption

CP_w	J/kg°K	921
CP_a	J/kg°K	1007.6
CP_v	J/kg°K	1756.6
CP_l	J/kg°K	6.4186
λ	w/m°K	4417.0
D_0	m^2/s	10^{-6} Ï 8.0
μ	kg/m.s	10^{-5} Ï 8.1

Figure 5 shows variations of COP for desiccant wheel with different lengths. With increasing length of wheel the coefficient of performance is risen, but from a special length this trend is changed and COP is decreased. Figure 5 also shows in different rotational speed. When the rotational speed is equal to18 rev/hr the maximum COP of the wheel is obtained with a length of about 32 cm. Meanwhile these results are presented with the simulation program.

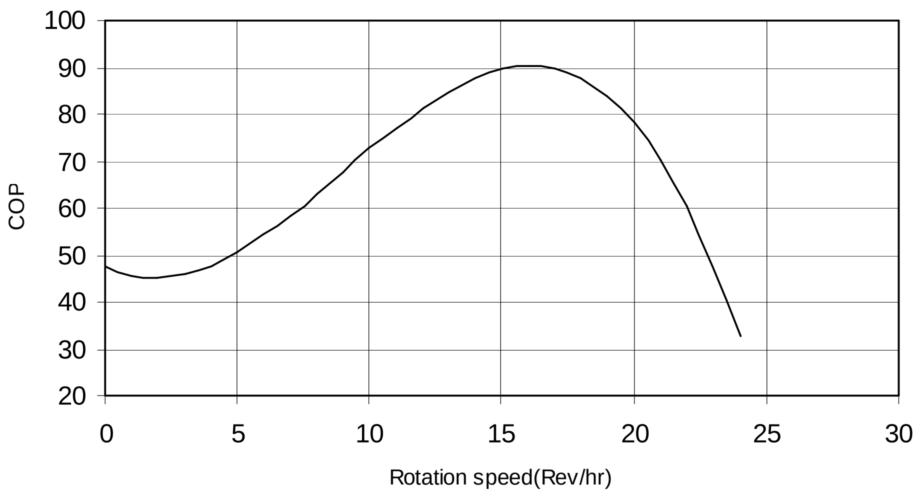

Figure 4: Coefficient of performance versus rotation speed

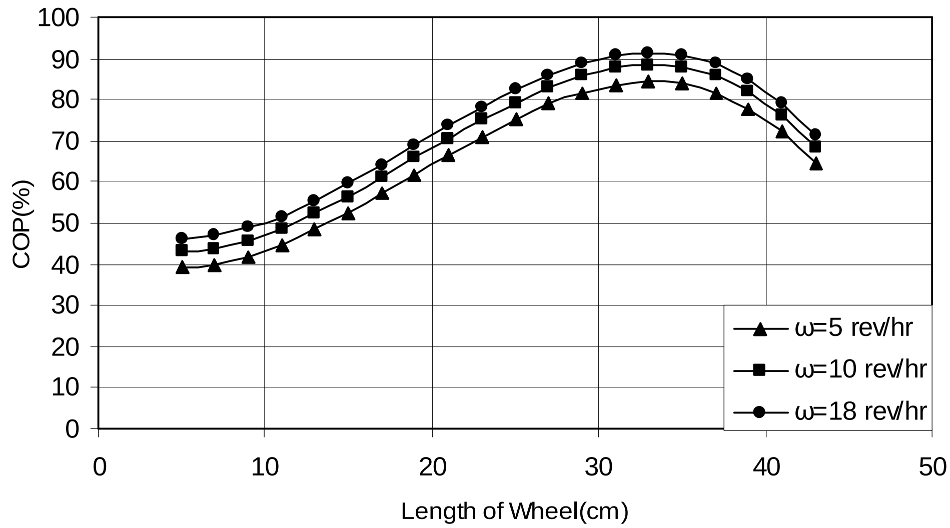

Figure 5: Coefficient of performance versus length of wheel in different rotation speed

Finally, Changes were made to the performance of the wheel by increasing and decreasing the speed at which both the adsorption and regeneration air passed through the channels of the desiccant dehumidifier. The operating air speed reflects the amount of time the air is allowed to be conditioned within the rotary wheel.

Performance analysis

The performance of a solid desiccant dehumidifier is not only a function of the geometric matrix design but also of the thermo-physical equilibrium properties of the desiccant. These properties include the adsorption isotherms for water vapor, heat of adsorption and the thermal capacity of the dry solid desiccant. Two kinds of adsorption exist, one is physical adsorption, the other is chemisorptions.
Physical adsorption is a reversible process and occurs when relatively weak intermolecular forces cause the adsorption. Chemisorptions involves a chemical reaction between the adsorbing and the adsorbed molecules, and the process is generally irreversible. However, the effect of chemisorptions on the enthalpy exchanger performance is negligible, and therefore it is not dealt with in this study. 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 below:

$$h = \frac{Y_{in} - \int_0^{180} Y(j)\,dj}{Y_{in} - \int_0^{180} Y_{opt}(j)\,dj}$$

In both the rotary wheel total enthalpy exchanger and fixed plate total enthalpy exchanger models, the exiting supply air temperatures and humidity ratios were used along with supply and exhaust air stream boundary conditions to determine exchanger efficiencies.

Three efficiencies were used to evaluate an exchanger's overall steady-state performance: sensible efficiency, latent efficiency, and total efficiency. The sensible efficiency of the system describes the exchanger's ability to remove heat from one air stream and transfer it to another. The sensible efficiencies for all numerical models evaluated in this project were determined by the sensible efficiency of a total enthalpy exchange system that represented as the difference in the inlet and exit temperatures of the supply air.

The latent efficiency of the system describes the exchanger's ability to remove moisture from one air stream and transfer it to another. The energy associated with the phase change between liquid and vapor states is called latent energy. The latent efficiency of a total enthalpy exchange system is represented as the difference in inlet and exit humidity ratios of the supply air. The total efficiency of the system describes the exchanger's ability to remove total sensible and latent energy from one air stream and transfer it to another. The total efficiency is represented as the difference in total enthalpy associated with the inlet and exit temperatures and humidity ratios of the supply air as compared to the total enthalpy associated with the inlet supply air and inlet exhaust air temperatures and humidity ratios. The thermal differences are converted into enthalpy changes by the multiplication of the temperature difference by the specific heat of air. The humidity ratios are multiplied by the enthalpy of vapor at the specified temperature to yield the enthalpy differences contributed by the humidity ratio differences.

Figure 6 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%. These results indicate that the system operation at the non-optimum conditions lead to deterioration of system performance.

When desiccant dehumidifier is operated at the non-optimum rotational speed, the dehumidification efficiency decreased. The dependence of the dehumidification performance on the wheel rotational speed can be easily explained by the results shown in figure6 which compares the variation of the outlet humidity profile for different rotational speed.

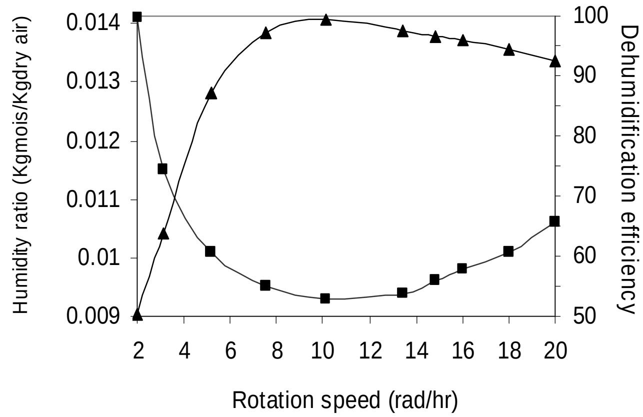

Figure 6. The average outlet humidity ratio of the adsorption section (■) and dehumidification efficiency (▲) versus rotational speed

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 specification is, of course, complex but computers can easily mange this problem. According to Figure 7, 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.

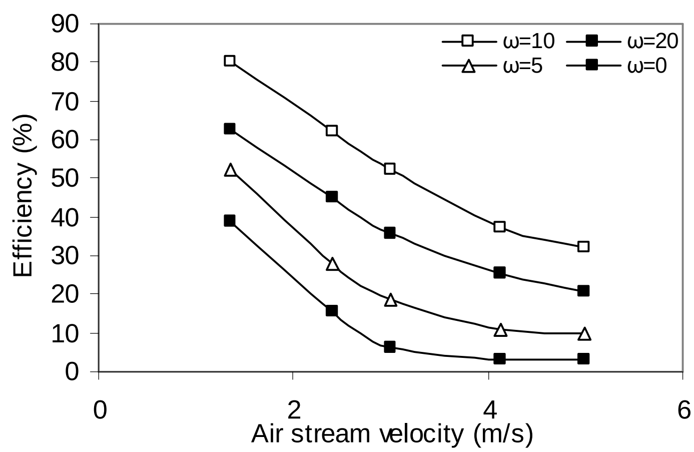

Figure7. Rotary wheel total enthalpy exchanger efficiency versus air stream velocity

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