EFFECT OF THE MASS DIFFUSION RESISTANCE IN ENTHALPY WHEELS FOR ENERGY RECOVERY

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Abstract. *The enthalpy wheels are heat and mass exchangers, honeycomb type, lightly impregnated of desiccant materia, which are used in adsorption air conditioning systems for energy recovery. The problem consists of a cylindrical wheel formed by a net of very small channels that rotate continuously with two counter air flows: process (adsorption) and exhaustion (desorption). The proposed mathematical model considers the mass diffusion resistance in the desiccant matrix. The system of equations were discretized by the finite-volume method in conjuction with a fully implicit scheme and staggered arrangement. The results are presented in terms of temperature and absolute humidity profiles in the adsorption and desorption sections. The results have been also compared with other existing models from the literature. From the several case studies, it was verified that the effect of the mass diffusion resistance change the amount of adsorbed and desorbed mass in the enthalpy wheel. Nevertheless, the difference verified between the model with and without resistance is small. Thus, we can neglect the mass diffusion resistance and employ the instantaneous equilibrium model which is computationally cheaper. The numerical results of the present work also showed an excellent agreement with the experimental data available in the literature.*

Keywords*: Air conditioning, Enthalpy wheels, Adsorption.*

1. INTRODUCTION

For global sustainable development, it is necessary to reduce the primary energy consumption and to introduce renewable energy. The reduction of primary energy could be performed with a reasonable utilization of waste thermal energy in a refrigeration process. In addition, the reasonable utilization of low temperature thermal energy would promote the introduction of renewable energy.

Most of the refrigeration and heat pump technologies are dominated by the vapor compressor systems. However, the vapor compressor systems are highly concerned with the environmental regulations, since many vapor compressor systems are still using CFCs or HCFCs, which are known as ozone depleting and global warming gases.

The current air conditioning systems, need one definitive outlet of exterior air introduced constantly and mixed to air already climatized to adjust to the temperature and humidity, to supply oxygen, to reduce odors of the body and others, to remove pollutant gases and to improve the quality of interior air. This supply of exterior air is determined in accordance with the type of application, number of people in the room, area of the room and conducted by the current law.

Conventional vapor-compression-based cooling systems are not able to cope with the current humidity standards required by the production and storage of humidity sensitive products. Currently, the interior air quality (IAQ) rigid standards made problematical the use of conventional air conditioning system, especially in extreme humidity conditions.

Moreover, a development of an independent-control technology of temperature and humidity is necessary from a standpoint of rational energy utilization and a comfortable environment control. The control of temperature and humidity independently using desiccant air conditioning system is possible. Desiccant air condition is also operated with relatively small amount of thermal energy. Therefore, it is recognized as a method for reducing the environmental pollution, and has received considerable attention.

Desiccant systems has several advantages relative to their closed-cycle counterparts: they are operated at ambient pressure; heat and mass transfer between the air and the desiccant take place by direct contact; both cooling and dehumidification of the conditioned air may be provided, in variable quantities, to adjust to, the load in the conditioned space. Disadvantages are the low COP, due to inherently inefficient regeneration; relatively large air volumes must be pumped, leading to potentially high parasitic losses; contamination of the desiccant by dirt and dust contained in the air can require its replacement after some period of operation.

Desiccant technology is now being applied in buildings and space where humidity levels are critical such as supermarket and cold food areas, hospital operating theatres, nursing homes, schools, hotels, convention centers and theatres. The technology is also applicable to buildings requiring a high fresh air intake in humid climate zones.

The major advantage of desiccant cooling is a significant potential for energy savings and reduced consumption of fossil fuels. The electrical energy requirement can be very low compared with conventional refrigeration systems. The source of thermal energy can be diverse (for instance, solar, waste heat, natural gas, among them).

Desiccant cooling systems work on the principles of desiccant dehumidification and evaporative cooling. A desiccant dehumidification system is a device, which contains an adsorbent to adsorb and desorb moisture in the process air and regeneration air, respectively. The core component of the dehumidification system is a rotary heat and moisture exchanger, which can be classified into enthalpy (energy) wheel and desiccant wheel. The classification is based on its function, operation condition and the wheel's configuration.

Another viable alternative for air conditioning systems is the use of hybrid equipment systems composites of responsible vapor compression for the load of sensible heat and responsible enthalpy wheels for the load of latent heat.

The desiccant wheels, basically consist of a rotary matrix composed of numerous corrugated channels, through which two process streams flow (usually in a counterflow arrangement), continuously exchange heat and moisture. The geometries of these channels resemble closely to sinusoidal functions. Although desiccant and enthalpy wheels are very similar devices, there is a fundamental difference on the proportions of heat and mass that they are designed to transfer: in desiccant wheels the main emphasis is the mass transfer, whereas enthalpy wheels are designed to transfer comparable fractions of heat and moisture.

The desiccant wheel rotates slowly to expose one portion of the desiccant material to the process air stream while the other portion is exposed to the regeneration air stream. A partition and flexible seals separate the process and regeneration air in the dehumidifier. Moist air enters the process side and passes ones the desiccant and is dehumidified. Regeneration occurs on the other side of the partition where heated air flows most often from the opposite direction, enters most often from the opposite direction, them passes over the desiccant, and finally laden with moisture is exhausted from the dehumidifier.

The enthalpy wheel rotates between the outside fresh air (process air) and the exhaust air from room. Heat and humidity would be recovered from the exhaust in winter and excess heat and moisture would be transferred to the exhaust to cool and dehumidify the process air in the summer.

In order to predict heat and moisture transfer in the enthalpy wheel, conservation equations for energy and mass need to be postulated. Convective heat and mass transfer are described by lumped coefficients, whereby heat and moisture transfer are coupled by the equilibrium condition of the desiccant material, the so called sorption isotherm. Conservation equations of heat and mass transfer, the sorption isotherm as well as thermodynamic state equations for air and desiccant material result in a complex, non-linear, differential algebraic system of equations.

The desiccant technology has being studied and implemented in diverse countries, such as the United States, China, India, Japan, Australia, Canada, and more recently in Brazil. The works using the desiccant technology are involved with adsorption refrigeration and desiccant air conditioning systems (Gurgel, 1994; Lima e*t al.*, 2002; Camargo and Ebinuma, 2005; Medeiros e*t al.*, 2006a; Medeiros e*t al.*, 2006b; Nóbrega and Brum, 2006). Pertinent literature to the enthalpy wheels can well be evaluated in (Klein *et al.*, 1990; Stiesch, 1994; Simonson, 1998; Zhang and Niu, 2002; Asiedu et al., 2004 and Sphaier and Worek, 2006).

In the present work, the performance of a desiccant air-conditioning system using an enthalpy wheel is investigated through a one-dimensional model with and without the effects of mass diffusion in conjunction the finite-volume method. The effect of the mass diffusion resistance in the enthalpy wheel was investigated by the addition of the mass diffusion equation to the grains of the wheel with a non-linear boundary condition on the surface of enthalpy wheel which is connected to the equation of mass conservation in the desiccant.

2. MATHEMATICAL MODEL

An enthalpy wheel is described as a rotating cylindrical porous-medium wheel of length L and diameter D with small channels whose walls are adhered with an adsorbent such as silica gel. The desiccant materials is usually dipped on a fibrous substrate, which is typically constructed in a matrix of alternating sinusoidal corrugated sheets and flat sheets forming channels for airflow. The rotary wheel revolves at a constant velocity ω and is cyclically exposed to two physically separated air streams. The two air streams are in a counterflow arrangement through the adsorption and exhaustion sections. A schematic of the enthalpy wheel together with one channel side view is shown in Fig. 1. The governing equations are written for simultaneous and coupled heat and moisture transfer in one channel rotates around the axis of the wheel.

The transport phenomena occurring in an enthalpy wheel are usually simplified by considering a single channel structure, ignoring radial and angular heat conduction between adjacent channels. An exact representation of the system can be extremely complex, requiring a prohibitive computational cost. In fact, the cross-sectional area of an elementary channel is relatively small, and the sorbent material is very thin. The Biot numbers (Bi) for both airflow channel and sorbent bed are close or small than 0.1. Therefore, the temperature and the mass gradient across the cross-section of desiccant and airflow channel are small. To avoid prohibitive costs of computation, it is reasonable to ignore the effect of heat and mass transfer across the thickness of desiccant and airflow channel. Consequently, a one-dimensional model is considered as an accurate method to describe the thermal conduction and diffusion that occur within the porous desiccant (Gao et al., 2005).

For convenience, the following simplifications are assumed for the mathematical model: one-dimensional model in humid air and the desiccant matrix; forced convection dominant; axial heat conduction and water vapor diffusion in the air are negligible; axial molecular diffusion within the desiccant is negligible; there are no radial temperature or moisture content gradients in the matrix; incompressible fluid; the laminar and fully developed into the channel; hysteresis in the sorption isotherm for the desiccant coating is neglected; the channels that make up the wheel are identical with constant heat and mass transfer surface areas; the physical of the fluids and matrix are constant; the channels are considered adiabatic and impermeable; the mass and heat transfer coefficient are constant; the carryover between two air flow neglected; the pressure drop along the channel is negligible, as a result of relative small temperatures the effects of radiation heat transfer is neglected; no chemical reaction takes place, nor there are any energy sources within the system.

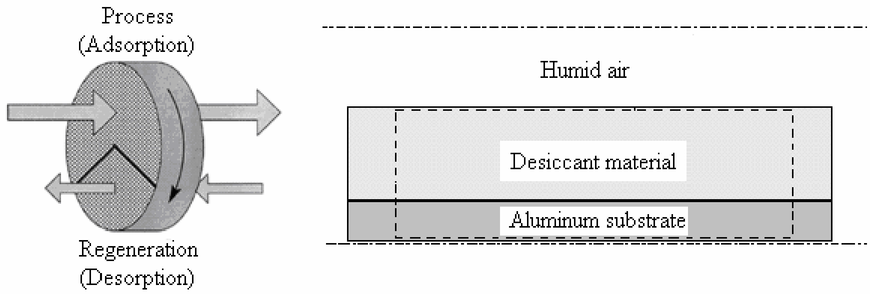


Figure 1. Schematic view of the enthalpy wheel and side view of one channel

2.1. Instantaneous equilibrium model

The governing equations for simultaneous heat and moisture transfer in the enthalpy wheels, based on the previous assumptions, Simonson, (1998) are the following:

Energy conservation equation between air stream and desiccant matrix

$$\left(\rho c_p A\right)_g \frac{\partial T_g}{\partial t} + U\left(\rho c_p A\right)_g \frac{\partial T_g}{\partial x} - \dot{m}' h_{ad}\eta + h\frac{A_s'}{L}\left(T_g - T_m\right) = 0 \quad (1)$$

Energy conservation equation on the desiccant matrix

$$\left(\rho c_p A\right)_m \frac{\partial T_m}{\partial t} - \dot{m}' h_{ad}\left(1-\eta\right) - \dot{m}' c_{p_w}\left(T_g - T_m\right) - h\frac{A_s'}{L}\left(T_g - T_m\right) = \frac{\partial}{\partial x}\left(k_{al} A_{al} \frac{\partial T_m}{\partial x}\right) \quad (2)$$

Mass conservation equation for the water vapor

$$A_g \frac{\partial \rho_v}{\partial t} + \frac{\partial}{\partial x}\left(\rho_v U A_g\right) + \dot{m}' = 0 \quad (3)$$

Mass conservation equation for the dry air

$$\frac{\partial \rho_a}{\partial t} + \frac{\partial}{\partial x}\left(\rho_a U\right) = 0 \quad (4)$$

Mass conservation equation for the desiccant matrix

$$\dot{m}' = \rho_{d,\mathrm{seco}} A_d \frac{\partial u}{\partial t} \quad (5)$$

During adsorption and desorption, the phase change rate of moisture ($\dot{m}'$) can be calculated by

$$\dot{m}' = h_m \frac{A_s^{'}}{L}\left(\rho_v - \rho_{v,m}\right) \tag{6}$$

where T_g, and T_m are respectively, the humid air temperature and the matrix temperature; ρ_v, $\rho_{v,m}$ and ρ_a are the vapor density in air, the water vapor density at the surface of matrix and dry air density; U denotes the air flow velocity and u the mass fraction of water in the desiccant; t time, x axial coordinate, L length of the enthalpy wheel, h the convective heat transfer coefficient, h_m convective mass transfer coefficient, h_{ad} heat of adsorption; η fraction of the phase change energy that is delivered directly to the air, ρ density, c_p specific heat, A cross-sectional area of one channel, A'_s heat and mass transfer surface area of one channel. In the equations, the subscripts "a", "al", "d", "g", "m", "v" e "w" denote respectively the dry air, aluminum, desiccant, gas (humid air), matrix (substrate +desiccant), water vapor and water.

The sorption isotherm that describes the equilibrium mass fraction of water in the desiccant is used to obtain the water vapor density on the desiccant surface as a function of moisture content and temperature. Since the desiccant is supposed to be type I isotherm, the isotherm can be described by the Langmuir equation, and using the fact that the water vapor density on the surface of the matrix is equal to the water vapor density in the air at equilibrium, the following equations are obtained:

$$u = \frac{f u_m}{1 - C + C/\varphi}; \quad \rho_{v,m} = \frac{P_{v,sat}\left(T_m\right)}{R_v T_m} \frac{C}{\left(\frac{f u_m}{u} - 1 + C\right)} \tag{7}$$

where, u_m is the empirical coefficient used in sorption isotherm describing maximum moisture absorbing capacity of desiccant; c is a constant describing the shape of the sorption curve and the relative humidity (φ).

The initial and boundary conditions for the problem are the supply and exhaust air stream at the inlet conditions:

Initial conditions

$$T_g(0,x) = T_0 \quad ; \quad T_m(0,x) = T_0 \quad ; \quad \rho_v(0,x) = \rho_0 \quad ; \quad u(0,x) = u_0 \tag{8}$$

Boundary conditions for process air

$$T_g(t,0) = T_{g_{p,in}}(t) \quad ; \quad \rho_v(t,0) = \rho_{v_{p,in}}(t) \quad ; \quad U(t,0) = U_{g_{p,in}}(t) \tag{9}$$

Boundary conditions for regeneration air

$$T_g(t,L) = T_{g_{e,in}}(t) \quad ; \quad \rho_v(t,L) = \rho_{v_{e,in}}(t) \quad ; \quad U(t,L) = U_{g_{e,in}}(t) \tag{10}$$

Boundary conditions for the desiccant matrix

$$\left.\frac{\partial T_m}{\partial x}\right|_{x=0} = \left.\frac{\partial T_m}{\partial x}\right|_{x=L} = 0 \quad ; \quad \left.\frac{\partial u}{\partial x}\right|_{x=0} = \left.\frac{\partial u}{\partial x}\right|_{x=L} = 0 \tag{11}$$

With the addition of the fundamental psychometric relationships for moisture air, thermodynamics and geometric relationships, the formulation of the desiccant problem is completed. Further details can be found in Simonson (1998).

2.2. Mass diffusion resistance model

The proposed mathematical model considers the mass diffusion resistance in the desiccant matrix. The gradient of humidity inside of the desiccant was modeled with the mass diffusion equation with a non-linear boundary condition in the surface of desiccant that was solved separately, but connected to the equation of conservation of mass in the desiccant matrix. In this model, it was admitted that the desiccant matrix is formed by a horizontal column for several volumes; each volume is represented by just one homogeneous desiccant spherical grain submitted to the humid air flow. A schematic of the desiccant wheel together with one grain side view is shown in Figure 2.

2.2.1 Grain model

The balance of mass in a desiccant grain was express by the following equation:

$$\frac{\partial u}{\partial t} = \frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2 D_{ef}\frac{\partial u}{\partial r}\right) \tag{12}$$

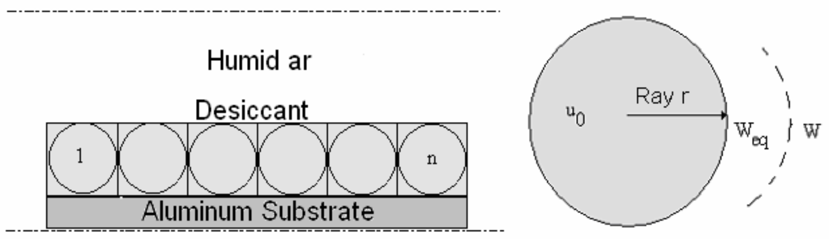


Figure 2. Schematic view of the enthalpy wheel and side view of one grain

Initial conditions

$$u(r, t = 0) = u_0 \tag{13}$$

Boundary conditions for the process air

$$\left.\frac{\partial u}{\partial r}\right|_{r=0} = 0 \tag{14}$$

$$-\rho_P D_{ef}\left.\frac{\partial u}{\partial r}\right|_{r=R} = h_m\left(w_{eq} - w\right) \tag{15}$$

Where the psychometric basic relation is given by:

$$w_{eq} = f(\varphi, T) = \frac{0.62198\varphi}{\left(P_{atm}/P_{ws} - \varphi\right)} \tag{16}$$

Using the adsorption isotherm, reorganizing the Eq. (7) and expressing φ as a function of f(u), we obtain

$$\varphi = \frac{uC}{fu_m - u(1-C)} \tag{17}$$

The psychometric basic relation can be written as

$$w_{eq} = f(u, T) = \frac{C_a\, u}{\left(C_b - C_c u\right)} \tag{18}$$

The next step was the linearization of the weq term using the series of Taylor. Further details can be found in Medeiros (2007).

3. NUMERICAL TREATMENT

The governing equations (1)-(5), subject to the initial and boundary conditions (8)-(11), were solved numerically by the finite-volume method (Patankar, 1980; Maliska, 1995) using a staggered variable arrangement. The velocity field is solved at the interfaces of each volume control and all the remaining dependent variables and properties are solved at the nodes. The upwind differencing scheme (UDS) is used for the air and the central differencing scheme (CDS) is used for the matrix. The linear systems were solved by Successive Over Relaxation (SOR) and, to speed up convergence, the energy equation in the matrix is solved through the Tridiagonal Matrix Algorithm (Patankar, 1980).

The iterative process to obtain the solution implied in the following steps:

1. Estimate the parameters and properties.
2. Solve Eqs. (1) to (5), respectively for the matrix temperature, water vapor density, air velocity, water fraction in matrix and air temperature in a segregated way.
3. Update the rate of phase change through Eq. (6).
4. Return to step 2 until a converged solution is reached. Then, the solver goes to next time step and repeat the procedure again until a quasi steady solution is reached.

The following convergence criterion was used to test the convergence of each time-step:

$$\sum_{i=1}^{n}\left(\chi_i^{j+1}-\chi_i^{j}\right)\Big/\left(n\left|\chi_{\max}-\chi_{\min}\right|\right)\leq 1x10^{-4} \tag{19}$$

where, χ is the estimated properties in the governing equations.

The steady-state solution is reached when the difference between the mass balance and energy are neglected for a complete cycle. To check if the steady-steady solution is obtained, the following criteria suggested by Simonson (1998) was used:

$$\left|\frac{\dot{m}_{pr}\left(W_{pr,in}-W_{pr,out}\right)-\dot{m}_{ex}\left(W_{ex,out}-W_{ex,in}\right)}{\dot{m}_{\min}\left(W_{pr,in}-W_{ex,in}\right)}\right|\leq 5x10^{-3}\ ;\left|\frac{\dot{m}_{pr}\left(H_{pr,in}-H_{pr,out}\right)-\dot{m}_{ex}\left(H_{ex,out}-H_{ex,in}\right)}{\dot{m}_{\min}\left(H_{pr,in}-H_{ex,in}\right)}\right|\leq 5x10^{-3} \tag{20}$$

where, W and H are respectively, the absolute humidity and the total enthalpy.

The effectiveness of the wheel can be calculated for the steady state regime (Klein et al, 1990; Simonson, 1998).

$$\begin{gathered}\varepsilon_s=\frac{\dot{m}_{pr}\left(T_{pr,in}-T_{pr,out}\right)+\dot{m}_{ex}\left(T_{ex,out}-T_{ex,in}\right)}{2\dot{m}_{\min}\left(T_{pr,in}-T_{ex,in}\right)}\ ;\varepsilon_l=\frac{\dot{m}_{pr}\left(W_{pr,in}-W_{pr,out}\right)+\dot{m}_{ex}\left(W_{ex,out}-W_{ex,in}\right)}{2\dot{m}_{\min}\left(W_{pr,in}-W_{ex,in}\right)};\\ \varepsilon_t=\frac{\dot{m}_{pr}\left(H_{pr,in}-H_{pr,out}\right)+\dot{m}_{ex}\left(H_{ex,out}-H_{ex,in}\right)}{2\dot{m}_{\min}\left(H_{pr,in}-H_{ex,in}\right)}\end{gathered} \tag{21}$$

4. RESULTS AND DISCUSSION

The results for the models with and without mass diffusion resistance are now presented.

4.1. Instantaneous equilibrium model

The instantaneous equilibrium model are compared with the results of Simonson (1998) and Zhang and Niu (2002).

4.1.1 Model validation – Simonson (1998)

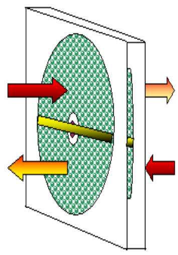
The parameters of the wheel and properties of the desiccant matrix are detailed here: the desiccant material is synthetic zeolites; the geometry of the channels in the wheel is sinusoidal with base of 3.62 mm and a height of 1.81 mm; the length L is 0.1 m and diameter D is 0.9 m of the wheel; the desiccant thickness is 0.044 mm and substrate thickness is 0.022 mm; the properties of the matrix (desiccant +substrate) had been ρ_d=350 Kg/m^3, Cp_d=600 J/kg K, m_d= 11.0 kg desiccant matrix, Cp_{al}=903 J/(kg K), ρ_{al}=2702 kg/m^3, k_{al}=237 W/(m K), σ_d=0.66 (fraction of the desiccant in the matrix); wheel rotational speed is 20 rpm; the sorption isotherm coefficients are C=20, f=1.0, u_{max}=0.2 kg water vapor / kg desiccant. Two cases studies were perfomed: a hot and cold test.

The following data were used for the hot test, which is a balanced configuration, the inlet temperature of the process air (35.0 ^{0}C), the humidity the desiccant wheel (25.2 g/kg); exhaustion air, 24.0 ^{0}C, 7.4 g/kg; the initial conditions (29.5 ^{0}C and 19.1 g/kg); mass airflow rates for both the adsorption and desorption sections (0.5 kg/s).

In Figure 3, the average properties in the inlet and outlet of the sections of the desiccant wheel are presented. From this figure, it is possible to observe the reduction of the temperature and humidity in the outlet of the adsorption section and the increase of humidity and the temperature in the outlet of the desorption section.

$T_{pr,in}$=35.0 ^{0}C
$W_{pr,in}$=25.2 g/kg
$H_{pr,in}$=98.5 kJ/kg

$T_{ex,out}$=31.2 ^{0}C
$W_{ex,out}$=18.9 g/kg
$H_{ex,out}$=78.7 kJ/kg



$T_{pr,out}$=27.7 ^{0}C
$W_{pr,out}$=13.7 g/kg
$H_{pr,out}$=62.1 kJ/kg

$T_{ex,in}$=24.0 ^{0}C
$W_{ex,in}$=7.4 g/kg
$H_{ex,in}$=42.6 kJ/kg

Figure 3. Hot test in the enthalpy wheel

Table 1 compares the results of present work with those obtained by Simonson (1998) in terms of effectiveness in function of the volumetric flows and speeds associates in the adsorption and desorption sections. The same conditions for the problem shown in the Figure 3 were imposed in the inlet and outlet of the wheel. From Table 1, it can be observed the results of the present work shown an excellent agreement with those obtained by Simonson (1998).

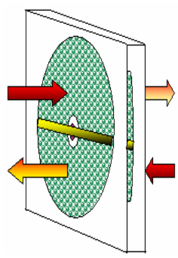
Table 1. Effectiveness of an enthalpy wheel, hot test

Rotation	Flow	Speed	Simonson (1998)			Present work		
20 rpm	(m^3/h)	(m/s)	Sensible	Latent	Total	Sensible	Latent	Total
Process	1635	1.68	66 %	65 %	65 %	66 %	65 %	65 %
Exhaustion	1533	1.58						

For the cold test, which is also a balanced configuration, the following parameters were used: inlet temperature of process air (5.0 ^{0}C), humidity of the desiccant wheel (1.1 g/kg); exhaustion (air, 20.0 ^{0}C, 5.8 g/kg); the initial conditions (12.5 ^{0}C and 1.8 g/kg); mass airflow rates for both the adsorption and desorption sections (0.5 kg/s).

$T_{pr,in}$=5.0 ^{0}C
$W_{pr,in}$=1.1 g/kg
$H_{pr,in}$= 7.7 kJ/kg

$T_{ex,out}$=9.4 ^{0}C
$W_{ex,out}$=2.2 g/kg
$H_{ex,out}$=14.9 kJ/kg



$T_{pr,out}$=15.6 ^{0}C
$W_{pr,out}$=4.7 g/kg
$H_{pr,out}$=27.3 kJ/kg

$T_{ex,in}$=20.0 ^{0}C
$W_{ex,in}$=5.8 g/kg
$H_{ex,in}$=34.5 kJ/kg

Figure 4. Cold test in the enthalpy wheel

In Figure 4, it can be observed, an inverse behavior to that it was described for the hot test. Therefore, in the outlet of the process it will occur the desorption and in the exhaustion exit the adsorption will occur.

Table 2 presents the effectiveness of enthalpy wheel for the cold test. The same data used for the results presented in Fig. 4 were used. From table 2, it can be verified that the results of the present work are identical to those obtained by Simonson (1998).

Table 2. Effectiveness of an enthalpy wheel, cold test

Rotation	Flow	Speed	Simonson (1998)			Present work		
20 rpm	(m^3/h)	(m/s)	Sensible	Latent	Total	Sensible	Latent	Total
Process	1420	1.46	71 %	76 %	73 %	71 %	76 %	73 %
Exhaustion	1509	1.55						

4.1.2 Model validation – Zhang and Niu (2002)

The parameters of the wheel and properties of the desiccant matrix are as follow: the desiccant material is silica gel; the geometry of the channels in the wheel are sinusoidal with base of 4.35 mm and a height of 1.74 mm; the length (L) and diameter of the wheel are respectively, (0.1 m and 0.7 m); the desiccant thickness is 0.01 mm and substrate thickness is 0.03 mm; the following properties of the matrix (desiccant +substrate) were chosen: ρ_d=1129 Kg/m^3, Cp_d=615 J/kg K, m_d= 15.0 kg desiccant matrix, Cp_{al}=903 J/(kg K), ρ_{al}=2702 kg/m^3, k_{al}=237 W/(m K), σ_d=0.77 (fraction of the desiccant in the matrix); wheel rotational speed is 10 rpm; the sorption isotherm coefficients are C=1, *f*=0.75, u_{max}=0.25 kg water vapor / kg desiccant.

For hot test in a balanced configuration, the inlet temperature and humidity the desiccant wheel are: process air, 35.0 ^{0}C, 25.2 g/kg; exhaustion air, 24.0 ^{0}C, 13.1 g/kg; the initial conditions were 29.5 ^{0}C and 19.1 g/kg; mass airflow rates for both the adsorption and desorption sections are 0.4 kg/s.

Figure 5 presents the average parameters in the inlet and outlet of enthalpy wheel, for the hot test.

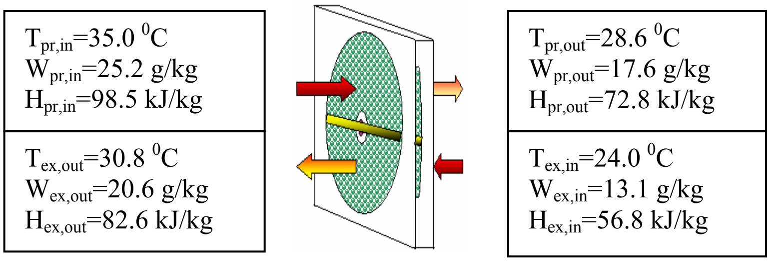


Figure 5. Enthalpy wheel

In Figure 5, it is presenting similar behavior to the described in the previous sub-section for the hot test.

In Table 3, the effectiveness for an enthalpy wheel is shown as function of the volumetric flows and air speeds. The same conditions of the problem shown in the Figure 3 had been imposed.

Table 3. Comparison is a commercially available desiccant dehumidification wheel, (1:1) configuration

Rotation	Flow	Speed	Effectiveness		
10 rpm	(m^3/h)	(m/s)	Sensible	Latent	Total
Process	1308	2.56	60 %	62 %	62 %
Exhaustion	1238	2.42			

In Figure 6, the temperature and absolute humidity profiles on the surface of desiccant of a representative channel of the enthalpy wheel in alternating cycles of adsorption and desorption are shown. It can be observed that in the periods of cooling a reduction of the temperature and humidity in the desiccant rotor occurs. This propagation of heat and mass obviously is faster in volume 1 (next to the entrance of the wheel) and slower in volume n (next to the exit to the wheel). For the period of heating, an increase of the temperature and the humidity in the wheel occurs. Being that now, the propagation of heat and mass are inverted, therefore the thermal source will be in the other extremity of the wheel. When the results of present work are compared to those of Zhang and Niu (2002), it can be observed a reasonable agreement.

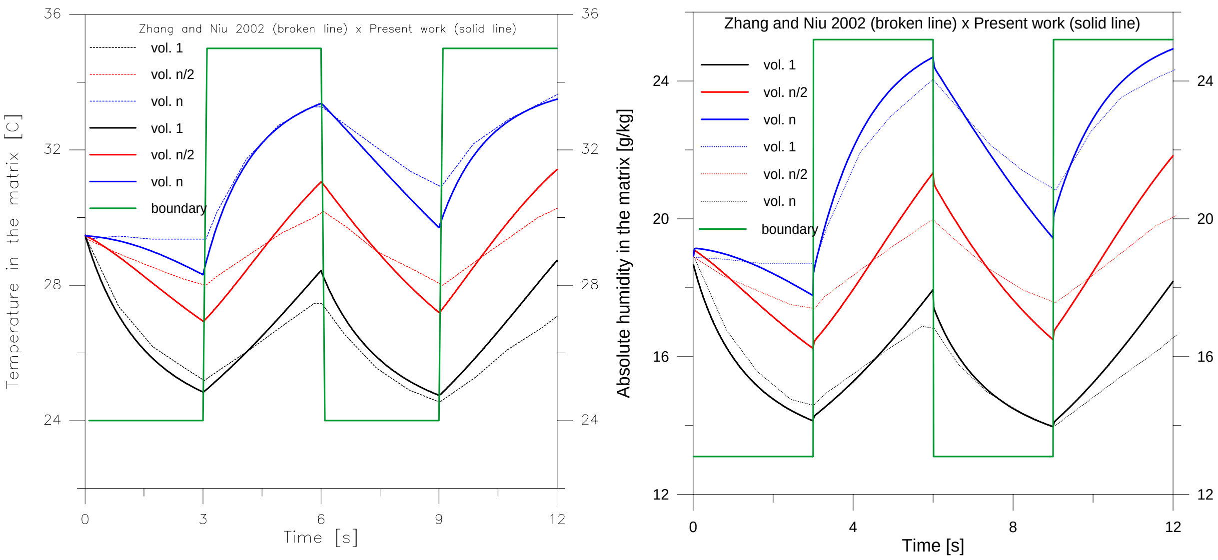


Figure 6. Temperature and absolute humidity into the desiccant matrix:
$T_{P,in}$=30.0 ^{0}C, $w_{P,in}$=21.0 g/kg, $T_{E,in}$=90.0 ^{0}C, $w_{E,in}$=21.0 g/kg e $\dot{m}_P = \dot{m}_E = 0.4\, kg/s$.

4.2. Mass diffusion resistance model

The effect of mass diffusion resistance model is presented in this section. Also, the model considering the effect of mass diffusion resistance is compared with numerical and experimental data from the literature.

4.2.1 Effect of the mass diffusion resistance

Table 4 presents the effects of mass diffusion resistance into the desiccant matrix. The parameter D_{ef} was changed from 10^{-14} to 10^{-1}. From Table 4, it is possible to infer that for $D_{ef} = 10^{-14}$ the water vapor accumulation in the desiccant will be small due to the high mass diffusion resistance. On the other hand, increasing the D_{ef} coefficient the amount of water vapor will increase in adsorption section which will be later on released in the desorption section. For D_{ef} values higher than 10^{-10}, the maximum amount of water vapor in the desiccant wheel will be reached; therefore the mass diffusion is fast.

When D_{ef} is equal to 10^{-10}, it is observed that in the exit of the process section will be had 18.3 g/kg and admitting the instantaneous equilibrium model will be had 17.6 g/kg. From Table 4, it can be observed that the effect of the mass diffusion resistance changes the amount of adsorbed and desorbed mass in the enthalpy wheel. However, when the differences between the results with and without mass resistance are compared, it can be observed that the instantaneous equilibrium model is still the best option since the differences between the models are small.

Table 4. Effect of the mass diffusion resistance in the performance of the enthalpy wheel: $T_{pr,in}$=35.0 ^{0}C, $w_{pr,in}$=25.2 g/kg, $T_{ex,in}$=24.0 ^{0}C, $w_{ex,in}$=13.1 g/kg and $\dot{m}_{pr} = \dot{m}_{ex} = 0.4\ kg/s$.

Outlet	Equilibrium	$D_{ef}=10^{-1}$	$D_{ef}=10^{-6}$	$D_{ef}=10^{-10}$	$D_{ef}=10^{-14}$
Process	28.6 ^{0}C	28.8 ^{0}C	28.8 ^{0}C	28.5 ^{0}C	28.1 ^{0}C
	17.6 g/kg	17.8 g/kg	17.8 g/kg	18.3 g/kg	23.1 g/kg
Exhaustion	30.8 ^{0}C	30.8 ^{0}C	30.8 ^{0}C	30.6 ^{0}C	29.8 ^{0}C
	20.5 g/kg	20.2 g/kg	20.2 g/kg	19.8 g/kg	15.4 g/kg

4.3 Experimental validation

Table 5 presents the results of the validation of mass diffusion resistance. 5. The results of the present work are compared with those obtained by Simonson (1998). From Table 5, it can be observed that a very good agreement of the results of the present work with the experimental and numerical results obtained by Simonson (1998) was obtained with the model without mass resistance diffusion for the outlet air process in terms of absolute humidity and temperature. Considering the mass resistance diffusion, it is expected an increase in the absolute humidity, but this behavior was not detected by the experimental procedure of Simonson (1998).

Table 5. Experimental validation of the enthalpy wheel: $T_{pr,in}$=39.7 ^{0}C, $w_{pr,in}$=1.8 g/kg, $T_{ex,in}$=26.2 ^{0}C, $w_{ex,in}$=7.4 g/kg and $\dot{m}_{pr} = \dot{m}_{ex} = 0.249\ kg/s$.

Outlet	Simonson, 1998		Present work	
	Experimental	Numerical	Without resistance	With resistance, $D_{ef}=10^{-10}$
Process	30.2 ^{0}C	30.6 ^{0}C	29.6 ^{0}C	29.6 ^{0}C
	6.6 g/kg	6.8 g/kg	6.9 g/kg	6.7 g/kg
Exhaustion	35.2 ^{0}C	35.3 ^{0}C	36.3 ^{0}C	36.1 ^{0}C
	1.6 g/kg	2.4 g/kg	2.3 g/kg	2.5 g/kg

5. CONCLUSIONS

In the present work, the performance of a desiccant air-conditioning wheel system has been investigated through the simulation of a one-dimensional model with and without the effects of mass diffusion in the enthalpy wheel. The effect of the mass diffusion resistance in the enthalpy wheel was investigated by the addition of the mass diffusion equation to the grains of the wheel with a non-linear boundary condition on the surface of desiccant wheel which is connected to the equation of mass conservation in the desiccant. From several simulation cases studies, it was verified that mass diffusion resistance changes the amount of adsorbed and desorbed mass in the enthalpy wheel, but this discrepancy is acceptable for the all tested cases. Therefore, the instantaneous equilibrium model can be used with an inferior computational cost compared mass diffusion resistance model.

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8. RESPONSIBILITY NOTICE

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