EFFECT OF THE MASS DIFFUSION RESISTANCE IN DESICCANT WHEELS FOR HUMID AIR DEHUMIDIFICATION

Jesus Marlinaldo de Medeiros, jesus_medeiros@yahoo.com.br
Federal Center of Technological Education of Sergipe, Sergipe, Brazil

José Maurício Gurgel, jmgurgel@pq.cnpq.br
Mechanical Technology Department, Federal University of Paraíba, Paraíba, Brazil

Francisco Marcondes, marcondes@ufc.br
Federal University of Ceará – Centro de Tecnologia - Campus do Pici – Bloco 714 – 60455-760 Fortaleza- CE –Brazil

Abstract. *The desiccant wheels are heat and mass exchangers, honeycomb type, impregnated with desiccant material used in adsorption air conditioning systems for humid air dehumidification. The problem consists of a cylindrical wheel formed by a net of very small channels that operate continuously with two contrary air flows: process (adsorption) and regeneration (desorption). The proposed mathematical model considers the mass diffusion resistance in the desiccant matrix. The system of equations was discretized by the finite-volume method using a fully implicit scheme in conjunction with a staggered arrangement. The results are presented in terms of temperature and absolute humidity profiles in the adsorption and desorption sections. The results have also been compared with other existing models found in the literature. It was observed that the effect of the mass diffusion resistance intervenes with the amount of adsorbed and desorbed mass in the desiccant wheel, and therefore, must be considered. In the experimental validation, the numerical results are physically coherent and adjust better to the experimental data than does the instantaneous equilibrium model.*

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

1. INTRODUCTION

For global sustainable development, it is necessary to reduce primary energy consumption and to introduce renewable energy as well. The reduction of the primary energy can 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 encourage the introduction of renewable energy.

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

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

Furthermore, the development of temperature and humidity independent control technology has been found necessary for rational energy utilization and comfortable environment control. It is possible to control temperature and humidity independently by means of desiccant air conditioning systems. Such systems can also be operated with relatively less thermal energy. Therefore, this method is recognized as being effective in reducing environmental pollution, and receives considerable attention nowadays.

Desiccant systems have many advantages relative to their closed-cycle counterparts: they operate at ambient pressure; heat and mass transfer between the air and the desiccant take place in a direct contact; both cooling and dehumidification of the conditioned air may be provided, in variable quantities, to fit the load in the conditioned space. Some 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 may require its replacement after some operation time.

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

The major advantage of desiccant cooling is its significant energy saving potentials, and reduced consumption of fossil fuels. The electrical energy requirement can be very low compared with the conventional refrigeration systems. The source of thermal energy can be diverse: solar energy, waste heat, and natural gas.

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

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

The desiccant wheel rotates slowly to expose one portion of the desiccant material to the process air stream while the other portion simultaneously passes through the regeneration air stream. A flexible seal separates the process air and regeneration air in the desiccant wheel. Moist air enters the process side and passes through the desiccant, where it is dehumidified. Regeneration occurs on the other side of the partition, where heated air enters most often from the opposite direction, then passes over the desiccant, and finally laden with moisture, is exhausted from the desiccant wheel.

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

The performance of the desiccant wheel depends on several parameters, such as ambient air condition (temperature and humidity), regeneration air, volumetric flow rates and rotation. Other wheel specific parameters are the geometry structure and the sorption properties of the material.

In order to predict heat and moisture transfer in the desiccant wheel, conservation equations for energy and mass must be postulated. Advective 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, 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 many countries, especially in the United States, China, India, Japan, Australia, Canada, and more recently in Brazil. Works using the desiccant technology are being used in adsorption refrigeration and in 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). Literature related to the desiccant wheels can well be evaluated in works by (Zheng and Worek, 1993; San and Hsian, 1993; Simonson, 1998; Grumback, 1999; Zhang and Niu, 2002; Sphaier and Worek, 2006).

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

2. MATHEMATICAL MODEL

A desiccant wheel is described as a rotating cylindrical porous-medium of length L and diameter D with small channels whose walls are coated 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 airflow channels. 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 regeneration sections. A schematic of the desiccant wheel together with one channel side view is shown in Figure 1. The governing equations are written for the simultaneous and coupled heat and moisture transfer in one channel as it rotates around the axis of the wheel. A typical operating condition would be that of a warm humid air supply, transferring energy and water vapor to the matrix, and energy and moisture being transferred from the matrix to the exhaust air during the second half of the cycle.

The transport phenomena occurring in a desiccant 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 exceedingly complex, requiring a prohibitive computational effect to obtain excessively detailed information. 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 felt are smaller than or close to 0.1. Therefore, the temperature and the mass gradient across the cross-section of the desiccant and airflow channel are small. To avoid excessive computational costs, it is reasonable to ignore the effect of heat and mass transfer across the thickness of the desiccant and airflow channel. Consequently, a one-dimensional model is considered an accurate method to describe thermal conduction and diffusion that occur within the porous desiccant (Gao et al., 2005).

For convenience, the following simplifications are assumed for the analysis and calculation: one-dimensional model in humid air and the desiccant matrix; forced advection 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 flow is laminar and fully developed into the channel; the 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 and thermodynamic properties of the matrix are constant; the channels are considered adiabatic and impermeable; the mass and heat transfer coefficient are constant; the carryover between two air flow negligible; the pressure drop along the channel is negligible, radiation effects are neglected, as a result of relatively small temperatures; no chemical reaction takes place, nor are there any energy sources within the system.

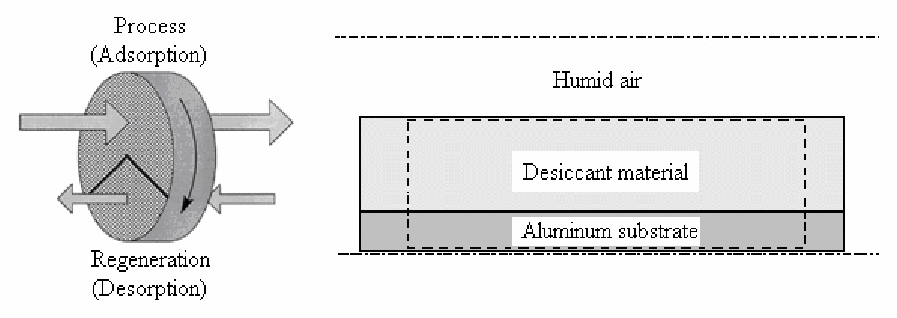

Figure 1. Schematic view of the desiccant wheel and a side view of one channel

2.1. Instantaneous equilibrium model

The governing equations for simultaneous heat and moisture transfer in desiccant wheels, based on the previous assumptions, are presented as follows, Simonson, 1998:
Energy conservation equation between air stream and desiccant matrix

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

Energy conservation equation on the desiccant matrix

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

Mass conservation equation for the water vapor

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

Mass conservation equation for the dry air

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

Mass conservation equation on the desiccant matrix

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

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

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

where T_g, and T_m are humid air temperature and matrix temperature; ρ_v, $\rho_{v,m}$ and ρ_a are vapor density in air, 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 desiccant wheel, h 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 into 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", and "w" are respectively 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 considering the fact that the water vapor density on the surface of the matrix will be equal to the water vapor density in the air at equilibrium, the following equations are obtained:

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

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

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 airflow:

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

Boundary conditions for regeneration airflow

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

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 psychrometric relationships for moist air, thermodynamics and geometric relationships, the formulation of the desiccant problem is completed. More details of the formulation can be found in Simonson, 1998.

2.2. Mass diffusion resistance model

The proposed mathematical model takes into acount the mass diffusion resistance in the desiccant matrix. The gradient of humidity inside the desiccant was modeled by the mass diffusion equation with a non-linear boundary condition on the surface of desiccant that was solved separately, but connected to the equation of conservation of mass in the desiccant matrix. In this model, the desiccant matrix assumed to be 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 is expressed by the following equation:

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

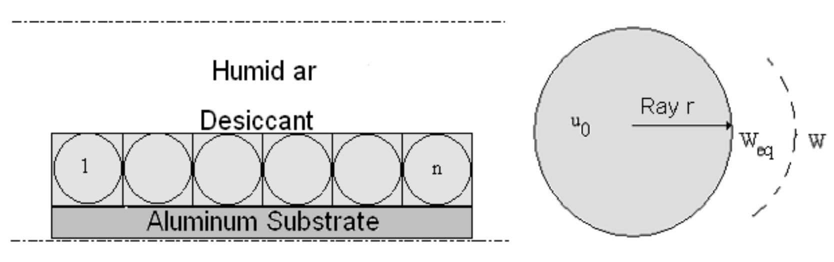

Figure 2. Schematic view of the desiccant wheel and side view of one grain

Initial conditions

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

Boundary conditions for process airflow

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

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

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, rewritten Eq. (7), and expressing φ as a function of the mass fraction of water into the desiccant

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

The psychrometric 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 is 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. Velocity is solved at the faces 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 the convergence, the energy equation in the matrix is solved by the Tridiagonal Matrix Algorithm (Patankar, 1980).

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

1. Estimate of 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 simulation process goes to next time step and repeats 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 the steady-steady solution, 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.

4. RESULTS AND DISCUSSION

The results for the models with and without mass diffusion resistance are now presented. For all results that will be shown in this section, it was used the isotherm presented by Simonson (1998). A mesh and time step study was carried out and based on the tests, all the results presented next have been obtained using 100 volumes and a time-step of 10^{-2} s.

4.1. Instantaneous equilibrium model

For all results that will be presented

The instantaneous equilibrium model proposed by Simonson (1998) is now compared with a commercial-desiccant-dehumidification wheel for two available data: balanced (1:1) and non-balanced (1:3).

The parameters of the wheel and properties of the desiccant matrix are detailed here: the desiccant material is silica gel; the geometry of the channels in the wheel is sinusoidal with of 3.62 mm and a height of 1.81 mm; the length (L) and Diameter of the wheel are respectively, 0.2 m and 0.55 m; the desiccant thickness is 0.1 mm and the substrate thickness is 0.03 mm; the properties of the matrix (desiccant plus substrate) are ρ_d=270 Kg/m^3, Cp_d=615 J/kg K, m_d= 10.0 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); the wheel rotational speed is 15 rph; the sorption isotherm coefficients are C=1, f=0.75, u_{max}=0.4 kg water vapor / kg desiccant.

For air dehumidification in a balanced configuration, the inlet temperature and humidity of the desiccant wheel are: process air, 30.0 ^{0}C, 22.0 g/kg; regeneration air, 100.0 ^{0}C, 14.0 g/kg; the initial conditions were 65.0 ^{0}C and 6.0 g/kg; the mass airflow rates for both the adsorption and regeneration sections are 0.25 kg/s.

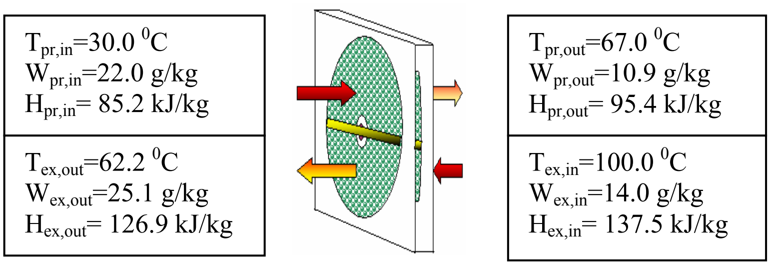

Figure 3. Desiccant wheel: (1:1) configuration

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

Table 1 compares the results of present work with those obtained by the commercial wheel and those obtained by Lima (2004). The same conditions for the problem shown in the Figure 3 had been imposed in the inlet and outlet of the wheel. From Table 1, it is possible to observe a good agreement between the results of the present work, those obtained by Lima, 2004, and those of the commercial wheel.

For an non-balanced configuration, the inlet temperature and humidity of the desiccant wheel are: process air, 30.0 ^{0}C, 22.0 g/kg; regeneration air, 120.0 ^{0}C, 14.0 g/kg; the initial conditions were 85.0 ^{0}C and 6.0 g/kg; the mass airflow rates for both the adsorption and regeneration sections are 0.25 kg/s and 0.065 kg/s, respectively. The results for this configuration are presented in Fig. 4, and comparisons with the available data are presented in Tab. 2.

Table 1. Commercial-desiccant-dehumidification wheel - (1:1) configuration

Process section			
Inlet	Outlet		
	DRI wheel	Lima, 2004	Present work
$T_{pr,in}$=30.0 ^{0}C	65.0 ^{0}C	68.2 ^{0}C	67.0 ^{0}C
$W_{pr,in}$=22.0 g/kg	11.0 g/kg	10.5 g/kg	10.9 g/kg

From Figure 4, it is possible to infer that a similar behavior obtained for the balanced configuration was observed. However, the humidity reduction was smaller than that observed in the balanced configuration (11.1 g/kg) compared to 6.1 g/kg. On the other hand, the adverse effect of temperature also was also smaller.

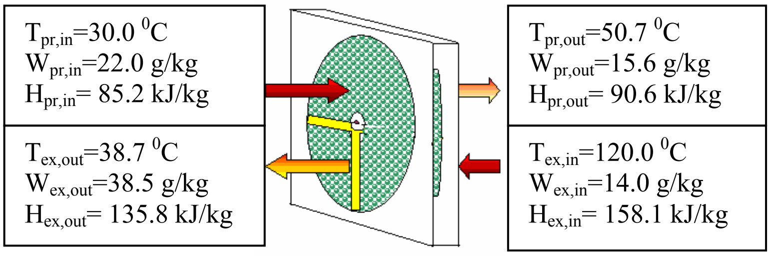

Figure 4. Desiccant (1:3) wheel configuration

Again from Table 2, for a non-balanced configuration, it is possible to observe that a close math between the simulated results and those obtained by Lima, 2004 and the commercial manufacturer was observed.

Table 2. Commercial-desiccant-dehumidification wheel - (1:3) configuration

Process section			
Inlet	Outlet		
	DRI wheel	Lima, 2004	Present work
$T_{pr,in}$=30.0 ^{0}C	55.0 ^{0}C	58.2 ^{0}C	50.7 ^{0}C
$W_{pr,in}$=22.0 g/kg	14.5 g/kg	13.4 g/kg	15.6 g/kg

4.1.2 Sensitivity of the individual parameters

After the validation of the model, the effect of several parameters in the wheel performance is presented in this sub-section. The following parameters will be investigated: mass flow rate or air velocity, wheel speed, regeneration/process relation, temperature, humidity, and inlet conditions of air temperature and humidity for process air stream as well as inlet conditions of air temperature and humidity of regeneration stream on the outlet air conditions. The effects of these parameters are presented in Tab. 3.

From Table 3, the following effects can be observed:

- By decreasing the relation R/P, one will decrease the outlet temperature of the process air. On the other hand, the outlet absolute humidity will increase, and as a result, the capacity of refrigeration of the system will also decrease.
- The ideal wheel rotational speed range to provide small outlet absolute humidity of the process air is located between 10 to 15 rph.
- By increasing the regeneration temperature, one will improve the dehumidification of the process air. Nevertheless, both the consumption of energy for regeneration, and the outlet temperature of the process air will increase. Therefore, the performance of the system can be decreased.
- The absolute humidity of regeneration does not affect significantly the performance of the desiccant wheel. However, an air leak in the regeneration section of the process air section can increase the outlet absolute humidity of the process air.
- By increasing the regeneration volumetric rate, one will increase the dehumidification of the process air. On the other hand, energy consumption for regeneration and the outlet temperature will also increase.

- By decreasing the inlet temperature, one will increase the effect of dehumidification because the desiccant will have a higher capacity of adsorption.

- By decreasing the entrance absolute humidity of the process, one will increase the dehumidification of the process air, and higher refrigerating capacity can be supplied to the system.

- By increasing the process volumetric rate, one will increase the outlet absolute humidity in the process air. Therefore, the performance of the system can be decreased.

Table 3. Analysis of several parameters that affect the performance of a desiccant wheel

Q_{pr}, Q_{ex} [m^3/h]; N [rph]; T_{pr},T_{ex} [^{0}C]; W_{pr}, W_{ex} [g/kg]				Process				Regeneration (exhaustion)			
				Inlet		Outlet		Inlet		Outlet	
R/P	Q_{pr}	Q_{ex}	N	T_{pr}	W_{pr}	T_{pr}	W_{pr}	T_{ex}	W_{ex}	T_{ex}	W_{ex}
1:1	1200	1200	15	30.0	22.0	58.5	13.1	100.0	22.0	64.4	32.9
1:1	1500	1500	15	30.0	22.0	55.1	14.0	100.0	22.0	68.4	31.8
1:1	1800	1800	15	30.0	22.0	52.4	14.8	100.0	22.0	71.7	30.9
1:3	1800	600	7.5	30.0	22.0	43.2	17.5	100.0	22.0	50.1	38.8
1:3	1800	600	10	30.0	22.0	44.1	17.1	100.0	22.0	46.9	39.8
1:3	1800	600	15	30.0	22.0	44.8	17.1	100.0	22.0	44.4	40.1
1:3	1800	600	30	30.0	22.0	45.4	17.3	100.0	22.0	42.3	39.4
1:3	1800	600	10	30.0	22.0	40.5	18.3	80.0	22.0	42.9	34.8
1:3	1800	600	15	30.0	22.0	41.0	18.3	80.0	22.0	41.1	35.0
1:3	1800	600	30	30.0	22.0	41.5	18.5	80.0	22.0	39.4	34.3
1:3	1800	600	10	30.0	22.0	47.4	16.3	120.0	22.0	51.0	44.4
1:3	1800	600	15	30.0	22.0	48.3	16.1	120.0	22.0	47.6	45.1
1:3	1800	600	30	30.0	22.0	49.1	16.3	120.0	22.0	44.7	44.0
1:3	1800	600	10	30.0	22.0	50.2	15.6	140.0	22.0	55.6	48.3
1:3	1800	600	15	30.0	22.0	51.4	15.3	140.0	22.0	50.6	49.5
1:3	1800	600	30	30.0	22.0	52.4	15.6	140.0	22.0	46.8	48.3
1:3	1800	600	15	30.0	22.0	51.7	15.2	140.0	18.0	49.6	45.8
1:3	1800	600	15	30.0	22.0	51.2	15.3	140.0	26.0	51.7	53.2
1:3	1800	600	15	30.0	22.0	51.0	15.5	140.0	30.0	52.6	56.9
1:3	1800	600	15	30.0	22.0	51.4	15.3	140.0	22.0	50.6	49.5
1:3	1800	1060	15	30.0	22.0	58.7	13.5	140.0	22.0	99.0	33.6
1:3	1800	1200	15	30.0	22.0	58.1	13.5	140.0	22.0	80.5	39.2
1:3	1800	1200	30	30.0	22.0	63.6	12.4	140.0	22.0	69.6	41.5
1:3	1800	1200	45	30.0	22.0	65.4	12.5	140.0	22.0	66.2	41.4
1:3	1800	600	10	29.0	19.0	48.9	12.8	140.0	19.0	56.6	44.5
1:3	1800	600	15	29.0	19.0	50.2	12.5	140.0	19.0	51.3	45.5
1:3	1800	600	20	29.0	19.0	50.8	12.6	140.0	19.0	49.2	45.3
1:3	1800	600	30	29.0	19.0	51.3	12.8	140.0	19.0	47.0	44.2
1:3	1500	500	10	29.0	19.0	50.2	12.4	140.0	19.0	51.7	46.0
1:3	1500	500	15	29.0	19.0	51.1	12.4	140.0	19.0	48.1	46.1
1:3	1500	500	20	29.0	19.0	51.5	12.5	140.0	19.0	46.5	45.6
1:3	1500	500	30	29.0	19.0	52.0	12.9	140.0	19.0	44.6	44.0
1:3	1200	400	10	29.0	19.0	51.3	12.2	140.0	19.0	47.6	46.9
1:3	1200	400	15	29.0	19.0	51.9	12.3	140.0	19.0	45.3	46.3
1:3	1200	400	20	29.0	19.0	52.2	12.6	140.0	19.0	44.0	45.4
1:3	1200	400	30	29.0	19.0	52.6	13.1	140.0	19.0	42.4	43.2
1:3	1800	600	10	27.0	14.0	8.6	45.5	140.0	19.0	61.9	42.5
1:3	1800	600	15	27.0	14.0	8.3	47.2	140.0	19.0	55.4	43.7
1:3	1800	600	30	27.0	14.0	8.7	48.6	140.0	19.0	49.9	42.0

4.2. Mass diffusion resistance model

The effects of mass diffusion are now evaluated. First, the effect of mass resistance parameter is investigated. Then, a comparison is performed between the model with mass diffusion resistance and the available 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^{-12} to 10^{-1} m^2/s. From Table 4, it is possible to infer that for $D_{ef} = 10^{-12}$ 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 the adsorption section which, will be later on released into 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.

For $D_{ef} = 10^{-10}$, one observes that in the exit of the process section the air has 16.9 g/kg, and considering the instantaneous equilibrium model, the air has 13.5 g/kg. This discrepancy between the models can be explained considering that the adsorption or desorption times are still inferior to the necessary time for the complete filling of the grains. It has been observed that the effect of the mass diffusion resistance intervenes with the amount of adsorbed and desorbed mass in the desiccant wheel, and therefore, must be considered.

Table 4. Effect of the mass diffusion resistance in the performance of the desiccant wheel: $T_{pr,in}$=30,0 ^{0}C, $W_{pr,in}$=21,0 g/kg, $T_{ex,in}$=90,0 ^{0}C, $W_{ex,in}$=21,0 g/kg e $\dot{m}_{pr} = \dot{m}_{ex} = 0{,}4\ kg/s$.

Outlet	Equilibrium	$D_{ef}=10^{-1}$ m^2/s	$D_{ef}=10^{-6}$ m^2/s	$D_{ef}=10^{-10}$ m^2/s	$D_{ef}=10^{-12}$ m^2/s
Process	59.2 ^{0}C	53.6 ^{0}C	53.6 ^{0}C	53.4 ^{0}C	49.1 ^{0}C
	13.5 g/kg	16.9 g/kg	16.9 g/kg	16.9 g/kg	18.8 g/kg
Regeneration	60.6 ^{0}C	62.6 ^{0}C	62.6 ^{0}C	63.0 ^{0}C	67.9 ^{0}C
	28.5 g/kg	26.3 g/kg	26.3 g/kg	26.2 g/kg	24.0 g/kg

4.2.2 Experimental validation

The validation of mass diffusion resistance is now presented in Tab. 5 and Tab. 6. The results of the present work are compared with the ones obtained by Zhang et al. (2002) and Kodama *et al. (*2003). From Table 5, it can be observed that a very good agreement between the results of the present work and the experimental and numerical results obtained by Zhang et al. (2002) was obtained with the model without mass resistance diffusion for outlet air process in terms of absolute humidity and temperature. Considering the mass resistance diffusion, an increase in the absolute humidity is expected, but this behavior has not been detected by the experimental procedure by Zhang and Niu (2002).

Table 5. Experimental validation of the desiccant wheel, (1:3) configuration: $T_{pr,in}$=34,3 ^{0}C, $W_{pr,in}$=19,2 g/kg, $T_{ex,in}$=100,0 ^{0}C, $W_{ex,in}$=19,2 g/kg, $\dot{m}_P = 0.160\ kg/s \quad e \quad \dot{m}_E = 0.032\ kg/s$.

Outlet	Zhang *et al,* 2002		Present work	
	Experimental	Numerical	Without resistance	With resistance, $D_{ef}=10^{-10}$ m^2/s
Process	44.0 ^{0}C	46.8 ^{0}C	45.4 ^{0}C	44.0 ^{0}C
	15.4 g/kg	15.5 g/kg	15.7 g/kg	17.0 g/kg
Regeneration	-	-	44.1 ^{0}C	40.6 ^{0}C
	-	-	36.6 g/kg	34.0 g/kg

Table 6. Experimental validation of the desiccant wheel, (1:1) configuration: $T_{pr,in}$=31,0 ^{0}C, $W_{pr,in}$=10,3 g/kg, $T_{ex,in}$=80,5 ^{0}C, $W_{ex,in}$=12,5 g/kg e $\dot{m}_{pr} = 0.072\ kg/s \quad e \quad \dot{m}_{ex} = 0.061\ kg/s$.

Outlet	Kodama *et al,* 2003		Present work	
	Experimental	Numerical	Without resistance	With resistance, $D_{ef}=10^{-10}$ m^2/s
Process	56.7 ^{0}C	55.8 ^{0}C	56.3 ^{0}C	55.8 ^{0}C
	4.8 g/kg	4.5 g/kg	4.9 g/kg	6.6 g/kg
Regeneration	56.2 ^{0}C	54.1 ^{0}C	51.0 ^{0}C	47.8 ^{0}C
	18.2 g/kg	19.2 g/kg	18.7 g/kg	17.8 g/kg

Comparing the results of the present work with and without mass diffusion resistance with the ones obtained by Kodama et al. (2003), shown in Tab. 6, it can be inferred again that the model without mass diffusion resistance presents an excellent agreement for the air process in terms of absolute humidity and temperature. Although the model with mass resistance presents almost the same outlet temperature for the air process, an extremely high absolute humidity has been obtained with this model. On the other hand, for the regeneration process, the model with mass diffusion resistance is closer to the experimental results.

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 desiccant wheel. The effect of the mass diffusion resistance in the desiccant 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 case studies, it was verified that the parameters affecting the wheel performance are: the R/P relation, the speed of rotation, the temperatures of process and regeneration, and the absolute humidity of process.

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8. RESPONSIBILITY NOTICE