PERFORMANCE EVALUATION OF ADSORPTIVE MATERIALS FOR DESICCANT COOLING SYSTEMS

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Abstract. *The continuous rise of of electric power rates worlwide and environmental issues related to the ozone layer depletion have increased the interest in natural gas cooling technologies over the last years . Accordingly, the present work analises the inluence of the adsorptive material over the coeficient of performance of an adsorptive cooling cycle. The governing equations for heat and mass transfer are solved employing a totally implicit finite volume technique,and the adsorption isotherms are represented by a general equation, in which the varition of the separation factor R allows the behavior of three different materials (silica-gel, molecular sieve and 1M) to be simulated. The results show that the separation factor R has a great influence over the coefficient of performance, for a fixed regeneration temperature. It is also shown that for one particular material (1M) the coeffcient of performance is relatively independent of the regeneration temperature, making it a natural choice for systems that employ a unsteady enegy source, such as solar energy.*

Keywords*: desiccant, adsorption, air-conditioning,*

1. Introduction

The present work is devoted to the modeling, simulation and analysis of desiccant wheels, as applied to thermal comfort systems. Desiccant wheels consist of a porous disc impregnated with hygroscopic material, which by definition attract and retain water vapor. Figure 1 shows a schematic of a desiccant wheel, operating between two air streams. At the "cold side", a fresh air stream is forced through the wheel, giving up its moisture to the hygroscopic material, leaving the wheel at a much lower absolute humidity. As the wheel rotates, the hygroscopic material eventually switches to the regeneration stream, where hot air from a hot source is forced through the wheel, drying the hygroscopic material and dumping the water vapor back to the atmosphere. As depicted in Figure 1, the porososity pattern is not random, but formed by small channels, evenly distributed throughout the disc. Each channel has a structural layer (usually aluminum) and an adsorptive layer, above which the air flows.

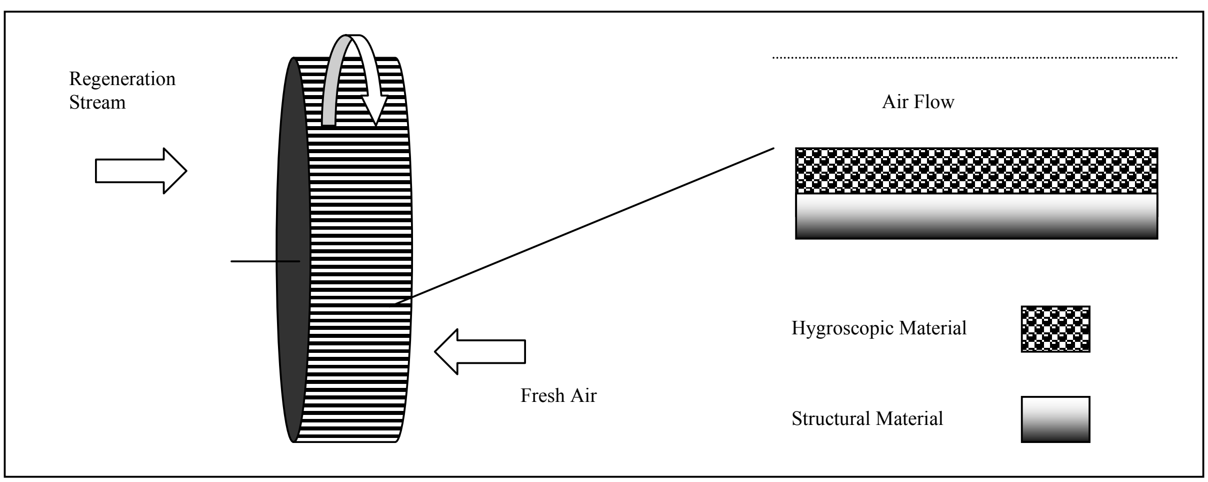

Figure 1: Schematic representation of a desiccant wheel

2. Mathematical Modeling

The adsorption phenomena occur in almost all solid-vapor interfaces, but its effects are negligible unless the solid possess certain proprieties that allow a great affinity with the vapor. For instance, the internal area of 1 cm^3 of silica-gel is estimated to be 400m^2. The average size of the porous is also of great importance, as it is the factor which determines the size of the molecules to be absorbed. The structural integrity in face of the continuous thermal inversion is also a factor to be considered. One of the earliest works devoted to the dehumidification modeling was performed by Bullock and Trelkheld (1966), and used a predictor-corrector method to solve a quite simplified model. Maclaine and Cross (1972) presented a characteristics potential method, which allowed the coupled problem to be solved in an analogous heat transfer problem. Jurinak et al (1984) proposed alternative design options combining evaporative coolers and desiccant wheels, expanding the range of applicability of such equipment. Zheng and Worek (1993) used an explicit finite difference scheme to solve the governing equations, using experimental data for the silica-gel heat of adsorption and isotherm shape. Zhang et al. (2003) presents experimental data for the heat of adsorption used by the present simulation. The present work uses a totally implicit finite volume technique to solve the non-dimensional governing equations, which contains less parameters when compared to previous efforts. It also proposes the use of a generalized equation for isotherm shape, which can be reduced to a particular material (silica-gel, molecular sieve 13 or 1M) by choosing an appropriate value for the separation parameter R. Consider an element of the channel of length Δx, in the direction of the flow, as shown in figure 2.

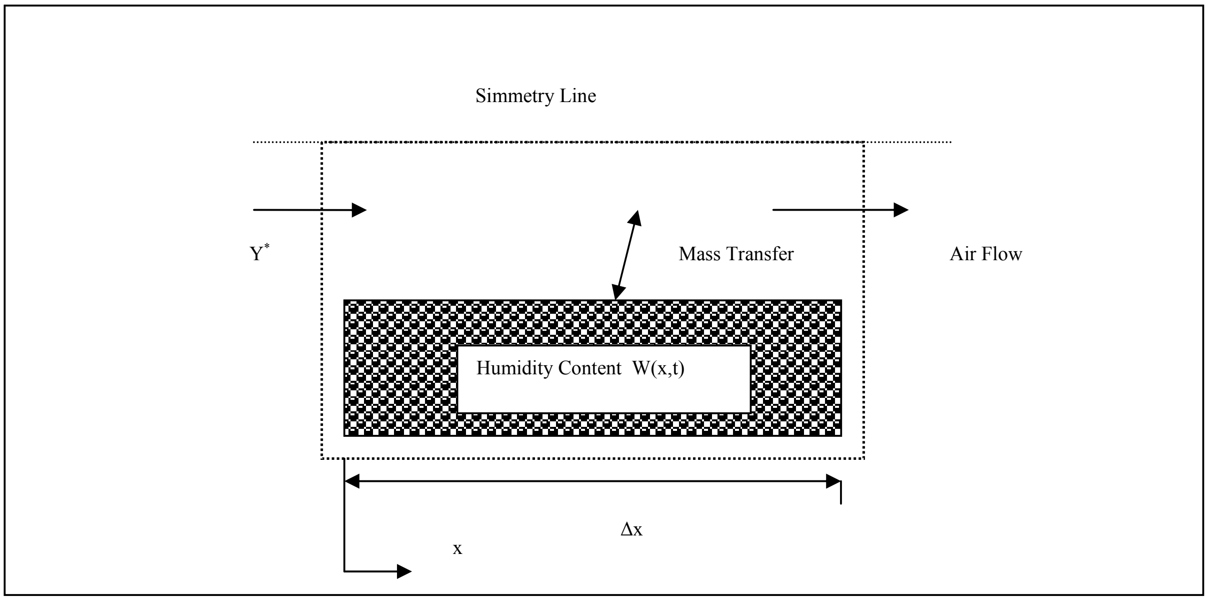

Figure 2: Elementary Control Volume

Some simplifying assumptions are necessary to the establishment of a mathematical model:

1) The flow is hydrodynamic developed.
2) The heat and mass transfer coefficients are constant along the channel.
3) Energy and mass storage within the air are negligible when compared to the solid.
4) Uniform temperature and concentration values in the direction perpendicular to the flow.
5) Symmetry lines represent perfectly adiabatic and impermeable surfaces.
6) Thermal and mass resistances are negligible in the direction perpendicular to the flow.
7) Thermal and mass resistances are infinite in the direction of the flow.

These assumptions can be justified one at time. Assumption (1) is reasonable because of the low viscosity of air and the (small) ratio between the air channel height and length. Assumption (2) holds because the flow is usually laminar in the range of interest, whereas assumption (3) reflects the smaller thermal and mass capacitances of air when compared to those of the hygroscopic material. Assumption (4) reflects the typical dimensions of desiccant wheels, in which the channel length is usually hundred or thousand times larger than the channel height and the desiccant layer. Assumption (5) reflects the uniformity of the porous media, and both assumptions (6) and (7) are consequences of assumption (3).

Accordingly, applying a mass balance to the control volume shown on Figure (2), enclosing both the flow channel and the hygroscopic material,

$$\underbrace{\rho_1\,\Delta x\; b\, a_1 \frac{\partial Y^*}{\partial t_A}}_{\substack{transient\\ within\,the\,channel}} + \underbrace{f\;\rho_w\,\Delta x\; b\, a_w \frac{\partial W}{\partial x}}_{\substack{transient\\ within\,the\,solid}} + \underbrace{\frac{\dot{m}_1}{n}\left[\left(Y^* + \frac{\partial Y^*}{\partial x}\Delta x\right) - Y^*\right]}_{net\;\;flux} = 0 \qquad (1)$$

in which

ρ_1 = *density of air*
Δx = *elementary length*
b = *channel and solid width*
a_1 = *channel height*
Y^* = *air absolute humidity*
W = *solid humidity content*
ρ_w = *density of solid*
f = *solid void fraction*
t_a = *time*

using the following definitions

$$\dot{m}_1 = \rho_1\, a_1\, y_{AF}\, u_1\, n$$

$$m_w = \rho_w\, a_w\, x_{AF}\, y_{AF}\, n$$

one obtain

$$\dot{m}_1 \left[\frac{1}{u_1}\frac{\partial Y^*}{\partial t_A} + \frac{\partial Y^*}{\partial x_A} \right] + \frac{f\;m_w}{x_{AF}}\frac{\partial W}{\partial t_A} = 0$$

The mass transfer between the solid ant the air stream is given by

$$f\;\rho_w\, b\,\Delta x_A\, a_w \frac{\partial W}{\partial t_A} = 2 h_y\, b\,\Delta x_A \left(Y^* - Y^*{}_w\right)$$

or

$$\frac{f\;m_w}{n\;x_{AF}\;y_{AF}}\frac{\partial W}{\partial t_A} = 2\,h_y \left(Y^* - Y_w^*\right) \qquad (2)$$

in which

m_w total mass of desiccant in the wheel
n number of channels
x_{af} length of the wheel
h_y mass transfer coefficient

Applying a mass balance to the control volume shown on Figure (2), enclosing both the flow channel and the hygroscopic material,

$$\underbrace{\rho_1\,\Delta x_A\,y_{AF}\,a_1\frac{\partial H_1}{\partial t_A}}_{\substack{transient\\ within\ the\ channel}}+\underbrace{\rho_w\,\Delta x_A\,y_{AF}\,a_w\frac{\partial H_w}{\partial t_A}}_{\substack{transient\\ within\ the\ solid}}+\underbrace{\frac{\dot{m}_1}{n}\left[\left(H_1+\frac{\partial H_1}{\partial x_A}\Delta x_A\right)-H_1\right]}_{net\ \ flux}=0$$

or

$$\dot{m}_1\left[\frac{1}{u_1}\frac{\partial H_1}{\partial t_A}+\frac{\partial H_1}{\partial x_A}\right]+\frac{m_w}{x_{AF}}\frac{\partial H_w}{\partial t_A}=0 \tag{3}$$

The heat transfer between the solid ant the air stream is given by

$$\underbrace{\rho_1\,b\,\Delta x_A\,a_1\frac{\partial H_1}{\partial t_A}}_{\substack{transient\\ within\ the\ channel}}+\underbrace{\frac{\dot{m}_1}{n}\left[\left(H_1+\frac{\partial H_1}{\partial x_A}\Delta x_A\right)-H_1\right]}_{\substack{net\ enthalpy\\ flux}}=\underbrace{2h_y\,b\,\Delta x_A\left(Y^*-Y_w^*\right)\frac{\partial H_1}{\partial Y}}_{\substack{heat\ released\ due\ to\\ adsorption}}+\underbrace{2\,h\left(T_w-T_1\right)}_{\substack{heat\ transfer\\ between\ air\ and\\ solid}}$$

$$\frac{\dot{m}_1}{nb}\left[\frac{1}{u_1}\frac{\partial H_1}{\partial t_A}+\frac{\partial H_1}{\partial x_A}\right]=2\,h_y\left(Y_w^*-Y^*\right)\frac{\partial H_1}{\partial Y}+2\,h\left(T_w-T_1\right) \tag{4}$$

in which

$$H_1=aT_1+Y^*\left(d+cT_1\right) \tag{5}$$

$$a=1.0046465\,KJ/Kg\ ^\circ C$$

$$d=2467.4304\,KJ/Kg$$

$$c=1.8837122\,KJ/Kg\ ^\circ C$$

Equations (1) through (4) are transformed into equivalent non-dimensional forms after extensive algebra,

$$\frac{\partial Y^*}{\partial x^*}=Y_w^*-Y^* \tag{6}$$

$$\frac{\partial W}{\partial t^*_{h,c}}=\lambda_2\left(Y^*-Y_w^*\right) \tag{7}$$

$$\frac{\partial T_1}{\partial x^*}=T_w-T_1 \tag{8}$$

$$\frac{\partial T_w}{\partial t^*_{h,c}}=\left(T_1-T_w\right)+\lambda_1\left(Y^*-Y_w^*\right) \tag{9}$$

where h and c refer to the hot and cold periods, respectively, and

$$\lambda_2=\frac{C_{wr}}{f\left(\dfrac{\partial H_1}{\partial T_1}\right)} \qquad\qquad \lambda_1=\frac{Q}{\left(\dfrac{\partial H_1}{\partial T_1}\right)}$$

The heat of adsorption Q is expressed in terms of the latent heat h_v and is experimentally obtained as (ZHANG et al., 2003)

$$Q = h_v\,(1.0 + 0.2843\,e^{-10.28W}) \tag{10}$$

We have now four equations (6) to (9) and five unknowns, Y^*, Y, W, T_1 and T_w . The missing equation is the adsorption isotherm, which relates the humidity content of the hygroscopic material, its temperature and the absolute humidity of the air layer in equilibrium with the solid,

$$W = W\left(T_w, Y_w^*\right) \tag{11}$$

with boundary conditions are given by

$$T_1(0,t^*) = T_{h\,in} \qquad ,0 < t^* < P_h^* \tag{12}$$

$$Y_1(0,t^*) = Y_{h\,in} \qquad ,0 < t^* < P_h^* \tag{13}$$

$$T_1(x_f^*,t^*) = T_{c\,in} \qquad ,P_h^* < t^* < P^* \tag{14}$$

$$Y_1(x_f^*,t^*) = Y_{c\,in} \qquad ,P_h^* < t^* < P^* \tag{15}$$

and the periodicity conditions are given by

$$T_{wc}(x^*,P^*) = T_{wh}(x^*,0) \tag{16}$$

$$W_c(x^*,P^*) = W_h(x^*,0) \tag{17}$$

The adsorption isotherm (Eq.11) is specific for each hygroscopic material. Simonson and Besant (1999) suggest that although the humidity content W is a function of both temperature and relative humidity of the air layer,its dependence on the later is much stronger than on the former. Accordingly, a variety of adsorptive materials can be represented by

$$\frac{W}{W_{\max}} = \frac{1}{(1 - R + \frac{R}{RH})} \tag{18}$$

in which

W_{max} maximum humidity content of the solid
R separation factor
RH relative humidity of the air layer

The separation factor R can be adjusted so as to make the curve a good representation of selected hygroscopic materials, as illustrated in Figure (3). The definition of effectiveness for a desiccant wheel is not as straightforward as it is in a heat exchangers. In a desiccant wheel both heat and mass are being exchanged, and mass and heat transfers have opposite directions. Taking the regenerative stream as an example, one would note that its temperature decreases as it flows through the channel, which tends to decrease its enthalpy. However, the hot stream is also humidified as it flows, which causes its enthalpy to increase. Results are expressed by the dehumidification effectiveness given by

$$\varepsilon_L = \frac{\left(Y_{ci} - Y_{co}\right)}{\left(Y_{ci} - Y_{hi}\right)} \tag{18}$$

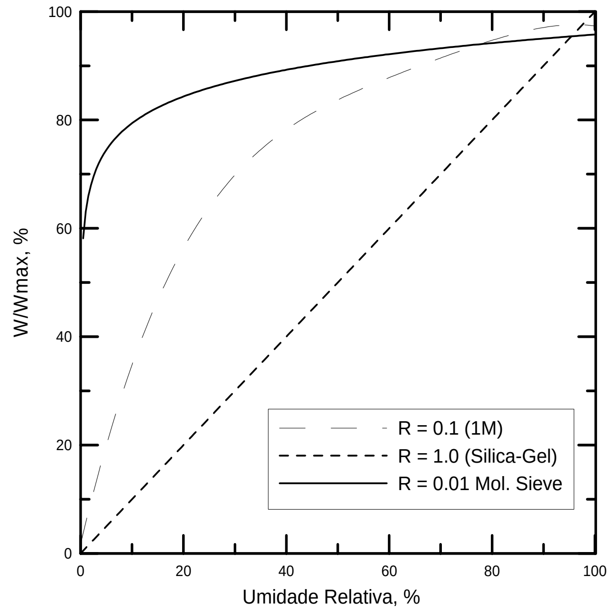

Figure 3: Generalized Adsorption Isotherm

The domain was discretized using a finite-volume technique, using a fully-implicit formulation for the transient terms and an upwind formulation for the convective terms. Since both initial mass and temperature distributions are not known a priori, an iterative procedure, which compares the initial fields with the fields calculated at 360° is required. The enthalpy flux of the inlet streams must equal the average enthalpy flux of the outlet streams, as otherwise the wheel wouldn't be operating in a cyclic condition. The heat balance error HBE given below was found to be smaller than 0,1% for all simulations carried out.

$$HBE = \frac{\dot{m}_h\, h_{hi} + \dot{m}_c\, h_{ci} - (\dot{m}_h \dfrac{1}{p_h}\displaystyle\int_0^{p_h} h_{ho}\, dt^* + \dot{m}_c \dfrac{1}{p_c}\displaystyle\int_0^{p_c} h_{co}\, dt^*)}{\dot{m}_h\, h_{hi} + \dot{m}_c\, h_{ci}} \tag{19}$$

3. Results

Figures (4) and (5) show the influence of the adsorptive material selection over the effectiveness of dehumidification for two regeneration temperatures. For both cases the best performance is obtained for R=0,1 (1M). For a moderate value of the regeneration temperature (100°C), silica-gel exhibits a better performance than molecular-sieve, whereas for a higher regeneration temperature (160°C) the results are reverted. This results are consistent with the fact that molecular sieves are much stronger adsorvents than silica-gel, which in turn also requires high regenerative temperatures for the desorption to take place. Interesting to note that a relatively weak adsorbent like silica-gel might nor be indicated, because of the low capacity of removing humidity. A strong adsorbent like molecular sieve might not be recommended as well, because a high affinity to the water vapor might result in an uncompleted desorption, compromising the cyclic operation of desiccant wheel. Accordingly, a moderate value for the separation factor seems to be the best choice. Figure (6) show the influence of non-dimensional revolution period P over the effectiveness for a given material (1M), and suggests the existence of an optimum period of revolution $P^* = 40.0$. The existence of an optimum period of revolution for a given regeneration temperature can be justified by observing that a high angular velocity (small revolution period) might implicate in uncompleted sorption and desorption processes, whereas a small angular velocity would result in overheating during desorption, increasing the average temperature of the desiccant material and reducing the dehumidifying capacity. Figure (7) shows that the increase in the regeneration temperature also cause the effectiveness to increase, up to a maximum at180°C.

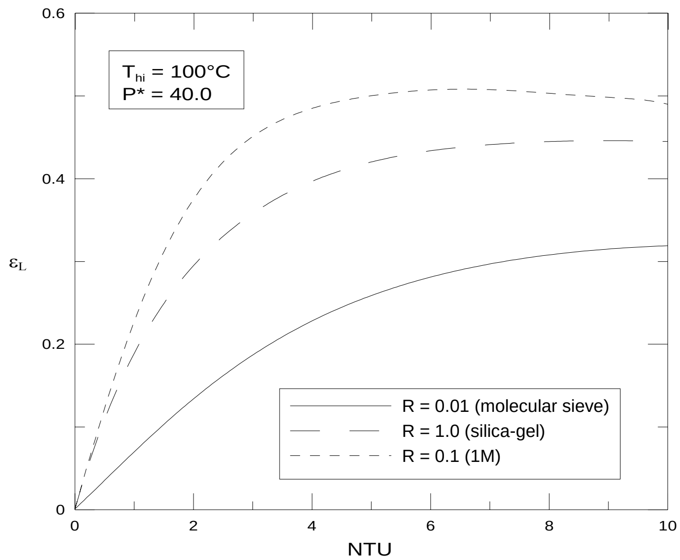

Figure 4: Selection of the adsorptive material

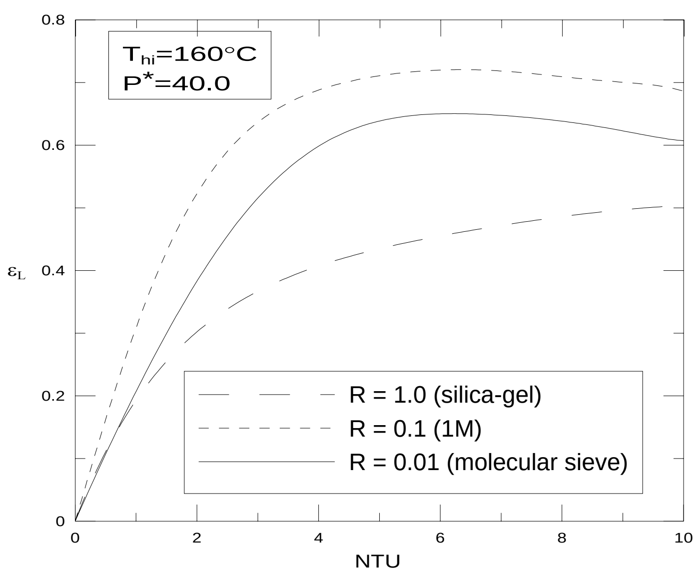

Figure 5: Selection of the adsorptive material

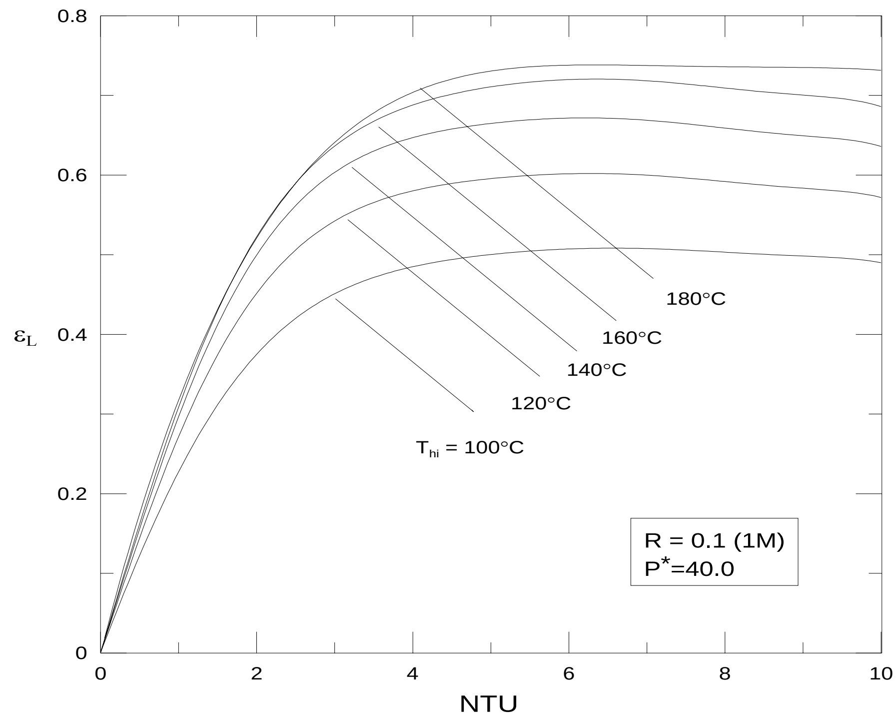

Figure 6: Influence of the non-dimensional period

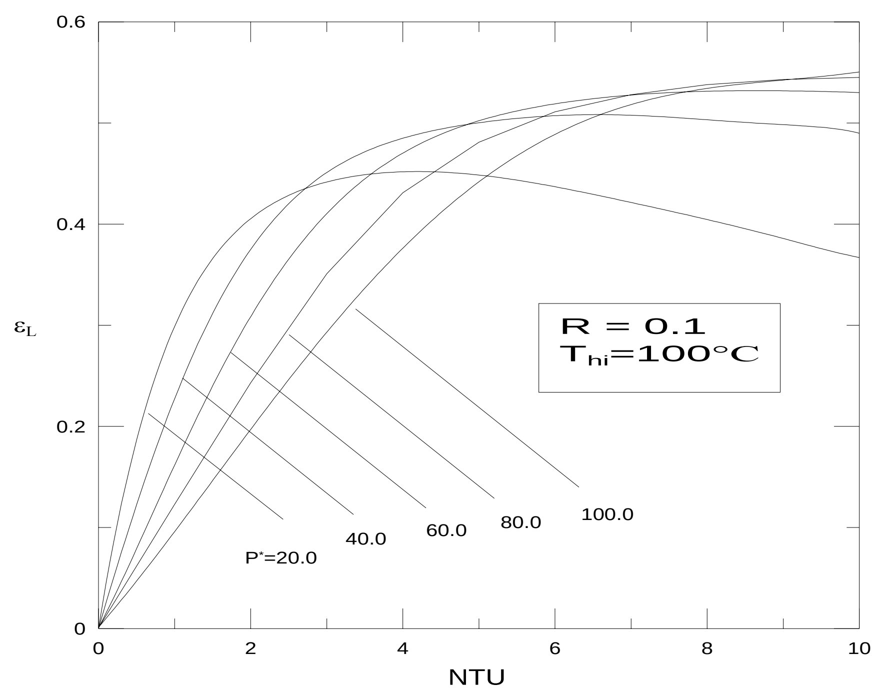

Figure 7: Influence of the regeneration temperature

4. Conclusion

A mathematical model for a rotary dehumidifier was developed and solved, the results showing that a balance between the properties of commercially available adsorptive materials is desirable, if the dehumidification effectiveness is to be maximized. It was also shown that for a given material, the effectiveness is strongly dependent on operation parameters such as the angular velocity and the regeneration temperature. This reinforces the need of a computational code as an important aid to the HVAC designer.

5. References

Bullock, C.E., Trelkheld, J.L., 1966, “Dehumidification of Moist Air by Adiabatic Adsorption”, Trans. ASHRAE, vol.72.

Jurinak, J.J.,1982, ”Open Cycle Solid Desiccant Cooling – Component Model and System Simulation” Ph.D. Thesis, The University of Winsconsin-Madison.

Maclaine-Cross,I.L.; Banks, P.J., 1972, “Coupled Heat and Mass Transfer in Regenerators”, Int. Journal of Heat and Mass Transfer,vol.15, 1972.

Simonson, J.C., Besant, R.W., 1997, “Heat and Moisture Transfer in Desiccant Coated Rotary Energy Exchangers Part I”, HVA&R Research, vol.3, No.4.

Zhang,X.J., et al., 2003, “A Simulation Study of Heat and Mass Transfer in a Honeycomb Rotary Dehumidifier, Applied Thermal Engineering, vol.23.

Zheng,W.; Worek, W.M.; 1993, “Numerical Simulation of Combined Heat and Mass transfer in a Rotary Dehumidifier”Num. Heat Transfer, A, vol.23, 1993.

6. Responsibility notice

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