SIMPLIFIED SIMULATION OF THE HEAT AND MASS TRANSFER IN A DESICCANT MEDIUM USED IN A DESICCANT-EVAPORATIVE COOLING SYSTEM

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ABSTRACT

This paper describes a simplified numerical simulation of the heat and mass transfer phenomena associated to both the humidification and dehumidification of the desiccant solid media used in air-conditioning installations. The algorithm is based on mass and energy balance equations, that are applied to the desiccant medium and to the airflows, and include the equilibrium moisture relationship for the desiccant. The convective mass and heat transfer coefficients are characterised by suitable correlations.

The developed model can be used to simulate a large set of working conditions, providing an efficient tool for studies of parametric sensibility, and for control and optimisation strategies.

KEYWORDS

Air-conditioning, rotary desiccant systems, numerical modelling

1-INTRODUCTION

The use of desiccant products in air-conditioning installations is a promising technique, which is able to replace in many cases the conventional refrigerating units. In the latter, the simultaneous cooling and dehumidification of the inlet air are obtained by contact with the cold surfaces of direct-expansion evaporators or of heat exchangers cooled by chilled water [1].

In the case of open cycle desiccant air conditioning systems, the desiccant medium is used to remove moisture from humid air. The dried air is then cooled through evaporative coolers and sensible heat exchangers to meet both sensible and latent air condition loads. Solar energy or other sources can be used to regenerate the desiccant, which represents an interesting possibility for diversification of energy demand [2].

A schematic representation of an air conditioning unit using a desiccant medium is shown in Figure 1. Outdoor clean air is dried and heated in the desiccant wheel B (evolution 0-1), cooled in the sensible heat regenerative wheel D (evolution 1-2), cooled in the evaporative cooler G (evolution 2-3) and then delivered into the room. The exhaust air is cooled by evaporation in a similar device E (evolution 4-5), in order to provide a low-temperature airflow that will cool the regenerative wheel D (evolution 5-6). The exhaust air is then heated (evolution 6-7) by a hot source C – using solar energy or some thermal effluent –, and is finally used to regenerate the desiccant wheel B (evolution 7-8) before being expelled outdoors. The just mentioned evolutions are represented in Figure 2 on a schematic psychrometric chart.

It is obvious that the global thermal behaviour of such a system depends on all particular processes that occur in a typical air conditioning unit. However, the behaviour of the desiccant medium is a crucial issue that needs a specific treatment, based on a dynamic analysis of the simultaneous heat and mass transfer.

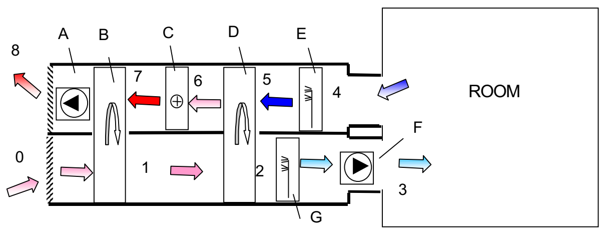

Figure 1 - Scheme of an air conditioning system with a desiccant medium and evaporative cooling.

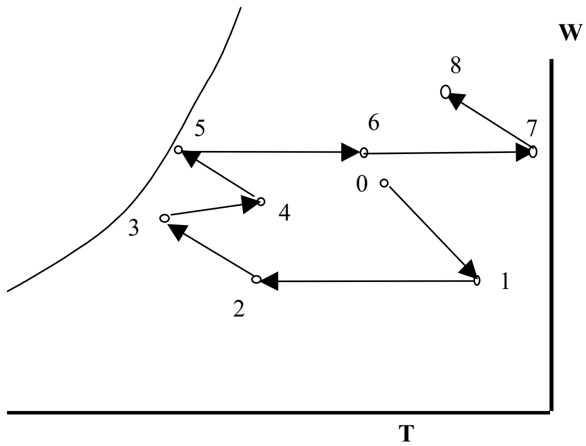

Figure 2 - Schematic representation of the air conditioning process with a desiccant medium and evaporative cooling.

2-MATHEMATICAL FORMULATION

The aim of this study is to propose an analytical model, and the corresponding numerical solution, to describe the higrothermal interaction between the airflows and the desiccant device.
The developed model enables the characterisation of the time and space evolutions of temperature and moisture content of both the desiccant wheel and the airflows. The desiccant medium is interpreted as a rotating cylindrical porous medium, with each half being crossed by counter axial flows, as represented in Figures 1 and 3.
The desiccant wheel rotates at a low velocity f (Hz). It is a porous medium characterised by a high value of transfer area to volume ratio, a_v, and a very low thickness of solid matrix.

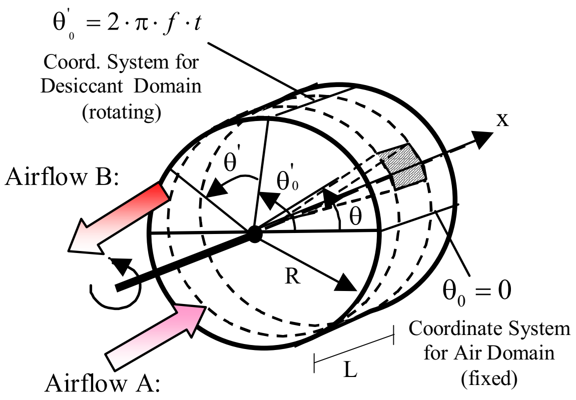

Figure 3 - Desiccant wheel.

A schematic representation of the discretization of the medium is shown in Figure 3. In each elementary volume, the temperature and the moisture content are assumed as uniformly distributed. The heat and mass transfer by diffusion within the solid matrix is considered to be negligible in the radial and axial directions. In each air pass, the flow is considered fully axial with uniform and constant velocity v_a.
The convective heat and mass transfer between the airflows and the porous medium is evaluated through specified heat and mass transfer coefficients, h_C and h_D, respectively. In order to evaluate the potential for the mass transfer, the relative humidity of the air on the surface of the porous medium is assumed by the model to be equal to that of the equilibrium condition at the moisture content and temperature of the desiccant medium.
The following mathematical formulation describes the system of equations, and its method of resolution, to characterise the interaction between the desiccant and both airflows. In the present study, an analysis is made of the system behaviour for a given set of specified initial conditions, namely the values of temperature and moisture content, together with constant inlet conditions for both airflows.

Referring to Figure 3, the initial conditions of the present problem are specified as follows:

$$t = 0 \wedge 0 \le x \le L \wedge 0 \le \theta \le 2\pi \Rightarrow \begin{cases} T_a(t,\theta,x) = T_0 \\ W_a(t,\theta,x) = W_0 \end{cases} \quad (1.a)$$

$$t = 0 \wedge 0 \le x \le L \wedge 0 \le \theta' \le 2\pi \Rightarrow \begin{cases} T_P\left(t,\theta',x\right) = T_0 \\ X_P\left(t,\theta',x\right) = X_0 \end{cases} \quad (1.b)$$

For the airflow A (e.g., outdoor, clean air), the inlet conditions are:

$$t > 0 \ \wedge \ x = 0 \ \wedge \ \pi \le \theta < 2\pi \ \Rightarrow \begin{cases} T_a(t,\theta,x) = T_{A\ in} \\ W_a(t,\theta,x) = W_{A\ in} \end{cases} \quad (2)$$

As for the airflow B (e.g., exhaust air), the inlet conditions are:

$$t > 0 \ \wedge \ x = L \ \wedge \ 0 \le \theta < \pi \Rightarrow \begin{cases} T_a(t,\theta,x) = T_{B\ in} \\ W_a(t,\theta,x) = W_{B\ in} \end{cases} \quad (3)$$

The fixed and the rotating reference systems, for the domains of the airflow and of the desiccant, respectively, are shown in Figure 3. They are related by:

$$\theta' = \theta - 2\cdot\pi\cdot f\cdot t = \theta - 2\cdot\pi\cdot t\frac{n}{3600}\cdot \quad (4)$$

The convective mass and heat transfers at the outer surface of the porous medium are evaluated by the following expressions, respectively:

$$v_a\cdot\frac{\partial W_a(t,\theta,x)}{\partial x} = h_D\cdot a_V\left[W_P\left(t,\theta',x\right) - W_a(t,\theta,x)\right] \quad (5)$$

$$\rho_a\, v_a\, c_{P_a}\frac{\partial T_a(t,\theta,x)}{\partial x} = h_C\, a_V\left[T_P\left(t,\theta',x\right) - T_a(t,\theta,x)\right] \quad (6)$$

The mass conservation principle is described by the following equation

$$\rho_a\cdot v_a\cdot\frac{\partial W_a(t,\theta,x)}{\partial x} + \rho_P\cdot\frac{\partial X_P\left(t,\theta',x\right)}{\partial t} = 0 \quad (7)$$

The energy conservation is written as:

$$\begin{aligned} &\rho_a\cdot v_a\frac{\partial\left(c_{P_a}\cdot T_a(t,\theta,x)\right)}{\partial x} + \\ &+\rho_a\cdot v_a\frac{\partial\left(W_a(t,\theta,x)\cdot\left(h_{fg} + c_{P_V}\cdot T_a(t,\theta,x)\right)\right)}{\partial x} + \\ &+\rho_P\cdot c_{P_P}\frac{\partial T_P\left(t,\theta',x\right)}{\partial t} + \\ &+\rho_P\cdot c_{P_L}\frac{\partial\left(X_P\left(t,\theta',x\right)\cdot T_P\left(t,\theta',x\right)\right)}{\partial t} = 0 \end{aligned} \quad (8.a)$$

$$
\begin{aligned}
&\rho_a \cdot v_a \cdot c_{p_a} \cdot \frac{\partial T_a(t,\theta,x)}{\partial x} + \\
&+ \rho_a \cdot v_a \cdot h_{fg} \frac{\partial W_a(t,\theta,x)}{\partial x} + \\
&+ \rho_a \cdot v_a \cdot c_{pV} \frac{\partial \left(T_a(t,\theta,x) \cdot W_a(t,\theta,x)\right)}{\partial x} + \\
&+ \rho_P \cdot c_{PP} \cdot \frac{\partial T_P\left(t,\theta',x\right)}{\partial t} + \\
&+ \rho_P \cdot c_{PL} \cdot \frac{\partial \left(X_P\left(t,\theta',x\right) \cdot T_P\left(t,\theta',x\right)\right)}{\partial t} = 0
\end{aligned}
\tag{8.b}
$$

The higrothermal equilibrium between the air and desiccant medium is evaluated by a typical expression relating the relative humidity of the air with the temperature and the moisture content of the desiccant medium. The present model uses the Henderson-Thompson equation [6]:

$$
\phi_P\left(t,\theta',x\right) = 1.0\text{-}\exp\left[-c_1 \cdot \left(T_P\left(t,\theta',x\right) + c_2\right) \cdot \left(\frac{X_P\left(t,\theta',x\right)}{0.01}\right)^{c_3}\right] \tag{9}
$$

To obtain the air moisture content, the following psychrometric relation is used:

$$
W_P\left(t,\theta',x\right) = \frac{0.622 \cdot \phi_P\left(t,\theta',x\right)}{\dfrac{P}{P_{VS}\left(T_P\left(t,\theta',x\right)\right)} - \phi_P\left(t,\theta',x\right)} \tag{10}
$$

The model evaluates the average values of the air temperature and of the air moisture content after each outlet section of the desiccant (fully mixed airflows), using the following expressions:

$$
\begin{cases} t > 0 \\ x = L \\ \pi \le \theta < 2\pi \end{cases} \Rightarrow \begin{cases} T_{A\ out} = \dfrac{1}{\pi} \displaystyle\int_0^{\pi} T_a(t,\theta,x)\, d\theta \\ W_{A\ out} = \dfrac{1}{\pi} \displaystyle\int_0^{\pi} W_a(t,\theta,x)\, d\theta \end{cases} \tag{11}
$$

$$
\begin{cases} t > 0 \\ x = 0 \\ 0 \le \theta < \pi \end{cases} \Rightarrow \begin{cases} T_{B\ out} = \dfrac{1}{\pi} \displaystyle\int_{\pi}^{2\pi} T_a(t,\theta,x)\, d\theta \\ W_{B\ out} = \dfrac{1}{\pi} \displaystyle\int_{\pi}^{2\pi} W_a(t,\theta,x)\, d\theta \end{cases} \tag{12}
$$

This is the set of equations that must be solved by a suitable algorithm to obtain the temperature and moisture content evolutions of the desiccant and of the two airflows.

3-RESULTS

Some simulations were performed to show the potentialities of the model. The results presented in this paper were obtained with the following relevant data:

Inlet conditions for airflow A
$T_{A\ in}$ from 20 ºC to 35ºC $W_{A\ in}$=0.010 kg/kg db

Inlet conditions for airflow B
$T_{B\ in}$ 40 ºC to 90ºC $W_{B\ in}$=0.013 kg/kg db

Desiccant Wheel
Rot. speed: n= 10 rph Total frontal Area: A_f=1 m^2 Thickness: L=0.1 m Apparent density: ρ_P=500 kg/m^3 Specific transfer area: a_v=900 m^2/m^3 Dried desiccant specific heat: c_{PP}=1300 J/kg.ºC Equilibrium constants: c_1=0.00011, c_2=10.0, c_3=1.45

Convective transfer coefficients (Mean air flow velocity: v_a=2m/s)	
Heat transfer: h_C = 7.73 W/m^2.ºC	Mass transfer: h_D = 6.44×10^{-3}m/s

Figures 4 and 5 show the average values of temperature and moisture content, respectively, of both airflows at the outlet of the desiccant wheel. In this calculation, the porous medium initial conditions were 50ºC and 0.2 kg/kg db., for the temperature and the moisture content, respectively. The inlet air temperature was fixed at 30ºC, for flow A, and at 80ºC, for flow B.

The plots of Figures 6, 7, 8 and 9 show the steady-state solutions obtained with the present model, in terms of the average outlet temperature and moisture content of each airflow, for different values of the inlet air temperature of flows A and B.

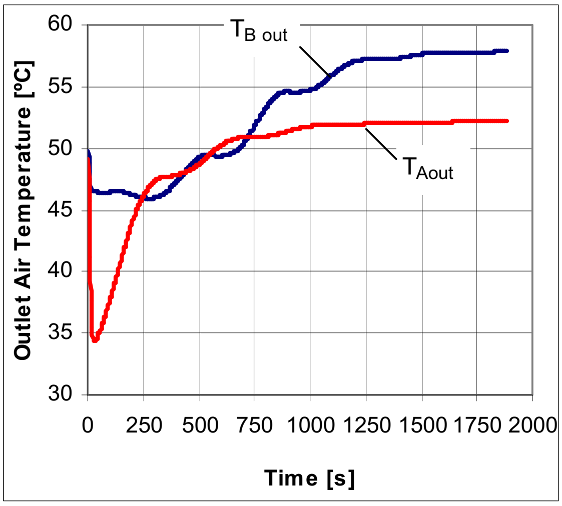

Figure 4 - Time evolutions of the outlet temperatures of airflows A and B (fully mixed)

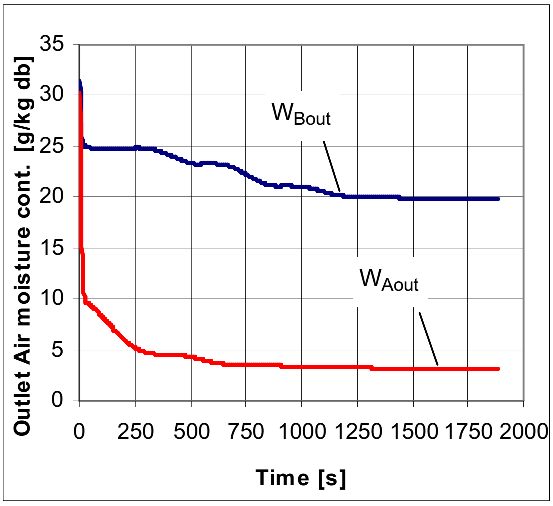

Figure 5 - Time evolutions of the outlet moisture contents of airflows A and B (fully mixed)

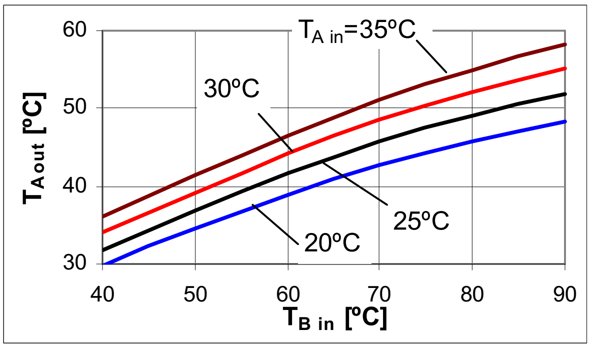

Figure 6 - Steady-state air temperature at the outlet of flow A, as a function of the inlet temperatures (WAin=0.01 kg/kg db, WBin=0.013 kg/kg db)

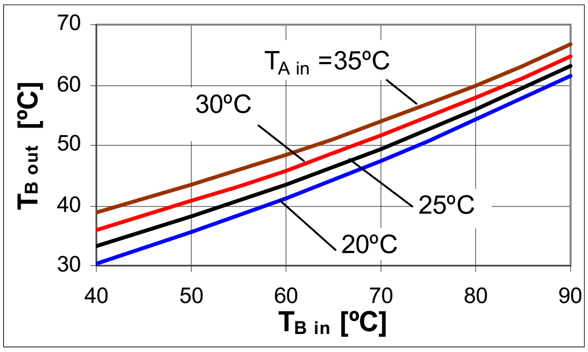

Figure 7 - Steady-state air temperature at the outlet of flow B, as a function of the inlet temperatures (WAin=0.01 kg/kg db, WBin=0.013 kg/kg db)

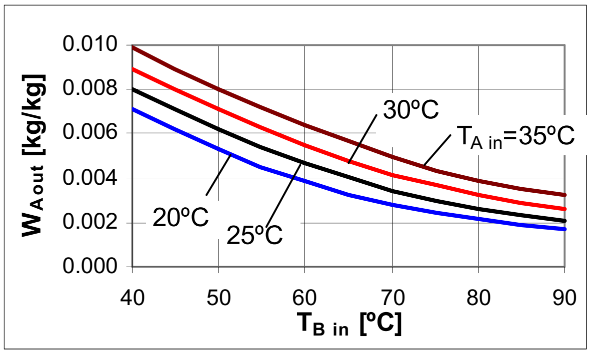

Figure 8 - Steady-state moisture content of air at the outlet of flow A (WA in=0.010 kg/kg db, WBin=0.013 kg/kg db)

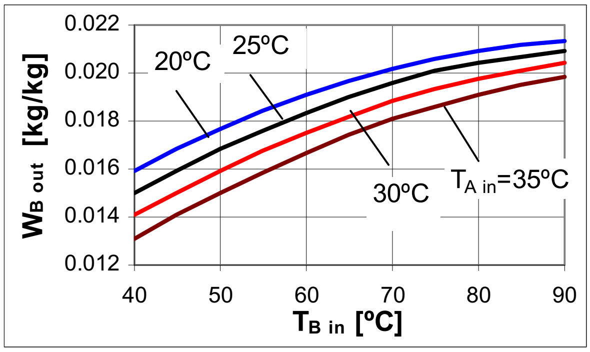

Figure 9 - Steady-state moisture content of air at the outlet of flow B (WA in=0.010 kg/kg db, WBin=0.013 kg/kg db)

4-CONCLUSIONS

The global higrothermal transient behaviour of a desiccant medium was modelled by a simplified technique that provides physically consistent and realistic results. The model can be used in parametric studies to preview the behaviour of the desiccant wheel, and it can provide a useful tool for preliminary design purposes of this type of installations.

Furthermore, the present model can also be integrated in a larger one, aiming to globally simulate the performance of air conditioning systems incorporating a desiccant medium, as well as evaporative cooling, rotary exchanger, heating coil, using, for example, solar energy or a heat recovery source from a particular process.

Experimental research is planned for a close future, in order to improve and validate the model. Particular attention will be addressed to the determination of the heat and mass transfer coefficients, the isothermal equilibrium curves of the desiccant medium, and some other parameters such as: void fraction, specific transfer area, specific mass and specific heat of the porous medium.

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NOMENCLATURE

A_f	Total frontal area, m^2
a_v	Specific transfer area, m^2/m^3
c_{P_a}	Constant-pressure specific heat of air, J/kg·ºC
c_{P_L}	Specific heat of liquid water, J/kg·ºC
c_{P_P}	Specific heat of the product (desiccant), J/kg·ºC
c_{P_V}	Specific heat of steam water, J/kg·ºC
c_1, c_2, c_3	Constants in the Henderson-Thompson equation
f	Rotation velocity, Hz
h_C	Convection heat transfer coefficient, W/m^{2}ºC
h_D	Convection mass transfer coefficient, m/s
h_{fg}	Vaporisation latent heat of water, J/kg
L	Length, m
n	Rotation velocity, rph
P	Atmospheric pressure, Pa
P_{VS}	Saturation pressure of water steam, Pa
t	Time, s
T_a	Air temperature, ºC
$T_{A\ in}$	Inlet temperature of airflow A, ºC
$T_{B\ in}$	Inlet temperature of airflow B, ºC
$T_{A\ out}$	Outlet temperature of airflow A, ºC
$T_{B\ out}$	Outlet temperature of airflow B, ºC
T_P	Temperature of the product, ºC
T_0	Initial temperature, ºC
v_a	Airflow velocity, m/s
W_a	Moisture content of air, kg/ kg db (dry basis)
$W_{A\ in}$	Inlet moisture content of airflow A, kg/ kg db
$W_{A\ out}$	Outlet moisture content of airflow A, kg/ kg db
$W_{B\ in}$	Inlet moisture content of airflow B, kg/ kg db
$W_{B\ out}$	Outlet moisture content of airflow B, kg/ kg db
W_P	Moisture content of air at higrothermal equilibrium conditions with the product, kg/ kg db
W_0	Initial moisture content of air, kg/ kg db
x	Axial coordinate, m
X_P	Product moisture content, kg/ kg db
X_0	Initial moisture content of the product, kg/ kg db
ϕ_P	Relative humidity of air at higrothermal equilibrium conditions with the product
θ	Angle (fixed coord. system – airflow domain), rad
θ'	Angle (rotating coord. system for the product domain), rad
ρ_a	Air density, kg/m^3
ρ_P	Apparent density of the product, kg/m^3

RÉSUMÉ

Cet article concerne une étude de simulation numérique des phénomènes de transfert de chaleur et de masse dans les roues dessicantes utilisés dans les centrales de traitement d'air avec refroidissement évaporatif. L'algorithme du modèle simplifié est basé sur les équations de conservation d'énergie et de masse, appliquées à la matrice solide hygroscopique et aux écoulements d'air, utilisant les courbes d'équilibre matrice solide-air humide et des coefficients d'échange de chaleur et de masse estimés par des expressions empiriques.

Le modèle est un outil efficient pour faire des études de sensibilité paramétrique et définir le système de contrôle, ainsi que pour l'optimisation des stratégies de fonctionnement.