Modeling and simulation of desiccant wheel for air conditioning

Fatemeh Esfandiari Nia a,*, Dolf van Paassen b,1, Mohamad Hassan Saidi a,2

a Mechanical Engineering Department, Sharif University of Technology, Tehran 11365-8639, Iran
b Mechanical Maritime and Materials Engineering, Delft University of Technology, 2628 CD Delft, The Netherlands

Received 29 January 2006; received in revised form 3 March 2006; accepted 4 March 2006

Abstract

This paper presents the modeling of a desiccant wheel used for dehumidifying the ventilation air of an air-conditioning system. The simulation of the combined heat and mass transfer processes that occur in a solid desiccant wheel is carried out with MATLAB Simulink. Using the numerical method, the performance of an adiabatic rotary dehumidifier is parametrically studied, and the optimal rotational speed is determined by examining the outlet adsorption-side humidity profiles. The solutions of the simulation at different conditions used in air dehumidifier have been investigated according to the previous published studies. The model is validated through comparison the simulated results with the published actual values of an experimental work. This method is useful to study and modelling of solid desiccant dehumidification and cooling system. The modeling solutions are used to develop simple correlations for the outlet air conditions of humidity and temperature of air through the wheel as a function of the physically measurable input variables. These correlations will be used to simulate the desiccant cooling cycle in an HVAC system in order to define the year round efficiency.

# 2006 Elsevier B.V. All rights reserved.

Keywords: Desiccant; Adsorption; Cooling; Modeling; Simulation; Correlation

1. Introduction

In desiccant cooling processes, fresh air is dehumidified and then sensibly and evaporatively cooled before being sent to the conditioned space. Since this technique works without conventional refrigerants, such as fluorocarbons and also since it allows the use of low-temperature heat (low-temperature industrial waste heat or solar energy) to drive the cooling cycle, it attracted increased attention especially in America, Japan, Europe and China [1]. Desiccants remove moisture from the surrounding air until they reach equilibrium with it. This moisture can be removed from the desiccant by heating it to temperatures around 60–90 °C and exposing it to a regenerative air stream. The desiccant is then cooled so that it can adsorb moisture again. Desiccant cooling cycles are particularly useful if they are used in humid regions.

The major advantage of desiccant cooling is significant potential for energy savings and reduced consumption of fossil fuels. The electrical energy requirement can be very low comparing with conventional refrigeration systems. The source of thermal energy can be diverse (i.e., solar, waste heat, natural gas) [2]. Having low coefficient of performance (COP) can be considered as the main disadvantage for desiccant cooling systems. COP values of 0.8–1 are commonly predicted for this cycle. COP or coefficient of performance is defined as the space cooling load divided by thermal energy required to regenerate the desiccant. Some investigators use the heat removed from the processes air stream divided by the thermal energy required to regenerate the desiccant.

Kang and Maclain-Cross [3] showed that the dehumidifier is the key component of a desiccant cooling system and the cooling COP (coefficient of performance) can be significantly improved by improving the performance of this component. Since the introduction of this technology, much research on the solid desiccant dehumidifiers has been accomplished. Maclaine-Cross and Banks [4] developed an analogy method for predicting the coupled heat and mass transfer process in desiccant dehumidifier wheel. Consecutively, Banks [5–7] analyzed the coupled heat and mass transfer processes in a porous medium using a nonlinear analogy method and predicted the performance of a silica gel air dryer. Neti and Wolf have reported [8] that the analogy method appears to be
good for only a small range of conditions. Van Den Bulck et al. [9,10] introduced a wave analysis to establish a one-dimensional transient model of the rotary heat and mass exchanger with infinite transfer coefficients, and with definite ones employing the \( \varepsilon \)-NTU method. Pesaran and Mills [11,12] established a solid-side resistance model and pseudo-gas-side controlled model to study the vapor transfer process in the silica gel packed bed, their studies revealed that the solid-side resistance model fits the experimental data much better than the pseudo-gas-side controlled model. Zheng and Worek [13] discussed the effect of rotary speed on the performance of the desiccant wheel by numerical simulation using an implicit finite differential method. Dai et al. [14] used a finite difference model to analyze and explain the “concentration” wave, “thermal” wave and middle zone point in the desiccant wheel in detail, and the rules to improve the performance of dehumidification were discussed using psychrometric charts. Zhang and Niu [15] presented a two-dimensional coupled heat and mass transfer model which takes into account the heat conduction, the surface and gas diffusion in both axial and radial directions and compared the performances of desiccant wheels as dehumidifier and enthalpy recovery. Chauch et al. [16] studied the relative importance of gas phase and solid-side resistance using two film models and showed the importance of solid-side resistances parametrically in small gap passages and when the gel size is large.

The purpose of this paper is to develop a model and obtaining the model solutions for heat and mass transfer in a desiccant wheel to study performance of a desiccant air-conditioning system. The numerical solutions for coupled nonlinear heat and mass transfer equations require significant computational efforts to calculate outlet air states. Correlating functions of temperature and humidity with input conditions are developed which allow simulation of the performance of components in air-conditioning cycles. Yearly simulation runs can then be made. This requires simple and fast correlation functions to limiting the computing time to an acceptable level. These functions are developed and can be used to model the cycle.

2. Governing equations

The dehumidifier is a rotating cylindrical wheel of length \( L \) and radius \( R \) with small channels which walls are adhered with an adsorbent such as silica gel. For simplicity it is divided into two equal sections: the adsorbing section and the regeneration section (desorption of water vapor). The regeneration and adsorption air streams are in a counter flow arrangement. The schematic of a balanced rotary dehumidifier is illustrated in Fig. 1 and the analysis is based on the following assumptions:

1. axial heat conduction and water vapor diffusion in the air are negligible;
2. axial molecular diffusion within the desiccant is negligible;
3. there are no radial temperature or moisture content gradients in the matrix;
4. hysteresis in the sorption isotherm for the desiccant coating is neglected and the heat of sorption is assumed constant;
5. the channels that make up the wheel are identical with constant heat and mass transfer surface areas;
6. the matrix thermal and moisture properties (support material/desiccant and adsorbed water) are constant;
7. the channels are considered adiabatic and impermeable;
8. the mass and heat transfer coefficients are constant;

<table>
<thead>
<tr>
<th>Nomenclature</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
</tr>
<tr>
<td>A_e</td>
</tr>
<tr>
<td>A_d</td>
</tr>
<tr>
<td>A_g</td>
</tr>
<tr>
<td>C</td>
</tr>
<tr>
<td>COP</td>
</tr>
<tr>
<td>d</td>
</tr>
<tr>
<td>D_h</td>
</tr>
<tr>
<td>h</td>
</tr>
<tr>
<td>h_m</td>
</tr>
<tr>
<td>H</td>
</tr>
<tr>
<td>k</td>
</tr>
<tr>
<td>L</td>
</tr>
<tr>
<td>Le</td>
</tr>
<tr>
<td>N</td>
</tr>
<tr>
<td>Nu</td>
</tr>
<tr>
<td>P</td>
</tr>
<tr>
<td>q_st</td>
</tr>
<tr>
<td>R</td>
</tr>
<tr>
<td>t</td>
</tr>
<tr>
<td>T</td>
</tr>
<tr>
<td>U</td>
</tr>
<tr>
<td>w</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Greek letters</th>
</tr>
</thead>
<tbody>
<tr>
<td>e</td>
</tr>
<tr>
<td>( \varphi )</td>
</tr>
<tr>
<td>( \rho )</td>
</tr>
<tr>
<td>( \omega )</td>
</tr>
<tr>
<td>( \omega_s )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Subscripts</th>
</tr>
</thead>
<tbody>
<tr>
<td>d</td>
</tr>
<tr>
<td>e</td>
</tr>
<tr>
<td>g</td>
</tr>
<tr>
<td>i</td>
</tr>
<tr>
<td>l</td>
</tr>
<tr>
<td>m</td>
</tr>
<tr>
<td>o</td>
</tr>
<tr>
<td>R</td>
</tr>
<tr>
<td>s</td>
</tr>
<tr>
<td>st</td>
</tr>
<tr>
<td>v</td>
</tr>
</tbody>
</table>

Van Den Bulck et al. [9,10] introduced a wave analysis to establish a one-dimensional transient model of the rotary heat and mass exchanger with infinite transfer coefficients, and with definite ones employing the \( \varepsilon \)-NTU method. Pesaran and Mills [11,12] established a solid-side resistance model and pseudo-gas-side controlled model to study the vapor transfer process in the silica gel packed bed, their studies revealed that the solid-side resistance model fits the experimental data much better than the pseudo-gas-side controlled model. Zheng and Worek [13] discussed the effect of rotary speed on the performance of the desiccant wheel by numerical simulation using an implicit finite differential method. Dai et al. [14] used a finite difference model to analyze and explain the “concentration” wave, “thermal” wave and middle zone point in the desiccant wheel in detail, and the rules to improve the performance of dehumidification were discussed using psychrometric charts. Zhang and Niu [15] presented a two-dimensional coupled heat and mass transfer model which takes into account the heat conduction, the surface and gas diffusion in both axial and radial directions and compared the performances of desiccant wheels as dehumidifier and enthalpy recovery. Chauch et al. [16] studied the relative importance of gas phase and solid-side resistance using two film models and showed the importance of solid-side resistances parametrically in small gap passages and when the gel size is large.

The objective of this paper is to develop a model and obtaining the model solutions for heat and mass transfer in a desiccant wheel to study performance of a desiccant air-conditioning system. The numerical solutions for coupled nonlinear heat and mass transfer equations require significant computational efforts to calculate outlet air states. Correlating functions of temperature and humidity with input conditions are developed which allow simulation of the performance of components in air-conditioning cycles. Yearly simulation runs can then be made. This requires simple and fast correlation functions to limiting the computing time to an acceptable level. These functions are developed and can be used to model the cycle.

2. Governing equations

The dehumidifier is a rotating cylindrical wheel of length \( L \) and radius \( R \) with small channels which walls are adhered with an adsorbent such as silica gel. For simplicity it is divided into two equal sections: the adsorbing section and the regeneration section (desorption of water vapor). The regeneration and adsorption air streams are in a counter flow arrangement. The schematic of a balanced rotary dehumidifier is illustrated in Fig. 1 and the analysis is based on the following assumptions:

1. axial heat conduction and water vapor diffusion in the air are negligible;
2. axial molecular diffusion within the desiccant is negligible;
3. there are no radial temperature or moisture content gradients in the matrix;
4. hysteresis in the sorption isotherm for the desiccant coating is neglected and the heat of sorption is assumed constant;
5. the channels that make up the wheel are identical with constant heat and mass transfer surface areas;
6. the matrix thermal and moisture properties (support material/desiccant and adsorbed water) are constant;
7. the channels are considered adiabatic and impermeable;
8. the mass and heat transfer coefficients are constant;
9. the adsorption heat per kilogram of adsorbed water is constant;
10. the carry over between two air flows is neglected.

Based on the above assumptions, the model used in this analysis is transient and one-dimensional.

One of the channels is divided into a number of equal step discrete elements or channels as shown in Fig. 2. For each discretized channel that identified in Simulink model as a framework with inlet conditions for air and storing outputs and initial condition for solid, as shown in Fig. 2, the energy and mass conservation equations can be written as follows. Mass transfer equation for the air stream:

$$\frac{d}{dr} \left( \frac{\rho_g A_g L}{\rho_g} \right) = U_g A_g \rho_g (\omega_i - \omega_o) + h_m A_c (\omega_s - \omega)$$  \hspace{1cm} (1)

$$\frac{d\omega}{dr} = U_g (\omega_i - \omega_o) + h_m A_c (\omega_s - \omega)$$  \hspace{1cm} (2)

where

$$\omega = \frac{\rho_s}{\rho_g}$$  \hspace{1cm} (3)

$$A_c = \frac{2L}{D_t}$$  \hspace{1cm} (4)

$$C_1 = \frac{U_g}{L}$$  \hspace{1cm} (5)

$$C_2 = \frac{h_m A_c}{\rho_g L A_g}$$  \hspace{1cm} (6)

Heat transfer equation for the air stream:

$$\frac{d(\rho_g A_g L C_g T_g)}{dr} = \rho_g U_g A_g C_g (T_g - T_{go}) + h_A (T_s - T_g)$$  \hspace{1cm} (7)

$$\frac{dT_g}{dr} = C_1 (T_g - T_{go}) + C_3 (T_s - T_g)$$  \hspace{1cm} (8)

$$C_3 = \frac{h_A}{\rho_g L A_g C_g}$$,  \hspace{1cm} (9)

Mass transfer equation for solid desiccant layer:

$$\frac{d(\rho_d w A_d L)}{dr} = h_m A_c (\omega - \omega_s)$$  \hspace{1cm} (10)

where \( w \) is the water content of desiccant material,

$$\frac{\rho_w}{\rho_d} = w$$  \hspace{1cm} (11)

So

$$\frac{dw}{dr} = \frac{h_m A_c}{\rho_d A_d L} (\omega - \omega_s)$$  \hspace{1cm} (12)

For having the equations according to the variables \( \omega_s, T_s, \omega, T \), it can be written as

$$dw = \frac{\partial w}{\partial \varphi} \frac{\partial \varphi}{\partial \omega_s} d\omega_s + \left( \frac{\partial w}{\partial \varphi} \frac{\partial \varphi}{\partial T} + \frac{\partial w}{\partial T} \right) dT_s$$  \hspace{1cm} (13)

or

$$dw = S1(\omega_s, T_s) d\omega_s + S2(\omega_s, T_s) dT_s$$  \hspace{1cm} (14)

where

$$S1(\omega_s, T_s) = \frac{\partial w}{\partial \varphi} \frac{\partial \varphi}{\partial \omega_s}$$, \hspace{1cm} (15)

$$S2(\omega_s, T_s) = \left( \frac{\partial w}{\partial \varphi} \frac{\partial \varphi}{\partial T} + \frac{\partial w}{\partial T} \right)$$

So mass transfer equation for desiccant layer will become

$$\frac{d\omega_s}{dr} = \frac{S2(\omega_s, T_s)}{S1(\omega_s, T_s)} dT_s + \frac{h_m A_c}{\rho_d A_d L S1(\omega_s, T_s)} (\omega - \omega_s)$$  \hspace{1cm} (16)

$$\frac{d\omega_s}{dr} = - \frac{S2(\omega_s, T_s)}{S1(\omega_s, T_s)} dT_s + \frac{C_4}{S1(\omega_s, T_s)} (\omega - \omega_s)$$  \hspace{1cm} (17)

where

$$C_4 = \frac{h_m A_c}{\rho_d L A_d}$$  \hspace{1cm} (18)

$$A_c = \frac{4D_h L}{D_h + d_v^2 - D_h^2}$$  \hspace{1cm} (19)

and \( Le \) is Lewis number that here is assumed equal 1 for air stream.
Heat transfer for solid desiccant layer:

\[
\frac{d}{dt} \left( \rho_d A_d L c_d T_s \right) = q_{st} \rho_d A_d L \frac{d\omega}{dt} + h A_c (T_g - T_s) \tag{20}
\]

\[
\frac{dT_s}{dt} = \frac{h_A c_d}{\rho_d A_d L c_d} (\omega - \omega_s) + \frac{h A_c}{C_d \rho_d A_d L} (T_g - T_s)
\]

\[
= C_5 (\omega - \omega_s) + C_6 (T_g - T_s) \tag{21}
\]

\[
C_5 = \frac{q_{st}}{C_d}
\]

\[
C_6 = \frac{h A_c}{C_d \rho_d A_d L} \tag{22}
\]

Relative humidity and saturation pressure can be calculated by [15]

\[
\psi = \frac{\omega_s P_0}{(0.622 + \omega_s) P_s} \tag{24}
\]

\[
P_s = 10^P_0 \exp \left( \frac{5294}{T_s} \right) \left( 1 + 1.61 \omega_s \right) \frac{(0.622 + \omega_s)}{(25)}
\]

For first channel (element) in adsorption period the initial conditions are known as

\[
T_{gi} = T_{\text{process air inlet}} = T_g \text{initial} \tag{26}
\]

\[
\omega_{gi} = \omega_{\text{process air inlet}} \tag{27}
\]

\[
\omega_{ai} = \omega_s \text{ regarding to inlet air Temperature} \tag{28}
\]

\[
T_{i\text{initial}} = T_{\text{process air inlet}} \tag{29}
\]

and initial conditions for regeneration period:

\[
T_{gi} = T_{\text{regeneration air inlet}} \tag{30}
\]

\[
\omega_{gi} = \omega_{\text{regeneration air inlet}} \tag{31}
\]

\[
\omega_{ai} = \omega_s \text{ regarding to inlet air Temperature} \tag{32}
\]

\[
T_{i\text{initial}} = T_{\text{process air inlet}} \tag{33}
\]

3. Simulation platform and subsystems

The model used in this simulation has been build up based on a main framework for each element shown in Fig. 2 that can be regarded as an interaction of four main subsystems. In Fig. 3, a block diagram shows the set-up of the simulation. Each subsystem is composed of a series of other, more detailed, subsystems. The calculation blocks are storing the response of an output. For simulation of the rotation of the wheel between two counter flow adsorption and regeneration air flow, the convenient user defined functions regarding the time step of revolution has been built up and is jointed to the main framework for each element. There are a number of techniques to solve the derived differential equations. The selected scheme of solutions in MATLAB for initial value problems is Runge–Kutta methods with variable time steps.

The output conditions of an individual channel can be considered as average values for each section of the wheel. The time step and the time each channel of wheel passes through the air stream can be considered to evaluate of the output conditions in different wheel angels.

4. Evaluation of the coefficients and step sizes

Regarding the derived nonlinear equations of combined mass and heat transfer we have many coefficients that are functions of $T_{\text{air}}, \omega_{\text{air}}$ that can be considered constant with considering the average values of them. It improves the accuracy and reduces the calculation time. Some of the coefficients are functions of $T_s, \omega_s$.

The present model was validated by comparing the simulated results with the published experimental values [17]. The adsorption capacity of a desiccant material is not only dependent on the relative humidity of the surrounding air but also on its temperature. Thus, the relation of capacity to both
temperature and relative humidity must be known. It is convenient to find a function that involves both of these parameters and allows prediction of the desiccant adsorption capacity at any given temperature–humidity combination. A correlation for silica gel given in the literature [18] is

\[ W = 0.106 \exp \left[ -\left( \frac{A}{8590} \right)^2 \right] + 0.242 \exp \left[ -\left( \frac{A}{3140} \right)^2 \right], \]

\[ A = -RT \ln \varphi \]  

(34)

So for Eq. (15) it can be written:

\[ S_1(\omega, T_s) = (2.9e - 9)A \frac{RT_s}{\varphi} \exp \left[ -\left( \frac{A}{8590} \right)^2 \right] \]

\[ + (4.9e - 8)A \frac{RT_s}{\varphi} \exp \left[ -\left( \frac{A}{3140} \right)^2 \right] \times \left[ -\left( \frac{A}{3140} \right)^2 \right] \left[ \frac{0.622P_0}{(0.622 + \omega_s)^2P_s} \right] \]  

(35)

\[ S_2(\omega, T_s) = \frac{\partial \omega}{\partial \varphi} \frac{\partial \varphi}{\partial T_s} + \frac{\partial \omega}{\partial T_s} \]

\[ = \left( 2.9e - 9 \right)A \frac{RT_s}{\varphi} \exp \left[ -\left( \frac{A}{8590} \right)^2 \right] \]

\[ + (4.9e - 8)A \frac{RT_s}{\varphi} \exp \left[ -\left( \frac{A}{3140} \right)^2 \right] \frac{5294\varphi}{T_s^2} \]

\[ + \left( 2.9e - 9 \right)A R \ln \varphi \exp \left[ -\left( \frac{A}{8590} \right)^2 \right] \]

\[ + (4.9e - 8)A R \ln \varphi \exp \left[ -\left( \frac{A}{8590} \right)^2 \right] \]  

(36)

In the experimental study [15], desiccant wheel has been used as a dehumidifier and the wheel has 20 cm width and the silica gel wall thickness is 0.2 mm. The pitch of the honeycomb shaped rotor used for experiments has been 3.2 mm × 1.8 mm. The heat of adsorption is calculated between 2100 and 2300 kJ/kg. The air velocity 2 m s\(^{-1}\) for air flow in both adsorption and regeneration periods have been reported. The important parameter in the modeling of the desiccant wheels is rotational speed or the time step which can be determined regarding to the time passed during each revolution of the wheel. As a simplifying assumption, the wheel in this modeling is considered a balanced wheel with two equal cross area and air flow rate for each part. Therefore the time step is two times more than the time needed for one complete matrix rotation. For desiccant dehumidifier wheel, the rotational speeds are in the range 1–30 revolution per hour (RPh). The number of time elements is determined by the maximum time of running the program and the time increments that are variable from 0.001 to 0.1 s. The number of the elements is 50 for 0.2 m length of the wheel.

The heat transfer coefficient is calculated by a correlation for forced convection in internal flows. As stated by Bejan [19], the Nusselt number is approximately between 3.63 and 4.364 for this laminar flow arrangement and the hydraulic diameter of the hexagonal passes is estimated to be 2.33 × 10\(^{-3}\) m. The heat transfer coefficient becomes

\[ h = \frac{Nu_{\text{air}} k_{\text{air}}}{D_h} = 43.8 \, \text{W m}^{-2} \text{K}^{-1} \]  

(37)

5. The behavior of solutions

The rotational speed of a rotary desiccant dehumidifier is inversely proportional to the sorption time. The rotational speed of a rotary desiccant dehumidifier is optimum when the average outlet humidity ratio of the process air flow is the minimum. A rotary desiccant dehumidifier operates such that the moisture absorbed during the adsorption process must be desorbed during the regeneration process, so when the average outlet humidity ratio in the adsorption stream is optimized, the average outlet humidity ratio of the regeneration stream is maximized. In other words, the adsorption and regeneration processes will be optimum at the same rotational speed. According to the published studies [13] the behavior of the Simulink model output can be estimated.

When a desiccant wheel rotates much faster than optimum speed, the adsorption and regeneration processes are too short, which results in poor performance. Also, when the rotational speed is low, the adsorption and regeneration processes are too long and less effective. The outlet humidity ratio profiles on the adsorption-side are shown in Fig. 4A–C. In Fig. 4C the outlet humidity profile at faster rotational speed than optimum speed (the case for the adsorption time is insufficient, is plotted versus the wheel angle). This profile shows that the outlet humidity ratio at the last point, or at an angle of 180, is less than the average value. This means that at the end of adsorption process, the desiccant wheel can still efficiently dehumidify the air stream.

Therefore the rotational speed should be slower to allow more adsorption time. In Fig. 4A the outlet humidity profile is in slower speeds than optimum rotational speed. In this case the outlet humidity at the end of adsorption is larger than the average value; this implies that the last portion of the adsorption process is inefficient. The rotational speed of the desiccant wheel should increase to remove this last ineffective portion of the adsorption process. Therefore, to improve dehumidification performance, the rotational speed should be in between the cases in Fig. 4A and C. As shown in Fig. 4B, at the optimum rotational speed, the outlet humidity ratio at the end of the adsorption process, or at an angle of 180, is equal to the average value. This result [13] is used to investigating the optimum rotational speeds of desiccant wheel, as the most important design parameter of the wheel, when operating conditions change.

Fig. 5 shows the prediction of the optimal rotation speed at various regeneration temperatures and compares the accuracy of the simulation results with experimental work [17]. As it can be seen in Fig. 5 the model can predict the
optimum speed of the wheel with very good accuracy in most conditions.

In addition the model has very good accuracy regarding to the temperatures and humidity of the outlet air in different conditions. The result of modeling presented here and the published experimental data agree within the acceptable margin of error. The user friendly and flexible programming environment of Simulink provides an excellent alternative. The advantages of such modeling platform include notion of adaptability, tenability to generate accurate solutions that are significantly concise with minimum calculation time as well.

6. Sensitivity of the individual parameters

In this section the effect and the importance of the individual parameters such as mass flow rate or air velocity, wheel speed, diameter of air passages, desiccant thickness and inlet conditions of air temperature and humidity for process air stream as well as inlet air temperature of regeneration stream on the outlet air conditions and dehumidification effectiveness will be examined. The evaluation is carried out for a dehumidifier wheel with silica gel as desiccant material and the sorption curve used in the simulation is a correlation for silica gel given in the literature [18] or Eq. (34). The dehumidifier effectiveness is defined as

$$e = \frac{\omega_{\text{inlet air}} - \omega_{\text{outlet air}}}{\omega_{\text{inlet air}}}$$  \hspace{1cm} (38)

Fig. 6a shows that the dehumidifier effectiveness decreases with the inlet air temperature increases. As shown in Fig. 7a, outlet air temperature increases with inlet air temperature. The wheel speed is one of the most important parameters in the dehumidification itself and in many other applications. The results in Figs. 6c and 7c suggest that the wheel speed affects the effectiveness and outlet air temperature and therefore on the outlet air relative humidity and enthalpy considerably. The dehumidification effectiveness for a desiccant wheel depends on the regeneration temperature as indicated in Fig. 6b. Fig. 7b shows the increase of outlet air temperature with regeneration air temperature increases. It can be proposed a simple linear function to show the variation of outlet air temperature with
inlet air temperature. Regarding to modeling in this paper and the mass transfer equations, the moisture diffusion in solid layer has been neglected. This simplification is not a good assumption when the gap or air passage diameter is small or the thickness is large [16]. Therefore for constant air passage gap for the solutions here (2.33 mm) there is limitation to select the range for size of wall thickness. The initial conditions of humidity for inlet air affects on the effectiveness and outlet temperature as it has been shown in Figs. 6g and 7g. In the dehumidifier the inlet air for both process air and regeneration air have same absolute humidity but different temperatures. The simulation data for different mass flow rate according to different air velocity are shown in Figs. 6f and 7f. The variation of outlet air conditions with mass flow rate is considerably high and it can be considered as an important parameter. According to the simulation data, the effect of air passages on the wheel effectiveness and outlet temperature on effectiveness and temperature shown in Figs. 6d and 7d. Regarding to modeling and the mass transfer equations in this paper, the moisture diffusion in solid layer has been neglected. And as mentioned before in this section, for constant wall thickness in the solutions here (0.2 mm) there is limitation to select the range for the size of air passages.

7. The correlations for effectiveness and outlet air conditions

In this section, the correlations for dehumidification effectiveness and outlet air temperature are presented according to the sensitivity of individual parameters presented in Figs. 6 and 7. The correlations presented in this section were determined by minimizing the maximum difference between over 500 simulated effectiveness and temperature values. The coefficients of the correlations were determined using optimization routine in Matlab.

The correlation for outlet air temperature is proposed by Eq. (39) and in Fig. 8 it has been shown comparing with the
outlet air temperatures from the simulation results:

\[ T_{out} = g_1(N)g_2(T_i)g_3(d_i)g_4(T_R)g_5(\omega_i)g_6(D_h)g_7(U), \]

\[ g_1(N) = -0.0002N^2 + 0.0112N + 0.4201, \]
\[ g_2(T_i) = -0.0001T_i^2 + 0.0275T_i + 0.7993, \]
\[ g_3(d_i) = -18.79d_i^2 + 7.92d_i + 1.75, \]
\[ g_4(T_R) = -0.0004T_R^2 + 0.1255T_R + 0.6757, \]
\[ g_5(\omega_i) = 594.48\omega_i^2 + 26.76\omega_i + 3.79, \]
\[ g_6(D_h) = -0.039D_h^2 + 0.026D_h^4 + 0.603D_h + 0.0912, \]
\[ g_7(U) = -0.060U + 0.7973 \quad (39) \]

The maximum difference between simulated and correlated outlet air temperature is \( \pm 2\% \). There is a significant error for some of the conditions of wheel and air. It is because the correlation can be proposed for only a limited range of individual parameters selected to obtain simulated data. The coefficients calculated from optimization routine are different in different range of parameters. Therefore the correlations proposed here can be considered for only the range of the conditions that covers the selected data to make that correlations and the modeling of cycle needs more correlations.

The correlation for outlet air humidity is derived through the following relation:

\[ \omega_{out} = \omega_{in} - \omega_{in} \quad (40) \]

The optimization can be proposed by polynomial fitting of the variables and propose a correlation as given in Eq. (41). Fig. 9b shows the outlet air absolute humidity calculated with Eqs. (40) and (41) that is approached by polynomial equations. The maximum difference between simulated and correlated outlet air humidity is \( \pm 2\% \). These equations are for silica gel.
Fig. 9. (a) Wheel efficiency calculated with Eq. (40) compared to the simulated efficiency; (b) outlet air absolute humidity calculated with Eqs. (40) and (41) compared to the simulated outlet air humidity.

wheel with the sorption curve given in Ref. [18]:

\[
e = f_1(N) f_2(T_i) f_3(d_i) f_4(T_R) f_5(\omega_i) f_6(D_h) f_7(U),
\]

\[
f_1(N) = -0.0001N^2 + 0.0042N + 0.4474,
\]

\[
f_2(T_i) = -0.0001T_i^2 - 0.0031T_i + 0.8353,
\]

\[
f_3(d_i) = -21.67d_i^2 + 6.93d_i + 1.34,
\]

\[
f_4(T_R) = -0.0001T_R^2 + 0.0355T_R - 0.4924,
\]

\[
f_5(\omega_i) = 592.77\omega_i^2 - 41.23\omega_i + 1.283,
\]

\[
f_6(D_h) = -0.0572D_h^2 + 0.0933D_h^2 + 0.6139D_h - 0.0922,
\]

\[
f_7(U) = -0.0611U + 0.8376
\]

(41)

8. Conclusions

In this paper a heat and mass transfer model has been presented which predicts the temperature and humidity states of the outlet air from a desiccant wheel and the optimum speed of the wheel when used as a dehumidifier. Furthermore, it has shown that the results of simulation and the published experimental data are agreeable within an acceptable margin of error. The Simulink code provides the possibility to couple the simulation model of one component such as this case, the drying wheel with models of total system. Another advantage is the tools for model reduction. Here the reduced physical module could be applied into simple correlation functions. This makes the simulation of complex systems such as the HVAC systems faster when year round analyses are needed. The modeling approach is adapted for a commercial cooling cycle, through development of simple correlation functions, to calculate the outlet air states according to individual parameters. The maximum difference between the results from the proposed correlations and simulation results are within ±2%, provided that the limited range of the variables are considered.

References

学霸图书馆
www.xuebalib.com

本文献由“学霸图书馆-文献云下载”收集自网络，仅供学习交流使用。

学霸图书馆（www.xuebalib.com）是一个“整合众多图书馆数据库资源，提供一站式文献检索和下载服务”的24小时在线不限IP图书馆。

图书馆致力于便利、促进学习与科研，提供最强文献下载服务。

图书馆导航：
图书馆首页 文献云下载 图书馆入口 外文数据库大全 疑难文献辅助工具