ABSTRACT

This paper presents the development of an equation based model to simulate the combined heat and mass transfer in the desiccant wheels. The performance model is one dimensional in the axial direction. It applies a lumped formulation in the thickness direction of the desiccant and the substrate. The boundary conditions of this problem represent the inlet outside/process and building exhaust/regeneration air conditions as well as the adiabatic condition of the two ends of the desiccant composite. The solutions of this model are iterated until the wheel reaches periodic steady state operation. The modeling results are obtained as the changes of the outside/process and building exhaust/regeneration air conditions along the wheel depth and the wheel rotation.

This performance model relates the wheel’s design parameters, such as the wheel dimension, the channel size and the desiccant properties, and the wheel’s operating variables, such as the rotary speed and the regeneration air flowrate, to its operating performance.

The impact of some practical issues, such as wheel purge, residual water in the desiccant and the wheel supporting structure, on the wheel performance has also been investigated.

INTRODUCTION

Desiccant wheels, including the enthalpy recovery and the active desiccant wheels, are commonly used devices in Heating Ventilating and Air Conditioning (HVAC) system. The performance of desiccant wheels can be modeled by a set of equations, which represent the conservation of energy and mass, the rates of heat and moisture transfer, and the adsorption equilibrium. The equations have been solved either by applying the analogy between the heat and mass transfer or numerically by using finite difference and finite volume techniques.

In early 1970’s, the modeling of the enthalpy recovery wheel was based on the analogy between heat and mass transfer. When the computational power was limited, using analogy between heat and mass transfer is necessary because the solutions for heat transfer are readily available. However, this advantage became less significant with the development of computers and numerical techniques[1].

Van den Bulck and Mitchell et al[2,3] developed an Effectiveness-Number of Transfer Units (ε-NTU) method for the active desiccant wheel. This model was then used to explore the impacts of different operating variables, such as the regeneration air flowrate, regeneration temperature and wheel rotary speed, in order to maximize the wheel performance based on the thermal and electrical energy input[4].

Zheng and Worek[5] developed a one dimensional transient model to simulate the simultaneous heat and mass transport processes involved in the rotary desiccant dehumidifier. The simulation results were compared with the predictions from other programs. As applications of this model, the impacts of certain design parameters, such as the separation factor and the maximum moisture uptake of the desiccant material and the number of transfer units, and certain operating variables, such as the inlet temperature and humidity ratio of the process and regeneration air, on the rotary dehumidifier performance were investigated[6,7,8].

Simonson and Besant[9,10] presented a one dimensional transient model to simulate the heat and moisture transfer in enthalpy wheels. Different from previous publications, they assumed that part of the heat of adsorption was conducted to the desiccant matrix and the rest was convected to the air stream. The model was validated using experimental data and reasonable agreement between the experiment and the model predictions was achieved.

The above models only included the convective heat and mass transfer resistance at the gas solid interface. Besides the
resistance at the interface, the solid side resistance for heat and mass transfer were also considered in Majumdar’s model[11]. The moisture transport in the desiccant matrix was represented by gas diffusion and surface diffusion resistances. Zhang and Niu[12] and Sphaier and Worek[13] presented two dimensional models that considered both heat and mass transfer resistances in both axial and thickness directions of the solid desiccant. These two models were claimed to be applicable for both the enthalpy recovery and the active desiccant wheels. Both models gave some insights into the heat and mass transfer processes in the desiccant matrix, which was not provided in previous publications. However, both models are more based on abstracted mathematics than physical understanding of the desiccant material. Whether the models truly described what is going on in reality remains unknown. As a matter of fact, very limited validation information, if at all, was provided in these papers.

In addition, none of the previous models considered the practical issues in the wheel operation, which include the wheel purge, the residual water contained in the desiccant material and the wheel supporting structure such as the spokes and the casing.

An equation based model has been developed to predict the operating performance of both the enthalpy recovery and the active desiccant wheels, based upon fundamental scientific and engineering principles. This model relates the desiccant wheel’s performance to its design parameters and operating conditions.

The design parameters include:
- the wheel dimension, such as the wheel depth, the wheel diameter and the split between adsorption and desorption sections;
- the channel dimension, such as the channel shape and size;
- the desiccant composite, such as the desiccant material properties and coating thickness, as well as the substrate properties and thickness.

The operating variables include:
- the rotary speed of the wheel;
- the inlet outside/process air temperature, humidity and flowrate;
- the building exhaust/regeneration air temperature, humidity and flowrate.

The practical issues considered in this model include:
- wheel purge;
- residual water contained in the desiccant materials;
- wheel supporting structure, such as wheel spokes and casing.

The results from the model prediction are:
- the temperature and humidity of the outside/process and building exhaust/regeneration air at any given time and location as well as the average values.

The model can be used:
- to explore different design alternatives of the enthalpy recovery and the active desiccant wheels;
- to predict the operating performance of the desiccant wheels once the wheel design and operating conditions are given;
- to select operating variables of the desiccant wheels for a given application.

**NOMENCLATURE**

**Abbreviations:**
- HVAC Heating, Ventilating and Air Conditioning
- NTU Number of Transfer Unit
- rpm revolution per minute

**Parameters/Variables:**
- $A$ cross sectional area of the airflow channel, the desiccant or substrate layer, m$^2$
- $c$ separation factor
- $C_p$ specific heat, J/kg-K
- $h$ enthalpy or convective heat transfer coefficient, J/kg or W/m$^2$-K
- $h_m$ convective mass transfer coefficient, kg/m$^2$-s
- $\Delta H$ heat of adsorption or vaporization, J/kg or kJ/kg
- $k$ thermal conductivity, W/m-K
- $L$ depth of the desiccant wheel, m
- $m$ mass flowrate, kg/s
- $N_u$ Nusselt number
- $p$ perimeter length of the airflow channel or pressure, m or Pa
- $R$ universal gas constant, $R = 8.314$ J/mole-K
- $t$ time or temperature, s or °C
- $T$ temperature in Kelvin, K
- $u$ air velocity, m/s
- $x$ distance in axial direction, m
- $\gamma$ moisture loading in the desiccant, kg moisture/kg dry desiccant
- $\varepsilon$ sensible, latent heat or enthalpy recovery effectiveness, regeneration efficiency, heat carryover ratio
- $\rho$ density, kg/m$^3$
- $\delta$ thickness of the desiccant layer, m
- $\phi$ relative water vapor concentration

**Subscripts:**
- amb ambient
- c carryover
- g air
- in inlet
- l latent
- m desiccant matrix or mass transfer
- max maximum
- min minimum
MODEL DEVELOPMENT

Problem Formulation

The schematic of the desiccant wheel and a cross section of its airflow channel used in the model development are shown in Fig. 1. The wheel is a rotating cylinder with depth L and diameter D. It rotates around its axis at a constant speed and the rotation of the wheel is represented by \( \alpha \). The wheel is split into the adsorption and desorption sections and the area of the two sections are not necessarily equal. The wheel is sometimes built with a purge section to prevent cross-contamination between the outside/process and the building exhaust/regeneration air streams, which is discussed later in this paper. The outside/process and the building exhaust/regeneration air streams are in counter flow arrangement in order to achieve improved performance of a rotating wheel.

The structure and composition of the wheel are assumed homogeneous. All the channels in the wheel are assumed identical. The wheel performance is modeled by tracking the air and desiccant conditions in a single channel as it rotates.

The channel can be of any shape, such as sinusoidal, rectangular or circular. The channel wall is made of desiccant composite, which consists of desiccant materials, and perhaps substrate to support the desiccants. Air flows through the channels in the direction of the wheel axis, exchanging energy and moisture with the desiccant composite.

The controlling process for the moisture transfer in the desiccant wheels is the moisture transport from the bulk air flowing in the channels to the hypothetical surface of the desiccant layer, due to the small thickness of the desiccant layer. For the same reason, both the heat and the mass transfer Biot numbers are small for this problem. Therefore, the temperature and moisture concentration gradient in the thickness direction (r direction in Fig. 1) is very small. Therefore the analysis can be considered as one dimensional in the axial direction (x direction in Fig. 1).

The model is formulated in the same way for both adsorption and desorption sections. Depending on the airflow direction, the air velocity is either positive or negative. Depending on the vapor concentration in the air and the desiccant, the moisture either adsorbs onto or desorbs from the desiccant. Depending on the temperature of the air and the desiccant, the energy either transfers from the air to the desiccant or from the desiccant to the air.

Model Assumptions

- The axial heat conduction and water vapor diffusion in the air are negligible.
- The axial water vapor and adsorbed water diffusion in the desiccant are negligible.
- The convective heat and mass transfer rates are represented using the bulk mean air temperature and humidity.
- Heat conduction in the desiccant is negligible. Heat may be conducted axially through the substrate.
- The mid plane, indicated as dash lines in Fig. 1 and two ends of the desiccant composite are adiabatic and impermeable.
- The airflow in the channel is fully developed laminar flow.
- The heat of adsorption is released in the desiccant composite.
- The inlet air conditions are uniform across the wheel surface, but they can vary with time.
• Thermodynamic properties of the dry air, desiccant material, and substrate, such as density, specific heat and heat of adsorption, remain constant during the wheel operation.
• The convective heat and mass transfer coefficients remain constant during the wheel operation. They are determined based on published coefficients between gases and solid surfaces.
• There is no heat or moisture storage in the wheel when it completes one rotation.

In essence, the model is one dimensional in the axial direction. It is transient, which means it calculates the time dependent conditions of the air and the desiccant composite. It models the simultaneous and coupled heat and mass transfer effect occurring in the wheel. It applies to both the enthalpy recovery and the active desiccant wheels, since none of the assumptions and the following governing equations is specific to either application. Furthermore, the wheel is in periodic steady-state operation, which means the wheel returns to its original condition in terms of temperature and moisture loading after one complete cycle.

**Governing Equations**

The governing equations, which describe the material and energy balance as well as the heat and mass transfer rates, are developed based on above assumptions.

Since there is no accumulation and depletion of dry air and the dry air properties remain constant during the wheel operation, the air velocity remains constant. The moisture balance of the air stream can be written as:

\[
h_{\text{m}} p (\rho_{\text{eg}} - \rho_{\text{eg}}) + u A \frac{\partial \rho_{\text{eg}}}{\partial x} + A \frac{\partial \rho_{\text{eg}}}{\partial t} = 0
\]  

(1)

The moisture balance of the desiccant is described as:

\[
h_{\text{m}} p (\rho_{\text{eg}} - \rho_{\text{eg}}) - A_{\text{m}} \rho_{\text{m}} \frac{\partial \gamma_{\text{m}}}{\partial t} = 0
\]  

(2)

The energy balance of the air stream is described as:

\[
h p (t_{\text{g}} - t_{\text{g}}) - u A p C_{p} \frac{\partial t_{\text{g}}}{\partial x} - A p C_{p} \frac{\partial t_{\text{g}}}{\partial t} = 0
\]  

(3)

The energy balance of the desiccant composite is described as:

\[
k_{\text{m}} A_{\text{m}} \frac{\partial^{2} \gamma_{\text{m}}}{\partial x^{2}} + h_{\text{m}} p (\rho_{\text{eg}} - \rho_{\text{eg}}) \Delta H_{\text{ads}} - h p (t_{\text{m}} - t_{\text{m}}) -
\]

\[
(\rho_{\text{m}} A_{\text{m}} C_{p} + \rho_{\text{eg}} A_{\text{eg}} C_{p}) \frac{\partial t_{\text{m}}}{\partial t} = 0
\]  

(4)

**Boundary Conditions**

The boundary conditions required to solve the previous governing equations include:

• the inlet outside/process and the building exhaust/regeneration air temperature and water vapor concentration

\[
t_{\text{g}} \big|_{x=0} = t_{\text{pair, in}}
\]  

(5)

\[
\rho_{\text{eg}} \big|_{x=0} = \rho_{\text{eg, pair, in}}
\]  

(6)

\[
t_{\text{g}} \big|_{x=L} = t_{\text{pair, in}}
\]  

(7)

\[
\rho_{\text{eg}} \big|_{x=L} = \rho_{\text{eg, pair, in}}
\]  

(8)

• the adiabatic and impermeable conditions of the two ends of the desiccant composite

\[
\frac{\partial t_{\text{g}}}{\partial x} \big|_{x=0} = \frac{\partial t_{\text{g}}}{\partial x} \big|_{x=L} = 0
\]  

(9)

\[
\frac{\partial \rho_{\text{eg}}}{\partial x} \big|_{x=0} = \frac{\partial \rho_{\text{eg}}}{\partial x} \big|_{x=L} = 0
\]  

(10)

As stated previously, the inlet air conditions can vary with time, but they are uniform across the wheel surface. The adiabatic and impermeable conditions are required in solving the energy and moisture balance equations of the desiccant composite.

The simulation is started with some initial conditions, but they do not influence the final modeling results since the wheel is in periodic steady-state operation.

**The Adsorption Isotherm of the Desiccant**

There are five unknown variables in the four governing equations: temperature of the air \(t_{\text{g}}\), temperature of the desiccant composite \(t_{\text{m}}\), water vapor concentration in the air \(\rho_{\text{eg}}\), water vapor concentration in equilibrium with the desiccant \(\rho_{\text{vm}}\) and moisture loading in the desiccant \(\gamma_{\text{m}}\). Another equation is therefore needed in order to solve for the five unknown variables. The fifth equation is the desiccant adsorption isotherm, which relates the moisture loading in the desiccant with the relative water vapor concentration of the air that is in equilibrium with the desiccant.

A general adsorption isotherm is described as[14,15]:

\[
\frac{\gamma_{\text{m}}}{\gamma_{\text{max}}} = \frac{1}{1 - e^{c \phi}}
\]  

(11)

\[
\phi = \frac{\rho_{\text{vm}}}{\rho_{\text{vm, sat}}}
\]  

(12)

The relative humidity can be represented by the partial pressure of water vapor based on the ideal gas law.

\[
\phi = \frac{p_{\text{vm}} R T}{p_{\text{vm, sat}}}
\]  

(13)

\(p_{\text{vm, sat}}\) is a function of temperature only. The relationship between \(p_{\text{vm, sat}}\) and \(t_{\text{m}}\) can be determined by applying the Clausius-Clapeyron equation.

\[
\ln \left( \frac{p_{\text{vm}}} {p_{2}} \right) = \frac{\Delta H_{\text{ads}}}{R} \left( \frac{1}{T_{2}} - \frac{1}{T_{1}} \right)
\]  

(14)

Assuming at the standard atmospheric pressure 101,325 Pa, water boils at 373.15 K (100°C), the saturation vapor pressure \(p_{\text{vm, sat}}\) at temperature \(T_{\text{m}}\) can be calculated as...
where $T_m = T_m + 273.15$ K.

The following relationship is obtained by combining Equations (12) and (15).

$$\phi = 4.09 \times 10^{-9} T_m \rho_{sat} e^{T_m\beta}$$  \hspace{1cm} (16)

Equation (11) relates the moisture loading of the desiccant material with its temperature and water vapor concentration through Equation (16). It supplements the governing equations shown in Equation (1) through Equation (4) and completes the formulation of the combined heat and mass transfer problem in the performance modeling of desiccant wheels.

In summary, the governing equation set consists of:
- four partial differential equations, which describe the energy and moisture balance of the air and the desiccant composite;
- one nonlinear algebraic equation, which represents the desiccant adsorption isotherm.

This governing equation set is subject to the boundary conditions listed in the previous section, which represent the inlet outside/process and building exhaust/regeneration air conditions as well as the adiabatic condition of the two ends of the desiccant composite. The initial conditions are not critical since the wheel is in periodic steady-state operation.

The convective heat transfer coefficient is calculated based on the Nusselt number for fully developed laminar flow in a tube with constant heat flux boundary conditions[16].

$$h = \frac{k_m \text{Nu}}{D_{ef}}$$  \hspace{1cm} (17)

The convective mass transfer coefficient is determined by applying the heat and mass transfer analogy[16]. Assuming unity Lewis number, which is related to the mechanisms for heat and mass transfer, $h_m$ is obtained as

$$h_m = \frac{h}{\rho_e \rho_e C_p}$$  \hspace{1cm} (18)

According to Simonson[1], the periodic steady state operation of the wheel can be determined either by the periodic conditions of the wheel or the conservation of energy and moisture across the wheel. The latter is a more stringent criterion than the former at typical operating conditions. Therefore, the conservation of energy and moisture across the wheel is used to determine the steady state operation in this performance model. In developing the model, the periodic steady state operation is defined as

$$\left| \frac{q_{gain} - q_{loss}}{q_{gain}} \right| \leq 3\%$$

$$\left| \frac{m_{gain} - m_{loss}}{m_{gain}} \right| \leq 3\%$$

The above model is programmed using explicit finite difference method. The temperature and water vapor concentration of the outside/process and the building exhaust/regeneration air, the temperature and moisture loading of the desiccant composite as functions of time and space are obtained as the modeling results, which are illustrated in the following section.

**MODELING RESULTS**

This performance model is applicable to both the enthalpy recovery and the active desiccant wheels. The design parameters of the enthalpy recovery and active desiccant wheels used in the simulation are shown in Table 1. The heat and mass transfer coefficients are relatively low compared to other heat exchangers, but the desiccant wheel makes up for this by the very high heat and mass transfer areas. Experimental confirmation of these transfer coefficients and discussions of numerical uncertainties can be found in [19].

This model can be used to simulate the performance of desiccant wheels with various design parameters and operating conditions. Those shown in this section are only one example.

The adsorption isotherms of the 3Å molecular sieves used in the enthalpy recovery wheel and the silica gel used in the active desiccant wheel are plotted in Fig. 2.

![Fig. 2 Adsorption Isotherm of 3Å Molecular Sieves and Silica Gel Used in the Simulation](image-url)

The psychrometric chart representation of the enthalpy recovery wheel operation, the inlet and predicted average outlet air conditions, is shown in Fig. 3. The temperature and humidity of the outside air are reduced after going through the wheel; those of the building exhaust air are increased. The performance of the enthalpy recovery wheel is represented by two parallel lines on the psychrometric chart, since the quantities of the outside and the building exhaust air are equal.

The schematic of the enthalpy recovery wheel used in the simulation is shown in Fig. 4. The outside air enters the wheel from one end $x=0$. The building exhaust air enters it from the other end $x=L$. The wheel split ratio, defined as the ratio between the face area for adsorption and the entire face area,
is 1/2. The adsorption and desorption sections are separated by brush seals. The wheel rotates clockwise at 30 rotations per minute (rpm).

Table 1. Design Parameters of the Desiccant Wheels Used in the Simulation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Enthalpy Recovery Wheel</th>
<th>Active Desiccant Wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel diameter (mm)</td>
<td>787</td>
<td>787</td>
</tr>
<tr>
<td>Wheel depth (mm)</td>
<td>152</td>
<td>102</td>
</tr>
<tr>
<td>Wheel split ratio</td>
<td>1/2</td>
<td>2/3</td>
</tr>
<tr>
<td>Rotary speed (rpm)</td>
<td>30</td>
<td>0.42</td>
</tr>
<tr>
<td>Channel size (mm)</td>
<td>1.8*4.2</td>
<td>1.5*3.4</td>
</tr>
<tr>
<td>Desiccant thickness (microns)</td>
<td>25</td>
<td>65</td>
</tr>
<tr>
<td>Substrate thickness (microns)</td>
<td>15</td>
<td>75</td>
</tr>
<tr>
<td>Desiccant material</td>
<td>3Å molecular sieves</td>
<td>Silica gel</td>
</tr>
<tr>
<td>Desiccant density (kg/m³)</td>
<td>760</td>
<td>700</td>
</tr>
<tr>
<td>Desiccant specific heat (J/kg-K)</td>
<td>1,000</td>
<td>1,000</td>
</tr>
<tr>
<td>Desiccant conductivity (W/m-K)</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Separation factor</td>
<td>0.1</td>
<td>1</td>
</tr>
<tr>
<td>Maximum moisture loading (kg/kg)</td>
<td>0.2</td>
<td>0.36</td>
</tr>
<tr>
<td>Heat of adsorption (J/kg)</td>
<td>2,791,000</td>
<td>2,791,000</td>
</tr>
<tr>
<td>Substrate material</td>
<td>Aluminum</td>
<td>Glass fiber paper</td>
</tr>
<tr>
<td>Substrate density (kg/m³)</td>
<td>2,700</td>
<td>500</td>
</tr>
<tr>
<td>Substrate specific heat (J/kg-K)</td>
<td>900</td>
<td>900</td>
</tr>
<tr>
<td>Substrate conductivity (W/m-K)</td>
<td>237</td>
<td>0</td>
</tr>
<tr>
<td>Convective heat transfer coefficient (W/m²-K)</td>
<td>36.2</td>
<td>43.3</td>
</tr>
<tr>
<td>Convective mass transfer coefficient (m/s)</td>
<td>0.030</td>
<td>0.036</td>
</tr>
</tbody>
</table>

The profiles of the air and desiccant conditions with respect to time, or the wheel rotation, are plotted for two complete cycles in Fig. 5 through Fig. 7. Since the wheel is in periodic steady state operation, the temperature and humidity conditions of the air and the desiccant wheel repeat themselves over one complete rotation.

Fig. 5 and 6 plot the profiles of the temperature of the air and the desiccant, and the water vapor concentration in the air and that at the air desiccant interface which is in equilibrium with the desiccant, at x=0 and x=L. In general, the temperature and water vapor concentration of the air and those in equilibrium with the desiccant at x=0 is higher than those at x=L, because the warmer and more humid outside air enters the wheel from x=0. The outside and building exhaust air outlet temperature and water vapor concentration are not uniform with the wheel rotation even though the inlet air streams are at uniform conditions.

Fig. 3 Psychrometric Chart Representation of the Enthalpy Recovery Wheel Operation

Fig. 4 Schematic of the Enthalpy Recovery Wheel Used in the Simulation

The changes of the outside and building exhaust air outlet temperature and water vapor concentration with the wheel rotation are directly related to the conditions of the desiccant. Since the outside air is warmer and more humid, heat and moisture are transferred from the outside air to the channels of the enthalpy recovery wheel. The temperature and water vapor concentration in the wheel increase. Therefore the wheel’s abilities to reduce the temperature and water vapor concentration of the incoming outside air stream decrease. The outside air outlet temperature and water vapor concentration therefore increase with the wheel rotation.

Since the building exhaust air is cooler and less humid, heat and moisture are transferred back from the wheel to the building exhaust air stream. The temperature and water vapor concentration in the wheel decrease. The wheel’s abilities to raise the temperature and water vapor concentration of the incoming building exhaust stream decrease with the wheel.
The building exhaust outlet temperature and water vapor concentration decrease accordingly.

The temperature and water vapor concentration of the desiccant appear to change linearly with time in one section of the wheel, but not linearly in another section. This phenomenon can be explained by the change of heat and mass transfer rates between the air and the desiccant. In the adsorption section, the differences in temperature and water vapor concentration between the incoming outside air and the desiccant at \( x=0 \) are at their largest values at the beginning of the cycle, the rotation angle \( \alpha=0^\circ \). The differences become smaller with the wheel rotation; so do the rates of heat and mass transfer. Therefore, the rates of change in the temperature of the desiccant and the water vapor concentration at the interface between the air and the desiccant become smaller. In the desorption section, the differences in temperature and water vapor concentration between the building exhaust air outlet and the desiccant at \( x=0 \) appear constant; so do the rates of heat and mass transfer. Therefore, the rates of change in the temperature and water vapor concentration of the desiccant remain constant. The changes in the temperature and water vapor concentration of the desiccant at \( x=L \) can be explained in a similar way.

When the wheel switches between the adsorption and desorption sections, the air contained in the channels of the wheel gets carried over. The directions of the airflow are different for the adsorption and desorption sections. Therefore, in a small region of the wheel the incoming outside air enters the building exhaust air outlet stream and the incoming building exhaust air enters the outside air outlet stream. This is presumably what happens when a purge section is not included in the wheel design. The setup of this model inherently includes this air carryover effect. The conditions of the wheel and the air contained in the channels of the wheel are preserved, and the directions of the airflow are reversed when the wheel switches between the adsorption and desorption sections.

When the wheel rotates from the building exhaust to the outside air section, for a short period of time the air leaving the wheel is the cooler and less humid building exhaust air contained in the channels of the wheel, which is represented by the outlet air conditions from 0 to 11.25 degree angle in Fig. 5 and Fig. 6. After that period, the outside air leaves the wheel at its characteristic outlet conditions. Its temperature and water vapor concentration increase gradually with the wheel rotation. Similarly, the effect of the air carryover from the outside to the building exhaust air section is represented by the warmer and more humid outlet air conditions from 180 to 191.25 degree angle. The 11.25 degree rotation angle corresponds to 0.0625 seconds for the enthalpy recovery wheel rotating at 30 rpm. The face velocity of the outside and the building exhaust air stream is 2.4 m/s and the depth of the wheel is 0.152 m, therefore it takes 0.0625 seconds for the air contained in the channels to be cleared out.

The two dashed lines in Fig. 5 and Fig. 6 indicate the average outlet outside and building exhaust air conditions, 25.1°C dry bulb temperature, 0.0104 kg/m³ water vapor concentration and 27.8°C, 0.0131 kg/m³, respectively. Seen from Fig. 5 and Fig. 6, the average outside air outlet condition is close to the outside air outlet condition at 90° rotation angle. The average building exhaust outlet condition is close to the building exhaust outlet condition at 270° rotation angle, 90° angle from the separation seal. This finding is consistent with previous research, which recommended the enthalpy recovery wheel performance be calculated based on the measured data taken at one angular position[17].

The profile of the desiccant moisture loading, corresponding to the desiccant conditions shown in Fig. 5 and Fig. 6, is plotted in Fig. 7. There is very limited variation in the moisture loading in the desiccant during the operation of the enthalpy recovery wheel. The desiccant is close to its maximum moisture loading during the operation of the wheel, which is indicated as the straight line on top of the graph. The moisture contained in the desiccant is referred to as the residual water. Its impact on the enthalpy recovery wheel performance is explored in the next section.

\[
E_s = \frac{m_{OA}(T_{OA,in} - T_{OA,out})}{\min(m_{OA}(T_{OA,in} - T_{OA,out}))} = \frac{m_{RA}(T_{RA,in} - T_{RA,out})}{\min(m_{RA}(T_{RA,in} - T_{RA,out}))}
\]

(21)

**PERFORMANCE INDICATORS**

According to ASHRAE[18], sensible heat, latent heat or moisture, and total heat recovery effectiveness, defined as the actual transfer of sensible heat, latent heat or moisture, and total heat divided by the maximum possible transfer between airstreams, are commonly used performance indicators for enthalpy recovery devices in HVAC systems.
The sensible, latent and total heat recovery effectiveness determine the slope and the extent of the lines in the psychrometric chart representation in Fig. 3. For the enthalpy recovery wheel indicated in Table 1, the sensible, latent and total heat recovery effectiveness are 0.695, 0.710 and 0.705, respectively. Heat carryover ratio and regeneration efficiency are the two performance indicators developed for the active desiccant wheel. Their definitions are detailed in [19]. The simulation results and the profiles of the air and desiccant conditions for the active desiccant wheel can also be found in [19].

**DISCUSSION**

**Wheel Purge**

The enthalpy recovery wheel is usually built with a purge section, in order to prevent cross contamination between the building exhaust air outlet and outside air intake. The required purge area is dependent on the depth of the wheel, the velocity of the purge airflow and wheel rotation speed. For the enthalpy wheel simulated in this paper, the required purge angle is calculated as 11.25°, as stated previously.

Fig. 8 shows the schematic of the purge section on which the model is based. The difference between this schematic and the one shown in Fig. 4 is that the brush seals at the two ends of the wheel are not aligned with each other. The air leaving the purge section is directed into the building exhaust inlet stream. It goes through the wheel one more time and ends up in the building exhaust air outlet stream. To simplify the computational model, the air leaving the purge section is directly put into the building exhaust air outlet stream. The outside air outlet flow rate is reduced since a portion of the outside air inlet is used as the purge flow.

![Diagram of the Purge Section in the Enthalpy Recovery Wheel](image)

The energy and moisture lost by the outside air stream is considered when calculating the wheel performance. Those lost by the purge airflow are not included because they are not useful transfer.

\[
\varepsilon_i = \frac{m_{GA}(h_{O,A,in} - h_{O,A,out})}{m_{min}(h_{O,A,in} - h_{O,A,out})} = \frac{m_{E,i}(h_{R,i,in} - h_{R,i,out})}{m_{min}(h_{R,i,in} - h_{R,i,out})}
\]

(22)

\[
\varepsilon_i = \frac{m_{GA}(w_{O,A,in} - w_{O,A,out})}{m_{min}(w_{O,A,in} - w_{O,A,out})} = \frac{m_{E,i}(w_{R,i,in} - w_{R,i,out})}{m_{min}(w_{R,i,in} - w_{R,i,out})}
\]

(23)
\[ e_i = \frac{m_{\text{out}} (h_{\text{in}} - h_{\text{out}})}{m_{\text{in}} (h_{\text{in}} - h_{\text{out}})} = \frac{\left(m_{\text{in}} - m_{\text{w}}\right) (h_{\text{in}} - h_{\text{out}})}{m_{\text{in}} (h_{\text{in}} - h_{\text{out}})} \]  

(26)

The outlet outside air conditions are different from the values presented in Fig. 3, since the outlet air in the purge section is directed into the building exhaust air outlet.

Fig. 9 compares the predicted performance of the enthalpy recovery wheel with and without the purge section. Inclusion of the purge reduced the wheel performance by about 5%.

**Residual Water in the Desiccant Material**

As stated previously, there is residual water contained in the desiccant material even after the wheel is regenerated. In the simulation, the density of the desiccant material can be adjusted to account for this residual water. The adsorption properties of the desiccant material remain unchanged, because the adsorption equilibrium relationship takes care of the impact of the residual water on moisture transfer. Seen from Fig. 7, the residual water in the enthalpy recovery wheel is high, 18% on average. The density of the zeolite molecular sieves is increased from 760 to 897 kg/m³ to account for this residual water. Assuming 15% residual water in the active desiccant wheel, the density of the silica gel is increased from 700 to 805 kg/m³ to account for the residual water.

Fig. 10 and Fig. 11 plot the impact of the residual water on the performance of the enthalpy recovery and active desiccant wheels, respectively. The residual water in the enthalpy recovery wheel has very limited impact on its energy recovery performance. The residual water increases the heat carryover ratio of the active desiccant wheel, since the wheel heat capacity is increased. The impact on the active desiccant wheel is also limited, but larger than the enthalpy recovery wheel, which is due to the larger ratio between the mass of the enthalpy recovery wheel media and the mass of the air, as a result of its high rotation speed.

**Wheel Supporting Structure**

The performance model developed in the previous section only considers the core of the desiccant wheels. In reality, the wheel also has supporting structure such as spokes and casing for structure integrity and strength. Assuming the supporting structure goes through the same temperature fluctuation as the substrate, it can be simulated as added thermal mass to the substrate.

For the wheels simulated in this paper, the weight of the enthalpy recovery and active desiccant wheels, including their cores and supporting structures, are assumed to be 13.6 kg (30 lbs), which is typical for their sizes. The core weight is calculated based on the channel composition, properties of the desiccant material and substrate and the number of channels in the wheel. The weight of the core and supporting structure are listed in Table 2. The structure of the active desiccant wheel is heavier than the enthalpy recovery wheel, because the spokes that are used to connect the different segments of the active desiccant wheel are not present in the enthalpy recovery wheel. The substrate density is adjusted to account for the added thermal mass of the supporting structure.
Table 2. Parameters of the Wheel Supporting Structure

<table>
<thead>
<tr>
<th></th>
<th>Enthalpy Recovery Wheel</th>
<th>Active Desiccant Wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total weight (kg)</td>
<td>13.6</td>
<td>13.6</td>
</tr>
<tr>
<td>Core weight (kg)</td>
<td>9.5</td>
<td>8.2</td>
</tr>
<tr>
<td>Structure weight (kg)</td>
<td>4.1</td>
<td>5.4</td>
</tr>
<tr>
<td>Original substrate density (kg/m³)</td>
<td>2700</td>
<td>500</td>
</tr>
<tr>
<td>Adjusted substrate density (kg/m³)</td>
<td>4400</td>
<td>1238</td>
</tr>
</tbody>
</table>

The impact of the supporting structure on the operating performance of the enthalpy recovery and active desiccant wheels are plotted in Fig. 12 and Fig. 13. The supporting structure has limited impact on the energy recovery performance of the enthalpy recovery wheel, because the mass of the wheel media is relatively large compared to the mass of the air, as stated previously.

![Fig. 12 The Impact of the Wheel Supporting Structure on the Predicted Performance of the Enthalpy Recovery Wheel](image)

The supporting structure has significant impact on the regeneration efficiency and the heat carryover ratio of the active desiccant wheel. For the wheel simulated in this paper, the supporting structure decreases the regeneration efficiency by 2% and increases the heat carryover ratio by 10%.

![Fig. 13 The Impact of the Wheel Supporting Structure on the Predicted Performance of the Active Desiccant Wheel](image)

**SUMMARY AND CONCLUSION**

In this paper, the combined heat and mass transfer problem in simulating the operating performance of the desiccant wheels has been formulated. The governing equations that describe the energy and material balance, the heat and mass transfer rates and the adsorption equilibrium relationship have been established. The boundary conditions have been identified. The governing equation set has been solved using the explicit finite difference analysis. The solution to this governing equation set has been iterated until a repetitive steady state solution has been reached. This model relates the operating performance of the desiccant wheel to its design parameters and operating conditions.

This model has been applied to predict the operating performance of both enthalpy recovery and active desiccant wheels once the wheel design and operating conditions are given. The changes of the air and desiccant conditions with regard to time, the rotation of the wheel, have been predicted and explained.

Practical issues might have significant impact on the desiccant wheel performance. Inclusion of a purge section in the enthalpy recovery wheel reduces the wheel performance by about 5%. The purge section is however an important design feature since the cross contamination between the building exhaust air and the outside air intake is prevented and the indoor air quality is improved.

The desiccant wheels are never fully regenerated. There is residual water contained in the desiccant material even after the regeneration process. This residual water can be modeled by adjusting the density of the desiccant material. The residual water in the enthalpy recovery wheel has very limited impact on its energy recovery performance. The residual water increases the heat carryover ratio of the active desiccant wheel, since the wheel heat capacity is increased. The impact on the active desiccant wheel is also limited, but larger than the enthalpy recovery wheel, which is due to the larger ratio between the mass of the enthalpy recovery wheel media and the mass of the air resulting from its high rotation speed.

The wheel supporting structures such as spokes and casing can be modeled as added thermal mass to the substrate, assuming they go through the same temperature fluctuation as the substrate. The supporting structure has limited impact on the energy recovery performance of the enthalpy recovery wheel, because the mass of the wheel media is relatively large compared to the mass of the air. The supporting structure has significant impact on the regeneration efficiency and the heat carryover ratio of the active desiccant wheel. For the wheel simulated in this section, the supporting structure decreases the regeneration efficiency by 2% and increases the heat carryover ratio by 10%.

The validation of this performance model has been presented in [19].
REFERENCES


