Parametric Evaluation of Governing Heat and Mass Transfer Resistances in Membrane Based Heat and Moisture Exchangers

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Parametric Evaluation of Governing Heat and Mass Transfer Resistance in Membrane Based Heat and Moisture Exchangers

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ABSTRACT

A heat and mass transfer resistance and pressure drop model is developed and implemented to evaluate the sensible and latent effectiveness of a counter-flow membrane based heat and moisture exchanger. The performance of pin-fin and bare high aspect ratio rectangular channel architectures are compared. Dominant heat and mass transfer resistances are also investigated. It was shown that the heat transfer is dominated by the convective resistance while the dominant mass transfer resistance shifted to the membrane at smaller hydraulic diameters. The results suggest that as membrane technology improves, enhancements to the air-side heat and mass transfer coefficients will be required to continue to realize performance gains.

1. INTRODUCTION

ANSI/ASHRAE standards (ASHRAE, 2016) require minimum ventilation rates for occupied spaces to maintain a comfortable and healthy indoor environment. Ventilation requires mechanical conditioning of incoming outdoor air to maintain thermal comfort. Energy Recovery Ventilators (ERVs) may be employed to reduce the energy required to maintain healthy building environments. ERVs are air-to-air heat and mass exchangers which transfer heat and moisture between the incoming outdoor air and outgoing exhaust air. Common form factors include desiccant based “enthalpy wheel” systems, and more recently the use of membrane based heat and mass exchangers (Woods, 2014). Membrane systems consist of stacked parallel membrane layers (Figure 1a). Indoor and outdoor air is split between alternating channels and heat and mass is exchanged. The bulk flow of air is in an either cross or counter-flow orientation.

The development of membranes with low mass transfer resistances (Huizing, Mérida, & Ko, 2014) has accelerated the market penetration of membrane based ERVs. As membrane mass transfer performance improves, the importance of the convective mass and heat transfer increases. This suggests overall performance of membrane ERVs could be enhanced by decreasing the flow passage hydraulic diameter or otherwise enhancing the air-side transport. Furthermore, advances in manufacturing techniques are opening up the possibility of entirely new flow geometries with enhancement features including pins, ribs and meshes (Woods & Kozubal, 2013). Predictive models of these geometries would help inform designs that balance heat transfer, mass transfer and pressure drop in advanced membrane ERV designs.

Prior work in this area includes numerical modeling of heat and mass transfer in bare parallel membrane channels (Niu & Zhang, 2001; Yang, Yuan, Gao, & Guo, 2013; Li-Zhi Z. Zhang, 2010; Li-Zhi Zhi Zhang, Liang, & Pei, 2010). A common architecture considered is bare cross-flow parallel channels. Yang et al. (2013) determined the counter-flow configuration to be the most effective but it is more difficult to route air streams to the inlets and outlets. Yang et al. reported typical assumptions made for numerical modeling of these exchangers including (1) 1-D process (2) uniform flow (3) negligible mass change in flow and (4) heat and mass transfer along flow direction are negligible. The exchanger architecture evaluated in this study most closely resembled the “quasi-counter” flow exchanger seen in Zhang (2010). Zhang used the finite volume method to solve the coupled momentum, energy and mass transport partial differential equations using FLUENT with conjugant mass transfer solved by the heat mass...
transfer analogy. In the present study, a discretized resistance network model of new ERV architecture is developed. The model is used to identify governing heat and mass transfer resistances and explore tradeoffs between improved performance and hydraulic resistance as hydraulic diameter is decreased and internal structural and enhancement features are added.

2. REPRESENTATIVE EXCHANGE DESIGN

The representative exchanger considered here consists of stacked layers of a polymer membrane forming multiple parallel channels (Figure 1a). A top view of a single layer with air flow paths overlaid is shown in Figure 1b. Two internal flow architectures were considered (1) adjacent flat membranes forming a high aspect rectangular channel and (2) adjacent flat membranes forming a high aspect duct filled with non-conducting micropins (Figure 2). The first design provides the most open membrane area and is a good baseline for other designs. The micropins in the second design provide structural support for the membrane and aid in flow distribution. For both designs, outdoor and indoor air enter the heat exchanger from opposite sides and are distributed into alternating parallel channels. The air streams then exchange sensible heat and moisture through the membrane in a counter-flow arrangement.

3. MODELING METHODOLOGY

We developed a segmented heat and mass transfer resistance network model to predict the heat and mass transfer across a single membrane layer (see Figure 3). Symmetry was then assumed to scale the predictions for a single layer to the global ERV performance. The segments were treated as individual heat exchangers in series. In each segment, expressions were formulated to calculate the local heat transfer, mass transfer and pressure drop using the Engineering Equation Solver (EES) platform (Klein, 2014).

Figure 1: (a) Representative isometric view of stacked membrane ERV and (b) top view of a single membrane layer showing counter-flow orientation and discretization approach in present study

Figure 2: Top and side view of a single channel with micropins
Figure 4 shows a schematic of the heat and mass transfer resistances considered. Energy and mass balances in each segment allowed the outlet conditions of each segment to be calculated. These values were then used as the inlet conditions of the adjacent segment. Due to the counter-flow orientation, the system of equations was solved iteratively to determine the total heat and mass transfer.

3.1 Segment Heat Transfer Model
Other than the inlet and outlet header section (see Figure 1b), each segment was modeled as a pure counter-flow heat exchanger ($F_{ht} = 1$ in Equation (1)) using the log-mean temperature difference method. The heat transfer in the inlet and outlet headers was found by approximating the headers as cross-flow heat exchangers, where the correction factor $F_{ht}$ was calculated using relationships from Nellis and Klein (2009).

$$\dot{Q} = F_{ht} \cdot U \cdot A \cdot \Delta T_{LM}$$

(1)

The $UA$ was defined as the inverse of the outdoor air convective ($R_{ht,od}$), membrane conductive ($R_{ht,mem}$), and indoor air convective ($R_{ht,id}$), thermal resistances in series (Figure 4). The outlet temperatures used in the log-mean temperature difference calculation are determined iteratively in EES. For the micropin architecture, the variable $A$ is defined as the open membrane area in a segment. As the micropins are made of non-conducting material, their contribution to heat transfer area is neglected. The factor of two in equations (3) and (4) accounts for the top and bottom layers of the channel segment.

$$UA = \left( \frac{1}{h_{od}A} + \frac{\delta}{k_{mem}A} + \frac{1}{h_{id}A} \right)^{-1}$$

(2)

$$A = 2 \left( l \cdot W - N_{pin} \cdot (\pi / 4) \cdot d^2 \right)$$

(3)

Figure 4: Schematic of heat and mass transfer resistances
With bare channels, the area is calculated according to Equation (4).

\[ A = 2l \cdot W \]  

(4)

A representative thermal conductivity of 0.0765 W m\(^{-2}\) K\(^{-1}\) for the membrane was assumed based on data supplied by an industrial partner. For the micropin array, the convective heat transfer coefficients were determined using the Colburn \(j\) factor developed by Short \textit{et al.} (2002b) for laminar flow through staggered pin arrays (Equation (5)).

\[ j = 0.760 \left( \frac{S}{d} \right)^{0.16} \left( \frac{T}{d} \right)^{0.20} \left( \frac{L_p}{d} \right)^{(-0.11)} \left( R_{ed} \right)^{(-0.67)} \text{ for } R_{ed} < 10^3 \]  

(5)

Here, \(S\) is the streamwise center-to-center pin spacing, \(d\) is the pin diameter, \(T\) is the transverse center-to-center pin spacing, and \(L_p\) is the pin height, as shown in Figure 2. Reynolds number (\(R_{ed}\)) is defined using the pin diameter as the characteristic length and the max velocity, defined as:

\[ V_{\text{max}} = \frac{V}{A_{\text{min}}} \]  

(6)

where,

\[ A_{\text{min}} = W \cdot L_p - N_t \cdot d \cdot L_p \]  

(7)

The \(j\)-factor is related to the Stanton number by:

\[ j = St \cdot Pr^{2/3} \]  

(8)

The Stanton number can then be used to find the Nusselt number and the convective heat transfer coefficient with the following equation:

\[ St = \frac{Nu}{Re \cdot Pr} = \frac{h}{\rho \cdot V_{\text{max}} \cdot c_p} \]  

(9)

Both the Nusselt number and Reynolds number are defined using the hydraulic diameter (Equation (10)) as the characteristic length in Equation (9). \(A_{\text{tot}}\) is defined as the total wetted surface area inside a channel, and is different from the heat and mass transfer area, \(A\), defined in Equation (3).

\[ D_h = 4A_{\text{min}} \cdot l / A_{\text{tot}} \]  

(10)

\[ A_{\text{tot}} = 2(l \cdot W) + N_{\text{pin}} \cdot \pi \left( d \cdot L_p - d^2 / 2 \right) \]  

(11)

For the bare channel model, the Nusselt number was calculated using a duct flow correlation defined by as a function of Reynolds number, Prandtl number, axial position within the exchanger, and the channel aspect ratio. Constant heat flux was assumed for each segment.

3.2 Segment Mass Transfer Model

The mass transfer resistances considered in each segment were shown previously in Figure 4. As it is with heat transfer, an overall mass transfer coefficient may be obtained considering both convective resistances and the diffusivity of the membrane:

\[ KA = \left( \frac{1}{k_{m,cd} \cdot A} + \frac{\delta}{D_{\text{mem}} \cdot A} + \frac{1}{k_{m,ja} \cdot A} \right)^{-1} \]  

(12)

Now the rate of mass transfer through the membrane may be calculated in the same fashion as the rate of heat transfer (Eq. (13)).
\[ \dot{n}_v = F_{mt} \cdot K \cdot A \cdot \Delta C_{LM} \]
\[ \dot{m}_v = \dot{n}_v \cdot MW_{H_2O} \]  
(13)

Where,

\[ \Delta C_{LM} = \left( C_{od,i} - C_{id,o} \right) - \left( C_{od,o} - C_{id,i} \right) \]
\[ \ln \left( \frac{C_{od,i} - C_{id,o}}{C_{od,o} - C_{id,i}} \right) \]  
(14)

The molar concentrations used in the log-mean concentration difference are derived from the average segment temperatures and relative humidity values assuming an ideal gas mixture:

\[ P_v = \phi \cdot P_{sat} \]  
(15)

\[ C = P_v / \left( T \cdot R \right) \]  
(16)

The membrane mass transfer resistance was calculated using a representative effective diffusivity of \( 4.3 \times 10^{-7} \text{ m}^2 \text{ s}^{-1} \), based on data supplied by an industrial partner. The convective mass transfer coefficients in the outdoor and indoor air streams were obtained using the Chilton-Colburn (1934) analogy:

\[ Le = \alpha / D_{nu} \]  
(17)

\[ Sh = Nu \cdot Le^{-1/3} \]  
(18)

\[ k_m = D_{nu} \cdot Sh / D_H \]  
(19)

### 3.3 Segment Pressure Drop Calculation

The frictional pressure drop through each segment was calculated using the Fanning friction factor correlation for laminar flow through pin-fins developed by Short et al. (2002a).

\[ f = 35.1 \left( \frac{S}{d} \right)^{-1.3} \left( \frac{T}{d} \right)^{-0.78} \left( \frac{L_p}{d} \right)^{-0.55} \left( \frac{Re_d}{d} \right)^{-0.65} \] for \( Re_d < 10^3 \)

(20)

Where,

\[ 1.9 < L_p / d < 9.6 \]
\[ 1.8 < S / d < 3.2 \]
\[ 2.0 < T / d < 6.4 \]

Once the friction factor is calculated, the segment pressure drop can be found. Here it should be noted that \( d \) is the pin diameter, not the hydraulic diameter. The friction factor for the bare channels was also calculated using a duct flow correlation (Shah & London, 1978) for laminar flow.

### 3.4 Segment Energy and Mass Balance

Energy and mass balances in each segment are required for closure of the iterative model. Figure 5 shows a diagram of the flow paths of mass and energy in two counter flow channels in the exchanger. Since the membrane only allows water vapor to pass from one channel to another, the mass flow rate of dry air \( (\dot{m}_{od,a}, \dot{m}_{id,a}) \) does not change in either of the channels. Also, any water vapor leaving one channel must enter one of the channels on either side of it. Equation (21) shows the mass balance for each channel.

\[ \dot{m}_v = \dot{m}_{od,a} \left( \omega_{od,i} - \omega_{od,o} \right) \]  
(21)
A first law analysis of the outdoor and indoor air streams yields the following conservation of energy equations:

\[
\dot{m}_{od,a} (i_{od,i} - i_{od,o}) - \dot{m}_{i} + \dot{Q} = 0 \tag{22}
\]

\[
\dot{m}_{id,a} (i_{id,i} - i_{id,o}) + \dot{m}_{i} + \dot{Q} = 0
\]

Here, the sensible heat transfer (\(\dot{Q}\)) calculated from Equation (1), the mass flow rate of vapor from Equation (13) and the segment inlet and outlet specific enthalpy are calculated at the local dry bulb temperature, pressure, and humidity ratio. The mass and energy balances provide closure to the system of equations and enable an iterative solution.

2.5 Effectiveness

Finally, the overall performance of the heat and mass exchanger was characterized according to the sensible and latent effectiveness as defined by AHRI Standard 1060 (AHRI, 2013).

\[
\varepsilon_s = \frac{\dot{m}_{od,a} c_{p,od,o} (T_{od,i} - T_{od,o})}{C_{\text{min,s}} (T_{od,i} - T_{id,i})} \tag{23}
\]

\[
C_{\text{min,s}} = \min \left( \dot{m}_{od,a} c_{p,od,o}, \dot{m}_{id,a} c_{p,id,i} \right)
\]

\[
\varepsilon_i = \frac{\dot{m}_{od,a} h_{fg,od,o} (\omega_{od,i} - \omega_{od,o})}{C_{\text{min,l}} (\omega_{od,i} - \omega_{od,o})} \tag{24}
\]

\[
C_{\text{min,l}} = \min \left( \dot{m}_{od,a} h_{fg,od,o}, \dot{m}_{id,a} h_{fg,id,i} \right)
\]

5. RESULTS

The developed model was used to evaluate the effect of decreasing the channel height for a fixed total exchanger volume. The base dimensions for the micropin exchanger are shown in Table 1. The inlet conditions (\(T_{id} = 23.9^\circ\text{C}, \phi_{id} = 51.4\%, T_{od} = 35^\circ\text{C}, \phi_{od} = 47.1\%\)) are the standard rating conditions for performance rating of air-to-air exchangers for energy recovery from AHRI (2013). A volumetric flow rate of 200 CFM was specified for both the outdoor and indoor air streams. Using overall geometry and inlet conditions, the results for varying channel height from 1 mm to 4 mm are shown in Figure 6. As the channel height decreases, the heat and mass transfer increase and the total membrane area increase, yielding the improved sensible and latent effectiveness shown in Figure 6. At the
same time, since the face area is fixed and the membrane thickness is small, the change in maximum flow velocity in each channel is small (1.178 to 1.149 m/s for channel height from 1 to 4 mm). Thus, the change in Reynolds number (67.28 to 264.2) and the related increase in pressure drop is primarily governed by the decreasing hydraulic diameter.

Table 1: External dimensions for baseline micropin exchanger

<table>
<thead>
<tr>
<th>W [mm]</th>
<th>H_ex [mm]</th>
<th>L_ex [mm]</th>
<th>δ [µm]</th>
<th>Lp/d</th>
<th>S/d</th>
<th>T/d</th>
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<tr>
<td>200</td>
<td>1775</td>
<td>500</td>
<td>25</td>
<td>2</td>
<td>3.15</td>
<td>6.35</td>
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</table>

5.2 Evaluation of Governing Resistance
To better understand the governing heat and mass transfer resistances, the relative contribution of the convective and diffusion resistances were evaluated for three channel heights in a pin-fin channel at a fixed mass flow rate. The results are shown in Figure 7. The conductive heat transfer resistance contributed by the membrane was negligible compared to the convective resistance for all channel heights evaluated, while diffusive resistance due to the membrane dominated the total mass transfer resistance at small channel heights. The membrane resistance became more comparable to the convective mass transfer resistance as channel height increased. This further reinforces that as membrane technology improves, the heat and mass transfer resistances in the air-side will become the limiting factors in membrane ERVs.

5.3 Enhancement due to Pin-Fins
Based on the results the parametric study in Figure 6, a channel height of 1.6 mm was chosen for further investigation as it provided a pressure drop less than 300 Pa, which is about the maximum ERV fans can reasonably achieve. Using this channel height, the performance of a heat exchanger with and without pin-fins was calculated assuming the same base dimensions and inlet conditions in Table 1. The results are shown in Table 2.

Table 2: Bare and pin-fin exchanger performance

<table>
<thead>
<tr>
<th>Type</th>
<th>εs [%]</th>
<th>εl [%]</th>
<th>ΔP_{max} [Pa]</th>
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</thead>
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<tr>
<td>Bare</td>
<td>97.90</td>
<td>91.39</td>
<td>40</td>
</tr>
<tr>
<td>Pin-Fin</td>
<td>98.93</td>
<td>91.49</td>
<td>297</td>
</tr>
</tbody>
</table>

The pin-fin heat exchanger yielded an increase in sensible (1.03 percentage points) and slight increase latent effectiveness (0.1 percentage points), at the expense of a large pressure drop penalty of 7.5x. While the pressure drop penalty is significant, the bare channel baseline is likely not achievable in practice, as the stacked membrane would have no structural support. In the present study the pins are assumed to be non-conducting, thus, they do not
increase heat transfer area but in fact decrease heat and mass transfer area by occupying membrane area (85.68 m$^2$ bare versus 79.05 m$^2$ open). Therefore, the observed enhancement is from higher heat (average of 131.7 vs. 115.4 W m$^{-2}$ K$^{-1}$) and mass transfer coefficients (average of 0.107 vs 0.094 m s$^{-1}$). The potential advantage of the pin-fin or similar internally structured geometry is that the features could be fabricated in a manner that provides structural support to the membrane ERV while simultaneously increasing heat and mass transfer and aiding in flow distribution, similar to the structures investigated by Woods and Kozubal (2013).

6. CONCLUSIONS

The model results here show that as membrane performance increases, air-side mass transfer resistance becomes increasingly important. This suggests that using smaller diameter channels and internal features to enhance air-side heat and mass transfer could improve overall membrane ERV effectiveness. In the present study, the convective enhancement of pin-fins in this study was advantageous for heat and mass transfer, increasing the heat and mass transfer coefficient compared to a bare channel baseline. However, the internal structures covered some fraction of the membrane area, reducing the active transfer area. Furthermore, the use of internal structures significantly increased pressure drop, requiring more power for operation. The developed model can be used to optimize membrane ERV envelope and pin-fin spacing to provide designs that meet desired performance goals while also providing internal structure to the heat and mass exchanger.

NOMENCLATURE

- $A$: segment open membrane area (m$^2$)
- $c_p$: specific heat (J kg$^{-1}$ K$^{-1}$)
- $C$: molar concentration (kmol m$^{-3}$)
- $C_{min,l}$: minimum capacity rate for latent effectiveness (W)
- $C_{min,s}$: minimum capacity rate for sensible effectiveness (W K$^{-1}$)
- $d$: pin diameter (m)
- $D$: vapor diffusivity (m$^2$ s$^{-1}$)
- $D_H$: hydraulic diameter (m)
- $f_F$: Fanning friction factor (–)
- $h$: convective heat transfer coefficient (W m$^{-2}$ K$^{-1}$)
- $h_{fg}$: heat of vaporization (J kg$^{-1}$)
- $H$: height (mm)
\( i \) specific enthalpy \((J \text{ kg}^{-1})\)

\( j \) Colburn j-factor \((-)\)

\( k \) thermal conductivity \((W \text{ m}^{-1}\text{K}^{-1})\)

\( K \) combined mass transfer coefficient \((m \text{s}^{-1})\)

\( k_n \) convective mass transfer coefficient \((m \text{s}^{-1})\)

\( l \) segment length \((m)\)

\( L \) length \((m)\)

\( Le \) Lewis Number \((-)\)

\( MW_{H_2O} \) molecular weight of \( H_2O \) \((kg \text{ kmol}^{-1})\)

\( \dot{m} \) mass flow rate \((kg \text{ s}^{-1})\)

\( \dot{n} \) molar flow rate \((\text{kmol} \text{ s}^{-1})\)

\( N_{pin} \) total number of pins in segment \((-)\)

\( N_p \) pin count in transverse row of segment \((-)\)

\( Nu \) Nusselt number \((-)\)

\( p \) pressure \((\text{Pa})\)

\( Pr \) Prandtl number \((-)\)

\( \dot{Q} \) rate of heat transfer through membrane \((W)\)

\( \overline{R} \) universal gas constant \((J \text{ kmol}^{-1}\text{K}^{-1})\)

\( \text{Re} \) Reynolds number \((-)\)

\( R_{mem} \) heat transfer resistance of membrane \((KW^{-1})\)

\( S \) streamwise center to center distance between pins \((m)\)

\( Sh \) Sherwood number \((-)\)

\( T \) transverse distance between pins \((m)\)

\( T \) temperature \((K)\)

\( U \) combined heat transfer coefficient \((W \text{ m}^{-2}\text{K}^{-1})\)

\( V \) velocity \((m \text{s}^{-1})\)

\( \dot{V} \) volume flow rate \((m^3 \text{s}^{-1})\)

\( W \) channel width \((m)\)

**Subscripts**

\( a \) air

\( D \) pin diameter

\( d \) channel diameter

\( ex \) exchanger

\( id \) indoor air stream

\( ht \) heat transfer

\( i \) inlet

\( l \) latent

\( LM \) log-mean

\( \text{max} \) maximum

\( \text{mem} \) membrane

\( \text{min} \) minimum

\( \text{mt} \) mass transfer

\( o \) outlet

\( od \) outdoor air stream

\( p \) pin
Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$\alpha$</td>
<td>thermal diffusivity</td>
<td>$(m^2 \cdot s^{-1})$</td>
</tr>
<tr>
<td>$\delta$</td>
<td>membrane thickness</td>
<td>$(m)$</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>effectiveness</td>
<td>$(\cdot)$</td>
</tr>
<tr>
<td>$\phi$</td>
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<tr>
<td>$\omega$</td>
<td>absolute humidity ratio</td>
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REFERENCES


