Membrane-type Total Heat Exchanger Performance Simulation with Consideration of Entrance Effects

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Abstract. Membrane-type total heat exchanger (THX) is an air-to-air heat exchanger used to reduce the building energy consumption associated with forced ventilation by recovering both heat and moisture from ventilation air. It contains a heat/moisture exchange core made of a water vapour permeable membrane, supply outdoor air and exhaust indoor air flow through the membrane channels in the core in a crossflow manner and exchange heat and moisture across the membranes. The present work numerically investigates the airflow channel entrance effects on the THX performance. The results show that such effects on the air temperature and humidity distributions are inconspicuous and so are they on the THX effectiveness, it is therefore appropriate to use the constant Nusselt number to evaluate the THX performance.

1. Introduction

With the developments of economy and technology, people spend more time in indoor environment [1]. Forced ventilation is often necessary to improve the indoor air quality but it simultaneously causes a huge energy consumption in fresh air conditioning. Membrane-type total heat exchanger (THX) is an air-to-air heat exchanger used to reduce such an energy consumption, it contains a heat and moisture exchange core made of a water vapor permeable membrane, supply outdoor air and exhaust indoor air flow through the channels in the core and exchange heat and moisture across the membranes.

Numerical methods are often used to evaluate the THX performance, in which equations governing the heat and moisture transfer in the THX core are set up and solved numerically to obtain various physical quantity distributions and exchanger effectiveness. Zhang and his co-workers [2, 3] numerically studied the THX performance. Min and his group [4-9] conducted a series of researches on the THX, they numerically investigated the effects of membrane thickness, channel spacing, membrane materials and outdoor air state on the THX performance [4-6], discussed the phenomena of heat and mass transfer in different directions across membranes [7], and compared the effectiveness-NTU and numerical methods for evaluating the THX performance and reported that the numerical method with consideration of the effect of adsorption heat was the best method [8]. They also carried out experimental investigations on the THX and validated their simulation model by comparing the calculations with the experiments [9]. More studies on the THX performance evaluation can be found in Refs. [10-13].

The existing studies all used the convective heat and mass transfer coefficients for fully developed laminar flow in rating the THX performance, no study has discussed the entrance effect on it. The present research attempts to investigate that effect by comparing the THX performance results obtained using the local heat and mass transfer coefficients and those for fully developed laminar flow.
2. Theoretical Model

Fig. 1 illustrates the basic physical model for the core of a typical THX, which consists of a supply airstream channel, an exhaust airstream channel, and membranes, with the two airstreams proceeding along the channels in a crossflow pattern and exchanging heat and moisture across the membrane separating the channels. Because of the periodicity and symmetry in geometry, half the volume of the exhaust airstream, half the volume of the supply airstream and the intermediate membrane are taken to constitute the computational domain.

![Figure 1. Basic physical model](image)

The mathematical model describing the heat and mass transfer in the Fig. 1 model is based on the assumptions that the physical properties of the air fluid and membrane are constant, and the heat and moisture transfer are both at steady state.

Supply air:

\[
\frac{m_s c_p}{n x_F} \frac{\partial T_s}{\partial x} + 2h_s (T_s - T_{m_s}) = 0, \quad \frac{m_s}{n y_F} \frac{\partial W_s}{\partial x} + 2k_s (W_s - W_{m_s}) = 0
\]

Exhaust air:

\[
\frac{m_e c_p}{n x_F} \frac{\partial T_e}{\partial y} + 2h_e (T_e - T_{m_e}) = 0, \quad \frac{m_e}{n y_F} \frac{\partial W_e}{\partial y} + 2k_e (W_e - W_{m_e}) = 0
\]

Membrane:

\[
q = -\lambda_m \frac{\partial T_m}{\partial z} = \lambda_m \frac{T_{m_s} - T_{m_e}}{\delta}, \quad J = -D_{m_s} \frac{\partial \theta}{\partial z} = D_{m_s} \frac{\theta_{m_s} - \theta_{m_e}}{\delta}
\]

Also

\[
q = h_s (T_s - T_{m_s}) + JL_{m_s} = h_e (T_{m_e} - T_e) + JL_{m_e}, \quad J = k_s (W_s - W_{m_s}) = k_e (W_{m_e} - W_e)
\]

where \(W\) is the air humidity ratio, \(m\) is the air mass flow rate, \(J\) is the moisture flux across membrane, \(n\) is the number of channels, \(x_F\) and \(y_F\) are the channel lengths in the \(x\) and \(y\) directions, \(D_{m_s}\) is the moisture diffusivity in membrane, \(\theta\) is the membrane moisture content, \(\delta\) is the membrane thickness, and \(L_{m_s}\) is the heat of adsorption of water vapor on membrane, which is assumed to be equal to the heat of vaporization of water. The subscripts \(s\), \(e\), \(m\) and \(w\) refer to the supply air, exhaust air, membrane and water vapor, respectively.

The equation describing moisture adsorption at the membrane surface can be represented by

\[
\theta = \frac{w_{\text{max}}}{1 - C + C/\phi}
\]

where \(\theta\) is the moisture uptake in membrane, \(w_{\text{max}}\) the maximum moisture content of the membrane material, \(C\) the adsorption constant, which determines the shape of the adsorption curve.

The THX performance is evaluated using the sensible heat transfer effectiveness, latent heat transfer effectiveness and enthalpy effectiveness, which can be calculated from

\[
\varepsilon_s = \frac{m_s (T_{s_l} - T_{sl}) + m_e (T_{m_e} - T_{sl})}{2m_{\text{min}} (T_{sl} - T_{sl})} \quad (6a); \quad \varepsilon_L = \frac{m_s (w_{sl} - w_{sl}) + m_e (w_{m_e} - w_{sl})}{2m_{\text{min}} (w_{sl} - w_{sl})} \quad (6b)
\]
\[
e_{H} = \frac{m_{i} (H_{i} - H_{w}) + m_{o} (H_{o} - H_{w})}{2m_{w} (H_{i} - H_{w})}
\]
where \( H \) is the specific enthalpy of moist air, given by
\[
H = 1.006T + W (2501 + 1.805T)
\]
in which \( T \) is the air temperature in °C.

According to Shah and London [14], for fully developed laminar flow between parallel plates with constant temperature boundary conditions, the Nusselt number is \( Nu_{c} = 7.54 \). According to Stephan and Karl [15], the mean Nusselt number for laminar flow between parallel plates with isothermal boundary conditions including the entrance can be calculated by
\[
Nu_{m} = 7.55 + \frac{0.024(x^{*})^{-1.14}}{1 + 0.0358(x^{*})^{-0.64} Pr^{0.17}}, \quad x^{*} = \frac{x}{D_{h} Re Pr}
\]
Since the mean Nusselt number is the local Nusselt number averaged along the channel, the local Nusselt number can be derived from Eq. (8) as
\[
Nu_{c} = 7.55 + \frac{0.00336(x^{*})^{-1.14}}{1 + 0.0358(x^{*})^{-0.64} Pr^{0.17}} - \frac{0.00055(x^{*})^{-1.78} Pr^{0.17}}{(1 + 0.0358(x^{*})^{-0.64} Pr^{0.17})^{2}}
\]
with \( Nu_{m} \) and \( Nu_{c} \) satisfying \( Nu_{c} = \frac{1}{x^{*}} \int_{0}^{x^{*}} Nu_{c} dx^{*} \). When the Nusselt number is known, the convective heat transfer coefficient can be obtained from
\[
h_{c} = h_{s} = h = \frac{Nu_{c} \lambda}{2d}
\]
while the convective mass transfer coefficient can be related with \( h \) by the heat and mass transfer analogy relation as [16-18]
\[
k_{s} = k_{c} = k = \frac{h}{c_{p} Le^{2/3}}
\]
where \( Le \) is the Lewis number, whose value is about 0.85 for a temperature range of 0-40°C [16].

3. Results and Discussion

The indoor and outdoor air dry-bulb temperatures are taken as 26.0 and 35.0 °C and the relative humidities are set as 50% and 70%. The airflow rate is specified to range 0.1-0.3 kg/s for both the supply and exhaust airstreams, generating approximately 1.0-3.0 m/s air velocities in the channel. Calculations are conducted on the THX with the core dimensions and membrane parameters presented in Table 1 for air velocities of \( V = 1.0-3.0 \) m/s, generating Reynolds numbers of 266-799.

| Table 1. THX core dimensions and membrane parameters |
|---------------------------------------------|
| THX core dimensions |
| \( x_{f}, y_{f} \) (m) | 0.25 |
| \( d \) (mm) | 2.0 |
| \( N \) | 180 |
| \( \delta \) (mm) | 0.1 |
| \( \lambda_{m} \) (W/m.K) | 0.1 |
| Membrane parameters |
| \( D_{som} \) (10^-7 kg/m.s) | 2.5 |
| \( w_{max} \) (kg/kg) | 0.25 |

Figure 2 illustrates various Nusselt numbers including the constant, local and average ones for \( V = 2.0 \) m/s air velocity, of which \( Nu_{c} = 7.54 \) is for fully developed laminar flow, \( Nu_{c} \) is given by Eq. (9) and varies along the channel, and \( Nu_{m} \) is given by Eq. (8) and is calculated as 7.73. It is appropriate to think that the local \( Nu \) can best reflect the reality whereas the constant \( Nu \) is widely used in the THX performance evaluations in literatures. Correspondingly, there are three convective heat transfer coefficients, \( h_{c}, h_{s} \) and \( h_{m} \), corresponding to \( Nu_{c}, Nu_{s} \) and \( Nu_{m} \), which are determined by Eq. (10). Fig.
3 presents their variations. Also, there are three convective mass transfer coefficients, $k_e$, $k_i$ and $k_m$, corresponding to $h_e$, $h_i$ and $h_m$, they are determined by Eq. (11). Their variations are similar to those of the heat transfer coefficients.

**Figure 2.** Various $Nu$ for $V=2.0$ m/s

**Figure 3.** Various $h$ for $V=2.0$ m/s

Figures 4 and 5 compare the air temperature and humidity ratio distributions obtained using the constant and local Nusselt numbers, $Nu_c$ and $Nu_t$, for $V=2.0$ m/s. The figures show that the air temperature and humidity distributions obtained using different Nusselt numbers are similar to each other, and the differences caused by the usage of $Nu_c$ and $Nu_t$ are minimal and inconspicuous.

**Figure 4.** Air temperature distributions obtained using $Nu_c$ and $Nu_t$ for $V=2.0$ m/s

**Figure 5.** Air humidity ratio distributions obtained using $Nu_c$ and $Nu_t$ for $V=2.0$ m/s
Table 2 compares the sensible, latent and enthalpy effectivities calculated using the local, average and constant Nusselt numbers ($\text{Nu}_L$, $\text{Nu}_m$ and $\text{Nu}_c$) for air velocity of $V=2.0$ m/s. When the local $\text{Nu}$ ($\text{Nu}_L$) is used, the calculated sensible, latent and enthalpy effectivities are 65.76%, 52.67% and 56.70%, respectively. When the average $\text{Nu}$ ($\text{Nu}_m$) is used instead of the local one ($\text{Nu}_L$), the sensible effectivity increases by 0.22 percentage points while the latent and enthalpy effectivities decrease by 1.76 and 1.68 percentage points. All effectivities become closer to those with $\text{Nu}_L$ when the constant $\text{Nu}$ ($\text{Nu}_c$) is employed. This is somewhat surprising, the explanation is that decrease of the Nusselt number acts to reduce the sensible effectivity but increases the latent effectivity, while use of the constant $\text{Nu}$ instead of the local one works to increase the sensible effectivity but reduce the latent and enthalpy effectivities, as seen in Table 2, these two effects act together and lead to the above results. No matter which Nusselt number is used, the enthalpy effectivity always lives between the sensible and latent effectivities. The Table 2 data support the use of the constant $\text{Nu}$ in the THX performance evaluation.

![Table 2](image)

Figure 6 compares the sensible, latent and enthalpy effectivities calculated using $\text{Nu}_L$, $\text{Nu}_m$ and $\text{Nu}_c$ for $V=1.0$-3.0 m/s. As expected, all effectivities decrease with increasing air velocity. The differences among the effectivities calculated using different Nusselt numbers tend to increase with increasing air velocity, but they are basically quite small and may not exceed 2 percentage points.

![Figure 6](image)

4. Conclusions
A mathematical model considering the effect of adsorption heat is provided to analyze the heat and mass transfer in the core of a membrane-type total heat exchanger. Calculations are conducted to investigate the airflow channel entrance effects on the total heat exchanger performance. The results suggest that such effects on the air temperature and humidity distributions are not obvious and those on the various effectivities including the sensible, latent and enthalpy ones are very limited and generally less than 2 percentage points. It is therefore appropriate to use the constant Nusselt number instead of the local one, which varies along the channel, to evaluate the total heat exchanger performance.
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