Development and Validation of a Minichannel Evaporator Model under Different Dehumidifying Conditions

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Minichannel heat exchangers are taken as a great area of research interest for their elevated heat transfer characteristics, higher heating surface area to volume ratio, smaller size, lighter weight, lower fluid inventory and energy efficiency compared to conventional heat exchangers with same heat exchange capacity.

However, the implementation of minichannels in evaporators lagged behind the condenser application because of many problems and concerns, such as the instability of refrigerant flow, accumulation of condensed water, and frosting.

Minichannel: 3 mm ≥ Dₚ > 200µm, (Kandlikar and Grande 2003)

Schematic of a compact minichannel condenser
To **overcome the drawbacks** that resulted from adopting classical modeling assumptions in the **ε-NTU approach**.

Hassan et al. (2015) and (2016) developed a comprehensive **two-dimensional** numerical model for minichannel evaporators, referred to as **Fin2D-W**.

They compared the **Fin2D-W results** with the **classical ε-NTU approach**, and concluded that:

- These widely used assumptions resulted in substantial deviations in heat transfer results between the two approaches, especially at the **region of partially wet fin**; and
- at the presence of a **temperature difference** between the adjacent tubes.
1. to develop a less complicated model compared to the Fin2D-W model, which will be referred to as the **Fin1D-MB** model. The proposed model will capture:
   - the partial dehumidification scenarios, and
   - tube-to-tube heat conduction as the Fin2D-W model does; but
   - with higher computational speed like the ε-NTU approach.

2. To validate the proposed model against experimental data for a minichannel evaporator operating with **R134a** at various test conditions.
The energy conservation equation within any of the tube wall cells \( t \), in contact with \( n_r \) refrigerant cells, \( n_a \) air cells, and \( n_f \) fin cells can be written as:

\[
\nabla \left( k_t \cdot \nabla T_{c,t} \right) \, dV + \sum_{r=1}^{n_r} U_{r,t} \left( T_r - T_{c,t} \right) \, dA_{r,t} + \sum_{a=1}^{n_a} U_{\text{wet},a,t} \left( T_{a,t}^* - T_{c,t} \right) \, dA_{a,t} + \sum_{f=1}^{n_f} dQ_{\text{cond},f} \bigg|_{\text{fin root}} = 0
\]

It should be noted that a linearization scheme is used to relate the saturated air humidity ratio to its corresponding surface tube wall temperature, where \( W_{\text{sat},s,t} = a_{a,t} + b_{a,t} T_{s,t} \):

\[
U_{r,t} (\text{W/m}^2 \cdot \text{K}) = \frac{1}{\left( t_t / 2 \cdot k_t \right) + \left( 1 / \alpha_{r,t} \right)}
\]

\[
U_{\text{wet},a,t} (\text{W/m}^2 \cdot \text{K}) = \frac{1}{\left( t_t / 2 \cdot k_t \right) + \left( 1 / \alpha_{\text{wet},a,t} \right)}
\]

\[
T_{a,t}^*(\circ C) = \frac{T_a + \beta_a \left[ W_a - (W_{\text{sat},s,t} - b_{a,t} \cdot T_{s,t}) \right]}{1 + \beta_a \cdot b_{a,t}}
\]

\[
\alpha_{\text{wet},a,t} (\text{W/m}^2 \cdot \text{K}) = \alpha_{a,t} \left( 1 + \beta_a \cdot b_{a,t} \right),
\]

\[
\beta_a (\text{K}) = \frac{h_{fg}}{C_p \cdot ma \cdot Le^{2/3}}
\]

\[
b_{a,t} (1/\text{K}) = \frac{(W_a - W_{\text{sat},s,t})}{(T_{dp} - T_{s,t})}
\]
The physical discretization of the fin is 1D. However, to capture the actual fin condition, it has to be virtually discretized into three portions, \( fp_1 \), \( fp_2 \), and \( fp_3 \), in the \( y \)-direction.

\[
\begin{align*}
\theta_{a,f1}(y_{fp1}) &= C_1 e^{M \cdot y_{fp1}} + C_2 e^{-M \cdot y_{fp1}} - \psi \\
0 &\leq y_{fp1} \leq \zeta_1 \\
\theta_{a,f2}(y_{fp2}) &= C_3 e^{M \cdot y_{fp2}} + C_4 e^{-M \cdot y_{fp2}} \\
0 &\leq y_{fp2} \leq H_f \left( \zeta_1 + \zeta_2 \right) \\
\theta_{a,f3}(y_{fp3}) &= C_5 e^{M \cdot y_{fp3}} + C_6 e^{-M \cdot y_{fp3}} - \psi \\
0 &\leq y_{fp3} \leq \zeta_2
\end{align*}
\]

where \( m = \sqrt{P_j \alpha_{a,f} / k_{f,j} A_{c,f}} \), \( M = m \sqrt{1 + \beta_a \cdot b_{a,f}} \), and
\[
\psi[K] = \frac{\beta_a \left[ W_a - \left(W_{a,f} - (b_{a,f} T_f) \right) \right] - b_{a,f} T_s}{1 + \beta_a \cdot b_{a,f}}
\]

The Fin1D-MB model assumes uniform air properties along \( y \)-direction, so the fin temperature profile can be expressed as:

\[
\begin{align*}
T_{fp1}(y_{fp1}) &= \bar{T}_a - \theta_{a,fp1}(y_{fp1}) \\
T_{fp2}(y_{fp2}) &= \bar{T}_a - \theta_{a,fp2}(y_{fp2}) \\
T_{fp3}(y_{fp3}) &= \bar{T}_a - \theta_{a,fp3}(y_{fp3}) \\
T_f(y) &= \begin{bmatrix} T_{fp1}(y_{fp1}) \\ T_{fp2}(y_{fp2}) \\ T_{fp3}(y_{fp3}) \end{bmatrix} = A \begin{bmatrix} y_{fp1}, y_{fp2}, y_{fp3} \end{bmatrix} \\
&= A \begin{bmatrix} T_a \\ T_{jB} \\ T_f \end{bmatrix}
\end{align*}
\]
Evaporator Discretization

Governing Equations

Solution Methodology

 Tube Wall

 Fin Wall

 Refrigerant-side

 Air-side

**Fin1D-MB Model**

- **Partially Wet Fin (3 pieces)**
  - $T_a$(avg) = 27 °C
  - $T_{dp}$(avg) = 17.2 °C
  - $\psi$(avg) = -6.33 K
  - $T_{fB}$ = 17 °C
  - $T_{fT}$ = 17.3 °C
  - $m.H_f$ = 0.5
- **Partially Wet Fin (2 pieces)**
  - $T_a$(avg) = 27 °C
  - $T_{dp}$(avg) = 17.2 °C
  - $\psi$(avg) = -6.33 K
  - $T_{fB}$ = 17 °C
  - $T_{fT}$ = 17.5 °C
  - $m.H_f$ = 0.5
- **Totally Wet Fin**
  - $T_{fp3}(y)$
  - $T_{fp2}(y)$
  - $T_{fp1}(y)$

- **Totally Dry Fin**
  - $T_{dp}$(avg) = 17.2 °C
  - $T_{fB}$ = 17 °C

- **Partially Wet Fin (2 pieces)**
  - $\zeta_1 = 13.8\% H_f$

- **Totally Wet Fin**
  - $\zeta_1 = H_f$

**Dimensionless fin height ($Y/H_f$)**

**Temperature ($T_f$ [°C])**

- $T_a$(avg) = 27 °C
- $T_{dp}$(avg) = 17.2 °C
- $\psi$(avg) = -6.33 K
- $T_{fB}$ = 17 °C
- $T_{fT}$ = 17.5 °C
- $m.H_f$ = 0.5

- $T_{fp3}(y)$
- $T_{fp2}(y)$
- $T_{fp1}(y)$

- $T_{dp}$(avg) = 17.2 °C
- $T_{fB}$ = 17 °C
- $T_{fT}$ = 17.5 °C
- $m.H_f$ = 0.5
The energy balance in each refrigerant cell \( r \) in contact with \( n_t \) tube wall cells (\( t=1-n_t \)) is:

\[
\dot{m}_r \cdot dh_r = -\sum_{t=1}^{n_t} \alpha_{r,t} \left(T_r - T_{s,t}\right) dA_{r,t}
\]

The total refrigerant-side pressure drop along the \textit{x-direction} consists of frictional, acceleration, and gravitational pressure drop terms.

In the \textit{superheat region}, the single-phase total pressure drop can be expressed as:

\[
\left(\frac{dp}{dx}\right)_{sp,tot} = \frac{f_G}{2} \frac{G_r^2}{D_h \cdot \rho_{G_{sp,fric}}} + G_r^2 \left[ \frac{1}{\rho_{G_{out}}} - \frac{1}{\rho_{G_{in}}} \right]_{sp,acc} + g \rho_G \sin \xi \bigg|_{sp,grav}
\]

However, in the \textit{two-phase region}, the total pressure drop for refrigerant-side can be expressed as:

\[
\left(\frac{dp}{dx}\right)_{tp,tot} = G_r^2 \frac{f_L (1-\dot{x})^2}{2D_h \cdot \rho_L} \Phi_L_{tp,fric} + G_r^2 \left( \frac{\dot{x}^2}{\rho_L \cdot \varepsilon} + \frac{(1-\dot{x})^2}{\rho_G (1-\dot{x})} \right)_{tp,acc} + g \left[ \varepsilon \rho_G + (1-\varepsilon) \rho_L \right] \sin \xi \bigg|_{tp,grav}
\]
The heat rate balance within an air cell \( a \) in contact with a fin cell \( f \), which is discretized into three portions \((fp=1-3)\), and \( n_t \) tube cells.

\[
\dot{m}_a \cdot C_{p,ma} \cdot dT_a = -\sum_{fp=1}^{3} \alpha_{a,fp} \cdot \theta_{a,fp} \cdot dzdy_{fp} - \sum_{t=1}^{n_t} \alpha_{a,t} \left( T_a - T_{s,t} \right) dA_{a,t}
\]

The mass balance, taking into account the Chilton-Colburn analogy (Sharqawy and Zubair, 2008), within any air cell is:

\[
\dot{m}_a \cdot dW_a = \frac{1}{Le^{2/3} \cdot C_{p,ma}} \left[ -\sum_{fp=1}^{3} \alpha_{a,fp} \left( W_a - W_{sat,fp} \right) \cdot dzdy_{fp} - \sum_{t=1}^{n_t} \alpha_{a,t} \left( W_a - W_{sat,s,t} \right) dA_{a,t} \right]
\]
**Fin1D-MB Model**

**Initialization process:** guessing the temperature and humidity ratio fields for both fluids, and walls.

**Iterative procedure:**

- **Fluids calculations:** calculating the outlet temperature and pressure for each fluid cell, besides the outlet humidity ratio for moist air cells.
- **Tube wall calculations:** calculating the temperature and saturated humidity ratio fields for the tube wall cells.
- **Fin wall calculations:** determining the dehumidifying condition (totally dry, totally wet, or partially wet), length of each fin portion $\zeta_1$ and $\zeta_2$, and average temperature for each fin cell.

**Solution Methodology**

**Governing Equations**

$max(|Residual|) \leq \text{Tolerance}$

Yes

Outputs

No

$\overline{\theta}_{dp} = \overline{T}_a - \overline{T}_{dp}$, $\theta_{a,fB} = \overline{T}_a - T_{fB}$, and $\theta_{a,fl} = \overline{T}_a - T_{fl}$.
Results & Discussion

Experimental Setup

Evaporator Dimensions & Inlet Conditions

Validation of the Fin1D-MB model

Symbols

- Compressor
- Chiller
- Heater
- Evaporator
- Pitot tube
- Expansion device
- 4-way valve
- Coriolis flowmeter
- Water-cooled condenser
- Air handling unit
- Pressure measurement
- Temperature measurement
- Humidity measurement
- Differential pressure measurement
Operating conditions for the R134a minichannel evaporator.

<table>
<thead>
<tr>
<th></th>
<th>Air</th>
<th>Refrigerant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet dry-bulb temperature (°C)</td>
<td>7</td>
<td>Inlet mass flow rate (kg/h)</td>
</tr>
<tr>
<td>Inlet relative humidity (%)</td>
<td>73—89</td>
<td>Inlet vapor quality (-)</td>
</tr>
<tr>
<td>Inlet flow rate (m³/h)</td>
<td>890—1890</td>
<td>Outlet superheat (K)</td>
</tr>
</tbody>
</table>
### Experimental Setup

#### Evaporator Dimensions & Inlet Conditions

Validation of the Fin1D-MB model

### Results & Discussion

#### Correlations used in the Fin1D-MB model for coefficients evaluation

<table>
<thead>
<tr>
<th>Fluid type</th>
<th>Heat transfer coefficient</th>
<th>Frictional pressure drop</th>
<th>Expansion/Contraction pressure losses</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Refrigerant:</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Air:</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
The Fin1D-MB model predicts the $T_{r,\text{in}}$ within $\pm 0.5 \, ^\circ\text{C}$ error bands. The mean absolute error (MAE) and standard deviation (SD) of the predicted values are $\pm 0.24 \, ^\circ\text{C}$ and $\pm 0.25 \, ^\circ\text{C}$, respectively.

The proposed model successively estimates the $\Delta P_r$ within $\pm 20\%$ error bands, with MAE and SD values of $\pm 9.12\%$ and $\pm 7.18\%$, respectively.

Although, the MAE and SD of the predicted $\Delta P_r$ are relatively high, it was found that their effect on the evaporator capacity was rather small.
Regarding the air-side, it can be observed that approximately all the predicted values of the $T_{a,\text{out}}$ are within $\pm 0.5$ °C error bands, with MAE and SD of $\pm 0.43$ °C and $\pm 0.34$ °C, respectively.

The good prediction of the refrigerant and air temperatures has a positive impact on the estimated cooling capacity. The Fin1D-MB model can predict the $Q_r$ with good agreement, with MAE and SD of $\pm 1.8\%$ and $\pm 0.3\%$, respectively.

For the current study, no adjustment factors were applied either to the heat transfer or frictional pressure drop coefficients.
The Fin1D-MB model is capable of predicting any dehumidifying condition for the fin and tube, this is the consequence of adopting the **technique of moving boundaries** between wet and dry portions along the fin height.

The proposed model predicted the outlet air temperature within ±0.5 °C error bands with a MAE of ±0.43 °C.

Regarding the refrigerant-side, the Fin1D-MB model successfully estimated:

- the inlet refrigerant temperature within error bands of ±0.5 °C with a MAE of ±0.24 °C,
- the pressure drop within error bands of ±20% with a MAE of ±9.12%, and
- the cooling capacity within error bands of ±5% with a MAE of ±1.8%.
“The great Pleasure in life is doing what people say you cannot do”

-Walter Bagehot

Thanks for Your Attention