A Comparative Study Between 2D Numerical Minichannel Evaporator Model and Classical Effectiveness-NTU Approach Under Different Dehumidifying Conditions

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In this paper a two-dimensional numerical model for a minichannel evaporator is implemented. This model takes into account the variation of wall temperature and moist air properties in both longitudinal and transverse directions. The verification of the current model is done with an analytical ε-NTU approach, and the results of the two approaches show a very good agreement and consistency. Different refrigeration and air conditioning applications have been chosen which represent various inlet conditions to the evaporator. A range of tube temperatures has also been selected to allow different dehumidifying scenarios for the tube and fin. A comparative study is conducted between the current model results and the results of the
traditional $\varepsilon$-NTU method based on two different models of simultaneous heat and mass transfer. Significant deviations in results between the current model and the traditional $\varepsilon$-NTU approach are found, especially in latent heat transfer. These deviations are mainly due to the assumptions which are normally adopted by the $\varepsilon$-NTU method and fin theory such as no variation in moist air temperature and humidity ratio along the direction between tubes, no accounting for partially wet fin conditions, and finally the assumption of a constant average saturation line slope within a specific evaporator segment.
Introduction

The minichannel heat exchanger is a promising technology which has recently been used to deal with many major challenges facing the refrigeration and air conditioning industry. Those challenges include improving the system performance, reducing the total refrigerant charge in the system, and reducing the footprint and size of the equipment.

When a minichannel heat exchanger is used as the evaporator, its wall surface temperature is usually below the average dew point of the surrounding moist air. This results in simultaneous heat and mass transfer, which subsequently leads to a partially or totally wet fin depending on the surface temperature and inlet air properties. The fin performance is dramatically influenced by water condensation on the surface. A lot of experimental and numerical studies have been implemented by many authors to analyze the performance and efficiency of a wet fin compared with a dry one, such as Liang et al. (2000), and Lin et al. (2001). The final common conclusions of their studies are as follows: firstly, in the case of a partially wet surface, a considerable influence of the relative humidity on the fin efficiency is encountered compared with a totally wet fin, and secondly, the dry fin efficiency is about 15–25% higher than that of the corresponding wet fin.

Currently, several minichannel evaporator models under dehumidification are available in the literature (e.g., Wu and Webb 2002; Zhao et al. 2012; Gossard et al. 2013). Most of them use the traditional ÕNTU approach and the adiabatic wet fin tip efficiency. However, the classical ÕNTU modeling does not usually account for the variation of air properties along the fin height, the partially wet scenario, or the heat conduction between different tubes, which is a consequence of employing the adiabatic fin tip assumption. Martínez-Ballester et al. (2010)
developed a detailed 2D model for a CO₂ minichannel gas cooler to identify the impact of those classical assumptions and their drawbacks on the performance of heat exchangers. Their study revealed large errors in the capacity prediction of individual tubes due to the adiabatic fin tip assumption, especially when the neighboring tubes were at different temperatures.

To evaluate the air-side performance of a minichannel evaporator under dehumidification, a comparative study between the proposed numerical (Fin2D) model results and the ÜNTU method is implemented for different air conditioning and refrigeration applications.

**Modeling of simultaneous heat and mass transfer**

Analyzing the performance of evaporators is complicated especially when they operate under wet conditions. Several simplifications of the original differential equations have been proposed, based on different assumptions, resulting in a variety of alternative heat and mass transfer models. The two most well-known models are the single potential model proposed by Threlkeld (1970), and the dual potential model proposed by McQuiston (1975).

**Single and dual potential models**

Figure 1 shows schematically a cold surface in contact with a moving stream of moist air. A moving film of water is formed on the surface by condensation of moisture from the air stream. Immediately next to the water film, we assume that the air is saturated at the condensate film temperature $T_{cf}$. Nevertheless, the thermal heat resistance associated with the presence of the thin water film due to condensation is very small and may be neglected, which results in a condensate film temperature equal to the wall surface temperature $T_s$. 
The derivation of both models may be divided into two major steps. First of all, an equation for the total heat transfer rate for a general increment of the wet surface area $dA$ is developed, according to the assumptions made in each model, as described in Table 1.

These expressions are then applied to an incremental fin area $dA_f$ to obtain and solve a differential equation describing the temperature distribution along the fin. With this solution it is possible to derive an expression for the fin efficiency using a solution similar to that obtained by Incropera and DeWitt (2006, pp. 147–153) for dry fins. Different fin efficiency expressions are obtained for each model, as shown in Table 2.

Finally, the total heat transfer for the heat exchanger may be calculated according to the model selected. With the single potential model, the total heat transfer rate is based on the log-mean enthalpy potential of the exchanger, making it difficult to separate its sensible and latent contributions. The dual potential model, on the other hand, allows for the independent evaluation of each contribution, and continuous evaluation of total heat in the transitions from humid to dry condition.

**Fin2D evaporator model development**

*Evaporator discretization*

Figure 2a presents a piece of the studied minichannel evaporator. It is discretized along the x-direction (refrigerant flow) in a number of segments $\hat{n}_\text{Ax}$. Each segment (Figure 2b) consists of two streams of refrigerant (top and bottom flows) that are split into $\hat{n}_\text{Bx}$ channels in the z-direction (air flow); two flat tubes (top and bottom) that are discretized into $\hat{n}_\text{Cz}$ cells in the z-direction; and both air flow and fins, which are discretized in two dimensions: $\hat{n}_\text{Dy}$ cells in the y-
direction and red cells in the z-direction. This is summarized in the text as a grid: \{a,b,c,d,e\}. For illustration of the nomenclature, the numerical example shown in Figure 2 corresponds to the grid \{2,5,3,6,5\}.

All grid dimensions are independent, with the only exception being that the air and fin have the same discretization. The heat is transferred by convection from the moist air cells to the fin cells as well as to the unfinned area of the tube wall in contact at the top and bottom. Then the fin cells conduct the heat along the plane ŷz and also to the wall cells in contact at the fin roots. The tube cells exchange the heat by conduction between each other in the x̂z plane and transfer it by convection to the refrigerant cells.

**Governing equations**

Every fluid cell has two nodes, one at the inlet and one at the outlet, while the wall cells have only one node, located in the centroid of the cell. Consequently, the governing equations for the fluids and wall can be written as follows:

**Air governing equations**

The dual potential model (McQuiston 1975) is adopted in the current work because it has many advantages, as discussed before.

The heat balance for any air cell \(i\) in contact with \(n_i\) wall cells \((j = 1, n_i)\) is given by

\[
\dot{m}_i \cdot \overline{C}_{p,ma} \cdot dT_i = \sum_{j=1}^{n_i} -\alpha_{a,ij} (\overline{T}_i - T_{s,j}) P_y \cdot ds_y
\]

where \(\overline{C}_{p,ma} = \left( C_{p,ma,\text{in}} + C_{p,ma,\text{out}} \right) / 2\) and \(\overline{T}_i = \left( T_{i,\text{in}} + T_{i,\text{out}} \right) / 2\) are the average moist air specific heat and temperature within the cell, respectively. \(T_s\) is the wall surface temperature in contact with the air cell, and \(P\) and \(s\) are the contact perimeter and length in the direction of fluid flow, respectively.

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The mass balance within any air cell can be evaluated by the following relation:

\[ \dot{m}_i \cdot dW_i = \sum_{j=1}^{n} -\alpha_{ij} (\bar{W}_j - W_{sat,j}) P_j \cdot ds_j, \quad \text{if} \ (T_j < \bar{T}_{dp}) \tag{2a} \]

\[ \dot{m}_i \cdot dW_i = 0, \quad \text{if} \ (T_j \geq \bar{T}_{dp}) \tag{2b} \]

where

\[ \alpha_{ij} = \frac{\alpha_{a,ij}}{Le_i^{2/3} \cdot C_{p,ma}}, \quad \text{the mass transfer coefficient based on the Chilton-Colburn analogy (Threlkeld 1970).} \]

\[ \bar{W}_i = \frac{W_{in} + W_{out}}{2}, \quad \text{the average air humidity ratio within the cell.} \]

\[ \bar{T}_{dp} = \frac{T_{dp}^{in} + T_{dp}^{out}}{2}, \quad \text{the average air dew point temperature within the cell.} \]

**Refrigerant governing equations**

The heat balance in each refrigerant cell is evaluated by Equation 3, which is similar to Equation 1; however, the enthalpy difference is used instead of the temperature difference.

\[ \dot{m}_i \cdot dh_j = \sum_{j=1}^{n} -\alpha_{r,ij} (\bar{T}_j - T_j) P_j \cdot ds_j \tag{3} \]

**Wall governing equations**

The 2D energy balance within any wall cell \( j \) in contact with \( n_j \) fluid cells \((i = 1, n_j)\) is represented by Equation 4:

\[ \nabla \left( k_{w,i} \cdot \nabla T_{w,j} \right) + \sum_{i=1}^{n_j} \frac{1}{t_{w,i}} \cdot \dot{q}_j = 0 \tag{4} \]

where \( T_{w} \) is the wall temperature evaluated at the cell centroid, and \( t_w \) and \( k_w \) are the wall cell thickness and thermal conductivity, respectively. The second term in Equation 4 represents the summation of total heat transfer to the current wall cell \( j \) from the surrounding fluid cells.

The total heat transfer term in Equation 4 could be expressed as follows:
a) for air cell:
\[
\dot{q}_j = k_{w,j} \frac{T_{s,j} - T_{w,j}}{t_{w,j}/2} = \alpha_{a,j} (T_i - T_s) + \alpha_{E,j} h_{fl} (\bar{W}_i - W_{sat,s})
\]  
\[\text{(5a)}\]

b) for refrigerant cell:
\[
\dot{q}_g = k_{w,g} \frac{T_{s,j} - T_{w,g}}{t_{w,g}/2} = \alpha_{r,j} (T_i - T_s) = U_g (T_i - T_{w})
\]  
\[\text{(5b)}\]

A linearization scheme (Patankar 1980) is required to simplify Equation 5a and to solve the wall differential equation. McQuiston (1975) considered the following scheme or assumption for \(W_{sat,s}\):

\[
W_{u} - W_{sat,s} = C(T_{u} - T_{s})
\]  
\[\text{(6)}\]

where \(C\) is a constant. Therefore, Equation 4 can be solved relatively easily. However, Wu and Bong (1994) argued that this assumption is physically inappropriate. Therefore, instead of the assumption of a constant \(C\) value they assumed a linear relation to relate the humidity ratio of saturated air to its temperature as follows:

\[
W_{sat,s}(T_{s,j}) = a_{ij} + b_{ij} \cdot T_{s,j}
\]  
\[\text{(7)}\]

Sharqawy and Zubair (2007) suggested the following relations to calculate the slope \(b\) and constant \(a\) in Equation 7:

\[
b_{ij} = \frac{\bar{W}_i - W_{sat,s,j}}{T_{dp,i} - T_{s,j}}
\]  
\[\text{(8)}\]

and
\[ a_{ij} = W_{s, t, s, j} - \left( \frac{W_t - W_{s, t, s, j}}{T_{dp, j} - T_{s, j}} \right) T_{s, j} \]  \quad (9)

where \( b \) is the average slope of the saturation line between the wall surface temperature \( T_s \) and average dew point of surrounding air \( T_{dp} \), as illustrated in Figure 3.

After substituting Equations 7 and 8 into Equation 5a and taking into account the Chilton-Colburn analogy, Equation 5a can now be expressed as follows:

\[ \dot{q}_{ij} = U_{w, ij} \left( T^*_{ij} - T_{w, j} \right) \]  \quad (10)

where

\[ U_{w, ij} = \frac{1}{t_{w, i} / 2 + \frac{1}{k_{w, j} + \alpha_{w, j}}} \]

the overall heat transfer coefficient for the wet case, which accounts for total (sensible and latent) convection resistance and conduction resistance within a half thickness of the wall cell.

\[ \alpha_{w, j} = \alpha_{s, j} (1 + \beta_{f, j} \cdot b_{ij}) \]

the total heat transfer coefficient for the wet case, which accounts for sensible and latent heat transfer; if there is no dehumidification then \( \alpha_w = \alpha_s \).

\[ \beta_{f, j} = h_{fg, j} \left( \frac{c_{p, w, j}}{\rho_{w, j}} \right)^{2/3} \cdot \bar{C}_{p, w, j} \]

the moist air temperature parameter; if there is no dehumidification then \( T^*_{ij} = T_{ij} \).

The numerical scheme corresponding to a linear fluid temperature variation (LFTV), as explained in Corberan et al. (2001), is employed for the discretization of the heat and mass transfer in Equations 5a and b. This numerical scheme is basically based on assuming a piecewise distribution of the fluid temperature and humidity ratio (in the case of air) along the fluid cell.

The discretization of the Laplacian operator in Equation 4 can be made by a classical finite difference (finite volume) approach. The corresponding boundary conditions are the prescribed inlet temperature and humidity ratio (in the case of air) beside the velocity distributions for both fluids. The open edges of the tubes to the air are considered adiabatic. The global solution
method employed in the current work is a semi-explicit method for wall-temperature linked equations (SEWTLE) (Corberan et al. 2001).

**Case study**

In this case study, a minichannel evaporator has been modeled. Its dimensions (Table 3) are based on the evaporator tested by Zhao et al. (2001). Only the tube length has been modified according to the scope of the current work, because the heat and mass transfer mechanisms are independent of the tube length and the number of tubes for the developed model.

**Verification of the Fin2D model**

Before using the newly developed model to produce detailed solutions of heat transfer in the analyzed portion of the minichannel evaporator, a verification process is required.

The verification procedure had to consist of two steps: firstly, air-side verification (V1), in which the conductivity of the tube and fin was assumed to be infinite and the fin was discretized only in the direction of air flow (z-direction), and secondly, fin temperature profile verification (V2), where the air heat capacity rate was assumed to be infinite and the fin was discretized only in the direction between tubes (y-direction). To allow a fair comparison against analytical solutions, the following additional assumptions were used: the longitudinal conduction on both fin and tube walls and the transverse conduction on the tube wall were disabled, and constant properties and heat transfer coefficients were used. On the other hand, conduction along the y-direction in the fin walls was kept enabled in order to verify the calculation of heat transferred to the fins.
In the case of a totally wet fin, V1 verification was divided into two further verifications: one for sensible heat transfer and the other for latent heat transfer. In both verifications, the numerical solution was compared with the analytical one based on a zero heat capacity ratio heat exchanger (Incropera and DeWitt 2006, p. 689), Ū= 1-exp (-NTU). Finally, the total heat transfer from the air (summation of the sensible and latent heats) was compared.

Figure 4 shows the error of the numerical solution with reference to the analytical one for the case of V1 under totally wet and totally dry fin conditions. The figure demonstrates that the error tends to diminish very quickly with the number of cells used. The abscissa shows the number of fin cells in the z-direction. As can be observed, the error is already very small for N = 5.

The results of V2 are depicted in Figure 5 for a totally wet fin as a function of the number of fin cells in the y-direction, considering two situations: equal tube temperatures at the bottom and the top, and a temperature difference of 5 K between tubes. θ is the difference between the fin temperature and the air temperature. The analytical solutions for both cases were taken from Wu and Bong (1994), and Sharqawy and Zubair (2008). As seen from the figure, there is a very good agreement between the numerical and analytical temperature profiles, especially with the increase in number of the fin cells in y-direction.

Comparative study between the classical ε-NTU method and Fin2D numerical results

Once the Fin2D model has been verified it can be used as the reference to check the error made by the classical methods, which are usually used to analyze the performance of any heat exchanger. The solutions to each operation scenario analyzed below are obtained with the Fin2D model using a detailed grid: {3,1,10,30,10}. 

The major disadvantage of the logarithmic mean temperature or enthalpy difference method (LMTD or LMHD) is that the outlet conditions for both air and refrigerant have to be known. Nevertheless this method could still be used for the analysis; however the procedure would require more iterations. One way to simplify the heat transfer rate calculations and reduce or eliminate the iterations in the solution of the discretized heat exchanger is to apply the effectiveness-NTU (UNTU) method (Kays and London 1984). This method is based on the heat transfer effectiveness (\( \varepsilon \)) and number of transfer units (NTU).

Most of the simulation models divide each evaporator tube into segments along the refrigerant flow with its corresponding fins. Once the evaporator has been divided into segments, the adiabatic fin tip assumption and classical UNTU relationships for heat exchangers are employed to solve the heat and mass transfer for each segment. This method simplifies the solution and the calculation time, but on the other hand it has many drawbacks, as discussed before. However the presence of the dehumidification process shows some other drawbacks such as:

1. **Constant air temperature and humidity ratio along the y-direction**: besides what was discussed in Martínez-Ballester et al. (2010) about this topic, now the constant temperature within the y-direction also results in a constant humidity ratio in the same direction.

2. **No accounting for partially wet fin condition**: actually, depending on the fin-base temperature, the fin-tip temperature, and the dew point of the air, the fin surface can be fully dry, fully wet, or partially wet. In the UNTU approach, the identification of the surface area below or above the dew point along both the tube and the associated fin appears to be difficult. Thus, in this approach the whole segment is usually assumed to be either
completely dry or completely wet based on the following condition proposed by Jiang et al. (2006) and usually used by many other authors:

If \( \bar{T}_s < \bar{T}_{dp} \) then the whole segment will be assumed to be totally wet; otherwise it will be assumed to be totally dry. \( \bar{T}_s \) is the average wall surface temperature for the tube and fin which is calculated under the dry fin condition assumption as follows:

\[
\bar{T}_s = \eta_{f, v} \left( T_{fb} - \bar{T}_a \right) + \bar{T}_a
\]

3. **Constant saturation line slope \( b \) within the entire segment:** in the \( \bar{\text{U}} \)NTU method, the slope \( b \) of the saturation line is assumed to be constant for the whole segment. It is usually evaluated at the fin mean temperature or as the average value between the fin base and tip conditions.

In the current study, three cases, which represent different applications of refrigeration and air conditioning, have been chosen. In each case a different range of tube temperatures has been selected to simulate and evaluate the evaporator performance under different dehumidifying conditions. Those cases are defined as follows:

- **Case (I), high-temperature refrigeration applications:** Most vegetables and fruits are stored, in the short term, in a temperature range varying from 7.2 to 15.6 °C and relative humidity range of 80 to 95% to prevent dehydration (Whitman et al. 2012, p. 753). In the current study, the evaporator inlet air conditions were kept at 14 °C and a dew point of 12.39 °C (\( \approx 90\% \) RH).

- **Case (II), air-conditioning (cooling and dehumidification) applications:** ASHRAE (2011, p. 3.2) specified general guidelines for temperature and humidity applicable to offices and
common areas; for example, in summer the indoor temperature should range from 23.3 to 26.7 °C while the relative humidity should range from 50 to 60%. According to the previous thermal comfort definition of ASHRAE, the inlet air conditions to the evaporator in the current case were set at 27 °C and a dew point temperature of 17.2 °C (≈ 55% RH).

- **Case (III), heat pump drying applications:** Heat pump dryers have been known to be energy efficient when used in conjunction with drying operations. The hot humid air which exits from the drying process is forced to flow through an evaporator for cooling and removing the excess moisture before heating it again through the condenser to increase its potential for humidity transport. Pendyala et al. (1986) have stated that in typical drying applications, for example cloth drying, the air conditions are in the range of 25 to 65 °C and 40 to 100% RH. So in the current situation the evaporator inlet air conditions were fixed at a dry bulb temperature of 40 °C and a dew point of 32.69 °C, which corresponds to ≈ 67% RH.

Table 4 summarizes the different inlet conditions for the evaporator, which were also used in the verification study. The air-side sensible heat transfer coefficient was estimated to be 48 W/m²·°C assuming fully laminar flow in non-circular tubes (Incropera and DeWitt 2006, p. 519). Since the air velocity does not change the value remained the same for all the studies. For air–water vapor mixtures, the Lewis number is close to unity (Coney et al. 1989), so a Lewis number of unity was assumed in all cases.

In the current study only the air-side of the evaporator is considered, assuming constant upper and lower tube temperatures. In ÚNTU modeling the fin is cut and adiabatic fin tip efficiency is adopted. The thermal effectiveness relationship used is similar to the one used in
verification study. This relationship is fundamentally correct for all cases with constant tube
temperature and flow boiling inside tubes, as encountered in evaporators.

**Results and discussion**

Figures 6 to 8 show the relative deviation (based on Fin2D results) in heat transfer between
the numerical model and the classical ŪNTU method for the three cases under study. It can be
noticed that the difference between the single and dual potential models is very small and does
not affect the results much. It is worth mentioning that Wu and Bong (1994), and Sharqawy and
Zubair (2008) reported a similar result in their analytical analysis of a totally wet straight fin.

**Totally dry fin analysis**

The fin is considered fully dry if the temperature of its surface at any location is equal to or
greater than the surrounding air dew point temperature. Under this situation, only sensible heat
transfer occurs between the air and the wall.

It can be concluded from Figures 6ī8 (a and c) that for each case the deviation in sensible
and total heat for the totally dry fin is not affected by the change in the tube wall temperature.
This deviation could result from the assumption of a constant air temperature between tubes
which is adopted by ŪNTU. However, the temperature of air close to the tube wall and fin roots
is actually very different from the bulk air temperature. This fact has an important impact on
local effects controlling the heat transfer and contributes significantly to the deviation between
the ŪNTU method and the current model results. Similar results and conclusions were reported
by Martínez-Ballester et al. (2010).
The average value of the deviation in sensible heat for the three cases is 3.94%. The deviation value decreases very slightly (0.1% in total) with increase in the moist air specific heat, which depends on the humidity ratio of the air.

*Partially wet fin analysis*

A partially wet fin is a transition condition between totally dry and totally wet conditions. In this scenario the tube surface temperature is slightly less than the dew point of air. This results in some locations on the fin surface having a temperature lower than the average dew point of the surrounding air, and thus simultaneous heat and mass transfer occurs. Meanwhile, the remaining fin surface still has a temperature higher than the dew point of the surrounding air, which results in only sensible heat transfer.

The range of partially wet fins is very small, as seen in Figures 6, 7, and 8. The maximum range is about 0.5 °C. Normally the multi-louvered fin which is employed in minichannel evaporators is short with an average height of 8 mm and a very small thickness of ≈ 0.1 mm. These specifications lead to a very high fin efficiency, even under wet conditions, which could result in this small partially wet fin range.

At the beginning of the partially wet region, the classical NTU method still considers the fin to be totally dry because the average segment wall temperature is still higher than the average dew point of air. This results in an underestimation of the latent heat by 100% compared with numerical results. Figure 9 shows the fin temperature profile and the amount of water condensed from the air in milligrams per hour for case III \( (T_{a,in} = 40 \text{ °C}, \ RH_{a,in} = 67\%) \) at \( T_{tube} = 32.60 \text{ °C} \). It can be clearly noticed that the NTU method fails to predict the fin status and the fin is actually under a partially wet condition.
With further decrease in the tube wall temperature, the percentage of the wet region is increased and the ÜNTU method starts to assume a totally wet fin. This assumption results in an overestimation of the latent heat compared with the Fin2D model and a rapid increase in the deviation between the two approaches for all cases, as seen in Figures 6b, 7b, and 8b. Numerical results for case II (\(T_{a,in} = 27 \, ^\circ\text{C}\) and \(\text{RH}_{a,in} = 55\%\)) are illustrated in Figure 10, where \(T_{\text{tube}} = 17 \, ^\circ\text{C}\). It can be seen that the wet region occupies nearly 40% of the fin surface area, while the rest is dry.

The misprediction of the fin condition and the amount of latent heat transfer by ÜNTU also affects the amount of sensible heat transfer in the partially wet region. The assumption of a totally wet fin, even though the fin is actually under the partially wet condition, results in an increase in the fin surface temperature due to the release of latent heat of condensation. A higher fin temperature results in a lower heat transfer potential between air and fin, which causes a decrease in the relative deviation of the sensible heat. As seen in Figure 8a, in the case of extremely high air temperature and relative humidity, as encountered in case III, the assumption of a totally wet fin results in a higher estimation of condensed water on the fin surface, which suppresses the sensible heat transfer substantially. Consequently, the ÜNTU method starts to underpredict the amount of sensible heat by 1.8% compared with Fin2D.

**Totally wet fin analysis**

With further decrease in the tube wall temperature, the whole fin surface temperature becomes much lower than the dew point of the surrounding air, and the fin becomes totally wet.

At the beginning of the totally wet region, the deviation in latent heat starts to decrease rapidly because the two approaches (Fin2D and ÜNTU) now predict the same fin condition.
Further lowering of the tube wall temperature results in a small decrease in the latent and total heat relative deviations and in an increase in the relative deviation of the sensible heat. The average values of deviation in sensible, latent, and total heat for the three cases under the totally wet condition are shown in Table 5.

It can be concluded from the results that generally the ŪNTU method overpredicts the amount of sensible, latent, and total heat in the fully wet region, except in case III at the beginning of the totally wet fin, where the ŪNTU method slightly underpredicts the amount of sensible heat. The assumption of no temperature variation of the air along the y-direction, which also results in a constant humidity ratio within the same direction, could be one of the reasons for this deviation in results. Also, the way of calculating the saturation line slope $b$, which is usually assumed by the ŪNTU method to be constant for the whole segment, could contribute to this deviation. However in the current Fin2D model this slope is calculated individually at each wall cell based on the surface wall temperature and the average conditions of the adjacent air cells.

Conclusions

A 2D numerical model which accounts for the air-side performance of minichannel evaporators under different dehumidifying conditions was presented. After verification against an analytical solution, a comparative study between the classical ŪNTU method and the current model results was done under different dehumidifying conditions. The following are the main conclusions of the study:

- The difference in results between the ŪNTU method based on the single potential model and the ŪNTU method based on the dual potential model is negligible.
• For a totally dry fin, the ÛNTU method always overestimates the amount of sensible heat for all cases under study with an average relative deviation of 3.94% regardless of the tube wall temperature. Only a small decrease is encountered with increase in the moist air specific heat, which depends on the humidity ratio of the air. The main source of this deviation could be the assumption of constant air temperature between tubes.

• For the partially wet fin, the ÛNTU method always fails to predict the right fin condition because it does not account for the partially wet scenario (drawback 2). Due to the assumption of a fully wet fin, the ÛNTU method always overestimates the amount of latent heat, and consequently the relative deviation in latent heat increases significantly. This assumption also affects the amount of sensible heat transfer due to the increase in fin surface temperature. The average deviation in total heat for the three cases is 3.77%.

• The high fin efficiency, even under the wet condition, results in a narrow range of the partially wet fin of a maximum of 0.5 °C. However, we anticipate a further increase in the partially wet fin range in the presence of a temperature difference between neighboring tubes.

• For the totally wet fin, generally the ÛNTU method overpredicts the amount of sensible, latent, and total heat for all cases under study. The average deviations in sensible, latent, and total heat transfer are 3.45, 22.91, and 4.11% respectively. The main sources of those deviations are the assumption of constant temperature and humidity ratio between tubes (drawback 1) and the assumption of a constant saturation line slope \( b \) within the whole segment which is evaluated at the average fin temperature (drawback 3).
Generally the őNTU model still has a lower computational time, for the current study it takes 1–4 iterations to converge, compared with the Fin2D model which takes 15–20 iterations. Also, the őNTU method predicts the amount of heat transfer quite well, especially for the totally dry and totally wet fins, compared with the current numerical results. However, in the partially wet region the deviations between the two approaches are increased. We expect more deviations, especially in latent heat, in the presence of a temperature difference between the upper and lower tubes (superheat).

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**Nomenclature**

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<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>$A$</td>
<td>surface area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$A_c$</td>
<td>cross-section area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$a$</td>
<td>parameter defined in Eq. (9)</td>
<td>kg$_w$/kg$_a$</td>
</tr>
<tr>
<td>$b$</td>
<td>slope of saturated humidity ratio line</td>
<td>1/°C</td>
</tr>
<tr>
<td>$b_h$</td>
<td>slope of saturated enthalpy line</td>
<td>J/kg·°C</td>
</tr>
<tr>
<td>$C$</td>
<td>constant defined in Eq.(6)</td>
<td>kg$_w$/kg$_a$·°C</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat</td>
<td>J/kg·°C</td>
</tr>
<tr>
<td>$D_f$</td>
<td>fin depth</td>
<td>m</td>
</tr>
<tr>
<td>$H_f$</td>
<td>fin height</td>
<td>m</td>
</tr>
</tbody>
</table>

Subscripts:

- $a$: air
- $fb$: fin base
- $dry$: dry surface condition
- $dp$: dew point
- $f$: fin
- $i$: fluid cell index
- $j$: wall cell index
- $k$: direction index
$h$ specific enthalpy (J/kg)  
$h_{fg}$ latent heat of water condensation (J/kg)  
$k$ conductivity (W/m·°C)  
$L_f$ fin length (m)  
$Le$ Lewis number (–)  
$M$ wet fin parameter (1/m)  
$m$ dry fin parameter (1/m)  
$m_a$ mass flow rate (kg/s)  
$NTU$ number of transfer units (–)  
$P$ perimeter (m)  
$Q$ heat transfer (W)  
$\dot{q}$ heat flux (W/m²)  
$RH$ relative humidity (%)  
$T$ temperature (°C)  
$T^*$ air temperature parameter defined in Eq. (10) (°C)  
$t$ thickness (m)  
$U$ overall heat transfer coefficient (W/m²·°C)  
$U_w$ overall heat transfer coefficient for wet case (W/m²·°C)  
$W$ humidity ratio (kgₜ/kgₚ)  
$x,y,z$ spatial coordinates (m)  
$\alpha$ sensible heat transfer coefficient (W/m²·°C)  
$\alpha_D$ mass transfer coefficient (kg/m²·s)  
$\alpha_w$ total heat transfer coefficient for wet case (W/m²·°C)  
$\beta$ parameter in Eq. (10) (°C)  
$\Theta$ thermal effectiveness (–)  
$\eta$ thermal efficiency (–)  
$\Delta$ temperature difference = $T_a - T_s$ (°C or K)
References


Table 1. Local heat transfer rates: single and dual potential models.

<table>
<thead>
<tr>
<th></th>
<th>Single potential</th>
<th>Dual potential</th>
</tr>
</thead>
<tbody>
<tr>
<td>General equation</td>
<td>$dq = dA \left[ \alpha_a (T_a - T_s) + \alpha_{D} \cdot h_{fg} \left( W_a - W_{sat,a} \right) \right]$</td>
<td></td>
</tr>
<tr>
<td>General assumptions</td>
<td>$\alpha_{D} = \frac{\alpha_a}{C_{p,ma} \cdot Le^{2/3}}$</td>
<td>$h_{fg} = h_{fg,water} (T_a) - h_{f,water} (T_s)$</td>
</tr>
<tr>
<td>Model assumptions</td>
<td>$\left[ \frac{(W_s - W_{sat}) (h_{fg} - h_{fg}^* \cdot Le^{2/3})}{Le^{2/3}} \right] &lt;&lt; h_s - h_{sat,s}$</td>
<td>$\bar{I}$</td>
</tr>
<tr>
<td>Local heat transfer rate for surface element ($dA$)</td>
<td>$dq = \frac{\alpha_a}{C_{p,ma}} \cdot dA (h_s - h_{sat,s})$</td>
<td>$dq = \alpha_{s} \cdot dA \left[ (T_s - T_a) + \frac{1}{C_{p,ma} \cdot Le^{2/3} \cdot h_{fg}} (W_s - W_{sat}) \right]$</td>
</tr>
</tbody>
</table>
Table 2. Fin analysis and efficiency: single and dual potential models.

<table>
<thead>
<tr>
<th></th>
<th>Single potential</th>
<th>Dual potential</th>
</tr>
</thead>
<tbody>
<tr>
<td>Local heat transfer</td>
<td>$dq_f = \frac{a_u}{C_{p,ma}} \cdot dA_f \left(h_a - h_{s,t,f} \right)$</td>
<td>$dq_f = a_u \cdot dA_f \left(T_a - T_{f}\right) + \frac{1}{C_{p,ma} \cdot Le^{2/3}} \cdot h_{b} \left(W_a - W_{s,t,f} \right)$</td>
</tr>
<tr>
<td>rate for fin element</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(dA_f)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Model assumptions</td>
<td>$dq_f = \alpha_u \cdot dA_f \left(T_a - T_{s,f} \right)$</td>
<td>$C = \frac{W_a - W_{s,t,f}}{T_a - T_{s,f}} = \frac{W_a - W_{s,t,b}}{T_a - T_{b}}$</td>
</tr>
<tr>
<td></td>
<td>$\alpha_u = \frac{a_u \cdot b_u}{C_{p,ma}}$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$b_u = \frac{h_u}{T_{ub} - T_{s,f}}$ or $= \frac{dh_ua}{dT} \bigg</td>
<td><em>{T=T</em>{s,f}}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(McQuiston 1975)</td>
</tr>
<tr>
<td></td>
<td></td>
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</tr>
<tr>
<td>Fin efficiency</td>
<td>$\eta_{sat,f} = \frac{\tanh \left(M \cdot L_f \right)}{M \cdot L_f}$</td>
<td>$\eta_{sat,f} = \frac{\tanh \left(M \cdot L_f \right)}{M \cdot L_f}$</td>
</tr>
<tr>
<td>(adiabatic fin tip</td>
<td></td>
<td></td>
</tr>
<tr>
<td>assumption)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$M = m \cdot \sqrt{\frac{b_u}{C_{p,ma}}} \quad \text{where} \quad m = \frac{P_f \cdot \alpha_s}{k_f \cdot A_{f}}$</td>
<td></td>
</tr>
<tr>
<td></td>
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</tbody>
</table>
Table 3. Geometry of the minichannel evaporator.

<table>
<thead>
<tr>
<th></th>
<th>Tube length (refrigerant flow direction)</th>
<th>Fin pitch (m)</th>
<th>Channel diameter (m)</th>
<th>Tube depth (air flow direction)</th>
<th>Fin thickness (m)</th>
<th>Number of channels (i)</th>
<th>Tube thickness (m)</th>
<th>Fin height (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube length</td>
<td>8.6 cm</td>
<td>1.5 m</td>
<td>1</td>
<td>1.5 m</td>
<td>0.1 m</td>
<td>52</td>
<td>0.5 m</td>
<td>8 m</td>
</tr>
<tr>
<td>Fin pitch</td>
<td>6 m</td>
<td>9 m</td>
<td></td>
<td>6 m</td>
<td>52 m</td>
<td></td>
<td>5 m</td>
<td>8 m</td>
</tr>
<tr>
<td>Channel diameter</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tube depth</td>
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<td></td>
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<td></td>
<td></td>
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<tr>
<td>Fin thickness</td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of channels</td>
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<td></td>
<td></td>
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<tr>
<td>Tube thickness</td>
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<tr>
<td>Fin height</td>
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<td></td>
</tr>
</tbody>
</table>
Table 4. Inlet conditions used in the simulations.

<table>
<thead>
<tr>
<th>Inlet pressure (kPa)</th>
<th>Inlet temperature (°C)</th>
<th>Inlet dew point (°C)</th>
<th>Inlet relative humidity (%)</th>
<th>Mass flux (kg/m²·s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>100</td>
<td>14, 27, 40</td>
<td>12.39, 17.20, 32.69</td>
<td>55%, 67%, 90%</td>
</tr>
</tbody>
</table>
Table 5. Average deviations in heat transfer under totally wet condition.

<table>
<thead>
<tr>
<th></th>
<th>Case I</th>
<th>Case II</th>
<th>Case III</th>
<th>Total average</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Sensible heat</strong></td>
<td>3.54</td>
<td>3.10</td>
<td>3.70</td>
<td>3.45</td>
</tr>
<tr>
<td><strong>Latent heat</strong></td>
<td>28.38</td>
<td>19.48</td>
<td>20.86</td>
<td>22.91</td>
</tr>
<tr>
<td><strong>Total heat</strong></td>
<td>4.12</td>
<td>4.11</td>
<td>4.10</td>
<td>4.11</td>
</tr>
</tbody>
</table>

* % Relative deviation = \[
\left\{\frac{Q_{\text{e-NTU}} - Q_{\text{e-2D}}}{Q_{\text{e-2D}}}\right\} \times 100
\]
Figure 1. Schematic of cooling and dehumidifying of moist air.
Figure 2. (a) A piece of the evaporator under study, (b) schematic of the discretization in a segment of the evaporator.
Figure 3. Linear approximation of the saturation curve in psychrometric chart.
Figure 4. V1 results for: (a) totally wet fin condition, (b) totally dry fin condition.
Figure 5. V2 results for totally wet fin: (a) no temperature difference between tubes, (b) temperature difference of 5 K between tubes.
Figure 6. The relative deviations in heat transfer results for Case I ($T_{a,in} = 14 \, ^\circ C$ and $RH_{a,in} \approx 90\%$).
Figure 7. The relative deviations in heat transfer results for Case II ($T_{a,in} = 27 \, ^{\circ}C$ and $RH_{a,in} \approx 55\%$).
Figure 8. The relative deviations in heat transfer results for Case III ($T_{a,in} = 40\,^\circ C$ and $RH_{a,in} \approx 67\%$).
Figure 9. (a) Fin temperature profile, (b) mass flow rate of condensed water for case III ($T_{a,in} = 40 \, ^{\circ}\text{C}$, $\text{RH}_{a,in} = 67\%$) at $T_{tube} = 32.60 \, ^{\circ}\text{C}$. 
Figure 10. (a) Fin temperature profile, (b) mass flow rate of condensed water for case II ($T_{a,in} = 27^\circ C$, $RH_{a,in} = 55\%$) at $T_{tube} = 17^\circ C$. 