INSTITUTO DE INGENIERÍA ENERGÉTICA
UNIVERSIDAD POLITÉCNICA DE VALENCIA

CPI, Edificio 8E, Cubo F, 5ª planta, Camino de Vera s/n, 46022 Valencia, Spain Tel. (34) 963877270, Fax (34) 963877272, Email: energeti@upvnet.upv.es

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(Institute for Energy Engineering)

Research Publications

Impact of Dehumidification Modelling on the Performance Prediction for Minichannel Evaporators

Abdelrahman HUSSEIN HASSAN[1*], Santiago MARTÍNEZ-BALLESTER[1], José GONZÁLVEZ-MACIÁ[1]

[1] Universitat Politècnica de València, Institute for Energy Engineering, Camino de Vera, s/n, Valencia, 46022, Spain
Contact Information (Tel: +34 963 879 121, Fax: +34 963 879 126, E-mail: abhusab1@upvnet.upv.es)

*Corresponding Author

ABSTRACT

In this paper a two-dimensional numerical model for a minichannel evaporator is implemented. This model takes into account the variation of wall (fin and tube) temperature and moist air properties (temperature and humidity ratio) in both longitudinal and transverse directions. The validation of the current model is done with a well-defined analytical case, the results of both cases show a very good agreement and consistency. A case study from literature has been chosen to compare between the current model results, and the results from traditional ε-NTU method with adiabatic fin-tip assumption which used generally by many authors for modeling and analyzing the evaporator performance. The tube wall temperature is varied from 1.4 ºC to 17 ºC, under constant inlet air temperature and humidity ratio. These conditions allow different scenarios for the tube and fin (totally wet, totally dry, or partially wet). Deviations in results between the current model and the traditional ε-NTU approach are noticed, especially under partially wet fin condition. These deviations are mainly due to the assumptions which normally adopted by the ε-NTU method and fin theory such as; no variation in moist air temperature and humidity ratio along the direction between tubes, the whole segment (tube and fin) is usually assumed to be either completely dry or wet and no accounting for partially wet scenario.

1. INTRODUCTION

Nowadays, the minichannel heat exchangers have started to replace in some applications, especially in refrigeration and air conditioning, the traditional ones (tube-fin, tube-wire, etc.) because of many advantages such as; improvement of the heat transfer, higher compactness and reduction of the refrigerant charge which decreases carbon footprint caused by direct refrigerant emissions.

When the surface temperature of the minichannel heat exchanger is below the dew point temperature of the incoming air (in the case of evaporator) the process of cooling and dehumidification is carried out. The fins and tubes of the evaporator become partially or totally wet depending on the surface temperature and inlet air properties (temperature and humidity ratio). The fin performance is dramatically influenced by the combined heat and mass transfer associated with the cooling and dehumidification of air. The condensation of moist air over the fin surface attributes to the decrease of fin efficiency. Example of calculations done by Hong and Webb (1996), showed that the wet fin efficiency can be as much as 35% below the dry surface fin efficiency. Since the fin performance under wet conditions may be very different from the dry conditions, special attention is needed for the wet fin performance analysis.

An extensive review on the theoretical aspects and design methods concerning heat and mass transfer process during cooling and dehumidification of moist air was presented by Kandlikar (1990).Two methods are proposed to model the combined heat and mass transfer. One is the single-potential method which considers the enthalpy difference between the flowing air stream and the saturated air at water surface temperature in the heat transfer calculation. The other is the two-potential method which considers the sensible heat transfer due to temperature difference of moist air and cooling surface and the latent heat transfer due to the specific humidity difference between the flowing air stream and the saturated air independently.

A lot of experimental and numerical studies were implemented by many authors to analyze and study the performance and efficiency of the fin under different (dry and wet) conditions. Numerically, Liang *et al.* (2000) presented a comparative study to investigate the wet-surface fin efficiency of a plate-fin-tube heat exchanger under a variety of fin geometric parameters and airflow conditions, especially for a wide range of air relative humidity. The results demonstrated the following; 1) there are two different moist condensation features exist for wet fin, i.e. partially wet and totally wet, for a partially wet fin, the fin efficiency decreases rapidly with the increase in air inlet relative humidity, for a totally wet fin, the effect of air inlet relative humidity on the fin efficiency is small. 2) The 2D numerical model takes into account the complex fin geometry and the variation of moist air properties over the fin, in contrast of 1D model. Experimentally, Lin *et al.* (2001) performed a detailed study concerning the performance of a rectangular fin in both dry and wet conditions. Their results showed that the dry fin efficiency is about 15–25 percent higher than that of the corresponding wet fin efficiency. In the case of partially wet surface, a considerable influence of the relative humidity on the fin efficiency is encountered.

Many authors introduced a lot of complete (finned-tube) evaporator models, for example; Domanski (1999) developed a model for an evaporator based on tube-by-tube discretization schema and traditional ε-NTU method to predict the evaporator capacity, his model allows for specification of complex refrigeration circuits and accounting for non-uniform air distribution. Another, numerical model was introduced by Jiang *et al.* (2006) in which a segment-by-segment approach was implemented to account for 2D non-uniformity of air distribution across the heat exchanger, and heterogeneous refrigerant flow patterns through a tube. Also, their model utilizes the adiabatic wet fin assumption based on two-potential method and ε-NTU heat flux equations for combined heat and mass transfer.

Vast amount of literatures including the above ones have analyzed extensively the performance of fins and macro (traditional) evaporators under dehumidification. On the other hand, the impact of dehumidification on the fin efficiency which reflects directly on the performance of minichannel evaporators is still under investigation by many authors.

Currently, several minichannel evaporator models are available in the literature; most of them use the traditional ε-NTU and the adiabatic wet fin tip efficiency approach depending on the enthalpy potential which proposed by Threlkeld (1970), such as; Wu and Webb (2002), Zhao *et al.* (2012), and Gossard *et al.* (2013). Although, the classical ε-NTU modeling does not account for; longitudinal conduction neither in the fin nor in the tube, transverse conduction in the tube, and the heat conduction between different tubes, which are a consequence of employing the adiabatic fin tip assumption. Emphasizing on those drawbacks, Martínez-Ballester *et al.* (2010) developed a detailed 2D numerical model for CO_2 minichannel gas cooler to capture heat conduction effects within its structure and a detailed representation of air properties. Their study revealed large errors in capacity prediction of individual tubes due to the adiabatic fin tip assumption, especially when the neighboring tubes are at different temperatures.

The current work presents a numerical model of a minichannel evaporator under dehumidification that accounts for 2D heat conduction in any element (fin or tube) and the variation of moist air properties (temperature and humidity ratio). The model, referred to as Fin2D model, subdivides the evaporator into segments, to which the corresponding system of energy-conservation equations is applied without traditional heat exchanger modeling assumptions. After validation, the numerical results of the current model will be compared with traditional ε-NTU method which commonly used, under different conditions; totally dry, totally wet and transition from dry to wet (partially wet).

2. FIN2D EVAPORATOR MODEL

2.1 Evaporator Discretization

Figure (1a), presents a piece of the studied minichannel evaporator. It is discretized along the X-direction (refrigerant flow) in a number of segments “a”. Each segment (Figure 1b) consists of: two streams of refrigerant (top and bottom flows) that are split into “b” channels in the Z-direction (air flow); two flat tubes (top and bottom) that are discretized into “c” cells in the Z-direction; and both air flow and fins, which are discretized in two dimensions: “d” cells in the Y-direction and “e” cells in the Z-direction. This is summarized in the text as; grid: {a,b,c,d,e}. For illustration of the nomenclature, the numerical example shown in Figure (1) corresponds to a grid: {2,5,3,6,5}.

All grid dimensions are independent, with the only exception that the air and fin have the same discretization. The heat (sensible and latent) 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 fin cells conduct the heat along the plane Y-Z and also to the wall cells in contact at the fin roots. The tube cells exchange the heat by conduction between each others in X-Z plane, and transfer it by convection to the refigerant cells.

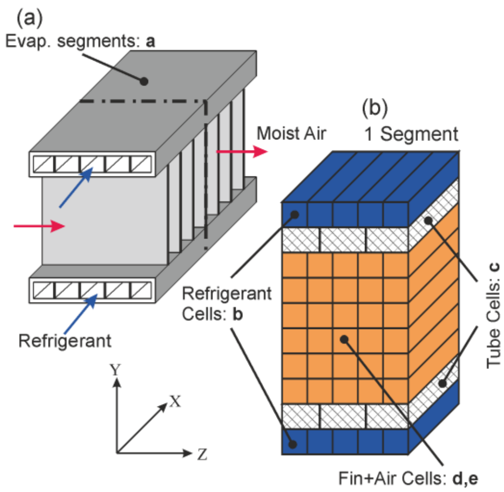


Figure 1: (a) a piece of the evaporator under study, **(b)** Schematic of the discretization in a segment of the evaporator.

2.2 Governing Equations

Every fluid cell (refrigerant or air) has two nodes, one at the inlet and one at the outlet. The wall cells (tube or fin) have only one node located in the centroid of the cell. In this situation, the governing equations for fluids and wall can be written as follows:

- *Air governing equations*:

The two-potential method proposed by McQuiston (1975) is adopted in the current work to analyze the total heat transfer from the moist air. The main advantages of this method that it allows for the independent evaluation of sensible and latent heat contribution, and continuous evaluation of total heat in the transitions from humid to dry condition. The sensible heat is obtained with a resistance-type equation based on the temperature difference between the moist air and corresponding wall temperature. The latent term, on the other hand, depends on the humidity ratio potential between the flowing air and the saturated air at the wall temperature. According to the previous discussion, the mass balance within any air cell i in contact with n_i wall cells ($j=1, n_i$) is given by:

$$\begin{aligned} &\dot{m}_i.dW_i = \sum_{j=1}^{n_i} -\alpha_{m,ij}.(\bar{W}_i - W_{sat,wj}).P_{ij}.ds_{ij} && \text{if } (T_{wj} < \bar{T}_{dp,i}) \\ &\dot{m}_i.dW_i = 0 && \text{if } (T_{wj} \geq \bar{T}_{dp,i}) \end{aligned} \qquad \text{where, } \alpha_{m,ij} = \frac{\alpha_{ij}}{Le^{2/3}.\bar{C}_{p,i}} \tag{1}$$

Chilton–Colburn analogy is used to relate the mass transfer coefficient (α_m) between the air and the wall surface to the heat transfer coefficient for dry case (α), based on Lewis number (Le) and moist air specific heat (C_p).

To evaluate the temperature variation within any air cell the following relation is proposed:

$$\dot{m}_i.\bar{C}_{p,i}.dT_i = \sum_{j=1}^{n_i} -\alpha_{ij}.(\bar{T}_i - T_{wj}).P_{ij}.ds_{ij} \tag{2}$$

where, $\bar{W}$, $\bar{T}$, *and* $\bar{C}_p$ are the average humidity ratio, temperature and specific heat of the moist air, respectively.

- *Refrigerant governing equations:*

The heat balance on each refrigerant cell is evaluated by Equation (3), which is similar to Equation (2) but the enthalpy difference is used instead of temperature difference.

$$\dot{m}_i.dh_i = \sum_{j=1}^{n_i} -\alpha_{ij}.(\bar{T}_i - T_{wj}).P_{ij}.ds_{ij} \tag{3}$$

- *Wall governing equations:*

Equation (4), represents a 2D energy balance within a wall cell *j* in contact with n_j fluid cells (*i=1,* n_j):

$$\nabla\left(k_{wj,k}.\nabla T_{wj}\right) + \sum_{i=1}^{n_j} \frac{1}{t_{wj}}.\dot{q}_{ij} = 0 \tag{4}$$

where, $k_{wj,k}$ is the thermal conductivity of the wall cell *j* in the *k* direction. The source/sink term (total heat flux) in Equation (4) could be expressed as following:

a) if the fluid cell (*i*) is air: $\dot{q}_{ij} = \alpha_{ij}\left(\bar{T}_i - T_{wj}\right) + \alpha_{m,ij}.h_{fg}\left(\bar{W}_i - W_{sat,wj}\right)$

b) if the fluid cell (*i*) is refrigerant: $\dot{q}_{ij} = \alpha_{ij}.\left(\bar{T}_i - T_{wj}\right)$ (5)

To linearize the source term in Equation (5a), Elmahdy and Biggs (1983) suggested a linear relation between the saturated humidity ratio and corresponding wall temperature:

$$W_{sat,wj} = a_{ij} + b_{ij}.T_{wj} \tag{6}$$

Where *b is* the average slope of the saturation line between the wall temperature and surrounding air dew point as proposed by Sharqawy and Zubair (2008):

$$b_{ij} = \frac{\bar{W}_i - W_{sat,wj}}{\bar{T}_{dp,i} - T_{wj}} \tag{7}$$

Substituting by Equations (6) and (7) into Equation (5a) for the saturated humidity ratio, rearranging and separating the term which depends on the wall temperature, we obtain:

$$\dot{q}_{ij} = \beta_{ij} - \alpha_{w,ij}.T_{wj}$$

$$\text{where, } \alpha_{w,ij} = \alpha_{ij}.\left[1 + \frac{h_{fg}}{Le^{2/3}.\bar{C}_{p,i}}.b_{ij}\right], \text{ and } \beta_{ij} = \alpha_{ij}.\left[\frac{h_{fg}}{Le^{2/3}.\bar{C}_{p,i}}\left(\bar{W}_i - a_{ij}\right) + \bar{T}_i\right] \tag{8}$$

α_w is the heat transfer coefficient for wet case which accounts for the heat and mass transfer between the moist air and the wall, in the case of no mass transfer it will be equal to the heat transfer coefficient for dry case (α).

Pressure drop is not considered since the paper only focuses on the understanding of possible differences in heat transfer. The LFTV numerical scheme, as explained in Corberan *et al.* (2001) is employed for the discretization of the heat and mass transfer in Equation (5). 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 prescribed inlet temperature and humidity ratio (in the case of air) beside the velocity distributions for both fluids, and that the open edges of the tubes to the air are considered adiabatic. The global solution method employed is called SEWTLE (for Semi Explicit method for Wall Temperature Linked Equations) and is outlined in Corberan *et al.* (2001).

3. CASE STUDY

In this case study, a minichannel evaporator has been modeled. Its dimensions are based on the evaporator tested by Zhao *et al.* (2001), where only the tube length has been modified according to the scope of the current work.

Table 1: Geometry of the minichannel evaporator

Tube Length	(cm)	8.6	Fin pitch	(mm)	1.59	Channel Diameter	(mm)	1
Tube Depth	(mm)	1.6	Fin thickness	(mm)	0.152	Channels Number	(-)	10
Tube Thickness	(mm)	0.5	Fin height	(mm)	8			

Table 2: Experimental test conditions

	Inlet Pressure (kPa)	Inlet Temperature (ºC)	Air Inlet Dew Point (ºC)	Inlet Quality (%)	G (kg/m2.s)
CO_2	3600	1.4	-	22	188.76
Air	100	26.7	16.2	-	3.34

Table (1), shows the geometric data of the evaporator under study, while Table (2) describes the experimental test conditions. Some data were estimated from the reported experimental data; namely, the heat transfer coefficients were estimated to be 8000 (W/m^2. ºC) for the CO_2 side and 48 (W/m^2. ºC) for the air side.

4. VALIDATION OF THE FIN2D MODEL

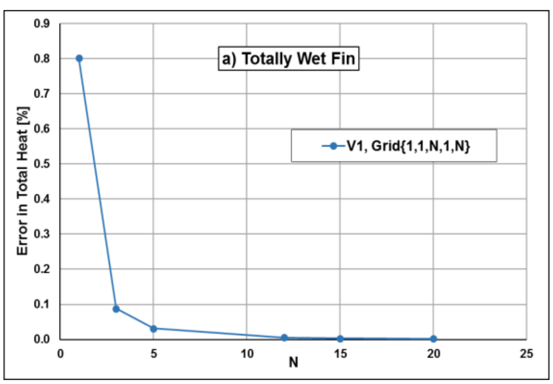
Before using the newly developed model to produce detailed solutions of heat transfer in the analyzed portion of the minichannel evaporator, it is necessary to validate it. With this purpose in mind a series of systematic checks were performed against operational cases for which an analytical solution can be obtained.

The detailed discretization of the air flow in the Y-direction adopted in Fin2D makes it difficult to compare Fin2D predictions with those of analytical solutions. The validation had to consist of two steps: air side validation (V1), and fin temperature profile validation (V2). To allow a comparison against analytical solutions the following assumptions were used; the longitudinal conduction on both fin and tube walls and the transverse conduction on the tube wall were disabled, 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 validate the calculation of heat transferred to the fins.

V1 validation is divided into two validations; one for sensible heat transfer and the other for latent heat transfer. In both validations the numerical solution was compared with analytical one based on single stream heat exchanger, $\varepsilon=1-\exp(-NTU)$, assuming constant tube wall temperature. Finally, the total heat transfer from air (summation of sensible and latent heat) was compared.

Figure (2), shows the error of the numerical solution with reference to the analytical one for V1 case under totally wet and totally dry fin condition. The figure demonstrates that the error tends to diminish very quickly with the number of cells used. The abscissa shows that the number of cells in the Z-direction (air flow direction). As it can be observed, the error is very small already for N=5.

V2 results are depicted in Figure (3) for totally wet fin as a function of the number of cells in Y-direction, considering two situations; equal tube temperatures at the bottom and the top, and a temperature difference between tubes of 5 K. θ is the difference between the fin temperature and the air temperature. The analytical solutions for both cases have been taken from Sharqawy and Zubair (2008). As seen from the figure, there is a very good agreement between the numerical and analytical temperature profiles, especially with increasing the number of fin cells in Y-direction.



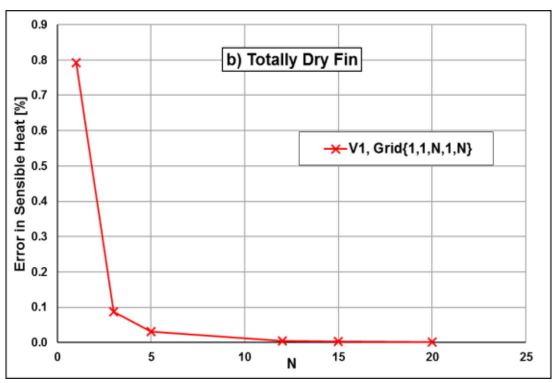


Figure 2: V1 results for; **(a)** totally wet fin condition, **(b)** totally dry fin condition

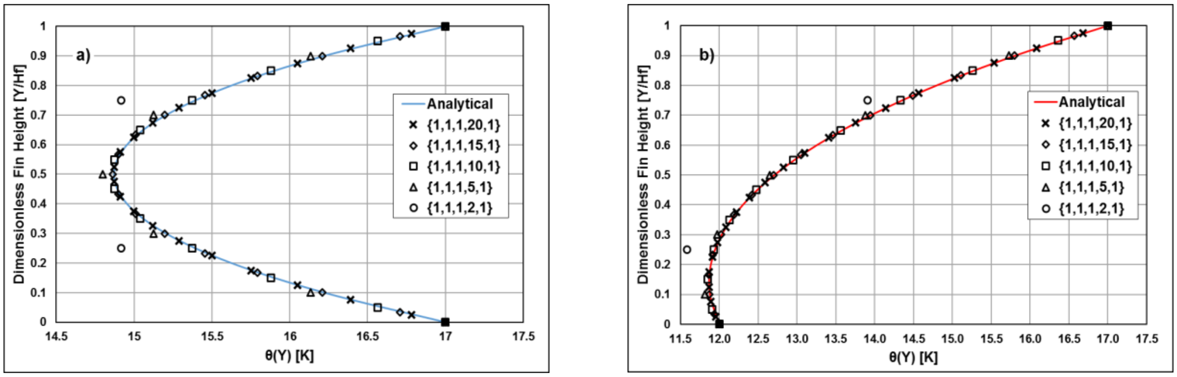


Figure 3: V2 results for totally wet fin; **(a)** $\theta(H_f) = \theta(0)$, **(b)** $\theta(H_f) = \theta(0)+5K$

5. ANALYSIS OF ε-NTU MODELLING BASED ON FIN2D MODEL RESULTS

Once the Fin2D model has been validated it can be used as the reference to check the error made by the classical segment-by-segment ε-NTU method which is widely used for modeling evaporators. The solutions to each operation scenario analyzed below were obtained with the Fin2D model using a detailed grid: {3,1,10,30,10}.

As mentioned before, most of simulation models divide each evaporator tube into segments along the refrigerant flow with its corresponding fins. Once the evaporator is divided into segments the adiabatic fin assumption and classical ε-NTU relationships for heat exchangers (Incropera and DeWitt, 1996) are employed to solve the heat and mass transfer for each segment. This method simplify the solution and the calculation time, but on the other hand, it has many drawbacks, e.g. neglecting the longitudinal conduction in the tube and fin, neglecting the transverse conduction in the tube, and assuming adiabatic fin tip. These drawbacks were extensively discussed by Martínez-Ballester *et al.* (2010). In addition to these effects, the presence of the dehumidification process shows some other drawbaks such as:

1. *Constant air temperature along the Y-direction:* besides what Martínez-Ballester *et al.* (2010) discussed about this topic, now constant temperature within the Y-direction results also in a constant humidity ratio. This assumption deviates from the reality because the actual temperature gradient in Y-direction will result in a gradient in the humidity ratio especially near the tube and fin roots.
2. *No accounting for partially wet fin:* 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 ε-NTU approach, the identification of surface area below or above the dew point both along 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 wet, based on one of the following conditions:
   - Condition (1): if, $\overline{T}_w < \overline{T}_{dp}$ then the whole segment will be assumed totally wet, otherwise it will be assumed totally dry. This condition was proposed by Jiang *et al.* (2006), where $\overline{T}_w = \eta_{f,dry}\left(T_b - \overline{T}_a\right) + \overline{T}_a$.
   - Condition (2): if, $T_b < \overline{T}_{dp}$ then the whole segment will be assumed totally wet, otherwise it will be assumed totally dry.

In the current study, the tube (top and bottom) temperature was changed from 1.4 ºC to 17 ºC, while the experimental test conditions for the case study in Table (2) were used to define the air status at the inlet, those conditions allowed us to capture different scenarios for the tube and fin. The resulted cases from the model are discussed in details and compared with classical ε-NTU approaches as following.

• Case (I): Totally Wet Tube and Fin

In this case the temperature of the tube was set to 1.4 ºC, as seen in Figure (4a) this results to a fin temperature profile below the average dew point of the air at any point. Under that condition the mass transfer (due to humidity ratio difference) occurs simultaneously with heat transfer (due to temperature difference) and the whole tube and fin surfaces become totally wet as depicted in Figure (4b).

The deviation in the ε-NTU results based on current model results has been analyzed, taking into account the two common conditions which are used generally to identify the segment status by ε-NTU approach (condition 1 and 2). The results in Table (3) show almost a similar deviation in the latent and sensible heat transfer. The assumption of a constant air temperature and humidity ratio along Y-direction (drawback no. 1), which is adopted usually by ε-NTU method and fin theory, could be the main source of this difference in results.

Table 3: Deviation in the heat transfer based on numerical results, case (I)

ε-NTU	Segment Condition	Sensible Heat Analysis		Latent heat Analysis		Total Heat Analysis	
		$Q_{sens,num}$ (W)	ΔQ_{sens} (%)	$Q_{lat,num}$ (W)	ΔQ_{lat} (%)	$Q_{tot,num}$ (W)	ΔQ_{tot} (%)
Condition (1)	Totally Wet	15.86	3.44	11.31	3.62	27.17	3.52
Condition (2)							

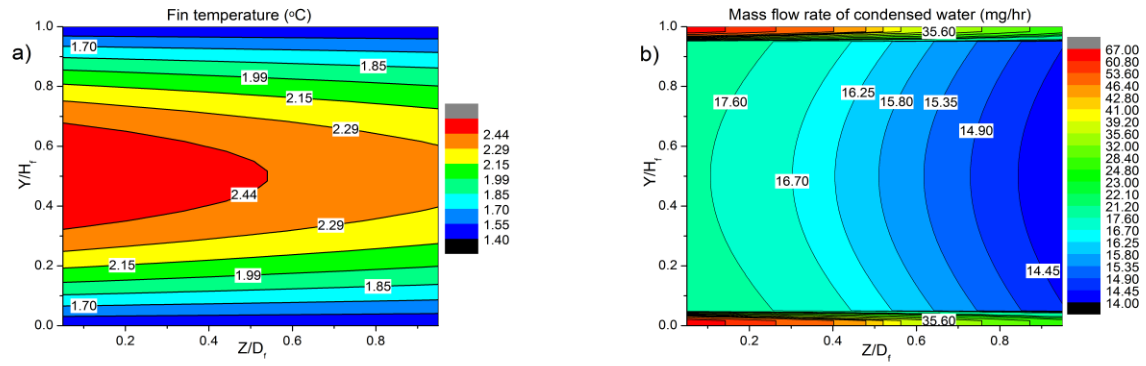


Figure 4: (a) fin temperature profile, **(b)** mass flow rate of condensed water for case (I)

• Case (II): Totally Wet Tube and Partially Wet Fin

In this situation the temperature of the tube was kept at 16.1 °C, which is almost close to the dew point of the inlet air. This scenario leads to a totally wet tube, however, the numerical results have shown that there are some areas on the fin which have a temperature bigger than the average dew point of the corresponding air cells, so sensible heat only is transferred between those areas and surrounding air resulting in a partially wet fin. Figure (5) shows the fin temperature profile and the amount of water condensed from the air in (mg/hr), it can be noticed that nearly less than 20% of the fin surface is wet and the remaining is totally dry.

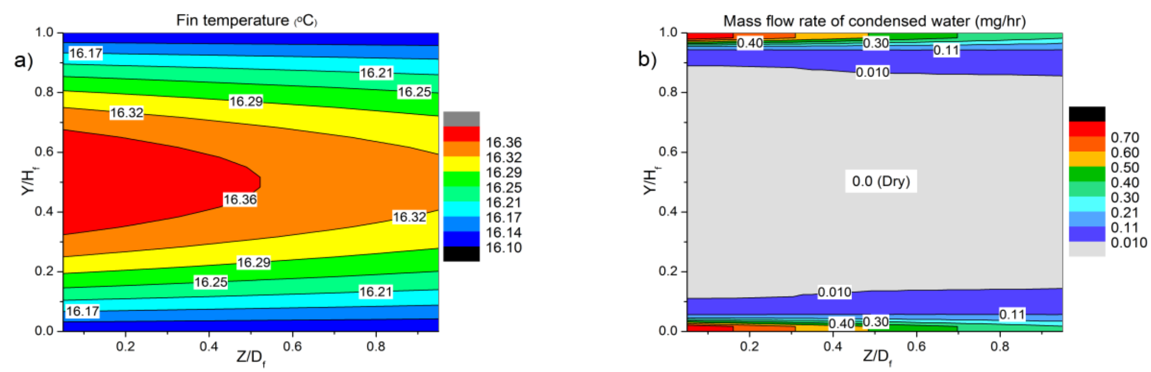


Figure 5: (a) fin temperature profile, **(b)** mass flow rate of condensed water for case (II)

As seen in Table (4), although ε-NTU methods give two completely different predictions for the segment condition, but finally the deviation in the total heat in both methods is very close. The domination of sensible heat and the small amount of condensation in this case can lead to this result. It can be noticed that condition (2) over predict the latent heat transfer comparing with numerical results because of assuming a totally wet fin (drawback no. 2). However, we believe that the deviation in results could be bigger and more significant, especially in the latent heat transfer, if a larger portion of the fin surface is wet and a more amount of water is condensed.

Table 4: Deviation in the heat transfer based on numerical results, case (II)

ε-NTU	Segment Condition	Sensible Heat Analysis		Latent heat Analysis		Total Heat Analysis	
		$Q_{sens,num}$ (W)	ΔQ_{sens} (%)	$Q_{lat,num}$ (W)	ΔQ_{lat} (%)	$Q_{tot,num}$ (W)	ΔQ_{tot} (%)
Condition (1)	Totally Dry	6.71	3.97	0.03	-	6.74	3.50
Condition (2)	Totally Wet		1.74		369		3.40

- **Case (III): Totally Wet Tube and Totally Dry Fin**

The tube temperature was set to 16.18 ^{o}C in the current case, which is very close to the dew point of the inlet air. This case gives similar results to case (II), but it can be noticed from Table (5) that in condition (1) the deviation in total heat increased to 3.86%. On the other hand, even though the deviation in the latent heat increased significantly in condition (2) up to 617%, but the latent heat transfer is very low comparing with the sensible heat transfer which has the biggest contribution in the total heat.

Table 5: Deviation in the heat transfer based on numerical results, case (III)

ε-NTU	Segment Condition	Sensible Heat Analysis		Latent heat Analysis		Total Heat Analysis	
		$Q_{sens,num}$ (W)	ΔQ_{sens} (%)	$Q_{lat,num}$ (W)	ΔQ_{lat} (%)	$Q_{tot,num}$ (W)	ΔQ_{tot} (%)
Condition (1)	Totally Dry	6.66	3.97	0.007	-	6.667	3.86
Condition (2)	Totally Wet		2.12		617		2.73

- **Case (IV): Totally Dry Tube and Fin**

17 ^{o}C was selected for the tube temperature, which is definitely bigger than the dew point temperature of the inlet air. Under this condition only sensible heat transfer occurs, and there is no mass transfer between the air and the wall surface which results in totally dry fin and tube. As demonstrated in Table (6), the ε-NTU method under both two conditions predicts well the segment status. However, the deviation in results is due to the assumption of constant air temperature between tubes, but actually the temperature of air close to the tube wall and fin roots is very different than the bulk air temperature. This fact has an important impact on local effects controlling the heat transfer, and contributes significantly in the deviation between ε-NTU method and current model results.

Table 6: Deviation in the heat transfer based on numerical results, case (IV)

ε-NTU	Segment Condition	Sensible Heat Analysis		Latent heat Analysis		Total Heat Analysis	
		$Q_{sens,num}$ (W)	ΔQ_{sens} (%)	$Q_{lat,num}$ (W)	ΔQ_{lat} (%)	$Q_{tot,num}$ (W)	ΔQ_{tot} (%)
Condition (1)	Totally Dry	6.14	3.97	-	-	6.14	3.97
Condition (2)							

6. CONCLUSIONS

A model for minichannel evaporator, Fin2D, accounting for two-dimensional variation of the tube and fin temperature, and moist air properties was presented. After validation against known analytical solution, the model was employed to quantify the deviation in the heat transfer between the traditional ε-NTU approach and its results. The following are the main conclusions of the study:

- For totally wet fin (case I), the deviations in the latent and sensible heat between ε-NTU method, under both two conditions (condition 1 and 2), and model are very similar. The deviation in the total heat is about 3.52% and mainly due to the assumption of constant air temperature and humidity ratio along the direction between tubes which is usually adopted in the ε-NTU approach and fin theory.

- For partially wet fin (case II), even the ε-NTU methods fail to predict the fin status; condition (1) assumes totally dry fin, while condition (2) assumes totally wet fin and over predict the latent heat by about 369%. However surprisingly, calculating locally (in the model) or globally (in ε-NTU approach) the mass transfer doesn't report big differences in results because of the small weight of the latent heat in this scenario.

- In general, the contribution of latent heat in the total heat deviation is less or at most equals to the sensible heat contribution. That indicates that the main responsible for this deviation between the two approaches is the assumption of no temperature variation of the air along Y-direction which results also to a constant humidity ratio within the same direction. In reality, this assumption is not true because the temperature and humidity ratio of the air close to the tube wall and fin roots are very different than the bulk air temperature and humidity ratio. Anyhow, we expect more deviation in results and more contribution of the latent heat for higher inlet relative humidity.

More studies will be held in the future, especially for different inlet moist air conditions, and also different fin roots temperatures.

The authors will follow working on a simplified model that will retain the most important effects. This will lead to much lower computation times while providing high accuracy of prediction of the complex heat and mass transfer phenomena taking place in air-to-refrigerant minichannel evaporators.

NOMENCLATURE

a	parameter defined in Eq. (6)	(kg_w/kg_a)
b	parameter defined in Eq. (7)	(1/ °C)
C_p	specific heat	(J/kg. °C)
D_f	fin depth	(m)
G	mass flux	($kg/m^2.s$)
H_f	fin height	(m)
h	specific enthalpy	(J/kg)
h_{fg}	latent heat of water condensation	(J/kg)
k	conductivity	(W/m. °C)
Le	Lewis number	(-)
$\dot{m}$	mass flow rate	(kg/s)
NTU	number of heat transfer units	(-)
P	perimeter	(m)
Q	heat transfer	(W)
$\dot{q}$	heat flux	(W/m^2)
s	length in the forward direction of a fluid	(m)
T	temperature	(°C)
t	thickness	(m)
W	humidity ratio	(kg_w/kg_a)
X,Y,Z	spatial coordinates	(m)
α	heat transfer coefficient for dry case	(W/m^2. °C)
α_m	mass transfer coefficient	($kg/m^2.s$)
α_w	heat transfer coefficient for wet case	(W/m^2. °C)
β	parameter defined in Eq.(8)	(W/m^2)
ε	thermal effectiveness	(-)
η	thermal efficiency	(-)
θ	temperature difference=T_a-T_w	(°C) or (K)

Subscripts

a	air
b	fin base
dry	dry surface condition
dp	dew point
f	fin
i	fluid cell index
j	wall cell index
k	direction index
lat	latent
num	numerical
sat	saturated
sens	sensible
tot	total
w	wall

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