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Research Publications

ANALYSIS OF SEGMENT-BY-SEGMENT ε-NTU MODELLING OF A MINICHANNEL CO_2 GAS COOLER

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ABSTRACT

Most of the current air-to-refrigerant heat exchanger models use the classic ε-NTU approach. These models do not account for longitudinal conduction neither in the fin nor in the tube, transverse conduction in the tube, and for the heat conduction between different tubes, which is a consequence of the employed adiabatic fin tip assumption. This paper presents a more detailed numerical approach to heat exchanger modeling with the goal to capture heat conduction effects within the heat exchanger structure and detailed representation of air properties. The new model uses a segment-by-segment approach and applies a 2-D discretization for each segment. The paper includes a presentation of the numerical scheme, validation, and a parametric study which tests the impact of the traditional heat exchanger model assumptions. The study revealed large errors in capacity prediction of individual tubes due to the adiabatic fin tip assumption, when the neighboring tubes are of different temperature.

1. INTRODUCTION

The use of minichannels heat exchangers is increasing because of their compactness and high effectiveness. In the case of transcritical CO_2 systems, minichannels have an additional merit related to their high mechanical strength. As with other products, reliable simulation models can provide substantial cost savings during the design and optimization process of heat exchangers. Currently, several heat exchanger models are available in the literature; most of them use the ε-NTU approach to solve the thermal problem. The ε-NTU methodology uses the adiabatic fin tip assumption, which fundamentally does not lend itself to accounting for heat transfer via fins between tubes of different temperatures. While the ε-NTU modeling approach can yield accurate predictions when the heat exchanger modeling assumptions are not significantly violated during the heat exchanger operation, it tends to overpredict heat exchanger capacity when significant temperature differences between tubes exist.

Several experimental studies indicated that the heat exchanger performance can be significantly degraded by the tube-to-tube heat transfer via connecting fins. For example, Domanski et al. (2007) measured as much as 23 % reduction in finned-tube evaporator capacity when different exit superheats were imposed on the individual refrigerant circuits. Park and Hrnjak (2007) reported a 3.9 % capacity improvement in a minichannel CO_2 gas cooler after introducing fin cuts between selected tubes. Also Zilio et al. (2007) concluded that heat conduction trough fins in a CO_2 gas cooler had a significant impact on the capacity. In fact, cut fin surfaces are increasingly being used in heat exchangers to reduce the heat conduction between tubes and improve the heat exchanger performance.

Several authors use different approaches to introduce heat conduction effects in their models. Asinari et al. (2004) proposed a minichannel model which takes into account heat conduction along all directions for all elements (fins and tubes). They investigated the impact of conduction effects on capacity, and also studied the prediction error due to the adiabatic fin tip assumption used in ε-NTU models. The authors concluded that when tube temperatures are different, the use of the adiabatic fin tip efficiency gives accurate predictions of the total heat capacity although it does not accurately represent the actual distribution of heat flow between fin roots. Regarding the conduction effects, they concluded that the impact of the individual heat conduction effects for each direction and element have negligible effects on the capacity, but the combined effect of all of them was not evaluated. Singh et al. (2008) presented a model, referred to as a "resistance model", to account for heat transfer between tubes through the fins in finned-tube heat exchangers using a segment-by-segment approach. Instead of using the ε-NTU approach, they applied energy equations to each segment and included a term for heat conduction through fins between neighboring tubes while still using the concept of adiabatic fin tip efficiency. The authors explained that using a set of energy conservation equations is better than using the ε-NTU methodology with the included heat conduction term because the ε-NTU relationship assumes all heat being transferred from one fluid to another without internal heat transfer within the heat exchanger wall structure itself. Their validation effort showed improved model predictions when heat conduction effects were included.

Not using the ε-NTU methodology has the disadvantage of losing an accurate fluid temperature function, which requires assuming some temperature profile for the fluids. This problem can be solved by dividing the heat exchanger into smaller segments, which improves the representation of non-uniform air and refrigerant properties. In most published models, this methodology improves only the representation of the refrigerant properties because no discretization is provided in the air flow direction. This leads to approximated air properties for the whole heat exchanger depth (air flow path) based on the average of the inlet and outlet temperatures.

Finally, it should be mentioned that the fin theory, applied in most models using fin efficiency, assumes uniform air temperature along the fin length, and this assumption is violated since there is a temperature variation along the tube pitch, as can be expected for the air close to the tube walls.

This paper presents a detailed model for minichannel heat exchangers used as gas coolers that accounts for two-dimensional (2D) heat conduction in any element (fin or tube). The model, referred to as Fin2D model, subdivides the heat exchanger into segments, to which the corresponding system of energy-conservation equations is applied without traditional heat exchanger modeling assumptions. After validation, the solution obtained with the Fin2D model is employed to assess the impact of the classical heat exchanger modeling assumptions on the accuracy of the performance predictions.

2. FIN2D HEAT EXCHANGER MODEL

2.1 Heat Exchanger discretization

Figure 1(a) presents a piece of the studied minichannel heat exchanger. It is discretized along the *X* direction (refrigerant flow) in a number of segments ‘*a*’. Each segment (figure 1(b)) 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: *{3,5,3,7,4}*.

All grid dimensions are independent, with the only exception that the air and fin have the same discretization. The refrigerant flows inside each channel (*b*=5 in the figure 1) along the *X* direction without any mixing between the channels. The heat is transferred from the refrigerant to the tube wall in contact, as well as from the air to the fin wall, and in the bottom and top cells to the outer surface of the tube wall. The fins then conduct the heat along *Y-Z* directions, and the bottom and top cells into the tube wall. The tube wall conducts the heat along *X-Z* directions and to the fin roots in contact.

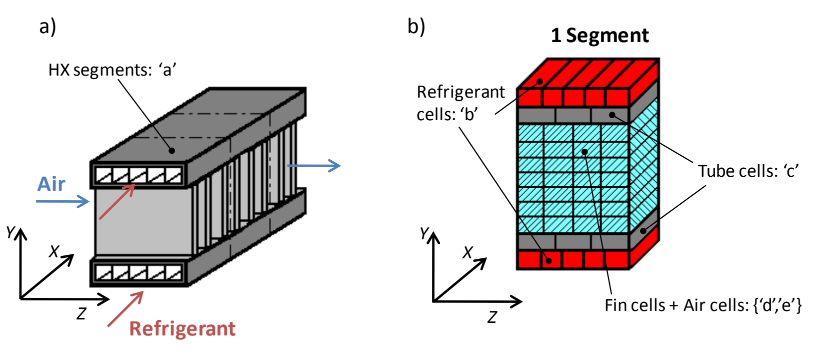

Figure 1. (a) Piece of the heat exchanger studied in the paper. (b) Schematic of the discretization in a segment of the heat exchanger.

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 at each fluid cell (refrigerant and air) and at each wall cell (tube and fin) can be written as follows:

$$\dot{m}_i \cdot dh_i = \sum_{j=1}^{n_i} \dot{q}_{ji} \cdot P_{ji} \cdot ds_i \tag{1}$$

$$\dot{q}_{ji} = U_{ji} \cdot (T_{wj} - T_i) \tag{2}$$

$$U_{ji} = \frac{1/A_{ji}}{\dfrac{t_j/2}{A_{ji} \cdot k_{j,k}} + \dfrac{1}{A_{ji} \cdot \alpha_{ji}}}$$

$$\nabla\left(k_{j,k} \cdot t_j \cdot \nabla T_{wj}\right) + \sum_{i=1}^{n_j} \dot{q}_{ji} = 0 \tag{3}$$

where any wall cell j is in contact with n_j fluid cells i=1, n_j; any fluid cell i is in contact with n_i wall cells j=1, n_i; $k_{j,k}$ is the thermal conductivity of the wall cell j in the k direction, thus it is possible to study the influence of longitudinal and transverse conduction at both fin and tube walls. Equation (1) states the energy conservation for a fluid cell, whereas equation (3) states the energy conservation equation for a wall cell. Equation (2) represents the heat flow in between a wall cell and a fluid cell. Pressure drop is not considered since the paper only focuses on the understanding of possible differences in heat transfer.

The discretization of the governing equations does not present any special difficulty, except for the estimation of the integral of the heat transferred to the fluids in contact with the considered piece of wall (equations (2) and (3)). This integration must be consistent with the integration of the coincident terms of the fluid energy equation (1). The LFTV numerical scheme, as explained in Corberan et al. (2001), is employed for the discretization of equation (2). This numerical scheme is basically based on assuming a piecewise distribution of the fluid temperature along the fluid cell, leading to the following expression:

$$A_{ji}\ \dot{q}_{ji} = U_{ji} P_{ji} \left(T_{wj} - \frac{T_i^{in} + T_i^{out}}{2} \right) \Delta s_i \tag{4}$$

The discretization of the Laplacian operator in equation (3) can be made by a classical finite difference (finite volume) approach. The corresponding boundary conditions are prescribed inlet temperature and 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). Basically, this method is based on an iterative solution procedure. First, a guess is made about the wall temperature distribution, and then the governing equations for the fluid flows are solved in an explicit manner, getting the outlet conditions at any fluid cell from the values at the inlet of the heat exchanger and the assumed values of the wall temperature field. Once the solution of the fluid properties are obtained for any fluid cell, then the wall temperature at every wall cell is estimated from the balance of the heat transferred across it (Eq.(3)). This procedure is repeated until convergence is reached. The numerical method employed for the calculation of the temperature at every wall cell is based on the line-by-line strategy (Patankar, 1980) following the *Y* direction for fin cells and the *X* direction for tube cells, so that the global strategy consists of an iterative series of explicit calculation steps. This method can be applied to any flow arrangement and geometrical configuration and offers excellent computational speed. Additionally, it can easily be extended to other cases, such as two-phase flow or humid air.

3. CASE STUDY

In this case study we modeled a minichannel gas cooler for which dimensions were extracted from Zhao et al. (2001). The operating conditions were correspond to the experimental data of the test for gas cooling n° 3b, HX1,

Table 1. Geometry of the minichannel heat exchanger

Tube Length (cm)	8	Fin pitch (mm)	1.56	Channel Diameter (mm)	1
Tube Depth (mm)	16	Fin thickness (mm)	0.152	Channels Number	10
Tube Thickness (mm)	1	Fin height (mm)	8		

Table 2. Operating conditions; Test for gas cooling n° 3b, HX1 (Zhao et al., 2001). *estimated value.

	Inlet pressure (kPa)	Inlet Temperature (°C)	Outlet Temperature (°C)	G (kg/m^2 s)
CO_2	8937	79.9	42.4*	132.56
Air	100	23.74*	32.4	3.05

from the same work. Table 1 shows the most important geometric data while Table 2 shows the considered operation conditions. Some data were estimated from the reported experimental data; namely, the heat transfer coefficients were estimated to be 537 (W/m^2 K) for the CO_2 side and 66 (W/m^2 K) for the air side.

4. VALIDATION OF THE FIN2D MODEL

Before employing the newly developed model to produce detailed solutions of heat transfer in the analyzed portion of the minichannel gas cooler, it is necessary to validate it. With this purpose in mind we performed a series of systematic checks 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 three steps: air side validation (V1), refrigerant side validation (V2), and fin temperature profile validation (V3). To allow a comparison against analytical solutions, we disabled the longitudinal conduction on both fin and tube walls and the transverse conduction on the tube wall, and used constant properties and heat transfer coefficients. 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 and V2 validations consisted of comparing the computed solution with the analytical solution for a single stream heat exchanger, $\varepsilon = 1 - \exp(-NTU)$, imposing infinite flow-stream capacity rate ($\dot{m} \cdot Cp$) for the other stream. Figure 3(a) shows the error of the numerical solution with reference to the analytical solution for V1 and V2 cases. The figure shows that the error tends to diminish very quickly with the number of cells used. In the case of V1, the abscissa shows the number of cells in the *Z* direction. As it can be observed, the error is very small already for *N*=5. In the case of V2, where the air has infinite flow-stream capacity rate, the abscissa was taken as the number of cells along the *X* direction. Again the analytical solution is almost reached with only five cells.

Figure 3(b) shows the error of the numerical solution for the heat transferred from the film to the fin wall as a function of the number of cells in the *Y* direction for two situations: equal tube temperatures at the bottom and the top, and a temperature difference between tubes of 15 K. θ is the difference between the fin temperature and the air temperature. The analytical solutions for both cases have been taken from (Incropera and DeWitt, 1996).

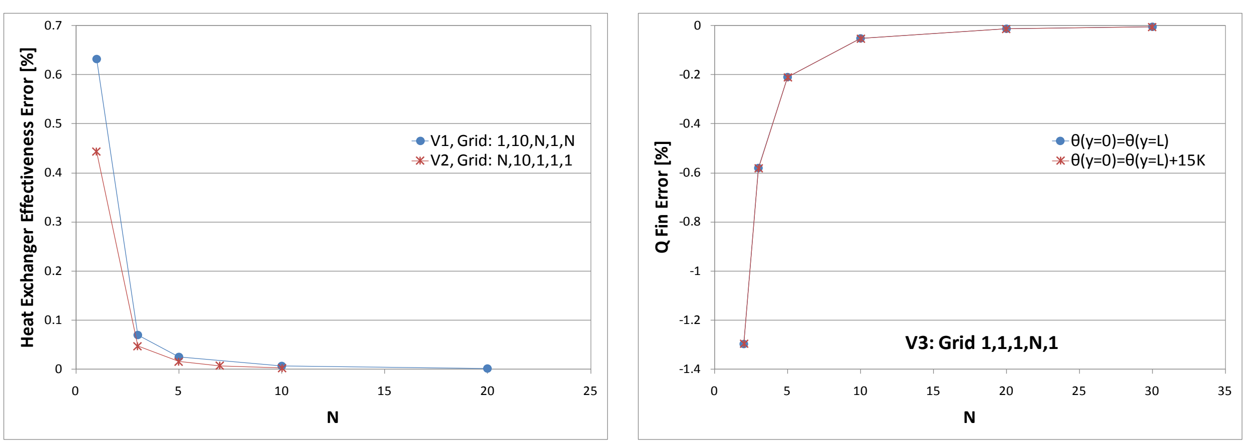

Figure 3. (a) V1 and V2 results. (b) V3 results in two cases: tube with the same temperature and with a difference of 15K.

As can be observed, the error is small, -0.2%, with only five cells in the *Y* direction, and quickly tends to as the solution. The calculated fin temperature profile was also compared with the analytical solutions proving the accuracy of the numerical model.

5. ANALYSIS OF THE SEGMENT-BY-SEGMENT ε-NTU MODELLING

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 modeling a gas cooler application. The solutions to each operation scenario analyzed below were obtained with the Fin2D model using a detailed grid: *{3,10,10,30,10}*.
The air side heat transfer coefficient was estimated by correlations for plain fin (Webb and Kim, 2005), adopting Petukhov's correlation for turbulent flow (Petukhov, 1970). For the refrigerant side, constant properties and heat transfer coefficients were used, as listed in Section 3.

The classical ε-NTU modeling approach divides each heat exchanger tube into segments along the refrigerant flow with its corresponding fins. Some modelers use only one segment per tube, which is commonly referred to the tube-by-tube approach. When the tube is discretized in more than one segment the approach is defined as segment-by-segment. Once the heat exchanger is divided into segments, the ε-NTU relationships for heat exchangers (Incropera and DeWitt, 1996) are employed for each segment. For cross-flow heat exchangers the air is always considered to be unmixed because the fins prevent the mixing, but there are two options for the refrigerant: to assume refrigerant as mixed (RMAU) or as unmixed (BU). Generally, RMAU is assumed for the segment-by-segment approach, e.g., Jiang et al. (2006), while BU is considered for the tube-by-tube approach.

The ε-NTU models used in this analysis were developed within a commercial equation solver (Klein, 1995). Both options available within the ε-NTU modeling methodology were included in this study: BU and RMAU. The ε-NTU models used the same properties and heat transfer correlations as those used in the Fin2D model.

The classical ε-NTU modeling presents the following drawbacks:

- Longitudinal and transverse conduction: As it was explained in the introduction, the ε-NTU method does not account for longitudinal conduction in the fin (along the *Z* direction), longitudinal conduction in the tube (along the *X* direction) and transverse conduction in the tube (along the *Z* direction).
- Adiabatic fin tip efficiency: This assumption is widely used even when a temperature difference between tubes exists.
- BU discretization inconsistency: discretizing along the *X* direction (i.e. number of segments) involves an implicit mixing of the refrigerant stream since the inlet temperature at one segment is evaluated as the averaged value at the outlet section of the preceding segment. Consequently, for the BU ε-NTU case, increasing the number of segments is inconsistent with the hypothesis of unmixed refrigerant stream. Therefore, if the unmixed condition for the refrigerant is the one which better represents the actual process, the best option for the discretization along the *X* direction would be to employ a tube-by-tube approach. This will lead to a full consistent BU solution at each tube with mixing at the outlet. This mixing would be perfectly consistent with the real operation in those minichannel heat exchangers where the tubes end in the collector/distributor head. For serpentine heat exchangers the solution would not be fully consistent. On the other hand, employing a tube-by-tube approach has the disadvantage that it is not possible to have a more discretized representation of the phenomena occurring inside one tube and therefore if any strong local variation of the flow or heat transfer happens in a tube, there is no way to account it.
- Air temperature variation along the *Y* direction: the ε-NTU approach assumes that the air temperature is constant along the *Y* direction. Furthermore, the fin theory is developed under the same assumption. This assumption deviates from the reality because the temperature of the air flowing close to the tube and the fin roots becomes much closer to the wall temperature.

Regarding the air-side heat transfer coefficient, two situations were considered: the reference value α_{air} for conditions cited in Table 2, and a value three times larger. This choice was made to cover large variations of possible fin surfaces including enhanced fin surfaces with a high heat transfer coefficient.

Figure 4 quantifies the errors obtained using the classical ε-NTU approach for the conditions explained above. For the RMAU case, the trend of the ε-NTU model is asymptotic to the solution with a final error of 2.5% for the α_{air} case and increasing to 3.5 % for the air-side heat transfer coefficient value increased threefold (about 180 W/m^2 K).

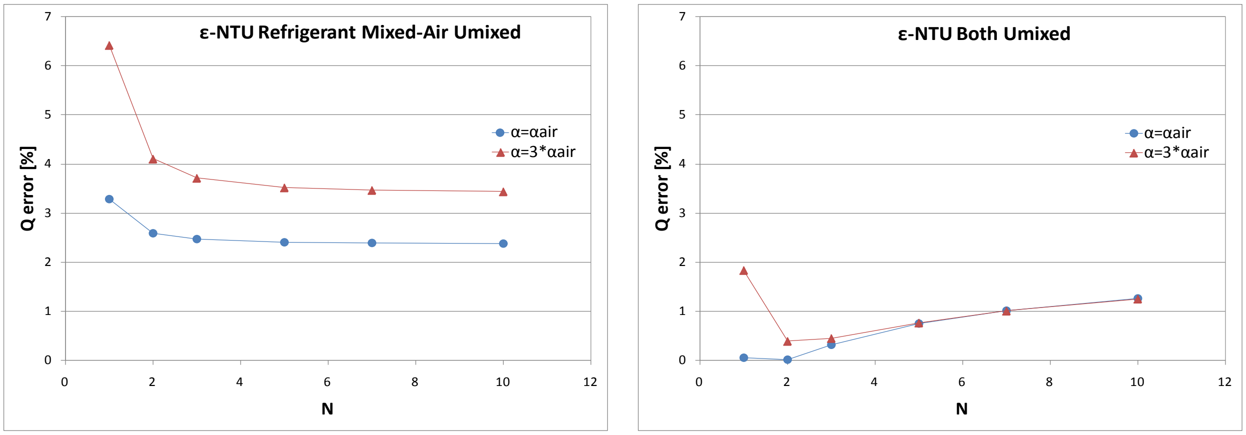

Figure 4. Comparison of Fin2D model and ε-NTU model for different number of refrigerant segments in the *X* direction (N) used in the ε-NTU model: (a) using RMAU relationships, (b) using BU relationships.

For the BU case, the errors are smaller, below 1.5 %, indicating that this approach is much closer to the solution and that the effect of the transverse temperature gradient of the refrigerant is important. However, as it can be observed in the figure 4(b), the error increases with the number of cells. This problem is a modeling inconsistency that was pointed out and explained above. Following that explanation, to be consistent with the assumption of unmixed refrigerant made for the BU case, the best way to avoid this problem is to use only one cell per tube. But on the other hand, to capture the unmixed air effect, which was also assumed, it is necessary to use more than one cell. For this dilemma Figure 4(b) indicates that *N*=2 gives the most accurate solution.

Table 3 shows the error in the capacity predictions associated with eliminating from consideration some of the heat conduction phenomena with respect to the complete solution. The cases studied are: No longitudinal conduction in fins (No LC in fins), No longitudinal conduction in tubes (No LC in tubes), No transverse conduction in tubes (No TC in tubes), and All transverse-longitudinal conduction effects disabled except conduction along the *Y* direction on the fin (No TC, No LC). The effect of heat conduction depends strongly on the air heat transfer coefficient. When the air heat transfer coefficient is equal to the reference, α_{air}, the influence of heat conduction is negligible. But when the air heat transfer coefficient has the higher value (about 180 W/m^2 K) the combined effect is noticeable, 2.54%. This increase in the prediction error due to neglecting the heat conduction effects when the air side heat transfer coefficient is increased is consistent with the increase in the prediction error shown in Figure 4 for the ε-NTU models. When the heat conduction has the largest influence, the dominant component is the transverse heat conduction in the tube. This observation also means that considering mixed refrigerant is not a good assumption since the transverse heat conduction in the tube (the dominant effect) involves a temperature gradient along the *Z* direction, affecting the refrigerant temperature profile in this direction. It is important to notice that the heat conduction effects are strongly non linear.

To study the effect of assuming the adiabatic tip at half length of the fin, as it is usually accepted, a case with a temperature difference between refrigerant inlets was simulated (for the case with the same inlet temperature the adiabatic fin tip assumption is exact). In this case, the air heat transfer coefficient was α_{air}, and the tube at *Y*=0 (lower tube) had a refrigerant inlet temperature 40 K lower than the upper tube. Figure 5(a) shows the wall temperature profile along the *Y* direction at the refrigerant inlet section (*X*=0) at three different locations along the *Z* direction. It can be observed how different the actual temperature profile is from the assumed profile when the adiabatic fin tip efficiency is used. The slope of these curves on the *Y* direction gives the local heat transfer along the

Table 3. Influence of longitudinal and transverse conduction in the capacity.

	Q error No LC, No TC [%]	Q error No LC in Fins [%]	Q error No LC in Tubes [%]	Q error No TC in Tubes [%]
$\alpha = \alpha_{air}$	0.66	0.03	0.12	0.09
$\alpha = 3 \cdot \alpha_{air}$	2.54	0.24	0.10	0.55

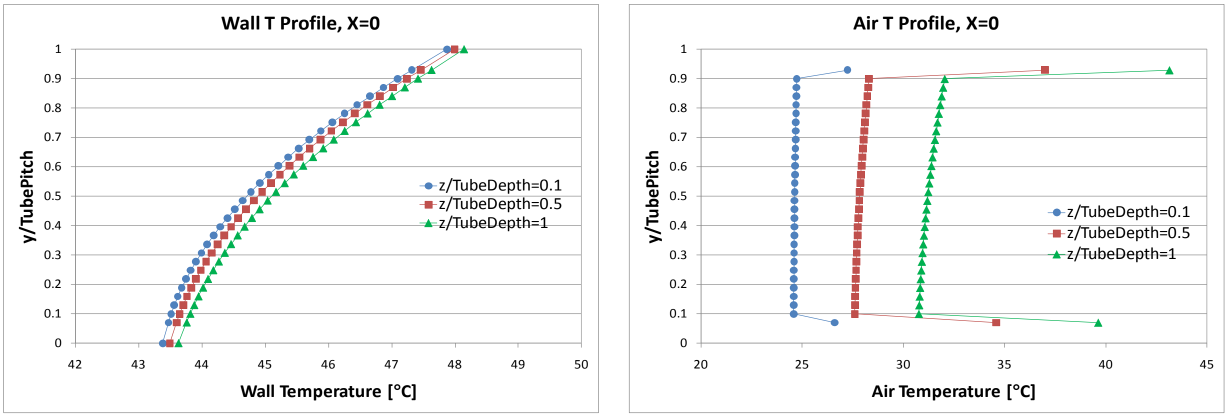

Figure 5. Temperature profiles at refrigerant inlet (*X*=0) in three positions along heat exchanger depth (*Z*) using α_{air}. (a) Wall (tube and fin) temperature profile, (b) Air temperature profile.

fin and from the fin to the tubes. Consequently, if the slope of the curves is analyzed is noticeable how far the assumption of half-fin-length idealization is from the actuality. In the figure 5(a) can be observed that the solution temperature slope does not change its sign in any section along the fin height, resulting in a wrong heat flux sign calculation (not only the absolute value) when adiabatic fin tip is used. The consequence of these differences is a large error in the heat capacity predicted for each tube and therefore in the refrigerant outlet properties. These errors were quantified for the BU case with two cells, for which Figure 4(b) shows the smallest error. For the case with 40 K temperature difference, the resulting errors were 40.6 % for the upper tube and -449.2 % for the lower tube, where a minus sign indicates that the result differs also in the direction on the heat flux.

Finally, to study the assumption of constant air temperature along the *Y* direction, Figure 5(b) presents the corresponding air temperature profile in the same locations where the wall temperature profile was analyzed. There is a large temperature variation in the air close to the tube wall, which is larger when the air has crossed more length. This difference of temperatures between bulk air and the air close to the tube wall could have an important influence in scenarios with the presence of dehumidification. The rest of the temperature profile is quite flat excepting at the air outlet.

5. CONCLUSIONS

A model for minichannel heat exchangers, Fin2D, accounting for heat conduction in all directions and in all heat exchanger elements was presented. The model allows for independent discretization for the refrigerant, tube and fins (air has the same discretization as the fins). After validation against known analytical solutions, the model was employed to quantify the prediction errors associated with the classical ε-NTU modeling approach. The following are the main conclusions of the study:

- The error obtained using the ε-NTU method depends on the ε-NTU relationship employed to calculate the heat exchanger effectiveness; it is smaller than 3.5% for BU and smaller than 1% for RMAU. In general, the best option for the studied case is to use a tube-by-tube approach and to consider both fluids as unmixed since the effect of the mixed refrigerant assumption turned out to be not negligible. However, this option can lead to larger errors when long length tubes are simulated because refrigerant properties and heat transfer coefficients can have significant variations, particularly when the refrigerant when the refrigerant undergoes a phase change.
- For the studied case, the error produced by the classical ε-NTU approach with both streams unmixed is smaller than 1% and, in general, becomes larger as the air side heat transfer coefficient increases.
- For the operating conditions studied, the impact of individual heat conduction effects in fins and tube walls, if considered separately, are not significant. The combined effect is more noticeable, which has an impact to be up to 2.5%, with the transverse heat conduction along the tube being the dominant effect. The impact of heat conduction depends on the air temperature variation thus on the heat transfer coefficient.
- Using the adiabatic fin tip efficiency, which is always the case in classical ε-NTU models, leads to large errors in heat distribution per tube when a temperature difference between tubes exists.

- The temperature of air close to the tube wall is very different than the bulk air temperature. This fact could have an important impact on local effects controlling the heat and mass transfer, e.g. dehumidification.
- The developed model is able to capture most of the secondary heat conduction effects not taken into account by the classical ε-NTU approach; however, simulation of the wall heat conduction problem requires a considerable computation time. 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 transfer phenomena taking place in air-to-refrigerant minichannel heat exchangers.

NOMENCLATURE

A	heat transfer area (m^2)	U	overall heat transfer coefficient (W/m^2 K)
G	mass flux (kg/m^2 s)	x,y,z	spatial coordinates (m)
c_p	specific heat (J/kg K)	α	convective heat transfer coefficient (W/m^2 K)
h	specific enthalpy (J/kg)	ε	heat exchanger effectiveness
k	conductivity (W/m K)	*Subscripts*	
$\dot{m}$	mass flow rate (kg/s)	i	fluid cell index
NTU	number of transfer units	in	inlet
P	wetted perimeter (m)	j	wall cell index
$\dot{q}$	heat flux (W/m^2)	k	direction index
s	length in the forward direction of a fluid	out	outlet
T	temperature (K)	w	wall
t	thickness (m)		

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