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NUMERICAL MODEL FOR A MICROCHANNEL GAS COOLER WITH ANY REFRIGERANT CIRCUITRY

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ABSTRACT

The present work presents a gas cooler model (Fin1Dx3) working with CO_2 in transcritical conditions for air-to-refrigerant microchannel heat exchangers with any refrigerant circuitry. The model applies a segment-by-segment discretization to the heat exchanger adding, in each segment, a specific bi-dimensional grid to the fluids flow, fin and tube wall. This methodology allows accounting for: 2D longitudinal heat conduction in the tube wall, the heat conduction between tubes along the fin, and the unmixed air influence on the heat capacity. The paper presents a short description of the heat exchanger discretization and the governing equations employed. The model has been validated against experimental data resulting in predicted capacity errors within ±2%. Differences on prediction results and computational cost were studied and compared with the previous authors's model, the Fin2D model. Simulation time of the proposed model was reduced one order of magnitude respect the Fin2D's time.

1. INTRODUCTION

Nowadays, simulation software is a very suitable tool for the design of products in which complex physical processes occur. These tools allow the saving of lots of costs and time in the laboratory working with expensive test benches. Currently, several models or simulation tools for heat exchanger are available in the literature: for finned tubes (Lee and Domanski 1997; Corberán et al. 2002; EVAP-COND 2003; Jiang et al. 2006; Singh et al. 2008) and microchannel heat exchangers (Yin et al. 2001; Shao et al. 2009; Fronk and Garimella 2010; García-Cascales et al. 2010). Some of them apply equations for conservation of energy to each control volume, while the rest of them apply directly the solution given by the ε-NTU methodology. Anyway, all models usually make the same assumptions for the thermal problem that those used by the ε-NTU model, the most important ones for the aim of this paper are the followings:

- Negligible effect of 2D longitudinal heat conduction (2D LHC).
- No heat conduction between tubes trough the fin (adiabatic-fin-tip assumption).
- Application of the fin theory, which assumes uniform air temperature along the fin height.

These assumptions are studied in the literature for many heat exchangers topologies such as fin and tube heat exchangers, and the studies conclude that in common applications they have a negligible effect, only with important impact in special working conditions, e.g. large superheat in evaporators, as reported Domanski et al. (2007). But the effects of these assumptions are not studied so extensively for microchannel heat exchangers, either multi-pass or serpentine. The use of microchannels heat exchangers is increasing because of their compactness and high effectiveness. In the case of transcritical CO2 systems, microchannels have an additional merit related to their high mechanical strength. Recently, the authors have worked in these issues. In the first work that they presented about these topics (Martínez-Ballester et al. 2010), they did a literature review in which all these problems were investigated theoretically and experimentally.

Martínez-Ballester et al. (2010), proposed a model for a microchannel gas cooler referred to as Fin2D model. The model subdivides the heat exchanger into segments, and these segments are divided into cells, to which the corresponding system of energy-conservation equations is applied without traditional heat exchanger modeling assumptions. In this manner, the model accounts for 2D LHC in the tube and fin wall. It does not use any fin efficiency so it can model consistently the heat conduction between tubes. Since it applies a 2D discretization for the air in each segment, it does a more accurate integration of the heat transferred to the air from the fin, since the air temperature is more uniform in a cell unlike classical ε-NTU approaches which apply the fin theory that assume intrinsically uniform air temperature along the fin height. Furthermore, the Fin2D model allows for independent discretization for the refrigerant and the air. This fact is interesting to capture the air properties variation along the air flow direction. The aim of developing the Fin2D model was to evaluate the prediction errors of classical modeling techniques in an equivalent piece of a microchannel gas cooler and identify error sources. The conclusions of the study of Martínez-Ballester et al. (2010), related to this work, were:

- The impact of LHC effects along each direction in fins and tube walls, if considered separately, was not significant. The combined effect was more noticeable and resulted in a capacity prediction error of as much as 2.5%, with the LHC in the tube, along the air flow direction, being the dominant effect.
- Using the adiabatic-fin-tip efficiency, which is widely used, leads to large errors in heat distribution per tube when a temperature difference between tubes exists. In addition, this assumption considerably affects the global capacity prediction of gas coolers with large number of refrigerant passes.
- The temperature of air close to the tube wall was 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 case study was sufficient to identify the modeling deficiencies sources of classical methodologies in such kind of heat exchangers, but it did not allow the simulation of a gas cooler with actual size: number of tubes, length and circuitry. The main reason was the computational cost of the Fin2D model.

Considering all the conclusions of Martínez-Ballester et al. (2010), the goal of this work was to develop a model based on the Fin2D model which could capture the same phenomena, with a similar accuracy, but with a high reduction of the computational cost. Consequently, given this computational time reduction, the model proposed in this paper is able to simulate a microchannel gas cooler with an actual size with any refrigerant circuitry.

The model was validated using experimental data available in the literature. Regarding the computational cost and accuracy, the model presented in this paper was compared against the Fin2D model in terms of the computation time and capacity results.

2. MODEL DESCRIPTION

The model proposed in this paper, referred as Fin1Dx3, is based on the Fin2D model presented by Martínez-Ballester et al. (2010) performing some changes in order to reduce the computational cost but preserving the accuracy. The changes are based on the following considerations:

- The study of Martínez-Ballester et al. (2010) revealed that the longitudinal conduction in the fin along the air flow direction resulted in a negligible effect on the predicted performance results. Thus, in the present model this effect is not taken into account, what means no thermal joints between neighbouring fin cells along the *Z* direction, even though a discretization of the fin and air exists along this direction.
- The study of Martínez-Ballester et al. (2010) revealed that the air temperature profile is quite flat along the *Y* direction, excepting the air close to the tube wall. The discretization of air along the *Y* direction increases the computational cost. On the other hand it would be quite interesting to capture the effect of accounting for the temperature difference between the air close to the tube wall and the rest of the air. The decision for this conflict of interests is to discretize the air with three air cells along the *Y* direction, shown in Figure 1(a). That is the reason to refer the model as Fin1Dx3 model. The height of the air cells close to the tube wall is unknown; this dimension should be that one which provides the best results. The only restriction is that the fin cells located at the bottom and top of a fin, will measure the same.
- The Fin2D model solved the heat transferred from the air to the fin without applying the fin theory. This fact gives more freedom in the processes formulation but it needs to be solved numerically. So, to

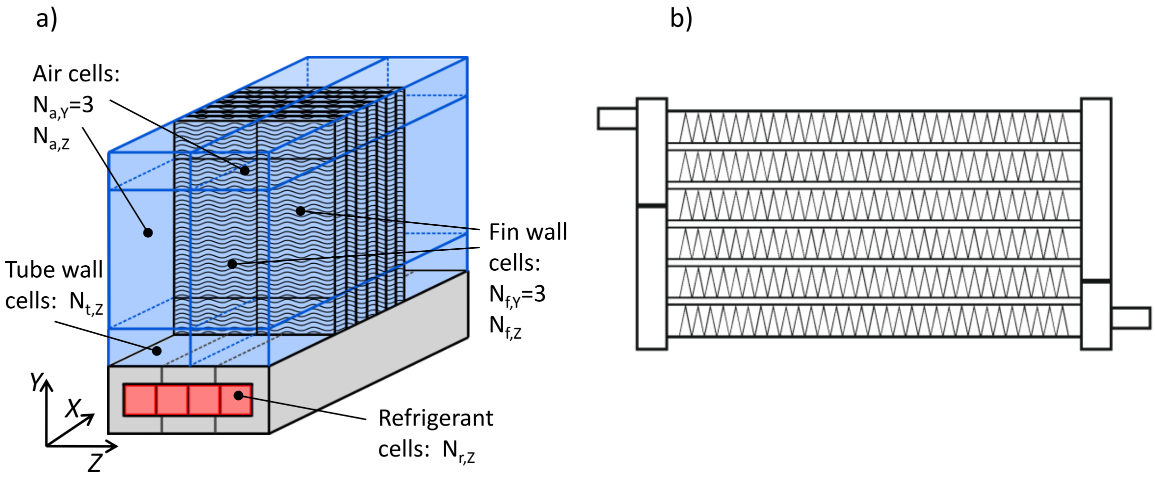

Figure 1. (a) Schematic of a heat exchanger segment. (b) Microchannel heat exchanger of three refrigerant passes.

solve accurately the 2D heat conduction in the fin a large number of fin cells along the *Y* direction were needed. This point is the one which introduced more computational cost in the Fin2D model. The wall temperature field, for a uniform cross-sectional fin, is governed by Eq. (1). Only when the air temperature and the heat transfer properties are constants, the solution for Eq. (1) is known and equal to Eq. (2), where θ represents the difference between the fin wall temperature and the temperature of the air. Thus, the main assumption of the fin theory, which is not satisfied in an actual fin, is that the air temperature is not uniform along the *Y* direction.

$$\frac{d^2T^f_{w,j}}{dY^2} - m^2_{j,i}\left(T^f_{w,j} - T_i\right) = 0 \quad (1)$$

$$\theta^f_{j,i}(Y) = C_{i,1}e^{m_{j,i}Y} + C_{i,2}e^{-m_{j,i}Y} \quad (2)$$

$$m^2_{j,i} = \frac{\alpha_{j,i}\ pw_{j,i}}{k_j\ A_{j,i}} \quad (3)$$

The discretization in the air, along the *Y* direction, has been chosen in order to represent air cells with no temperature variation in the *Y* direction. Since the discretization for the fin and the air is the same, it is possible to apply the fin theory solution (Eq. (2)) for each fin cell without failing the assumption of uniform air temperature. The result of this methodology is a big reduction of the grid size and consequently of the computational cost. Note, that Eq. (2) does not imply the adiabatic-fin-tip assumption, since boundary conditions have still not been applied. The evaluation of the constants $C_{i,1}$ and $C_{i,2}$ will be exposed in the section 2.2.

2.1. Heat Exchanger discretization

Figure 1(b) presents a sample of a microchannel gas cooler that can be simulated with the proposed model. The present model can simulate any refrigerant circuitry arrangement: any number of refrigerant inlets and outlets, and any connection between different tube outlets/inlets at any location.

First, the heat exchanger is chopped into segments along the *X* direction (refrigerant flow), resulting *Ns* segments per tube. The discretization for each segment is the same and it is shown in Figure 1(a). Each segment consists of: one refrigerant stream that is split into N_r channels in the *Z* direction (air flow); a flat tube that is discretized into N_{tw} cells in the *Z* direction; and both air flow and fins, which are discretized in two dimensions: three cells in the *Y* direction and N_a cells in the *Z* direction. Since the discretization for the air and fin wall is the same, $N_{fw} = N_a$. This discretization is summarized in the text as; grid: $\{Ns, N_r, N_{tw}, N_a\}$. For illustration of the nomenclature, if the heat exchanger of Figure 1(b) is cut into three pieces along the *X* direction and the discretization illustrated in Figure 1(a) is applied to it, the correspondent grid would be: {3,4,3,2}.

The refrigerant flows inside the channels along the *X* direction without any mixing between the channels, and it exchanges heat with the tube cells in contact; these tube cells transfer this heat to the air cells in contact by convection, to its neighbouring tube cells on the plane *X-Z* by conduction, and to the fin roots in contact by conduction. The air exchanges heat by convection with the fin cells, and the air cells at the bottom and top also exchange heat with the tube cells in contact. The fin cells conduct the heat along the *Y* direction, and the bottom and top fin cells also conduct heat to the adjoined tube wall.

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. The governing equations for the refrigerant side and the tube wall are the same that the described in the Martínez-Ballester et al.'s Fin2D model. Consequently, the proposed model accounts also for 2D LHC in the tube wall. Regarding the air flow and fin wall, the proposed governing equations are,

$$\dot{m}_{\mathrm{i}} dh_{\mathrm{i}} = d\dot{Q}^{f}_{j,i} + \sum_{\mathrm{j=1}}^{n_{\mathrm{i}}} \left(T_{w,j} - T_{i}\right) U_{\mathrm{j,i}}\, pw_{\mathrm{j,i}}\, ds_{\mathrm{i}} \tag{4}$$

$$d\dot{Q}^{f}_{j,i} = \alpha_{\mathrm{j,i}}\, pw_{\mathrm{j,i}}\, \theta^{f}_{j,i}\, dY \tag{5}$$

$$\sum_{\mathrm{j=1}}^{n_{\mathrm{fB}}} A_{\mathrm{j,fB}} \left.\frac{\partial\left(k_{j,fB} T_{w,j}\right)}{\partial Y}\right|_{Y=0} = A_{\mathrm{fB}} \left.\frac{d\left(k_{fB} T^{f}_{w,j}\right)}{dY}\right|_{Y=0} \tag{6}$$

$$\sum_{\mathrm{j=1}}^{n_{\mathrm{fT}}} A_{\mathrm{j,fT}} \left.\frac{\partial\left(k_{j,fT} T_{w,j}\right)}{\partial Y}\right|_{Y=Hf} = A_{\mathrm{fT}} \left.\frac{d\left(k_{fT} T^{f}_{w,j}\right)}{dY}\right|_{Y=Hf} \tag{7}$$

where any air cell *i* is in contact with n_{i} tube wall cells *i*=1, n_{i}; any fin bottom cell *fB* or fin top cell *fH* is in contact with n_{fB} or n_{fT} tube cells; the variable with the superscript *f* indicates that it is evaluated applying the specific functions for the fin wall; $k_{\mathrm{j,fT}}$ is the thermal conductivity of the tube cell *j* in the direction of *fH*. Eq. (4) states the energy conservation for an air cell.

In order to solve Eq. (5) the constants $C_{\mathrm{i,1}}$ and $C_{\mathrm{i,2}}$ must be known, resulting six constants for each fin. The value of these constants is obtained by imposing continuity in first derivative of $\theta^{\mathrm{f}}_{\mathrm{j,i}}$ respect *Y* along all the fin height. If two new variables are introduced, the temperature at the bottom T_{fB} and top T_{fT} of the fin, the restriction of continuity in first derivative gives six equations. In this manner, the fin temperature at any location can be obtained. Note that no adiabatic assumption has been done, so heat conduction between tubes is taken into account by this model. Since we have introduced two new variables, two new equations has to be introduced to solve the general problem. These equations are Eq. (6) and Eq. (7) which state a heat balance at the bottom and top of the fin respectively.

Pressure drop is considered in both fluids using the same equation,

$$\Delta P_{i} = \Delta P_{fr} + \Delta P_{acc} + \Delta P_{contr} + \Delta P_{\mathrm{exp}} \tag{8}$$

Where ΔP_{i} is the total pressure drop for a fluid cell *i*, ΔP_{fr} corresponds to the frictional term, ΔP_{acc} corresponds to the acceleration term, $\Delta P_{\mathrm{contr}}$ and ΔP_{exp} correspond to the sudden contraction and expansion that the air suffer at the inlet and outlet of the heat exchanger, and that the refrigerant experiments in the connections between tubes and the header.

The corresponding boundary conditions are: prescribed inlet conditions and velocity distributions for both fluids; and that the open edges of the tubes to the surrounding are considered adiabatic. For both fluids, uniform distribution is assumed.

The discretization of the governing equations applied in the refrigerant side and in the wall is described by Martínez-Ballester et al. (2010), but the discretization of the equations for the air side and fin wall is quite different from the one explained in that work. This methodology is not introduced in the present paper since it would require a deep analysis out of the scope for the present work.

The global solution method employed is called SEWTLE (Semi Explicit method for Wall Temperature Linked Equations) and it is outlined in Corberán 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 fluids flow 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 is obtained for any fluid cell, then the wall temperature at every wall cell is updated with the new fluid temperature field and using the correspondent equations for tube and fin wall. 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 *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.

3. MODEL VALIDATION

In order to validate the proposed model, a set of existing experimental results are going to be compared with the thermal capacity predicted by the model. The experimental data used and the gas cooler simulated are extracted from Yin et al. (2001), who measured a wide range of operating conditions. The experimental error reported by them was ±5%. Basically, the gas cooler modelled is a microchannel multitube heat exchanger with three refrigerant passes and a total number of 34 tubes with 11 channels per tube.

The grid applied to the gas cooler corresponds to: {5,1,3,3}. The authors studied the effect of simulating the actual number of channels or just one channel with identical hydraulic diameter, and they concluded that the differences were negligible. Thus, only an equivalent channel is modelled, although the actual tube has 11 channels. To determine the suitable size of the fin wall cells close to the tube, the results of Martínez-Ballester et al. (2010) were analyzed. In these studies it could be observed that the height close to the tube and occupied by air with a temperature different from the rest of the air was about 1/30 the fin height. Thus, for the experimental validation, the bottom and top of the fin had a height equal to 0.33% of the total fin height. This dimension should be object of future studies to determine a more justified and accurate value.

The different correlations to evaluate the heat transfer and pressure drop coefficients are listed in Table 1.

Figure 2(a) presents the predicted gas cooler capacity against the experimental values. Most of the errors are within the error bound of ±2 %. The accuracy is quite high since a linear function fitted to the predicted capacity had a slope of 0.9972, what represents an error of -0.28%, for the used data. The model underpredicts slightly the gas cooler capacity.

The exit refrigerant temperature is also compared against experimental data in Figure 2(b). In the figure are plotted the bounds of ± 1 K respect the measured temperature. The major part of the points deviates from the experimental data less than 1 K.

The predicted pressure losses of refrigerant were far from the experimental data, with a mean error of -80%. These errors are similar to those errors reported by Asinari et al. (2004) and Yin et al. (2001) when they evaluated this error with their own models. Yin et al. (2001) solve this disagreement introducing some dimensional changes in ports, produced by manufacturing defects. Asinari et al. (2004) demonstrate that introducing arbitrary multiplying factors solves the pressure losses disagreement with a negligible effect on the heat capacity results. They argue that the reason is based on underestimation pressure losses when traditional correlations are used for such a situations.

Table 1. Correlations used in the model for coefficients evaluation.

	Heat transfer coefficient	Friction coefficient	Expansion/Contraction pressure losses
CO_2	Gnielinski (1976)	Churchill (1977)	Kays and London (1984)
Air	Kim and Bullard (2002)	Kim and Bullard (2002)	Kays and London (1984)

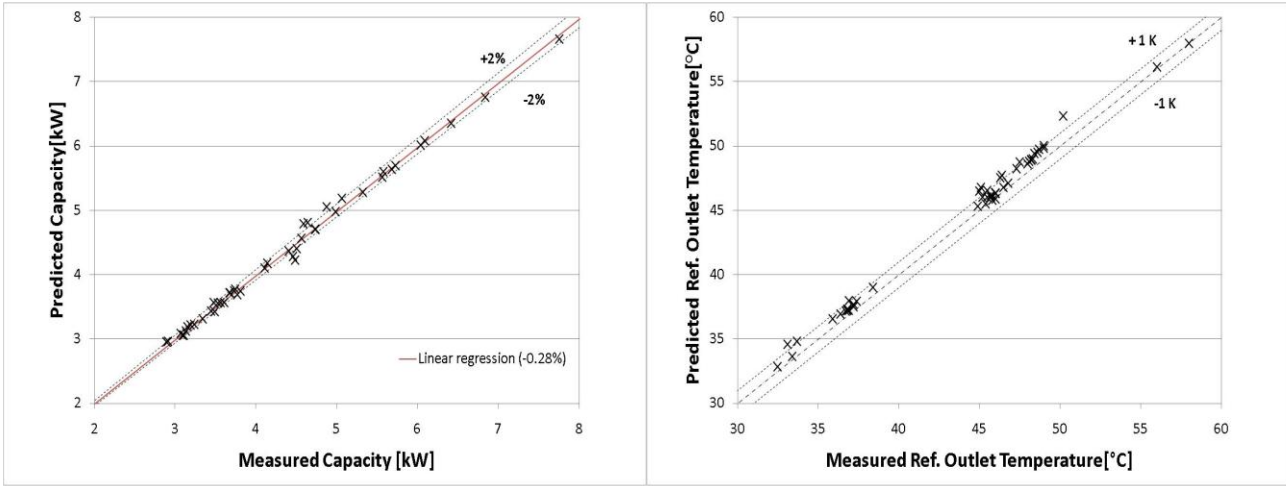

Figure 2. (a) Predicted capacity validation. (b) Predicted gas cooler outlet temperature validation.

When a gas cooler with high effectiveness is used to compare predicted results and experimental data, a good accuracy can be expected even with a simple model. When a model overpredicts heat capacity, the highest capacity that any model can predict is that one which results from a gas cooler operating with unitary effectiveness. Thus, an important factor to take into account in a model validation is the effectiveness of the gas cooler employed for obtaining the experimental data. If we define temperature approach as the temperature difference between the refrigerant outlet and the air inlet, this factor will be quite representative of the gas cooler effectiveness. A robust validation of a model will imply large approach values. For experimental data used, the approach is between 1 K and 7 K, with an average value of 4.1 K. This value indicates a high gas cooler effectiveness, in fact it had, for the data used, an average value of 83%

4. COMPUTATIONAL COST COMPARISON

The main reason to develop the proposed model is to achieve, preserving the accuracy, a computational cost reduction with respect to the Fin2D model, which requires a large computational effort. To this end, the computing speed and prediction results for each model were studied. Regarding the problems to solve, the Fin2D model was capable to solve only a pair of tubes with the correspondent fin surface between both of them. Thus, to evaluate the computing time only two tubes are going to be simulated. The geometry of these tubes and the correspondent fin surface are the same as those pointed out in the section 3. The operating conditions for the simulations are those used for the tests nº: 9, 17, 25, 33 and 41 (Yin et al. 2001). Both models applied the same grid, with the exception of the fin and air cells along *Y* direction. Due to the model differences, Fin2D model needs a large number of these cells; Martínez-Ballester et al. (2010) proposed using 30 cells in the *Y* direction. The grids applied for these scenarios are: {5,1,3,30,3} for Fin2D model and {5,1,3,3} for Fin1Dx3 model.

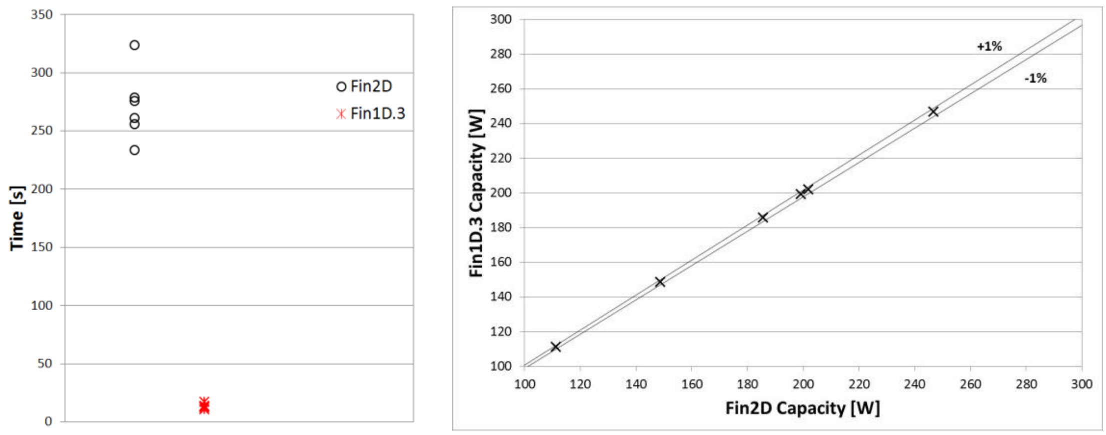

Figure 3. (a) Computing time comparison for both models. (b) Capacity results comparison between models.

The results for this study are presented in Figure 3 (a) and (b). In Figure 3(a), a large computing time reduction, from Fin2D model to Fin1Dx3 model, is noticeable. This reduction represents one order of magnitude. Regarding the accuracy of Fin1Dx3 model, Figure 3 (b) depicts the predicted capacity by each model. The differences in the results between both models, for all the simulated scenarios, resulted to be less than 0.3%.

5. CONCLUSIONS

The goal of the present work was to achieve a model that could reduce significantly the computational cost of the Fin2D model retaining its accuracy. In this manner, it allows using that model to analyse microchannel gas coolers with any refrigerant circuitry, including serpentine heat exchangers.

The motivation to develop this new methodology is based on the drawbacks that, in the authors' opinion, existing models have when they are applied to some recent designs of heat exchanger such as microchannel heat exchangers: multi-pass and serpentine. The model has been worked out following the research line of the authors of developing a new modeling methodology for heat exchangers which can take into account the heat transfer processes in a more fundamental way, paying also attention to the computational cost.

To this end, the main conclusions are the following:

- The Fin1Dx3 model accounts for the same processes than Fin2D model excepting the LHC in the fin along Z direction. However introduces a new methodology to describe the air-side heat transfer, using a composed function for the fin temperature and it only needs three air cells along the Y direction.
- The Fin1Dx3 model was validated with experimental data with the predicted capacity within ±2% error. The approach of the analysed data ranged between 1 and 7 K, with an average value of 4.1 K. Although pressure drop was drastically underpredicted, it did not affect the heat transfer results.
- The solving time of Fin1Dx3 has been reduced one order of magnitude respect to the Fin2D's time, whereas the differences on the results are less than 0.3%, considered as negligible for practical applications.

6. NOMENCLATURE

A	heat transfer area (m^2)	Greek symbols	
h	specific enthalpy (J kg^{-1})	α	convective heat transfer coefficient (W m^{-2} K^{-1})
Hf	fin height (m)	ε	heat exchanger effectiveness
k	thermal conductivity (W m^{-1} K^{-1})	θ	temperature difference between air and fin wall (K)
$\dot{m}$	mass flow rate (kg s^{-1})	Superscript	
N	number of cells	f	evaluated with a fin function
Ns	number of segments per tube	Subscript	
NTU	number of transfer units	a	air
P	Pressure (Pa)	fB	section at bottom of a fin
pw	wetted perimeter (m)	fT	section at top of a fin
$\dot{Q}$	heat (W)	fw	fin wall
s	length in the forward direction of a fluid (m)	i	fluid cell index
T	temperature (K)	j	wall cell index
U	overall heat transfer coefficient (W m^{-2} K^{-1})	r	refrigerant
X,Y,Z	spatial coordinates (m)	tw	tube wall
		w	wall

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