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# INSTITUTO DE INGENIERÍA ENERGÉTICA
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# A Novel Approach to Model the Air-Side Heat Transfer in Microchannel Condensers

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**Abstract**. The work presents a model (Fin1Dx3) for microchannel condensers and gas coolers. The paper focusses on the description of the novel approach employed to model the air-side heat transfer. The model applies a segment-by-segment discretization to the heat exchanger adding, in each segment, a specific bi-dimensional grid to the air flow and fin wall. Given this discretization, the fin theory is applied by using a continuous piecewise function for the fin wall temperature. It allows taking into account implicitly the heat conduction between tubes along the fin, and the unmixed air influence on the heat capacity. The model has been validated against experimental data resulting in predicted capacity errors within ±5%. Differences on prediction results and computational cost were studied and compared with the previous authors' model (Fin2D) and with other simplified model. Simulation time of the proposed model was reduced one order of magnitude respect the Fin2D's time retaining its same accuracy.

## 1. Introduction

The use of microchannels heat exchangers (MCHX) is increasing because of their compactness and high effectiveness. In the case of transcritical $CO_2$ systems, microchannels have an additional merit related to their high mechanical strength.

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 MCHX are available in the literature [1-6]). 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 $\varepsilon$-NTU methodology. Regardless the approach applied, all models usually make the same assumptions for the thermal problem that those used by the $\varepsilon$-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 Domanski et al. [7] reported. But the effects of these assumptions are not studied so extensively for microchannel heat exchangers. Recently, the authors have worked in these issues. Martinez-Ballester et al. [8] did a literature review in which all these problems were investigated theoretically and experimentally for microchannel heat exchangers.

Martinez-Ballester et al. [8] 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 unlike classical $\varepsilon$-NTU based approaches. 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 Martinez-Ballester et al. [8], related to this work, were:

- Impact of LHC in the fin on results was negligible.
- 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. The reason is that it does not take into account itself the heat conduction between tubes.
- The temperature of air close to the tube wall was very different than the bulk air temperature. This fact makes to fail the assumption of uniform air temperature used in the fin theory. Furthermore, it 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 required a large computational cost. Considering all the conclusions of Martinez-Ballester et al. [8], the goal of this work was to develop a model, in forth referred to as Fin1Dx3, based on the Fin2D model which could capture the same phenomena, with a similar accuracy, but with a high reduction of the computational cost. The governing equations of the whole microchannel condenser model it out of the scope for this paper, and the present paper focusses on the explanation of the approach used to model the air-to-fin heat transfer. For a full description of the whole MCHX model, and air-side approach in more detail the author is referred to Martinez-Ballester et al. [9].

The model was validated using experimental data available in the literature. Regarding the computational cost and accuracy, the Fin1Dx3 model was compared against the Fin2D and with a simplification based on Fin1Dx3, in terms of the computation time and predicted results.

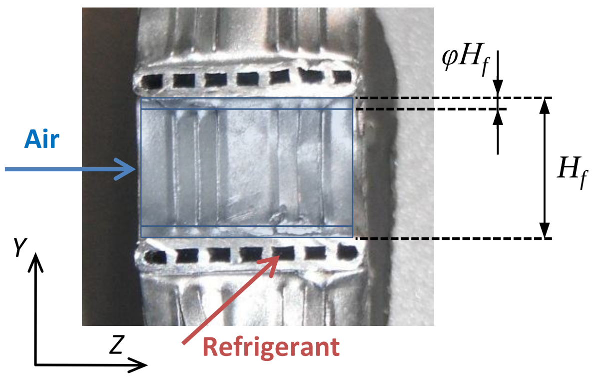


**Figure 1.** Detail of a louvered fin surface in a microchannel heat exchanger, where the non-louvered height and the total fin height are depicted.

## 2. Model Description

Based on conclusions of Martínez-Ballester et al. [8], the changes to obtain a model faster and with similar accuracy than Fin2D attends to the following considerations:

- Longitudinal heat conduction in the fin along the air flow direction is cancelled, what means in practice no thermal connections between neighbouring fin cells along the air flow direction.
- It is proposed to discretize the air with three air cells along the *Y* direction, as shown in figure 1. For this discretization, the height of the air cells close to the tube wall is unknown: this dimension ($\varphi H_f$) should be adjusted either experimentally, numerically, or even by observation. What actually these three air cells represent is the consideration of non-mixed air along the *Y* direction between them.
- The temperature field for a uniform cross-sectional fin is governed by equation (1). Only when the air temperature and coefficients are constant, or they are evaluated with mean values, the general solution for the equation (1) is equation (3) [10], where $\theta$ represents the difference between the fin wall temperature and the air temperature.

$$\frac{d^2T_f}{dY^2} - m_{f,a}^2 \left(T_f - T_a\right) = 0 \tag{1}$$

$$m_{f,a}^2 = \frac{\alpha_{f,a}\ \mathrm{pw}_{f,a}}{k_f\ A_{f,a}} \tag{2}$$

$$\theta_{f,a}(Y) = C_1 e^{m_{fa}Y} + C_2 e^{-m_{fa}Y} \tag{3}$$

In the Fin1Dx3 model, the discretization for the fin and the air is the same which has been chosen in order to represent air cells with uniform temperature, so it could be possible to apply the fin theory solution (equation (3)) for each air-fin cell connection without failing the assumption of uniform air temperature.

### 2.1. Governing Equations for the Air-Side Heat Transfer

First, each tube of the MCHX is chopped into segments along the *X* direction (refrigerant flow). The discretization for each segment is the same and it is shown in figure 2. Each segment consists of: one refrigerant stream that can be split into channels in the *Z* direction; a flat tube that is discretized into cells in the *Z* direction; and both air flow and fins, which are discretized in two dimensions: three cells in the *Y* direction and any number of cells in the *Z* direction. The discretization for the air and fin wall is the same.

Equation (3) is applied individually to each fin cell, resulting in a piecewise function (4) for a column of fin cells located at same *Z*,

$$\theta_{f,a}(Y) = \begin{cases} \theta_{f,a1}(Y) = T_{f1}(Y) - \bar{T}_{a1} = C_1\, e^{m_{f,a1}Y} + C_2\, e^{-m_{f,a1}Y},\ 0 \le Y \le \varphi H_f \\ \theta_{f,a2}(Y) = T_{f2}(Y) - \bar{T}_{a2} = C_3\, e^{m_{f,a2}\left(Y - \varphi H_f\right)} + C_4\, e^{-m_{f,a2}\left(Y - \varphi H_f\right)},\ \varphi H_f \le Y \le (1-\varphi) H_f \\ \theta_{f,a3}(Y) = T_{f3}(Y) - \bar{T}_{a3} = C_5\, e^{m_{f,a3}\left[Y - (1-\varphi) H_f\right]} + C_6\, e^{-m_{f,a3}\left[Y - (1-\varphi) H_f\right]},\ (1-\varphi) H_f \le Y \le H_f \end{cases} \tag{4}$$

In equation (4) $H_f$ is the fin height; $\varphi$ is the non-dimensional height of fin and air cells at the bottom and top of the fin; and $\bar{T}_a$ corresponds to an integrated value, of air temperature crossing the attached fin cell, along the *Z* direction. The air temperature for each corresponding region is assumed as uniform along the *Y* direction.

From application of equation (3) will result six unknown constants: $C_1$, $C_2$, $C_3$, $C_4$, $C_5$ and $C_6$. These constants must be evaluated from the boundary conditions of the heat transfer problem along the fin height, i.e. the temperature field must be continuous and derivable. Note that no adiabatic assumption is necessary to define these boundary conditions, so that heat conduction between tubes is taken into account. In this way it is possible to define $T_f(Y)$ as follows,

$$T_f(Y) = [T_{f1}(Y), T_{f2}(Y), T_{f3}(Y)]^{\mathrm{T}} = [\mathbf{A}(Y)][\bar{T}_{a1}, \bar{T}_{a2}, \bar{T}_{a3}, T_{fB}, T_{fT}]^{\mathrm{T}} \tag{5}$$

Where $T_{fB}$ and $T_{fT}$, illustrated in figure 2(c), correspond to the temperature at the bottom and top of the fin. [$\mathbf{A}(Y)$] is a 3x5 matrix that depends on $Y$, geometry, air-side heat transfer coefficient and fin conductivity. Note that $T_f$ has the interesting feature that is a pseudo-linear function with respect to $\bar{T}_a$, $T_{fB}$ and $T_{fT}$. Since the paper only focusses on the air-to-fin heat transfer, in the following it is assumed that $T_{fB}$ and $T_{fT}$ are given. Actually, in the whole model [9] for the MCHX these variables are obtained by imposing energy conservation at fin-to-tube connection.

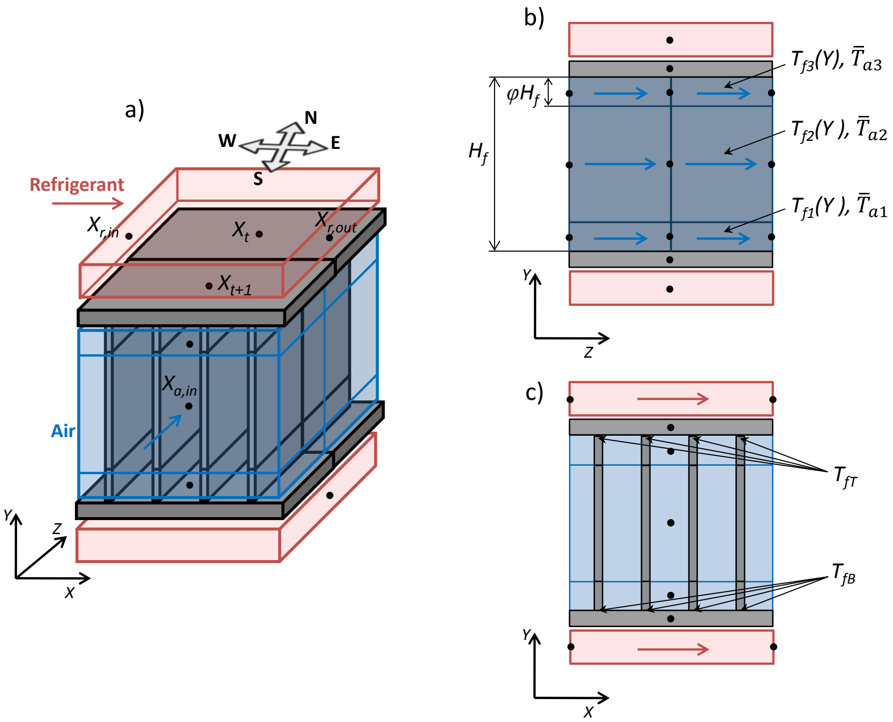


**Figure 2.** Different views of a discretized portion of heat exchanger: (a) global view illustrating fluid nodes and tube directions; (b) $Z$-$Y$ plane, which shows main geometric data of the fin and regions where is defined the corresponding Tf (Y) and ; (c) $X$-$Y$ plane, which shows the location of the $T_{fT}$ and $T_{fB}$ temperatures.

Equation (6) states the energy conservation in an air cell $a$ in contact with a fin cell $f$ and $n_t$ tube cells $t$=1, $n_t$.

$$\dot{m}_a dh_a = d\dot{Q}_{f,a} + \sum_{t=1}^{n_t} \dot{q}_{t,a} pw_{t,a} dZ \tag{6}$$

The heat flux $\dot{q}_{t,a}$ corresponds to the heat flux exchanged with each tube cell t, which is explained by Martínez-Ballester et al. [9], while the heat transferred to the neighbour fin cell $f$, can be evaluated by:

$$d\dot{Q}_{f,a} = \alpha_{f,a} pw_{f,a} \theta_{f,a} dY \tag{7}$$

### 2.2. Discretization of Governing Equations

In order to discretize the governing equations presented in the previous subsection, a finite volume method (FVM) was applied. To discretize the set of governing equations, first it is necessary to assume a temperature profile for the fluids or for the wall in order to estimate the integrals of the heat transferred. The linear fluid temperature variation scheme (LFTV) has been assumed for both fluids, as Corberán et al. [11] suggested for this application.

To obtain the outgoing temperature of the air, equation (7) has to be solved, so the integration of equation (8) must be done previously. The total heat transfer along the fin cell can be expressed as:

$$\int d\dot{Q}_{f,a} = \int \alpha_{f,a} p w_{f,a} \theta_{f,a} dY = \alpha_{f,a}\ A_{f,a}\ \bar{\theta}_{f,a} \tag{8}$$

A novel aspect of this model is that in order to include heat transfer from fin to air, integration of temperature difference $\theta_{f,a}$ is implemented in the model, while rest of models use directly a fin efficiency by applying the analytical relationship for adiabatic-fin-tip assumption, $\tanh(mL)/mL$ [10]. The advantage of using the integration of $\theta_{f,a}$ is that allows taking into account the heat conduction between tubes more easily and fundamentally than other fin efficiency based approaches. Furthermore a fin efficiency cannot be always be defined, e.g. when temperature at fin roots are not identical. This fact leads to some models, which use the adiabatic-fin-tip efficiency, to apply more or less artificial approaches in order to include heat conduction between tubes. It is important notice that this idea is independent on the discretization applied in air and fin, i.e. with just one air cell, instead of three as this paper proposes, is possible to apply this idea. Thus, there is neither accuracy nor computational cost reason to apply an approach based on the use of an adiabatic-fin-tip efficiency instead of the previous methodology, which is fundamentally more appropriated.

Once $\bar{T}_f$ is obtained by integration of $\bar{T}_f$ in equation (5) along the fin height, if $\bar{T}_a$ is subtracted from $\bar{T}_f$ and rearranging the result, $\bar{\theta}_{f,a}$ can be expressed as,

$$\left[\bar{\theta}_{f,a1}, \bar{\theta}_{f,a2}, \bar{\theta}_{f,a3}\right]^{\mathrm{T}} = \left[\bar{T}_{f1} - \bar{T}_{a1}, \bar{T}_{f2} - \bar{T}_{a2}, \bar{T}_{f3} - \bar{T}_{a3}\right]^{\mathrm{T}} = [\mathbf{B}]\left[\bar{T}_{a1}, \bar{T}_{a2}, \bar{T}_{a3}, T_{fB}, T_{fT}\right]^{\mathrm{T}} \tag{9}$$

[**B**] is a 3x5 matrix that depends on the same parameters as [**A**(Y)] excepting *Y*. $\bar{\theta}_{f,a}$ depends on the outlet temperatures of all the air cells located at the same *Z* (of the same segment), however note that $\bar{\theta}_{f,a}$ has the interesting characteristic, same as $T_f(Y)$, that is a pseudo-linear function with respect to the variables of the proposed problem. This fact will report interesting computation capabilities. Now all terms of equation (6) are known and it can be rewritten [9], as,

$$[\mathbf{\Omega}]\left[\bar{T}_{a1}, \bar{T}_{a2}, \bar{T}_{a3}\right]^{\mathrm{T}} = [\boldsymbol{\omega}]^{\mathrm{T}} \tag{10}$$

The solution of equation (10) gives the average air temperature and consequently the outlet air temperature of each air cell along the fin height of a fin. For a time step, equation (10) is a system of three linear equations, whose solution is known and easy to compute. Thus, the outlet temperatures of the three air cells of each fin column (in a segment) are obtained explicitly.

The presentation of this work has been assumed that fin roots temperatures are known, but in the global MCHX model they have to be calculated. An interesting point of the proposed fin discretization is that though fin wall is discretized into three cells, computationally it behaves as just one fin cell. In fact, the total number of unknown variables for each fin, regardless the number of fin cells, are two: temperatures at fin and bottom of fin.

For a full description of the global solution method, the reader is referred to [9].

## 3. Model Validation and Comparison with Others Authors' Approaches

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 was measured by García-Cascales et al. [4]. Two one-row condensers working with R410A were tested and have been simulated. Full description of working conditions, geometry and correlations used by the model can be found in Martínez-Ballester et al. [9].

The fin height ratio $\varphi$ could be adjusted experimentally, numerically or even by observation. According to the corresponding explanations above, it is possible to get a first approach from: typical dimensions of louvered fins used in this type of heat exchangers; a value of φ equal to 4% was assumed for the validation.

Figure 3 presents the predicted capacity against the whole set of experimental values, for both condensers. The model predicts the condenser capacity with errors within an error band of ±5%.

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. The other studied approach Fin1D is basically the same model as Fin1Dx3, and same phenomena modeled, but now neither air nor fins are discretized along the fin height. The analytical solution given by fin theory, for the case of given temperatures at fin roots, is used to get the fin temperature profile instead of a piecewise function. It also takes into account heat conduction between tubes.

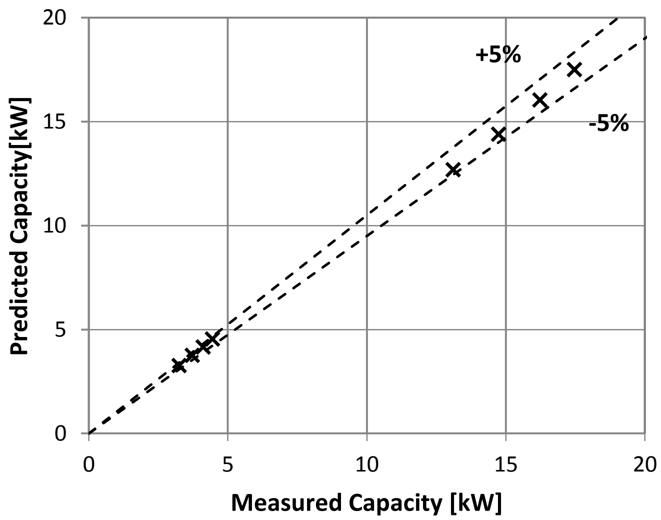


**Figure 3.** Model validation for two condensers by means of comparison between experimental and predicted capacity.

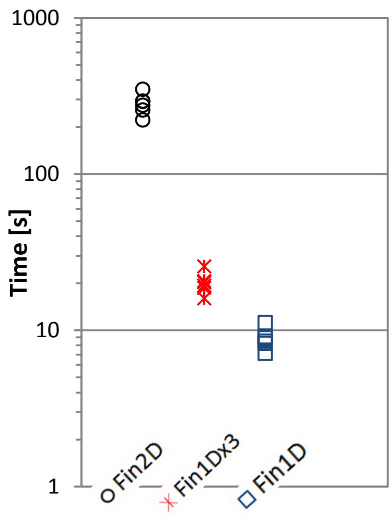


**Figure 4.** Comparison of the simulation time employed by each model.

Regarding the problems to solve; data of the MCHX geometry and the operating conditions are explained in [12]. All the compared models applied equivalent discretization grid [12].

In Figure 4, a large computing time reduction, from Fin2D model to Fin1Dx3 model, is noticeable. This reduction represents one order of magnitude. The main reason is the large difference of air and fin cells used by both models. In the case of Fin1Dx3 model, a piecewise function which consists of three one-dimensional functions is enough to capture accurately the actual fin temperature profile and consequently the heat transfer from fin to air. However, in practice Fin2D needs to apply 30 fin and air cells to get accurate results.

The simulation time reduction from Fin1D to Fin1Dx3 model is not as drastic as for Fin2D case, Fin1D needs half the time spent by Fin1Dx3. A priori, a larger simulation time reduction could be expected. However an interesting fact of the piecewise function applied in the Fin1Dx3 model is the following; the piecewise function uses as unknown variables the temperatures of the three air cells and the fin roots. The Fin1D model also includes as unknown variables the fin roots temperatures but only one air temperature value. Thus only two variables are saved in the Fin1D model with regard to Fin1Dx3 model, which corresponds to the air temperature values. In other words, the only cells that add computational cost to the Fin1Dx3 models are the air cells whilst the three fin cells behave numerically as just one.

Regarding the accuracy of Fin1Dx3 model, differences in results between Fin1Dx3 and Fin2D models, resulted to be less than 0.3% for all the simulated scenarios [12]. However, this deviation turns to be up to 2% in the case of Fin1D model [12].

Fin1D and Fin1Dx3 take into account same phenomena but with different fin/air discretization. Thus, the deviation between predicted results of both models is consequence only of a more accurate application by Fin1Dx3 model of fin theory for the air-to-fin heat transfer evaluation. In other words, this difference could be interpreted as the effect of non-mixed air along $Y$ direction. Nevertheless, this deviation can be interpreted as small, though the effect would depend on the operating conditions, heat exchanger and application. For an evaporator, dehumidification appears and plays an important role and what happens depends strongly on local properties.

## 4. 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. To this end, the main conclusions are the following:

- The large number of fin cells needed by Fin2D model to solve accurately the air-side heat transfer, is compensated in Fin1Dx3 with a novel methodology to describe the air-side heat transfer, using a piecewise function for the fin temperature. This piecewise function, together with the employed air discretization, allows applying in a more fundamental way the fin analytical solution.
- The main capabilities of Fin1Dx3 are: non-mixed air effects due to temperature difference between bulk air and the air close to the tubes; and it accounts fundamentally for heat conduction between tubes since it does not apply adiabatic-fin-tip assumption.
- The equations have been discretized, with the interesting characteristic of resulting in a system of pseudo-linear equations with respect to the variables of the problem. The numerical scheme proposed allows computing the three fin cells with the computational effort of just one fin cell.
- Fin1Dx3 model was validated with experimental data, for a condenser. The predicted capacity is within ±5% error.
- The solving time of Fin1Dx3 has been reduced one order of magnitude with regard to the Fin2D's time, whereas the differences on the results are less than 0.3%.
- The computation time difference between Fin1Dx3 and Fin1D turned out to be twice with difference on results for a condenser as much as 2%. For an evaporator where

dehumidification appears and what happens depends strongly on local properties, the difference could be higher.

The alternative methodology, proposed in this work, consists in evaluating the heat transfer by integration of the corresponding fin temperature profile instead of using a fin efficiency which cannot always be defined, e.g. when temperature at fin roots are different. It has been shown that this integration does not represent an obstacle since it can be easily discretized consistently with the rest of governing equations; therefore there is neither accuracy nor computational cost reason to apply adiabatic-fin-tip assumption when it is not satisfied. The approach proposed in this paper is developed for a three air-fin cells discretization but this conclusion is equally applicable to a single air-fin discretization.

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