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

INSTITUTO DE INGENIERÍA ENERGÉTICA
(Institute for Energy Engineering)

Research Publications

INTERNATIONAL JOURNAL OF REFRIGERATION 36 (2013) 191–202

Available online at www.sciencedirect.com

SciVerse ScienceDirect

journal homepage: www.elsevier.com/locate/ijrefrig

Numerical model for microchannel condensers and gas coolers: Part II – Simulation studies and model comparison

*Santiago Martínez-Ballester**, José-M. Corberán, José Gonzálvez-Maciá*

Instituto de Ingeniería Energética, Universidad Politécnica de Valencia, Camino de Vera s/n, Valencia 46022, Spain

ARTICLE INFO

Article history:
Received 31 January 2012
Received in revised form
26 August 2012
Accepted 27 August 2012
Available online 7 September 2012

Keywords:
Circuitry
Gas cooler
Simulation
Microchannel
Fin cut
Design

ABSTRACT

For a microchannel heat exchanger (MCHX), given the working conditions, main geometric data of the fin and tubes, heat transfer and face areas, there are multiple choices for the refrigerant circuitry and aspect ratio. Numerical studies using the Fin1Dx3 model, presented in Part I, are undertaken in order to assess the impact on the heat transfer of these design parameters for a microchannel gas cooler. The effect of fin cuts in the gas cooler performance has also been studied numerically as function of the refrigerant circuitry, where it has been found that an optimum circuitry for the use of fin cuts exists. Finally, with the aim of presenting the Fin1Dx3 model as a suitable design tool for MCHX, the model has been compared against the authors' previous model (Fin2D) and other representative models from the literature in terms of accuracy and computational cost. The Fin1Dx3 model has reduced the simulation time by one order of magnitude with regard to Fin2D, and in terms of accuracy deviates less than 0.3%.

Modèle numérique pour les condenseurs à microcanaux et les refroidisseurs de gaz : Partie II – Études de simulation et comparaison des modèles

Mots clés : Circuits ; Refroidisseur à gaz ; Simulation ; Microcanal ; Coupure d'ailette ; Conception

1. Introduction

Currently, an increasing interest in microchannel heat exchangers (MCHXs) has arisen in refrigeration and air conditioning applications due to their high compactness and high effectiveness. The high effectiveness is a consequence of large heat transfer coefficients as a result of using small hydraulic diameters. Given an air side heat transfer area, high compactness means a reduced volume, resulting in light heat exchangers with high mechanical strength being able to operate with low refrigerant charges.

Natural refrigerants are considered more environmentally friendly than other commonly-used refrigerants with similar or even better performance. However, working with some

* *Corresponding author*. Tel.: +34 963 879 121; fax: +34 963 879 126.
E-mail address: sanmarba@iie.upv.es (S. Martínez-Ballester).

0140-7007/$ – see front matter

http://dx.doi.org/10.1016/j.ijrefrig.2012.08.024

Nomenclature

A	heat transfer area (m^2)
H	height (m)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
L	length (m)
LHC	longitudinal heat conduction
N	number of refrigerant passes
P	wetted perimeter (m)
$\dot{Q}$	heat transfer (W)
R	thermal resistance ($K\ W^{-1}$)
T	temperature (K)
t	thickness (m)
v	air velocity ($m\ s^{-1}$)
d	tube depth (m)
X, Y, Z	spatial coordinates (m)
Greek symbols	
α	convective heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
η	fin efficiency
λ	multiplier
θ	temperature difference (K)
Subscript	
a	air
f	fin
fB	fin base
r	refrigerant
t	tube index

natural refrigerants has the following chief drawbacks: ammonia is toxic in large quantities; propane is highly flammable, and in fact IEC 60335-1 (2010) restricts the amount of hydrocarbon that can be used in a system to 150 g; carbon dioxide is neither toxic nor flammable but it works at high pressure, requiring of high mechanical strength components. Therefore, the features of MCHXs play an important role in the use of natural refrigerants: reduced volumes for getting low refrigerant charges in the case of flammable refrigerants like propane, and high mechanical strength in the case of transcritical CO_2 systems. Additionally, a suitable heat exchanger design for obtaining low refrigerant charges is a serpentine MCHX. This kind of heat exchanger minimises the refrigerant charge because it has no headers, thus saving this volume and the corresponding refrigerant charge.

Nowadays, simulation software is an appropriate tool for the design of products in which complex physical phenomena occur. These tools allow the saving of a lot of cost and time in the laboratory. Currently, some models for MCHXs are available in the literature: Asinari et al. (2004); CoilDesigner (2010), Fronk and Garimella (2011), García-Cascales et al. (2010), MPower (2010), Shao et al. (2009), and Yin et al. (2001). The modelling approaches and assumptions employed by them were extensively discussed in Part I (Martínez-Ballester et al., 2012), where the authors of the current work presented the fundamentals of the new proposed model: Fin1Dx3. This model is based on the previous Fin2D model (Martínez-Ballester et al., 2011) but introduces a new formulation, which allows the same accuracy to be retained with a large reduction in the computational cost. In the Fin1Dx3 model, the main heat transfer processes, which are modelled in a different and novel way with respect to other MCHX models available in the literature, are:

- 2D longitudinal heat conduction (LHC) in the tube.
- Heat conduction between tubes along the fin in contrast with the usual adiabatic-fin-tip assumption.
- Consideration of an air temperature zone close to each tube wall, in addition to the air bulk temperature.

In air-to-refrigerant heat exchangers, heat conduction between tubes along the fins appears when a temperature difference exists between the tubes, which always degrades the heat exchanger effectiveness. Several experimental studies indicated that the heat exchanger performance can be significantly degraded by the tube-to-tube heat transfer via connecting fins. Domanski et al. (2007) measured as much as a 23% reduction in the capacity of a finned-tube evaporator when different exit superheats were imposed on individual refrigerant circuits. This heat conduction and its negative effects can be avoided by cutting the fins, what has been studied in the literature. For a finned tube gas cooler, Singh et al. (2010) reported heat load gain of up to 12% and fin material savings of up to 40%, for a target heat load, by cutting the fins. However, not so large improvements have been achieved for MCHXs, namely: Asinari et al. (2004) concluded that the impact of using the adiabatic-fin-tip, which assumes no heat conduction, in predicted results can be considered negligible for a wide range of applications; Park and Hrnjak (2007) reported measurements of capacity improvements of up to 3.9% by cutting the fins in a CO_2 serpentine microchannel gas cooler.

Application of the fin theory is an assumption widely used and necessary when a model uses fin efficiency to evaluate the heat transfer from fins to air. The fin efficiency is based on the fin theory that assumes uniform air temperature along the fin height, which is not always satisfied, as explained in Part I (Martínez-Ballester et al., 2012) (Sections 1 and 2). In the literature, only a few models discretize the governing equations along the fin height and do not use the fin efficiency theory.

The Fin1Dx3 model proposed in Part I (Martínez-Ballester et al., 2012) takes into account all previously explained effects, and it can simulate any refrigerant circuitry regarding the number of refrigerant passes, tubes and tube connections. In addition, the model has the option of working in two different modes: continuous fin or fin cut. The reason for these two modes is to be able to evaluate the improvements by cutting the fins on the heat transfer.

Through the design process of an MCHX, the geometric data of tubes and fins are usually imposed by the manufacturer. Fin pitch, heat transfer area and face area of an MCHX is usually obtained by consideration of performance requirements. However, given a working conditions, multiple choices exist for the number of refrigerant passes, refrigerant connections and the aspect ratio (L/H) of the MCHX. In fact, some simulation software like EVAP-COND (2010) has the capability of optimising the heat load, varying the circuitry of a finned tube heat exchanger. Shao et al. (2009) studied the effect of the number of refrigerant passes for a serpentine MCHX working as a condenser, with the same face area and heat transfer area. The authors obtained up to 30% differences on heat load only by changing the number of refrigerant passes. Given that the circuitry has an important influence on the heat exchanger performance, the usefulness of simulation software for this purpose is clearly justified, since optimisation via experimentation would take too long, is difficult and expensive.

On the other hand, depending on the model's assumptions some parameter can be studied or not, e.g. the impact of the aspect ratio (L/H) on the heat transfer of a heat exchanger would be null if it is evaluated with a model which applies the adiabatic-fin-tip efficiency. This design parameter can only be assessed if the model adequately accounts for the heat conduction between tubes.

A model that uses the adiabatic-fin-tip without any correction term, to take into account the heat conduction between tubes, is always predicting results as if the heat exchanger had all fins cut, hence these models always overpredict the heat transfer (Domanski et al., 2007). In order to evaluate the effect of cutting fins, the model has to be able to simulate both scenarios; with and without the fin cut. There are few models that can estimate the impact of cutting fins on the prediction results. For finned tubes, Singh et al. (2008) presented a model, referred to as a "resistance model", to account for heat transfer between tubes through the fins. They included a term for heat conduction along fins between neighbouring tubes while still using the concept of adiabatic-fin-tip efficiency. The drawback of this methodology is the use of a set of multipliers that are dependent on the problem, which have to be determined either experimentally or numerically. Asinari et al. (2004) proposed a three-dimensional model for microchannel gas coolers using CO_2 as the refrigerant; the model employs a finite-volume and finite-element hybrid technique. They applied this model to evaluate the effect of heat conduction between tubes for one gas cooler, without any modification, operating in the operating conditions of one test. Martínez-Ballester et al. (2011) presented a model referred to as Fin2D which did not apply fin theory and was able to assess the impact of the fin cuts, but with a large computational cost since it needs to use a large discretization of the fin surface.

According to the ideas previously put forward, the authors considered studying some design parameters of an MCHX such as: aspect ratio and number of refrigerant passes. The influence of fin cuts was also studied for different refrigerant circuitry. The impact of all these parameters depends strongly on the heat conduction between tubes, LHC and air-side heat transfer. Hence the need for the use of a model which accurately takes into account all previous phenomena, otherwise it would not be possible to evaluate the effects of some of the aforementioned parameters on the MCHX performance. To this end, the simulation studies were carried out with the new proposed model Fin1Dx3.

The more sensitive the case study to LHC and heat conduction between tubes, the larger the impact will be on the performance due to variations in the defined parameters. The impact of LHC and heat conduction between tubes will increase as the temperature gradient on a tube and the temperature difference between tubes increases. That is the reason why a microchannel gas cooler working with CO_2 in transcritical pressures has been chosen as the case study. The reasons are based on the temperature glide of CO_2 during supercritical gas cooling, in contrast with a condenser where the temperature during condensation remains approximately constant. Representative values can be extracted from the experimental results of Zhao et al. (2001), where CO_2 undergoes temperature variations along a single tube from 25 K up to 85 K while the maximum temperature difference between two neighbouring tubes ranges from 30 K to 100 K. These kinds of numerical study on MCHXs are barely available in the literature. The goal of the case studies selected is to contribute to a better understanding of the influence of some of the design parameters on MCHX performance.

The goal of Part I (Martínez-Ballester et al., 2012) was to present the Fin1Dx3 model as a tool for the simulation of MCHXs. In Part I (Martínez-Ballester et al., 2012) it was explained that the model discretization is based on the Fin2D model (Martínez-Ballester et al., 2011), which had large computational requirements. However the new discretization of Fin1Dx3 allows, for same accuracy, a considerable reduction in the number of both air and fin cells, so that a large reduction in computational cost is expected. In order to assess the degree of accomplishment achieved regarding these statements, a comparison study between the accuracy and simulation time of the Fin2D and FinDx3 models has been carried out.

Part I (Martínez-Ballester et al., 2012) extensively discussed how other authors model the heat conduction between tubes and the air-side heat transfer, thus that Part II presents a comparison between the Fin1Dx3 model and an alternative approach that is representative of others models from the literature, regarding the modelling of these phenomena. This alternative approach is based on the work of Singh et al. (2008) and Lee and Domanski (1997). Although these approaches were originally proposed for finned tube heat exchangers, in this paper they have been adapted to MCHXs.

2. Simulation studies

In an MCHX design, the first step is to choose the geometric data of the tubes and fins, such as the minor tube dimension, major tube dimension, fin height and fin depth. This choice is based on manufacturing requirements, e.g. costs, tooling and volume production. The rest of the geometric parameters are related to the air side heat transfer area and face area. Given the inlet conditions and mass flow rates for both refrigerant and air, the heat transfer area can generally be fixed by imposing a target heat load, while the face area of the MCHX is obtained from pressure drop criteria.

Once these areas have been chosen, there are multiple circuit designs that satisfy the target heat load so that the refrigerant circuitry can be designed in order to optimise the heat exchanger effectiveness by maximising the heat transfer, with some restrictions regarding pressure drop. In the same way, parameters like the aspect ratio (*L*/*H*) play a similar role to the circuitry: several aspect ratios satisfy the performance requirements but just one optimises the effectiveness.

Fin cuts are another possible improvement to be introduced in an MCHX design. Obviously, the improvement in heat transfer will be null for a single-pass heat exchanger since all the tubes have the same temperature and the heat conduction between them would be zero. Thus it is worth assessing the improvements due to fin cuts in an MCHX for a different number of refrigerant passes.

2.1. Simulation methodology and case study description

The MCHX chosen for these studies corresponds to a gas cooler, according to the reasons presented in the introduction. The gas cooler geometry is based on the gas cooler tested by Yin et al. (2001), which corresponds to a microchannel gas cooler used in automotive applications, with CO_2 as the working fluid in transcritical conditions. This gas cooler consists of 34 tubes with 3 refrigerant passes. The number of refrigerant passes is one parameter to be studied, from one pass up to the limit that corresponds to a serpentine gas cooler, i.e. the refrigerant passes equal the tube number, without changing the rest of the gas cooler dimensions and inlet conditions. Increasing the number of refrigerant passes leads to larger velocities of refrigerant flow. This fact, besides the increase in refrigerant path length, produces a much larger pressure drop. The limiting case (serpentine MCHX) would be, for this reason, of no practical use.

The total number of tubes and some geometric dimensions of the gas cooler tested by Yin et al. (2001) have been modified so that the change in number of refrigerant passes will not produce excessive pressure losses for the serpentine case. The total number of tubes was reduced to 12 and the rest of dimensions, such as gas cooler width and height, were obtained by rescaling the original ones proportionally to the number of tubes. The resulting geometric data is shown in Table 1. The rest of the geometric data concerning fins and tubes was the same as tested by Yin et al. (2001).

For all scenarios the refrigerant and air side areas, face area and the rest of the geometry are the same. Inlet conditions for both fluids in the gas cooler are going to be identical for all simulation studies. Regarding the operating conditions, those corresponding to test no. 2 from Yin et al. (2001) have been chosen. Both the mass flow rate and air flow rate have been modified in order to obtain the same fluid velocities as the original values, according to the new geometry. The operating conditions are listed in Table 2. In relation to the air, there are two scenarios: with the mass flow rate given in Table 2, and with that mass flow rate divided by three.

The correlations used by the model are listed in Table 3.

Table 1 – Geometric characteristics of the gas cooler studied.

Face area (cm^2)	242.5	Refrigerant side area (cm^2)	609
Airside area (cm^2)	6465	Tubes number of tubes	12
Tube length (mm)	192	Core depth (mm)	16.5
Fin type	Louvred	Fin density (fins/in)	22
Number of ports	11	Port diameter (mm)	0.79
Wall thickness (mm)	0.43	Fin height (mm)	8.89
Fin thickness (mm)	0.1		

2.2. Number of refrigerant passes

The number of refrigerant passes is varied from one pass up to the maximum possible number, i.e. 12 passes, which corresponds to a serpentine configuration. Fig. 1 depicts two samples of cases studied. The performance differences will only be due to the number of passes, since the refrigerant area, air side area, face area and the rest of the geometry do not change.

Fig. 2 shows the results of this study for two different values of air velocity. As the air velocity is increased, the heat transfer is also increased for all cases due to: the mass flow rate rising since the air velocity is increased with the same face area; the overall heat transfer coefficient increasing because the greater the air velocity the larger the air side heat transfer coefficient. When the number of passes is increased the total refrigerant cross-sectional area is reduced so that the refrigerant velocity rises to keep the mass flow rate constant, which improves the heat transfer coefficient. Thus for this case study, the figure clearly shows that the heat transfer is always raised, with an asymptotic trend, by increasing the number of passes. Regarding refrigerant pressure losses, Fig. 3 shows the total pressure drop along the heat exchanger when the number of refrigerant passes is modified. Only the scenario corresponding to the air velocity of 3 m/s has been plotted because the results are very similar since the impact of the heat transfer on the pressure drop in the refrigerant side is negligible for this scenario. It should be noted that the case study corresponds to a gas cooler, which does not undergo a phase change.

In a condenser, the pressure drop leads to a temperature drop during the phase change, therefore the temperature difference between the air and refrigerant would reduce, and the heat transfer would be reduced. In this way, for condensers/evaporators the pressure drop plays an important role in heat transfer, in fact there is an optimum value on the heat transfer when the number of refrigerant passes is studied, due to the opposing influence of the heat transfer coefficients and pressure drop. This conclusion was also made by Shao et al. (2009) in their investigations for a serpentine microchannel condenser, where they studied the influence of the number of passes on the heat transfer.

2.3. Influence of fin cuts

A technique to improve the effectiveness of air-to-refrigerant heat exchangers is the cutting of the fins. The heat conduction between tubes, due to temperature differences from the bottom to the top of the fin roots, degrades the heat exchanger

Table 2 – Operating conditions for the simulation studies.

	Inlet pressure (kPa)	Pressure drop (kPa)	Inlet temperature (°C)	Outlet temperature (°C)	Mass flow rate (g/s)
CO_2	10,792	421.6	138.6	48.2	5.64
Air	100	61×10^{-3}	43.5	–	87.3

effectiveness. By cutting the fins, this heat conduction is avoided.

This technique is suggested for heat exchangers which have large temperature differences between tubes. For example, in a condenser there are tubes with superheated vapour flowing inside which are connected through fins to other tube with saturated vapour inside. Under these conditions large temperature differences can be expected. An extreme case corresponds to a gas cooler arrangement, in which the refrigerant undergoes a temperature variation along the whole gas cooler length, since there is no phase change. Thus, the temperature difference between two neighbouring tubes can be as large as 50 K.

As mentioned in the introduction, only a few models exist that take into account the heat conduction between tubes. The rest of the models always overpredicts the heat transfer for the same conditions, since they do not account for the degradation in effectiveness caused by heat conduction. The impact expected on the effectiveness by the cutting of fins is not the same for a finned tube as for an MCHX. In a finned tube heat exchanger the fin cuts can be made perpendicularly to the air flow direction, thus longitudinal heat conduction between rows of tubes is avoided, which always degrades effectiveness. The degree of degradation depends on many factors such as geometry of fins and tubes, operating conditions and fluids arrangement. In an MCHX the fins are cut along the air flow direction so that the effect introduced by them is not fundamentally the same as in the finned tube case, in fact the improvements on the capacity are lower: Singh et al. (2010) reported capacity improvements of up to 12% for a finned tube heat exchanger, whereas Park and Hrnjak (2007) obtained an improvement of 3.9% for a serpentine microchannel gas cooler. Note that fin surfaces commonly used for MCHXs are louvred, which have louvres that already prevent longitudinal heat conduction in the fin in the air flow direction.

The fin cuts can be customised according to the working conditions and heat exchanger circuitry. Singh et al. (2010) analysed different fin cut arrangements for a finned tube gas cooler. In the present study the fin cuts studied are arranged along the middle section between two neighbouring tubes for all the fins of the heat exchanger. Fig. 4 shows an example of this fin cut arrangement. The Fin1Dx3 model is developed for a continuous fin, but can be slightly modified to incorporate a cut in a section at half the fin height. This change implies changing two boundary conditions of the piecewise function for the fin temperature, which was presented in Part I (Martínez-Ballester et al., 2012). As a consequence of changing the boundary conditions it is also necessary to obtain the new matrixes of the model: [A], [B], [C] (Part I (Martínez-Ballester et al., 2012)).

To the authors' knowledge there are no numerical studies for MCHXs concerning the influence of refrigerant circuitry on the impact of fin cuts. To this end, the impact of cutting the fins has been evaluated for the same refrigerant passes studied in the previous subsection.

The results are shown in Fig. 5, where the heat transfer improvement by cutting fins has been plotted with respect to the solution given by the same model under the same conditions but without fin cuts, i.e. continuous fins. The heat improvement for one pass is zero because with this arrangement all the tubes have same temperature evolution, resulting in a null temperature difference between tubes at the same X coordinate. In such a case the adiabatic-fin-tip assumption is fundamentally correct.

The first interesting fact is that the influence of the air velocity on the parameter studied does not change the trend of the curves, it only moves them vertically. Thus, if we study the plot for $v = 1$ m/s, when the number of passes is different from one there is always an improvement in the heat transfer as a result of the cutting of the fins and, for the studied conditions, there is a maximum value for 3 passes, regardless of air velocity. A possible explanation for the presence of a maximum in the heat improvement is described below.

When the number of passes is two, the fin roots which connect two tubes of different passes (central tubes of the heat exchanger) have a large temperature difference that produces a heat conduction flux. As the number of passes is increased the temperature difference between tubes decreases, but the number of fins with such a temperature difference rises. Fig. 1 illustrates this explanation, where the heat exchanger with 3 passes has two zones with a large temperature difference, regions “a” and “b”. The serpentine heat exchanger has a similar and smaller temperature difference between all the tubes, which can be represented by the temperature difference at zone “c”. The heat exchanger with 3 passes will have only two zones with a temperature difference, but the temperature difference between the

Table 3 – Correlations for coefficients evaluation used in the model.

	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)

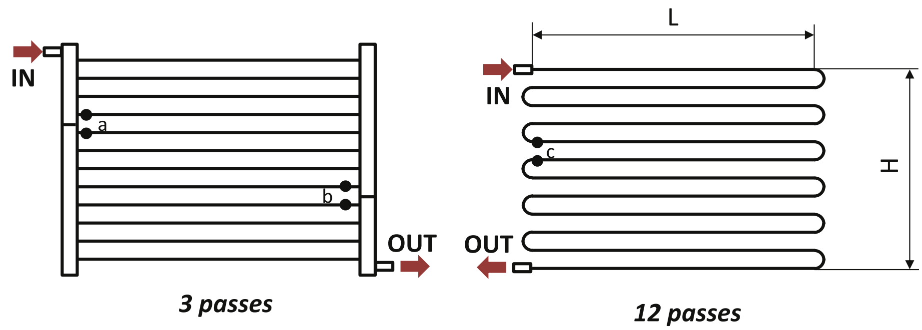

Fig. 1 – Schematics of two gas cooler arrangements studied: 3 and 12 refrigerant passes.

bottom and top of zones "a" or "b" is much higher than the corresponding value for region "c" of the serpentine case, though the serpentine MCHX has 11 regions with a similar temperature difference to the "c" zone. These opposing effects could be one of the reasons explaining the presence of the maximum depicted in Fig. 5.

Regarding the influence of air velocity on these results, Fig. 5 shows that the lower the velocity the larger the improvement. This fact was already pointed out by Singh et al. (2010) in their simulation studies for a finned tube gas cooler.

The maximum improvement that can be obtained depends on the air velocity, but for the scenarios studied this improvement is as much as 3%. Similar values were reported by Park and Hrnjak (2007), who measured capacity improvements of up to 3.9% for a serpentine gas cooler.

2.4. *Influence of aspect ratio for a serpentine gas cooler*

A serpentine MCHX corresponds to an MCHX with a single tube which is bent in order to provide a specific number of refrigerant passes. It has the peculiarity of not having headers, therefore it is highly recommended for saving refrigerant charge thanks to its reduced internal volume.

A restriction to these studies is the fact that the air side and face area are constant, while the aspect ratio (L/H) changes. From observations of the serpentine MCHX design, it is deducible that the air-side heat transfer area is proportional to the product: $N\ L$, which means the total refrigerant path length. Therefore, to study the isolated effect of the aspect ratio on performance, $N\ L$ will have to remain unchanged for all cases studied. The baseline gas cooler corresponds to the twelve-pass gas cooler studied in subsection 2.2. When the aspect ratio changes the gas cooler length becomes larger or shorter, so the number of refrigerant passes will have to change to keep $N\ L$ constant. Table 4 lists the corresponding length L, gas cooler height H, and aspect ratio, when the number of refrigerant passes N is varied according to the previously stated restrictions.

Since the tube length changes, the number of segments used by the model to discretize the gas cooler also change in order to maintain same accuracy for all cases.

Fig. 6 shows the results for the predicted heat transfer as a function of the aspect ratio. The figure shows the results for the two cases analysed: with the fin cut and a continuous fin. The figure shows that the aspect ratio has no effect on heat transfer when the fin is cut, thus models that apply the adiabatic-fin-tip assumption will not be able to study this influence since the results are always the same.

For the case of a continuous fin, Fig. 6 shows that heat transfer has a strong dependence on the gas cooler aspect ratio. According to Table 4, the highest value of aspect ratio corresponds to $N = 2$, while lowest value corresponds to $N = 16$, therefore Fig. 6 shows that is preferable to use many

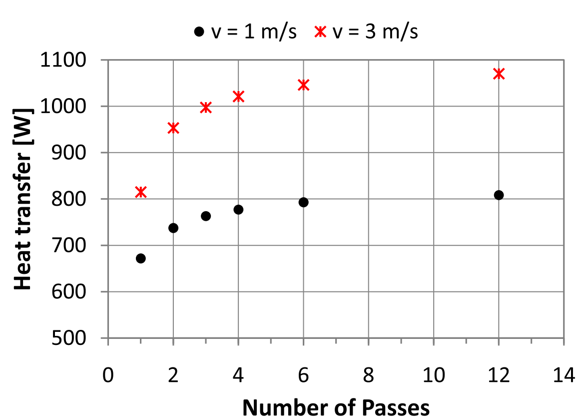

Fig. 2 – Heat transfer when the number of refrigerant passes is changed for two scenarios: air velocity of 3 m/s and 1 m/s.

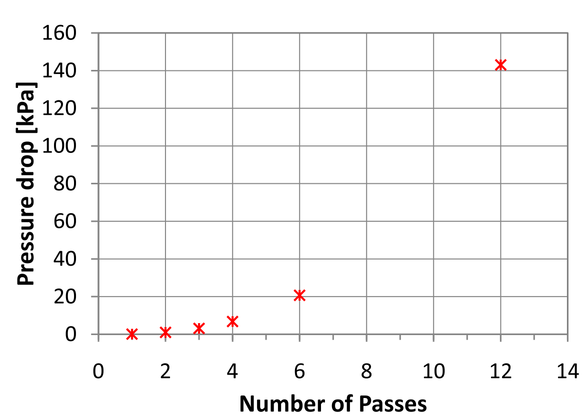

Fig. 3 – Refrigerant pressure drop along the heat exchanger when the number of refrigerant passes is changed.

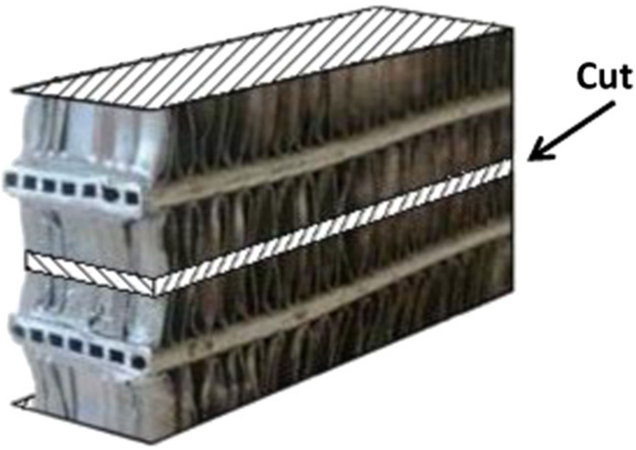

Fig. 4 – Schematic of the fin cut arrangement studied.

refrigerant passes with short heat exchanger lengths instead of a few passes with a long length, resulting in an asymptotic trend. An interesting observation is that the asymptote looks to be the capacity of the fin cut case. This fact means that the aspect ratio that maximises the heat transfer corresponds to the value that minimises the heat conduction between tubes.

The refrigerant cross-sectional area and the total length of the refrigerant path are the same for all the cases, therefore the pressure losses will change only because of the number of bends. However, note that these conclusions are not affected by pressure losses phenomena, because the analysed case is a gas cooler, where effect of pressure drop on heat transfer is negligible.

3. Numerical comparison of models

This section will discuss and compare the accuracy and computation time for two groups of models for MCHX.

The first group of models to be compared are those developed by the present authors for MCHX modelling: Fin2D, Fin1Dx3 and Fin1D. The reasons for developing the Fin1Dx3 model were to obtain suitable simulation times for designing purposes and to retain a similar accuracy as the Fin2D model (Martínez-Ballester et al., 2011). Therefore, the preliminary results presented in this section are orientated to assess the degree of accomplishment towards this end. Another way to reduce the computational cost is to reduce the number of cells employed in the discretization, which a priori means accuracy degradation. To this end the authors developed a model referred to as Fin1D, which applies the same assumptions as the Fin1Dx3 model but it discretizes the whole fin and air column of each segment into just one cell along the fin height direction.

The second group of models compared in this section consists of the models proposed in this paper (Fin1Dx3 and Fin1D) and a model which represents the approach that other authors (Lee and Domanski, 1997; Singh et al., 2008) apply in their models in order to consider heat conduction between tubes. In the introduction of Part I (Martínez-Ballester et al., 2012), it was explained that the majority of models available in the literature do not take into account heat conduction between tubes. With respect to finned tube heat exchangers, only a few authors (Lee and Domanski, 1997; Singh et al., 2008) model this phenomenon by using a correction term which takes into account, in a more or less artificial way, the heat conduction between tubes, despite using the adiabatic-fin-tip assumption in the governing equations.

Table 4 – Geometric variables in the aspect ratio study for a serpentine gas cooler.

Number of passes	Length (m)	Height (m)	Aspect ratio
2	1.15E + 00	2.02E − 02	5.70E + 01
4	5.76E − 01	4.04E − 02	1.43E + 01
6	3.84E − 01	6.06E − 02	6.34E + 00
8	2.88E − 01	8.08E − 02	3.56E + 00
10	2.30E − 01	1.01E − 01	2.28E + 00
12	1.92E − 01	1.21E − 01	1.58E + 00
14	1.65E − 01	1.41E − 01	1.16E + 00
16	1.44E − 01	1.62E − 01	8.91E − 01

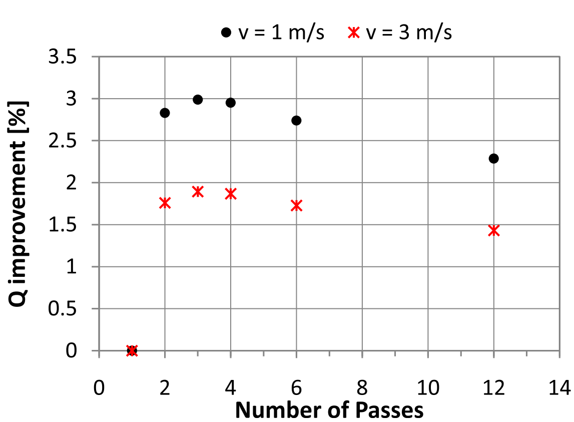

Fig. 5 – Improvement in heat transfer by cutting fins with regards to the same conditions but with a continuous fin for a different number of refrigerant passes and for two scenarios: air velocity of 1 m/s and 3 m/s.

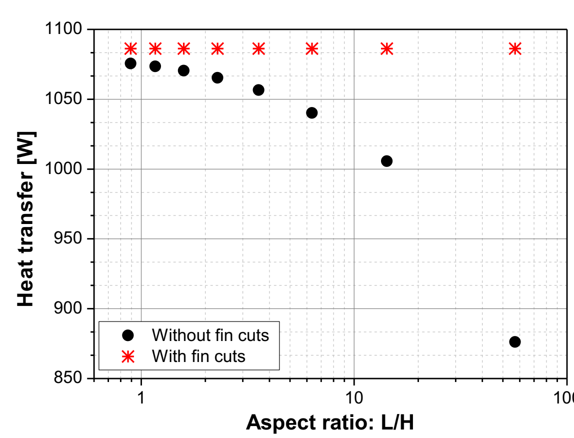

Fig. 6 – Heat transfer of the gas cooler when the aspect ratio is varied for two scenarios: continuous fin and fin with cuts.

The comparison of accuracy between the different models has been calculated by comparing the models with regard to the most detailed model, which depends on the scenario studied. Experimental results have not been used as a reference to perform the accuracy comparison, since the deviation between results is affected by several factors that are hard to identify, such as experimental uncertainty, moreover this deviation could be non-linear thus adding a complex factor in order to draw conclusions.

3.1. *Comparison among the different models developed: Fin2D, Fin1Dx3 and Fin1D*

Below are listed and briefly summarised each of the models compared in this subsection:

- Fin2D: Corresponds to the model presented by Martínez-Ballester et al. (2011). It is a very detailed model which discretizes fin and air into a two-dimensional grid. Its main capabilities are: it takes into account 2D longitudinal heat conduction (LHC) in both fin and tube wall, it does not apply fin theory and it accounts for heat conduction between tubes. Its main drawback is the simulation time employed to solve a case, due to the detailed grid adopted in the fin and air elements.
- Fin1Dx3: For each segment it discretizes air and fin into three cells along the direction between tubes, while for the tube wall it applies the same discretization as the Fin2D model. The phenomena modelled are the same as the Fin2D model, except the LHC in the fin along the air flow direction, with a large reduction in the number of cells employed.
- Fin1D: Basically is the same model as Fin1Dx3, and same phenomena are modelled, 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 the fin roots, is used to obtain the fin temperature profile. Thus it also takes into account heat conduction between tubes.

The geometry of the tubes and fins, of the case study in this section, are the same as used by Yin et al. (2001). The operating conditions for the simulations are those used for tests no. 9, 17, 25, 33 and 41 (Yin et al., 2001). All the models applied the same grid, with the exception of the fin and air cells in the Y direction. Due to the model differences, the Fin2D model needs a large number of these cells; Martínez-Ballester et al. (2011) proposed using 30 cells in the Y direction. The grids applied for these scenarios are: {5,1,3,30,3} for the Fin2D model and {5,1,3,3} for Fin1Dx3 and Fin1D models. The correlations used by the model are listed in Table 3.

The results of the accuracy comparison are presented in Fig. 7. The figure shows the deviation on predicted capacity for models Fin1D and Fin1Dx3 with respect to the predicted results of the Fin2D model. Therefore, the zero for the ordinates axis corresponds to the predicted results of the Fin2D model. The Fin2D model has been chosen as the reference because it is the most accurate, since it applies the finest discretization to the heat exchanger.

Fig. 7 shows that the deviation between the Fin2D and Fin1Dx3 models is at most 0.2%, which means that the predicted results could be considered the same. However, this deviation turns out to be as much as 2% in the case of the Fin1D model. The negligible difference between the Fin2D and Fin1Dx3 models means that longitudinal heat conduction in the fin surface along the air direction, which is not modelled in the Fin1Dx3 model, can be neglected for this scenario. These results also confirm that the approach of using three fin/air cells with a piecewise function for the fin temperature profile gives a good solution with a much lower computational cost.

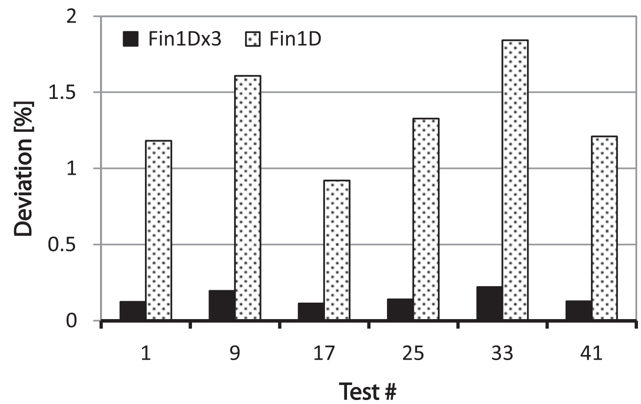

Fig. 7 – Heat transfer deviation, for different test conditions, of the Fin1Dx3 and Fin1D models with regard to the Fin2D model.

Fin1D and Fin1Dx3 take into account the same phenomena, and the differences between them are only due to the fin/air discretization. According to this, the deviation between the predicted results of both models is only a consequence of a more accurate application by the 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 in the Y direction. Nevertheless, this deviation can be interpreted as small, though the effect would depend on the operating conditions, heat exchanger and application. The present work analyses the case of a gas cooler, which corresponds to a case with an expected impact of these phenomena larger than for the case of a condenser. For an evaporator, dehumidification appears and plays an important role, and what occurs strongly depends on local properties, thus the authors foresee the inclusion of dehumidification in future work.

With regard to the computational cost, Fig. 8 presents the simulation time employed by each model to solve the several cases described above. In the figure, a large computing time reduction, from the Fin2D model to the Fin1Dx3 model, is noticeable. This reduction represents one order of magnitude. The main reason is the large difference in the number of air and fin cells used by both models. In the case of the Fin1Dx3 model, a piecewise function which consists of three one-dimensional functions is enough to accurately capture the actual fin temperature profile and consequently the heat transfer from fin to air. However, as explained in the introduction of Part I (Martínez-Ballester et al., 2012), Fin2D needs to apply a large discretization to the fin height, in practise 30 fin and air cells are required to get accurate results.

The simulation time reduction from the Fin1D to Fin1Dx3 model is not as drastic as in the Fin2D case, Fin1D needs half

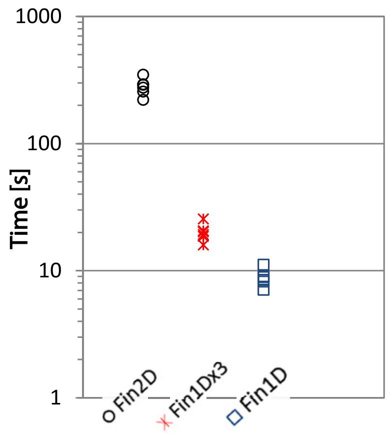

Fig. 8 – Comparison of the simulation time employed by each model.

the time spent by Fin1Dx3. A priori, a larger simulation time reduction could be expected since Fin1D uses just one air and fin cell along the fin height direction instead of three air and fin cells as in the Fin1Dx3 model. 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 root temperatures, since it takes into account heat conduction between tubes but only one air temperature value. Thus only two variables are saved in the Fin1D model with regard to the Fin1Dx3 model, which corresponds to the air temperature values. These temperature values are obtained in the same manner as undertaken by the Fin1D model, i.e. with an explicit calculation given the wall temperature field, so that in the Fin1Dx3 model there are only two more explicit calculations. 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.

If both factors of accuracy and computational cost are taken into account, the Fin2D model is not a cost effective solution since Fin1Dx3 provides the same results with a simulation time reduction of one order of magnitude. In contrast, Fin1Dx3 offers better results than Fin1D with only double the simulation time, thus it is considered by the authors as the best option for the modelling of this kind of heat exchanger.

3.2. *Comparison with other authors' approaches*

This subsection compares, in a similar way as in the previous subsection, the models proposed in the paper (Fin1D and Fin1Dx3) against other approaches used in the literature for heat exchanger modelling. To this end, it has been necessary to develop two new models:

- Fin1D_Cut: Reproduces the results of the most common models available in the literature (Corberán et al., 2002; Fronk and Garimella, 2011; García-Cascales et al., 2010; Jiang et al., 2006; Yin et al., 2001). It applies a segment-by-segment discretization, uses the adiabatic-fin-tip assumption and it does not take into account heat conduction between tubes. The model is the same as Fin1D but includes a cut along the fin to always reproduce the adiabatic-fin-tip assumption. The required changes in the model to include this fin cut are the same as those explained in the previous section, when the Fin1Dx3 model was modified to simulate an MCHX with fin cuts.
- Corrected-Fin: Based on the approaches proposed by Singh et al. (2008) and Lee and Domanski (1997). They have been chosen as references since they account for heat conduction between tubes in a different way to that proposed in the present paper, though these approaches model that phenomenon in a more artificial way. This model tries to be representative of what the referenced authors' models achieve. It is based on the Fin1D model and it applies the same discretization, but it uses the analytical solution given by fin theory when the adiabatic-fin-tip is assumed. In order to account for heat conduction in the same way as the referenced authors, correction terms are included in the corresponding energy conservation equations, which will be described in detail below.

The approaches of Singh et al. (2008) and Lee and Domanski (1997) were originally developed for fin-and-tube heat exchangers, but they have been adapted in this paper for an MCHX. Fig. 9 shows the geometric parameters of both arrangements regarding the heat conduction phenomenon.

These approaches (Singh et al., 2008; Lee and Domanski, 1997) apply the fin theory to each volume control and use fin efficiency to include the fin-to-air heat transfer, which is evaluated with Eq. (1), where $\theta_{fB,a}$ is the temperature difference between the bulk air temperature and the corresponding fin root temperature, and η_f is the fin efficiency. The relationship used for the evaluation of the fin efficiency corresponds to the case of adiabatic-fin-tip assumption, Eq. (2).

$$\dot{Q}_{f,a} = \eta_f \alpha_{f,a} A_{f,a} \theta_{fB,a} \quad (1)$$

$$\eta_f = \frac{\tan h(m_{f,a} H_f/2)}{m_{f,a} H_f/2} \quad (2)$$

$$m_{f,a}^2 = \frac{\alpha_{f,a} P_{f,a}}{k_f A_f}$$

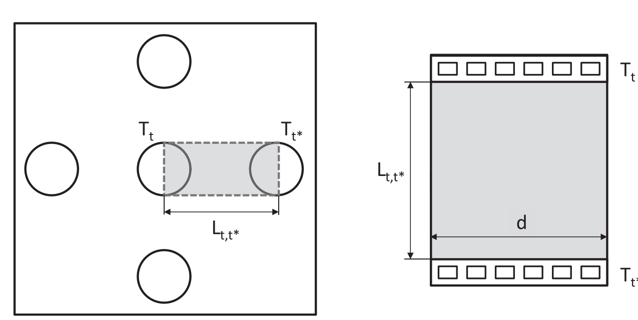

Fig. 9 – Analogy between a finned tube and an MCHX for the heat conduction resistance evaluation between two neighbouring tubes along the fin.

Eq. (3) establishes the energy conservation in a segment. The segment consists of the wall tube cell t, the corresponding fin wall cell f attached to the tube, and the fluids in contact with it: refrigerant cell r and air cell a.

$$\dot{Q}_{f,a} + \dot{Q}_{t,a} + \dot{Q}_{t,r} + \dot{Q}_{t,t^*} = 0 \tag{3}$$

Since the tubes have different temperatures, a correction term $\dot{Q}_{t,t^*}$ is introduced in Eq. (3) in order to take into account the heat conduction between tubes, which corresponds to the total heat transfer by conduction between neighbouring tubes. Fig. 9 shows 4 tubes t^* connected to a central tube t by the fin surface. For this example the total heat conduction between central and neighbouring tubes can be modelled as Eq. (4).

$$\dot{Q}_{t,t^*} = \sum_{t^*} \lambda \left(\frac{T_t - T_{t^*}}{R_{t,t^*}} \right) \tag{4}$$

Different approaches could be applied to obtain the value of thermal resistance R_{t,t^*} together with the use of λ, which is a multiplier that can be used to adjust the heat conduction term. Singh et al. (2008) explain that this multiplier has to be adjusted either numerically or experimentally, which depends on the heat exchanger simulated. The need to use this correction factor which a priory is unknown and its dependency on the modelled case are the main drawbacks of this methodology.

The Corrected-Fin model evaluates R_{t,t^*} with Eq. (5) and applies $\lambda = 1$.

$$R_{t,t*} = \frac{L_{t,t*}}{t_f \, d \, k_f} \tag{5}$$

The simulations were carried out for the gas cooler (Yin et al., 2001) that was validated in Part I (Martínez-Ballester et al., 2012). The operating conditions for the simulations are those used for tests no. 9, 17, 25, 33 and 41 by Yin et al. (2001). The correlations for heat transfer and pressure losses coefficients were also the same as described in the previous subsection.

All the cases analysed have tubes with different temperatures, and heat conduction is present, therefore Fin1D will be more accurate than Fin1D_Cut since the adiabatic-fin-tip assumption is not valid. The Fin1D model should also be more accurate than Corrected-Fin because the latter applies a correction term to take into account heat conduction between tubes, while Fin1D implicitly takes into account the heat conduction without simplifying assumptions.

The first study compares the models that apply the same level of discretization, i.e. the fin is discretized in just one cell. Fig. 10 shows the deviation on predicted capacity for models Fin1D_Cut and Corrected-Fin with regard to the predicted results of the Fin1D model, which is expected to be the most accurate. First, it is noticeable that deviations between these models for these conditions are quite small, which means that the adiabatic-fin-tip assumption despite not being valid does not have a large impact on the solution, it being less than 0.8%. The deviation between Fin1D_Cut and Fin1D is always positive, which implies that by cutting the fins, heat transfer is always increased. As can be observed in Fig. 10, Corrected-Fin can take into account heat conduction between tubes with negligible deviations, which means that the approaches of Singh et al. (2008) and Lee and Domanski (1997) are good alternatives for the modelling of finned tube heat exchangers in the presence of heat conduction between neighbouring tubes.

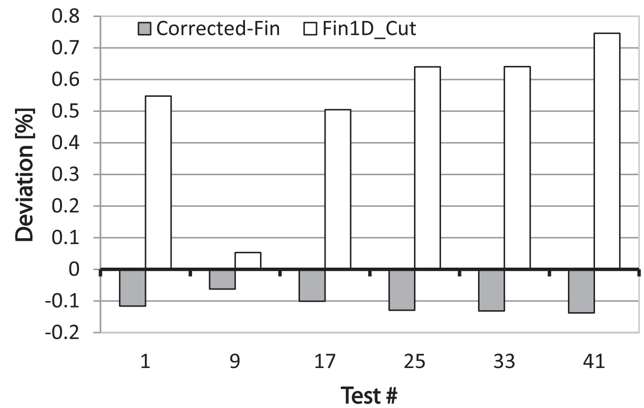

Fig. 10 – Deviation of predicted heat transfer of Fin1D_Cut and Corrected-Fin models with regard to Fin1D for different test conditions.

The following study compares the models that apply the same level of discretization (Fin1D, Fin1D_Cut and Corrected-Fin) as the Fin1Dx3 model, which uses a more detailed discretization, resulting in it being the most accurate. Fig. 11 presents the deviation in capacity for models Fin1D, Fin1D_Cut and Corrected-Fin with regard to the predicted results of the Fin1Dx3 model. Therefore, the zero for the ordinates axis corresponds to the predicted results of the Fin1Dx3 model.

As can be observed in Fig. 11, the accuracy of all the models is good, only resulting in errors as much as 2% with respect to the Fin1Dx3 model. The largest deviation is produced by the Fin1D_Cut model which uses the adiabatic-fin-tip. The Fin1D and Corrected-Fin models have a similar deviation, ranging from 1% to 2%. This deviation indicates that the major impact in the prediction error is the consideration of the air as being mixed in the direction between tubes; in fact this is the only difference between the Fin1D and Fin1Dx3 models; Fin1Dx3

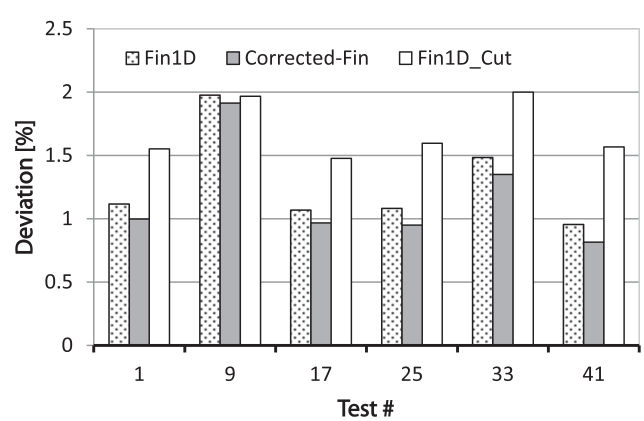

Fig. 11 – Deviation of predicted heat transfer of Fin1D, Fin1D_Cut and Corrected-Fin models with regard to Fin1Dx3 for different test conditions.

discretizes the fin height into 3 cells that are able to account for non-mixed air along the fin height.

Regarding computational cost, it is not necessary to perform an evaluation of the simulation time required by the Fin1D_Cut and Corrected-Fin models because they apply the same discretization as Fin1D and therefore have the same computational cost, which was presented above (Fig. 8).

4. Conclusions

The present work analyzes the impact of some design parameters of MCHXs on its performance. These parameters were the number of refrigerant passes, aspect ratio and the effect of fin cuts. The success of simulation tools to this end depends on the assumptions of the model, i.e. some parameters produce effects due to phenomena not taken into account by the model. For instance, a model that does not account for heat conduction between tubes cannot study the effect of the aspect ratio; otherwise, the results would be always the same.

The numerical studies presented were carried out using a model for the MCHXs that uses the novel approach Fin1Dx3, presented in Part I (Martínez-Ballester et al., 2012), which takes into account: heat conduction between tubes, fin cut or continuous fin, detailed air discretization, 2D longitudinal heat conduction along the tube and the effects of non-mixed air in the Y direction.

For a gas cooler working with CO_2 under transcritical pressures, the main conclusions of the simulation studies were:

- For a gas cooler where no phase change occurs, heat transfer is always increased by increasing the number of refrigerant passes, regardless of the increase in pressure drop.
- The fin cuts always increase the heat transfer. In the gas cooler analysed, the improvement with regard to the continuous fin depends on the air velocity and number of refrigerant passes: the lower the velocity the greater the improvement in capacity. There is an optimum value for the number of refrigerant passes, regardless of air velocity, which is 3 passes for the case analysed. The improvement in heat transfer was as much as 3%.
- Regarding the aspect ratio of a serpentine heat exchanger, given a heat transfer area and a face area, the best aspect ratio corresponds to a gas cooler with a reduced length (L) and large height (H). The reason is based on the fact that this configuration reduces heat conduction between tubes.

Numerical studies on the accuracy and computational cost were presented in order to compare the proposed models of Fin1D and Fin1Dx3 with regard to the authors' previous models and other representative models from the literature. The main conclusions of these comparisons were the following:

- The solution time of Fin1Dx3 has been reduced by one order of magnitude with regard to Fin2D, whereas the differences in the results are less than 0.3%, which are considered negligible for practical applications. The computation time difference between Fin1Dx3 and Fin1D was determined to be double.
- Corrected-Fin can lead to accurate results when compared with a model with an equivalent approach that models heat conduction between the tubes in a more fundamental way, such as Fin1Dx3. The difference between the predicted results from both models was between 1% and 2%. The computational costs of the Fin1D and Corrected-Fin models are the same.
- Nevertheless, the authors would like to emphasise the fact that the present work shows no computational saving or advantage in accuracy by adding correction terms to an approach that uses adiabatic-fin-tip efficiency rather than a more fundamental approach, regarding the phenomena of heat conduction between tubes, like Fin1D.
- By comparison of deviations between the Fin1D, Corrected-Fin and Fin1Dx3 models, it was concluded that the main factor responsible for the differences between them was the effect of non-mixed air along the fin height.

Acknowledgements

The first author's work on this project was partially supported by the Ministry for Education of Spain, under the Training of University Professors program (FPU). Financial support from the Ministry for Education of Spain, project numbers: DPI2008-06707-C02-01 and DPI2011-26771-C02-01, is also gratefully acknowledged. Santiago Martínez-Ballester gratefully acknowledges the National Institute of Standards and Technology where this work was partially developed and funded, and especially Dr. Piotr A. Domanski for hosting Santiago at NIST as well as for giving his valuable time and advice.

REFERENCE

Asinari, P., Cecchinato, L., Fornasieri, E., 2004. Effects of thermal conduction in microchannel gas coolers for carbon dioxide. Int. J. Refrigeration 27 (6), 577–586.

Churchill, S.W., 1977. Friction-factor equation spans all fluid flow regimes. Chem. Eng. 7, 91–92.

CoilDesigner, 2010. Tool to Aid in the Design, Simulation and Optimization of Air-cooled Heat Exchangers. University of Maryland, Center for Environmental Energy Engineering, MD, USA. http://ceee.umd.edu/isoc/software/index_coildesigner.htm.

Corberán, J.M., Gonzálvez, J., Montes, P., Blasco, R., 2002. 'ART' a computer code to assist the design of refrigeration and A/C Equipment. International Refrigeration and Air Conditioning Conference at Purdue, IN, USA.

Domanski, P.A., Choi, J.M., Payne, W.V., 2007. Longitudinal Heat Conduction in Finned-Tube Evaporator. 22nd IIR International Congress of Refrigeration, Beijing, China.

EVAP-COND, 2010. Simulation Models for Finned Tube Heat Exchangers with Circuitry Optimization, National Institute of Standards and Technology. Building and Fire Research Laboratory, Gaithersburg, MD, USA. http://www.nist.gov/el/building_environment/evapcond_software.cfm.

Fronk, B.M., Garimella, S., 2011. Water-coupled carbon dioxide microchannel gas cooler for heat pump water heaters: part II – model development and validation. Int. J. Refrigeration 34, 17–28.

García-Cascales, J.R., Vera-García, F., Gonzálvez-Maciá, J., Corberán-Salvador, J.M., Johnson, M.W., Kohler, G.T., 2010. Compact heat exchangers modeling: condensation. Int. J. Refrigeration 33, 135–147.

Gnielinski, V., 1976. New equations for heat and mass transfer in turbulent pipe and channel flow. Int. Chem. Eng. 16 (2), 359–368.

IEC 60335-1, 2010. International Standard: Household and Similar Electrical Appliances - Safety –Part 1: General Requirements. International Electrotechnical Commission.

Jiang, H.B., Aute, V., Radermacher, R., 2006. CoilDesigner: a general-purpose simulation and design tool for air-to-refrigerant heat exchangers. Int. J. Refrigeration 29 (4), 601–610.

Kays, W.M., London, A.L., 1984. Compact Heat Exchangers, third ed.. McGraw-Hill, New York.

Kim, M.-H., Bullard, C., 2002. Air-side thermal hydraulic performance of multi-louvered fin aluminum heat exchangers. Int. J. Refrigeration 25, 390–400.

Lee, J., Domanski, P.A., 1997. Impact of Air and Refrigerant Maldistributions on the Performance of Finned-tube Evaporators with R-22 and R-407C. Report No.: DOE/CE/23810-81.

Martínez-Ballester, S., Corberán, J.M., Gonzálvez-Macia, J., 2012. Numerical model for microchannel condensers and gas coolers: part i – model description and validation. Int. J. Refrigeration 36 (1), 173–190.

Martínez-Ballester, S., Corberán, J.M., Gonzálvez-Macia, J., Domanski, P.A., 2011. Impact of classical assumptions in modelling a microchannel gas cooler. Int. J. Refrigeration 34, 1898–1910.

MPower, 2010. Modine's Custom Vapor Compression System Design. Modine Manufacturing Company/the Universitat Politècnica de València, Racine, WI, USA/Spain. http://www.modine.com/v2portal/page/portal/hvac/hvacCoolingCoilsDefault/hvac_com/cooling_coils/level_3_content2_040.htm.

Park, C.Y., Hrnjak, P., 2007. Effect of heat conduction through the fins of a microchannel serpentine gas cooler of transcritical CO_2 system. Int. J. Refrigeration 30 (3), 389–397.

Shao, L.L., Yang, L., Zhang, C.L., Gu, B., 2009. Numerical modeling of serpentine microchannel condensers. Int. J. Refrigeration 32 (6), 1162–1172.

Singh, V., Aute, V., Radermacher, R., 2008. Numerical approach for modeling air-to-refrigerant fin-and-tube heat exchanger with tube-to-tube heat transfer. Int. J. Refrigeration 31 (8), 1414–1425.

Singh, V., Aute, V., Radermacher, R., 2010. Investigation of effect of cut fins on carbon dioxide gas cooler performance. HVAC&R Res. 16 (4), 513–527.

Yin, J.M., Bullard, C.W., Hrnjak, P.S., 2001. R-744 gas cooler model development and validation. Int. J. Refrigeration 24 (7), 692–701.

Zhao, Y., Ohadi, M.M., Radermacher, R., 2001. Microchannel Heat Exchangers with Carbon Dioxide. Report No.: ARTI-21CR/10020-01.