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Research Publications

IMPACT OF FIN CUTS AND REFRIGERANT LAYOUT ON THE PERFORMANCE OF A MICROCHANNEL GAS COOLER WORKING WITH TRANSCRITICAL CO_2

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ABSTRACT

Microchannel heat exchangers (MCHX) play an important role in topics about charge reduction and transcritical cycles due to its high compactness and high mechanical strength. From a designer's point of view, given an air-side area and faced area, there are many options to design the refrigerant circuitry and aspect ratio of a MCHX. The paper presents numerical studies about the influence on the MCHX's performance of these design parameters for a gas cooler working with CO_2 under transcritical pressures. A technique to improve the MCHX effectiveness is to cut the fins along the middle section between neighboring tubes. Following same methodology as was used in previous study, the paper analyzes the improvement by cutting fins for different gas cooler layouts. A key point of the paper is that the parameters analyzed can only be assessed by models which take into account heat conduction between tubes otherwise their effects would be hidden. Results illustrate some general design recommendations in order to maximize effectiveness of gas coolers working in similar conditions. Improvement on gas cooler effectiveness by cutting fins depends on the gas cooler layout, ranging from an optimum value for a specific configuration to negligible for other configurations.

1. INTRODUCTION

Currently, an increasing interest on microchannel heat exchangers (MCHXs) is arising in refrigeration and air conditioning applications due to their high compactness and high effectiveness. The high effectiveness is consequence of large heat transfer coefficients as result of using small hydraulic diameters. Given an air side heat transfer area, high compactness means reduced volumes what will result in light heat exchangers and low refrigerant charges.

Getting low refrigerant charges plays an important role on the use of natural refrigerants which are flammable like propane. Natural refrigerants are considered as more environmentally friendly than others refrigerants commonly used, with a similar or even better performance. The main drawback of working with some natural refrigerants is that they are dangerous in large quantities: ammonia is toxic and propane is highly flammable, in fact IEC 60335-1 restricts the amount of a hydrocarbon that can be used in a system to 150 g. To this end, a suitable heat exchanger design is a serpentine MCHX. This kind of heat exchangers minimizes the refrigerant charge because it has no headers, thus saving all this volume and the corresponding refrigerant charge. In the case of transcritical CO_2 systems, microchannels have an additional merit related to their high mechanical strength.

Along the design process of a MCHX, geometric data of tubes and fins are usually imposed by manufacturer. Fin pitch, heat transfer area and face area of a MCHX is usually obtained attending to performance requirements. However, multiple choices exist for the number of refrigerant passes, refrigerant connections and aspect ratio (L/H) of the MCHX. In fact, some simulation software like EVAP-COND (2003) has the capability to optimize the heat load by 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. Since the circuitry has an important influence on the heat exchanger performance, the usefulness of simulation software for this purpose is clearly justified, since the optimization via experimentation takes too long, it is difficult and expensive.

On the other hand, depending on the model's assumptions some parameter can be studied or not. Most of models for air-to-refrigerant heat exchangers apply the fin theory by using the analytical solution for adiabatic-fin-tip assumption. In this way, these models do not account for heat conduction between tubes. Performance results of these approaches do not depend on the parameters studied in the present paper. For instance, the impact of the aspect ratio (L/H) on the heat transfer of a heat exchanger would be null if it is evaluated with such models. This effect can only be assessed if the model adequately accounts for the heat conduction between tubes.

The authors presented a detailed model referred to as Fin2D model (Martínez-Ballester et al., 2011a) in order to analyze the influence of these classical assumptions in a microchannel gas cooler. In the air-side the heat transfer was modeled without applying any of these assumptions but it required a detailed discretization and large computational cost. The conclusions of that work allowed continuing working in a model which could retain only the most important effects but with an interesting ratio between accuracy and computational cost: Fin1Dx3 (Martínez-Ballester et al., 2011b). The main difference of this approach is the discretization applied to the air and fin elements. This approach reduces one order of magnitude the simulation time with regard the Fin2D model and predicts same results. Fin1Dx3 model is the model employed to carry out the numerical studies presented in this paper.

Cutting the fins between tubes for air-to-refrigerant heat exchangers is an improvement studied in literature. Cutting the fins avoids the heat conduction between tubes along the fins, which degrades the heat exchanger effectiveness. Heat conduction between tubes along fins appears when a temperature difference between tubes exists. 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 23 % reduction in finned-tube evaporator capacity when different exit superheats were imposed on individual refrigerant circuits. However, not so large improvements have been achieved for MCHXs, namely: Asinari et al. (2004) concluded that the impact of this heat conduction can be assumed as negligible in 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.

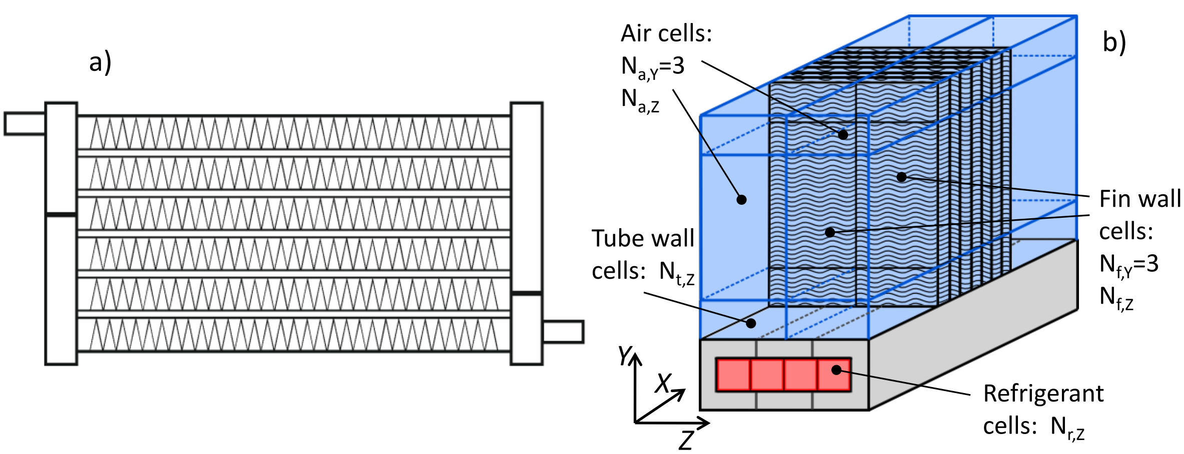

Figure 1. (a) Microchannel heat exchanger with three refrigerant passes. (b) Schematic of a heat exchanger segment.

Numerical studies of the parameters analyzed in this paper, for a MCHX, are hardly available in the literature. The goal of the selected case studies is contributing to a better understanding of the influence of some design parameters on transcritical CO_2 microchannel gas coolers performance.

2. MODEL DESCRIPTION

The model used for the present work applies the Fin1Dx3 approach (Martínez-Ballester et al., 2011b). The 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. Fig. 1(a) shows a MCHX that can be simulated with this model which is used as example to explain the discretization applied by the model.

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 Fig. 1(b). Each segment consists of: a 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. In this way, it allows to capture the most significant air temperature values: the air temperature close to each tube and the bulk air temperature. Since the discretization for the air and fin wall is the same, $N_{fw}= N_{a.}$

The Fin1Dx3 approach applies a novel methodology to model the air side heat transfer. Fin1Dx3 adopts a composed function for the fin wall temperature which applies the fin theory for each fin-to-air connection by using the analytical solution of the fin theory for the case of given different fin roots temperature. Thus heat conduction between tubes along the fin is taken into account. Finally, the specific discretization in the air side captures the unmixed air (along *Y* direction) influence in the air-to-tube heat transfer evaluation. For more details regarding governing equations and its discretization, reader is referred to Martínez-Ballester et al. (2011b).

The main advantage of the proposed approach is that the heat conduction between tubes is modeled in a more fundamental way than rest of current models, which apply more or less artificial correction terms to account for this phenomenon. A second advantage is that, by following the proposed discretization of equations, the simulation time only is increased to the double with regard to the simplest 1D discretization.

3. SIMULATION STUDIES

3.1 Case Study

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 working fluid in transcritical conditions. This gas cooler consists of 34 tubes with 3 refrigerant passes.

Table 1. Geometric characteristics of gas cooler (based on data of Yin et al., 2001).

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	Louvered	Fin pitch (mm)	1.15
Number of ports	11	Port diameter (mm)	0.79
Wall thickness (mm)	0.43	Fin height (mm)	8.89
Fin thickness (mm)	0.1		

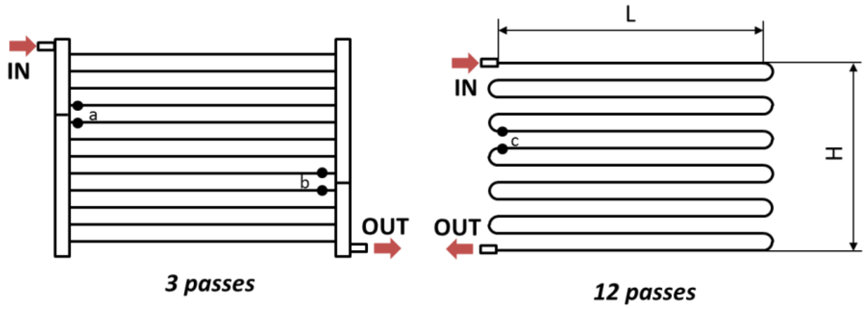

Figure 2 – Schematics of two gas cooler arrangements studied: 3 and 12 refrigerant passes.

The number of refrigerant passes is a parameter to be studied, from one pass up to the limit that corresponds to a serpentine gas cooler. Increasing number of refrigerant passes leads to larger velocities of the refrigerant flow. This fact, besides the increase of refrigerant path length in a serpentine, produces a much larger pressure drop. The limit case (serpentine MCHX) would be, for this reason, of no practical use. Therefore, 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 the 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 tubes number. The resulting geometric data is shown in Table 1. The rest of geometric data for fins and tubes are the same as Yin et al. (2001) tested. For all scenarios the refrigerant and air-side areas, face area and 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, conditions of test nº 2 from Yin et al. (2001) have been chosen. Both the CO_2 mass flow rate and air flow rate have been modified in order to get the same mass velocities as the original values according to the new geometry. The operating conditions are listed in Table 2. Regarding the air, there are two scenarios: with the mass flow rate given in Table 2, and with this mass flow rate divided by three.

3.2 Number of Refrigerant Passes

The number of refrigerant passes is varied from 1 pass up to the maximum possible number; 12 passes which corresponds to a serpentine configuration. Fig. 2 depicts two samples of the cases studied. The performance differences will be only due to the number of passes since refrigerant area, air-side area, face area and rest of the geometry do not change.

Table 2. Operating conditions: based on Test n° 2 (Yin et al., 2001).

	Inlet Pressure (kPa)	Pressure drop (kPa)	Inlet temperature (°C)	Outlet temperature (°C)	Mass flowrate (g/s)
CO_2	10792	421.6	138.6	48.2	5.64
Air	100	$61\ 10^{-3}$	43.5	-	87.3

Fig. 3 shows the results of this study for two different values of the air velocity. As the air velocity is increased, the heat transfer is also increased for all cases 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 area is reduced so that the refrigerant velocity rises to keep constant the mass flow rate, and it improves the heat transfer coefficient. Thus, for this case study, the figure depicts clearly that the heat transfer is always raised, with an asymptotic trend, by increasing the number of passes.

It should be noticed that the case study corresponds to a gas cooler, which does not undergo a phase change. In a condenser, the pressure drop leads a temperature drop during the phase change, therefore the temperature difference between air and refrigerant would decrease and the heat transfer would be reduced. In this way, in condensers/evaporators there is an optimum on the heat transfer when the number of refrigerant passes is studied according to the opposite influence on the heat transfer and pressures drop. This conclusion was also exposed by Shao et al. (2009) in their studies for a serpentine microchannel condenser, where they studied the influence of the number of passes on heat transfer.

3.3 Influence of Cutting the Fins

A technique to improve the effectiveness in air-to-refrigerant heat exchangers is by cutting the fins. The heat conduction between tubes, due to temperature differences from bottom to top fin roots, degrades the heat exchanger effectiveness. By cutting the fins, this heat conduction is avoided. This technique is indicated for heat exchangers which have large temperature differences between tubes. For example, in a condenser there are tubes with superheated vapor flowing inside which are connected through fins to other tube with saturated vapor 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 all the gas cooler length, since there is no phase change. Thus the temperature difference between two neighboring tubes can be as large as 50 K.

There only exist few models that take into account the heat conduction between tubes. The rest of models always overpredict the heat transfer for the same conditions since they do not account for the effectiveness degradation caused by the heat conduction.

In the present study the fin cuts studied are disposed along the middle section between two neighbour tubes for all the fins of the heat exchanger. The Fin1Dx3 model is developed for a continuous fin, but can be slightly modified to incorporate a cut in the section at half the fin height. To the authors' knowledge there are no studies for MCHXs about the influence of the refrigerant circuitry on the impact of fin cuts in the predicted results. To this end, the impact of cutting the fins has been evaluated for the same refrigerant passes studied in previous subsection.

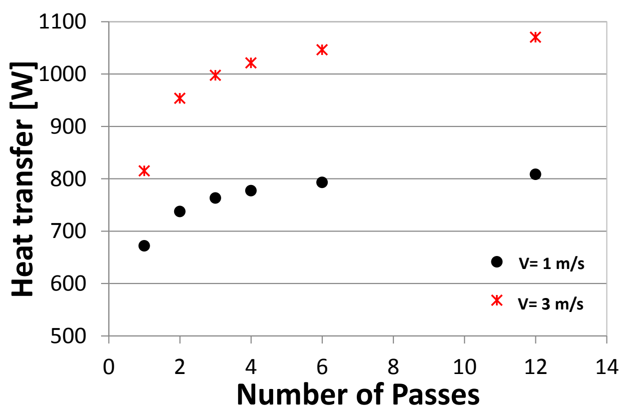

Figure 3 – Heat transfer when number of refrigerant passes is changed in two scenarios: air velocity of 3 m/s and 1 m/s.

The results are shown in Fig. 4, where it has been plotted the heat transfer improvement by cutting fins with respect to the solution without fin cuts. The heat improvement for one pass is zero because in a one pass arrangement all tubes have the same temperature evolution resulting in a negligible 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. Fig. 4 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.

If we analyze the plot for v=1 m/s, when the number of passes is different from one, there is always an improvement on the heat transfer by cutting the fins, and in this case there is a maximum value for 3 passes, regardless the air velocity. 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. 2 illustrates this explanation, where the heat exchanger with 3 passes has two zones with large temperature difference, regions “a” and “b”. The serpentine heat exchanger has a similar temperature difference between all the tubes, which can be represented by the temperature difference at zone “c”. Heat exchanger with 3 passes will have only two zones with temperature difference, but the temperature difference between bottom and top of zones “a” or “b” is much higher than corresponding value for region “c” of the serpentine case, though serpentine MCHX has 11 regions with a similar temperature difference to the “c” zone. These opposite effects could be one of the reasons to explain the presence of a maximum in the heat improvement depicted in Fig. 4.

3.4 Influence of aspect ratio for a serpentine gas cooler

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

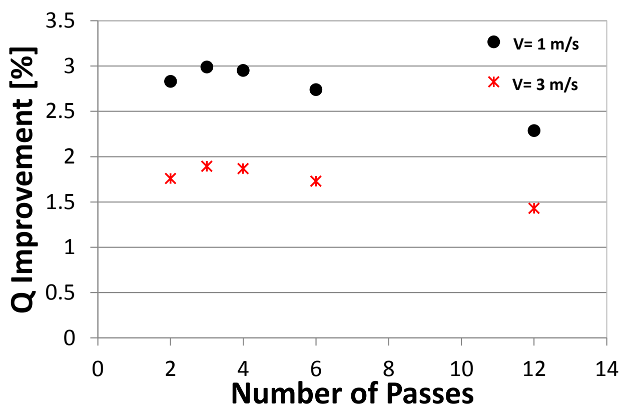

Figure 4–Improvement of heat transfer by cutting fins with respect to continuous fin for different number of refrigerant passes and for two scenarios: air velocities of 1 m/s and 3 m/s.

A restriction for these studies is that air-side and face area are constants while aspect ratio (*L/H*) changes. By observation of the serpentine MCHX design, it is deducible that the air-side heat transfer area is proportional to the product: *N L*. Therefore, to study the isolated effect of the aspect ratio on the performance, *N L* will have to remain unchanged for all the cases studied.

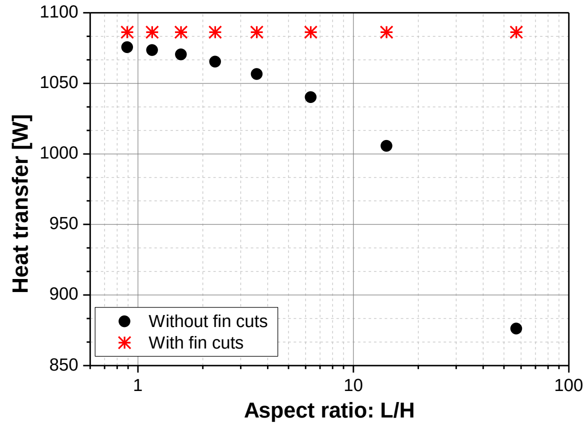

Figure 5 – Heat transfer of the gas cooler when the aspect ratio is varied for two modeling scenarios: continuous fin and fin with cuts.

Fig. 5 shows the results for the predicted heat transfer as function of the aspect ratio. The figure shows the results for the two analyzed cases: the fin is cut and a continuous fin. The figure shows that aspect ratio has no effect on heat transfer when the fin is cut, thus models that apply adiabatic-fin-tip assumption will not be able to study this influence since results are always the same.

For the case of continuous fin, Fig. 5 shows that heat transfer has a strong dependence on the gas cooler aspect ratio. It shows that is preferable to use a short heat exchanger length (large number of passes) instead of a large length (few passes). For the case analyzed, the highest value of aspect ratio corresponds to N=2 while lowest value correspond to N=16. An interesting observation is that the asymptote looks to be the capacity for the fin cut case. This fact means that the aspect ratio which maximizes the heat transfer corresponds to the value that minimizes the heat conduction between tubes.

In a one-phase flow and for the conditions studied, the heat transfer and pressure drop phenomena can be decoupled. Thus previous conclusions are not affected by pressure losses phenomena. Furthermore, frictional term of pressure drop would be rather the same for all cases since the refrigerant cross area and the total length are always the same, though not the pressure drop due to singularities (return bends and sudden contraction/expansion in headers). In case of a two-phase flow, pressure drop due to singularities (return bends and sudden contraction/expansion in headers) could play an important role since for this situation the pressure drop and heat transfer phenomena are coupled.

4. CONCLUSIONS

MCHX minimizes the impact of some disadvantages of some natural refrigerants: flammability and high pressures due to its high compactness, reduced volumes and high strength.

From the designer's point of view, some parameters of MCHX such as number of refrigerant passes, aspect ratio and effect of fin cuts, are hard and expensive to determine by experimentation. On the other hand, success of simulation tools to this end depends on the model's assumptions, i.e. some parameters produce effects that are due to some phenomena which is not taken into account by the model.

The proposed numerical studies were carried out by using a model for MCHX which uses a novel approach (Fin1Dx3) that takes into account: heat conduction between tubes, fin cut or continuous fin and effects of no-mixed air along Y direction.

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

- For a gas cooler where no phase change occurs, heat transfer is always increased by increasing the number of refrigerant passes regardless the increase of pressure drop. However, this increase of pressure drop for a compression vapor system would penalize the power of compressor.

- The fin cuts always increase the heat transfer. In the gas cooler analyzed, 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 capacity improvement. Regarding the number of refrigerant passes, there exists an optimum value regardless the air velocity, which is 3 passes for the analyzed case. Anyway, the improvement is as much as 3%.

- When the aspect ratio is analyzed, given the face and heat transfer areas, the best aspect ratio corresponds to a gas cooler with reduced length (*L*) and large height (*H*). The reason is based on the fact that this configuration reduces the heat conduction between tubes. For a gas cooler, effect of pressure drop on heat transfer can be neglected for the conditions studied.

ACKNOWLEDGEMENTS

Santiago Martínez-Ballester's work on this project was partially supported by Ministry for Education of Spain, under the training for university professors program (FPU). Financial support from Ministry for Education of Spain, project number: DPI2008-06707-C02-01, is also gratefully acknowledged.

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