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

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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

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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,