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Numerical model for microchannel condensers and gas coolers: Part I – Model description and validation

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ABSTRACT

The present work presents a model (Fin1Dx3) for air-to-refrigerant microchannel condensers and gas coolers, with any refrigerant circuitry. The model applies a segment-by-segment discretization to the heat exchanger, adding in each segment a novel bi-dimensional discretization to the fluids flow, fin and tube wall. Fin1Dx3 introduces a new approach to model the air-side heat transfer by using a composed function for the fin wall temperature, which allows to take into account more fundamentally the heat conduction between tubes. The proposed model accounts for: 2D longitudinal heat conduction in the tube wall, the heat conduction between tubes along the fin, and the unmixed air influence on performance. The paper presents the heat exchanger discretization, the governing equations, the numerical scheme employed to discretize equations and the solving methodology. The model has been validated against experimental data for both a condenser and a gas cooler, resulting in predicted capacity errors within ±5%.

Modèle numérique pour les condenseurs à microcanaux et les refroidisseurs de gaz : Partie I – Description et validation du modèle

Mots clés : Condenseur ; Modèle ; Conductivité thermique ; Microcanal ; Ailette ; Refroidisseur à gaz

1. Introduction

The use of microchannel heat exchangers (MCHXs) 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 us to save lots of costs and time in the laboratory working with expensive test benches. Currently, several models or simulation tools for heat exchangers are available in the literature: for finned tubes (CoilDesigner, 2010; Corberán et al., 2002; EVAP-COND, 2010; IMST-ART, 2010; Jiang

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Nomenclature

A	heat transfer area (m^2)
Ac	cross-sectional area (m^2)
Cp	specific heat ($J\ kg^{-1}\ K^{-1}$)
D	hydraulic diameter (m)
f	Darcy–Weisbach friction factor
g	gravitational constant ($m\ s^{-2}$)
H	height (m)
h	specific enthalpy ($J\ kg^{-1}$)
k	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
l	distance between two wall cells (m)
LHC	longitudinal heat conduction
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
N	number of cells
N_s	number of segments
NTU	number of transfer units
P	wetted perimeter (m)
p	pressure (Pa)
$\dot{q}$	heat flux ($W\ m^{-2}$)
s	length in the forward direction of a fluid (m)
T	temperature (K)
t	thickness (m)
U	overall heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
V	volume (m^3)
X, Y, Z	spatial coordinates (m)

Greek symbols

α	convective heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
β	tube orientation (deg)
ε	heat exchanger effectiveness
φ	fin height ratio
θ	temperature difference (K)
ρ	density ($kg\ m^{-3}$)

Subscript

a	air, air cell index
acc	acceleration
cont	contraction
exp	expansion
f	fin, fin cell index
fB	fin cell at bottom
fr	friction
fT	fin cell at top
g	gravitational
i	fluid cell index
in	inlet
j	matrix column index
k	tube direction index
N, S, W, E, J_B, J_T	directions of neighbour wall cell
out	outlet
r	refrigerant, refrigerant, cell index
t	tube, tube cell index
X, Y, Z	spatial coordinates directions

et al., 2006; Lee and Domanski, 1997; Singh et al., 2008) and microchannel heat exchangers (Asinari, 2004; Fronk and Garimella, 2011; García-Cascales et al., 2010; Shao et al., 2009; Yin et al., 2001). Some of them (Asinari, 2004; CoilDesigner, 2010; Corberán et al., 2002; IMST-ART, 2010; Shao et al., 2009; Singh et al., 2008; Yin et al., 2001) apply energy conservation equations to each control volume, while others (Fronk and Garimella, 2011; García-Cascales et al., 2010; Jiang et al., 2006; Lee and Domanski, 1997; EVAP-COND, 2010) apply directly the solution given by the ε-NTU methodology. The main difference between the two methodologies is that the ε-NTU model has several implicit assumptions resulting in less freedom to describe the actual processes. Despite this fact, the models that do not apply the ε-NTU methodology usually make the same assumptions for the thermal problem as those used by ε-NTU based models, of which the most important for the aim of this paper are the following:

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

Some of these assumptions have been studied in the literature for some heat exchanger topologies, such as finned tubes heat exchangers, whilst the effects of these assumptions are not studied so extensively for parallel tubes and serpentine MCHXs. These heat exchangers have a different thermal behaviour since the thermal and geometric conditions are different. Thus, it is interesting to evaluate the impact of the classical assumptions, which were explained previously, on the model results. Martínez-Ballester et al. (2011) performed a literature review in which the influence of all these effects was investigated theoretically and experimentally for both finned tubes and MCHXs.

The main motivation for this work is based on the drawbacks that, in the authors' opinion, existing models have when they are applied to some recent designs of heat exchanger, such as serpentine and parallel tubes MCHXs.

Firstly, Martínez-Ballester et al. (2011) proposed a model for a microchannel gas cooler referred to as the 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 modelling assumptions: the model accounts for 2D LHC in the tube and fin wall; it does not use any fin efficiency so it can model consistently the heat conduction between tubes. Since it applies a 2D discretization for the air in each segment, it does a more accurate integration of the heat transferred from the fin to the air, since the air temperature is more uniform in a cell. In contrast, most of the models apply the fin theory that assumes intrinsically uniform air temperature along the fin height. Furthermore, the Fin2D model allows independent discretization to be applied for refrigerant and air. This fact is interesting to capture the variation in air properties along the air flow direction.

The aim of developing the Fin2D model was to evaluate the prediction errors of the classical modelling assumptions and techniques described above, in an equivalent piece of