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Evaluation of microchannel condenser characteristics by numerical simulation

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ABSTRACT

This paper presents the development of a finite-volume-based numerical condenser model that considers important factors such as non-uniform air temperature and velocity at the front, fin conduction, refrigerant-side maldistribution caused by pressure balance between tubes, and air-side distribution for multislabs. Air-side and refrigerant-side microscale heat transfer and pressure drop correlations are carefully compared. The model results match well with lab test results for one-slab and two-slab microchannel heat exchangers on heat transfer and pressure drop. Several simulations are conducted to determine the impact of return air temperature, tube wall temperature, and non-uniform refrigerant flow rate. In addition, optimization results are analyzed by changing the number of flat tubes in each pass for both types of heat exchangers.

Evaluation des caractéristiques d'un condenseur à microcanal par une simulation numérique

Mots-clés : Simulation numérique ; Microcanal ; Condenseur ; Multi-plaque

1. Introduction

The implementation of microchannel heat exchangers has increased in automotive and building air conditioning systems during the past few decades. Microchannel condenser research for automotive air conditioning systems resulted in the use of CO_2, due to its smaller volume, compact structure, heat resistance, and smaller refrigerant charge inventory. As interest in the technology rapidly has expanded, the importance of developing and optimizing efficient, low-cost microchannel heat exchangers has become apparent.

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Nomenclature

A	area (m^2)
C	capacity rate ($W\ K^{-1}$)
C_l	parameter in Lockhart-Martinelli [35] correlation
C_p	specific heat ($J\ kg^{-1}\ K^{-1}$)
f	friction factor
F_l	fin length (mm)
F_p	fin pitch (mm)
G	mass velocity ($kg\ m^{-2}\ s^{-1}$)
h	heat transfer coefficient ($W\ m^{-2}\ {}^\circ C^{-1}$)
h	specific enthalpy ($kJ\ kg^{-1}$)
L	length (m)
L_l	louver length (m)
L_p	louver pitch (m)
m	mass flow rate ($kg\ s^{-1}$)
NTU	number of transfer units
P	pressure (Pa)
P_{crit}	critical pressure (Pa)
P_{sat}	saturated pressure (Pa)
P_r	$=p_{sat}/p_{crit}$, reduced pressure
Pr	Prandtl number
Q	heat transfer rate (W)
R	heat resistance ($m^2\ K\ W^{-1}$)
Re	Reynolds number
Re_f	superficial liquid Reynolds number
Re_{fo}	liquid only Reynolds number
Re_g	superficial vapor Reynolds number
Su	Suratman number
T	temperature (°C)
T_d	tube depth (mm)
T_p	tube pitch (mm)
U	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
v	specific volume ($m^3\ kg^{-1}$)
We	Weber number
X	Lockhart-Martinelli parameter

Greek symbols

Δ	difference
β	channel aspect ratio ($\beta < 1$)
λ	thermal conductivity ($W\ m^{-2}\ K^{-1}$)
ε	effectiveness
δ	thickness of fin (mm)
θ	louver angle (deg)
σ	surface tension
ρ	density ($kg\ m^{-3}$)

Subscripts

a	air
e	entrance
f	saturated liquid
fo	fluid only
g	saturated vapor
in	inlet
it	inlet tube
i, j, k	element tube coordinate
max	maximum
l	leaving
ot	outlet tube
out	outlet
ref	refrigerant
t	tube
tt	turbulent liquid-turbulent vapor
tv	turbulent liquid-laminar vapor
vt	laminar liquid-turbulent vapor
vv	laminar liquid-laminar vapor
w	tube wall

Previous research has been conducted on various models for finned tube (Jiang. 2003; Jiang et al., 2006; Singh et al., 2008) and microchannel heat exchangers (Yin et al., 2001; Wu and Webb, 2002; Kim and Bullard, 2001; Asinari et al., 2004; Park and Hrnjak, 2007; Shao et al., 2009; Fronk and Garimella, 2010; Garcia-Cascales et al., 2010; Martinez-Ballester et al., 2011, 2013; Ren et al., 2013; Huang et al., 2014). Wu and Webb (2002) presented an analytical model for a microchannel evaporator that operates under dehumidifying conditions. The in-tube refrigerant flow was divided into superheat, two-phase, and liquid regions, and the effects of refrigerant pressure drop at the tube entrance and exit were considered. For the superheated and liquid regions, the single-phase Petukhov equation (1973) was used to predict heat transfer and friction factors. The Kays and London correlation (1984) was used to calculate pressure losses at the tube entrance and exit. The dry air heat transfer was calculated by using the semi-analytical correlation reported by Webb et al. (1995). This program over-predicted cooling capacity by 8%, which may have been caused by flow maldistribution in the tubes. Because the heat exchanger was divided into three zones, tube-wall temperature, non-uniform air, refrigerant flow, and fin heat conduction could not be quantifiable.

Kim and Bullard (2001) also developed a microchannel evaporator model for CO_2 systems. The physical evaporator model employed numerous slabs, each of which was divided into multiple segments along the direction of the refrigerant and air flow. The researchers used air-side heat transfer and pressure drop correlations that they developed, Hwang (1997) correlation for refrigerant-side heat transfer, and the Tran et al. (1999) correlation for refrigerant-side pressure drop. The calculated air-side heat transfer data were in good agreement with measured data such that the root-mean-square (RMS) error for cooling capacity was within ±2.6%. However, the refrigerant-side pressure drop was usually underestimated, in that the RMS error was within ±13.1%. In that study, tube-wall temperature was not considered, and uniform air-flow temperature and velocity and uniform refrigerant mass flow rates in each tube were established as assumptions. To obtain more accurate estimations of microchannel performance, several researchers used the finite element method to analyze the process, which is able to consider key parameters such as fin conduction, tube wall temperature, and non-uniform air flow. Yin et al. (2001) developed a single-phase mathematical model for a CO_2 microchannel gas cooler. They used Gnielinski (1976) correlation for refrigerant-side heat transfer, Churchill (1977)