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A numerical study of condensation heat transfer and pressure drop in horizontal round and flattened minichannels

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Condensation heat transfer and pressure drop characteristics for R410A and R134a in horizontal round (d_h = 3.78 mm) and flattened tubes [aspect ratio (AR) = 3.07, 4.23, and 5.39] at saturation temperature T_{sat} = 320 K were investigated numerically. Liquid-vapor interfaces and stream traces are also presented to give a better understanding of the condensation process inside these tubes. The results indicate that local heat transfer coefficients and pressure gradients increase with increasing mass flux, vapor quality, and aspect ratio. The enhancement of the flattened tube is more pronounced at higher mass flux and vapor quality. The liquid film thickness decreases with increasing aspect ratio, mass flux, and vapor quality. The average liquid film thickness of R134a is about 40% thinner than R410A. For round tubes, a vortex with its core lying at the bottom of the tube is observed in the vapor phase region, while this vortex is only noticeable at lower mass flux in flattened tubes. The transportation of vapor phase from the core region to the wall region, like the flow pattern of longitudinal vortex, will also enhance heat transfer. The numerically-computed heat transfer coefficients and frictional pressure gradients for flattened tubes are compared with their counterparts from empirical correlations.

1. Introduction

Plenty of research works on the heat transfer and pressure drop characteristics of heat exchangers and the way to improve their performance have been carried out. The reason for this is the large variety of industrial applications in which heat exchangers are utilized, including power plants, air conditioning systems, electronic equipment, and space vehicles. Among different types of heat exchangers, finned-tube heat exchangers are widely adopted as condensers or evaporators for air conditioning systems. Most of the experimental and numerical works on finned-tube heat exchangers focus on round tubes, while studies about oval or flattened tubes are limited. A general configuration of finned flattened tube heat exchangers is shown in Fig. 1. As pointed by Kim et al. [1], the usage of such flattened tubes can enhance the air-side performance and reduce the refrigerant charge on the fluid-side.

The heat transfer and hydrodynamic performance on the air-side of finned flattened-tube heat exchangers have been studied experimentally and numerically. Recently, Tahseen et al. [2] carried out an overview of thermal and fluid flow characteristics of the tube banks heat exchangers. The effects of tube diameters, tube rows, tubes pitch and fins pitch, and external velocities on heat transfer coefficients and pressure drops in these heat exchangers were discussed in detail. They also pointed out that more future studies are needed on flattened tubes. Webb and Iyengar [3] measured the air-side performance of an automotive radiator with round and oval tubes. The pressure drop for oval tubes was 10% lower than that of round tubes, while the heat transfer performance was similar. Li (corresponding author) and Wang [4] experimentally studied the air-side heat transfer and pressure drop characteristics of aluminum heat exchangers with multi-region louver fins and flat tubes. Two correlations for Colburn j factor and Fanning friction factor f were developed based on the experimental results. Zhang et al. [5] investigated experimentally and numerically the heat transfer and flow characteristics around oval-shaped tubes. The streamlined shapes of these tubes can delay or avoid boundary separations, leading to a lower pressure drop on the air-side. Wang et al. [6] performed numerical studies of heat transfer

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Nomenclature

Ac	Cross-section area for round tubes, m^2
AR	Aspect ratio, W/H –
C	Chisholm number, –
Cp	Specific heat capacity, J/kg K
d_h	Hydraulic diameter, m
d_e	Equivalent diameter, m
E	Specific sensible enthalpy, J/kg
f	Fanning friction factor, –
F_E	Friedel parameter, $(1-x)^2 + x^2 \rho_l f_{vo}/(\rho_v f_{lo})$ –
F_F	Friedel parameter, $x^{0.78}(1-x)^{0.224}$ –
F_H	Friedel parameter, $(1-\mu_v/\mu_l)^{0..7}(\rho_l/\rho_v)^{0..91}(\mu_v/\mu_l)^{0.19}$ –
Fr	Froude number, $G^2/(gd\rho_H^2)$ –
g	Gravitational acceleration, m/s^2
G	Mass flux, kg/m^2 s
h	Heat transfer coefficient, W/m^2 K
H	Tube height, m
h_{lv}	Latent heat of vaporization, J/kg
J_G	Dimensionless modified Froude number, $xG/[gd\rho_v(\rho_l-\rho_v)]^{0.5}$ –
J_G^T	Transition dimensionless gas velocity, $\{2.6^{-3} + [7.5/(4.3X_{tt}^{1.111}+1)]^{-3}\}^{-1/3}$ –
k	Thermal conductivity, W/m K
L	Length of computational model, m
Lo^*	Non-dimensional Laplace constant, $\left[\sigma/g(\rho_l-\rho_v)\right]^{0.5}/d$ –
m	Mass source due to phase change, kg/m^3 s
MAD	Mean average deviation, –
MRD	Mean relative deviation, –
P	Pressure, Pa
P_{red}	Reduced pressure, –
P_w	Wetted perimeter, m
Pr	Prandtl number, –
Pr_t	Turbulent Prandtl number,–
q	Heat flux, W/m^2
r	Coefficient of mass source, 1/s
R	Tube corner radius, m
Re_l	Liquid Reynolds number, $Gd(1-x)/\mu_l$ –
Re_{lo}	All liquid Reynolds number, Gd/μ_l –
Re_v	Vapor Reynolds number, Gdx/μ_v –
Re_{vo}	Liquid Reynolds number, Gd/μ_v –
Su_{vo}	Suratman number, $\rho_v\sigma d/\mu_v^2$ –
T	Temperature, K
v	Velocity, m/s
W	Tube width, m
We	Weber number, $G^2d/(\sigma\rho_H)$ –
x	Vapor quality, –
X_{tt}	Lockhart-Martinelli parameter, $(1/x-1)^{0.9}(\rho_v/\rho_l)^{0.5}(\mu_l/\mu_v)^{0.1}$ –

Greek letters

α	Volume fraction, –
δ	Liquid film thickness, μm
ε	Void fraction of vapor, –
κ	Curvature of the interface, 1/m
μ	Dynamic viscosity, Pa s
μ_t	Turbulent viscosity, Pa s
ρ	Density, kg/m^3
σ	Surface tension, N/m
Φ	Two-phase pressure drop multiplier

Subscripts

a	Acceleration
annular	Annular flow regime
cal	Results predicted by correlation
conv	Convective
h	Homogenous
l	Liquid phase
lo	All refrigerant assumed to be liquid
sat	Saturation status
sim	Results obtained by simulation
strat	Stratified flow regime
tp	Two phase
v	Vapor phase
vo	All refrigerant assumed to be vapor
wall	Wall
wavy	Wavy flow regime

characteristics in a finned flattened tube heat exchanger using both the uniform temperature boundary condition and the conjunctive numerical method on fin surfaces. Similar numerical works about finned flattened tube heat exchangers were carried out by Delac et al. [7], Bahaidarah et al. [8], Wen et al. [9], etc.

Compared with the air-side performance, the research works on heat transfer and flow characteristics in flat tubes are limited, especially for two-phase flow. Most of condensation, evaporation, and flow boiling works focus on round tubes. Reviews of condensation in horizontal tubes covering flow regimes, pressure drops, and heat transfer characteristics in round tubes can be found in Cavallini et al. [10], Doretti et al. [11], Dalkilic and Wongwises [12]. However, there are limited works on condensation or evaporation in flattened tubes.

Quibén et al. [13,14] investigated flow boiling heat transfer and pressure drop characteristics in horizontal flattened tubes (equivalent diameters $d_e = 8.6$, 7.17, 6.25, and 5.3 mm) for R22 and R410A at mass velocities ranging from 150 to 500 kg/m^2 s, heat fluxes from 6 to 40 kW/m^2, and saturation temperature of 278.15 K. Similar to round tubes, the mass and heat flux had significant effects on the heat transfer coefficients and on the onset of dryout in flattened tubes. A modified flow boiling heat transfer model which is capable at a reduced pressure up to 0.19 was developed based on the experimental data. Tibiriçá et al. [15] experimentally studied saturated flow boiling and critical heat flux (CHF) in small flattened tubes with an equivalent diameter of 2.2 mm using R134a and R245fa at $T_{sat} = 304.15$ K. For mass fluxes higher than 200 kg/m^2 s, similar heat transfer coefficients were found for flattened and round tubes. The heat transfer correlation for round tubes could predict the data of flattened tubes well with a mean absolute error lower than 20%. A new correlation for CHF was proposed for flattened tubes using the equivalent diameter to account for the effect of the geometry. Kim et al. [16] measured the heat transfer coefficients and pressure gradients in flattened tubes made from 5 mm round tubes for R410A at mass fluxes ranging from 150 to 500 kg/m^2 s, heat fluxes ranging from 5 to 15 kW m^{-2}, and saturation temperatures ranging from 283.15 to 288.15 K. Results show that the heat transfer coefficients and pressure drops increased