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Condensation heat transfer and pressure drop characteristics of R-134a in horizontal smooth tubes and enhanced tubes fabricated by selective laser melting

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ABSTRACT

This paper reports a study of condensation heat transfer and pressure drop of R134a inside four enhanced tubes and one plain tube fabricated by Selective Laser Melting (SLM). The results are compared to a plain commercial aluminum tube. The enhanced tubes consist of a tube with a metallic foam structure, a tube with eight short circumferential pin fins, a tube with five long circumferential pin fins and a tube with five twisted pin fins. The experiments were conducted at mass fluxes from 50 to 150 kg/m^2 s. Throughout the experiments, the inlet and outlet vapor qualities were maintained at 0.9 and 0.3, respectively. Two saturation pressures of 13.4 bar and 11.6 bar were investigated. The effects of fin height, refrigerant flow direction and mass flux on the heat transfer coefficient and pressure drop were studied. Our results show that for the wavy flow pattern, the saturation pressure, vapor quality, mass flux, refrigerant flow direction and fin structure have significant effects on the condensation heat transfer coefficient and pressure drop. At higher saturation pressures, the head impact on the fins with shorter fin height has a higher heat transfer coefficient than the back impact. For the longer and twisted fins, a reversed trend was observed. With an increase in the mass flux or a decrease in the saturation pressure, the difference in heat transfer coefficients between the head and back impact for the same tube structure reduces. The heat transfer coefficients of the metallic foam tubes are higher than that of the plain SLM tube with a large penalty of pressure drop. The eight-fin tubes yield higher efficiency indices in terms of heat transfer over pressure drop when compared to the other tubes.

1. Introduction

Condensation heat transfer is widely used in many industrial processes such as refrigeration, power plant and water desalination. In a condenser, heat is removed from the refrigerant which resulted in a change in phase in the refrigerant from vapor to liquid. During the phase change process, the refrigerant experiences a variety of flow patterns, which depend on its mass flow rate, vapor quality and various properties of the fluid. For higher refrigerant mass velocity, the flow patterns include annular flow, slug flow, plug flow and bubbly flow. For lower mass velocity, the flow patterns consist of stratified or wavy flow and plug flow. Since heat transfer coefficients and pressure drop vary with the flow pattern, it is useful to obtain the different flow patterns based on the mass flow rate, tube size, vapor quality and liquid and vapor properties [1]. Numerous flow pattern maps have been proposed over the years for predicting the transition in two-phase flow regimes in horizontal tubes, such as those reported in Refs. [2,3]. Xiao and Hrnjak [4] made use of flow visualizations and liquid film thickness measurements to study the flow characteristics of R134a condensing in a 6.1 mm inner diameter horizontal round tube. The mass fluxes were in the range of 50–200 kg/m^2·s. Their results indicated that the onset of condensation, flow regime, film distribution and void fraction were affected by the change in mass flux and heat flux.

With decreasing tube diameter, the flow pattern varies as surface tension forces become dominant as compared to the gravitational force. The Confinement number ($Co = (\sigma/[g(\rho_L/\rho_G)])0.5/d_e$) can be used as a criterion to distinguish between microscale and macroscale two-phase flows, where σ is the surface tension, g is the acceleration due to gravity, d_e denotes the tube diameter, and ρ_L and ρ_G are the liquid and vapor densities [5]. Kawahara et al. [6] carried out an investigation on two-phase flow performance

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Nomenclature

A	area (m^2)
c	specific heat (kJ/kg·K)
d	diameter (mm)
D	hydraulic diameter (m)
f	Fanning friction factor
G	mass flow (kg/m^2·s)
h_{lv}	latent heat of vaporization (kJ/kg)
h	heat transfer coefficient (W/m^2·K)
H	enthalpy of refrigerant (kJ/kg)
k	thermal conductivity (W/m·K)
L	length (m)
P	pressure (bar)
Q	heat rate (W)
R	thermal resistance (K/W)
x	vapor quality
v	velocity (m/s)
V	volume (m^3)
W	mass flow rate (kg/s)

Dimensionless number

Co	confinement number
Gz	Graetz number
Nu	Nusselt number
Fr	Froude number
Pr	Prandtl number
Re	Reynolds number
X	Lockhart and Martinelli number

Greek symbols

Δ	difference
η	efficiency index
Φ	friction multiplicator
ρ	density
μ	dynamic viscosity

Subscripts

en	enhanced tube
fric	frictional pressure drop
g	vapor side
h	hot water
i	inner
in	segment inlet
l	liquid side
lat	latent heat
ltt	liquid side parameter
mom	momentum pressure drop
o	outer
out	segment outlet
p	constant pressure
pre	preheater
ref	refrigerant
sat	saturation
sens	sensible heat
sm	smooth tube
static	static pressure drop
test	test section
w	water

in a microchannel circular tube using de-ionized water and nitrogen gas and developed a two-phase flow map based on the superficial velocities of gas and liquid. Their results showed that the single-phase friction factor and two-phase friction multiplier data were in good agreement with the conventional correlations. Revellin et al. [7] presented a two-phase flow pattern map for microtubes using an optical measurement method. The tests were run in a glass channel of 0.5 mm internal diameter with R134a. The results showed that transitions observed in the two-phase flow patterns did not agree well with the leading macroscale flow map for refrigerants nor did it agree with the microscale map for air and water flows. Lee et al. [8] conducted a study on annular condensation of FC-72 in microgravity under the influence of different gravitational accelerations. Their results showed that the influence of gravity was very pronounced at low mass velocities. However, at high mass velocities, the thickening of the condensate film was nonexistent for Lunar and Martian gravities due to the increase in vapor shear on the film interface. Under microgravity conditions, the condensation heat transfer coefficient was the highest near the inlet, and decreased along the axial direction.

For condensation inside tube channels, the tube surfaces can be modified to enhance the condensation heat transfer. These enhanced features include the use of macroscale inserts, micro-fins [9], herringbone structures and corrugated tubes. However, the pressure drop in an enhanced heat transfer tube is usually higher than that of a smooth tube. For an efficient condenser tube design, the benefits of heat transfer enhancement and the penalty of higher pressure drop should be simultaneously considered. Xiao and Hrnjak [10] studied the heat transfer and pressure drop during condensation in a horizontal smooth round tube of 6.1 mm inner diameter using R134a. The mass fluxes were varied from 50 kg/m^2·s to 200 kg/m^2·s and the heat fluxes were applied from 5 kW/m^2 to 15 kW/m^2. Their results showed that in the superheated region, even though the liquid film for higher mass flux was much thinner, the heat transfer coefficient (h) was not affected by the change in mass flux. On the other hand, in the two-phase region, higher mass flux clearly yielded higher h. When the mass flow was constant, an increase in heat flux increased h in the superheated region but h did not change with heat flux in the two-phase region.

For the micro-fin tube, Li et al. [11] performed experiments to evaluate the condensation heat transfer coefficients of smooth and micro-fin tubes using R22 and R410a as the working fluids. The refrigerant mass fluxes were varied from 100 to 400 kg/m^2·s for a tube of 9.52 mm outer diameter and from 300 to 550 kg/m^2·s for a tube of 5 mm outer diameter. Their results showed that the heat transfer coefficients of the micro-fin tube were about 1.65–2.55 times those of the smooth tube. However, the micro-fin tube also resulted in about 30% higher pressure drop as compared to the smooth tube. Olivier et al. [12] investigated condensation heat transfer coefficients of smooth, micro-fin and herringbone tubes using R22, R407C, and R134a with mass fluxes ranging from 400 to 800 kg/m^2·s. Their results indicated that heat transfer coefficients of the herringbone tube had an average enhancement factor of 1.7 as compared to the smooth tube values and about 1.4 as compared to the helical micro-fin tube values for all the refrigerants tested. The pressure drops of the herringbone tube were about 80% higher than those for the smooth tube, and about 27% higher as compared to the helical micro-fin tube. Subsequently, Han and Lee [13] conducted experiments to evaluate the