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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/m²·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.

Keywords: Condensation; R134a; metallic foam; pin fins, flow pattern; selective laser melting

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] and [3]. Xiao and Hrňjak [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/m²·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.

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Nomenclature			
<i>A</i>	area (m ²)	<i>Greek symbols</i>	
<i>c</i>	specific heat (J/kg·K)	Δ	difference
<i>d</i>	diameter (mm)	η	efficiency index
<i>D</i>	hydraulic diameter (m)	Φ	friction multiplier
<i>f</i>	Fanning friction factor	ρ	density
<i>G</i>	mass flow (kg/m ² ·s)	μ	dynamic viscosity
<i>h_{lv}</i>	latent heat of vaporization (kJ/kg)	<i>Subscripts</i>	
<i>h</i>	heat transfer coefficient (W/m ² ·K)	<i>en</i>	enhanced tube
<i>H</i>	enthalpy of refrigerant (kJ/kg)	<i>fric</i>	frictional pressure drop
<i>k</i>	thermal conductivity (W/m·K)	<i>g</i>	vapor side
<i>L</i>	length (m)	<i>h</i>	hot water
<i>P</i>	pressure (bar)	<i>i</i>	inner
<i>Q</i>	heat rate (W)	<i>in</i>	segment inlet
<i>R</i>	Thermal resistance (K/W)	<i>l</i>	liquid side
<i>x</i>	vapor quality	<i>lat</i>	latent heat
<i>v</i>	velocity (m/s)	<i>ltt</i>	liquid side parameter
<i>V</i>	volume (m ³)	<i>mom</i>	momentum pressure drop
<i>W</i>	mass flow rate (kg/s)	<i>o</i>	outer
<i>Dimensionless number</i>		<i>out</i>	segment outlet
<i>Co</i>	Confinement number	<i>p</i>	constant pressure
<i>Gz</i>	Graetz number	<i>pre</i>	preheater
<i>Nu</i>	Nusselt number	<i>ref</i>	refrigerant
<i>Fr</i>	Froude number	<i>sat</i>	saturation
<i>Pr</i>	Prandtl number	<i>sens</i>	sensible heat
<i>Re</i>	Reynolds number	<i>sm</i>	smooth tube
<i>X</i>	Lockhart and Martinelli number	<i>static</i>	static pressure drop
		<i>test</i>	test section
		<i>w</i>	water

With decreasing tube diameter, the flow pattern varies as surface tension forces become dominant 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 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/m²·s to 200 kg/m²·s and the heat fluxes were applied from 5 kW/m² to 15 kW/m². 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.

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/m²·s for a tube of 9.52 mm outer diameter and from 300 to 550 kg/m²·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 to 2.55 times those of the smooth one. 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/m²·s. Their results indicated that heat transfer coefficients of the herringbone tube had an average enhancement factor of 1.7 compared to the smooth tube values and about 1.4 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 tube-side condensation heat transfer performance of HFC134a and R404a. They found that the heat transfer coefficient increased with mass flux but decreased with increasing condensing temperature for both smooth and micro-fin tubes. The average condensation heat transfer coefficients of the micro-fin tubes were 1.5 to 2.5 times and 1.3 to 2 times larger than that in the smooth tube for HFC134a and R-404a, respectively. In addition to the features of the enhancement devices, the orientation of the tubes also influences the flow pattern and the magnitude of the condensation heat transfer coefficient. For instance, Sapali and Patil [14] investigated the condensation heat transfer coefficients of an inclined micro-fin tube using R134a. Their results showed that the inclination angle (θ) had a significant effect on the condensation heat transfer and the maximum heat transfer coefficient was achieved at $\theta = 30^\circ$. Yang et al. [15] experimentally investigated the condensation heat transfer and pressure drop of R410a for vertical upward flow of vapor in a smooth tube and a micro-fin tube. Their results showed that heat transfer enhancement factors were in the range of 1.3 to 2.05 at the saturation temperature (T_{sat}) of 40°C and 1.4 to 2.2 at T_{sat} of 48°C. The pressure drop penalty factors were in the ranges of 1.0 to 1.6 and 1.03 to 1.5 at $T_{sat} = 40^\circ\text{C}$ and $T_{sat} = 48^\circ\text{C}$, respectively. Colombo et al. [16] performed experimental studies on boiling and convective condensation for oil-free R134a in two micro-fin tubes and one smooth tube of 9.52 mm outer diameter. They found that the heat transfer coefficient and pressure drop were different from the correlations obtained from the literature and the tube with the largest number of fins showed the poorest condensation heat transfer performance. They suggested the existence of an optimal fin number for maximum condensation performance. Guo et al. [17] performed experiments to study convective condensation and evaporation of R22, R32 and R410a inside a smooth tube, a herringbone tube and a newly developed enhanced surface EHT tube with mass fluxes ranging from 57 to 181 kg/m²·s. Their results showed that heat transfer coefficient of the herringbone tube was 2.0 to 3.0 times larger than that of the smooth tube and 1.3 to 1.95 times larger for the EHT tube than that of the smooth tube.

Miyara et al. [18] investigated condensation heat transfer and pressure drop of R410a and R22 in a new type of herringbone tube. They found that heat transfer coefficients of the herringbone tube were higher than those of the helical micro-fin tube at high mass velocity. However, the herringbone tube had slightly lower heat transfer coefficients in the low mass velocity region. Apart from micro-fin and herringbone tubes, condenser tubes with other enhancement features were also studied. For example, Sarmadian et al. [19] studied condensation heat transfer and frictional pressure drop of refrigerant R600a inside a helically dimpled tube and a plain tube. Refrigerant mass fluxes were varied from 114 to 368 kg/m²·s. Their results showed that the heat transfer coefficients of the dimpled tube were 1.2 – 2 times those of the smooth tube with pressure drop penalties between 58% and 195%. Li et al. [20] conducted experimental studies to investigate the tube side condensation and evaporation of two three-dimensional enhanced heat transfer tubes and a smooth surface copper tube using R410A. The mass fluxes ranged from 80 to 260 kg/m²·s. Their results showed that the ratios of the condensation heat transfer coefficient of the enhanced heat transfer tubes to those of the smooth tube were 1.1 to 1.16 and the frictional pressure drop values of the two EHT tubes were very similar to one another. Zheng et al. [21] conducted a numerical investigation of heat exchanger tubes with different internal ribbed structures to determine the effects of rib arrangements on the flow pattern and heat transfer performance. They concluded that the average Nusselt number and friction factor of the V-type ribbed tubes were about 57 to 76% and 86 to 94% higher than those of the P-type ribbed tube, respectively. Aroonrat and Wongwises [22] investigated heat transfer and pressure drop of R-134a during condensation inside a dimpled tube. The mass fluxes were 300, 400, and 500 kg/m²·s. Their results showed that the heat transfer coefficient and frictional pressure drop of the dimpled tube were higher than those of the smooth tube. The dimpled tube enhanced the Nusselt number by about 1.3 to 1.4 times as compared to the smooth tube. When the equivalent Reynolds number was decreased, the friction factor increased from 2.8 to 4.1 times as compared to the smooth tube. Shi et al. [23] studied heat transfer and pressure drop of condensation of steam in horizontal tubes filled with annular metal foams of three different sizes. The vapor mass flow rates ranged from 20 kg/h to 100 kg/h. Their results showed that the unit mass efficiency coefficient of heat transfer in the metal foam tube was approximately 1.3 times higher than that in the micro-fin tube. Ji et al. [24] used a mastoid array to control the wettability of a heat pipe. The results showed that the newly designed heat pipe behaved excellently in response to the heating loads. Xie et al. [25-28] provided a phase separation method to enhance condensation heat transfer in horizontal tubes. For annular flows, liquid droplets in the vapor core can be captured by a mesh screen surface to increase the vapor void fractions near the tube wall. Their results showed that the heat transfer coefficients could be more than two times those in the bare smooth tube. Xie et al. [29] also performed a condensation experiment using R245fa with a modulated heat transfer tube that had a porous membrane tube in the center core. With the increase in vapor void fractions near the tube wall, the modulated heat transfer tube had up to 85% improvement in condensation heat transfer coefficients as compared to the bare tube.

Based on the above brief review, it can be seen that extensive investigations on smooth tubes and tubes of various enhanced features have been conducted. However, most enhanced tubes were limited to those with herringbone, micro-fin and dimpled features. To the best of the authors' knowledge, the use of tubes with larger pin fins of millimeter length scale or with fully-filled metal foam inserts to improve condensation heat transfer has not been reported in the literature. In addition, research on larger pin fins to modify the condensation flow pattern in order to increase heat transfer coefficients has also not been published. In order to study pin fin performance and make use of pin fins to enhance the inside condensation heat transfer coefficient, four enhanced fin tubes were fabricated by Selective Laser Melting (SLM). This is the first time the SLM technique is used to produce enhanced tubes for promoting internal condensation with high pressure refrigerants. This paper presents the results of experiments conducted with R134a and with mass fluxes ranging from 50 and 150 kg/m²·s. The inlet and outlet vapor qualities of the refrigerant were maintained at 0.9 and 0.3, respectively and two saturation pressures of 13.4 bar and

11.6 bar were investigated. The effects of fin height, fin twist, flow direction, and mass flux on heat transfer coefficient and pressure drop are discussed.

2. Experimental apparatus and procedure

2.1 Test facility

Fig. 1 shows the test facility used in the present study. It consists of one refrigerant system and four water circulating systems. The components of the refrigerant system are a refrigerant accumulator, a gear pump, a Coriolis mass flow meter, a preheater, a test section, two post condensers, view glasses and valves. The liquid refrigerant was recirculated through the system by a gear pump. A mass flow meter with a nominal flow range of 0 to 20 kg/min and a relative accuracy of $\pm 0.1\%$ was used to measure the refrigerant flow rate. The subcooled refrigerant was first heated by a preheater to the required vapor quality at the preheater outlet. Thereafter, the liquid-vapor mixture was channeled into the test section where internal condensation occurs. After leaving the test section, two post condensers were installed to further condense the refrigerant to a subcooled temperature. Finally, the refrigerant was directed back to an accumulator which also served as a gas-liquid separator. Three K-type thermocouples, calibrated to an accuracy of $\pm 0.1^\circ\text{C}$, were installed in the refrigerant system to measure the refrigerant inlet temperature to the preheater, refrigerant inlet temperature to the test section and refrigerant outlet temperature from the test section. The vapor quality at the inlet and outlet of the test section can be obtained from energy balance. Three pressure sensors with a range of 0 to 20 bar and accuracy of $\pm 0.1\%$ were used to measure the pressures at different locations. The four water circulating systems supplied water to the preheater, test section and the two post condensers. The preheater and test section are double pipe heat exchangers with refrigerant flowing through the internal tubes and water flowing through the outer tubes. Four K-type thermocouples and two water flow meters were installed on the water-side of the preheater and test sections. The water flow rate was measured by two turbine wheel flow meters with a nominal flow range of 0.4 to 5.0 L/min. The refrigerant pressure in the system was adjusted by the two heat exchangers that served as the post condensers. One of the post condensers is a plate type heat exchanger with a constant heat load, which can cool the refrigerant. The other post condenser consists of a coil of copper tube inserted into a chiller. The refrigerant in the copper tube can be cooled and heated to achieve different experimental conditions.

Fig. 1. Schematic of experimental apparatus

2.2 Test section

Fig. 2 shows the cross-sectional images and perspective drawings of the different enhanced features in the enhanced tubes produced by SLM. The enhanced tubes comprise 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. A plain SLM tube and a commercial aluminum tube were also fabricated for comparison. The geometrical parameters of these tubes are summarized in Table 1. For example, the pin fin tubes consist of eight pin fins located along the circumference of the inner tube. Each fin is 2.5 mm in length and has a diameter of 2.0 mm and is tilted at 45° with respect to the tube central axis. Experiments were conducted for the eight-fin tube, five-fin tube and twisted-fin tube at two different orientations. The first orientation is such that the refrigerant flow first comes into contact with the tip of the fin and this flow configuration is named 'head impact'. The second orientation is where the refrigerant first comes into contact with the fin base and this flow configuration is named 'back impact'.

The tubes with enhanced structures and the plain SLM tube employed in the present investigation were fabricated using the SLM250HL selective laser melting facility manufactured by SLM Solutions GmbH and housed in the Singapore Centre for 3D Printing of Nanyang Technological University, Singapore. An aluminum alloy (AlSi10Mg) powder of 20 μm to 63 μm size distribution was used as the base powder. A Gaussian distributed Yb:YAG laser with a maximum power of 400 W and a laser beam spot size of 80 μm was utilized to melt and fuse the powder layer-by-layer based on the input of a Computer Aided Design (CAD) model [30, 31]. In the present investigation, a laser power of 200 W, a scanning speed of 1300 mm/s and a hatching spacing of 0.08 mm were selected. Due to the size of the built chamber, five tubes of the same design were fabricated. Each tube is 170 mm long and all five tubes were then welded to form the test specimen. The internal and external diameters of the enhanced and plain SLM tubes are 8.7 mm and 12.7 mm, respectively.

Each condenser tube was enclosed by a larger copper pipe of outer diameter 19 mm and thickness 1.2 mm to form a counter-flow double pipe heat exchanger. Cooling water flowed through the copper pipe for condensation to take place and the effective length of the condenser tubes which were enclosed by the copper pipe was fixed at 670 mm. At the entrances and exits of the test section, four K-type thermocouples were inserted into the working fluid to obtain the refrigerant and water temperatures. The refrigerant saturation temperature and pressure can be adjusted by controlling the preheater and the post condensers.

For the metallic foam tubes, the metallic foam consists of repetitions of the Rhombi-Octet unit cell structure of 7 mm unit cell size to form the porous matrix as shown in Fig.2. A detailed description of the Rhombi-Octet unit cell structure can be found in Ref. [31].

The condensation experiments with R134a were conducted at saturation pressures of 11.6 bar and 13.4 bar, refrigerant mass flux of 50 - 150 kg/m²·s and the vapor qualities at the test section inlet and outlet were maintained at 0.9 and 0.3, respectively.

Fig. 2. Cross-sectional images and perspective drawings of the enhancement tubes

Table 1. Structure parameters of the test sections

Test tubes	d_o (mm)	d_i (mm)	Fin height (mm)	Fin diameter (mm)	Fin angle with flow (degree)	Tube thickness (mm)
Five- fin	12.7	8.7	4.5	2	45	2
Eight-fin	12.7	8.7	2.5	2	45	2
Twisted-fin	12.7	8.7	6	2	45	2
Metallic foam	12.7	8.7	-	2	45	2

Plain SLM	12.7	8.7	-	-	-	2
Commercial Al	12.7	8.7	-	-	-	-

2.3 Data reduction

The following equations are employed to determine the vapor quality, heat transfer coefficient and uncertainty of the measured data during each test run under steady state condition.

The test section inlet vapor quality, x_{in} , was determined from an energy balance in the preheater section given by

$$x_{in} = \frac{\frac{Q_{pre}}{W_{ref}}(H_{sat} - H_{ref,in})}{h_{lv}} \quad (1)$$

where Q_{pre} , W_{ref} , H_{sat} , $H_{ref,in}$ and h_{lv} are total heat transfer rate in the preheater, refrigerant flow rate, refrigerant saturated liquid enthalpy, subcooled enthalpy and latent heat, respectively.

The total heat transfer rate in the preheater is determined from an energy balance for the water flowing through the annulus as

$$Q_{pre} = c_{p,pre} W_{w,pre} (T_{w,pre,in} - T_{w,pre,out}) \quad (2)$$

where $c_{p,pre}$, $W_{w,pre}$, $T_{w,pre,in}$ and $T_{w,pre,out}$ are the specific heat of water taken at the average water temperature, the water flow rate, the annulus inlet water temperature and annulus outlet water temperature of the preheater. The total heat transfer rate in the preheater includes both the refrigerant sensible heat Q_{sens} and the latent heat Q_{lat} as given in Eqs. (3) - (5).

$$Q_{pre} = Q_{lat} + Q_{sens} \quad (3)$$

$$Q_{sens} = W_{ref} (H_{sat} - H_{ref,in}) \quad (4)$$

$$Q_{lat} = W_{ref} h_{lv} x_{in} \quad (5)$$

The vapor outlet quality, x_{out} , can be calculated from

$$x_{out} = x_{in} - \frac{Q_{test}}{W_{ref} h_{lv}} \quad (6)$$

The total heat transfer rate in the test section can be determined from an energy balance for the annulus given by

$$Q_{test} = c_{p,test} W_{w,test} (T_{w,test,out} - T_{w,test,in}) \quad (7)$$

where $c_{p,test}$, $W_{w,test}$, $T_{w,test,in}$ and $T_{w,test,out}$ are the specific heat of water taken at the average water temperature, the water flow rate, the annulus inlet water temperature and annulus outlet water temperature of the test section, respectively.

2.3.1 Heat transfer coefficient for condensation (h_{ref})

The log-mean temperature difference (LMTD) for the double pipe heat exchanger is determined from

$$LMTD = \frac{(T_{ref,test,out} - T_{w,test,in}) - (T_{ref,test,in} - T_{w,test,out})}{\ln \left[\frac{(T_{ref,test,out} - T_{w,test,in})}{(T_{ref,test,in} - T_{w,test,out})} \right]} \quad (8)$$

where $T_{ref,test,in}$, $T_{ref,test,out}$, $T_{w,test,in}$, $T_{w,test,out}$ are the inlet refrigerant temperature, outlet refrigerant temperature, inlet water temperature and outlet water temperature, respectively. Assuming no fouling resistance, the inner enhanced tube side heat transfer coefficient for condensation is determined from

$$h_{ref} = \frac{1}{A_i \left[\frac{LMTD}{Q_{test}} - \frac{1}{h_o A_o} - \frac{\ln(d_o/d_i)}{2\pi L k} \right]} \quad (9)$$

where A_i is the nominal internal surface area based on d_i . A_o , d_o , L and k are the outer surface area, diameter of the test tube, length of the test tube and thermal conductivity of the tube material, respectively.

Only one term is unknown on the right side of the Eq. (9), which is the water-side heat transfer coefficient (h_o) in the annulus of the double pipe heat exchanger. The heat transfer coefficient on the water side heat transfer coefficient can be obtained using the Wilson plot technique [32]. Before performing the internal condensation experiments, hot water was first used as the working fluid on the inner side of the condenser tubes. The overall thermal resistance (R) in the double pipe heat exchanger can be divided into three parts, such as the inside tube thermal resistance (R_i), outside tube thermal resistance (R_o) and conduction resistance (R_c), which is shown in Eq. (10).

$$R = R_o + R_c + R_i \quad (10)$$

The total thermal resistance (R) can be obtained from the energy conservation principle and the log-mean temperature difference (LMTD) and conduction resistance (R_c) can be calculated from the heat conduction equation. In addition, the outside tube thermal resistance (R_o) can be expressed as Eq. (11) and the constants c and m can then be determined by a series of experiments with the water flow rate on the inner side of the tube maintained constant.

$$R_o = \frac{1}{c v^m} \quad (11)$$

2.3.2 Water-side heat transfer coefficient (h_o)

Before the condensation experiment with refrigerant was run, an energy conservation experiment was conducted with hot water flowing in the inner side of the tube to obtain the water-side heat transfer coefficient for the commercial aluminum tube and plain SLM tube.

The water-side heat transfer coefficient (h_o) for the plain SLM tube and commercial aluminum tube can be calculated from

$$\frac{\Delta T}{Q_h} = \frac{1}{h_o A_o} + \frac{\ln(d_o/d_i)}{2\pi L k} + \frac{1}{h_i A_i} \quad (12)$$

where ΔT , Q_h , and h_i are the water-to-water log-mean temperature difference, heat transfer rate for the inner tube and heat transfer coefficient for the inner side of the tube, respectively. The heat transfer rate for the tube inner side is given by

$$Q_h = c_{p,h} W_{w,h} (T_{w,h,in} - T_{w,h,out}) \quad (13)$$

where Q_h , $c_{p,h}$, $W_{w,h}$, $T_{w,h,out}$, and $T_{w,h,in}$ are the heat transfer rate, specific heat of water, water flow rate, hot water outlet temperature and hot water inlet temperature. The log mean temperature difference is given by

$$\Delta T = \frac{(T_{w,test,in} - T_{w,h,out}) - (T_{w,test,out} - T_{w,h,in})}{\ln \left[\frac{(T_{w,test,in} - T_{w,h,out})}{(T_{w,test,out} - T_{w,h,in})} \right]} \quad (14)$$

In order to reduce the error due to a small temperature difference between the water inlet and outlet, the water in the annulus was maintained at laminar flow. For the entrance region in developing laminar flow, the Nusselt number for single flow is determined by the Graetz number [33]. In the present study, the Nusselt number for the outer side is expressed as

$$Nu_o = aGz^b \quad (15)$$

The Graetz number is defined as

$$Gz = (D/L)RePr \quad (16)$$

where Re , Pr , D , and L are the Reynolds number, Prandtl number, hydraulic diameter and the total length of the experimental tube, respectively.

For internal forced convection heat transfer with $3000 < Re < 5 \times 10^6$ and $0.5 < Pr < 2000$, the Gnielinski equation [33] gives an accurate prediction of Nu as

$$Nu_i = \frac{(f/8)(Re-1000)Pr}{1+12.75(f/8)^{0.5}(Pr^{2/3}-1)} \quad (17)$$

The Fanning friction factor is determined from the Petukhov equation [33] and is valid for $3000 < Re < 5 \times 10^6$.

$$f = (0.790 \ln Re - 1.64)^{-2}, \quad 3000 < Re < 5 \times 10^6 \quad (18)$$

For the plain SLM and commercial aluminum tubes, the root mean square roughnesses are $7.32 \mu m$ and $0.25 \mu m$, respectively [30]. As the Reynolds number inside the tube was held constant at 4490 and the flow was in the transition regime, the roughness had little effect on the friction factor. Using Eqs. (17) and (18), the inner heat transfer coefficient (h_i) can be calculated from

$$h_i = \frac{k_w}{d_i} Nu_i \quad (19)$$

where k_w is the thermal conductivity of water.

Finally, using Eqs. (12) and (19), the water-side heat transfer coefficient (h_o) of the plain SLM tube and the commercial aluminum tubes in the experiment can be obtained. For the present experiment, the heat transfer coefficients (h_o) for the commercial aluminum tube and the plain SLM tube can be expressed as Eqs. (20) and (21), respectively.

$$h_o = 3030 \cdot v^{0.3811}, \text{ W/m}^2 \cdot \text{K}, \text{ (commercial aluminum)} \quad (20)$$

$$h_o = 5056 \cdot v^{0.5026}, \text{ W/m}^2 \cdot \text{K}, \text{ (plain SLM)} \quad (21)$$

where v is water flow rate.

2.3.3 Uncertainties of dependent quantities

The uncertainty of the dependent quantities can be calculated by the following propagation of error equation:

$$\delta R = \pm \sqrt{\sum_{i=1}^n \left(\frac{\partial R}{\partial x_i} \delta x_i \right)^2} \quad (22)$$

where δR is the total uncertainty and the uncertainty of variable R can be calculated based on the uncertainty of primary measurements of n number of variables x_i .

The uncertainty of the heat transfer rate can be derived as

$$\frac{\delta Q_{test}}{Q_{test}} = \pm \sqrt{\left(\frac{\partial Q_{test}}{\partial W_{w,test}} \delta W_{w,test} \right)^2 + \left(\frac{\partial Q_{test}}{\partial T_{w,test,out}} \delta T_{w,test,out} \right)^2 + \left(\frac{\partial Q_{test}}{\partial T_{w,test,in}} \delta T_{w,test,in} \right)^2} \quad (23)$$

The uncertainty of LMTD can be obtained from

$$\frac{\delta \text{LMTD}}{\text{LMTD}} = \pm \sqrt{\left(\frac{\partial \text{LMTD}}{\partial T_{ref,test,out}} \delta T_{ref,test,out} \right)^2 + \left(\frac{\partial \text{LMTD}}{\partial T_{w,test,in}} \delta T_{w,test,in} \right)^2 + \left(\frac{\partial \text{LMTD}}{\partial T_{ref,test,in}} \delta T_{ref,test,in} \right)^2 + \left(\frac{\partial \text{LMTD}}{\partial T_{w,test,out}} \delta T_{w,test,out} \right)^2} \quad (24)$$

The uncertainties of the vapor qualities at the inlet and outlet of the test section can be obtained from the following equations:

$$\frac{\delta x_{in}}{x_{in}} = \pm \sqrt{\left(\frac{\partial x_{in}}{\partial Q_{pre}} \delta Q_{pre} \right)^2 + \left(\frac{\partial x_{in}}{\partial W_{ref}} \delta W_{ref} \right)^2} \quad (25)$$

$$\frac{\delta x_{out}}{x_{out}} = \pm \sqrt{\left(\delta x_{in} \right)^2 + \left(\frac{\partial x_{out}}{\partial Q_{test}} \delta Q_{test} \right)^2 + \left(\frac{\partial x_{out}}{\partial W_{ref}} \delta W_{ref} \right)^2} \quad (26)$$

The uncertainty of h_{ref} can be obtained from

$$\frac{\delta h_{ref}}{h_{ref}} = \pm \sqrt{\left(\frac{\partial h_{ref}}{\partial Q_{test}} \delta Q_{test} \right)^2 + \left(\frac{\partial h_{ref}}{\partial h_o} \delta h_o \right)^2 + \left(\frac{\partial h_{ref}}{\partial \text{LMTD}} \delta \text{LMTD} \right)^2} \quad (27)$$

The uncertainties of the measured and calculated parameters, which are evaluated by the sum of the root mean square method, are shown in Table 2.

Table 2. Uncertainties of measured quantities and calculated parameters

Parameter	Uncertainty
Temperature, T	$\pm 0.1^\circ\text{C}$
Mass flow rate of refrigerant, W_{ref}	$\pm 0.1\%$
Vapor at inlet of the test section, x_{in}	$\pm 2\%$
Vapor at outlet of the test section, x_{out}	$\pm 3\%$
Log mean temperature difference, ΔT_{lm}	$\pm 10\%$
Heat transfer rate of preheater, Q_{pre}	$\pm 2\%$

Heat transfer rate of test section, Q_{test}	$\pm 2.5\%$
Heat transfer coefficient, h_{ref}	$\pm 15\%$
Pressure drop, ΔP	$\pm 56\%$ at 0.05 bar

3. Results and discussion

Condensation heat transfer is closely related to the two-phase flow patterns. The flow pattern map of Ref. [34] indicates that two-phase flow in the plain and commercial aluminum tubes display the stratified/wavy flow pattern for the mass fluxes in the present investigation. However, for the enhanced tubes, the presence of pin fins and metallic foam reduces the available flow area and may alter the refrigerant flow pattern. In addition to the increased heat transfer area, the introduction of enhancement features also served to alter the two-phase flow pattern to increase the condensation heat transfer coefficients.

3.1 Comparison of the experimental data (h_{ref}) to existing correlations

Fig. 3 presents the heat transfer coefficients (h_{ref}) of R134a condensing inside the commercial aluminum tube at saturation pressures of 13.4 bar and 11.6 bar between mass fluxes of 50 and 130 kg/m²·s. The vapor quality at the inlet is 0.9 ± 0.01 and at the outlet, it is 0.3 ± 0.01 . It can be seen from the figure that as the mass flux increases, the condensation heat transfer coefficient (h_{ref}) increases slightly and the change in saturation pressure has no noticeable effects on h_{ref} . In the present investigation, the h_{ref} values of the commercial aluminum tube range from 1400 to 2136 W/m²·K and the increase in mass flux has a positive effect on the heat transfer coefficients. This is because the increase in the mass flux increases the vapor and liquid velocities and results in enhanced heat transfer by convection.

Figs. 4 and 5 compare the experimental data with the correlations of Cavallini and Zecchin [35] and Traviss et al. [36] for saturation pressures of 11.6 bar and 13.4 bar, respectively, for commercial aluminum tube. It can be seen that for all the 28 data points (in Figs. 4 and 5) obtained for the two different saturation pressures, there is better agreement with the correlation of Cavallini and Zecchin [35] with most of the data falling within $\pm 30\%$. The maximum deviations between the experimental results and the Cavallini and Zecchin [35] and Traviss et al. [36] correlations are 35% and 37%, respectively.

Fig. 3. Comparison of the heat transfer coefficients of commercial aluminum tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 4. Comparison of commercial aluminum tube experimental results against existing correlations at $P_{sat} = 11.6$ bar.

Fig. 5. Comparison of commercial aluminum tube experimental results against existing correlations at $P_{sat} = 13.4$ bar.

3.2 Results of metallic foam tube

Figs. 6 and 7 compare the experimental results of the condensation heat transfer coefficient and pressure drop for the plain SLM tube, commercial aluminum tube and metallic foam tube with P_{sat} at 13.4 bar and 11.6 bar, respectively. Due to the limitation of the gear pump and high pressure drop of the metallic foam tube, a limited range of mass flux of 50 to 125 kg/m²·s was investigated. For both the P_{sat} values, the heat transfer coefficients of the plain SLM tube are higher than those of the commercial aluminum tube. This is because the plain SLM tube is significantly rougher than the commercial aluminum tube.

On the other hand, the metallic foam tube has significantly higher heat transfer coefficients than the plain SLM and the commercial aluminum tubes. With increasing mass flux, it can be observed that the heat transfer coefficients of the metallic foam tube also vary. When the mass flux is higher than 90 kg/m²·s, the heat transfer coefficients increase greatly. The average enhancement factor is about 1.5 times as compared to the commercial aluminum tube. However, it can be seen from Fig. 7 that the pressure drop of the metallic foam tube is also significantly higher than that of the plain SLM tube and its pressure drop also increases with increasing mass flux. When the mass flux is about 123 kg/m²·s, the pressure drop of the metallic foam tube is about 8 times higher than that of plain SLM tube. Even though the overall heat transfer area of the metallic structure is significantly higher, only moderate heat transfer enhancement is achieved. Therefore, it can be concluded that within the range of mass flux tested, a substantial portion of metallic structure did not participate in the heat transfer process. However, three meaningful observations can be deduced from this investigation. First, although the average inlet mass flux of the metallic foam tube is the same as that of the plain SLM and commercial aluminum tubes, the local mass fluxes in between the intertwining pores of the porous matrix are much higher. These intertwining features may result in irregular flow patterns. Second, due to surface tension forces and low vapor velocities, the condensate film can be easily trapped onto the metallic structures. The liquid film increases the thermal resistance and significantly reduces heat transfer performance. Finally, with increasing mass flux, it can be seen that the heat transfer coefficient increases greatly. This is because the high vapor velocity at

higher mass flux pushes the liquid film in the radial direction and toward the internal tube wall, thereby achieving a vapor core and increases the effective heat transfer area of the metallic foam.

Fig. 6. Comparison of the heat transfer coefficients of plain SLM, commercial aluminum and metallic foam tubes at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 7. Comparison of the pressure drop of plain SLM, commercial aluminum tube and metallic foam tubes at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

With the reduction in saturation pressure, the heat transfer coefficients of the metallic foam tube increase slightly due mainly to the variation in the vapor and liquid densities. When the saturation pressure decreases, the vapor density decreases significantly but the liquid density increases slightly. For example, the decrease in saturation pressure resulted in the change in the vapor density from 76 to 57 kg/m³ (about

25%) but the liquid density only increases from 1078 to 1125 kg/m³ (about 4%). According to the conservation of mass, this resulted in a significant increase in vapor velocity. The higher vapor velocity increases the vapor shear on the liquid-vapor interface which enhances the heat transfer.

3.3 Results of the eight-fin tube

Figs. 8 and 9 compare the experimental results of the condensation heat transfer coefficient and pressure drop of the plain SLM and eight-fin tubes at the P_{sat} of 13.4 bar and 11.6 bar. It can be seen from Fig. 8 that the heat transfer coefficients increase gradually with increasing mass flux. The heat transfer coefficient for the eight-fin tube is higher than that of the plain SLM tube. In addition, the heat transfer coefficient for the ‘head impact’ flow configuration is almost two times that of the plain SLM tube. It should also be noted that the heat transfer coefficient of the ‘head impact’ configuration is higher than that of the ‘back impact’ configuration. For example, when the mass flux is at 117 kg/m²·s and saturation pressure is 13.4 bar, the heat transfer coefficients of ‘head impact’ and ‘back impact’ flow are about 4000 and 3000 W/m²·K, respectively. This result can be explained using the schematic diagrams of Figs. 10 and 11. For annular flow, the most important mechanism is that the thickness of the liquid film and condensation are controlled by vapor shear [37]. In the wavy flow pattern, the condensate liquid in the horizontal tube is simultaneously driven by pressure drop and vapor shear. However, comparing with the liquid flow rate in annular flow, the flow rate in wavy flow is significantly lower. Hence, the liquid film covers the tube wall and forms a thermal barrier that limits heat transfer from the tube wall. Under such circumstances, the large fin protruding into the vapor plays an important role in condensation heat transfer. As shown in Fig. 10, for the ‘head impact’ flow configuration, the flow vapor impacts on the fin tip first and then flows along the fin structure, which can drive the condensate liquid flow along the fin and keep the liquid film at the fin tip and fin stem thin. The liquid film flows down along the fin and mixes together with the main liquid flow near the tube wall. The fin tip can be deemed as the critical area for condensation even though its area is not large [38]. However, as shown in Fig. 11, for the ‘back impact’ flow, the vapor shear force will thicken the liquid film along the fin.

Finally, from Fig. 8, it can be seen that the pressure drop of the eight-fin tube is almost 2 times that of the plain SLM tube at different saturation pressures. In addition, the deviation of the ‘head impact’ and ‘back impact’ flow configurations at lower saturation pressure is smaller than that at higher saturation pressure. This is also due to the increasing vapor velocity (as mentioned in Section 3.2) which drives the liquid from the wavy flow pattern to the annular flow pattern.

Fig. 8. Comparison of the heat transfer coefficients of plain SLM and eight-fin tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 9. Comparison of the pressure drop of plain SLM and eight-fin tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 10. Liquid film on the fin tip of the 'head impact' flow configuration

Fig. 11. Liquid film on the fin tip of the 'back impact' flow configuration

3.4 Results for the five-fin tube

Figs. 12 and 13 compare the experimental data of condensation heat transfer coefficient and pressure drop of the plain SLM tube and five-fin tubes at the P_{sat} of 13.4 bar and 11.6 bar. It can be seen from Fig. 2 that the fins for the five-fin tube are larger than those of the eight-fin tube and the fins for the five-fin tube protrude into the hollow space in the center of the tube. When P_{sat} is 11.6 bar, the heat transfer coefficients increase gradually with the increasing mass flux. However, the performance is different when the P_{sat} is 13.4 bar. In addition, the heat transfer coefficient for the 'head impact' flow configuration is lower than that of the 'back impact' flow configuration at the same mass flux and the same P_{sat} . For example, when the mass flux is at $80 \text{ kg/m}^2\cdot\text{s}$ and P_{sat} is 13.4 bar, the heat transfer coefficients of 'head impact' and 'back impact' flow are about 1580 and $4132 \text{ W/m}^2\cdot\text{K}$, respectively. It can be seen that the heat transfer coefficient for the 'back impact' flow configuration is almost 2 times that of the smooth tube at the same mass flux. These results are different from those of the eight-fin tube, which can be explained using the schematic diagrams of Figs. 14 and 15. As shown in Fig.14, for the 'head impact' flow configuration, the vapor not only flows through the hollow space in the tube center but also through the space between the fin roots. The velocity of the vapor impacting on the fin tip decreases compared to the same circumstance as the eight-fin tube. In addition, the fin tip temperature increases because of the longer heat conduction distance along the fin height. The critical heat transfer area for the fin tips was influenced. When the flow is in the 'back impact' configuration, as shown in Fig.15, the condensate liquid will be easily driven from the region near the wall to the center of the tube along the fin surface. By

means of surface shear force and the fin's tip support, it is likely to form a second flow region such as flow column or liquid droplets in the center of the tube. Due to the decreasing thickness of the liquid film near the wall surface, the heat transfer coefficient is higher than that of the 'head impact' flow. Under this circumstance, the larger fins act mainly as a 'bridge' for the second flow region. The flow pattern, which is similar to the enhanced condensation tube investigated by Xu and his co-workers [25-29], is likely to be modified from the wavy flow pattern.

With the mass flux increasing, the deviation between the 'back impact' and 'head impact' flow becomes smaller, which means that the flow pattern would evolve into annular flow gradually. After that, there is no distinct difference between the 'head impact' flow and 'back impact' flow. The pressure drop of the five-fin-tube is almost 4 times higher than that of the plain SLM tube.

Fig. 12. Comparison of the heat transfer coefficient of five-fin tube and plain SLM tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 13. Comparison of the pressure drop of five-fin tube and plain SLM tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 14. Liquid film at the contact fin tip of the 'head impact' flow configuration

Fig. 15. Liquid film at the contact fin tip of the 'back impact' flow configuration

3.5 Results for twisted-fin tube

Figs. 16 and 17 compare the experimental results of condensation heat transfer coefficient and pressure drop of the twisted-fin and plain SLM tubes at P_{sat} of 13.4 bar and 11.6 bar. For higher saturation pressure, the heat transfer coefficients increase gradually with increasing mass flux. In addition, similar to the five-fin tube, at the same mass flux and saturation pressure, the heat transfer coefficient of the 'head impact' configuration is lower than that of the 'back impact' configuration. For example, when the mass flux is $73 \text{ kg/m}^2\cdot\text{s}$ and the saturation pressure is 13.4 bar, the heat transfer coefficients of the 'head impact' and 'back impact' flows are about 3226 and 4398 $\text{W/m}^2\cdot\text{K}$, respectively. The heat transfer coefficient of the 'back impact' flow configuration is almost 2.5 times that of the plain SLM tube at the same mass flux. Similar to that shown in Fig. 15, the centrifugal force produced by the twisted fins spreads the liquid to the upper portion of the fins, and the pressure drop along tube and the surface shear force push the condensate from the tip of one fin to the tip of another, and the stem of the fin along the flow direction. In the cross section of the tube, there are two liquid flow streams: one is the liquid layer near the wall surface, while the other is the liquid flow column or droplets at the center of the tube. Hence, the film layer thickness near the wall decreases greatly and the heat transfer coefficients increase. In addition, Fig. 17 shows that the pressure drop of the twisted fin tube is almost 4 times that of the plain SLM tube. This may be due to turbulent eddies near the fin tips and the inner flow droplets.

Fig. 16. Comparison of the heat transfer coefficient of twisted tube, plain SLM tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 17. Comparison of the pressure drop of twisted fin tube, plain SLM tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

3.6 Comparison for various fin tubes

3.6.1 Evaluation criteria

Figs. 18 and 19 compare the experimental data of condensation heat transfer coefficient and pressure drop for the metallic foam tube, eight-fin tube, five-fin tube, twisted-fin tube, commercial aluminum tube and

plain SLM tube at P_{sat} of 13.4 bar and 11.6 bar. As the heat transfer enhancement must be accompanied by a penalty of increased pressure drop, the thermal-hydraulic performance should be considered [39]. In order to evaluate the performance of heat transfer and pressure drop, Chamra et al. [40] proposed the following criterion:

$$\eta_1 = \frac{h_{en}/h_{sm}}{\Delta P_{en}/\Delta P_{sm}} \quad (28)$$

where h_{en} and h_{sm} are the heat transfer coefficients for condensation of the enhanced tube and plain SLM tube, respectively while ΔP_{en} and ΔP_{sm} are the pressure drops of the enhanced tube and plain SLM tube, respectively. Sometimes, the following criteria of the constraints of identical pressure drop and identical pumping power for driving the refrigerant are used:

$$\eta_2 = \frac{h_{en}/h_{sm}}{(\Delta P_{en}/\Delta P_{sm})^{1/2}} \quad (29)$$

$$\eta_3 = \frac{h_{en}/h_{sm}}{(\Delta P_{en}/\Delta P_{sm})^{1/3}} \quad (30)$$

Fig. 18. Comparison of the heat transfer coefficients of various tubes fabricated by SLM, commercial aluminum tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

Fig. 19. Comparison of the pressure drops of various tubes fabricated by SLM and commercial aluminum tube at P_{sat} of 13.4 bar and 11.6 bar, inlet vapor quality of 0.9 ± 0.01 and outlet vapor quality 0.3 ± 0.01 .

The h_{en} and h_{sm} values are based on the average values for all the experimental data while ΔP_{en} and ΔP_{sm} are for mass fluxes higher than $110 \text{ kg/m}^2\cdot\text{s}$ due to the large error of pressure drop at lower mass flux. The efficiency indices for different criteria are shown in Table 3, where the heat transfer coefficients and pressures for mass flux close to $111.6 \text{ kg/m}^2\cdot\text{s}$ are selected. For these criteria, it can be seen from Table 3 that the ‘head impact’ flow configuration of eight-fin tube has the best efficiency index as compared to the flow configurations of the other enhanced tubes, meanwhile, the left is the value of 11.6 bar and the right is the value of 13.4 bar.

Table 3. Efficiency indices for different criteria

Efficiency index	Metallic foam	Eight-fin tube head impact	Eight-fin tube back impact	Five-fin tube head impact	Five-fin tube back impact	Twisted head impact	Twisted back impact
η_1	0.24/0.16	0.80/0.83	0.80/0.55	0.26/0.16	0.26/0.26	0.25/0.24	0.29/0.22
η_2	0.60/0.46	1.20/1.3	1.18/0.92	0.61/0.44	0.64/0.70	0.72/0.69	0.78/0.70
η_3	0.81/0.66	1.38/1.51	1.34/1.09	0.82/0.62	0.87/0.96	1.02/0.99	1.08/1.02

3.6.2 Analysis on the obtained data

It is known that flow patterns have a strong connection with the condensation heat transfer coefficient. In addition, for the horizontal smooth tube the flow pattern is mainly related with the mass flux, vapor quality [2-4]. The heat transfer coefficient correlations for microfin tubes can be affected by several parameters, such as inertia force, gravitational force, surface tension force, operating pressure, refrigerant properties and so on. Therefore, several dimensionless parameters such as Froude number, Bond number, Weber number, Lockhart and Martinelli number are considered to present a few new correlations of the heat transfer coefficient and pressure drop. For example, Cavallini et al. [41] presented a condensation correlation for microfin tubes including Froude number, Bond number, and the area enhancement

parameter. Yu and Koyama [42] presented a heat transfer model with the Lockhart and Martinelli number. Kedzierski and Goncalves [43] have considered the ratio of the operation pressure and critical pressure in their model. In addition, Xing et al. [44] conducted condensation experiments with an inclination tube and a correlation with inclination angle was obtained. In the present study, the results show that additional parameters should be included for enhanced tubes such as the impact angle (θ), which is the head impact angle and back impact angle, the fin length to tube diameter ratio (l/d) and the contact condition of the fins, where l is the fin length.

With different fins inserted, the flow pattern may be changed at the same mass flux due to the cross-sectional area or the fin structure. From the result of the eight-fin tube, it can be predicted that the head impact on the short fins can prevent condensate retention on the fin tip and a more effective fin surface area is obtained when compared with the back impact. In the present study, the head impact angle can be defined as 45° and back impact angle as 135° according to the vapor flow direction and fin center line direction. It can also be predicted that the heat transfer coefficient should not vary linearly with the impact angle with the possibility of an optimal angle for the fin configuration.

The parameter of fin length to diameter ratio (l/d) could be another important factor that affects the flow pattern. A large fin length to diameter ratio enables the fin to act as a liquid bridge and results in a new liquid column which takes the place of the original central vapor core thus, reducing the thickness of the liquid film near the wall.

The metallic foam tube can be interpreted as a large number of randomly placed longer fins set on a smooth tube. The regular or irregular contact fins destroy the original two-phase pattern and cause most of the fin surface to be covered with a liquid film due to surface tension, which increases significantly with decreasing the hydraulic diameter of flow channels. Although the metallic foam can provide more area than the other enhanced tubes, the effective area for condensation will be decreased significantly for the present mass flux range.

The pressure for 11.6 bar is slightly higher than that of 13.4 bar, which can be explained by the two-phase flow pressure drop correlation of Lockhart and Martinelli [45]. The two-phase pressure drops for R134a inside tubes are the sum of three contributions, which are the static pressure drop Δp_{static} , the momentum pressure drop Δp_{mom} and the frictional pressure drop Δp_{fric} , as shown in Eq. (31). For a horizontal tube, there is no change in static head, so the static pressure drop is zero. The momentum pressure drop is determined by total mass flux, void fraction and R134a physical properties such as vapor density, liquid density and vapor quality. Due to the slight variation in the operation pressure, the changes in the momentum pressure drop can be neglected. For higher two-phase flow, Lockhart and Martinelli predicted that the two-phase frictional pressure drop is based on a friction multiplier Φ_{lt}^2 and pressure drop calculated by liquid phase Δp_l , such as Eq. (32).

$$\Delta p = \Delta p_{static} + \Delta p_{mom} + \Delta p_{fric} \quad (31)$$

$$\Delta p_{fric} = \Phi_{lt}^2 \Delta p_l \quad (32)$$

$$\Phi_{lt}^2 = 1 + \frac{c}{x_{tt}} + \frac{1}{x_{tt}^2} \quad (33)$$

$$\Delta p_l = 2f_1 \left(\frac{L}{d}\right) G^2 \frac{(1-x)^2}{\rho_l} \quad (34)$$

where the Lockhart and Martinelli number X_{tt} is shown as

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_G}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_G}\right)^{0.1} \quad (35)$$

For the uniform mass flux, it was found that the increasing ratio of X_{tt} from Eq. (35) between 11.6 bar and 13.4 bar is slightly higher than the decreasing ratio of Δp_f from Eq. (34). It means that vapor density decreases in 11.6 bar directly determine the increasing in frictional pressure drop. With mass flux increasing, all of the momentum pressure drop and frictional pressure drop increase greatly, which leads to the total two-phase pressure drop increases simultaneously.

4. Conclusions

An experimental study of the inside flow condensation performance of four enhanced heat transfer tubes, one plain tube produced by SLM and a commercial aluminum tube was carried out. The investigated mass fluxes ranged from 50 to 150 kg/m²·s. The inlet and outlet vapor qualities were 0.9 and 0.3, respectively, and the saturation pressures of 13.4 bar and 11.6 bar were investigated. The following conclusions can be drawn:

- 1) For the metallic foam tube, the heat transfer coefficients are higher than those of the plain SLM tube and the commercial aluminum tube. The average heat transfer coefficient of metallic foam tube is about 1.5 times that of the plain SLM tube. However, the pressure drop is about 8 times higher than that of plain SLM tube.
- 2) For the eight-fin tube, the heat transfer coefficient for the ‘head impact’ flow configuration is almost two times that of the smooth tube. In addition, the heat transfer coefficients of the ‘head impact’ configuration are also higher than that of the ‘back impact’ flow configuration at the same mass flux and saturation pressure. With a decrease in saturation pressure or with an increase in mass flux, the differences in heat transfer coefficients between the ‘head impact’ and ‘back impact’ flow configurations are also reduced.
- 3) For the five-fin tube and twisted fin tube, the heat transfer coefficients for the ‘back impact’ flow condition are almost 2 and 2.5 times, respectively higher than that of the plain SLM tube at the same mass flux. The heat transfer coefficient of ‘back impact’ is higher than that of the ‘head impact’ flow at the same mass flux and the same saturation pressure. With pressure decreasing or mass flux increasing, the difference is smaller.
- 4) Based on the different criteria, the ‘head impact’ flow configuration of the eight-fin tube is the best flow configuration compared to the other enhanced tubes.

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