INVESTIGATION ON HEAT TRANSFER AND PRESSURE DROP OF R-134A DURING CONDENSATION IN A DIMPLED TUBE

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Abstract - The experimental study was conducted to investigate the condensation heat transfer and pressure drop of R-134a flowing inside a dimpled tube. The test section is a counter-flow double-tube heat exchanger, which the refrigerant flows in the inner tube and water flows in the annulus. The inner tubes are one smooth tube and one dimpled tube. The experiments are performed over a mass flux of 400 kg/m^{2}s, heat flux of 10 kW/m^2, and condensing temperature of 50 ˚C. The results show that the dimpled tube provides maximum heat transfer enhancement and pressure drop augmentation up to around 90% and 607% over the smooth tube, respectively.

Index Terms - Condensation, dimpled tube, heat transfer, pressure drop.

I. INTRODUCTION

In recent years, numerous researchers use various heat transfer enhancement techniques to improve the heat exchanger performance. Typically, heat transfer enhancement techniques can be categorized as active and passive methods. Active methods involve the use of external power, while the passive methods are those that use of channel surface modification to enhance the heat transfer. The tube with surface modification is generally referred as enhanced tube. Dimpled tube is one of many kinds of enhanced tube. The protrusions on the inner surface of dimpled tube disturb the flow, which provide higher turbulence level of the flow over the smooth surface, resulting in heat transfer improvement.

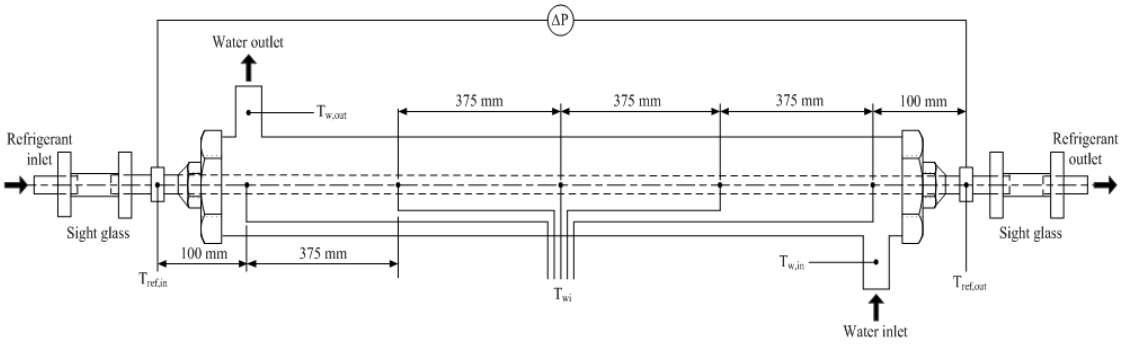


Fig. 1. Schematic diagram of the test section.

Over the years, heat transfer enhancement of refrigerant during condensation in enhanced tubes has been the subject of some researchers. However, there have been very few works relating to condensation of refrigerant in dimpled tubes, which are summarized as follows:

The performance of enhanced tube having protrusion on the inner surface developed by Vipertex was investigated by some group of researchers [1-4]. The refrigerants used in these experiments were R22, R32 and R410A. The results reported that tube with the surface covered by dimples and petals arrays gave the maximum condensation heat transfer enhancement and pressure drop penalty up to 1.95 and 4.0 times of the smooth tube. Sarmadian [5] explored the performance of a helically dimpled tube using R-600a during condensation. The results report that the dimpled tube provided the maximum heat transfer enhancement and pressure drop augmentation around 2 and 1.95 times of the smooth tube, respectively. Aroonrat and Wongwises [6-8] performed the experimental evaluation on condensation heat transfer and pressure drop of R-134a in dimpled tubes with different geometrical parameters. They found that the increment in heat transfer and pressure drop can be obtained with increasing protrusion height and decreasing helical and dimpled pitches. The maximum heat transfer and pressure drop augmentation were up to 88% and 892% over the smooth tube, respectively.

Although, some information according to condensation of refrigerant in dimpled tube was

reported in the literature, there still remains room for further research for different refrigerants and experimental conditions. Hence, the main aim of this study is to investigate the heat transfer enhancement and pressure drop penalty of R-134a during condensation in a dimpled tube.

II. EXPERIMENTAL APPARATUS AND METHODS

A detailed of the test section was shown in Fig. 1. The test section was a counter-flow double-tube heat exchanger with water flowing in an annulus and refrigerant condensing in the inner tube, which was placed horizontally. The inner tubes were one smooth and one dimpled tube. The sketch of dimpled tube was depicted in Fig. 2. The details of test tubes were illustrated in Table 1. Ten T-type thermocouples were installed at five positions along the outer surface of the test tube with the axial interval of 375 mm to measure the wall temperatures. The two-phase pressure drop was examined by a differential pressure transducer. The experiments were conducted for mass fluxes of 400 kg/m^{2}s, heat flux of 10 kW/m^2, condensing temperature of 50 ˚C, and the vapor quality of 0.1 to 0.8.

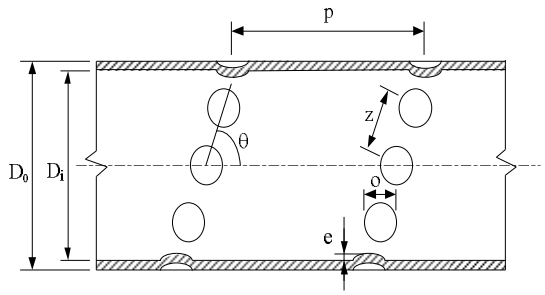


Fig. 2. Sketch of dimpled tube.

Table 1 Dimensions of the test tubes.

Parameters	Smooth tube	Dimpled tube
Outer diameter, mm	9.52	9.52
Inner diameter, mm	8.1	8.1
Length of test tube, mm	1,500	1,500
Inside tube area, mm^2	38,170	42,304
Helical pitch, mm	-	5.08
Dimple pitch, mm	-	3.24
Dimple depth, mm	-	0.75
Dimple diameter, mm	-	1.5
Helix angle, deg	-	79.47

III. DATA REDUCTION

The average heat transfer coefficient (h_{avg}) was calculated by the relationship between the heat transfer rate in the test section ($Q_{TS,w}$) obtained from the heat gained by water, the inner surface area of the tube (A_i), the average inner wall temperature ($T_{avg,wi}$) equal to the arithmetical mean value of the reading of the ten thermocouples at the tube wall, and the average saturation temperature of the refrigerant in the test tube obtained by averaging the temperature measured at the inlet and exit of the test tube ($T_{avg,sat}$):

$$h_{avg} = \frac{Q_{TS,w}}{A_i(T_{avg,wi} - T_{avg,sat})} \quad (1)$$

The frictional pressure drop (ΔP_F) was computed by:

$$\Delta P_F = \Delta P_T - \Delta P_G - \Delta P_A \quad (2)$$

where ΔP_T, ΔP_G, and ΔP_A are the total pressure drop, gravitational pressure drop, and accelerational pressure drop, respectively. For the horizontal tube, the gravitational pressure drop is equal to zero. The accelerational pressure drop was calculated as:

$$\Delta P_A = G^2 \left\{ \left(\frac{(1-x)^2}{\rho_l(1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right)_{out} - \left(\frac{(1-x)^2}{\rho_l(1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right)_{in} \right\} \quad (3)$$

where G is the refrigerant mass flux, ρ_l and ρ_g are the density of liquid and vapor phase, x is the vapor quality, and the void fraction (α) was computed from Zivi [9] correlation, as follows:

$$\alpha = \frac{1}{1 + \frac{(1-x)}{x}\left(\frac{\rho_g}{\rho_l}\right)^{2/3}} \quad (4)$$

IV. RESULTS AND DISCUSSION

The heat transfer coefficient of the smooth tube was compared with the values predicted by Cavallini et al. [10] correlation, as shown in Fig. 3. It was found that Cavallini et al. [10] correlation overestimates the data at high vapor quality, whereas it underestimates the data at low vapor quality. Although, this correlation shows a very good agreement with the experimental results by predicting almost all of data in the error range of 15%. Similarly, the frictional pressure drop

results for the smooth tube were compared with the correlation proposed by Laohalertdecha et al. [11], as depicted in Fig. 4. It was found that Laohalertdecha et al. [11] correlation predicts more than 90% of the present data within an error band of ±15%. Based on the above analysis, this confirms that this apparatus and procedures are reliable.

Fig. 5 shows the variation of the average heat transfer coefficient for smooth and dimpled tubes with average vapor quality at mass flux of 400 kg/m^{2}s, heat flux of 10 kW/m^2, and condensing temperature of 50 ˚C. The results indicate that the heat transfer coefficient rises with the increment of vapor quality. This is because the liquid film thickness gradually decreases when the vapor quality increases, resulting in lower liquid thermal resistance, which enhances the heat transfer. Moreover, as the vapor quality increases, the two-phase flow velocity tends to increase due to lower fluid density, which improves the convective heat transfer. This figure also indicates that the heat transfer enhancement is obtained from dimpled tube.

This is due to the protrusion on the inner surface of dimpled tube creates secondary flow, turbulence augmentation, and better mixing of the flow. The dimpled tubes generate maximum heat transfer coefficients around 90% higher than that of smooth tube.

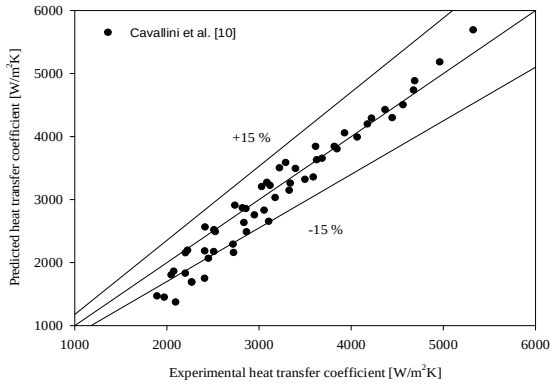


Fig. 3. Comparison of heat transfer coefficient of smooth tube with the correlation of Cavallini et al. [10].

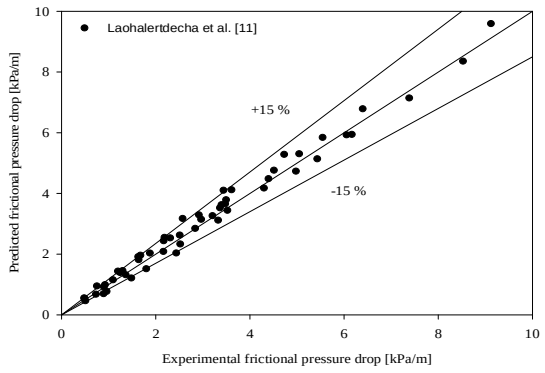


Fig. 4. Comparison of frictional pressure drop of smooth tube with the correlation of Laohalertdecha et al. [11].

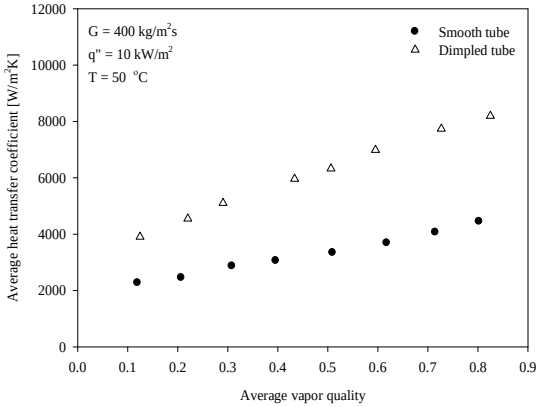


Fig. 5. Comparison of heat transfer coefficient for the smooth and dimpled tubes.

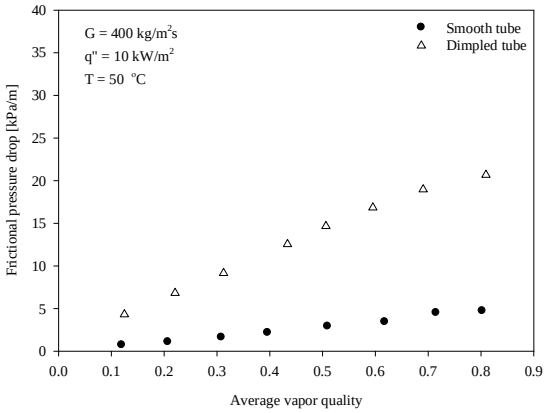


Fig. 6. Comparison of frictional pressure drop for the smooth and dimpled tubes.

Fig. 6 displays the variation of the frictional pressure drop for smooth and dimpled tubes with average vapor quality at mass flux of 400 kg/m^{2}s, heat flux of 10 kW/m^2, and condensing temperature of 50 ˚C. This graph shows that the frictional pressure drop continuously increases with a rise of vapor quality. This occurs because the velocity of vapor is higher with an increase of vapor quality, resulting in higher interfacial shear stress between the vapor core and liquid film, and hence more pressure loss. In addition, it can be observed that the dimpled tube presents higher frictional pressure drops over the smooth tube. This is due to the flow blockage by the protrusions, which causes the pressure drop to increase. The dimpled tube produces the frictional pressure drops of up to 607% higher than that of smooth tube.

CONCLUSIONS

The heat transfer coefficient and frictional pressure drop increase with a rise of vapor quality. The heat transfer coefficient and frictional pressure drop of dimpled tube are higher than those of the smooth tube. The dimpled tube produces the maximum heat transfer coefficient and frictional pressure drop

around 90% and 607% over the smooth tube, respectively.

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NOMENCLATURE

A	surface area of the test section, m^2
D	tube diameter, mm
e	depth of dimple, mm
G	mass flux, kg/m^2 s
h	heat transfer coefficient, W/m^2 K
L	length of test tube, m
o	diameter of dimple
P	pressure, Pa
p	helical pitch, mm
Q	heat transfer rate, W
q"	heat flux, kW/m2
T	temperature, ºC
x	vapor quality
z	dimpled pitch, mm

Greek symbols

α	void fraction
μ	dynamic viscosity, kg/m s
θ	helix angle, deg
ρ	density, kg/m^3
ΔP	pressure drop, Pa

Subscripts

A	accelerational
avg	average
d	dimpled tube
F	frictional
G	gravitational
g	gas/vapor
i	inside
in	inlet
l	liquid
o	outside
out	outlet
sat	saturation
TS	test section
w	water
wi	inner wall

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