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Experimental study of evaporation pressure drop characteristics of refrigerants R-134a and R-407C in horizontal small tubes

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Abstract

An experiment is carried out in the present study to investigate the characteristics of the frictional pressure drops for the evaporation of refrigerants R-134a and R-407C in horizontal small tubes having the same inside diameter of 0.83 mm or 2.0 mm. In the experiment for the 2.0-mm tubes, the refrigerant mass flux G is varied from 200 to 400 kg/m^2 s, imposed heat flux q from 5 to 15 kW/m^2, inlet vapor quality x_{in} from 0.2 to 0.8, and refrigerant saturation temperature T_{sat} from 5 to 15 °C. While for the 0.83-mm tubes, G is varied from 800 to 1500 kg/m^2 s with the other parameters varied in the same ranges as those for $D_i = 2.0$ mm. In this study, the effects of the inlet refrigerant vapor quality, mass flux, saturation temperature and imposed heat flux on the measured frictional pressure drops are examined in detail. Our experimental data clearly show that both the R-134a and R-407C frictional pressure drops increase significantly with the inlet vapor quality of the refrigerant, except at low mass flux and high heat flux. Besides, the effect of the imposed heat flux on the frictional pressure drop is rather weak. Moreover, a significant decrease in the frictional pressure drop results for a rise in T_{sat}. Furthermore, both the R-134a and R-407C frictional pressure drops increase substantially with the refrigerant mass flux. We also note that under the same x_{in}, T_{sat}, G, q and D_i, refrigerant R-407C has a lower frictional pressure drop when compared with that for R-134a. For the same refrigerant, a reduction in the duct size from 2.0-mm to 0.83-mm causes a significant increase in ΔP_f. Finally, an empirical correlation for the friction factor for the R-134a and R-407C evaporation in the 0.83-mm and 2.0-mm small tubes is proposed.

1. Introduction

Recently, there is growing interest in the use of ultra-compact heat exchangers in various thermal systems because of their very high heat transfer density. Thus the heat transfer and pressure drop characteristics in small and capillary tubes have been extensively investigated for various fluids such as air, water and some refrigerants [1]. But the pressure drop characteristics associated with the evaporation and condensation of the HFCs refrigerants in the small tubes are less explored. Data for the two-phase pressure drop in the small tubes are still scarce. In the present study, we conduct experiments to measure the evaporation pressure drop of the HFC refrigerants R-134a and R-407C in small tubes. We choose these two refrigerants in this study because they are regarded as the major substitutes for refrigerants R-12 and R-22.

In the following, the relevant literature on the evaporation pressure drop in small channels is briefly reviewed. Lazarek and Black [2] developed empirical correlations for frictional, spatial acceleration and bend pressure drops for saturated boiling of R-113 in a vertical U-tube with the hydraulic diameter $D_h = 3$ mm. Yan and Lin [3,4] found that the frictional pressure drop for R-134a evaporation increased with the refrigerant mass flux and imposed wall heat flux in a tube bank with $D_h = 2$ mm for each tube. Pressure drop of R-123 in a horizontal small tube with inside diameter of 1.12-mm was measured by Fujita et al. [5]. They indicated that the two-phase friction factor is nearly constant. Warrier et al. [6] reported experimental data and developed pressure drop correlation for saturated flow boiling of FC-84 in a horizontal tube bundle consisting

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Nomenclature

a, b, c, d, e, f	constants in Eq. (8)
C_c	coefficient of contraction
C_r	contraction ratio
D_h	hydraulic diameter (mm)
D_i	inside diameter of small tube (mm)
f_{tp}	two-phase friction factor (dimensionless)
g	gravitational acceleration (m/s^2)
G	mass flux (kg/m^2 s)
G_{eq}	equivalent mass flux (Eq. (10))
i_{fg}	enthalpy of vaporization (J/kg)
L	length of small tubes (m)
N_{conf}	confinement number, $N_{conf} = \frac{\left[\frac{\sigma}{g(\rho_l-\rho_g)}\right]^{0.5}}{D_h}$ (dimensionless)
q	average imposed heat flux (W/m^2)
Re_{eq}	equivalent Reynolds number, $Re_{eq} = \frac{G_{eq} \cdot D_i}{\mu_l}$ (dimensionless)
T	temperature (°C)
$T_{r,sat}$	saturated temperature of refrigerant (°C)
x	vapor quality

Greek symbols

α	void fraction
ΔP	pressure drop (Pa)
Δx	total quality change in the small tubes
μ	viscosity (Ns/m^2)
v	specific volume (m^3/kg)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

a	acceleration
exp	total
f	frictional
fg	phase change
g	vapor phase
i, in	at inlet of the test section
l	liquid phase
m	liquid–vapor mixture
o	exit
sat	saturation

of five parallel channels with a hydraulic diameter of 0.75 mm and length to diameter ratio of 409.8 for each channel. Two-phase boiling pressure drop measurements were made by Tran et al. [7] for three refrigerants R-134a, R-12, and R-113 in two round tubes ($D_i = 2.46$ and 2.92 mm) and one rectangular channel ($D_h = 2.4$ mm). Their data were used to develop new correlation for flow boiling frictional pressure drop in small channels. Chang and Ro [8] investigated the pressure drop of refrigerants R-32, R-125, R-134a and their mixtures in capillary tubes with inside diameters ranging from 1.0 to 2.0 mm and derived a model to predict their experimental data. A new correlation which modified the Friedel correlation for two-phase friction pressure drop was developed by Zhang and Webb [9]. The correlation predicts the data for refrigerants R-22, R-404a and R-134a flowing in a multi-port extruded aluminum tube ($D_h = 2.13$ mm) with a mean deviation of 11.5%. Recently, Qu and Mudawar [10] investigated the pressure drop in multi-port parallel micro-channels ($D_h = 0.348$ mm). They identified two types of two-phase hydrodynamic instability: severe pressure drop oscillation and mild parallel channel instability. They further noted that the two-phase pressure drop increased appreciably upon commencement of boiling in micro-channels. The pressure drop characteristics of refrigerants R-236ea, R-410A, and R-134a flowing in parallel mini-channels with $D_h = 1.4$ mm were reported by Cavallini et al. [11]. They compared their data with several models and showed that no one could fit the data of R-410A. Recently, a comprehensive review of the experimental data and prediction methods reported in the literature for two-phase frictional pressure drop and flow boiling heat transfer in micro-scale channels was conducted by Ribatski et al. [12]. The data were analyzed and compared against the prediction methods.

The above literature review clearly indicates that the experimental data for the evaporation pressure drop of the HFC refrigerants in small tubes are still in urgent need. In this study, we measure the evaporation pressure drop of refrigerants R-134a and R-407C in horizontal small tubes of inside diameter 2.0 and 0.83 mm. The effects of the vapor quality, refrigerant mass flux, imposed heat flux and system pressure on the evaporation frictional pressure drop in the small tubes will be examined in detail.

2. Experimental apparatus and procedures

The experimental system employed in our recent study of evaporation heat transfer in small tubes [13] is also used here to investigate the evaporation pressure drop of the HFC refrigerants in small tubes. It is schematically depicted in Fig. 1. The experimental apparatus consists of three main loops, namely, a refrigerant loop, a water–glycol loop, and a hot-water loop. Refrigerant R-134a or R-407C is circulated in the refrigerant loop. In order to control various test conditions of the refrigerants in the test section, we need to control the temperature and flow rate in the other two loops. The test section along with the entry

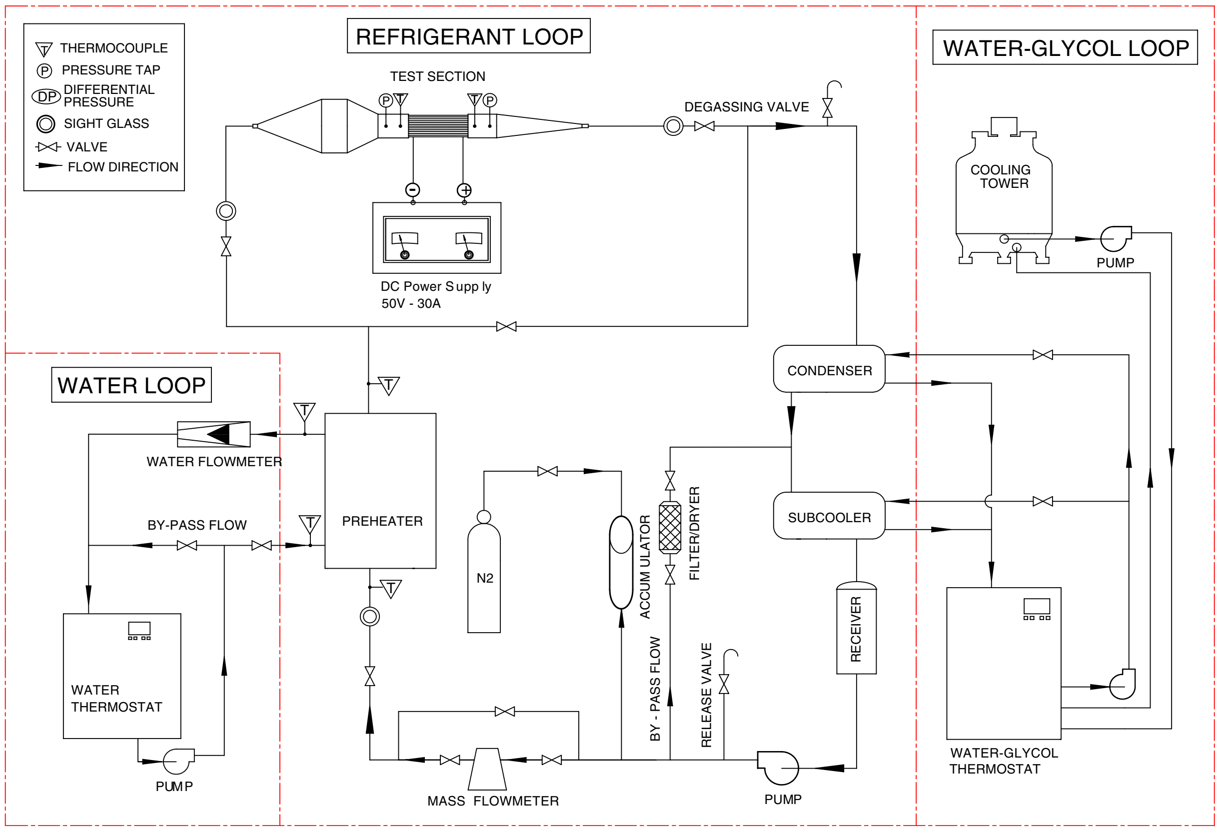

Fig. 1. Schematic layout of the experimental system.

and exit sections attached to it are schematically shown in Fig. 2. The test section is a tube bundle forming from 28 parallel side by side contacting small copper tubes having the same inside diameter of 0.83 or 2.0 mm, outside diameter of 1.83 or 3.0 mm, and length of 150 mm. Besides, the pressure transducers and thermocouples are placed at the inlet and exit of the test section to measure the pressure drops and refrigerant temperatures. Two copper plates of 5-mm thick are, respectively, soldered onto the upper and lower sides of the tube bundle. The copper plates are heated directly by an electric-resistance heater with a 500-W DC power supply. The power input to the heater is measured by a power meter with an accuracy of ±0.5%. In order to reduce the heat loss from the heaters, the whole test section is wrapped with a 10-cm thick polyethylene layer. It should be noted that the heated section of the tube bundle is only 100-mm long and there are two unheated sections each having 25-mm in length upstream and downstream of the heated section. Axial heat conduction in the tube walls can be important in affecting the measured data

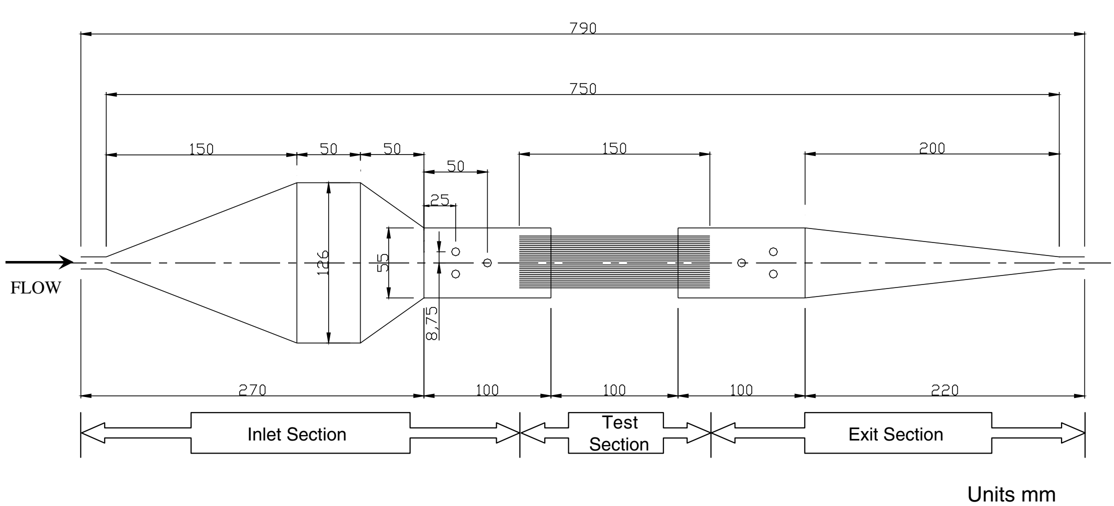

Fig. 2. Schematic diagram of test section along with the inlet and exit sections.

in view of the thermal conductivity of the copper being much higher than that of R-134a and R-407C. In the present study, however, the liquid refrigerant flow in the small tubes is at a very high Peclet number (>5000). Thus the conjugation effects between the convection in the flow and conduction in the tube walls are expected to be small, as evident from our early studies [14,15]. The details of the apparatus, test section, and experimental procedures are already available in our early studies [3,13] and are not repeated here.

3. Data reduction

Note that in the evaporation of R-134a or R-407C refrigerant in the tubes the flow accelerates, causing the pressure drop, as it moves downstream. Besides, the refrigerant pressure also drops due to the contraction at the test section inlet and rises due to the expansion at the exits of the small tubes. Thus, in the refrigerant flow the two-phase frictional pressure drop ΔP_f associated with the refrigerant evaporation in the small tubes is calculated by subtracting the pressure drop due to flow acceleration ΔP_a and the pressure drop at the test section inlet ΔP_i and by adding the pressure rise at the test section exit ΔP_o from the measured total pressure drop ΔP_{exp}. The frictional pressure drop is hence given as

$$\Delta P_f = \Delta P_{exp} - \Delta P_a - \Delta P_i + \Delta P_o \tag{1}$$

Note that the acceleration pressure drop is estimated by the homogeneous model for two-phase flow [16] as

$$\Delta P_a = G^2 v_{fg} \Delta x \tag{2}$$

Moreover, the pressure drop associated with the sudden contraction ΔP_i and the pressure rise associated with the sudden expansion ΔP_o for two-phase flow moving through the inlet and exit ports estimated by Collier [16] based on a separated flow model are chosen here. They can be expressed as

$$\Delta P_i = \left(\frac{G}{C_c}\right)^2 (1 - C_c) \left\{ \frac{(1 + C_c)[x_{in}^3 v_g^2/\alpha^2 + (1 - x_{in})^3 v_l^2/(1-\alpha)^2]}{2[x_{in} v_g + (1 - x_{in}) v_l]} - C_c \left[\frac{x_{in}^2 v_g}{\alpha} + \frac{(1 - x_{in})^2 v_g}{(1-\alpha)} \right] \right\} \tag{3}$$

and

$$\Delta P_o = G^2 C_r (1 - C_r) v_l \left[\frac{(1 - x_{in})^2}{(1-\alpha)} + \left(\frac{v_g}{v_l}\right) \frac{x_{in}^2}{\alpha} \right] \tag{4}$$

where C_c in Eq. (3) is the coefficient of contraction and it is a function of the contraction ratio C_r. The void fraction α in the above equations is calculated from the correlation given by Zivi [17] as

$$\alpha = \frac{1}{1 + \left(\frac{1-x_{in}}{x_{in}}\right)\left(\frac{\rho_g}{\rho_l}\right)^{2/3}} \tag{5}$$

Table 1
Summary of the uncertainty analysis

Parameter	Uncertainty
Small tubes geometry	
Length, width and thickness	±0.5%
Area	±1.0%
Parameter measurement	
Temperature, T	±0.2 °C
Temperature difference, ΔT	0.28 °C
System pressure, P	±2 kPa
Pressure drop, ΔP	±200 Pa
Mass flux of refrigerant, G	±2%
Evaporation heat transfer in small tubes	
Imposed heat flux, q	±4.5%
Inlet vapor quality, x_{in}	9.5%
Friction pressure drop, ΔP_f	±16.5%

Finally, for the evaporation of R-134a or R-407C in the small tubes the two-phase friction factor is expressed as

$$f_{tp} = \frac{\Delta P_f D_i}{2 G^2 v_m L} \tag{6}$$

where L is the length of the small tubes and v_m is the mean specific volume of the vapor–liquid mixture in the small tubes when they are homogeneously mixed and can be expressed as

$$v_m = [x_m v_g + (1 - x_m) v_l] = (v_l + x_m v_{fg}) \tag{7}$$

More detailed description of the data reduction is available from our earlier study [3]. Uncertainties of the measured pressure drops are estimated according to the procedures proposed by Kline and McClintock [18]. The detailed results from this uncertainty analysis are summarized in Table 1.

4. Results and discussion

The frictional pressure drops for the R-134a and R-407C evaporation in the small tubes deduced from the measured raw data for ΔP_{exp} are presented in the following. The present experiments are performed for refrigerant R-134a or R-407C in the tube bank forming from the 2.0-mm diameter tubes with the refrigerant mass flux G varied from 200 to 400 kg/m^2 s, imposed heat flux q from 5 to 15 kW/m^2, inlet vapor quality x_{in} from 0.2 to 0.8, and refrigerant saturated temperature T_{sat} from 5 to 15 °C. While for the other tube bank forming from the 0.83-mm diameter tubes, G is varied from 800 to 1500 kg/m^2 s with the other parameters varied in the same ranges as those for $D_i = 2.0$ mm. Note that different ranges of the refrigerant mass flux are chosen for the different sizes of the tubes. Since at the low mass flow rate the evaporating refrigerant flow in the smaller tubes for $D_i = 0.83$ mm is somewhat unsteady, leading to the unstable intermittent flow in the system. In the following the effects of the refrigerant vapor quality, imposed heat flux, and refrigerant mass flux and saturated temperature on the R-134a and

R-407C evaporation frictional pressure drops are to be examined in detail.

4.1. Frictional pressure drops in 2.0-mm tubes

The variations of the frictional pressure drops with the inlet vapor quality for R-134a evaporation in the 2.0-mm tubes are shown in Fig. 3 for various refrigerant saturated temperatures, mass fluxes and imposed heat fluxes. The results indicate that for all T_{sat}, G and q tested here the frictional pressure drop of R-134a in the tubes increases with the inlet vapor quality. The increase is more significant for a lower refrigerant saturated temperature and a higher mass flux. For instance, at $T_{sat} = 5\,°C$, $G = 400\ kg/m^2\,s$ and $q = 5\ kW/m^2$ the data in Fig. 3a show that the frictional pressure drop ΔP_f is increased by 48% when x_{in} is raised from 0.2 to 0.8. This large increase in ΔP_f with x_{in} is attributed to the noticeable increase in the vapor void fraction for x_{in} raised from 0.2 to 0.8, as evident from Eq. (5). At the higher void fraction the liquid film on the tube wall is thinner and more liquid–vapor interfacial area is available for liquid film evaporation. Thus the vapor flow is at a higher speed for a higher x_{in}, resulting in a larger ΔP_f. Fig. 3a also shows that an increase in the R-134a saturated temperature causes a large reduction in the frictional pressure drop. For example, the quality-averaged frictional pressure drop at $G = 400\ kg/m^2\,s$ and $q = 5\ kW/m^2$ is reduced by 37% for T_{sat} raised from 5 to 15 °C. This mainly reflects the fact that the dynamic viscosity of the liquid R-134a is lower and the density of the R-134a vapor is higher at a higher saturated temperature. Thus the speed of the vapor flow in the duct core is lower. Hence the two-phase flow frictions at the liquid–vapor interface and at the wall are reduced at increasing T_{sat}, resulting in a significant reduction in ΔP_f. It is also noted from Fig. 3b that a substantial increase in ΔP_f results for

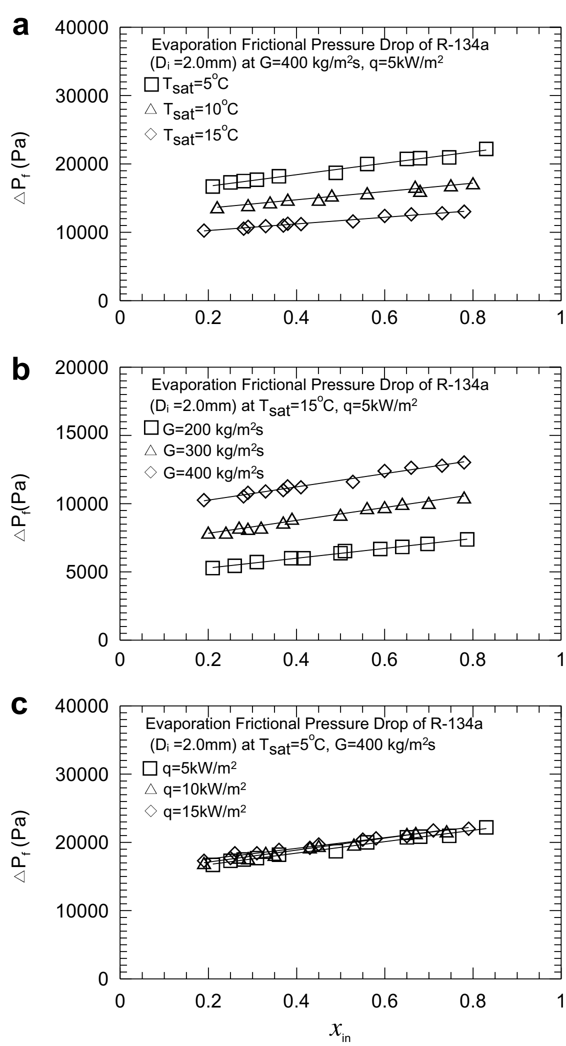

Fig. 3. Variations of R-134a evaporation frictional pressure drop with inlet vapor quality in 2.0-mm small tubes: (a) for various T_{sat} at $G = 400\ kg/m^2$ s and $q = 5\ kW/m^2$, (b) for various G at $T_{sat} = 15\,°C$ and $q = 5\ kW/m^2$, and (c) for various q at $T_{sat} = 5\,°C$ and $G = 400\ kg/m^2$ s.

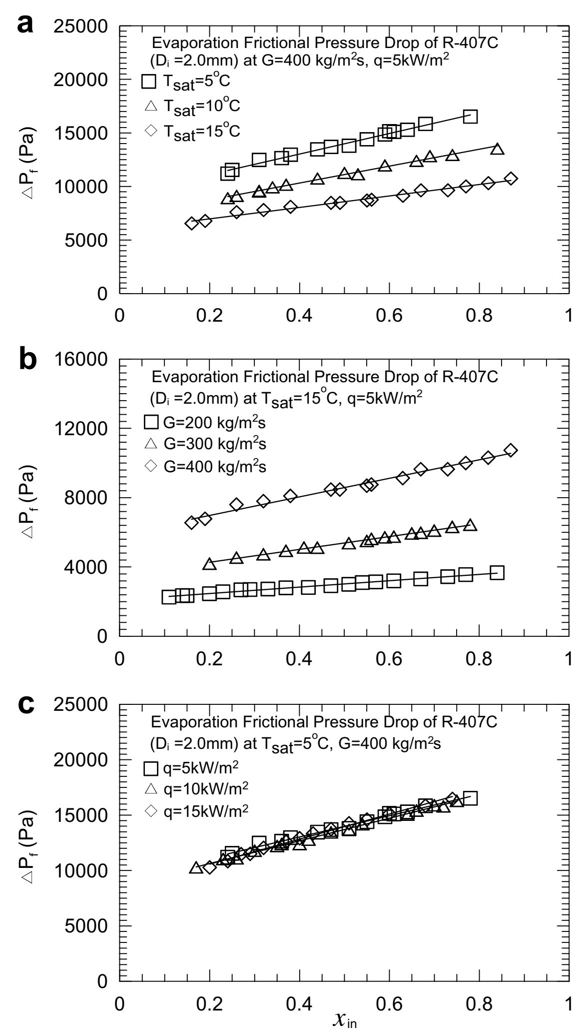

Fig. 4. Variations of R-407C evaporation frictional pressure drop with inlet vapor quality in 2.0-mm small tubes: (a) for various T_{sat} at $G = 400\ kg/m^2$ s and $q = 5\ kW/m^2$, (b) for various G at $T_{sat} = 15\,°C$ and $q = 5\ kW/m^2$, and (c) for various q at $T_{sat} = 5\,°C$ and $G = 400\ kg/m^2$ s.

an increase of the R-134a mass flux. According to Fig. 3b, at $T_{sat} = 15\ °C$ and $q = 5\ kW/m^2$ the quality-averaged ΔP_f is increased by 83.1% for G raised from 200 to 400 kg/m^2 s. The higher ΔP_f for a high refrigerant mass flux is attributed to the fact that both the speeds of the liquid and vapor flows in the tubes are directly proportional to the refrigerant mass flux. Finally, the frictional pressure drop is not affected to a noticeable degree by the imposed heat flux, as is evident from the results in Fig. 3c. In fact, this change of the frictional pressure drop with the imposed heat flux is within the experimental uncertainty. This small change of ΔP_f with q results from the fact that the increase in the total rate of liquid film vaporization in the entire tube bundle is very small compared with the vapor flow rate at the inlet since the tubes are short even for q raised from 5 to 15 kW/m^2. It is generally <1%.

The ΔP_f data for the R-407C evaporation in the 2.0-mm tubes are shown in Fig. 4 for comparison. The results manifest that the effects of the refrigerant inlet vapor quality, saturated temperature, mass flux and heat flux on the frictional pressure drop associated with the R-407C evaporation in the 2.0-mm tubes qualitatively resemble those presented above for R-134a. Contrasting the magnitudes of ΔP_f for the corresponding cases with the same T_{sat}, G and q shown in Figs. 3 and 4, respectively, for the R-134a and R-407C evaporation in the 2.0-mm tubes reveals that the frictional pressure drop in the R-134a flow is much higher. The higher ΔP_f for the R-134a evaporation is considered to result from the fact that R-134a has a much higher dynamic viscosity and a slightly lower latent heat of vaporization.

4.2. Frictional pressure drops in the smaller tubes ($D_i = 0.83$ mm)

The variations of the frictional pressure drops with x_{in}, T_{sat}, G and q for R-134a and R-407C evaporation in the

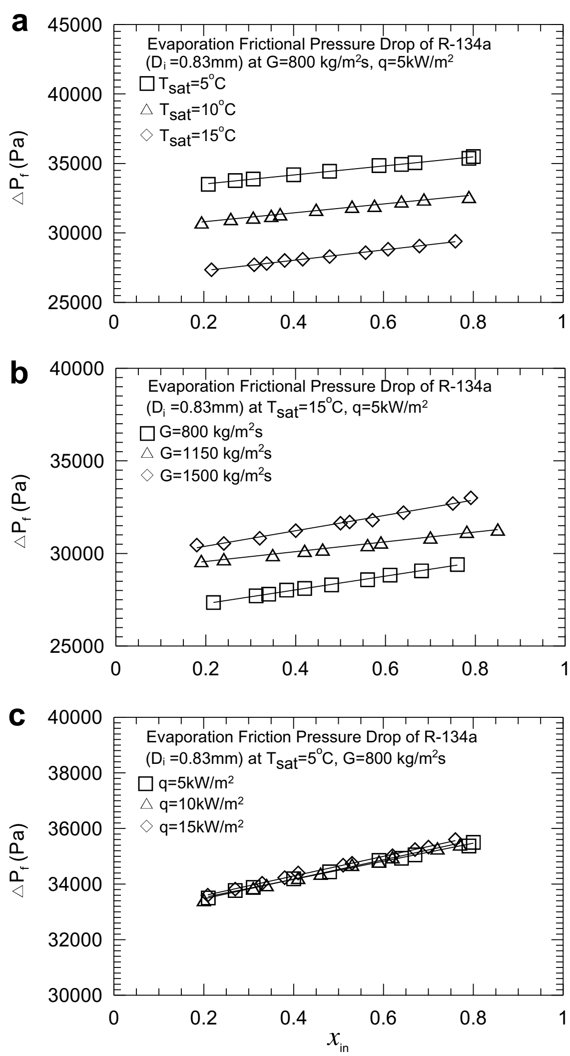

Fig. 5. Variations of R-134a evaporation frictional pressure drop with inlet vapor quality in 0.83-mm small tubes: (a) for various T_{sat} at $G = 800\ kg/m^2$ s and $q = 5\ kW/m^2$, (b) for various G at $T_{sat} = 15\ °C$ and $q = 5\ kW/m^2$, and (c) for various q at $T_{sat} = 5\ °C$ and $G = 800\ kg/m^2$ s.

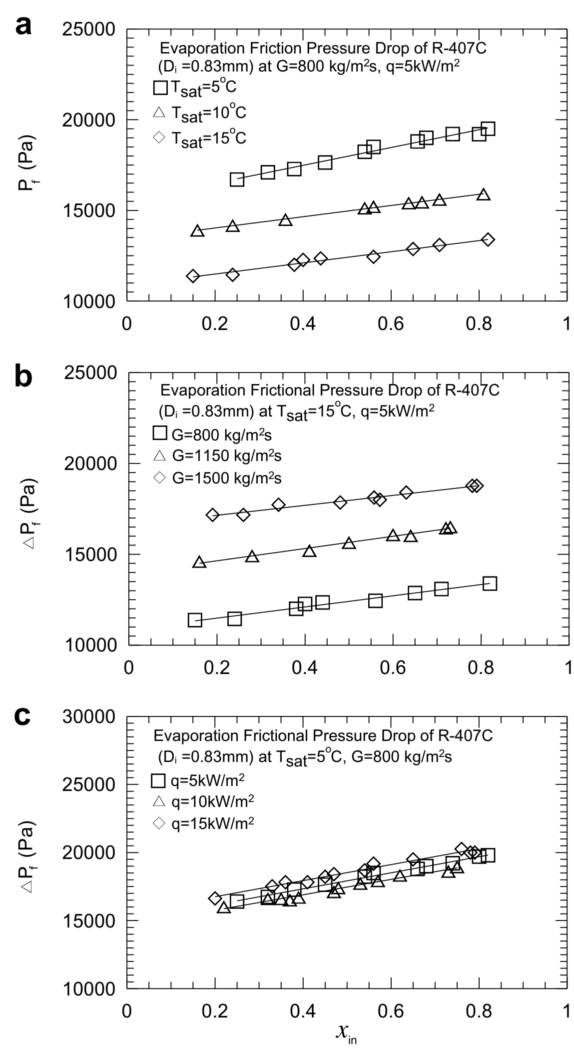

Fig. 6. Variations of R-407C evaporation frictional pressure drop with inlet vapor quality in 0.83-mm small tubes: (a) for various T_{sat} at $G = 800\ kg/m^2$ s and $q = 5\ kW/m^2$, (b) for various G at $T_{sat} = 15\ °C$ and $q = 5\ kW/m^2$, and (c) for various q at $T_{sat} = 5\ °C$ and $G = 800\ kg/m^2$ s.

smaller tubes with $D_i = 0.83$ mm are presented in Figs. 5 and 6. The results also indicate that the frictional pressure drops for both R-134a and R-407C in the smaller tubes increase significantly with the refrigerant inlet vapor quality, saturated temperature and mass flux. Besides, the frictional pressure drop for R-134a is also substantially higher than that for R-407C. Some quantitative data are given here to illustrate the effects of the important parameters on ΔP_f. For the typical cases with $T_{sat} = 15\,°C$, $G = 800\ kg/m^2\,s$ and $q = 5\ kW/m^2$ the data in Fig. 5a and Fig. 6a show that for the inlet vapor quality raised from 0.2 to 0.8, the frictional pressure drop experiences a 10% increase for R-134a and a 14% increase for R-407C. Fig. 5a and Fig. 6a also show that the quality-averaged frictional pressure drops reduce substantially for R-134a and for R-407C when T_{sat} is increased from 5 to 15 °C for $G = 800\ kg/m^2\,s$ and $q = 5\ kW/m^2$. These, respectively, correspond to 18% and 44% reductions in ΔP_f. Moreover, according to Fig. 5b and Fig. 6b, the quality-averaged frictional pressure drops increase noticeably for R-134a and for R-407C when G is raised from 800 to 1500 $kg/m^2\,s$ for $T_{sat} = 15\,°C$ and $q = 5\ kW/m^2$. These, respectively, correspond to 12% and 46% increases in ΔP_f. Furthermore, the above results clearly manifest that the refrigerant saturated temperature and mass flux exhibit more significant effects on the ΔP_f for R-407C. The frictional pressure drop is not affected to a noticeable degree by the imposed heat flux, as is evident from the results in Fig. 5c and Fig. 6c. Finally, comparing the magnitudes of ΔP_f data in Figs. 3 and 4 with that in Figs. 5 and 6 reveals that the frictional pressure drop in the smaller tubes for $D_i = 0.83$ mm is much higher than that in the larger tubes for $D_i = 2.0$ mm for both refrigerants.

4.3. Correlation equation for frictional pressure drops

Based on the present data for R-134a and R-407C evaporation in the 2.0-mm and 0.83-mm tubes, an empirical correlation for the dimensionless frictional pressure drop is proposed in terms of the friction factor. It is expressed as

$$f_{tp} = a + b \cdot Re_{eq}^{c} + d \cdot N_{conf}^{e} + f \cdot Re_{eq}^{c} \cdot N_{conf}^{e} \tag{8}$$

where Re_{eq} is the equivalent Reynolds number for the evaporating flow and is defined as

$$Re_{eq} = \frac{G_{eq} \cdot D_i}{\mu_f} \tag{9}$$

in which

$$G_{eq} = G \cdot \left[(1 - x_{in}) + x_{in} \cdot \left(\frac{\rho_f}{\rho_g} \right)^{0.5} \right] \tag{10}$$

Here G_{eq} is an equivalent mass flux at the test section inlet which is a function of the refrigerant mass flux, inlet vapor quality and density at the test condition. The values for the constants in Eq. (8) are determined from a least-square fit

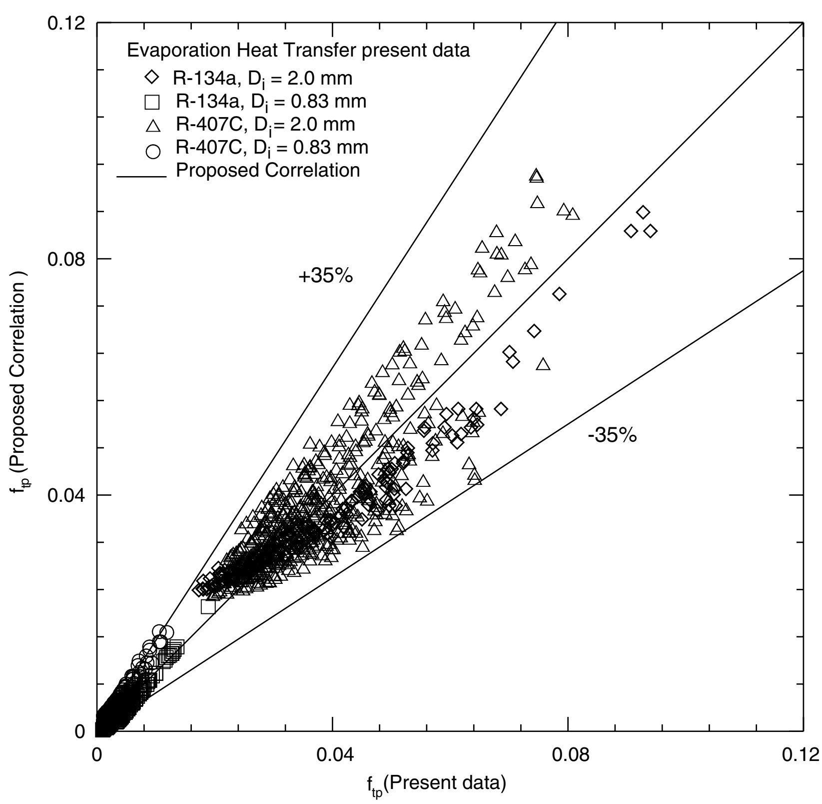

Fig. 7. Comparison of the measured data for frictional pressure drops for the evaporation of R-134a and R-407 in 0.83-mm and 2.0-mm small tubes with the proposed correlation.

of the present data and they are $a = -0.037$, $b = -147341$, $c = -1.859$, $d = 0.039$, $e = -0.508$ and $f = 327726$. Besides, x_{in} is the inlet vapor quality of the flow. Fig. 7 shows that the present data for f_{tp} fall within ±35% of Eq. (8), and the mean deviation between the present data for f_{tp} and the proposed correlation is about 19.4%.

A close inspection of the data given in Fig. 7 reveals that the friction factor f_{tp} is much smaller for the smaller tubes with $D_i = 0.83$ mm, contrary to the trend presented above for the frictional pressure drops. These opposite trends in f_{tp} and ΔP_f are simply due to the definition for f_{tp} in Eq. (6).

5. Concluding remarks

Experimental measurement has been carried out here to investigate how the frictional pressure drops of R-134a and R-407C evaporation in the small tubes are affected by the inlet vapor quality, refrigerant saturated temperature and mass flux, and imposed heat flux. The results show the significant increase of the frictional pressure drops for R-134a and R-407C evaporation in the small tubes with the inlet vapor quality and the increase is larger for a higher refrigerant mass flux. But an opposite trend is noted for a rise in the refrigerant saturated temperature. Furthermore, the imposed heat flux exhibits a negligible effect on the frictional pressure drops. Besides, we also note that on an average the frictional pressure drop for R-134a is substantially higher than that for R-407C in both tubes. Finally, an empirical correlation is proposed to correlate the present data for the frictional pressure drops in terms of the friction factor.

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