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

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Abstract

An experiment is carried out here to investigate the characteristics of the evaporation heat transfer for refrigerants R-134a and R-407C flowing in horizontal small tubes having the same inside diameter of 0.83 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 the study the effects of the refrigerant vapor quality, mass flux, saturation temperature and imposed heat flux on the measured evaporation heat transfer coefficient $h_r$ are examined in detail. The experimental data clearly show that both the R-134a and R-407C evaporation heat transfer coefficients increase almost linearly and significantly with the vapor quality of the refrigerant, except at low mass flux and high heat flux. Besides, the evaporation heat transfer coefficients also increase substantially with the rises in the imposed heat flux, refrigerant mass flux and saturation temperature. At low R-134a mass flux and high imposed heat flux the evaporation heat transfer coefficient in the smaller tubes ($D_i = 0.83$ mm) may decline at increasing vapor quality when the quality is high, due to the partial dryout of the refrigerant flow in the smaller tubes at these conditions. We also note that under the same $x_{in}$, $T_{sat}$, $G$, $q$ and $D_i$, refrigerant R-407C has a higher $h_r$ when compared with that for R-134a. Finally, an empirical correlation for the R-134a and R-407C evaporation heat transfer coefficients in the small tubes is proposed.

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1. Introduction

In the past decade following the signing of the Montreal Protocol in 1996, extensive research has been undertaken to search for the alternatives that can replace the chlorofluorocarbons (CFCs) refrigerants. It was well known at that time that the use of the CFCs refrigerants, which contain chlorine and carbon, would lead to the ozone depletion and a consequent increase in ultraviolet radiation, and would cause global warming. Some hydrochlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs) refrigerants have been developed. The HCFCs refrigerants contain less chlorine than the CFCs and have shorter atmospheric life time. They were considered as interim for the CFCs. The HFCs refrigerants have zero ozone depletion potential...
and can replace the CFCs for some period of time. In order to properly use these new refrigerants, we need to understand their thermodynamic, flow and heat transfer properties. In particular, a detailed understanding of the characteristics of the evaporation and condensation heat transfer for the HFCs refrigerants is very important and the evaporation heat transfer in the small tubes is less explored. Data for these two-phase contacting pipes \((D_i = 2.0\) mm) revealed that both the refrigerant mass flux and imposed heat flux were important and the evaporation heat transfer in the small pipes was significantly higher than that in large tubes. A similar study for subcooled flow boiling of R-134a in a vertical multiport parallel rectangular channels \((D_i = 2.01\) mm) was carried out by Agostini and Bon temps \(6\). In a visualization investigation Nino et al. \(7\) examined R-134a flow boiling in a multiport minichannel tube with \(D_h = 1.5\) mm. They proposed a method to describe the fraction of time or the probability that a flow pattern existed in a particular flow condition. A recent study from Fujita et al. \(8\) for R-123 boiling in a horizontal small tube with an inside diameter of
confined boiling occurred when...The above literature review clearly indicates that
the experimental data for the evaporation heat transfer of
the HFC refrigerants in small tubes are still in urgent
need. To complement our earlier investigations [4,5], in
this study we move further to measure the evaporation
heat transfer coefficients 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 heat transfer in the small tubes will be examined in detail.
2. Experimental apparatus and procedures

The experimental system modified slightly from that used in the previous study [4] is employed here to investigate the evaporation heat transfer 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. The 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 detailed description of the apparatus is available from our earlier study [4]. Here only the modified test section employed in the experiment is described in detail.

The modified test section along with the entry and exit sections attached to it are schematically shown in Fig. 2. Due to the tubes to be tested being relatively small, the refrigerant flow rates in them are very low and direct measurement of evaporation heat transfer coefficient in the tubes is difficult and can be subject to large error. Thus 28 small tubes all made of copper, each having the same diameter and length, are put together side by side to form a plane tube bundle acting as the test section, as shown in Fig. 2. Each small tube has the same diameter of 0.83 or 2.0 mm, outside diameter of 1.83 or 3.0 mm, and length of 150 mm. In order to allow the refrigerant to flow smoothly into the small tubes, a section including divergent, convergent and straight portions is connected to the inlets of the tubes. Besides, another section including straight and convergent parts is attached to the exits of the tubes. Both the entry and exit sections are formed by the stainless steel plates. Note that the addition of the entry and exit sections in the present study is expected to improve the flow distribution among the tubes in the bundle [5]. At the middle axial location of the small tubes 14 thermocouples are soldered onto the outer surface of the tubes. Specifically, these thermocouples are soldered onto 14 selected tubes at the circumferential position of 45° from the top of the tube or from the bottom of the tube, as shown in Fig. 3. Two copper plates of 5-mm thick are respectively soldered onto the upper and lower sides of the tube bundle also shown in Fig. 3. The copper plates are heated directly by an electric-resistance heater of 2.6-mm wide, 0.5-mm thick and 2.5-m long. The heater is connected to a 500 W DC power supply. Mica sheet is placed in the narrow space between the heater and copper plates to prevent the electric current leaking to the
copper plates. 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 evaporation heat transfer coefficient 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 [27,28].

Before a test is started, the system temperature is compared with the saturation temperature of refrigerant R-134a or R-407C corresponding to the measured saturation pressure of the refrigerant and the allowable difference is kept in the range of 0.2–0.3 K. Otherwise, the system is re-evacuated and then re-charged to remove some noncondensible gases possibly existing in the refrigerant loop. In each test the liquid refrigerant leaving the subcooler is first maintained at a specified temperature by adjusting the water-glycol temperature and flow rate. In addition, we adjust the thermostat in the water loop to stabilize the refrigerant temperature at the test section inlet. Next, the temperature and flow rate of the water loop for the preheater are adjusted to keep the vapor quality of R-134a or R-407C at the test section inlet at the desired value. Then, we regulate the refrigerant pressure at the test section inlet by adjusting the gate valve located right after the exit of the test section. Meanwhile, by changing the current of the DC motor connecting to the refrigerant pump, the refrigerant flow rate can be varied. The imposed heat flux from the heater to the refrigerant is adjusted by varying the electric current delivered from the DC power supply to the heater. By measuring the current delivered to and voltage drop across the heater, we can calculate the heat transfer rate to the refrigerant. All tests are run when the experimental system has reached statistically steady state. Finally, all the data channels are scanned every 5 s for a period of 50 s.

3. Data reduction

Before the two-phase experiments, the total heat loss from the test section is evaluated by comparing the total power input from the power supply with that calculated from the energy balance in the single phase refrigerant flow. The measured results indicated that for all runs in the energy balance test the heat loss was within 2%. The average single-phase liquid refrigerant convection heat transfer coefficient in the small tubes is defined as

$$h_l = \frac{Q_n}{A_t \cdot (T_{wall} - T_{ave})}$$  \hspace{1cm} (1)

Here $Q_n$ is the net power input to the liquid refrigerant R134a or R-407C, $A_t$ is the total inside surface area of the small tubes in the test section, $T_{wall}$ is the average of the measured tube wall temperatures at all detected locations, and $T_{ave}$ is the average refrigerant temperature in the tubes, which in turn is estimated from the measured refrigerant temperatures at the inlet and exit of the test section as $(T_{in} + T_{out})/2$.

The vapor quality of refrigerant R-134a or R-407C entering the test section is evaluated from the energy balance for the preheater. The total change of the refrigerant vapor quality in the test section is deduced from the net heat transfer rate from the heater to the refrigerant in the test section. Finally, the average heat transfer coefficient for the evaporation of refrigerant R-134a or R-407C in the test section is determined from the definition

$$h_l = \frac{Q_n}{A_t \cdot (T_{wall} - T_{satu})}$$  \hspace{1cm} (2)

More detailed description of the data reduction is available from our earlier study [4]. Uncertainties of the measured heat transfer coefficients are estimated according to the procedures proposed by Kline and McClintock [29]. The detailed results from this uncertainty analysis are summarized in Table 1.

4. Results and discussion

In what follows selected data obtained here are presented to illustrate the evaporation heat transfer of R-134a or R-407C in the horizontal small circular tubes. The present experiments are performed for refrigerant R-134a or R-407C in the tube bank forming by the 2.0-mm diameter tubes with the refrigerant mass flux $G$ varied from 200 to 400 kg/m$^2$ s$^{-1}$, 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 $T_{satu}$ from 5 to 15 °C. While for the other tube bank forming by
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 the 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 vapor quality, imposed heat flux and refrigerant mass flux and saturated temperature on the R-134a and R-407C evaporation heat transfer coefficients are to be examined in detail.

4.1. Single phase heat transfer

Before measuring the R-134a and R-407C evaporation heat transfer coefficients, single-phase liquid R-134a and R-407C convection heat transfer coefficients in the small tubes were obtained first for the refrigerant inlet temperature fixed at 15 °C and imposed heat flux of 5 kW/m$^2$ with the refrigerant mass flux respectively varied from 200 to 800 kg/m$^2$s (corresponding to $Re_l$ ranging from 1786 to 9227) and from 400 to 3000 kg/m$^2$s (corresponding to $Re_l$ ranging from 1482 to 14,360) in the 2.0-mm and 0.83-mm diameter tubes. Note that the single-phase heat transfer test is conducted here to check the energy balance in the test section and the suitability of the experimental system for the present measurement. The measured single-phase liquid R-134a and R-407C heat transfer coefficients are compared with the correlations from Dittus–Boelter [30] and Gnielinski [31] in Figs. 4 and 5.

![Graph showing comparison of single-phase heat transfer](image)

The well known Dittus–Boelter correlation is

$$Nu_l = 0.023 \cdot \frac{Re_l^{0.8} \cdot Pr_l^{0.4}}{f \cdot 2}$$

applicable for $Re_l > 10^5$ and $0.7 < Pr_l < 16,700$ and the Gnielinski correlation is

$$Nu_l = \frac{(f/2)(Re_l - 1000)Pr_l}{1.07 + 12.7\sqrt{f/2(Pr_l^{1/3} - 1)}}$$

applicable for $2300 < Re_l < 10^6$ and $0.6 < Pr_l < 10^5$, where

$$f = (1.58 \ln Re_l - 3.28)^{-2}$$

It is of interest to note from the results in Figs. 4 and 5 that for the 2.0-mm tubes the measured single-phase heat transfer coefficients are close to the Dittus–Boelter correlation. While for the 0.83-mm tubes the data are well fitted with the Gnielinski correlation. Finally, in the single-phase heat transfer tests to check the energy balance in the test section the relative heat loss is found to be within 2% for all runs. Thus the heat loss from the test section is small.

4.2. Evaporation heat transfer in 2.0-mm tubes

The measured heat transfer data for the R-134a and R-407C evaporation in the 2.0-mm tubes are presented first. Figs. 6 and 7 respectively illustrate how the refrigerant saturated temperature, mass flux and imposed heat flux affect the heat transfer coefficients for R-134a and R-407C evaporation in the 2.0-mm tubes. The measured heat transfer coefficients are examined by checking their variations with the vapor quality at the test section inlet $x_{in}$. Since the tubes are short, the total quality change $Δx$ between the inlet and exit of the test section is relatively
small, ranging from 0.01 to 0.03 in the present study. The results given in Figs. 6 and 7 show that for given $q$, $T_{\text{sat}}$, and $G$ the evaporation heat transfer coefficients for both R-134a and R-407C increase almost linearly with the inlet quality. Moreover, at higher $T_{\text{sat}}$, $q$ and $G$ the increase of $h_r$ with $x_{\text{in}}$ is more significant. The significant increase of the evaporation heat transfer coefficients with the inlet vapor quality is considered to result from the fact that at a higher inlet quality the liquid film thickness of the refrigerant on the inside surface of the tubes becomes thinner. Hence the thermal resistance of the liquid film is reduced and heat transfer across the film is improved. Besides, at a higher vapor quality the mass flux of the vapor and the velocity of vapor flow are faster. This also improves the interfacial heat transfer. More specifically for the case with R-134a at $T_{\text{sat}} = 15$ °C, $G = 400$ kg/m$^2$s and $q = 15$ kW/m$^2$, an increase of 31% in $h_r$ occurs for $x_{\text{in}}$ raised from 0.2 to 0.8 (Fig. 6(a)). The effects of each parameter on $h_r$ are examined in the following. At first, the effects of the saturated temperature $T_{\text{sat}}$ are presented in Fig. 6(a) by showing the variations of the R-134a evaporation heat transfer coefficients with the inlet vapor quality at $T_{\text{sat}} = 5, 10, 15$ °C for given refrigerant mass flux and imposed heat flux. The results in Fig. 6(a) indicate that at fixed $q$
4.3. Evaporation heat transfer in the smaller tubes

and G the R-134a evaporation heat transfer coefficients rise with the saturated temperature of the refrigerant. A similar trend is also noted by Agostini and Bonfemps [6]. This increase in \( h_i \) with \( T_{\text{sat}} \) is ascribed to the fact that at a higher \( T_{\text{sat}} \) the latent heat of vaporization \( h_l \) is lower, which in turn results in a higher evaporation rate of the liquid R-134a for a fixed \( q \). Hence the R-134a vapor in the tubes flows at a higher speed, producing a higher convection effect and therefore a higher \( h_i \). To be more quantitative on the effects of \( T_{\text{sat}} \) on the R-134a evaporation heat transfer coefficient, the quality-averaged evaporation heat transfer coefficients \( h_i \) at \( G = 400 \text{ kg/m}^2\text{s} \) and \( q = 15 \text{ kW/m}^2\text{m}^2 \) are calculated from the data in Fig. 6(a). For \( T_{\text{sat}} \) raised from 5 °C to 15 °C, \( h_i \) is increased by 19%.

Next, the effects of the refrigerant mass flux on the R-134a evaporation in the 2.0-mm tubes are shown in Fig. 6(b). The results indicate that the increase of \( h_i \) with the R-134a mass flux is rather significant, suggesting that the interfacial evaporation is effectively enhanced by the rise in the refrigerant mass flux. Hence the convection mechanism is important in the flow. Quantitatively according to the data in Fig. 6(b) for \( T_{\text{sat}} = 15 \text{ °C} \) and \( q = 15 \text{ kW/m}^2\text{m}^2 \), the quality-averaged evaporation heat transfer coefficient is increased by 27% for \( G \) raised from 200 to 400 kg/m²s for R-134a.

Then, the results presented in Fig. 6(c) indicate that the R-134a evaporation heat transfer coefficient increases rather significantly with the imposed heat flux. This significant increase of \( h_i \) with \( q \) reflects that the evaporation at the liquid–vapor interface in the refrigerant flow is substantially augmented by the increase in the imposed heat flux. According to the data in Fig. 6(c) for \( T_{\text{sat}} = 15 \text{ °C} \) and \( G = 400 \text{ kg/m}^2\text{s} \), with \( q \) raised from 5 to 15 kW/m², \( h_i \) is increased by 44%.

Checking with the heat transfer data given in Fig. 7 for R-407C, we note that the effects of the vapor quality, refrigerant saturated temperature and mass flux, and imposed heat flux on the evaporation heat transfer coefficients of R-407C are qualitatively similar to those for R-134a. A close inspection of the data in Figs. 6 and 7, however, reveals some differences. The R-407C evaporation heat transfer coefficient is noticeably higher. Besides, the effects of \( T_{\text{sat}} \), \( G \), and \( q \) on \( h_i \) for R-407C are stronger for most cases. Moreover, for R-134a the evaporation heat transfer coefficient varies with \( G \) more significantly at a high refrigerant mass flux (Fig. 6(b)). While the opposite is the case for R-407C, as evident from Fig. 7(b).

4.3. Evaporation heat transfer in the smaller tubes

\( (D_i = 0.83 \text{ mm}) \)

The measured heat transfer data for the smaller tubes with \( D_i = 0.83 \text{ mm} \) are illustrated in Fig. 8 for the R-134a evaporation and in Fig. 9 for the R-407C evaporation, covering the effects of various parameters on the evaporation heat transfer coefficients for these two refrigerants. First, it is noted from the results in Fig. 8 that for given \( T_{\text{sat}} \), \( G \), and \( q \) the R-134a evaporation heat transfer coefficient also increases noticeably with the inlet vapor quality except at high vapor quality for some cases at low \( G \) and high \( q \) (Fig. 8(b)). A close inspection of data in Fig. 8 reveals that at low mass flux and high imposed heat flux, \( h_i \) even decreases with a rise in \( x_{\text{in}} \) at a high vapor quality for \( x_{\text{in}} > 0.6 \). This is conjectured to result from the partial dryout of the refrigerant on the tube wall at high \( x_{\text{in}} \) at these conditions. In fact, the data
Evaporation heat transfer coefficient of R-407C only occurs in the smaller tubes with $D_i$ of the test section. Hence the partial refrigerant dryout has been improved by modifying the entry and exit sections. In the present study, this bad flow distribution has mass flux distribution at the inlet of the previous test section. This is apparently due to the bad refrigerant distribution in the tubes, as already mentioned. These earlier poor data result from the significant refrigerant dryout in the tubes, as already mentioned. For R-134a evaporation in the 2.0-mm tubes show the decline of $h_t$ with a rise in $x_{in}$ for many cases. These earlier poor data result from the significant refrigerant dryout in the tubes, as already mentioned. This is apparently due to the bad refrigerant mass flux distribution at the inlet of the previous test section. In the present study, this bad flow distribution has been improved by modifying the entry and exit sections of the test section. Hence the partial refrigerant dryout only occurs in the smaller tubes with $D_i = 0.83$ mm at high $x_{in}$ and $q$ and low $G$ (Fig. 8(b)). Moreover, at an intermediate quality the increase of $h_t$ with $x_{in}$ is rather large for the cases at high $T_{sat}$, $q$ and $G$. However, for these cases the increase is rather mild at low quality. Quantitatively for the case with $T_{sat} = 15 \, ^\circ\text{C}$, $G = 1500 \, \text{kg/m}^2\text{s}$ and $q = 15 \, \text{kW/m}^2$, the increase in $h_t$ is 24\% for $x_{in}$ raised from 0.2 to 0.8 (Fig. 8(a)).

According to the results in Fig. 9, the R-407C evaporation heat transfer coefficients increase almost linearly with the inlet vapor quality for most cases. The increase is also rather substantial at high $T_{sat}$, $q$ and $G$. Checking the numerical values for $h_t$ in Fig. 9 reveals the quantitative increase of $h_t$ with $x_{in}$ for R-407C. For example, at $T_{sat} = 15 \, ^\circ\text{C}$, $G = 1500 \, \text{kg/m}^2\text{s}$ and $q = 15 \, \text{kW/m}^2$, we have a much smaller $h_t$ increase of 11\% for R-407C for the same rise in $x_{in}$ (Fig. 9(a)).

An overall inspection of the data presented in Fig. 8 discloses that for R-134a evaporation in the smaller tubes with $D_i = 0.83$ mm the evaporation heat transfer in the flow can be significantly increased by raising the refrigerant saturated temperature, mass flux and heat flux. These trends are similar to that for the large tubes with $D_i = 2.0$ mm already examined above. A further inspection of the measured data for R-134a evaporation in the smaller tubes reveals that the effects of the refrigerant saturated temperature and mass flux on $h_t$ are more pronounced at a higher heat flux. Finally, the heat flux variation on $h_t$ is more important at high $T_{sat}$ and $G$.

For R-407C evaporation in the smaller tubes a substantial increase in $h_t$ with the increasing refrigerant saturated temperature, mass flux and heat flux is also noted from the results in Fig. 9. It should be mentioned that for R-407C evaporation at high vapor quality $h_t$ does not decline for a rise in $x_{in}$ unlike that for R-134a. This is attributed to the fact that R-407C has a high latent heat of vaporization and the partial dryout on the tube wall is less likely to occur in the R-407C evaporation. Moreover, in the smaller tubes the effects of $T_{sat}$, $G$ and $q$ on the evaporation heat transfer coefficients for R-407C are also stronger than that for R-134a. Again in the small tubes R-407C has a higher $h_t$.

To be more quantitative on the effects of various parameters on the heat transfer data in the smaller tubes, the quality-averaged evaporation heat transfer coefficients for various cases are evaluated. The results from this evaluation show that for R-134a at $G = 1500 \, \text{kg/m}^2\text{s}$ and $q = 15 \, \text{kW/m}^2$, $h_t$ experiences a 21\% increase when $T_{sat}$ is raised from 5 to 15 $^\circ\text{C}$ (Fig. 8(a)). While for R-407C the corresponding increase is 37\% (Fig. 9(a)). Next, we note that for R-134a at $T_{sat} = 15 \, ^\circ\text{C}$ and $q = 15 \, \text{kW/m}^2$, there is a 41\% increase in $h_t$ for $G$ raised from 800 to 1500 $\text{kg/m}^2\text{s}$ (Fig. 8(b)). The corresponding increase for R-407C is 48\% (Fig. 9(b)). Finally, a 33\% increase in $h_t$ is noted for $q$ raised from 5 to 15 $\text{kW/m}^2$ for R-134a at $T_{sat} = 15 \, ^\circ\text{C}$ and $G = 1500 \, \text{kg/m}^2\text{s}$ (Fig. 8(c)). For R-407C the corresponding increase is 67\% (Fig. 9(c)).

![Fig. 9. Variations of R-407C evaporation heat transfer coefficient with inlet vapor quality in 0.83-mm small tubes: (a) for various $T_{sat}$ at $G = 1500 \, \text{kg/m}^2\text{s}$ and $q = 15 \, \text{kW/m}^2$, (b) for various $G$ at $T_{sat} = 15 \, ^\circ\text{C}$ and $q = 15 \, \text{kW/m}^2$, and (c) for various $q$ at $T_{sat} = 15 \, ^\circ\text{C}$ and $G = 1500 \, \text{kg/m}^2\text{s}$.](image-url)
4.4. Correlation equation for evaporation heat transfer coefficients

For practical application the present data for the R-134a and R-407C evaporation in the 2.0-mm and 0.83-mm tubes need to be correlated empirically. The data presented above indicate that \( h_r \) varies linearly with the vapor quality for most cases and the correlation is thus expressed as

\[
Nu_r = \frac{h_r \cdot D_i}{k_i} = m_1 x_{in} + m_2
\]  

(6)

where \( m_1 \) and \( m_2 \) can be correlated as

\[
m_1 = f(B_o, R_e) = a_1 + b_1 B_o c_1 R_e^{d_1}
\]

(7)

\[
m_2 = f(B_o, R_e) = a_2 B_o^{b_2} R_e^{c_2}
\]

(8)

The values for the coefficients in \( m_1 \) and \( m_2 \) are \( a_1 = 1.39 \), \( b_1 = 1.87 \times 10^{-3} \), \( c_1 = 1.82 \), \( d_1 = 3.14 \), \( a_2 = 28.6 \), \( b_2 = 0.706 \), \( c_2 = 0.888 \). The Boiling number and Reynolds number are defined respectively as

\[
B_o = \frac{q}{G \cdot \kappa}
\]

(9)

\[
R_e = \frac{G \cdot D_i}{\mu_1}
\]

(10)

Comparison of the above correlation with the present experimental data shown in Fig. 10 indicates that more than 80% of the present data for \( h_r \) fall within ±35% of Eq. (6), and the mean absolute error (MAE) between the present data for \( h_r \) and the proposed correlation is 19%.

5. Concluding remarks

Experiments have been conducted here to investigate the evaporation heat transfer of R-134a and R-407C in the small tubes with \( D_i = 0.83 \) and 2.0 mm. The effects of the refrigerant saturated temperature, mass flux, imposed heat flux, and vapor quality of R-134a and R-407C on the evaporation heat transfer coefficients have been examined in detail. The results show that the R-134a and R-407C evaporation heat transfer coefficients in the small tubes increase almost linearly with the vapor quality and the increases are significant except at low imposed heat flux and low refrigerant mass flux. Moreover, the increases of the R-134a and R-407C evaporation heat transfer coefficients in both tubes with the imposed heat flux, refrigerant mass flux and saturated temperature are also substantial. Besides, the evaporation heat transfer coefficients for R-134a are noticeably lower than that for R-407C at the same \( T_{sat}, G \) and \( q \). Furthermore, for R-134a in the smaller tubes (\( D_i = 0.83 \) mm) partial refrigerant dryout may occur, resulting in the decline of the evaporation heat transfer coefficient at increasing inlet vapor quality at high \( x_{in} \). This is normally seen at high imposed heat flux and saturated temperature and low mass flux. Finally, an empirical correlation is proposed to correlate the present data for R-134a and R-407C evaporation heat transfer coefficients in the small tubes.

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References


