Evaporation Heat Transfer and Pressure Drop of Refrigerant R-410A Flow in a Vertical Plate Heat Exchanger

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Experiments are carried out here to measure the evaporation heat transfer coefficient h_r and associated frictional pressure drop ΔP_f in a vertical plate heat exchanger for refrigerant R-410A. The heat exchanger consists of two vertical counterflow channels which are formed by three plates whose surface corrugations have a sine shape and a chevron angle of 60 deg. Upflow boiling of refrigerant R-410A receives heat from the hot downflow of water. In the experiments, the mean vapor quality in the refrigerant channel is varied from 0.10 to 0.80, the mass flux from 50 to 100 kg/m^2s, and the imposed heat flux from 10 to 20 kW/m^2 for the system pressure fixed at 1.08 and 1.25 MPa. The measured data indicate that both h_r and ΔP_r increase with the refrigerant mass flux except at low vapor quality. In addition, raising the imposed heat flux is found to significantly improve h_r for the entire range of the mean vapor quality. However, the corresponding friction factor f_{tp} is insensitive to the imposed heat flux and refrigerant pressure. Based on the present data, empirical correlations are provided for h_r and f_{tp}, for R-410A in the plate heat exchanger. [DOI: 10.1115/1.1518498]

Keywords: *Boiling, Evaporation, Heat Transfer, Heat Exchangers*

1 Introduction

Over the past decades the HCFC (hydrochlorofluorocarbon) refrigerant R-22 has been used as the working fluid in many refrigeration and air-conditioning systems. But it will be phased out in a short period of time (before 2020) since the chlorine it contains has an ozone depletion potential (ODP) of 0.055 and comparatively high global warming potential (GWP) of 1500 based on the time horizons of 100 years [1,2]. As a result, the search for a replacement for R-22 has been intensified in recent years. Owing to the fact that there are no single-component hydrofluorocarbons (HFCs) that have thermodynamic properties close to those of R-22, binary or ternary refrigerant mixtures have been introduced. The technical committee for the Alternative Refrigerants Evaluation Program (AREP) has proposed an updated list of the potential alternatives to R-22 [2]. Some of the alternatives on the AREP's list are R-410A, R-410B, R-407C and R-507. Currently, R-134a is extensively used in many systems. A number of investigations have been reported in the literature dealing with the phase change heat transfer of R-134a in ducts of various geometries. However, the two-phase boiling and condensation heat transfer characteristics for R-410A, R-410B, and R-407C have not been studied extensively. It should be mentioned here that refrigerant R-410A is a mixture of R-32 and R-125 (50 percent by mass) which exhibits azetropic behavior with a temperature glide of about 0.1°C.

A number of studies have been reported in the open literature on the R-22 evaporation heat transfer in various enhanced ducts such as microfin tubes [3,4], internally-fin tubes [5], and axially grooved tubes [6]. The measured data were compared with other common refrigerants. Recently, Sami and Poirier [7] compared the evaporation and condensation heat transfer data inside an enhanced tubing for several refrigerant blends proposed as substitutes for R-22, including R-410A, R-410B, R-507 and the quanternary mixture R-32/125/143a/134a. They showed that the two phase heat transfer coefficient and pressure drops all increased with the refrigerant mass flux. In a continuing study [8] they showed that in a double fluted tube for the refrigerant Reynolds number higher than 4.2×10^6, R-410A had better evaporation heat transfer than R-507. Wang et al. [9] compared the measured data for R-22 and R-410A flowing in a horizontal smooth tube and indicated that the heat transfer coefficients for R-410A were 10–20 percent higher than those for R-22 and the pressure drop of R-410A was about 30–40 percent lower than that of R-22. This outcome was attributed to the higher latent heat of vaporization, thermal conductivity and specific heat, and lower liquid viscosity for R-410A. In a similar study Ebisu and Torikoshi [10] indicated that the evaporation heat transfer coefficient of R-410A was 20 percent higher than that of R-22 up to the vapor quality of 0.4, while the heat transfer coefficients for both R-410A and R-22 became almost the same at the quality of 0.6. Furthermore, the pressure drop for R-410A was about 30 percent lower than that of R-22 during evaporation. The quantitative differences in the pressure drops between R-410A and R-22 were mainly attributed to the lower vapor density for R-410A. Wijaya and Spatz [11] reached a similar conclusion.

Plate heat exchangers (PHEs) have been widely used in food processing, chemical reaction processes and many other industrial applications due to their high effectiveness, compactness, flexibility, and cost competitiveness. Furthermore, they have been introduced to the refrigeration and air conditioning systems as evaporators or condensers. Recently, a number of investigations on PHEs were reported in the open literature. Unfortunately, these studies were mainly focused on the single-phase liquid-to-liquid heat transfer [12–15]. In view of this scarcity in the two-phase heat transfer data for PHEs, Yan and Lin [16] recently investigated the evaporation of R-134a in a vertical plate heat exchanger. They showed that that evaporation heat transfer for R-134a flowing in the PHE was much higher than that in circular tubes, particularly in high vapor quality convection dominated regime. Both the evaporation heat transfer coefficient and pressure drop increased

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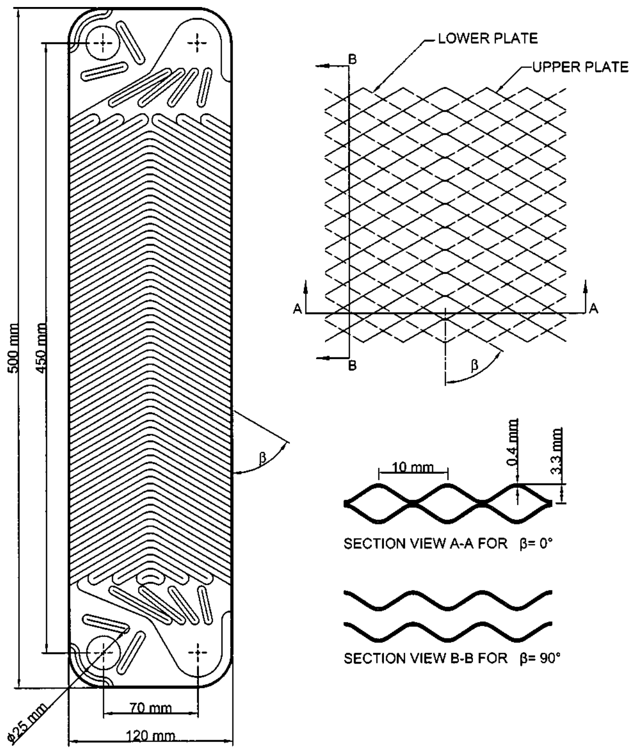

Fig. 1 Schematic diagram of the plate

with the refrigerant mass flux and vapor quality. Moreover, the rise in the heat transfer coefficient with the quality was larger than that in the pressure drop.

The above literature review clearly reveals that although R-410A is one of the most possible substitutes for R-22, the two-phase heat transfer data for R-410A are still scarce especially for PHEs. To complement our previous study on the two-phase heat transfer in the PHE [16], the evaporation heat transfer of R-410A in a vertical PHE is investigated in this study.

2 Experimental Apparatus and Procedures

The experiment apparatus established previously to explore the R-134a evaporation in a PHE [16] is used here to investigate the evaporation heat transfer and associated frictional pressure drop of R-410A in a vertical PHE. It includes a refrigerant loop, two water loops (one for preheater and the other for the test section), and a cold water-glycol loop. R-410A is circulated in the refrigerant loop. In order to control various test conditions of R-410A in the test section, we need to control the temperature and flow rate in the other three loops. The detailed description of the apparatus is available from our earlier study [16]. The refrigerant flow rate is measured by an accurate mass flux meter manufactured by Mircomotion (Type UL-D-IS) with a reading accuracy of ± 1 percent. Here only the test section employed in the experiment is described in some detail.

Three commercial SS-316 plates manufactured by the Kaori Heat Treatment Co. Ltd., Taiwan, form the plate heat exchanger (test section). The plate surfaces are pressed to become grooved with a corrugated sinusoidal shape and 60 deg of chevron angle β. The detailed configuration of the PHE can be seen in Fig. 1. The corrugated grooves on the right and left outer plates have a V shape but those in the middle plate have a contrary V shape on both sides. This arrangement allows the flow stream to be divided into two different flow directions along the plates. Thus, the flow moves mainly along the grooves in each plate. Due to the contrary V shapes between two neighbor plates the flow streams near the two plates cross each other in each channel. This cross flow results in significant flow unsteadiness and randomness. In the PHE the upflow of R-410A in one channel is heated by the downflow of the hot water in the other channel. The heat transfer rate in the test section is calculated by measuring the total water temperature drop in the water channel and the water flow rate.

In each test the system pressure at the test section is first maintained at a specified level. Then, the vapor quality of R-410A at the test section inlet is kept at the desired value by adjusting the temperature and flow rate of the hot water loop for the preheater. Next, the heat transfer rate between the counterflow channels in the test section can be varied by changing the water temperature and flowrate in the water loop for the tests section. Meanwhile, the R-410A mass flow rate in the test section is maintained at a desired value.

In the test any changes of the system variables will lead to fluctuations in the temperature and pressure of the refrigerant flow. It takes about 20–100 minutes for the system to reach a statistically stable state at which variations of the time-average inlet and outlet temperatures are both less than $\pm 0.2°C$, and the variations of the pressure and imposed heat flux are within 1 percent and 4 percent, respectively. Then the data acquisition unit is initiated to scan all the data channels for ten times in 50 sec. The mean value of the data for each channel is used to calculate the evaporation heat transfer coefficient and the associated frictional pressure drop. Additionally, the flow rate of water in the test section should be high enough to have turbulent flow in the water side so that the associated single-phase heat transfer in it is high enough for balancing the evaporation heat transfer in the refrigerant side.

Before examining the R-410A evaporation heat transfer characteristics, a preliminary test for single-phase water-to-water convective heat transfer in the vertical PHE is performed. The Wilson's method [17] is adopted to calculate the relation between the single-phase heat transfer coefficient and the flow rate from these data. The result obtained can then be used to analyze the data acquired from the evaporation heat transfer experiments.

The uncertainties of the experimental results are analyzed by the procedures proposed by Kline and McClintock [18]. This analysis indicates that the uncertainties for the data of the imposed heat flux q, mass flux G, pressure P, pressure drop ΔP, average vapor quality X_m single phase heat transfer coefficient h_w, evaporation heat transfer coefficient h_r, and friction factor f_{tp} are respectively ± 6.5 percent, ± 2 percent, ± 1 percent, ± 1.5 percent, ± 8 percent, ± 11 percent, ± 14.5 percent, and ± 16.5 percent.

3 Data Reduction

The data reduction analysis detailed in our earlier study for R-134a evaporation [16] is also used here to deduce the R-410A evaporation heat transfer coefficient and associated frictional pressure drop from the measured raw data. Specifically, the data from the single-phase water-to-water heat transfer tests are analyzed by the modified Wilson plot [17]. The single phase convection heat transfer coefficient in a PHE, suggested by Shah and Focke [12], can be expressed empirically as

$$h_w = C \cdot \left(\frac{k}{D_h}\right) \cdot \mathrm{Re}^n \cdot \mathrm{Pr}^{1/3} \cdot \left(\frac{\mu}{\mu_{\mathrm{wall}}}\right)^{0.14} \quad (1)$$

Here the constants C and n can be determined from the Wilson plot.

To evaluate the evaporation heat transfer coefficient of the refrigerant flow, the total heat transfer rate Q_w between the counter flows in the PHE is calculated first from the hot water side. Then, the refrigerant vapor quality at the test section inlet is evaluated from the energy balance for the preheater. The change in the refrigerant vapor quality in the test section is then deduced from the total heat transfer rate to the refrigerant in the test section,

$$\Delta X = \frac{Q_w}{W_r \cdot i_{fg}} \quad (2)$$

The determination of the overall heat transfer coefficient for the evaporation of refrigerant R-410A in the PHE is based on the heat transfer between the counter-flow channels and is expressed as

$$U=\frac{Q_w}{A\cdot \mathrm{LMTD}} \tag{3}$$

The log-mean temperature difference (LMTD) is determined from the inlet and exit temperatures in the two channels. According to the thermal resistances for heat transfer across the channel, the evaporation heat transfer coefficient in the flow of R-410A is evaluated from the equation

$$\left(\frac{1}{h_r}\right)=\left(\frac{1}{U}\right)-\left(\frac{1}{h_w}\right)-R_{\mathrm{wall}}\cdot A \tag{4}$$

where h_w is calculated from the single-phase water-to-water heat transfer test.

In order to obtain the friction factor associated with the R-410A evaporation in the refrigerant channel in the vertical PHE, the frictional pressure drop ΔP_f is calculated by subtracting the acceleration pressure drop, the pressure losses at the test section inlet and exit manifolds and ports, and the elevation pressure rise from the measured total pressure drop in the refrigerant channel. The acceleration pressure drop and elevation pressure rise are estimated by the homogeneous model for two phase gas-liquid flow [19]. The pressure drop in the inlet and outlet manifolds and ports was empirically suggested by Shah and Focke [12]. Based on the above estimation, the acceleration pressure drop and the pressure losses at the test section inlet and exit manifolds and ports are found to be rather small. In fact, the summation of these two pressure losses ranges from 1 percent to 3 percent of the total pressure drop. The pressure drop due to the elevation difference between the inlet and outlet ports of the PHE is smaller than 1 percent of the total pressure drop. According to the definition

$$f_{tp}=-\frac{\Delta P_f\cdot D_h}{2G^2\cdot v_m\cdot L} \tag{5}$$

the friction factor for the evaporation of R-410A in the PHE is obtained

4 Results and Discussion

The single phase water-to-water convection heat transfer coefficient for the present vertical plate heat exchanger deduced from the modified Wilson plot can be correlated as

$$h_w=0.2092\cdot\left(\frac{K_1}{D_h}\right)\cdot \mathrm{Re}^{0.78}\cdot \mathrm{Pr}^{1/3}\cdot\left(\frac{\mu_m}{\mu_{\mathrm{wall}}}\right)^{0.14} \tag{6}$$

with the regression accuracy of 0.997. Here the viscosities μ_m and μ_{wall} are, respectively, based on the average bulk water and wall temperatures estimated by averaging the measured inlet and outlet temperatures in the hot and cold sides. The present single-phase heat transfer data well agree with Eq. (18) in the study of Muley and Manglik [13], as is clear from Fig. 2.

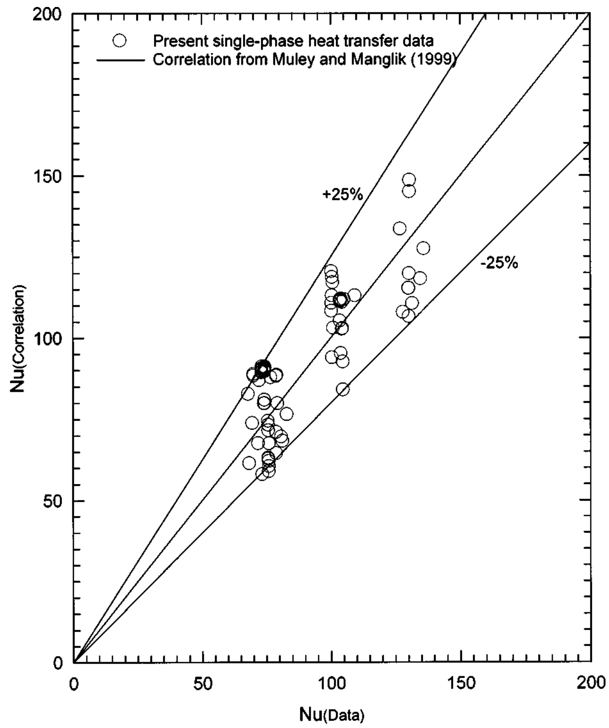

Fig. 2 Comparison of the present single-phase water convection heat transfer data with the correlation from Muley and Manglik [13]

Effects of the refrigerant mass flux, imposed heat flux and system pressure on the evaporation heat transfer of R-410A in the vertical PHE are illustrated in Fig. 3 by presenting the changes of the R-410A evaporation heat transfer coefficient with the refrigerant vapor quality at the imposed heat fluxes $q=10$ and 20 kw/m^2 for refrigerant mass fluxes $G=50$ to 100 kg/m^{2}s, system pressures $P=1.08$ and 1.25 MPa ($T_{\mathrm{sat}}=10°$C and 15°C). In these plots X_m denotes the mean R-410A vapor quality in the PHE. The total change in the vapor quality ΔX in the test section for the present study ranges from 0.126 and 0.337. The data in Fig. 3 clearly indicate that a given heat flux, mass flux and system pressure the evaporation heat transfer coefficient increases noticeably with the mean vapor quality of the refrigerant in the PHE. For example, at $G=75$ kg/m^{2}s, $P=1.08$ MPa, and $q=10$ kW/m^2 the evaporation heat transfer coefficient at X_m of 0.8 is about 60 percent higher than that at 0.1 (Fig. 3(a)).This significant increase of h_r with X_m obviously results from the fact that in the high vapor quality regime, intense evaporation at the liquid-vapor interface diminishes the thickness of the liquid film on the plate surface to a noticeable degree. This, in turn, reduces the resistance of heat transfer from the channel surface to the refrigerant. Furthermore, the data also show that a rise in the refrigerant mass flux always produces an evident increase in the evaporation heat transfer coefficient except at the low vapor quality regime. In fact, at low vapor quality ($X_m<.20$) the evaporation heat transfer coefficient is insensitive to the refrigerant mass flux. This can be ascribed to the fact that the interfacial evaporation of the liquid film on the plate is largely suppressed at low vapor quality. Moreover, the evaporation heat transfer coefficient for the higher mass flux rises more quickly with the vapor quality than that for the lower mass flux. This larger increase in h_r with X_m at a higher G is considered to result from the more intense turbulence in the flow for a higher G. Similar results were noted for other system pressures. The results in Fig. 3 further show that the evaporation heat transfer coefficient increases significantly with the imposed heat flux for both pressures. For example, at $G=75$ cg/m^{2}s and $P=1.08$ MPa the quality-average evaporation heat transfer coefficients at 20 kW/m^2 is about 32 percent higher than that at 10 KW/m^2. This large increase in the evaporation heat transfer coefficient is ascribed to the higher wall superheat and thinner liquid film on the plate surface for a higher imposed heat flux. It is also noted that the evaporation heat transfer is slightly better at the lower pressure for a higher vapor quality ($X_m>0.6$). At the low vapor quality for $X_m<0.5$ the effects of the pressure on the evaporation heat transfer is insignificant except for $q=20$ kW/m^2 and $G=100$ kg/m^{2}s (Fig. 3(b)). This is attributed to the fact that the density of the R-410A vapor is lower at a lower saturated pressure, which causes the vapor flow to move in a higher speed and hence the higher evaporation heat transfer coefficient.

We also compare the present data for the R-410A evaporation heat transfer in the PHE with those for R-134A in the same PHE

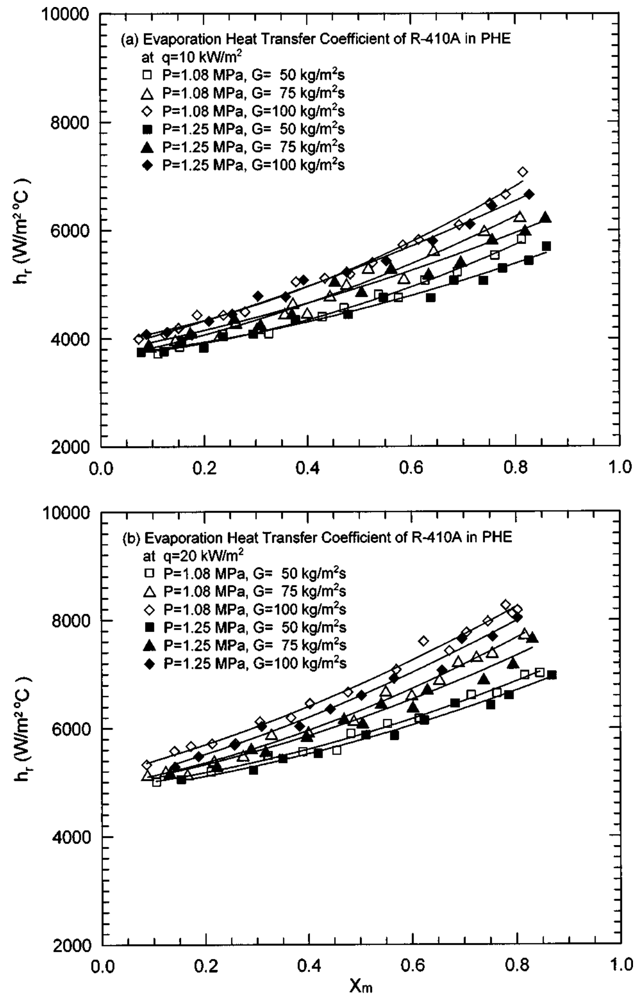

Fig. 3 Variations of evaporation heat transfer coefficient with mean vapor quality for various system pressures and refrigerant mass fluxes at (*a*) $q=10$ kW/m^2 and (*b*) $q=20$ kW/m^2

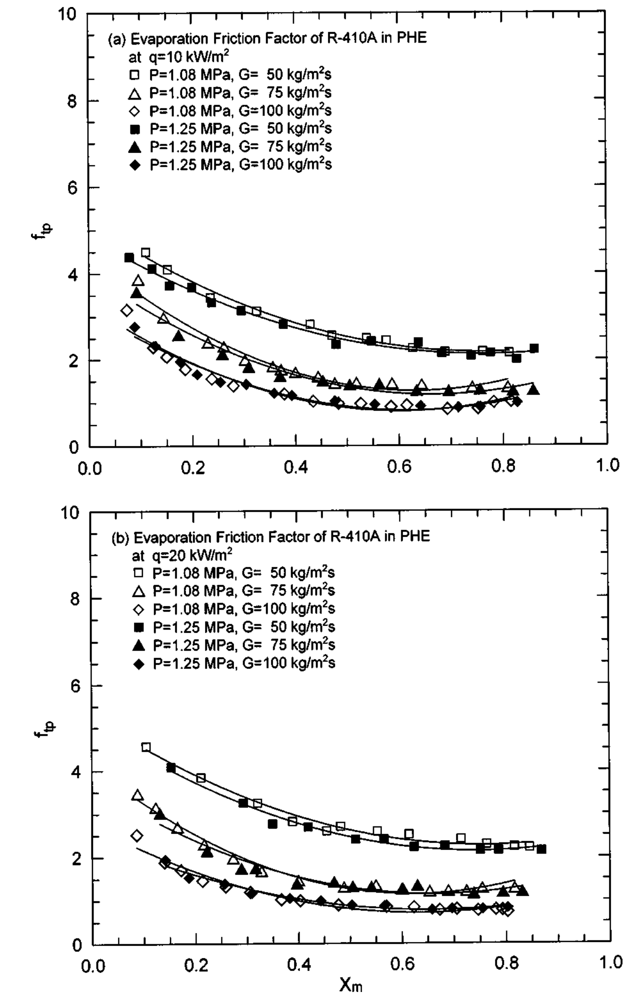

Fig. 4 Variations of friction factor with mean vapor quality for various system pressures and refrigerant mass fluxes at (*a*) $q=10$ kW/m^2 and (*b*) $q=20$ kW/m^2

reported by Yan and Lin [16] and with those for R-410A in a horizontal smooth pipe collected by Ebisu and Torikoshi [10]. This comparison indicates that the R-410A evaporation heat transfer coefficient is higher than that for R-134a in the PHE to a noticeable degree except at high vapor quality for $X_m>0.75$. More specifically, at high X_m the R-134a evaporation is more effective. These opposite trends in different vapor quality ranges are attributed mainly to the different thermal conductivities of the two refrigerants for the liquid and vapor phases. Specifically, the liquid thermal conductivity for R-410A is higher than that for R-134a by about 20 percent. However, the vapor thermal conductivity for R-410A is lower than that for R-134a. Hence, at lower quality the evaporation heat transfer coefficient for R-410A is higher than that for R-134a. The results suggest that the evaporation heat transfer in the PHE is dominated by the heat transfer associated with the liquid film evaporation. The comparison also shows that for R-410A the evaporation heat transfer coefficient in the PHE is substantially higher than that in a horizontal smooth tube.

The friction factor, defined in Eq. (5), associated with the R-410A evaporation in the PHE obtained in the present study are presented in Fig. 4. The results indicate that the friction factor significantly decreases with the increase in the refrigerant mass flux. For example, at $P=1.08$ MPa, $G=100$ kg/m^{2}s, and $q=10$ kW/m^2, the quality-average fraction factor is respectively about 50 percent and 30 percent lower than those for $G=50$ and 75 kg/m^{2}s (Fig. 4(*a*)). Besides, the friction factor decreases significantly with the increase in the mean vapor quality at low vapor quality ($X_m<0.4$). At high vapor quality for $X_m<0.5$, the effect of the vapor quality on the friction factor is insignificant. It is of interest to note that the friction factor is not affected to a noticeable degree by the imposed heat flux and refrigerant pressure. We also note that for $X_m>0.5$ the frictional pressure drop of R-410A evaporation in the PHE is substantially lower than that for R-134a in the same PHE, but is much higher than in a smooth horizontal pipe for all X_m. The lower frictional pressure drop for R-410A evaporation can be attributed to the fact that the densities and viscosities of R-410A vapor and liquid are lower than those of R-134a.

Correlation equations for the heat transfer coefficient and friction factor associated with the R-410A evaporation in the vertical PHE are important for thermal design of evaporators in various air conditioning and refrigeration systems. Based on the present data, an empirical correlation for the evaporation heat transfer coefficient is proposed by considering the convective and nucleate boiling contributions [20]. It is expressed as

$$h_r=E\cdot h_1+S\cdot h_{\mathrm{pool}}, \quad \text{for} \quad 2000<\mathrm{Re}<12{,}000$$
$$\text{and} \quad 0.0002<\mathrm{Bo}<0.0020 \tag{7}$$

Here h_1 and h_{pool} are respectively given by the Dittus-Boelter Eq. [21] and Cooper [22] as

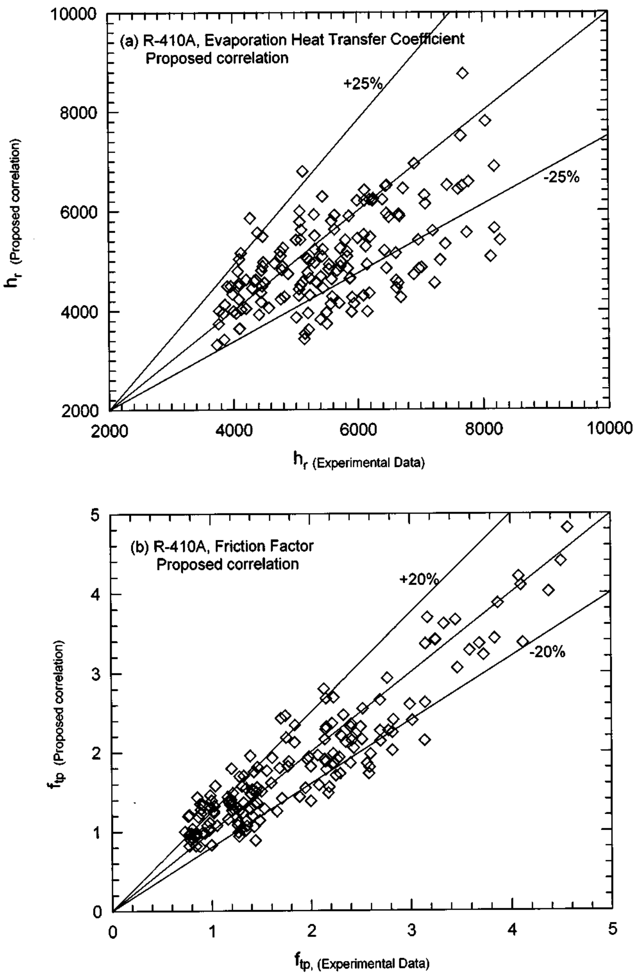

Fig. 5 Comparison of the proposed correlations with the present data for (*a*) the heat transfer coefficient and (*b*) the friction factor

$$h_1 = 0.023 \cdot \mathrm{Re}_1^{0.8} \cdot \mathrm{Pr}^{0.4} \cdot (k_1/D_h) \tag{8}$$

$$h_{\text{pool}} = 55 \cdot P_r^{0.12} \cdot (-\log_{10} P_r)^{-0.55} \cdot M^{-0.5} \cdot q^{0.67} \tag{9}$$

Besides, E and S respectively represent the enhancement and suppression factors, which are dependent on the boiling number Bo, the Martnelli parameter X_{tt} and liquid Reynolds number Re_1. The expressions for E and S are

$$E = 1 + 24{,}000 \cdot \mathrm{Bo}^{1.16} + 1.37\left(\frac{1}{X_{tt}}\right)^{0.86} \tag{10}$$

$$S = (1 + 1.15 \times 10^{-6} \cdot E^2 \cdot \mathrm{Re}_1^{1.17})^{-1} \tag{11}$$

The friction factor is correlated as

$$f_{tp} = 23{,}820 \cdot \mathrm{Re}_{\text{eq}}^{-1.12}, \quad \text{for} \quad 2000 < \mathrm{Re} < 12{,}000$$

$$\text{and} \quad 0.0002 < \mathrm{Bo} < 0.0020 \tag{12}$$

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

$$\mathrm{Re}_{\text{eq}} = \frac{G_{\text{eq}} \cdot D_h}{\mu_1} \tag{13}$$

in which

$$G_{\text{eq}} = G\left[(1 - X_m) + X_m \cdot \left(\frac{\rho_1}{\rho_g}\right)^{1/2}\right] \tag{14}$$

Here G_{eq} is an equivalent mass flux which is a function of the R-410A mass flux, mean vapor quality and densities at the saturated conditions. Figure 5 shows that more than 74 percent of the present experimental data for h_r fall within ± 25 percent of Eq. (7), and the average deviation between the present data for f_{tp} and the proposed correlation is about 18 percent.

5 Concluding Remarks

Experiments have been carried out here to investigate the evaporation heat transfer and the associated frictional pressure drop for the ozone friendly refrigerant R-410A in a vertical plate heat exchanger. The effects of the refrigerant mass flux, imposed heat flux, system pressure and vapor quality of R-410A on the evaporation heat transfer coefficient and friction factor were examined in detail. A summary of the major findings is given in the following.

1. The evaporation heat transfer coefficient and frictional pressure drop normally increases with the refrigerant mass flux and vapor quality. It is also noted that the evaporation heat transfer coefficient is only slightly affected by the refrigerant mass flux at low vapor quality. Furthermore, the increase of the frictional pressure drop with the vapor quality is more evident than the rise of the heat transfer.
2. A rise in the imposed heat flux results in a significant increase in the evaporation heat transfer coefficient. Nevertheless the influences of the imposed heat flux and system pressure on the friction factor are rather slight.
3. Empirical correlation for the R-410A evaporation heat transfer coefficient and friction factor in the PHE were provided to facilitate the design in various thermal systems.

Acknowledgments

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Nomenclature

- A = heat transfer area of the plate, m^2
- Bo = boiling number, Bo$= q/G \cdot i_{fg}$, dimensionless
- c_p = specific heat, J/kg°C
- D_h = hydraulic diameter, $D_h = 2b, m$
- E, S = enhancement and suppression factors
- f_{tp} = two-phase friction factor
- G = refrigerant mass flux, kg/m^{2}s
- G_{eq} = equivalent all liquid mass flux in Eqs. (13, 14)
- h = heat transfer coefficient, W/m^2°C
- i_{fg} = enthalpy of evaporation, J/kg
- k = conductivity, W/m°C
- L = plate length from center of inlet port to center of exit port, m
- LMTD = log mean temperature difference, °C
- M = molecular weight
- Nu = Nusselt number, Nu$= h_w \cdot D_h / k_1$, dimensionless
- P = pressure, Pa
- Pr = Prandtl number, Pr$= \mu \cdot c_p / k$, dimensionless
- Q_w = total heat transfer rate, W
- q = imposed heat flux, W/m^2
- R_{wall} = thermal resistance of the wall
- Re = Reynolds number, Re$= G \cdot D_h / \mu$, dimensionless
- Re_{eq} = equivalent all liquid Reynolds number in Eqs. (12, 13)
- U = overall heat transfer coefficient, W/m^2°C
- v_m = specific volume of the vapor-liquid mixture, m^3/kg
- W = mass flow rate, kg/s
- X_m = mean vapor quality
- X_{tt} = Martinelli parameter, dimensionless

Greek Symbols

ΔP_f = frictional pressure drop
ΔX = total quality change in the exchanger
β = chevron angle
μ = viscosity, Ns/m^2
ρ = density, kg/m^3

Subscripts

g = vapor phase
l = liquid phase
m = mean value for the two-phase mixture in the exchanger
pool = pool boiling
r = reduced, refrigerant
w = water
wall = wall/fluid near the wall

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