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Critical heat flux of R-407C in upflow boiling in a vertical pipe

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Abstract

The critical heat flux (CHF) in forced convective upflow has been investigated in a uniformly heated vertical tube of 12.7 mm internal diameter and 3 m length for different reduced pressures ranging from 0.32 to 0.98, with non-azeotropic ternary refrigerant mixture R-407C as the working fluid. The onset of CHF was determined by the sudden rise in the wall temperature of the electrically heated tube. Experiments were performed over a wide range of parameters: mass flux values from 200 to 2000 kg/m^2 s, pressure from 15 to 45.6 bar and heat flux from 5 to 80 kW/m^2. The results show considerably lower critical heat flux at higher pressures as in the case of pure fluids. Finally, a comparison of the experimental data with the available pure fluid correlations has been performed.

Keywords: R407C; Critical Heat Flux; Vertical Pipe; Upflow

1. Introduction

Global concern over damage to the ozone layer has resulted in the replacement of chlorine containing refrigerants (CFCs and HCFCs) with non-chlorinated alternatives such as the hydrofluorocarbons (HFCs) [1–3]. In some cases, binary and ternary refrigerant mixtures have been employed since the required properties could be easily obtained. While it would be preferable to have azeotropes as replacements, it is rather difficult to find such mixtures in practice. Hence, many non-azeotropic refrigerant mixtures (NARMs) are currently being used. In addition to satisfying the basic requirements, NARMs may also improve the system performance by means of their gliding temperature during phase change [4,5]. The fluid chosen for the present study, R-407C is a 23/25/52 wt% ternary mixture of R-32, R-125 and R-134a. This offers a close match to the widely used refrigerant R-22 in terms of thermodynamic properties [6–8] allowing the utilization of this mixture in existing machinery without major modifications.

Studies on the boiling of NARMs under vertical forced flow are scarce. A knowledge of the pressure drop, heat transfer coefficient and parametric behavior can reduce costs by avoiding both under-design and over-design of evaporators, boilers and other two-phase process equipments. In addition, the critical heat flux (CHF) information is essential as this limiting condition forms an important boundary when considering the performance of heat exchange equipment in which evaporation is occurring. The CHF condition is characterized by a sharp reduction of the local heat transfer coefficient which results from the replacement of the liquid adjacent to the heat transfer surface by vapor. For the case where the surface heat flux is the independent variable, the condition manifests itself by a sharp increase in surface temperature as the critical heat flux value is reached. Likewise, a considerably reduced heat flux will result when the condition is reached using a temperature-controlled heating surface.

There are basically two classes of CHF, namely departure from nucleate boiling (DNB) and dry out (DO). While DNB occurs at low qualities or lower fluid enthalpy, DO is generally experienced at high qualities or higher fluid

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Nomenclature

C_p	specific heat capacity (kJ/kg K)
d	diameter (m)
G	mass flux (kg/m^2 s)
g	acceleration due to gravity (m/s^2)
h	enthalpy (kJ/kg)
I	current (A)
L	length of the tube (m)
m	mass flow rate (m/s)
P	absolute pressure (bar)
Pr	Prandtl number
P_R	reduced pressure
Q	heat supplied (W)
q	heat flux (kW/m^2)
Re	Reynolds number
T	temperature (°C)
V	voltage (V)
We	Weber number
x	quality
z	location (m)
ΔT_{sub}	subcooling (°C)

Greek symbols

μ	viscosity of liquid (Ns/m^2)
λ	latent heat of evaporation (kJ/kg)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

cr	critical
CHF	critical heat flux
Exp	experimental
imp	imposed
l	liquid
Pre	predicted
R	reduced pressure
s	saturation
v	vapor
0	corresponding to zero quality

enthalpies where nucleation is suppressed and the flow pattern is likely to be annular [9]. The term 'dryout' is used to imply the drying out of the liquid film that causes the CHF condition and is the subject of present study.

While CHF has been studied extensively for pure fluids, very few studies on the CHF of mixtures in forced convective boiling have been reported. Earlier experimental studies reported by Collier and Thome [9] on CHF in forced flow with binary mixtures were mainly related to low exit quality conditions, and did not include refrigerant mixtures. A non-linear behavior of CHF was observed for both non-azeotropic and azeotropic mixtures. Auracher and Marroquin [10] measured the CHF for an R-13B1/R-114 mixture flowing in an electrically heated vertical stainless steel tube. The experimental results showed, in effect, a linear behavior of CHF with varying mixture composition. Thus, the literature reports present apparently contradictory results on the dependence of CHF on mixture composition.

Celata et al. [11] investigated the CHF in forced convective upflow at different compositions of binary mixtures of R-12 and R-114. Their experiments showed nearly linear dependence of the critical heat flux on the composition of the mixture even though deviations from linearity were detected as a function of thermal hydraulic conditions. Their study focused on the CHF in the annular flow regime, and on the relationship between the mixture composition and the onset of the boiling crisis. A linear dependence of CHF on the inlet mole fraction was observed in the case of DNB type crisis, while this dependency was negligible under the DO conditions.

Auracher and Marroquin [12] studied the effects of length of the heated section, local quality or subcooling and mole fraction on CHF and minimum heat flux of film boiling (MHF) for the binary refrigerant mixture R-13B1/R-114. Their investigation was limited to CHF caused by DNB.

Most of the correlations and models that predict the CHF in forced convective boiling have been developed for pure fluids and are limited to low pressures. Recently, Vijayarangan et al. [13] conducted experimental studies to obtain CHF data for R-134a over a wide range of pressures approaching the critical conditions. This study discusses the four principal approaches available for the prediction of CHF and presents a new correlation, a modified version of the existing correlation by Katto and Ohno [14]. Hence, this study provides CHF information at near critical pressures that has been lacking in the literature. It should be noted that the literature mentioned above only deal with pure fluids and binary mixtures, and CHF data for ternary mixtures such as R-407C has not been reported so far.

The objective of the present study is to obtain CHF data for R-407C in vertical upflow over a range of system pressure, mass flux and heat flux conditions with a fixed inlet subcooling. Since the CHF of mixtures at high qualities is independent of the inlet mole fraction of the components, an attempt is made to obtain the CHF data for ternary refrigerant mixture R-407C considering it to be an equivalent pure fluid with an overall bulk fluid composition known to be 23/25/52% by wt. However, the changes in the local composition of the individual components along the boiling length of the test section and the corresponding phase properties of the mixtures are being evaluated in this study.

2. Experimental set-up

The experimental set-up used in the present investigation has been described in detail in reference [13]. It consists of the primary loop (or the working fluid loop), the chilling unit loop, the cooling water loop and the data acquisition system. The schematic diagram of the experimental set-up is shown in Fig. 1.

In the primary loop, the working fluid, R-407C flows in a closed circuit. The loop consists of a refrigerant pump, an accumulator, a mass flow meter, the test section, pressure transducers, flow regulating valves and a receiver tank. The refrigerant is circulated through the loop by a hermetically sealed oil-free canned motor pump. A piston type accumulator is used to vary and maintain the desired pressure level of the loop. A pre-calibrated micromotion mass flow meter is used to measure the mass flow rate at the delivery of the pump. A flow and pressure regulating valve is positioned in between the mass flow meter and the pump. The required flow rate at the test section can be set by operating the main and bypass valves provided at the pump delivery. The vertical upflow test section is positioned after the mass flow meter as shown in Fig. 1. The liquid–vapor mixture from the test section is condensed in the cooling loop and the liquid is fed to the receiver tank connected to the suction end of the pump.

The entire closed loop of the test rig was subjected to a hydrostatic pressure test at 60 bar to ensure leak free operation at 45 bar with R-407C. Before filling up the rig with R-407C, a fine vacuum of the order of 0.1 Pa was created using a direct driven vacuum pump to ensure proper filling of the refrigerant.

The chilling unit loop is another closed loop system which enables the test fluid, R-407C, in the primary loop to be operated at pressures from atmospheric to pressures as high as the critical pressure (46.3 bar). It consists of three condensers connected in parallel and designed to work at different pressures and temperatures. The first condenser works in the temperature range of 33–86 °C which corresponds to a test section pressure of range 15–46.3 bar, and uses tap water as the heat sink. The condenser works with cooling water on the shell side and the refrigerant R-407C on the tube side. The cooling water is circulated using a 3 HP pump. The required flow rate of the cooling water can be controlled by adjusting the main valve and the bypass valves. The flow rate is measured by a rotameter positioned on the delivery side of the pump. The inlet and outlet temperatures of the cooling water are measured using T-type thermocouples. The second and the third condensers are designed to operate in the temperature ranges of 18–33 °C and −3 to 18 °C, which correspond to test section pressures of 10–15 bar and 5–10 bar, respectively. These two condensers form a closed loop and use the refrigerants R-22 and R-404a respectively to provide the heat sink to the primary coolant. The chilling unit loop is equipped with a hermetically sealed compressor to circulate the refrigerants through the condenser units. The condensers are shell-and-tube heat exchangers in which R-407C flows on the tube side and R-22 and R-404a on the shell side. Depending on the operating pressure and temperature of the test fluid in the primary loop, only one of the condensers is operated at any time.

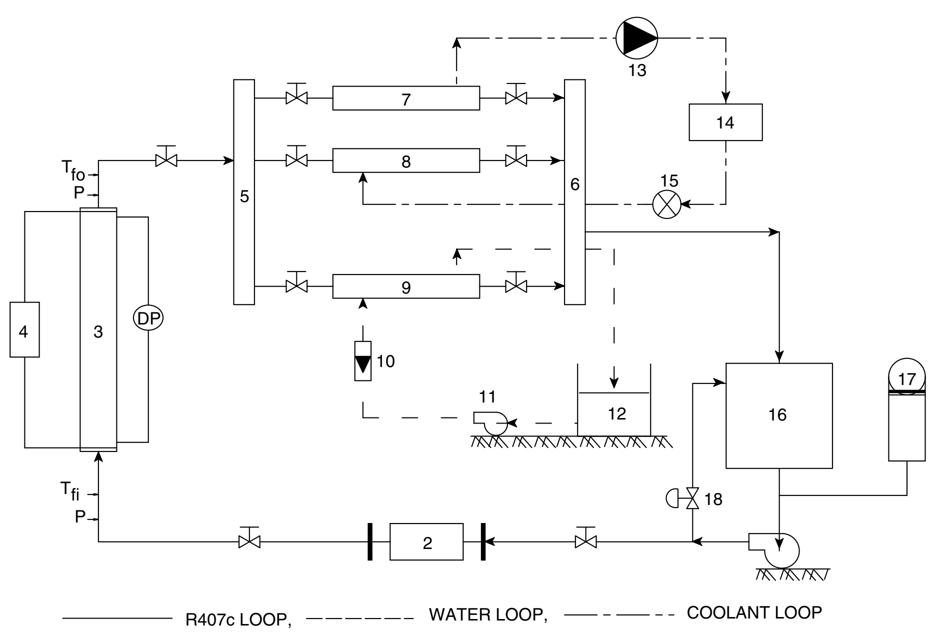

Fig. 1. Schematic diagram of the experimental set-up: 1, refrigerant circulating pump; 2, mass flow meter; 3, test section; 4, D.C. rectifier; 5, inlet header; 6, outlet header; 7, low temperature condenser; 8, medium temperature condenser; 9, water cooled condenser; 10, rotameter; 11, water circulating pump; 12, thermostatic container; 13, compressor; 14, condenser of refrigeration system; 15, expansion valve; 16, R-40C receiver tank; 17, piston type accumulator; 18, by-pass valve; P, pressure indicator; DP, Differential pressure indicator; T_{fi}, inlet temperature indicator; T_{fo}, oulet temperature indicator.

The data reported here have been obtained using the water-cooled condenser alone and cover the pressure range of 15–45.6 bar.

The test set-up was subject to a hydrostatic pressure test at up to 60 bar to ensure leak free operation at 45.6 bar with R-407C. Before filling up the test loop with R-407C, a vacuum of the order of 0.1 Pa was created using a vacuum pump to ensure no other gases are present.

The test section where the two-phase flow is studied is made of stainless steel (SS-304) and has an inner diameter of 12.7 mm and an outer diameter of 16.7 mm. The heated length of the test section is 3.24 m. A low voltage, high current DC power supply (maximum of 17 V DC, 800 A) is used to heat the test section. The DC power is supplied through two copper bus bars of 100 mm width and 7 mm thickness connected across the test section. The test section is well insulated with asbestos rope and polyurethane foam to minimize heat losses to the surroundings.

Details of the instrumentation on the test section are shown in Fig. 2. The bulk fluid temperature is measured at the inlet (T_{fi}), and outlet (T_{fo}) of the test section and at four (T_{f1}–T_{f4}) intermediate locations using mineral insulated T-type thermocouples of 1.10 mm outside diameter. The wall temperature of the test section is measured at 39 axial locations (T_{w1}–T_{w39}) along the heated length using T-type thermocouples with bead diameter of 0.8 mm. The thermocouples were calibrated before installation as well as in situ. The nominal accuracy of each thermocouple is 0.1 °C.

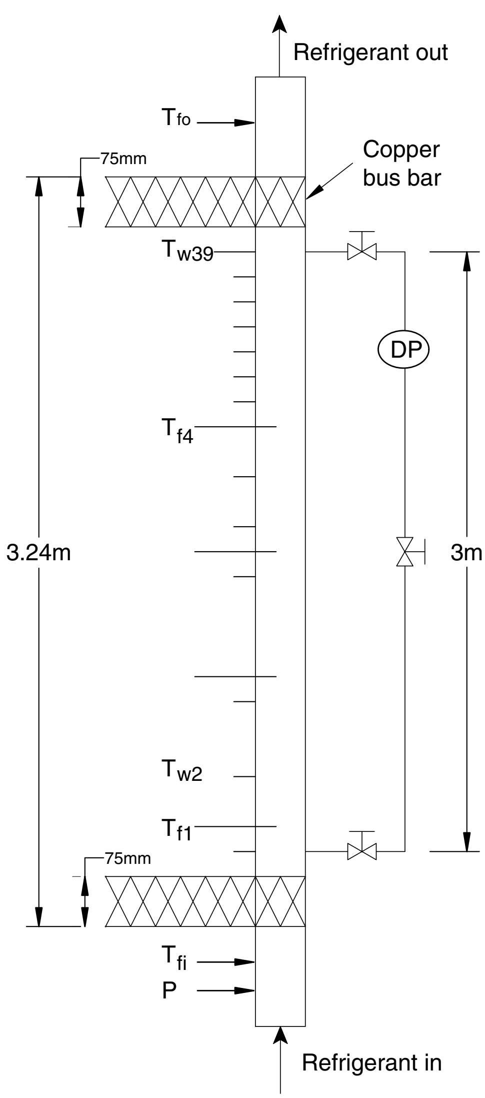

Fig. 2. Schematic diagram of vertical upflow test section.

A differential pressure transducer is used to measure the pressure drop across the test section. In addition, absolute pressure transducers are located at the test section inlet and at the outlet. The pressure transducers are electrically isolated by means of a teflon seating and bushes in between the flanges and bolts to which the transducers are connected.

3. Experimental procedure

The critical heat flux experiments were conducted as follows. Liquid R-407C from the receiver tank is circulated through the electrically heated test section with the help of the hermetically sealed canned motor pump. Once the system is stabilized at predetermined values of inlet temperature, mass flux and pressure, the readings from the absolute pressure transducer, differential pressure transducer, inlet and outlet temperatures and mass flow rates are recorded through the data acquisition system. For set values of pressure, flow rate, heat flux and inlet subcooling, the rise in the tube wall temperature is monitored. If a sudden rise in the wall temperature at any location is not detectable, the heat flux value in terms of the product of ($V \times I$) power input is increased and the other flow parameters are adjusted if necessary and the wall temperature is again observed. This process is continued until one of the thermocouples shows a significant rise in wall temperature. The power supply is automatically shut-off when one of the measured wall temperatures exceeds a given limiting value (viz. 150 °C). These experiments are repeated for different flow conditions. The pressure in the test loop is adjusted by turning on the piston accumulator. The temperature of the fluid at the inlet of test section is maintained by controlling the temperature and the mass flow rate of cooling water flowing through the heat exchangers.

The critical heat flux experiments were conducted over a pressure range of 15–45.6 bar (corresponding to reduced pressures of 0.32–0.98) and a mass flux range of 200–1200 kg/m^2 s. The inlet subcooling was maintained at 3 °C for all cases. The heat flux at which a significant rise in the wall temperature was recorded, was taken as the critical heat flux. Due to the limitation in the DC power supply, experiments in the higher mass flux range of 1400–2000 kg/m^2 s were conducted at fixed heat fluxes of 50, 60, 70 and 80 kW/m^2 by varying the mass flux for a fixed test section pressure and inlet subcooling and recording the wall temperatures. In some of these cases, the wall temperature showed a sudden rise after a particular distance from the inlet and the CHF here corresponds to the DO condition.

4. Data reduction

In order to calculate the experimental CHF and the CHF by predictive methods, thermodynamic properties of the liquid and vapor mixtures are necessary. For binary or multicomponent mixtures, these properties change along the boiling length of the tube and are dependent on the mole fraction of the individual components in addition to the primary system variables such as pressure and temperature. As this evaluation procedure is complicated for ternary mixtures and cannot be done by simple means, these are calculated by using REFPROP [16].

During the evaporation of non-azeotropic mixtures, the vapor composition of the more volatile component is in most cases greater than its composition in the liquid phase. Consequently, the local bubble point temperature increases as the concentration of the less volatile component in the liquid phase rises during evaporation along the heated tube. The local change in enthalpy during evaporation of a binary/multicomponent mixture as given by Collier and Thome [9], is due to (a) latent heat to the fraction of liquid vaporized, (b) sensible heat to the fraction of fluid in the liquid phase heated to a higher bubble point temperature and (c) sensible heat to the fraction of fluid in the vapor phase heated to a higher bubble point temperature. This local change in enthalpy, dh is given by the equation below

$$\mathrm{d}h = \mathrm{d}x\lambda + (1-x)C_{pl}\,\mathrm{d}T + xC_{pg}\,\mathrm{d}T \tag{1}$$

where $\mathrm{d}T$ – rise in the bubble point temperature.

Where only a small change in the enthalpy is considered (or for a very small heated section of the tube), the isobaric specific heats and the vapor quality may be assumed constant and the above equation reduces to

$$h = h_0 + x\lambda + (1-x)C_{pl}(T-T_0) + xC_{pg}(T-T_0) \tag{2}$$

where h_0 – enthalpy at zero vapor quality and T_0 – corresponding temperature.

The fluid always enters the test section under subcooled conditions with a known inlet enthalpy. The bulk enthalpy of the fluid at any location is obtained by adding the enthalpy rise due to heating of the fluid to the inlet enthalpy.

$$h = h_0(T,P,z) + \left(\frac{Q}{m}\right) \tag{3}$$

where $h_0(T,P,z)$ – enthalpy at zero vapor quality as a function of bulk fluid temperature, pressure and fluid composition at the inlet conditions (subcooled).

The inlet pressure and the pressure drop along the test section are measured and the local pressure is evaluated by assuming a linear pressure drop.

Assuming a thermodynamic equilibrium, the local quality (x) at any location z, can be evaluated using the energy balance

$$x = \left(\frac{1}{\lambda}\right)\left[h_0 + \left(\frac{Q}{m}\right) - h_{\mathrm{l,s}}\right] \tag{4}$$

where $h_{\mathrm{l,s}}$ – saturation enthalpy at location z.

A reasonable evaluation of the required properties is possible if a closed system evaporation and thermodynamic equilibrium at various locations along the test section is assumed, as suggested by Auracher and Marroquin [10]. As the pressure, bulk fluid enthalpy and the overall bulk composition are known, the desired properties such as the local quality and the composition of the individual components and other thermodynamic and transport properties of the mixture can be calculated from the above equations and the REFPROP simulation tool [16].

5. Results and discussion

CHF experiments have been carried out over a range of test section pressures and mass fluxes at fixed inlet subcooling of 3 °C. The overall test matrix is summarized in Table 1 in terms of the range of parameters investigated.

The location of the sudden rise in wall temperature along the length of the tube is shown in Fig. 3 for various test section pressures and for a mass flux of 200 kg/m^2 s. The wall temperature is also shown plotted against the local quality calculated along the length of the tube for a

Table 1
Range of parameters investigated in the present study

P (bar)	15, 20, 25, 30, 35, 40, 42, 44, 45.6
P_{R}	0.32, 0.43, 0.54, 0.64, 0.75, 0.86, 0.90, 0.95, 0.98
G (kg/m^2 s)	200, 400, 600, 800, 1000, 1200, 1400, 1600, 1800, 2000
q (kW/m^2)	50, 60, 70, 80 and variable in the range of 5 to 80
ΔT_{sub} (°C)	3

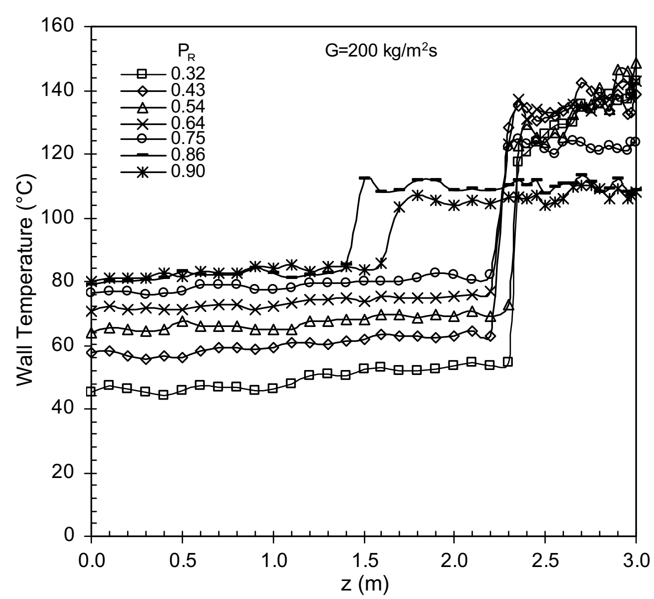

Fig. 3. Location of sudden rise in wall temperature along length of the tube for different pressures at $G = 200$ kg/m^2 s.

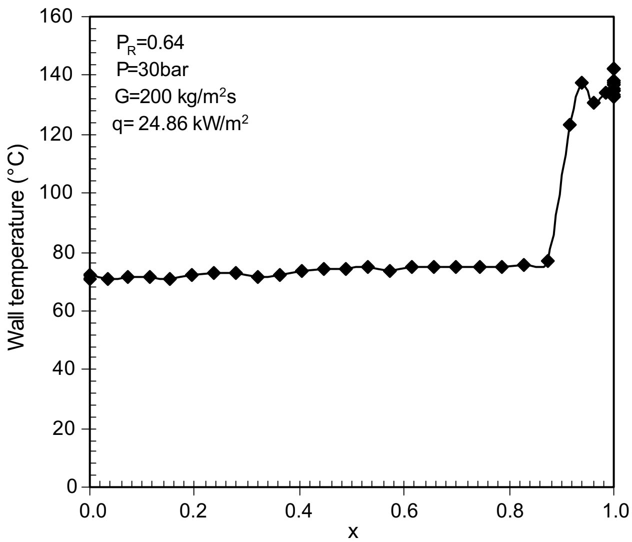

Fig. 4. Variation of wall temperature with quality.

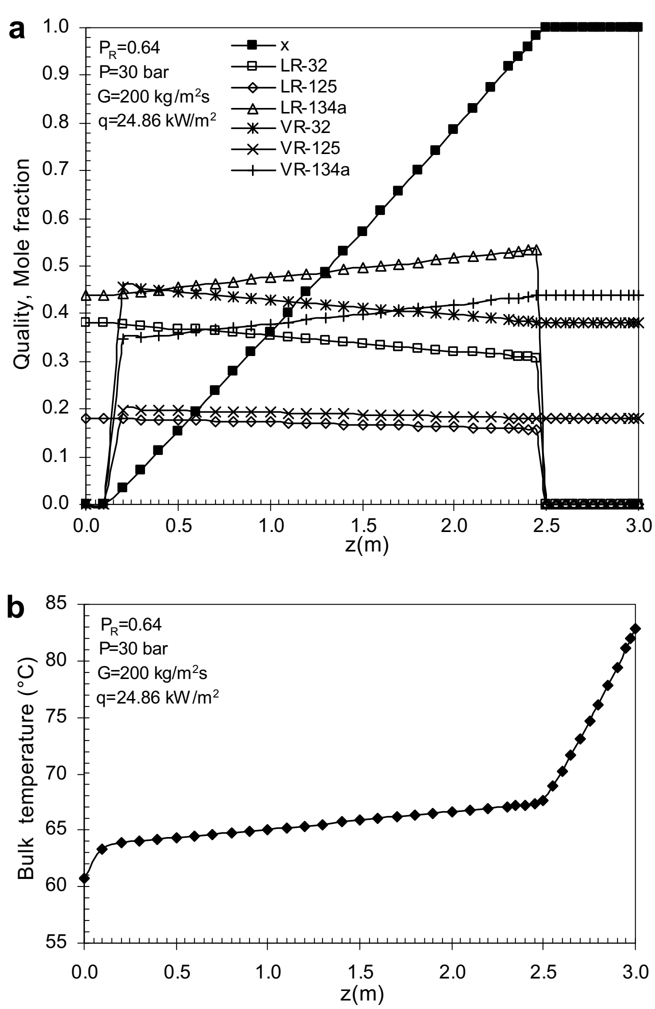

Fig. 5. Change in the (a) quality, concentration of the individual components and (b) the bulk fluid temperature along the length of the tube for $G = 200$ kg/m^2 s and $P = 30$ bar. (LR-32, LR-125 and LR-134a and VR-32, VR-125 and VR-134a represent the local mole fractions of the individual components R32, R-125 and R-134a in the liquid phase and vapor phases, respectively.)

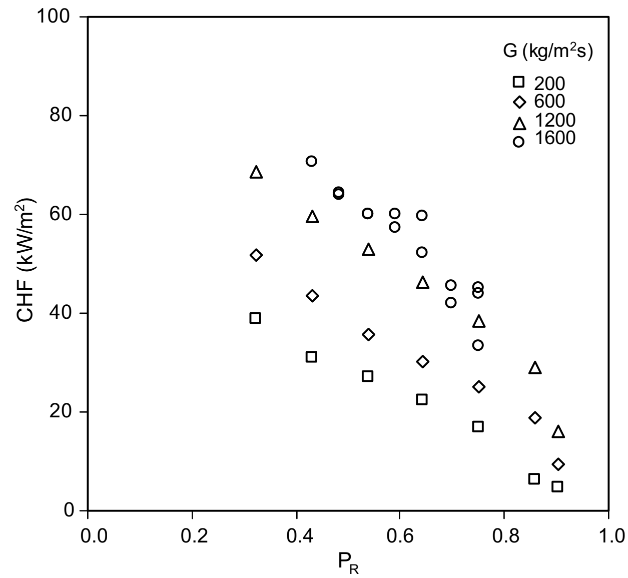

Fig. 6. Variation of measured CHF with reduced pressure for different mass flux values.

test section pressure of 30 bar and mass flux 200 kg/m^2 s in Fig. 4. The CHF here corresponds to the DO condition. Assuming that the dryout condition occurred at the same quality for a given set of pressure, mass flux, inlet subcooling and overall bulk composition at the inlet, the CHF (q_{CHF}) is given as

$$q_{\mathrm{CHF}} = q_{\mathrm{imp}}\left(\frac{z}{L}\right) \tag{5}$$

Assuming that thermodynamic equilibrium conditions exist at every location, the composition changes of the individual components along the length of the test section, as well as change in the bulk fluid temperature have been calculated using REFPROP and are shown in Fig. 5 for a test section pressure of 30 bar and mass flux 200 kg/m^2 s.

Fig. 6 shows the dependence of CHF on system pressure and inlet mass flux. The measured CHF values are plotted along the test section pressure (in terms of reduced pressure) for mass flux values of 200, 600, 1200 and 1600 kg/m^2 s. The CHF follows a similar trend for all the mass fluxes tested. For a given mass flux, CHF decreases, as expected, as the system pressure increases. The experimental data of Celata et al. [11] for binary refrigerant mixtures and Vijayarangan et al. [13] for pure refrigerant, as well as the current data for ternary refrigerant mixture show similar results. In general, the CHF is less for low mass fluxes and also a significant effect of mass flux on the CHF is seen at high pressures (reduced pressure >0.75).

6. Experimental data prediction

Most of the correlations and models developed to predict CHF in forced convective boiling are available only for pure fluids and no method is available in the literature to predict the CHF for ternary mixtures. In this study, an attempt is made to employ the pure fluid correlations for the case of non-azeotropic ternary mixtures using the physical properties of the mixture as suggested by Celata et al. [11]. Among a number of pure fluid correlations available in the literature, two well-established methods of predicting CHF, the dimensionless look-up table method of Groeneveld et al. [15], the generalized CHF correlation of Katto and Ohno [14], as well as Vijayarangan et al. [13] correlation for pure refrigerant, have been used for the prediction of present data.

The overall comparison of all the three prediction methods with the CHF data of the present study is shown in Figs. 7–9. The deviation from the experimental value is

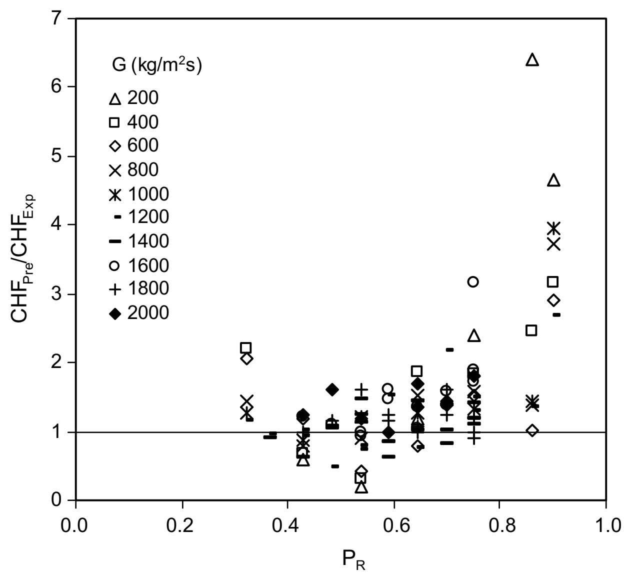

Fig. 7. The overall comparison of present data with predictions from Groeneveld et al. [15] in terms of relative deviation of reduced pressure.

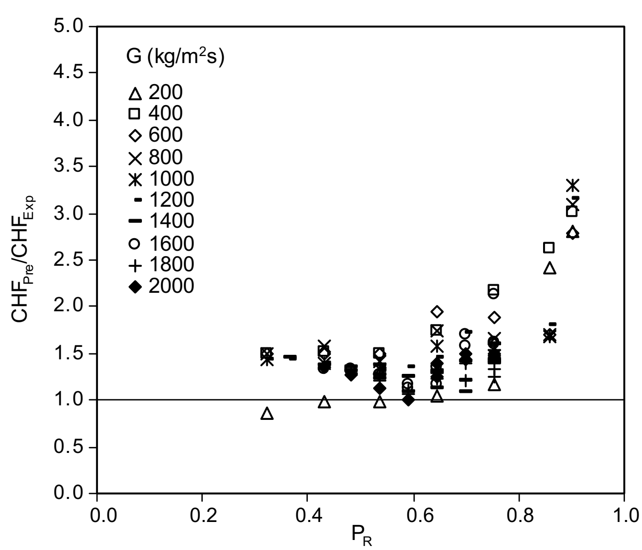

Fig. 8. The overall comparison of present data with predictions from Katto and Ohno [14] in terms of relative deviation of reduced pressure.

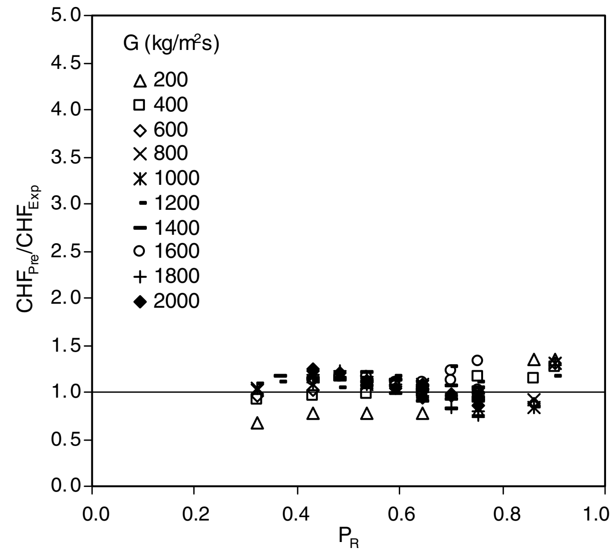

Fig. 9. The overall comparison of present data with predictions from Vijayarangan et al. [13] in terms of relative deviation of reduced pressure.

plotted as a function of the reduced pressure. It is clear from these figures that both Katto and Ohno [14] and Groeneveld et al. [15] prediction methods suffer from over prediction, which is severe especially at high reduced pressures. The pure fluid correlation of Vijayarangan et al. [13] was developed taking into consideration the strong effect of mass flux on the CHF and also the significant over prediction of the CHF at high reduced pressures by earlier correlations. This correlation is in good agreement with the experimental data over the entire range of pressures investigated. The correlation proposed by Vijayarangan et al. [13] is shown in the following equation

$$\frac{q_{\mathrm{CHF}}}{G\lambda}=0.0051\left\{\left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}}\right)^{0.133}\left(\frac{\sigma\rho_{\mathrm{l}}}{G^{2}L}\right)^{1/3}\left[\frac{1}{1+0.0031(L/d)}\right]\mathrm{Pr}^{0.147}Re^{0.25}\right\} \tag{6}$$

The range of dimensionless groups for which the above correlation is valid are given below:

$$1.4\times10^{4}<Re_{\mathrm{l}}<1.4\times10^{5}$$
$$0.13<P_{\mathrm{r}}<0.95$$
$$0.01<\rho_{\mathrm{v}}/\rho_{\mathrm{l}}<0.44$$
$$1.09\times10^{-8}<We_{\mathrm{l}}<1.47\times10^{-5}$$
$$6.1\times10^{-5}<q/G\lambda<8.05\times10^{-4}$$

The good agreement between the Vijayarangan et al. [13] correlation and the experimental data may be attributed to the fact that DO and the corresponding CHF are mainly a result of hydrodynamic behavior and mixture effects are relatively negligible. Similar results were observed by Celata et al. [11] for binary refrigerant mixtures of R-12/R-114. From the data prediction discussed here, it can be seen that the Vijayarangan et al. [13] correlation can be successfully used to predict the CHF values of ternary mixture used.

7. Uncertainties involved in the measurements

In the present study the measured variables are the liquid temperature, wall temperature, absolute pressure, mass flow rate, voltage and current. The uncertainty in the measurement of temperature is ±0.1%. The uncertainty in the measurement of mass flow rate is ±0.5%. The uncertainty in the measurement of absolute and differential pressure is ±0.25%. The uncertainty in the measurement of voltage and current is ±1% and ±1%, respectively. Therefore, the uncertainty in the measurement of critical heat flux using the Moffat procedure [17] lies within ±4.21%.

8. Conclusions

Experiments to determine CHF have been conducted for a uniformly heated vertical tube with ternary non-azeotropic refrigerant mixture R-407C. The CHF decreases as the system pressure increases as in the case of pure fluids and binary mixtures. Existing predictive methods for pure fluids, such as the look-up table method of Groeneveld et al. [15], the generalized CHF correlation of Katto and Ohno [14] and Vijayarangan et al. [13] correlation for pure refrigerants, have been used for the prediction of the experimental data. Vijayarangan et al. [13] correlation developed for pure refrigerant agrees well with the present data over the entire range of investigation.

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