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Flow boiling heat transfer of Freon R11 and HCFC123 in narrow passages

Z.Y. Bao, D.F. Fletcher*, B.S. Haynes

Department of Chemical Engineering, University of Sydney, Sydney, NSW 2006, Australia

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Abstract

Flow boiling heat transfer coefficients for Freon R11 and HCFC123 in a smooth copper tube with an inner diameter of 1.95 mm have been experimentally investigated. The parameter ranges examined are: heat fluxes from 5 to 200 kW m^{-2}; mass fluxes from 50 to 1800 kg m^{-2} s^{-1}; vapour quality from 0 to 0.9; system pressures from 200 to 500 kPa; and experimental heat transfer coefficients from 1 to 18 kW m^{-2} K^{-1}. It was found that the heat transfer coefficients are a strong function of the heat flux and the system pressure, while the effects of mass flux and vapour quality are very small in the range examined. This suggests that the heat transfer is mainly via nucleate boiling. The present experimental data were compared with some existing correlations and recommendations on their use is made.

Keywords: Freon R11; HCFC123; Narrow passage; Flow boiling; Nucleate boiling

1. Introduction

Heat exchangers with fine passages have some remarkable advantages and are finding ever more applications in industry. However, because of the complexity of the phenomenon of boiling heat transfer and the lack of research work in this field, the process of flow boiling heat transfer in fine passages is still far from well understood. Despite recent work carried out in order to investigate the behaviour of flow boiling heat transfer in small diameter ducts [1–4], the lack of information and reliable data for a wide range of engineering design and other applications is still very clear. The aim of most experimental investigations has been to examine a particular fluid and geometry of relevance to a specific heat exchanger or evaporator. For example, Yan and Lin [4] have recently investigated boiling of R-134a in a bundle of 2.0 mm diameter tubes. They obtained data on the average heat transfer coefficients over the length of the tube, as a function of mass flux, saturation temperature and heat flux. They fitted their data by a correlation, based on that of Kandlikar [5], in which the average heat transfer coefficient is a function of the convection number, boiling number and liquid Froude number.

This paper describes an experimental investigation of the behaviour of flow boiling heat transfer in fine passages, and examines the effects of operation parameters, such as flow mass flux, vapour quality, heat flux, and system pressure. However, in this case local heat transfer data have been obtained. The suitability of some existing correlations for flow boiling heat

* Corresponding author. Tel.: +1-61-2-9351-4147; fax: +1-61-2-9351-2854.

E-mail address: davidf@chem.eng.usyd.edu.au (D.F. Fletcher).

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Nomenclature

Bo	boiling number
C_p	specific heat capacity (J kg^{-1} K^{-1})
d	diameter of flow passages (m)
F	convective heat transfer correlation factor
G	mass flux (kg m^{-2} s^{-1})
H	specific enthalpy (J kg^{-1})
H_{fg}	enthalpy of vaporization (J kg^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
M	molecular weight of fluid (g mol^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
Nu	Nusselt number
P	fluid pressure (Pa)
Q_F	rate of fluid heat transfer (W)
q	heat flux to fluid (W m^{-2})
Re	Reynolds number
S	nucleate-boiling suppression factor
T	temperature (°C)
U_S	superficial velocity (m s^{-1})
X	Lockhart–Martinelli parameter
x	vapour quality
x_{th}	thermodynamic vapour quality

Greek symbols

ΔT	temperature difference (°C)
ρ	density (kg m^{-3})

Subscripts

CB	convective boiling
cr	critical value
exp	experimental
F	fluid
in	inlet
L	liquid phase
NB	nucleate boiling
out	outlet
PB	pool boiling
pred	predicted
sat	saturation
V	vapour phase
W	wall

transfer are assessed against the experimental data. This work builds upon our earlier work [6], in which we investigated pressure drop and non-boiling heat transfer in the same geometry.

2. The experimental apparatus

A schematic of the experimental apparatus is shown in Fig. 1. The system consists of a circulating fluid test loop and an open water cooler loop. The working fluids employed in the present experiments are Freon R11 and HCFC123. Subcooled working fluid was pumped into the test section, where some of the fluid was vaporized. The test section was followed by a cooler in which the vapour was recondensed and the fluid was cooled to below its saturation temperature. Finally, the fluid was recirculated to the test section by a pump.

The system pressure was set and controlled to within $\pm$ 3% by controlling the electrical power input to a vapour/liquid reservoir connected to the main circulation loop. The heater employed for this purpose was encased in polymer and was incapable of creating surface temperatures capable of causing thermal degradation of the working fluid. This system allowed the pressure to be controlled independently of other conditions. The total length of the test section is 870 mm, where the first 400 mm is an entrance section, the next 270 mm is the actual heated test section, and the last 200 mm is an unheated exit section.

The heated test section is divided into 10 separate heating zones as shown in Fig. 2. Each zone is surrounded by a copper block which is heated by an electrical band heater. The length and outside diameter of the copper block are each 25 mm. The test duct passed through the center of the blocks. In order to get good thermal contact between the tubing and the blocks, the components were soldered together. Adjacent blocks were separated by a 2 mm thickness of insulation to minimize the interaction between adjacent zones.

The flow was metered using a K-FLOW K2 coriolis-effect mass flow meter (LMFM). The pressure and pressure drop were measured with pressure gauges (P) and differential pressure transducers (DP). The inlet and outlet fluid temperatures, and the temperatures of the ten heating blocks were measured using Type K thermocouples (T). The thermocouples used for determining the block temperature were located at the base of narrow wells (< 1 mm diameter), within 2 mm of the duct wall. The block temperature measurements were made differentially with respect to the entrance temperature of the fluid with a minimum detectable temperature difference of 0.02 K. All the thermocouples outputs were acquired by a data logger (Datataker 100) and stored in a personal computer.

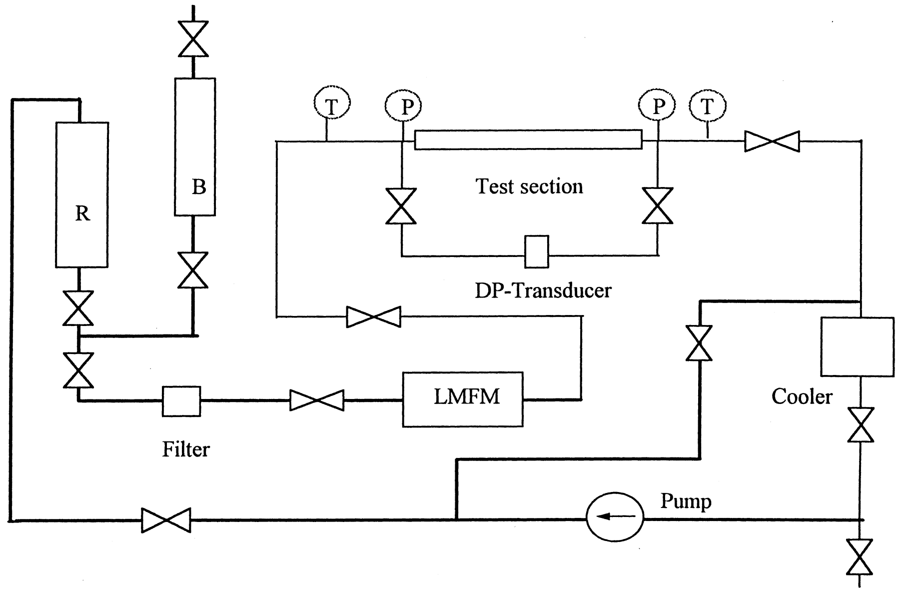

Fig. 1. A schematic of the apparatus used for boiling heat transfer measurements. The key to the symbols is as follows: B, boiler used to pressurize the fluid; DP, differential pressure; LMFM, liquid mass flow meter; P, pressure transducer; R, fluid reservoir; and T, thermocouple.

Heat released by the resistance heaters attached to the blocks may be transferred to the adjacent blocks and to the surroundings, as well as to the working fluid. The inter-block and block-to-ambient conductances were determined in an extensive series of calibration experiments, performed with and without fluid flowing in the test duct (see Refs. [6] and [7] for details). In this work, the heat input is transferred predominantly into the fluid stream passing through the blocks, with the heat transferred between the heating zones and to the surroundings each being less then 5% under all circumstances, except for the first and the last zones, for which larger corrections (<20%) were required.

The system was tested with single phase flows of water and of Freon R11 and these results were compared with established correlations, as reported in [6,7]. The overall heat balance between the electrical heat input (corrected for losses to the surroundings) and the sensible heat transferred to the fluid in the complete test section was satisfied to within 5% in all cases.

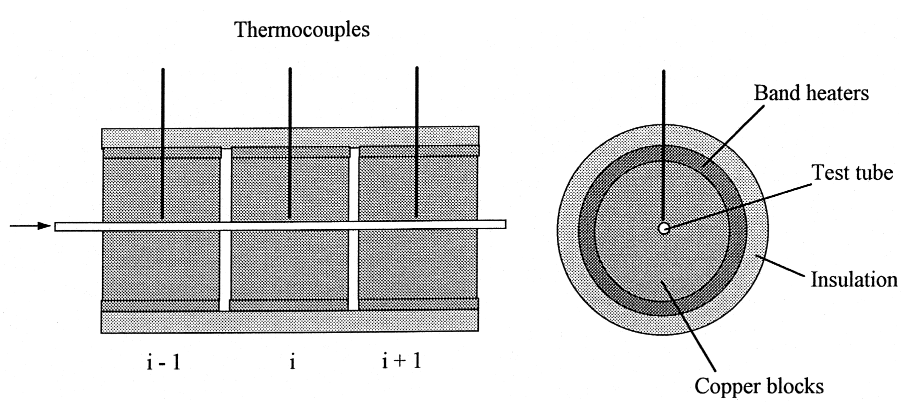

Fig. 2. A schematic of the heating arrangement used to provide a controlled local heat flux.

The local heat transfer coefficient at each heating section is determined from the following equation

$$h = \frac{q}{T_W - T_F} \tag{1}$$

where T_W is the wall temperature, T_F is the mean fluid temperature, and q is the inner wall heat flux to the fluid. Although the difference between the measured copper block temperature and the inner wall temperature was evaluated to be very small, the temperature difference is still considered and evaluated according to the heat flux and the wall heat transfer resistance. The mean fluid temperature was calculated from the inlet fluid temperature and the heat transferred to the fluid. If the fluid is subcooled, then the outlet temperature is given by

$$T_{F,\,out} = T_{F,\,in} + \frac{Q_F}{\dot{m}C_{pL}} \tag{2}$$

where Q_F is the heat transfer rate to the fluid, $\dot{m}$ is fluid mass flow rate and C_{pL} is the fluid specific heat capacity. The mean fluid temperature in a section is given by

$$T_F = \frac{T_{F,\,in} + T_{F,\,out}}{2} \tag{3}$$

For the case where the calculated mean fluid temperature is higher than the saturation temperature, the saturation temperature is used as the mean fluid temperature, and the vapour quality is determined according to the heat transferred to the fluid. The thermodynamic vapour quality is given by

$$x_{th} = \frac{H - H_{sat,\,L}}{H_{sat,\,V} - H_{sat,\,L}} \tag{4}$$

where $H_{sat,\,L}$ and $H_{sat,\,V}$ are the specific enthalpy of the saturated liquid and vapour, respectively. H is the total specific enthalpy of the fluid, which is determined from the inlet enthalpy and the heat transferred to the fluid.

Table 1
The range of parameters considered in the boiling heat transfer tests

Parameter	Range	Unit
Diameter (d)	1.95	mm
System pressure (P)	200–500	kPa
Mass flux (G)	50–1800	kg m^{-2} s^{-1}
Heat flux (q)	5–200	kW m^{-2}
Vapour quality (x_{th})	−0.3–0.9	–
Temperature difference (ΔT_{sat})	5–15	°C
Superficial vapour velocity (U_{SV})	0–22	m s^{-1}
Superficial liquid velocity, (U_{SL})	0.04–1.2	m s^{-1}
Vapour Reynolds number (Re_v)	0–68,000	–
Liquid Reynolds number (Re_L)	200–10,000	–
Heat transfer coefficient (h)	1–18	kW m^{-2} K^{-1}

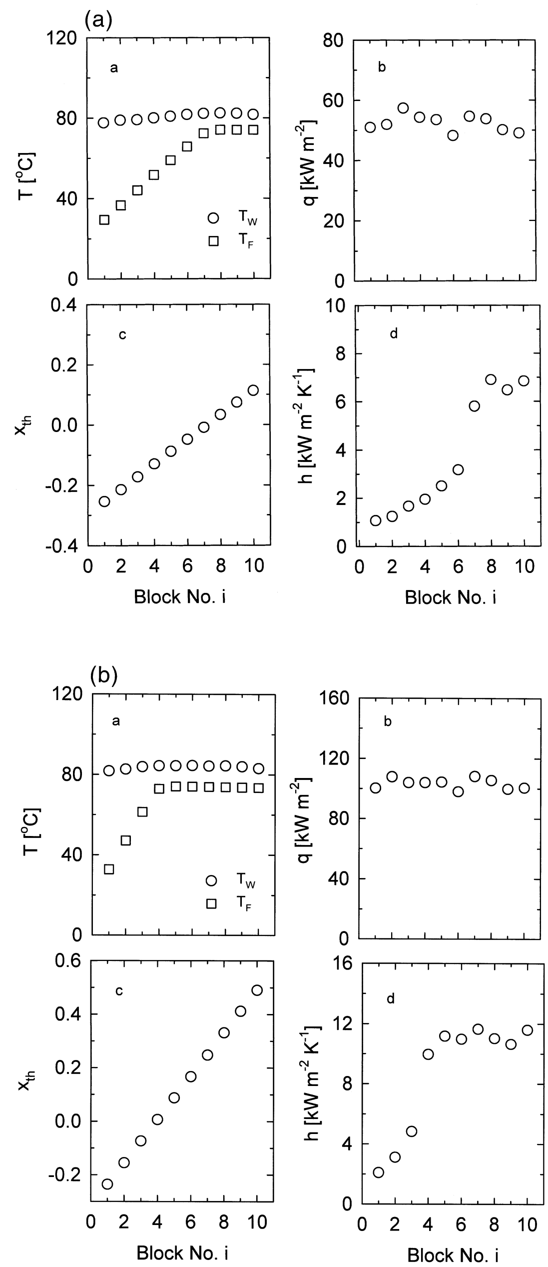

Fig. 3. Typical results for wall and fluid temperatures, vapour quality, heat flux, and heat transfer coefficient in individual heating sections. The working fluid is R11, the mass flux is 446 kg m^{-2} s^{-1} and the inlet pressure is 463 kPa: (a) heat flux is 55 kW m^{-2}; (b) heat flux is 105 kW m^{-2}.

3. The experimental results

Local flow boiling heat transfer coefficients for Freon R11 and HCFC123 have been examined experimentally. The main variable parameters in the present work are the heat flux, the mass flow velocity, the vapour quality and the system pressure. The ranges of these and other relevant parameters are summarized in Table 1.

Fig. 2 shows two sets of typical block-to-block summaries of data from the experiments. The data are the wall temperature, the predicted fluid temperature, the thermodynamic vapour quality, the heat flux and the heat transfer coefficient. The flow is of Freon R11 with $G = 446$ kg m^{-2} s^{-1}, $P_{in} = 463$ kPa. Case (a) is for a heat flux of 55 kW m^{-2} and case (b) is for a heat flux of 105 kW m^{-2}. The heat fluxes are not perfectly constant across the heating sections because of the difference in heat inputs and the effect of inter-block heat transfer, but typically they are constant to within 10%.

From these plots, it is clear that along the direction of the flow, the wall temperature T_W increases, reaches a maximum at some point, and then decreases very slowly. In comparison, for the calculated fluid temperature, T_F, and thermodynamic vapour quality, x_{th}, obtained from an energy balance, it is found that the maximum wall temperature always occurs around where $x_{th} = 0$, up to which point the flow is subcooled. In this region, the experimental heat transfer coefficients are relatively low but they increase rapidly as the saturation condition ($x_{th} = 0$) is approached. This behaviour and the magnitude of the heat transfer coefficients obtained in this region are indicative of the occurrence of subcooled boiling, since the single-phase heat transfer coefficients expected in this region are of the order of 0.1 kW m^{-2} K^{-1}. However, the fact that $h_{x<0} < h_{x>0}$ may not be mechanistically significant because, for $x_{th} < 0$, h is defined in terms of the bulk mean temperature, while for $x_{th} > 0$ the saturation temperature is used.

The results obtained for a working fluid of HCFC123 are qualitatively very similar to those for R11, as shown in Fig. 4, for which the inlet pressure is 360 kPa, the mass flux is 335 kg m^{-2} s^{-1} and the average heat flux is 85 kW m^{-2}.

3.1. The effects of mass flux and vapour quality

Fig. 5 shows the heat transfer coefficient as a function of quality for a range of mass fluxes. For the R11 case the inlet pressure is maintained at a constant

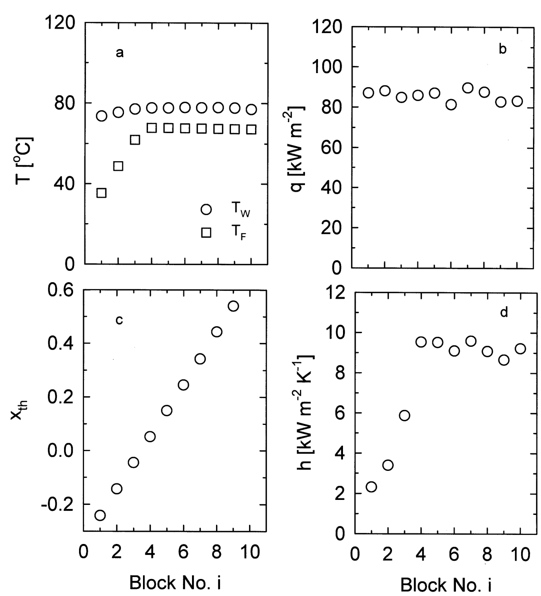

Fig. 4. Typical results for wall and fluid temperatures, vapour quality, heat flux, and heat transfer coefficient in individual heating sections. The working fluid is HCFC123, the mass flux is 335 kg m^{-2} s^{-1} and the inlet pressure is 360 kPa.

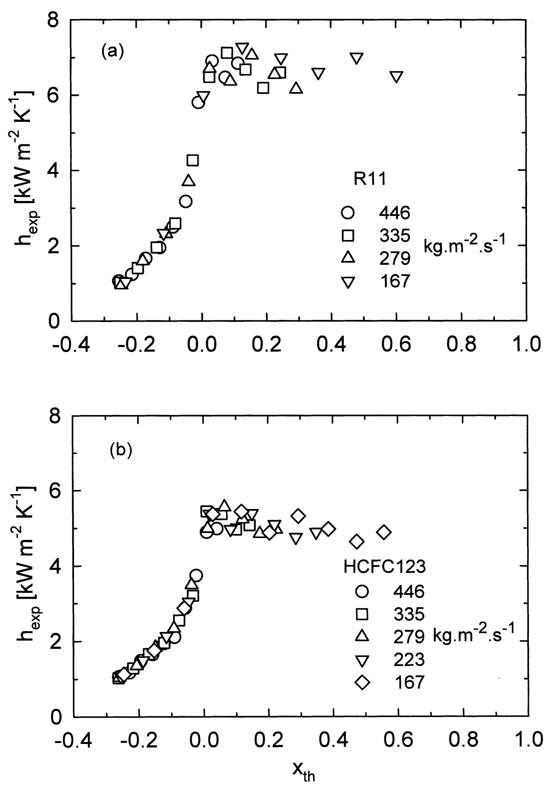

Fig. 5. A plot of experimental heat transfer coefficient, h, against vapour quality, x_{th}. (a) R11 flows: the inlet pressure is 470 kPa and the heat flux is 55 kW m^{-2}; (b) HCFC123 flows: the inlet pressure is 350 kPa and the heat flux is 39 kW m^{-2}.

value of 470 kPa, the heat flux is 55 kW m^{-2} and the mass flux ranges from 167 to 446 kg m^{-2} s^{-1}. For the HCFC123 case, the inlet pressure is 350 kPa and the heat flux in each heating section is 39 ± 5 kW m^{-2}. It is evident that, in both cases, the heat transfer coefficient is independent of the mass flow rate, with all other conditions fixed.

For both fluids, the fact that the heat transfer coefficient is independent of the mass flux for subcooled conditions suggests that the contribution of convective heat transfer to the overall transfer rate is small. For the saturated conditions, the independence of the heat transfer coefficient on both the mass flux and the quality strongly suggests that heat transfer is dominated by nucleate boiling, with convective heat transfer again being very weak.

3.2. The effect of heat flux

In contrast to the lack of influence of the mass flux on the heat transfer coefficients, the heat flux, q, has a significant effect. Fig. 6 shows the experimentally determined heat transfer coefficient plotted against quality as a function of the heat flux, at constant mass flux and system pressure. For the Freon R11 case the mass flux was 446 kg m^{-2} s^{-1} and the inlet pressure was 460 kPa, whilst for the HCFC123 case the mass flux was 452 kg m^{-2} s^{-1} and the inlet pressure was 450 kPa. The data show that the heat transfer coefficients increase steadily with the heat flux and are independent of the quality once the fluid becomes saturated in both cases.

For flow boiling heat transfer, the nucleate component is often expressed as a function of the boiling number, $Bo = q/G_L \times H_{fg}$ [1,8,9] or as a function of the pool boiling heat transfer coefficient, h_{PB} [10–12]. In either case, h is a function of the heat flux, q. In order to determine the effect of q on h in this work, the measured heat transfer coefficients (for a constant inlet pressure) in the saturated region ($0 < x_{th} < 1$) are correlated using

$$h = C_1 q^{C_2} \quad (5)$$

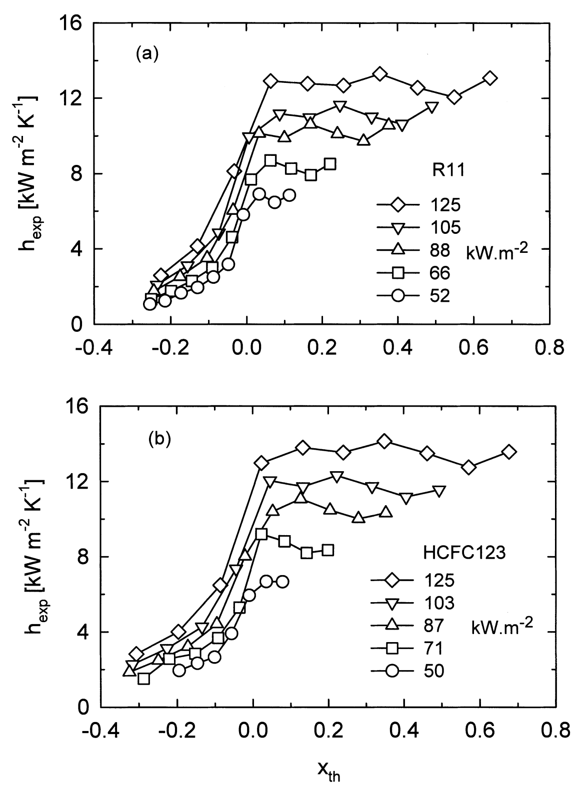

Fig. 6. A plot of the experimental heat transfer coefficient, h against x_{th} for different heat fluxes. (a) R11 flows: the inlet pressure is 460 kPa and the mass flux is 446 kg m^{-2} s^{-1}; (b) HCFC123 flows: the inlet pressure is 450 kPa and the mass flux is 452 kg m^{-2} s^{-1}.

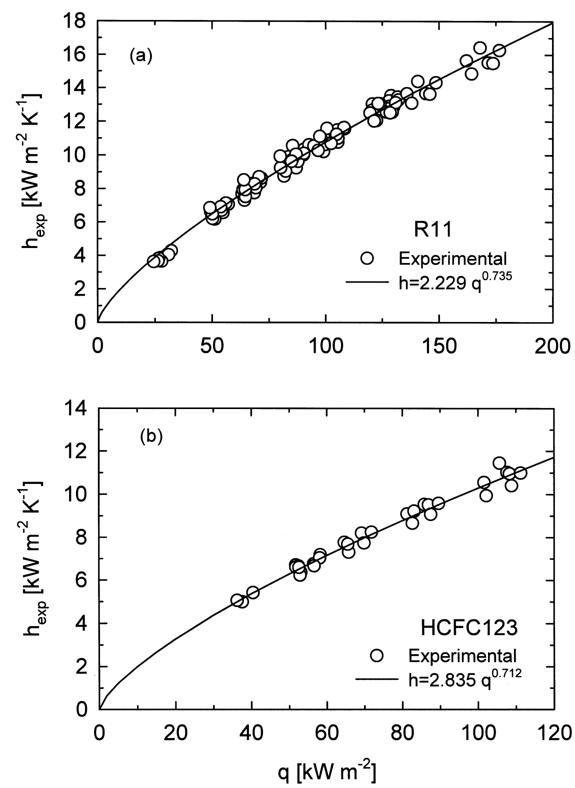

Fig. 7. A plot of the experimental heat transfer coefficient, h, against the heat flux, q, at different heat fluxes: (a) R11 flows: the inlet pressure is 460 kPa and the mass flux is 446 kg m^{-2} s^{-1}; (b) HCFC123 flows: the inlet pressure is 450 kPa and the mass flux is 452 kg m^{-2} s^{-1}.

The least-squares best fit to Eq. (5) using the current data, shown in Fig. 7, gives $C_1 = 2.23$ and $C_2 = 0.735$ for R11, and $C_1 = 2.84$ and $C_2 = 0.712$ for HCFC123. Note that in the correlations q is expressed in W m^{-2} and h is in W m^{-2} K^{-1}. The values for C_2 obtained here fall within the conventional range of 0.67–0.8 suggested by Steiner and Taborek [12] for nucleate flow boiling.

3.3. The effect of system pressure

In order to investigate the effect of system pressure, experimental measurements were carried out by increasing or decreasing the system pressure at constant mass velocity and heat supply rate. Fig. 8 shows experimental data for both fluids. For the Freon R11 case the mass flux is 560 kg m^{-2} s^{-1} and the heat flux is 125 ± 6 kW m^{-2}, and for the HCFC123 case the mass flux is 335 kg m^{-2} s^{-1} and the heat flux is 86 ± 6 kW m^{-2}. In the region of saturated flow boiling, $0 < x_{th} < 0.55$, it is clear that the heat transfer coefficient increases with increasing pressure.

This result is consistent with the results for nucleate flow boiling by Klimenko [13], Liu and Winterton [11] and Steiner and Taborek [12], and with trends of some well-known correlations for pool boiling, for example, Cooper's correlation [12] and that of Kutateladze [14].

For the region of subcooled nucleate boiling, unlike in the region of saturated nucleate boiling, an effect of the system pressure is not apparent in Fig. 8. The definition of h is different in this region from that in the region of saturated nucleate boiling, with $\Delta T = T_W - T_F$, which does not depend explicitly on pressure. In Fig. 9, the same sets of measured heat transfer data as shown in Fig. 8 are plotted against heating section number, i. Now, in contrast to the saturated region, the subcooled heat transfer coefficient decreases with an increase in system pressure. The reason for this may be that, because the inlet temperature of the fluid is maintained approximately constant (32.5°C), at a given location a higher system pressure

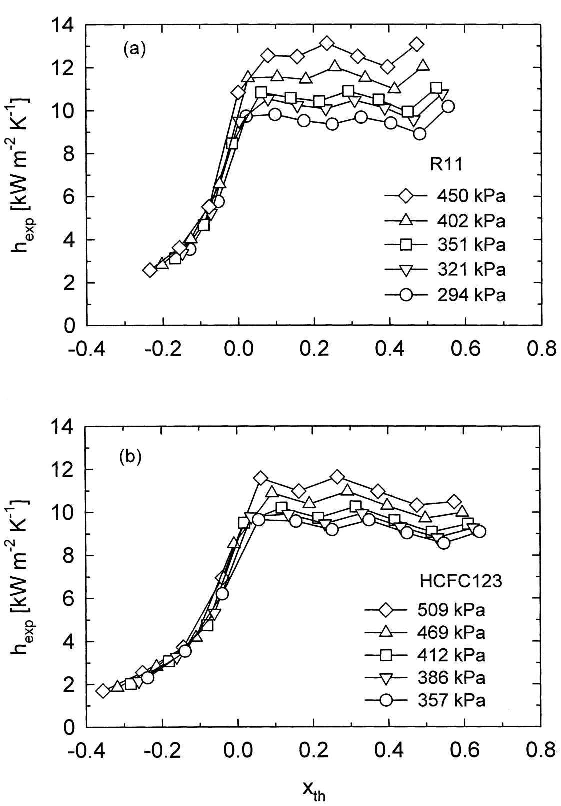

Fig. 8. A plot of the experimental heat transfer coefficient, h, versus vapour quality, x_{th}, at different pressures: (a) R11 flows: the heat flux is 125 ± 6 kW m^{-2} and the mass flux is 560 kg m^{-2} s^{-1}; (b) HCFC123 flows: the heat flux from 86 ± 6 kW m^{-2} and the mass flux is 335 kg m^{-2} s^{-1}.

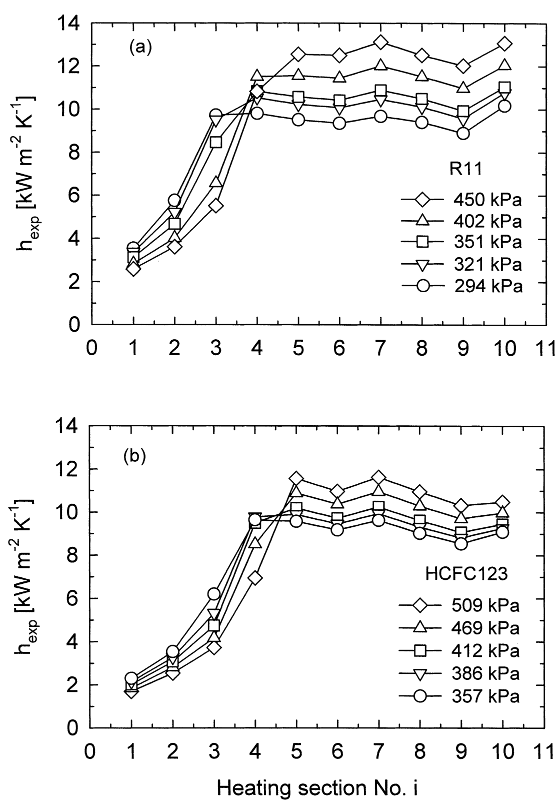

Fig. 9. The experimental heat transfer coefficient, h plotted against heating section number, i, at different pressures: (a) R11 flows: the heat flux is 125 ± 6 kW m^{-2} and the mass flux is 560 kg m^{-2} s^{-1}; (b) HCFC123 flows: the heat flux is 86 ± 6 kW m^{-2} and the mass flux is 335 kg m^{-2} s^{-1}.

gives a more highly subcooled fluid which is further from fully developed nucleate boiling. The present results suggest therefore that, in the region of subcooled nucleate boiling, the heat transfer coefficients defined as above are considerably affected by the degree of subcooling.

For the remainder of this paper all discussion of flow boiling will be restricted to saturated conditions.

3.4. Comparison of the Freon R11 and HCFC123 results

The forgoing discussion has demonstrated that the two most important system parameters in saturated flow boiling are the heat flux and the pressure. In order to examine the effect of different fluids on the heat transfer coefficient, comparisons are now carried out at constant pressure and heat flux. In Fig. 10(a), the saturated ($x_{th} > 0$) flow boiling heat transfer coefficients for R11 and HCFC123 are compared for the same conditions of mass flux ($G = 335$ kg m^{-2} s^{-1}) and inlet pressure ($P_{in} = 355$ kPa) by plotting h against the heat flux, q. As shown in the figure, the heat transfer coefficients for HCFC123 flows are on average about 10% higher than those for R11 at the same heat flux.

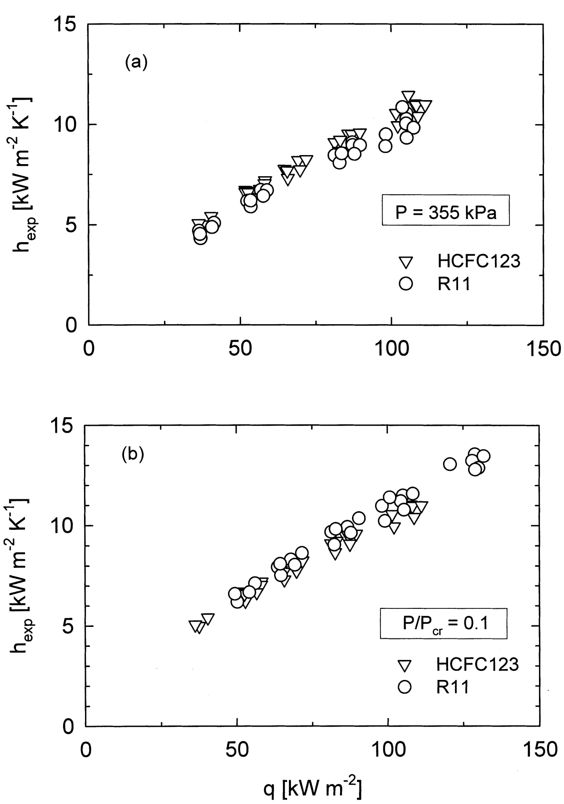

Fig. 10. A comparison of heat transfer coefficient of R11 and HCFC123 flows of the same mass flux (335 kg m^{-2} s^{-1}) versus heat flux q. The data cover the range $0.03 < x_{th} < 0.85$: (a) The same inlet pressure (355 kPa) is set in both cases; (b) the same reduced pressure ($P_{in}/P_{cr} \approx 0.1$) is set in both cases.

The only published comparisons of heat transfer of boiling R11 and HCFC123 was made by Webb and Pais [15] for nucleate pool boiling of the two fluids at a relatively low temperature and pressure. For boiling on the surface of a copper tube ($d = 9.53$ mm) in a copper boiling cell ($d = 76.2$ mm), $T_{sat} = 4.4$ and 26.7°C and heat fluxes in the range 8–100 kW m^{-2}, they found the difference in nucleate pool boiling coefficients to be within 10%. However, the saturation pressure of HCFC123 is lower than that of R11 at the same temperature. Therefore, since Webb and Pais [15] also found higher heat transfer coefficients at higher temperatures (corresponding to higher pressures), comparison of the two fluids at the same pressure would give a boiling heat transfer coefficient for HCFC123 somewhat higher than for R11. In addition, it should be pointed out that the present system pressure is higher than the pressure in the experiments of Webb and Pais, and the measured heat transfer coefficients are also higher than those of Webb and Pais. Thus, the present results are entirely consistent with the nucleate pool boiling data for R11 and HCFC123 reported by Webb and Pais [13], even though the difference in h between R11 and HCFC123 is not very large over the parameter range examined here.

Heat transfer coefficients for nucleate pool and flow boiling are often correlated in terms of the reduced pressure (P/P_{cr}) [11,16]. In Fig. 10(b), the measured heat transfer coefficients of saturated R11 and HCFC123 flows at similar reduced pressure, $P_{in}/P_{cr} \sim 0.1$ ($P_{in} = 460$ kPa for R11, and 355 kPa for HCFC123) are plotted against the heat flux. There is now less difference between the data obtained using the two different fluids at a given heat flux, implying that for saturated Freon R11 and HCFC123 flow boiling heat transfer, the heat transfer coefficient could be correlated using only the heat flux and the reduced pressure in the region examined here.

4. Comparison with the data of other workers

The present experimental results can be summarized by stating that R11 and HCFC123 flow boiling heat transfer in fine passages is dominated by a nucleate boiling mechanism for a wide range of values of system parameters. There are no other experimental data available for the same conditions as the present work, but some related studies have been reported recently. A study of boiling heat transfer of refrigerant R113 in a horizontal small-diameter (2.92 mm) stainless steel tube was reported by Wambsganss et al. [3]. Local heat transfer coefficients were measured over the range $q = 9$–90 kW m^{-2}, G = 50–300 kg m^{-2} s^{-1}, $x_{\mathrm{th}} = 0$–0.9, and outlet saturation pressure 124–160 kPa. They also found that the local heat transfer coefficient is a strong function of heat flux, and is only weakly dependent on the mass flux and quality. Based on the flow pattern map of air–water flow developed by Damianides and Westwater [17] and their own work, they concluded that the dominant flow pattern was slug flow, with heat transfer due to nucleate boiling in the relatively thick liquid layer present in slug flow.

Experimental results for boiling heat transfer to flowing refrigerant R12 in a small horizontal, rectangular channel was reported by Tran et al. [2]. The channel, made of brass, had a height of 4.06 mm, a width of 1.7 mm and a length of 0.9 m. The range of parameters studied was: $q = 4$–34 kW m^{-2}, $G = 50$–400 kg m^{-2} s^{-1}, $x_{\mathrm{th}} = 0.15$–0.8 and $P_{sat} = 760$–950 kPa. Again, it was found that the nucleation mechanism dominates the heat transfer in flow boiling in small channels, and it was suggested that this occurs in the thin liquid layer present in annular flow, as has been investigated by Mesler [18,19].

The investigation carried out by Ross and Radermacher [20] examined the possibility of suppression of nucleate boiling of pure and mixed refrigerants in turbulent annular flow. The test section employed was an electrically heated horizontal stainless steel tube of diameter 9 mm and length 2.7 m and the working fluid was R152a. Both nucleate boiling and forced convective evaporation were observed. The results indicated that nucleate boiling may be easier to achieve at high

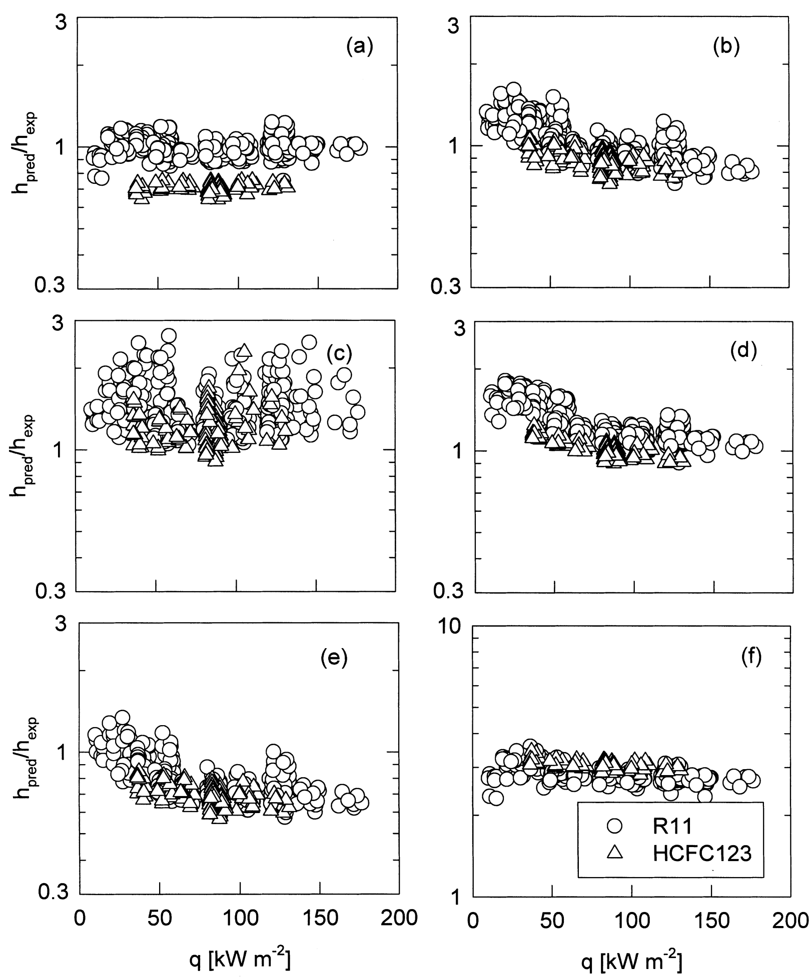

Fig. 11. Comparison of the experimental heat transfer coefficients h_{exp} with those predicted from correlations: (a) Lazarek and Black [17] correlation; (b) Chen [10] correlation (with h_{PB} calculated from Cooper's correlation); (c) Gungor and Winterton [9] correlation; (d) Klimenko [13] correlation; (e) Liu and Winterton [11] correlation; (f) Steiner and Taborek [12] correlation.

pressure and is dominant at low quality, while forced convection evaporation is favoured at high quality. In addition, it was shown that in the region of saturated nucleate boiling, the heat transfer coefficient increases with pressure.

5. Comparison of the experimental data with existing correlations

There are numerous correlations available for saturated flow boiling heat transfer, such as those of Chen [10], Lazarek and Black [1], Gungor and Winterton [9], Liu and Winterton [11], Klimenko [13] and Steiner and Taborek [12]. These correlations are compared with the present experimental data, and the results are given in Fig. 11.

The Lazarek and Black [1] correlation is compared with the present measurements since it has been reported to work well for boiling heat transfer in small diameter tubes [3]. According to this correlation, the Nusselt number, Nu, is a function of the Reynolds number, Re, and the boiling number, Bo, and is given by

$$Nu = 30Re^{0.857}Bo^{0.714} \tag{6}$$

This correlation is based on data obtained from saturated flow boiling of R113 under conditions similar to those employed in the present study. Apart from the different working fluid (R113), the other differences from the present study are tube diameter (3.15 mm) and flow orientation (vertical upwards and downwards). However, it has been reported by Wambsganss et al. [3] that this correlation also works well for boiling heat transfer of refrigerant R113 flowing in a horizontal tube with a diameter of 2.92 mm.

Comparison with the present data, given in Fig. 11(a), shows that this correlation works well for our horizontal R11 flows, even though the properties of R11 are significantly different from those of R113 (e.g., the saturation pressure of R113 is considerably lower than that of R11 at the same temperature; and the critical pressures are different, $P_{cr} = 3411$ kPa for R113 and 4409 kPa for R11). However, the correlation systematically underpredicts the heat transfer in the HCFC123 case. The failure of this correlation for HCFC123 indicates that it does not account well for the influence of fluid properties. This correlation suggests that $h \propto q^{0.714}$, which is very close to the relationship found in this work. However, the correlation also has $h \propto G^{0.143}$, but no influence of G is observed in the present study. Furthermore there is no allowance for system pressure in the correlation. Therefore, it is suggested that care must be taken when using this correlation, even in the region of saturated nucleate boiling of refrigerants in small diameter tubes.

The unmodified Chen [10] correlation considerably overpredicts the actual heat transfer coefficient (not shown here). However, the method does correctly identify the dominant mechanism as nucleate boiling and there is very little scatter in the comparison. The overprediction is in fact mainly caused by an overprediction of nucleate boiling rates. It was found by Liu and Winterton [11] that the Chen correlation does overpredict a large database of R11 flow boiling heat transfer rates by, on average, 70%.

In this method the heat transfer coefficient is given as the sum of the two heat transfer contributions:

$$h = h_{CB} + h_{NB} = Fh_L + Sh_{PB} \tag{7}$$

In the comparison, the calculation of h follows the method of Collier [21] with the replacement of X_{tt} by X. Although this correlation overpredicts the present data, it indicates that the present measurements are all in a region dominated by nucleate boiling, since $h_{CB} \ll h_{NB}$. Furthermore, the correlation provides a reasonable prediction of air–water non-boiling heat transfer data [6,7] and S is always less than unity. Therefore, it seems that the overprediction of heat transfer coefficients is the result of inaccuracy in the prediction of h_{PB}. If h_{PB} is calculated using Cooper's method [16], as shown in Fig. 11(b), the predictions are much improved.

In Fig. 12, the experimental data are compared with the predictions of Cooper's correlation for nucleate pool boiling. Clearly, $h_{exp} > h_{PB}$ and h_{exp}/h_{PB} increase with increasing h_{exp} over the entire range. This suggests that either the contribution of forced convection may be significant or that Cooper's correlation slightly underpredicts the present nucleate boiling in fine passages. In Cooper's correlation, the pool boiling heat

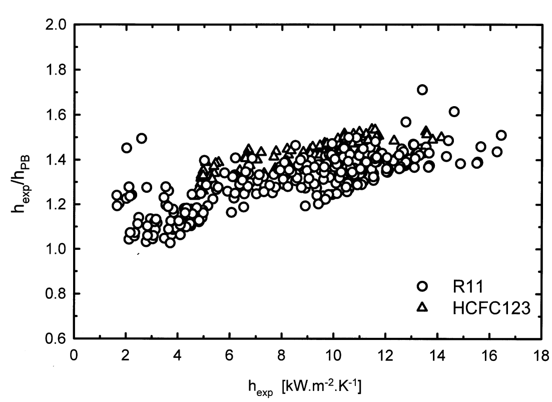

Fig. 12. A plot of h_{exp}/h_{PB} (with h_{PB} calculated from Cooper's correlation [14]) plotted against the experimentally determined heat transfer coefficients.

transfer coefficient is given by

$$h_{\mathrm{PB}} = 55\left(\frac{P}{P_{\mathrm{cr}}}\right)^{0.12} q^{2/3}\left[-\log\left(\frac{P}{P_{\mathrm{cr}}}\right)\right]^{-0.55} M^{0.5} \tag{8}$$

where M is the molecular weight of the fluid. The power dependence here on the heat flux (0.667) is somewhat lower than found in our experiments, which may explain why the underprediction is progressively worse as the heat flux increases. We cannot comment on the form of the pressure dependence in the correlation because it is weak and not quantitatively assessable in our data.

The comparison of the data, with the correlation of Gungor and Winterton [9], is presented in Fig. 11(c). A reasonable average prediction of the experimental data is achieved, but there is also considerable scatter. This correlation assumes that the reduced two-phase heat transfer coefficient, h/h_{L}, is a function of the boiling number, Bo, the vapour quality, x, and the density ratio of the two phases via

$$\frac{h}{h_{\mathrm{L}}} = 1 + 3000Bo^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}}\right)^{0.41} \tag{9}$$

where h_{L} is the liquid phase heat transfer coefficient, while ρ_{L} and ρ_{V} are the liquid and vapour densities, respectively. In this correlation, the nucleate boiling contribution is represented by the second term on the right-hand side of the equation. For heat transfer dominated by nucleate boiling, h should be a strong function of h_{L} and $q^{0.86}$. However, the present data do not show any effect of h_{L} and there is a slight difference in the exponent of the heat flux with the experimental data giving $h \propto q^{0.72}$ whilst the correlation gives $q^{0.86}$.

The comparison with the Klimenko [13] correlation is shown in Fig. 11(d). Although this correlation is based on 3215 data points using a variety of fluids, and with diameters down to 0.47 mm, it tends to overpredict the measurements in R11 flows, especially in the region of low heat transfer coefficients. The prediction for the flows of HCFC123 are substantially better. Overall, there is little scatter in the predictions for either fluid.

In contrast with the methodology of the Chen correlation, the Klimenko correlation uses only the maximum of the nucleate and convective components to estimate the heat transfer rates in two-phase flow boiling. This correlation also identifies all the experimental data as relating to nucleate boiling conditions. The overprediction of the data is therefore due to the inaccuracy in the prediction of the nucleate boiling heat transfer rate. For example, according to this correlation $h \propto q^{0.6}$, whereas the present data follow $h \propto q^{0.72}$. The effect of q on h for HCFC123 flows is similar to that for R11 flows in both the measurements and the prediction. Thus, the failure to reproduce the R11 data as well as the HCFC123 data indicates that the correlation does not describe the effects of fluid properties well, at least in the parameter range examined here.

The experimental data are compared with the predictions of the Liu and Winterton [11] correlation in Fig. 11(e). This correlation uses the method suggested by Kutateladze [14] to combine the two contributions of heat transfer asymptotically. i.e.

$$h = \left(h_{\mathrm{CB}}^{n} + h_{\mathrm{NB}}^{n}\right)^{1/n} \tag{10}$$

where $n = 2$. This correlation applies equally well to the prediction of R11 and HCFC123 flow, but underpredicts the experimental data in the whole range examined. The scatter in the predictions is small. The predictions are closer to the measurements when the heat transfer coefficient is lower. The correlation does identify the dominance of nucleate boiling under most of the conditions of the present study, but some of the data are placed in the region dominated by convection, or by both of the mechanisms, especially when the heat transfer coefficient is low. According to the comparison with heat transfer data for air–water flows in fine passages, this correlation tends to overpredict the convective component, h_{CB} [6].

The Steiner and Taborek [12] correlation is again based on an asymptotic model (Eq. (10)), but in this case $n = 3$. The comparison of this correlation with the experimental data is presented in Fig. 11(f). Consistent with the present observations, the correlation identifies that the present data are dominated by nucleate boiling, but the predictions are about 2–3 times higher than the measured values, and the correlation does not work equally well for R11 and HCFC123.

In summary, the present data for R11 and HCFC123 flow boiling show the heat transfer process to be dominated by nucleate boiling over the wide range of parameters considered. The heat transfer coefficient is seen to depend strongly on the heat flux and system pressure. Comparison of the experimental data with a range of correlations yields no correlation that can satisfactorily predict, without adjustment, all of the experimental data for both R11 and HCFC123 flow boiling over the ranges examined. The dominance of nucleate boiling behaviour in R11 and HCFC123 flow boiling derives from the low liquid phase thermal conductivities of these two fluids and the low Reynolds numbers of the flows in ducts with small diameter. However, most of the correlations do correctly identify nucleate boiling as the dominant mechanism.

Use of Cooper's special pool boiling correlation, without adjustment for flow effects such as "suppression", provides the best overall description of the present data.

6. Conclusions

The flow boiling heat transfer to a gas–liquid two-phase flow in a narrow duct has been studied experimentally. The investigation of R11 and HCFC123 flow boiling heat transfer shows that:

(a) the heat transfer coefficients are independent of mass flux and vapour quality, but are strong functions of the heat flux and system pressure in the saturated region.
(b) Nucleate boiling is the dominant mechanism over a wide range of flow conditions. Convective effects are less important because of the relatively low Reynolds numbers and liquid conductivity.
(c) The difference in heat transfer coefficients for R11 and HCFC123 flow boiling in fine passage are not very significant under the same conditions.
(d) None of the correlations examined can predict the present experimental data over the whole range examined, but most of them do identify the dominance of nucleate boiling. The Cooper pool boiling correlation describes our flow boiling data with reasonable accuracy.

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