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STUDY OF FLOW BOILING CHARACTERISTICS OF R134*a* IN ANNULUS OF ENHANCED SURFACE TUBING

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SUMMARY

The paper presents an experimental study of the flow-boiling heat-transfer characteristics of R12 and R134*a* in the annulus of a horizontal enhanced-surface-tubing evaporator. The test section has an inner-tube bore diameter of 17·5 mm, an envelope diameter of 28·6 mm and an outer smooth tube of 32·3 mm inside diameter. The ranges of heat flux and mass velocity covered in the tests were 5–25 kW/m^2 and 180–290 kg/m^2/s, respectively, at a pressure of 365 kPa. In order to establish the flow regime conditions at the inlet to the test section, the test rig allows for the visualization of refrigerant flow through the preheater. The experiments show two regions of heat transfer: a nucleate boiling region where the heat transfer depends mainly on heat flux, and a forced convective region where the heat transfer depends only on the refrigerant flow rate.

KEY WORDS Boiling R134*a* Pressure drop Heat transfer Enhanced surface

INTRODUCTION

During the next few years, chlorofluorocarbon (CFC) refrigerants will be phased out owing to environmental considerations. The impending CFC restriction necessitates the evaluation of new compounds as potential alternatives to regulated refrigerants. In the area of refrigeration, air conditioning and heat pumps, R12 accounts for nearly 50% of the total use of all controlled CFCs [1]. At present, the most likely substitute for R12 (CCl_2F_2) is R134*a* (CH_2FCF_3), which has no ozone depletion potential, low greenhouse warming potential, and appears to have acceptable thermodynamic properties.

In order to evaluate the performance of R134*a* as a working fluid in refrigeration applications, knowledge of the thermophysical properties is required. In particular, the heat-transfer coefficients and pressure drops have a major impact on the design of heat exchangers for vapour-compression systems.

Numerous experimental studies on the horizontal flow-boiling heat transfer in tubes and annulus have been reported in literature, e.g. by Ross [2], Anderson *et al.* [3] and Chaddock and Noerager [4]. These studies dealt only with smooth tubes. On the other hand, some investigations have treated single-phase flow in enhanced-surface tubing (double-fluted tubes) [5–7]. Bergles [5] presented a literature survey and assessed the thermal/hydraulic performance of spiral-fluted tubes. Richards *et al.* [7] conducted an experimental study of pressure drop and heat transfer for turbulent flow inside 12 different double-fluted tubes.

In the present paper, single- and two-phase flow-boiling characteristics of the new alternative refrigerant R134*a*, as well as R12 and R22 in a horizontal annulus of an enhanced-surface tube, are presented and discussed. Generalized correlation for predicting the local heat-transfer coefficient and the pressure drop during evaporation are developed to predict the relative performance of these fluids under different operating conditions.

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EXPERIMENTAL APPARATUS

A schematic view of the experimental apparatus is presented in Figures 1*a* and *b*. The test rig was made up of three closed flow loops: a refrigerant flow loop, a heating water flow loop, and a brine flow loop.

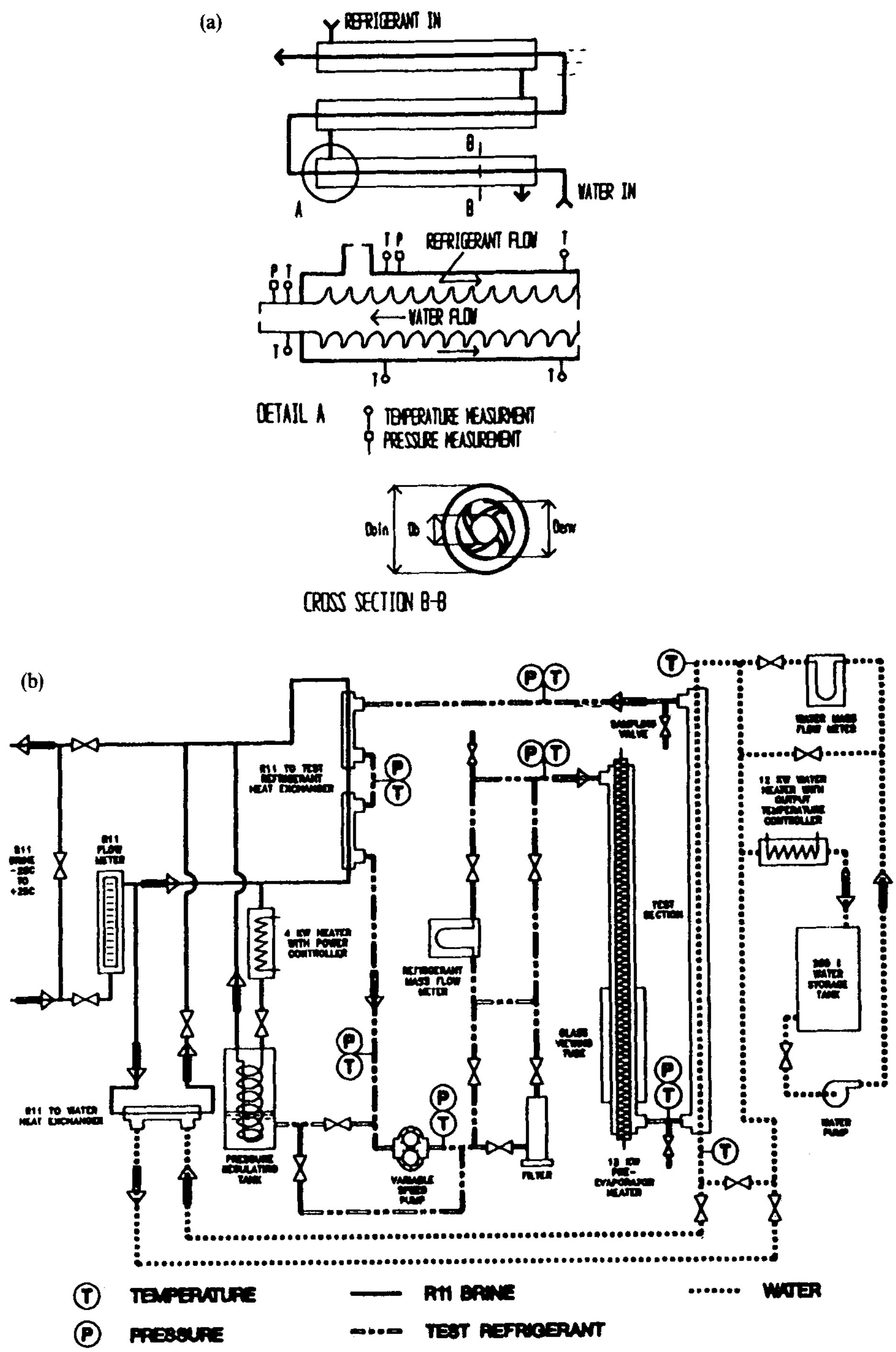

Figure 1. Schematic diagram of experimental apparatus (*a*), and schematic view of test section (*b*)

The refrigerant flow loop consisted of a variable speed gear pump which delivered subcooled liquid to the preheater. A Coriolis type flow meter was installed between the pump and the preheater. The subcooled liquid was heated by passing a direct current through the inner tube of the preheater section, which controls the refrigerant condition at the entry to the test section. The flow visualisation of the flow regime entering the test section was observed in the annulus of the preheater through a 1·8 m long, 28·6 mm-outer-diameter glass tube which surrounds the heated inner tube. Refrigerant flow conditions were also observed through a sight glass located at the test section outlet.

A horizontal double-fluted tube test section, 1·2 m long, and of equivalent diameter 0·024 m, with the test fluid flowing through the annular passage, and the heating water flowing countercurrently through the inner tube, was used. Eight copper–constantan (T-type) thermocouples were equally spaced at 120 mm intervals for measuring the local test fluid conditions. 16 T-type thermocouples were soldered on the top and bottom of the spiral grooves at eight locations for measuring the outer-tube wall temperatures of the double-fluted tube. The saturation pressure at the inlet of the test section was measured by a pressure transducer. A differential pressure transducer was used to measure the pressure drop across the test section. The mean temperatures of the heating water and refrigerant at the inlet and outlet of the test section were measured by platinum RTD and T-type thermocouples, respectively. In order to reduce heat loss to the surroundings, the test section was covered with 12·7 mm-thick foam insulation.

The refrigerant leaving the test section was condensed and subcooled by an R11 brine circuit, which forms part of the refrigeration system of a psychrometric calorimeter. Two pressure transducers and two T-type thermocouples were used to measure the refrigerant temperatures at the inlet of the subcooler and at the inlet of the variable speed pump.

The water flow loop consisted of a pump, a Coriolis flow meter, and a controller to maintain a constant water temperature. A heat exchanger was connected to the brine flow circuit for heat-removal purposes.

Test procedure

Data collection was carried out using an HP computer and data acquisition system. Each scan of the various channels recorded pressures, temperatures, mass flow rates, power etc. The instrumentation accuracy was first checked by making energy balance measurements in single-phase flow. During the tests, it usually took less than 1 h for the system to reach steady-state conditions, which were obtained when the last three scans, taken at 5 min intervals, indicated small variations ($<2\%$ deviation) from the set-up conditions (inlet saturation pressure, heat flux, and mass flow rate). The pressure at the inlet of the test section was kept at 365 kPa for R12 and R134*a* and at 570 kPa for R22. Three mass velocities of 181, 242 and 288 kg/m^2 were covered in the experiments. Heat fluxes ranged from 7 to 24 kW/m^2. The test section inlet quality varied from 0·1 to 0·6. The refrigerant capacities were deduced from the water/refrigerant energy balance. The maximum deviation of energy balance was less than 5% in the calculations, the double-fluted tube was treated as a smooth tube of equivalent cross-section. The cross-section was calculated by measuring the volume of water and finding the diameter of a cylinder with the same length that gives an equivalent volume. This diameter permits a direct comparison of the performance of the augmented tube with that of a plain tube.

SINGLE PHASE

Heat transfer

Wiegand [8] studied a large amount of annular heat-transfer data and recommended, for flow in smooth annuli,

$$h_l = c\left(\frac{k_l}{d_n}\right) Re^b \, Pr^{0\cdot4}\left(\frac{d_o}{d_n}\right)^{0\cdot45} \tag{1}$$

where $c = 0{\cdot}023$ and $b = 0{\cdot}8$.

The present study uses the general form as in equation (1), and a regression analysis has been carried out to obtain the b and c coefficients. The R134*a* single-phase liquid heat-transfer coefficient in enhanced-annulus tubes, based on the flow, can be described by the following correlation:

$$h_l = 0{\cdot}0254\left(\frac{k_l}{d_n}\right)Re^{0{\cdot}835}\,Pr^{0{\cdot}4}\left(\frac{d_o}{d_n}\right)^{0{\cdot}45} \tag{2}$$

where $Re > 5000$, and properties were evaluated at the bulk fluid temperature. The data for the transport properties of R12 are available from ASHRAE and those for R134*a* are taken from Shankland *et al.* [10] and McLinden *et al.* [11]. Only the liquid thermal conductivity and vapour viscosity of R134*a* are not currently available. The liquid thermal conductivity is a key parameter where the comparison of heat-transfer coefficients between refrigerants is concerned. Hence, an attempt has been made to predict the liquid thermal conductivity of R134*a*. Reid *et al.* [12] suggested a correlation for nonpolar fluids, relating the thermal conductivity to the reduced temperature and reduced normal boiling temperature of the fluid concerned:

$$k_l = \frac{1{\cdot}1051}{M^{1/2}}\left(\frac{3 + 20(1 - T_r)^{2/3}}{3 + 20(1 - T_{rb})^{2/3}}\right) \tag{3}$$

where $T_r = T/T_c$ and $T_{rb} = T_b/T_c$.

The application of equation (3) can lead to deviations approaching 25% from precisely established data. Therefore, recently published data by Cross *et al.* [13] on the thermal conductivity for saturated liquid R134*a* was employed in this research work. The following proposed correlation is accurate to within $\pm$ 1·5%:

$$k_l = 0{\cdot}09189 - 4{\cdot}0807 \times 10^{-4}T \tag{4}$$

where T is the refrigerant temperature in degrees celsius, and k_l represents the thermal conductivity of saturated liquid in watts per kilometre.

The accuracy of the proposed correlation, equation (2), for single-phase flow was verified by comparing the experimentally determined heat-transfer coefficient with those from the proposed correlation. The results presented in Figure 2 show that the deviation between the experimental and predicted heat transfer coefficients was less than 5%.

The heat transfer with liquid refrigerant in smooth tubes was predicted using the Weigand correlation [8], compared with that of fluted tubes (see equation (1)) and plotted in Figure 3. The heat-transfer enhancement factors lay between 1·60 and 1·90 at a Reynolds number of $1{\cdot}5 \times 10^4$. It appears that the enhancement factors

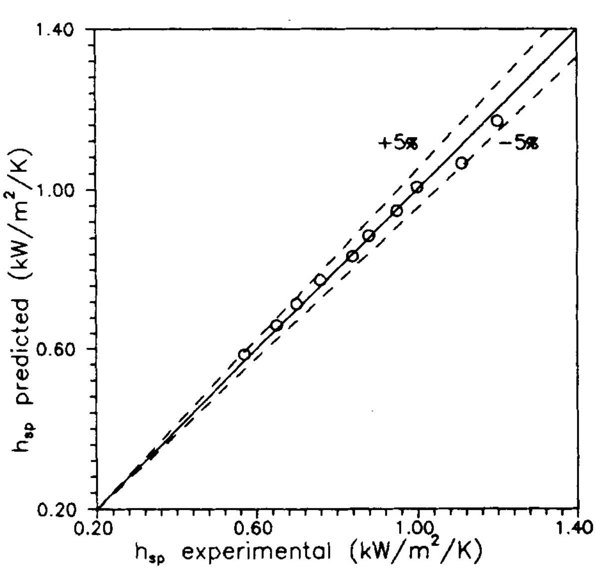

Figure 2. Comparison of experimental and predicted liquid-phase heat-transfer coefficients for R12 in enhanced-surface tube

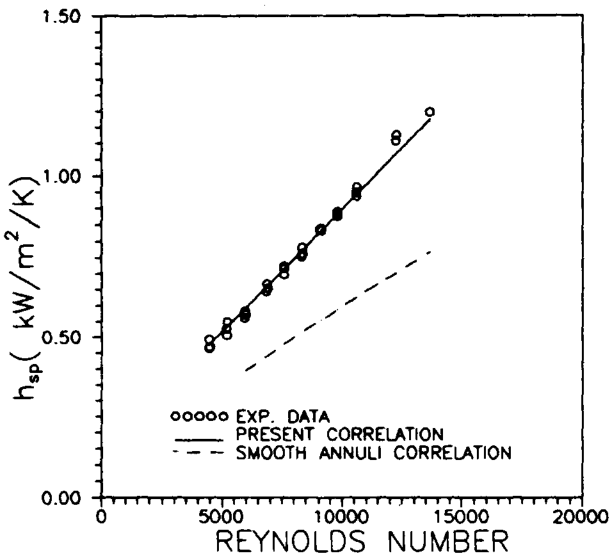

Figure 3. Liquid-phase heat-transfer coefficient of R12 in enhanced-surface tube

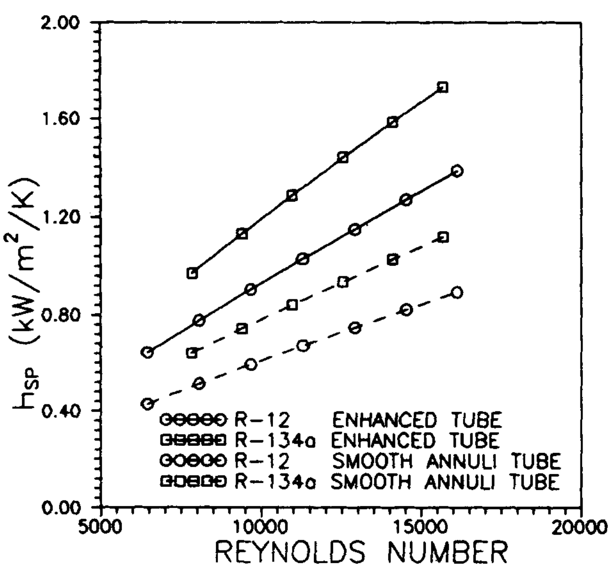

Figure 4. Comparison of liquid-phase heat-transfer coefficient of R12 and R134*a* between smooth-annulus tube and enhanced-surface tube

were a function of the Reynolds number. Figure 4 shows a comparison between liquid refrigerant R12 and R134*a* in the annuli of smooth and enhanced-surface tubes. It appears from this Figure that the enhanced surface displays a significant increase in the heat-transfer coefficient compared with smooth tubes. Heat-transfer factors were found to be a functional dependent of the Reynolds number. On the other hand, the enhanced factors were more significant with R134*a* than with R12 at the same Reynolds number.

Friction factor

Moody friction factors were calculated using the pressure-drop data and the following equation:

$$f = \frac{1}{2}\left(\frac{dP}{dZ}\right)\frac{\rho D_e}{G^2} \tag{5}$$

Inlet and outlet pressure losses for the test section were neglected owing to their insignificant effect on the friction pressure losses. Figure 5 presents the single-phase pressure drop of refrigerants R12, R22 and R134*a* as a function of the Reynolds number. In addition, a comparison between the single-phase frictional pressure losses in smooth-annulus tubes and double-fluted tubes is shown in Figure 5. As expected, the fluted tube surface leads to a pressure drop higher than that of the smooth tube. On the other hand, Figure 5 shows that the pressure penalty factor is increased on increasing the Reynolds number. In addition, it appears from these data that the pressure penalty R12 and R22 at the same Reynolds number.

The friction factor data in the annulus of a double-fluted tube fit an equation of the following form:

$$f = aRe^{1/n} \tag{6}$$

where $a = 0{\cdot}471$ and $n = 0{\cdot}236$. The data covered Reynolds numbers between 5000 and 15000. The results for the friction factor of the annulus of an enhanced-surface tube are depicted in Figure 6, and can be compared with those of the single-phase flow in a smooth-annulus tube. The smooth-annulus-tube pressure drop was predicted using the Kays *et al.* [14] correlation.

$$f = 0{\cdot}34Re^{-0{\cdot}25} \tag{7}$$

A significant increase in the friction factor of the enhanced-surface tube over that of a smooth tube can be observed in Figure 6.

TWO-PHASE FLOW

Previous studies have revealed that the transition quality from a partial boiling region to a convective evaporation region greatly influences the fluid properties, i.e. the thermal conductivity and the heat of evaporation. In the following, the boiling-flow characteristics and the heat transfer will be outlined.

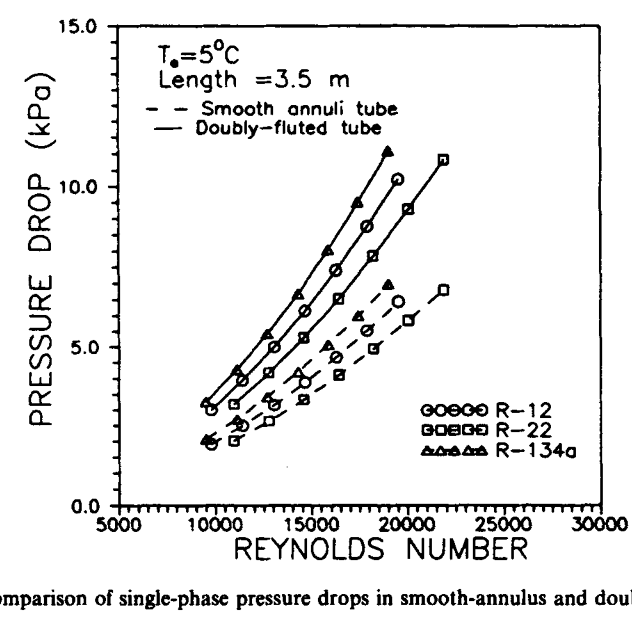

Figure 5. Comparison of single-phase pressure drops in smooth-annulus and double-fluted tubes

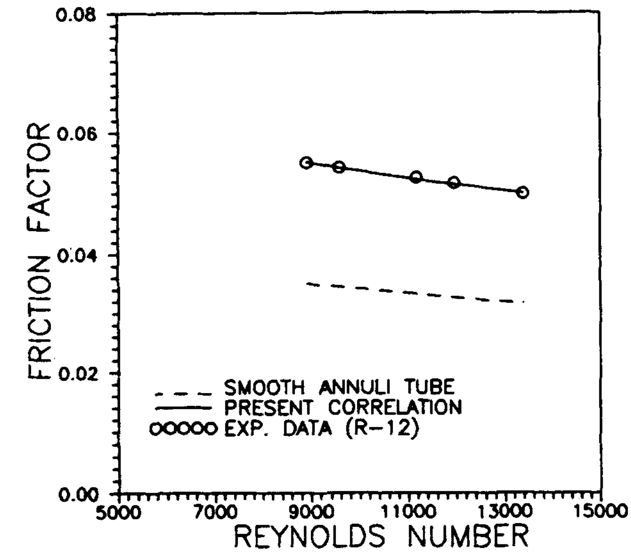

Figure 6.

Two-phase pressure drop

The static pressure gradient can be calculated in terms of the frictional and momentum components:

$$\left.\frac{dP}{dZ}\right|_{TP} = \left(\frac{dP}{dZ}\right)_f + \left(\frac{dP}{dZ}\right)_a \tag{8}$$

where $dP/dZ|_f$ and $dP/dZ|_a$ are the frictional and acceleration pressure drops along the flow path.

The present model assumes that the working fluid is evaporated from a saturated liquid to a saturated vapour. During the vapour–liquid mixture region, the vapour mass fraction has a linear dependence on the quality over the heated length. Therefore the following expression gives the total two-phase pressure drop:

$$\Delta P_{tp} = G^2\left\{\frac{x_e^2}{\alpha_e \rho_v} + \frac{1}{\rho_l}\left(\frac{(1-x_e)^2}{(1-\alpha_e)} - 1\right)\right\} + \frac{2f_{flo}G^2 L_e}{D\rho_l}\frac{1}{x_e}\int_0^{x_e} \Phi_{flo}^2\, dx$$

and

$$\Phi_{flo} = \left(\frac{\Delta P_{tp}}{\Delta P_{flo}}\right)^{1/2} \tag{9}$$

where ΔP_{tp} and ΔP_{flo} are the frictional two-phase pressure drop and the frictional pressure drop of the liquid phase flowing alone at the same mass flux.

Frictional pressure drop: Lockhart and Martinelli [15] employed the following two-phase flow-multiplier expression to calculate the frictional pressure drop:

$$\Phi_{flo}^2 = \frac{\Delta P_f}{\Delta P_{flo}} \tag{10}$$

where ΔP_f is the frictional two-phase flow pressure drop, and ΔP_{flo} represents the frictional liquid pressure drop flowing with the same mass flow rate in the annulus.

On the other hand, Jung *et al.* [16] recently introduced a multiplier to calculate the frictional pressure drop in terms of the total pressure drop as follows:

$$\Phi_{lo}^2 = \frac{\Delta P_{TP}}{\Delta P_1} \tag{11}$$

Φ_{to}^2 can be correlated to the quality function X_{tt}:

$$\Phi_{to}^2 = f(X_{tt}) \tag{12}$$

where

$$X_{tt} = \left(\frac{1-x}{x}\right)^{1-n/2} \left(\frac{\mu_1}{\mu_v}\right)^{n/2} \left(\frac{\rho_v}{\rho_l}\right)^{1/2} \tag{13}$$

A regression analysis using the measured data yielded the following expression:

$$\Phi_{to} = \varepsilon \left(\frac{1}{X_{tt}}\right)^{\beta} \tag{14}$$

where $\varepsilon = 2{\cdot}9$, and $\beta = 0{\cdot}62$.

On the other hand, the expression developed by Martinelli and Nelson [17] is also based on the total liquid flow:

$$\Phi_{tp}^2 = \frac{\Delta P_{TP}}{\Delta P_{flo}} \tag{15}$$

On introducing the ζ which relates the Jung *et al.* expression to the Martinelli–Nelson equation,

$$\zeta = \frac{\Phi_{tp}^2}{\Phi_{to}^2} = \frac{\Delta P_l}{\Delta P_{flo}} \tag{16}$$

where $\zeta = (1-x)^{2-n}$, and $n = 0{\cdot}236$, one can obtain

$$\Phi_{tp}^2 = (1-x)^{2-n} \Phi_{to}^2 \tag{17}$$

On applying our data to the different refrigerants, the following expression for Φ_{tp}^2 can be obtained:

$$\Phi_{tp}^2 = 8{\cdot}41(1-x)^{1{\cdot}764} X_{tt}^{-1{\cdot}24} \tag{18}$$

where

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0{\cdot}882} \left(\frac{\mu_1}{\mu_v}\right)^{0{\cdot}118} \left(\frac{\rho_v}{\rho_1}\right)^{0{\cdot}5} \tag{19}$$

Acceleration pressure drop: In order to calculate the acceleration pressure drop, the void fraction needs to be known. The following expression, proposed by Wallis [19], has been employed in this study:

$$\alpha = (1 + x_{tt}^{0{\cdot}8})^{-0{\cdot}378} \tag{20}$$

where α is the void fraction and X_{tt} is defined in equation (13).

This expression is limited to separated turbulent flow. However, for X_{tt} values higher than 10, the following Lockhart–Martinelli [15] equation has been used:

$$\alpha = 0{\cdot}823 - 0{\cdot}157 \ln(X_{tt}) \tag{21}$$

Numerical calculations presented in Figure 7 showed that the calculated phase acceleration pressure drop was less than 5% of the measured two-phase pressure drop. Therefore only the two-phase frictional pressure is considered in this study ($\Phi_{flo}^2 = \Phi_{tp}^2$).

Combining equations (18) and (19) with equation (9) yields

$$\Delta P_{TP} = 2 \frac{f_{flo} G^2 L}{\rho_l D} \frac{1}{\Delta x} \int_{x_i}^{x_e} \Phi_{tp}^2 \mathrm{d}x \tag{22}$$

where

$$f_{flo} = 0{\cdot}471\, Re^{-0{\cdot}236} \tag{23}$$

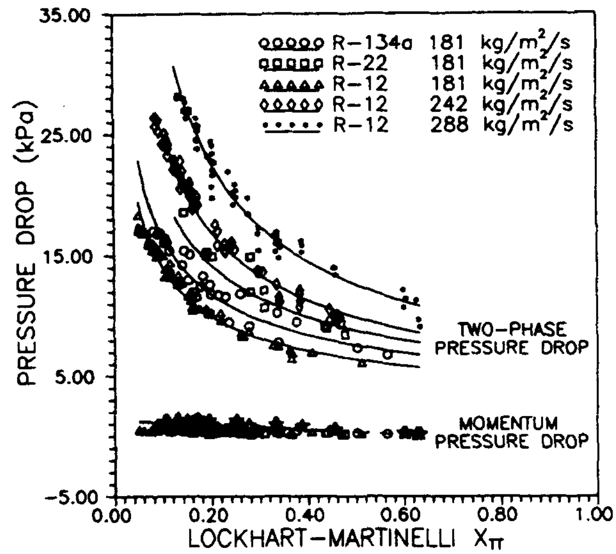

Figure 7. Effect of acceleration pressure drop on two-phase pressure drop

and

$$Re = GD/\mu_1. \tag{24}$$

Figure 8 shows a comparison between the measured two-phase pressure drop in the annulus of the enhanced-surface tubing, and those predicted by the proposed correlation (equation (22)). It is evident from this Figure that the proposed correlation was found to correlate the data with a mean deviation of 10%.

No attempt has been made to compare the proposed data with other experimental data or correlations because, to the authors' knowledge, no data exist.

Heat transfer

Two different correlations are proposed to cover the two regions of heat transfer, namely, the transition boiling regime and the fully forced-convective regime.

The correlation form which was developed by Chen [6] for two-phase saturated flow boiling, states that the two-phase heat-transfer coefficient is composed of the sum of two terms covering the nucleate boiling and forced-convective boiling contributions:

$$h_{tp} = S\, h_{NB} + \Omega_{FC}\, h_l \tag{25}$$

Jung and Raderrmacher [18] employed equation (25) with some modifications in the nucleate boiling term:

$$h_{tp} = \Omega_{NB}\, h_{SA} + \Omega_{FC}\, h_l \tag{26}$$

The present study uses the general form of equation (26) to predict the local heat-transfer coefficients in the transition boiling region.

The values of h_l are as defined in equation (2).

Stephan and Abdelsalam [20] collected data from different investigations and, with the aid of regression analysis techniques, derived the following correlation for nucleate pool boiling:

$$h_{SA} = 207\frac{k_l}{b_d}\left(\frac{qb_d}{k_l T_{sat}}\right)^{0.745}\left(\frac{\rho_v}{\rho_l}\right)^{0.581} Pr_l^{0.533} \tag{29}$$

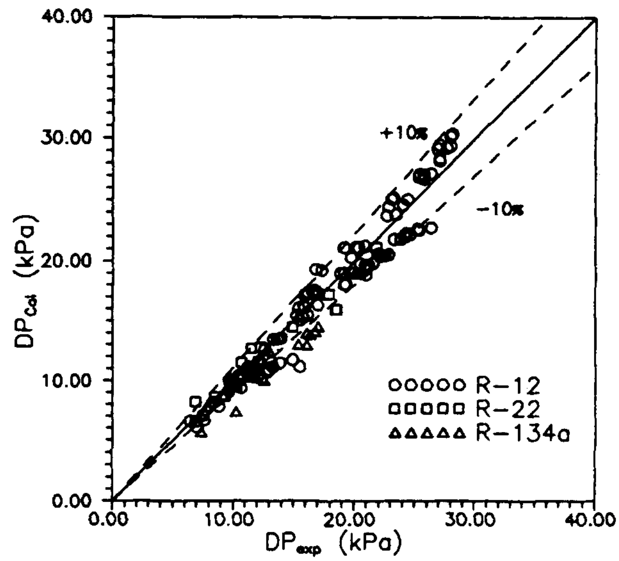

Figure 8. Comparison of measured and predicted pressure drops in enhanced-surface tubing

where

$$b_d = 0{\cdot}0146\beta\left(\frac{2\sigma}{g(\rho_l - \rho_v)}\right)^{0\cdot5}$$

and $\beta = 35°$.

The following generalized forms are suggested for the multiplying factors Ω_{NB} and Ω_{FC}:

$$\Omega_{NB} = a_{NB} Bo^{n_{NB}} X_{tt}^{m_{NB}} \tag{28}$$

$$\Omega_{FC} = a_{FC}\left(\frac{1}{X_{tt}}\right)^{m_{FC}} \tag{29}$$

At a higher quality, the two-phase boiling heat-transfer coefficients depend only on the mass-flow rate. Hence, the first term of the right-hand side of equation (26) disappears, and the correlation for fully forced convective boiling (i.e. full suppression of nucleate boiling) is expressed as

$$h_{tp} = \Omega_{FFC} h_l \tag{30}$$

where

$$\Omega_{FFC} = a_{FFC}\left(\frac{1}{X_{tt}}\right)^{m_{FFC}} \tag{31}$$

Using the current experimental data obtained for boiling on double fluted tubing, a regression analysis was carried out to determine the appropriate factors for R12 and R134*a* as defined in equations (28), (29) and (31). The corresponding *a*, *n* and *m* values which fit these equations are given in Table 1.

EXPERIMENTAL RESULTS AND DISCUSSION

Figures 9–12 illustrate the average heat-transfer coefficients of R12 and R134*a* as a function of inlet quality for different mass velocities and heat fluxes, at a saturation pressure of 365 kPa. As seen in these Figures, two distinct mechanisms of heat transfer are indicated: a transition boiling regime and a fully forced-convective

Table 1. Constants and exponents applied in equations (28), (29) and (31)

Regime	*a*		*n*		*m*	
Refrigerant	R12	R134*a*	R12	R134*a*	R12	R134*a*
NB	1·6	3·0	0·13	0·2	—	—
FC	4·2	4·1	—	—	0·45	0·43
FFC	6·2	5·36	—	—	0·28	0·29

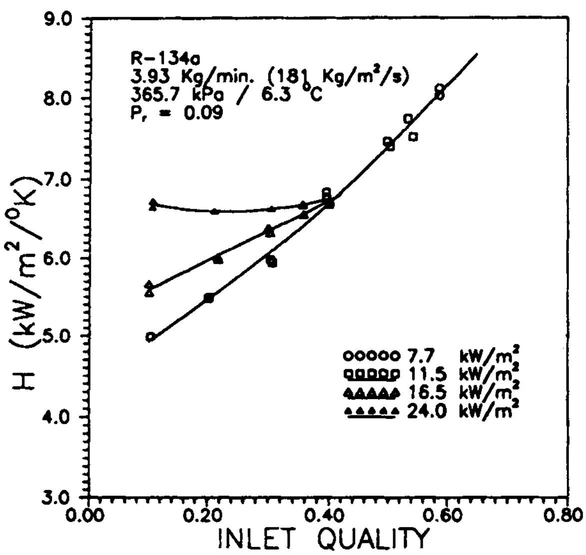

Figure 9. Average heat-transfer coefficient for R134*a* in double-fluted heat-exchanger tubing

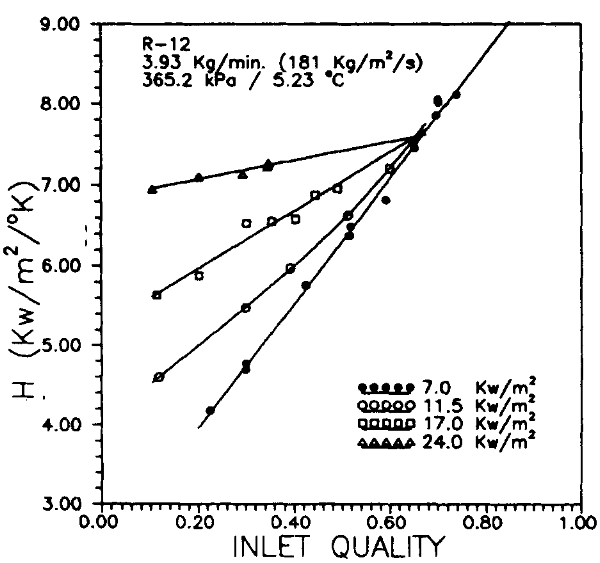

Figure 10. Average boiling heat-transfer coefficient for R12 in double-fluted heat-exchanger tubing

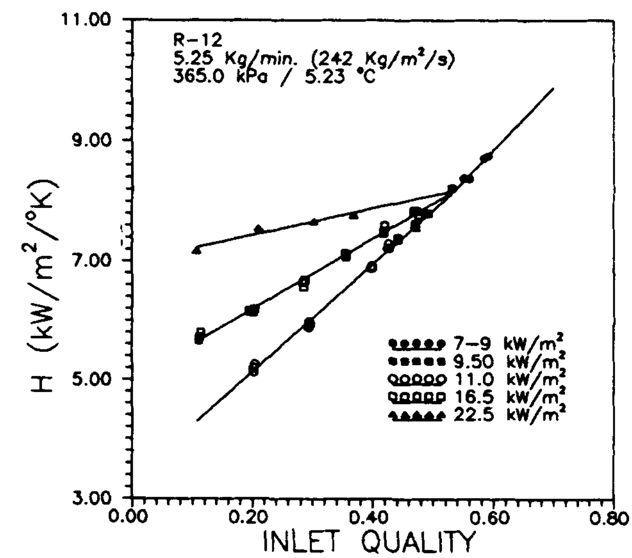

Figure 11. Average boiling heat-transfer coefficient for R12 in double-fluted heat-exchanger tubing

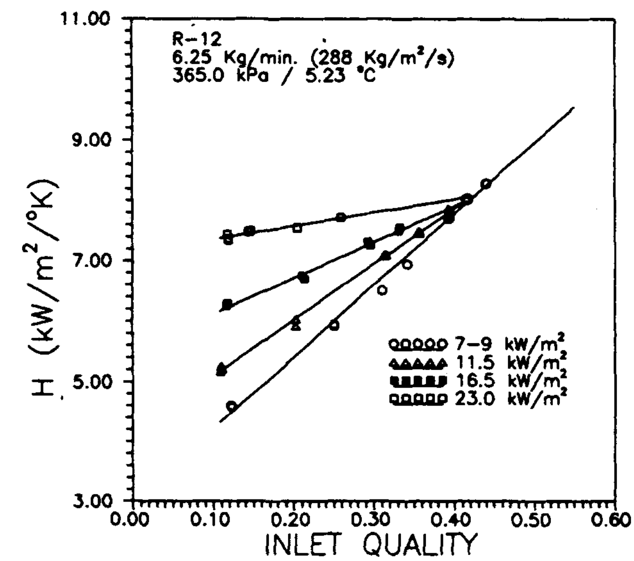

Figure 12. Average boiling heat-transfer coefficient for R12 in double-fluted heat-exchanger tubing

boiling regime. In the transition boiling regime, the heat-transfer coefficients depend mainly on the heat flux and also on the mass-flow rate. As the quality increases, the dependence of heat transfer coefficients on the heat flux is less significant. Shah [21] and Jung *et al.* [16] also noticed a strong dependence of heat-transfer coefficients on heat flux in the transition boiling region. Beyond a transition point, the forced-convection regime takes over, and the heat-transfer coefficients depend only on the mass-flow rate.

As shown in Figures 9 and 10, the transition point of R134*a* occurs at a lower inlet quality than for R12 at a mass velocity of 181 kg/m^2/s. Also, the heat-transfer coefficients for R12 are more dependent on heat flux than for R134*a*, under the same conditions of mass velocity and pressure.

The influence of the boiling number Bo on the transition point is shown in Figure 13. The boiling number expresses the ratio of the vapour mass flux perpendicular to the wall due to boiling compared to the total mass flux parallel to the heated surface, and is defined as

$$Bo = \frac{q}{Gh_{fg}} \tag{32}$$

In the case of R12, the boiling numbers are higher for the same heat flux and mass flow rate, since the heat of vaporization of R134*a* is higher than that of R12 at a given pressure. The curves indicate that the higher is the value of Bo, the higher is the quality at which transition to fully suppressed nucleate boiling occurs.

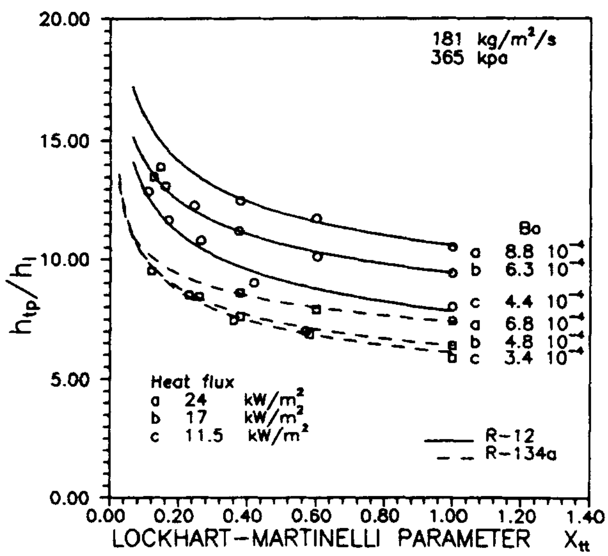

Figure 13. Flow-boiling heat-transfer results for R134*a* and R12 using dimensionless parameters

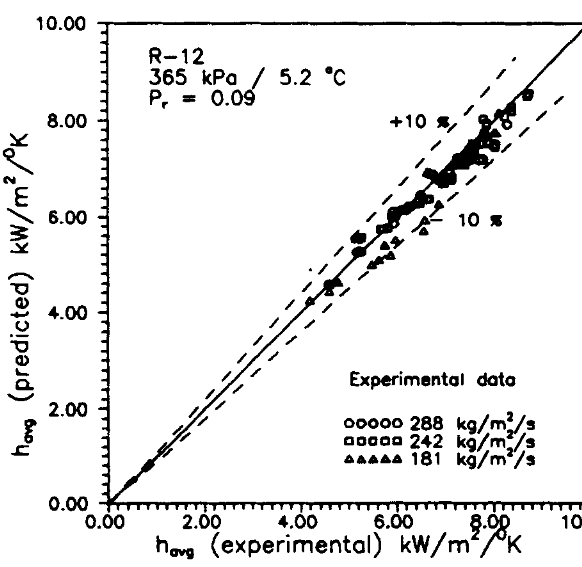

Figure 14. Comparison of experimental and predicted boiling heat-transfer coefficients for R12 in double-fluted heat-exchanger tubing

In order to compare the predicted and experimental heat-transfer coefficients, a numerical integration is carried out along the length of the test section to obtain a prediction of average heat-transfer coefficients. As shown in Figures 14 and 15, the deviation between the predicted and experimental values is within 10%.

The above experimental results and derived empirical correlations are specific to the particular tube geometry used in this series of tests. However, it is more useful to extend this work to provide a comparison of the heat-transfer performance of R12 and R134*a* which would have general applicability.

Figures 16 and 17 show the two-phase heat-transfer coefficient ratios of R134*a* to R12 which are obtained from equations (26) and (30) for the transition boiling regime, and the fully forced-convective boiling regime,

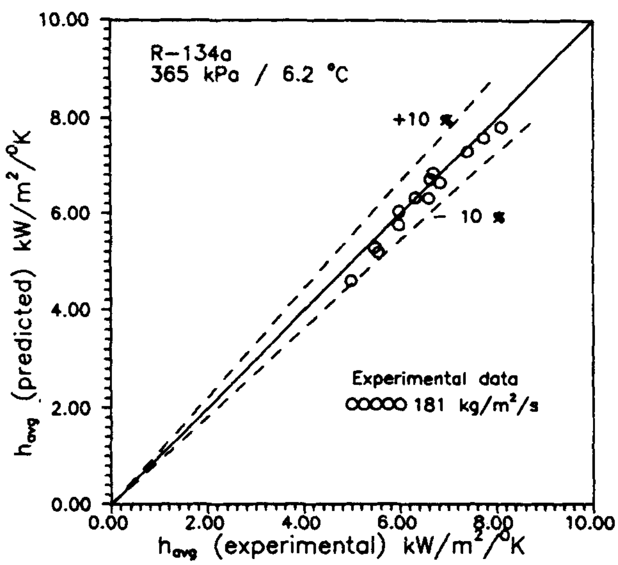

Figure 15. Comparison of experimental and predicted boiling heat-transfer coefficients for R134*a* in double-fluted heat-exchanger tubing

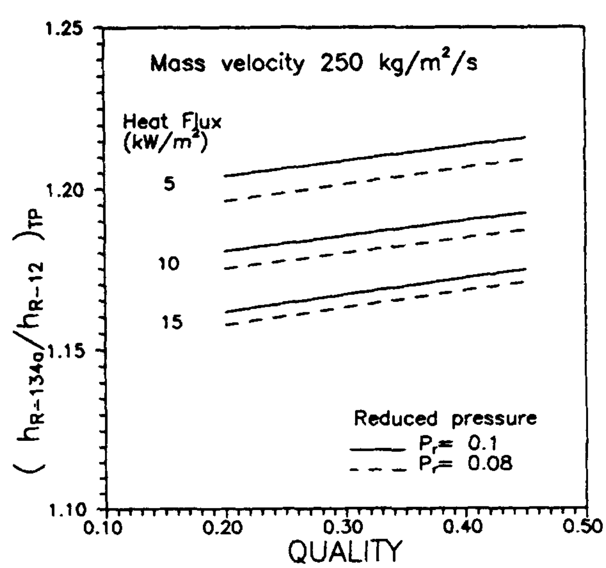

Figure 16. Two-phase heat-transfer coefficient ratio (R134*a*/R12) in transition boiling region

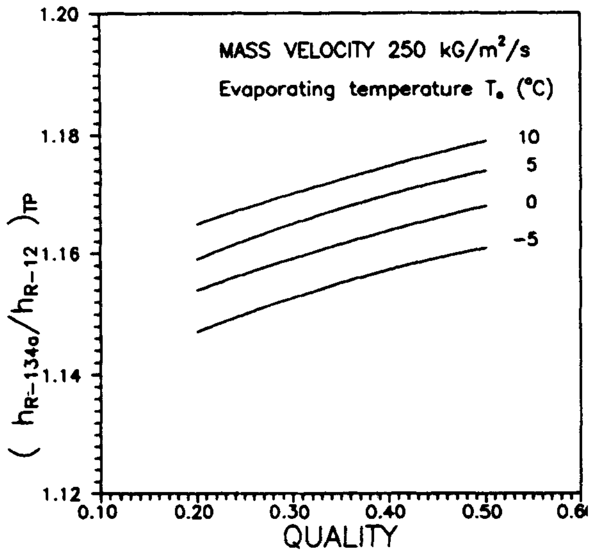

Figure 17. Two-phase heat-transfer coefficient ratio (R134*a*/R12) in fully forced-convective region

Table 2. Comparisons between liquid thermal conductivity of R12 and predicted R134*a*

Temp °C	R12*	R134*a*†	k_{lR134a}/k_{lR12}
1·7	0·0787	0·0912	1·16
4·4	0·0777	0·0900	1·16
7·2	0·0767	0·0089	1·15
10·0	0·0757	0·0878	1·15

* ASHRAE (1976) data † Predicted

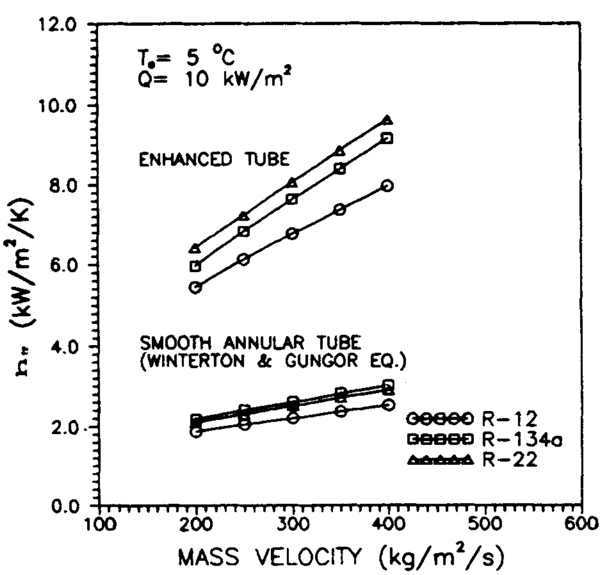

Figure 18. Evaporation heat-transfer coefficients for smooth annular tube and double-fluted tube

respectively. These graphs indicate that the heat transfer coefficients for R134*a* are typically 15–20% higher than for R12 throughout the entire evaporating temperature test range, except at higher heat flux conditions (above 20 kW/m^2). The ratio of the two-phase heat-transfer coefficients in the fully forced-convective region increases slightly with the evaporating temperature. The same trend is also observed in the transition boiling region as the reduced pressure increases, but in this region the two-phase heat-transfer coefficient ratio decreases as the heat flux increases. This is due to the nucleate-boiling heat-transfer component of R12 contributing more than that for R134*a* as the heat flux increases. In general, the improvement in the heat-transfer performance of R134*a* over R12 in both regimes is generally attributable to the effect of liquid thermal conductivity. Data presented in this study show that the thermal conductivity of R134*a* is predicted to be approximately 15% higher than that of R12 (see Table 2). The thermal conductivities of the two fluids are implied in equations (26) and (30), in evaluating h_l and h_{SA}.

Finally, in an attempt to demonstrate two-phase heat-transfer enhancement factors, Figure 18 has been constructed. As expected, the results depicted in Figure 18 showed that the heat-transfer coefficient increases with increasing mass velocity. It appears also from this Figure that the enhanced surface significantly augmented the heat-transfer coefficient compared with those of smooth tubes. The boiling heat-transfer coefficient in the annuli of smooth tubes has been predicted using the Gungor–Winterton correlation [22]. On the other hand, the enhancement factors seem to increase with increasing mass velocity. The heat-transfer enhancement factors range from 3 to 3·30. Also, Figure 18 shows that the heat-transfer enhancement factors are more significant for R134*a* than for R12.

CONCLUSIONS

Heat-transfer coefficients were experimentally determined for R134*a* and R12 as well as for R22 during evaporation in the annulus of an enhanced'-surface heat-exchanger tubing. Empirical correlations were proposed to calculate local boiling heat-transfer coefficients for both fluids in the transition boiling region and fully forced-convective boiling region, as well as pressure drop.

Comparison of local boiling heat-transfer coefficients at the same mass velocities indicated a 15–20% higher performance for R134*a* relative to R12 for the range of conditions tested. This implies that, for the same evaporator capacity, it may be possible to reduce the heat exchanger size, or conversely maintain a constant heat-transfer surface area and operate with a higher evaporating temperature.

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NOMENCLATURE

Bo = boiling number
C_p = specific heat, kJ/kg/K
D = hydraulic diameter, m
d_e = equivalent diameter, m
d_n = nominal diameter, m
d_o = inner diameter of outer tube, m
f = friction factor
g = gravitational force, m/s^2
G = mass velocity, kg/m^2/s
h = heat-transfer coefficient, kW/m^2/K
h_{fg} = heat of vaporisation, kJ/kg
K = thermal conductivity, kW/m/K
L = test-section length, m
M = molecular weight
P = pressure, kPa
Pr = Prandtl number
q = heat flux, kW/m^2
Re = Reynolds number
S = suppression factor
T = refrigerant temperature, K
T_b = normal boiling temperature, K
T_c = critical temperature, K
T_r = reduced temperature
T_{rb} = reduced normal boiling temperature
x = quality
X_{tt} = Lockhart–Martinelli parameter
z = length, m
α = void fraction
ρ = density
μ = viscosity

Ω = two-phase heat-transfer multiplier
ΔP = pressure drop

Subscripts

FC = forced-convective
FFC = fully forced-convective
l = liquid
NB = nucleate-boiling
sat = saturated
tp = two-phase
v = vapour