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Prediction of pressure drop during horizontal annular flow boiling of pure and mixed refrigerants

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Abstract—An experimental study on pressure drop during horizontal flow boiling of pure and mixed refrigerants of R22, R114, R12, and R152a is reported. More than 600 pressure drop data are taken for annular flow under uniform heat flux at reduced pressures of 0.08–0.16 (200–800 kPa). The range of heat flux and mass flow rate is 10–45 kW m^{-2} and 16–46 g s^{-1} (corresponding to 230–720 kg m^{-2} s^{-1} in terms of mass flux). The results are compared against well-known correlations; Bo Pierre's correlation failed to correlate half of the present data while Martinelli and Nelson's correlation overpredicted it by 20%. Pressure drops with both pure and mixed refrigerants, however, are well correlated by Martinelli's parameter, X_{tt}. Furthermore, no composition dependence of pressure drop is found with mixtures. A simple correlation adopting the thermodynamic corresponding states principles is developed with a chart to facilitate the estimation of pressure drop during flow boiling. The prediction agrees with the measured data with a mean deviation of 8.4%.

1. INTRODUCTION

THE PREDICTION of pressure drop during flow boiling is essential for the design of various industrial heat exchangers including evaporators in refrigeration and heat pump systems. Knowing the pressure drop determines a pumping load and enables a variety of physical properties needed for many transport correlations to be calculated at various locations in a heat exchanger. Early two-phase flow studies dealt mostly with adiabatic flow to determine the pressure drop through a pipeline for estimating a pumping load in the petrochemical industry. Since nuclear power plant technology developed in the 1960s, much work has been carried out with water–steam mixtures at high pressures and low qualities in a vertical geometry. Due to these efforts, some generalized correlations for the prediction of pressure drop with water–steam mixtures were established.

Recently, the use of non-azeotropic refrigerant mixtures as working fluids in heat pumps has shown theoretical promise, which has been verified experimentally [1]. Studies in an experimental heat pump have demonstrated an increase in the coefficient of performance of up to 32% for an optimized mixed refrigerant as compared to that of pure R22 [1]. Several other advantages have been claimed with non-azeotropic mixtures in refrigeration systems. Among these are improved thermodynamic efficiency, lower pressure ratios across the compressor, and the ability to change refrigerant composition allowing the working media's capacity to match load requirement. Despite these advantages, however, the use of refrigerant mixtures has not yet been popular due in part to the lack of such design data as pressure drop.

In heat pumps, evaporator coils are made of a small diameter tube, typically 0.01 m, and usually in a horizontal geometry, and the quality range of interest is typically 20–100%, which is in sharp contrast to the large pipe diameter, vertical geometry, and low quality range found in power plant applications. In spite of this significant difference, the pressure drop correlations based on water–steam or water–air mixture data have been adopted widely in the refrigeration and heat pump industry.

These correlations are usually provided with charts to simplify the estimation of pressure drop at various pressures and qualities. These charts are developed based upon the physical properties of water–air and/or water–steam mixtures. Since water–air and water–steam and commonly used chlorofluorocarbon (CFC) refrigerants are fundamentally different fluids, these charts may not provide the relevant information for the prediction of pressure drop during evaporation of CFC refrigerants.

Experimental studies on pressure drop during flow boiling of commonly used CFC refrigerants have been undertaken by Altman *et al.* [2], Anderson *et al.* [3], Bo Pierre [4], and Chaddock and Noerager [5]. Most of the studies used only one working fluid and some calculation methods were developed to fit their own data. The correlations thus made would not necessarily predict the other data obtained with different instrumentation and working fluids. In order to eliminate the inconsistency in measurements and develop a general correlation, data must be taken using the same instrumentation with a variety of working fluids.

One of the most important steps in the correlation development is the selection of thermophysical properties. Since most of these measurements were carried out 20–30 years ago and now fairly accurate property values of various refrigerants are available, a new

NOMENCLATURE

D	diameter [m]
f	friction factor
G	mass flux [kg m^{-2} s^{-1}]
h_{fg}	heat of evaporation [J kg^{-1}]
J	mechanical equivalent of heat, 1 in SI units
K_f	Bo Pierre's boiling number, $(J\Delta x h_{fg})/L$ [N kg^{-1}]
L	tube length
M	molecular weight
$\dot{m}$	mass flow rate [g s^{-1}]
p_r	reduced pressure
ΔP	pressure drop [Pa]
Re	Reynolds number, $G\mu_l/D$
T	temperature [K or °C]
x	mass fraction of vapor
Δx	quality change between inlet and outlet, $x_2 - x_1$
X	liquid composition based on mole
X_M	liquid composition based on mass
Y	vapor composition based on mole
Y_M	vapor composition based on mass.

Subscripts

a	accelerational
av	average
BP	Bo Pierre
f	frictional
fo	total flow assumed as liquid
l	liquid
lo	liquid phase only flowing
MN	Martinelli and Nelson
m	mixture
tp	two phase
v	vapor
vo	vapor phase only flowing
1	inlet
2	outlet.

Greek symbols

α	void fraction
μ	viscosity [Pa s]
ρ	density [kg m^{-3}]
ϕ	pressure drop multiplier.

correlation based upon the measured data using the same instrumentation with a variety of working fluids and accurate (best available) properties may be needed to better design heat exchangers.

Regarding the pressure drop during flow boiling of refrigeration mixtures, measured data are scarce and no reliable correlation is available. Singal *et al.* [6] conducted experiments with R12/R13 mixtures. The concentrations of R13 in R12 were 0–20% in intervals of 5%. The range of heat and mass flux was 5–17 kW m^{-2} and 234–454 kg m^{-2} s^{-1}, respectively. Singal *et al.* found that the Martinelli–Nelson correlation [7] overpredicted the pressure drop data with pure R12 while it underpredicted those with R12/R13 mixtures.

The measured pressure drop with the R12/R13 mixture was shown to be a function of mixture composition. Two correlations for mixtures were developed which explicitly employed a composition as a main variable. The mean deviations with the measured data were 30%. As Singal *et al.* [6] concluded, the main drawback of their correlations was the inability of extrapolating pressure drop at other compositions. This implies that these correlations would not predict adequately the pressure drop with other refrigerant mixtures at various compositions. Instead of correlating the data with mixtures employing a composition as a main variable, a more fundamental approach of using properties evaluated at local liquid and vapor phase compositions may be preferred in the development of the correlation.

This study is undertaken to investigate mixture effects (if there are any) on pressure drop as well as to provide design data with proper correlations. Of particular concern in this study are:

(1) the measurement of pressure drop during evaporation of pure and mixed refrigerants for annular flow under uniform heat flux;

(2) the comparison of pressure drop between pure and mixed refrigerants under the same condition;

(3) the comparison of the measured data with available correlations;

(4) the development of a simple correlation for design engineers.

2. EXPERIMENTS

2.1. *Experimental apparatus*

An 8 m long test section was constructed to eliminate entry length effects. It also facilitated the measurements by enabling data in a wide range of quality to be taken in a single test, which would otherwise have to be taken in many tests with a short test section. Figure 1 shows the schematic diagram of the test rig.

A semi-hermetic, oil free pump delivers subcooled liquid refrigerant to the test section. Just before the test section inlet, a calibrated turbine flow meter and a flow regulating valve are installed to measure and control the flow rate. The subcooled liquid is heated by passing a d.c. current through the tube. The vapor generated in the test section is condensed in the condenser which is cooled by a brine at −20 to −60°C. The pump then draws the liquid from the bottom of the condenser to complete the cycle.

The flow patterns are observed through the sight glasses installed before and after the test section. In order to reduce the heat gain (or loss) from (or to)

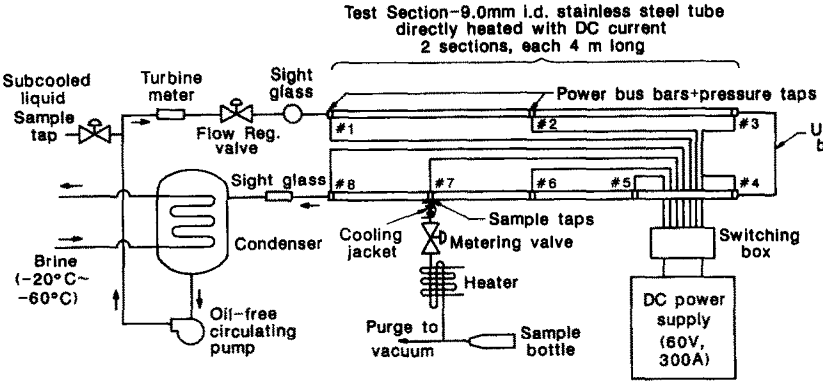

FIG. 1. Schematic diagram of the experimental apparatus.

the surroundings, the test section is covered with 0.2 m diameter fiberglass insulation.

The test section was made of two identical 4 m long, 9.1 mm i.d., type 304 stainless steel tubes (specification ASTMA 269/213) with a nominal wall thickness of 0.25 mm. Due to space limitation, these two sections were connected by a 180° U-turn bend, made of copper tube with a similar inside diameter. The test section was rigidly attached to the frame near the condenser side but was otherwise free to move axially to accommodate any thermal expansion (or contraction). It was electrically isolated from the frame and the U-turn bend by using plastic bushings at its ends.

There are three bus connections to the d.c. power supply on the first 4 m section and five on the second 4 m section. These bus connections effectively created a variable length test section, which is one of the distinctive features of the current test rig. For instance, for high heat flux and low mass flow rate, only a part of the 8 m length was heated to yield the exit quality of up to 90%. For low heat flux and high mass flow rate, however, the full length was heated to achieve a reasonably high exit quality. Further details of the test facility are available in ref. [8].

2.2. *Measurement*

2.2.1. *Pressure and pressure drop.* Figure 2 illustrates a detailed diagram of the pressure tap soldered on the power bus connection. Only the top port was used for pressure measurement, side and bottom ports were used for the local liquid sampling (see ref. [8] for further details). Fine holes (< 0.5 mm) were machined through the 0.25 mm thick stainless steel test section by electrical discharge machining (or spark erosion) to ensure a smooth, burr-free inner surface. A 3.1 mm o.d. stainless steel tube was silver-soldered on a copper bushing (18.8 mm wide, 9.4 mm i.d. and 15.6 mm o.d.) which was, in turn, soft-soldered to the copper strip which served as a power cable connection. The pressure lines were electrically isolated from the test section by using plastic bushings over a copper tube as shown in Fig. 2.

Four pressure transducers and one differential pressure transducer in conjunction with two 5-way valves were used for the pressure measurements at 8 bus connections. Two pressure transducers (0–3447 kPa, absolute pressure) were fixed at the inlet and outlet of the test section (#1 and #8 in Fig. 1). Pressures at points 2 and 3 in Fig. 1 were measured by another pressure transducer (0–3447 kPa, absolute pressure) with the aid of a 5-way valve. All absolute pressure transducers were calibrated using a dead weight pressure tester three times over 18 months. They drifted

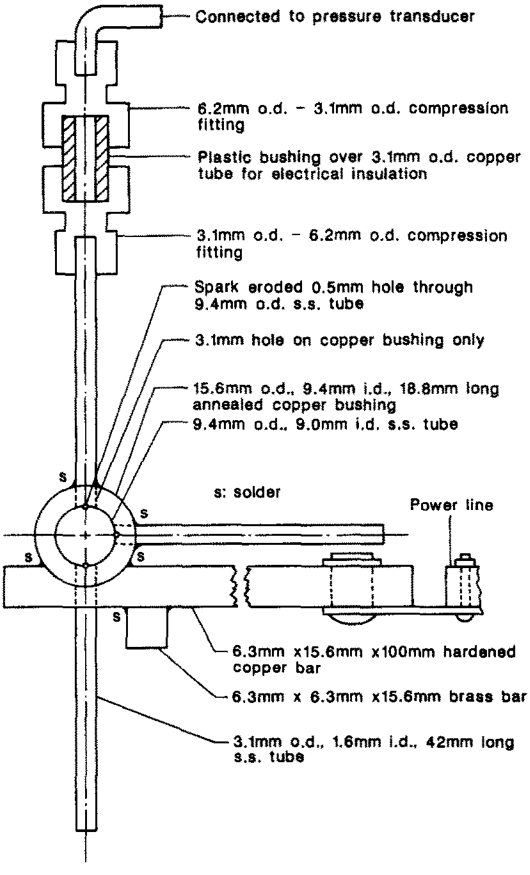

FIG. 2. Diagram of pressure tap on bus connection.

by less than ±3 kPa. The accuracy of the pressure transducers is ±1 kPa.

The differential pressure transducer (0–69 kPa, differential pressure) was used to measure the pressure drops in the second 4 m section between #4 and #8 (ΔP_{48}) and for each 1 m segment between #4 and #5, #5 and #6, #6 and #7, and #7 and #8 (ΔP_{45}, ΔP_{56}, ΔP_{67}, ΔP_{78}) with the aid of another 5-way valve. This differential pressure transducer was referenced to the outlet pressure (#8). With these pressure drops and absolute pressures at points 4 and 8, pressures at points 5–7 were determined. The accuracy of the differential pressure transducer was 1.7 kPa.

2.2.2. *Mass flow rate.* A turbine meter was installed in the subcooled liquid line about 80 diameters upstream of the test section inlet as shown in Fig. 1 to measure the mass flow rate. This measurement was used in conjunction with an energy balance on the test section to determine the specific enthalpy of the refrigerant. The enthalpy is needed to calculate the thermodynamic properties of working fluids, which will be discussed later. The signal (frequency, Hz) from the turbine meter was sent to a pulse counter as well as a counter integral to the data acquisition system. The readings from these two counters agreed within 1%. The volume flow rates were interpolated from these calibration points. The mass flow rate was then calculated by multiplying the volume flow rate to the subcooled liquid density at the inlet.

2.2.3. *Power.* A low voltage, high current power supply (60 V d.c., 300 A) was used to heat up the test section. Voltage drops across the various segments of the test section and a standard resistor (0.001 Ω) in series with the test section were measured by the digital voltmeter. The current through the test section was calculated from the voltage drop across the standard resistor and consequently the total power input to the test section was determined.

2.2.4. *Composition.* For each mixture test, upon a test being completed, a liquid sample was drawn from the subcooled liquid line before the turbine meter as shown in Fig. 1. The liquid was expanded to superheated vapor and this vapor sample was analyzed by gas chromatography to give the overall composition of the working fluid entering the test section. The reproducibility of the measurement was within 0.2% for the same sample. The variation of composition from sample to sample for the same test was less than 0.5% in mole composition.

2.3. *Data collection*

All tests were run at steady-state conditions. For most of the tests, the pressure at the outlet of the test section was kept at a reduced pressure, p_r, of 0.08, which corresponds to 400, 260, 330, and 360 kPa for pure R22, R114, R12, and R152a, respectively. To determine the effect of pressure, several tests were performed at reduced pressures of 0.12 and 0.16. The reduced pressures for mixtures were determined from the critical pressures calculated by a linear mole fraction weighting (ideal mixing rule) of the pure components' critical pressure values. Due to pressure drops across the test section, however, the pressures upstream of the outlet were up to 30% higher than those of the outlet.

Data collection and system control were carried out using a micro-computer and data acquisition system. A single scan took 90 s to record pressure drops, ΔP_{45}, ΔP_{56}, ΔP_{67}, ΔP_{78}, ΔP_{48}, absolute pressures at each pressure tap, mass flow rate, and heat flux. It usually took 1–2 h for the system to reach steady state which was indicated by monitoring several key system temperatures and also the sum of the differences between the successive scans of the recorded variables. When steady-state condition was observed for more than 20 scans, these scans were stored and constituted a test.

The primary experimental parameters were overall composition, mass flow rate, heat flux, and quality. The overall compositions for R22/R114 and R12/R152a mixtures were 0, 23, 48, 77, and 100 mole % R22 and 0, 21, 60 (azeotrope, R500), 89, and 100 mole % R12, respectively. Mass flow rates were within ±5% of the target values of 16, 23, 33, and 46 g s^{-1}, equivalent to 250–720 kg m^{-2} s^{-1} in terms of mass flux. Heat fluxes were 10, 17, 26, 36, and 45 kW m^{-2}. The quality ranged up to 95%.

2.4. *Property evaluation*

For the analysis of the data and comparison with the predictive methods, all thermodynamic properties for both pure and mixed refrigerants were calculated by the Carhahan–Starling–DeSantis (CSD) equation of state [9]. One of the characteristics of mixtures is that evaporation under constant pressure occurs in a range of temperatures, called a gliding temperature effect. During phase change, the liquid and vapor phase compositions vary continuously and are significantly different from each other except for an azeotrope.

To determine the thermodynamic properties at a certain location along the test section, a specific enthalpy calculated by an energy balance and a measured pressure were used for pure fluids. For mixtures, however, an overall composition was needed in addition to the enthalpy and pressure. With these data, the CSD equation of state calculated such equilibrium data as the vapor and liquid phase compositions and quality at the desired location.

Transport properties for pure refrigerants were taken from Johns [10] and Phillips and Murphy [11]. There are often large discrepancies in transport property values among different sources; special care was exercised to select the most accurate source of transport properties of pure refrigerants.

Transport properties of mixed refrigerants, however, are rarely found in the literature. Consequently, they are calculated by using the best available mixing rules suggested by Reid *et al.* [12] and Kandlikar *et al.* [13]. Care was also taken in evaluating

Table 1. Physical properties of pure refrigerants at a reduced pressure of 0.08

Property	Unit	R22	R114	R12	R152a
Molecular weight	$\mathrm{kg\ kmol^{-1}}$	86.47	170.92	120.91	66.05
Pressure	kPa	400	260	330	360
Saturated temperature	°C	−6.47	31.25	2.1	8.64
Liquid density	$\mathrm{kg\ m^{-3}}$	1304	1435	1389	938
Vapor density	$\mathrm{kg\ m^{-3}}$	17.1	19.3	19.2	11
Liquid specific heat	$\mathrm{J\ kg^{-1}\ {}^{\circ}C^{-1}}$	1155	1003	940	1644
Heat of evaporation	$\mathrm{J\ kg^{-1}}$	211 000	124 100	150 800	300 000
Liquid thermal conductivity	$\mathrm{W\ m^{-1}\ {}^{\circ}C^{-1}}$	0.101	0.0626	0.0748	0.112
Liquid viscosity	Pa s	2.32×10^{-4}	3.36×10^{-4}	2.58×10^{-4}	2×10^{-4}
Vapor viscosity	Pa s	1.17×10^{-5}	1.19×10^{-5}	1.19×10^{-5}	9.6×10^{-6}
Surface tension	$\mathrm{N\ m^{-1}}$	0.0128	0.01	0.0117	0.0116
Prandtl number		2.66	5.38	3.25	2.93

the transport properties of liquid and vapor phases in the two-phase region. They are calculated at their local compositions (not at the overall compositions). A detailed procedure for determining physical properties of refrigerants is given in the Appendix. Table 1 lists some of the properties for pure refrigerants at a reduced pressure of 0.08. For mixtures, it is felt inappropriate to list the properties since they vary with quality during evaporation under constant pressure.

2.5. *Data verification*

The validity of the pressure drop data was checked indirectly by measuring single phase pressure drops and comparing these results with existing correlations which have accurately correlated single phase pressure drops by other investigators. Five tests for pure refrigerants and eight for mixtures were performed with the subcooled liquid heated throughout the test section. The results were compared with the Blasius type correlation suggested by McAdams [14] which is valid for a smooth tube in the range of $5000 < Re < 200\,000$

$$\Delta P_{\mathrm{fo}} = \frac{2 f_{\mathrm{fo}} G^2 L}{D \rho_{\mathrm{l}}} \tag{1}$$

where

$$f_{\mathrm{fo}} = 0.046 Re^{-0.2}.$$

The correlation predicted the present results with a mean deviation of 8%, indirectly substantiating the pressure drop measurements.

3. RESULTS AND DISCUSSION

A series of tests were performed with R22/R114 and R12/R152a mixtures at several compositions and corresponding pure components to measure more than 600 pressure drop data. Only the data obtained at qualities more than 5% were considered in the analysis to assure that they were in the annular flow regime, which is based on Berenson and Stone's observation [15].

The effect of the unheated U-bend was investigated in detail in ref. [8]. In general, the results indicated that the flow was re-established within 0.3 m downstream from the inlet of the second test section. Thus, it is believed that the influence of the U-bend was minimal.

In order to compare the results on the same basis, the pressure at the outlet was kept at reduced pressures of 0.08–0.16. The reason for setting the test condition at the same reduced pressure is in part based upon the recent studies by Soumerai [16] that the thermodynamic corresponding states principle based on a reduced pressure is particularly useful in evaporator design optimization.

3.1. *Comparison with pressure drop models*

In a horizontal geometry, the two-phase pressure drop during flow boiling or condensation is composed of two components

$$\Delta P_{\mathrm{tp}} = \Delta P_{\mathrm{f}} + \Delta P_{\mathrm{a}} \tag{2}$$

where ΔP_{f} and ΔP_{a} are the frictional and accelerational pressure drops, respectively.

There have been two types of frictional pressure drop models in two-phase flow—homogeneous and separate flow models. The homogeneous model assumes that both liquid and vapor phases move at the same velocity. Consequently, it has also been called the zero slip model. It considers the two-phase flow as a single phase flow having average fluid properties depending on quality. Thus, the frictional pressure drop is calculated by assuming a constant friction coefficient between the inlet and outlet.

One of the best accepted pressure drop correlations, based on the homogeneous model, is Bo Pierre's semi-empirical correlation [4]. The correlation, equation (3), was developed based on the measured pressure drop data with R12 and R22 flowing inside 12 and 18 mm diameter tubes

$$\Delta p_{\mathrm{BP}} = \left[f_{\mathrm{av}} + \frac{(x_2 - x_1) D}{x_{\mathrm{av}} L} \right] \frac{G^2 L}{D \rho_{\mathrm{av}}} \tag{3}$$

where $x_{\mathrm{av}} = (x_1 + x_2)/2$ and f_{av}, ρ_{av}, and K_{f} are defined as

$$f_{\mathrm{av}} = 0.0185 K_{\mathrm{f}}^{0.25} Re^{-0.25} \tag{4}$$

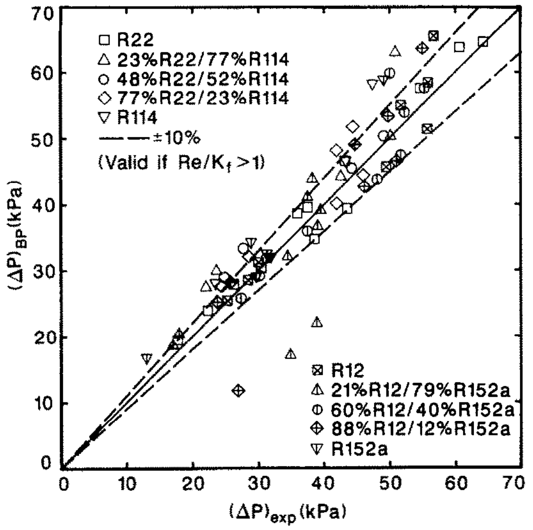

FIG. 3. Comparison between measured data and that calculated by Bo Pierre's correlation.

$$\rho_{av} = \frac{\rho_v \rho_l}{x_{av}\rho_l + (1 - x_{av})\rho_v} \tag{5}$$

$$K_f = \frac{J \Delta x h_{fg}}{L}. \tag{6}$$

In equation (6) K_f is not dimensionless but has units of N kg^{-1}, J is 1 in SI units with h_{fg} and L in J kg^{-1} and m, respectively. Equation (3) is valid only if $(Re/K_f) > 1$.

Figure 3 illustrates the comparison between the measured pressure drop in the second 4 m of the test section, ΔP_{48}, and calculated ones by Bo Pierre's correlation. Bo Pierre's correlation was found to correlate the data for both pure and mixed refrigerants with a mean deviation of 12%. It, however, had one serious drawback that more than half of the present data could not be correlated since the ratio, Re/K_f, was less than 1 for these data. This would happen when heat flux is high with a small mass flux and hence the quality change between inlet and outlet, Δx, is relatively large.

The fact that Bo Pierre's correlation is only applicable to half of the present data, obtained in typical ranges of heat and mass fluxes in heat pumps, indicates that Bo Pierre's correlation is not complete. Thus, care should be exercised in the use of Bo Pierre's correlation because of this limitation.

The separate model considers two phases to be divided into liquid and vapor streams and hence has been referred to as a slip flow model. It was originated from the classical work of Lockhart and Martinelli [17], which was followed by Martinelli and Nelson [7]. Their empirical correlations were shown to be as reliable as any annular flow pressure drop correlation [18].

One of the basic assumptions made by Martinelli and Nelson was that the static pressure drop of the liquid phase was the same as that of the vapor phase. Due to the nature of this assumption, their model would be better suited to annular flow. Many modifications have been made to the basic Martinelli and Nelson model. Their correlations with and without modifications were applied to the conditions much different from those assumed in the original work with fairly good success.

For the special case where a working fluid is evaporated from saturated liquid ($x = 0$) to a vapor–liquid mixture containing a vapor mass faction of x, with a linear change of x over a length L, the Martinelli and Nelson correlation can be simplified to (see Collier [18])

$$\Delta P_{MN} = \frac{2 f_{fo} G^2 L}{D \rho_l}\left[\frac{1}{x}\int_0^x \phi_{fo}^2 \, dx\right] + \frac{G^2}{\rho_l}\left[\frac{x^2}{\alpha}\left(\frac{\rho_l}{\rho_v}\right) + \frac{(1-x)^2}{(1-\alpha)} - 1\right] \tag{7}$$

where α is the void fraction, f_{fo} the friction coefficient given in equation (1), and ϕ_{fo} the frictional multiplier defined as

$$\Delta P_f = \Delta P_{fo}\phi_{fo}^2 = \left[\frac{2 f_{fo} G^2 L}{D \rho_l}\right]\phi_{fo}^2 \tag{8}$$

where ΔP_{fo} is the single phase pressure drop for the flow considered as all liquid as in equation (1).

Generally, equation (7) requires a stepwise integration. To present the pressure drop in a graphical form, the following key parameters were defined by Lockhart and Martinelli [17]

$$\phi_{lo}^2 = \frac{\Delta P_f}{\Delta P_{lo}} \tag{9}$$

$$\phi_{vo}^2 = \frac{\Delta P_f}{\Delta P_{vo}} \tag{10}$$

$$X^2 = \frac{\Delta P_{lo}}{\Delta P_{vo}} \tag{11}$$

where ΔP_{lo} and ΔP_{vo} are the pressure drops for either the liquid or vapor phase only considered to flow in the pipe.

By using the Blasius type relation given in equation (1) and assuming turbulent flow in both liquid and vapor phases, Martinelli's parameter, X_{tt}, was obtained from equation (11) as

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}. \tag{12}$$

Lockhart and Martinelli correlated ϕ_{lo}, ϕ_{vo}, and α as a function of X_{tt} and presented these results in a graphical form. Their correlation, however, was valid for the adiabatic flow of low pressure air–liquid mixtures.

For the prediction of pressure drop during forced convective boiling of water, Martinelli and Nelson [7] developed a correlation in terms of the frictional multiplier based upon the total flow in the liquid

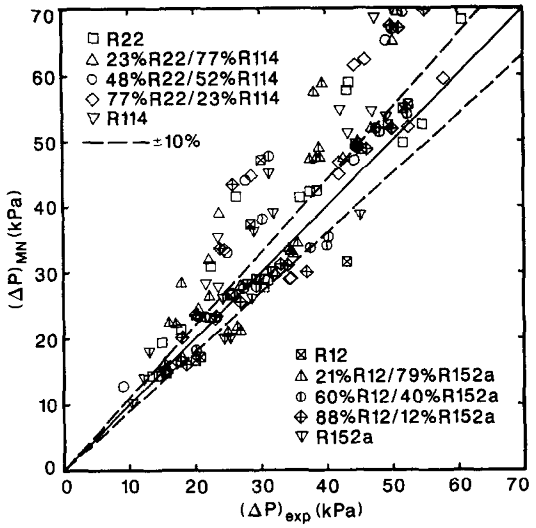

FIG. 4. Comparison between measured data and that calculated by Martinelli and Nelson's correlation.

phase, ϕ_{fo}. For flow boiling with heat addition, ϕ_{fo} would be more convenient than ϕ_{lo} or ϕ_{vo} which are based on single phase only flow in a pipe. By using equations (1) and (8), the following relationship between ϕ_{fo} and ϕ_{lo} is obtained:

$$\phi_{fo}^2 = \phi_{lo}^2(1-x)^{1.8}. \tag{13}$$

The term

$$\frac{1}{x}\int_0^x \phi_{fo}^2\,dx$$

in equation (7) was evaluated for the water–steam system and presented in a graphical form.

Figure 4 shows the comparison between the measured pressure drop in the second 4 m of the test section, ΔP_{48}, and estimated ones by Martinelli and Nelson's correlation. Even if it overpredicted the present data for both pure and mixed refrigerants by 20%, it was applicable to the entire heat and mass flux ranges employed in the present study.

3.2. *Correlation development*

Since Bo Pierre's correlation based on the homogeneous model seems to be inapplicable to the entire range of heat and mass fluxes considered in this study, the separate model developed by Martinelli *et al.* is used as a basis for correlation development. The separate model would be more physically appropriate because the present data were taken in the annular flow regime in which the basic assumption in the separate model is closely met. Collier [18] also states that the separate model would also provide more accurate pressure drop estimation in the low mass flux range ($G < 1300\ \mathrm{kg\,m^{-2}\,s^{-1}}$) in which the present data were taken.

Recently, Soumerai [16] showed that in thermodynamically efficient forced convection evaporators, the aspect ratio, L/D, would typically be in the range of 500–1000. For this range of aspect ratio, the ratio of ($\Delta P_a/\Delta P_f$) would be 0.025–0.05. Thus, the frictional pressure drop, ΔP_f, alone would constitute most of the two-phase pressure drop in evaporators

$$\Delta P_{tp} \approx \Delta P_f. \tag{14}$$

Since the aspect ratio of the present test section was 400–900, it would be interesting to check the validity of Soumerai's supposition, equation (14). From the measured data, the accelerational pressure drop, the second term in equation (7), was determined to be always less than 10% of the measured pressure drop, supporting the view of Soumerai [16]. Consequently, we feel that it is not necessary to divide the total pressure drop into the two components as in equation (2).

Furthermore, Wallis [19] has shown that the void fraction, α, can be well correlated by using only X_{tt} indicating that the accelerational pressure drop in equation (7) may be correlated by X_{tt}. Based upon this reasoning, the multiplier for total pressure drop (not just for frictional pressure drop), ϕ_{to} is correlated as a function of X_{tt} (note the difference in numerators of equations (9) and (15))

$$\phi_{to}^2 = \frac{\Delta P_{tp}}{\Delta P_{lo}}. \tag{15}$$

Figure 5 illustrates the two-phase pressure drop multiplier, ϕ_{to}, based upon the measured total two-phase pressure drop in each 1 m segment for both pure and mixed refrigerants. For the calculation of ϕ_{to}, the average values of quality, fluid temperature, and thermodynamic and transport properties at the average fluid temperature were used for each 1 m segment. For mixtures, both liquid and vapor phase compositions and equilibrium temperature vary as the quality is increased. The best available mixing rules (see Appendix) were used to calculate the transport properties of both phases at thermodynamic equilibrium compositions of both phases at the average fluid temperature.

It is remarkable for all data with both pure and mixed refrigerants to be well correlated by Martinelli's parameter, X_{tt}. This indicates that the method of Martinelli *et al.* for correlating pressure drop, favored by the nuclear power industry, is also relevant for correlating the pressure drop in evaporators of heat pumps. This is probably due to the coincidence between the flow regime (annular flow) of the present data and that of Martinelli *et al.*'s model.

Another point to be noted is that there is no significant mixture effect on the two-phase pressure drop. This indicates that a single correlation can be used for both pure and mixed refrigerants as long as the mixture properties are evaluated properly at local compositions. A regression analysis was carried out to get the functional dependence of ϕ_{to} on X_{tt} with a correlating coefficient of 0.985

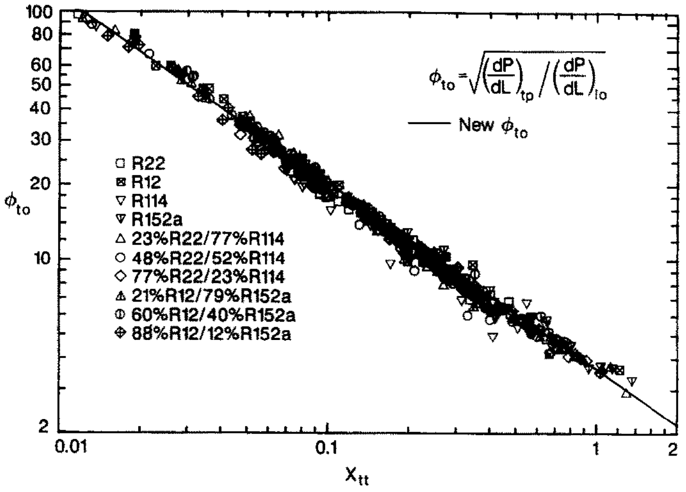

FIG. 5. Two-phase pressure drop multiplier, ϕ_{lo}, as a function of X_{tt} for pure and mixed refrigerants of R22, R114, R12, and R152a at reduced pressures of 0.08–0.16.

$$\phi_{to} = 3.58X_{tt}^{-0.735}. \tag{16}$$

As suggested by Martinelli and Nelson [7], the two-phase multiplier based on total liquid flow, ϕ_{tp}, is obtained for the flow boiling case

$$\phi_{tp}^2 = \phi_{to}^2(1-x)^{1.8} = 12.82X_{tt}^{-1.47}(1-x)^{1.8}. \tag{17}$$

Recently, Cooper [20] has shown that for pure fluids the property group in X_{tt} can be well correlated by using a reduced pressure, p_r. Following Cooper, the property group in X_{tt} is correlated for pure fluids (R22, R114, R12, R152a) as

$$\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} = 0.551p_r^{0.492}. \tag{18}$$

Equation (18) is applicable in the range of $0.06 < p_r < 0.7$ with a maximum deviation of 5%.

By using equation (18), X_{tt} of pure refrigerants can be rewritten as

$$X_{tt} = 0.551\left(\frac{1-x}{x}\right)^{0.9} p_r^{0.492}. \tag{19}$$

It has been shown that the Martinelli parameter, X_{tt}, of the mixtures of R22/R114 and R21/R152a was independent of overall composition [8]. Consequently, equation (19) could also be used for mixed refrigerants with the same accuracy. With equation (19) as X_{tt} of both pure and mixed refrigerants, the two-phase multiplier, equation (17), can be rewritten as

$$\phi_{tp}^2 = 30.78x^{1.323}(1-x)^{0.477}p_r^{-0.7232}. \tag{20}$$

Figure 6 shows ϕ_{tp}^2 as a function of quality for various reduced pressures. It should be noted that for $x = 0$ and 1, ϕ_{tp}^2 becomes 1 and $3.294p_r^{-0.984}$, respectively.

Finally, the two-phase pressure drop for pure and mixed refrigerants becomes

$$\Delta P_{tp} = \frac{2f_{fo}G^2L}{D\rho_l}\left[\frac{1}{x}\int_0^x \phi_{tp}^2\,dx\right]. \tag{21}$$

Equation (21) is valid only if the inlet vapor quality is 0%. Figure 7 illustrates the integral

$$\int_0^x \phi_{tp}^2\,dx$$

with a simplified ϕ_{tp} in equation (20) as a function of reduced pressure at various qualities.

Design engineers are often asked to estimate the pressure drop between the inlet vapor quality larger than zero, x_1, and the outlet quality, x_2. To be more general, equation (21) is modified to calculate the pressure drop between x_1 and x_2

$$\Delta P_{tp} = \frac{2f_{fo}G^2L}{D\rho_l}\left[\frac{1}{\Delta x}\int_{x_1}^{x_2} \phi_{tp}^2\,dx\right]. \tag{22}$$

Equation (22) is evaluated by subtracting the integral between 0% to the inlet vapor quality, x_1, from the integral between 0% and the outlet vapor quality, x_2.

Figure 8 illustrates the comparison between the measured pressure drop in the second 4 m of the test section, ΔP_{48}, and calculated ones by the present method for pure and mixed refrigerants. The mean deviation for both pure and mixed refrigerants was 8.4%. The advantage of the present correlation is that it requires the least number of physical properties (only liquid density and viscosity). With these values, the physical dimensions of the heat exchanger, the range of quality, mass flux, and the average operating pressure, the pressure drop during flow boiling of both pure and mixed refrigerants could be calculated simply by using equation (22) with the aid of the chart provided in Fig. 8.

4. CONCLUSIONS

The following conclusions can be drawn based upon the present experimental study of two-phase

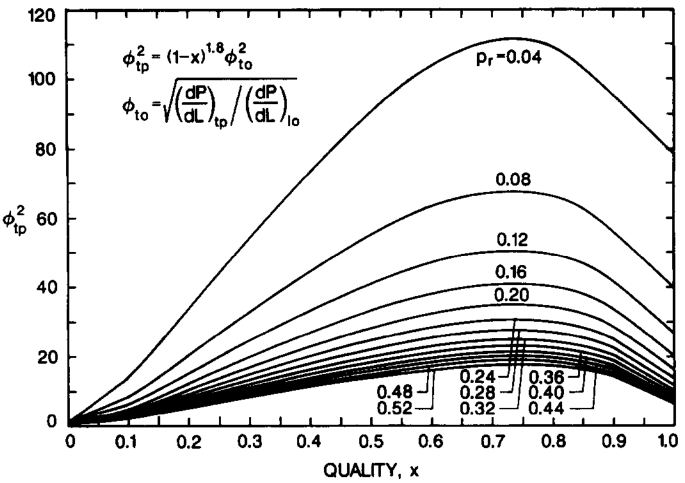

FIG. 6. Two-phase pressure drop multiplier, ϕ_{tp}, as a function of quality for various reduced pressures.

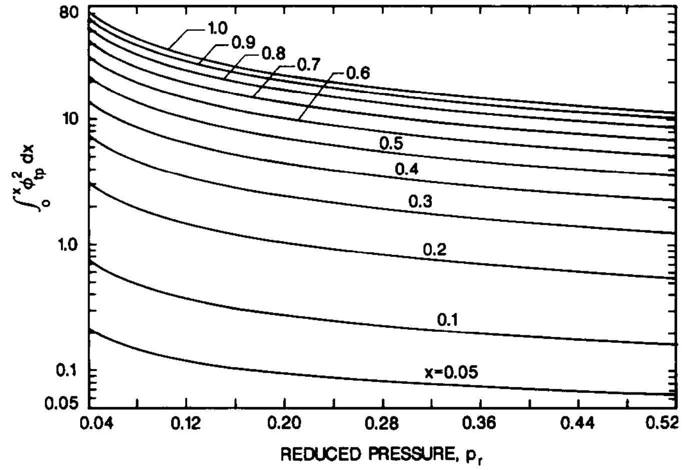

FIG. 7. Integral $\int_0^x \phi_{tp}^2\,dx$ as a function of reduced pressure at various qualities.

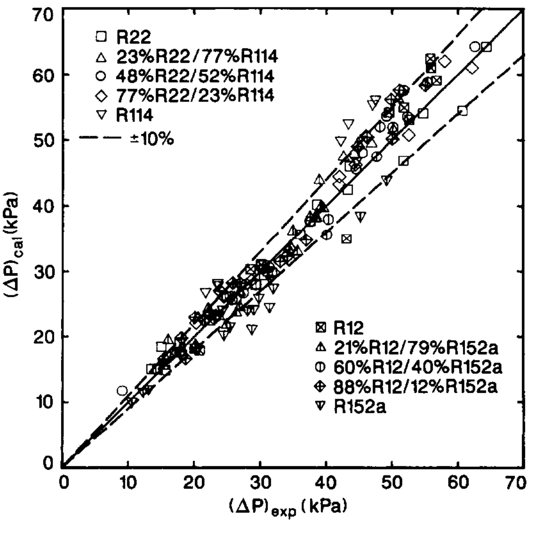

FIG. 8. Comparison between measured and predicted pressure drops.

pressure drop during evaporation of pure and mixed refrigerants.

(1) Bo Pierre's correlation based on the homogeneous model failed to correlate half of the present data. Care should be taken in its use especially when heat flux is high with a low mass flow rate since the parameter, Re/K_f, becomes less than 1. When $Re/K_f > 1$, it, however, predicted the present data well with a mean deviation of 12%.

(2) Martinelli and Nelson's correlation overpredicted the present data by 20%. It, however, was applicable in the entire heat and mass flux ranges typical in heat pumps. This proved that the separate model, popular in the power plant industry, would be also relevant for the prediction of pressure drop in evaporators in refrigeration and heat pump systems. The success of the separate model comes from the fact that the basic assumptions in the model are closely met by the flow patterns observed in the major portion of the evaporators.

(3) The present results indicate no composition

dependence of pressure drop with mixed refrigerants. When pressure drop data are nondimensionalized by introducing proper dimensionless variables with the properties evaluated at local compositions, pressure drops with both pure and mixed fluids are equally well correlated by a single parameter, X_{tt}.

(4) A new correlation is developed by modifying Martinelli and Nelson's correlation. The thermodynamic corresponding states principle is applied to correlate the property group commonly encountered in two-phase flow by using a reduced pressure. A chart is provided to facilitate the estimation of pressure drop. The correlation predicts the present data for both pure and mixed refrigerants with a mean deviation of 8.4%.

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REFERENCES

1. M. Kauffeld, W. Mulroy, M. McLinden and D. Didion, An experimental evaluation of two nonazeotropic refrigerant mixtures in a water-to-water, breadboard heat pump, NBSIR, NBS, Gaithersburg, MD 20899 (1989), to be published.
2. M. Altman, R. H. Norries and F. W. Staub, Local and average heat transfer and pressure drop of refrigerants, *J. Heat Transfer* **82C**, 189–198 (1960).
3. S. W. Anderson, D. G. Rich and D. F. Geary, Evaporation of Refrigerant 22 in a horizontal 3/4-in. OD tube, *ASHRAE Trans.* 28–42 (1961).
4. Bo Pierre, Flow resistance with boiling refrigerants, *ASHRAE J.* 58–77 (1964).
5. J. B. Chaddock and J. A. Noerager, Evaporation of refrigerant 12 in a horizontal tube with constant wall heat flux, *ASHRAE Trans.* 90–103 (1966).
6. L. C. Singal, C. P. Sharma and H. K. Varma, Pressure drop during forced convection boiling of binary refrigerant mixtures, *Int. J. Multiphase Flow* **9**, 309–323 (1983).
7. R. C. Martinelli and D. B. Nelson, Prediction of pressure drop during forced-circulation boiling of water, *Trans. ASME* **70**, 695–702 (1948).
8. D. S. Jung, Mixture effects on horizontal convective boiling heat transfer, Ph.D. thesis, Dept. of Mech. Engng, Univ. of Maryland, College Park, Maryland, May (1988).
9. G. Morrison and M. McLinden, Application of a hard sphere equation of state to refrigerants and refrigerant mixtures, NBS Technical Note 1226, NBS, Gaithersburg, Maryland (1986).
10. A. Johns, Private communication, National Engineering Laboratory, East Kilbride, Scotland.
11. T. W. Phillips and K. P. Murphy, Liquid viscosity of halogenated refrigerants. Paper presented at the ASHRAE Annual Meeting in Kansas City (1970).
12. R. C. Reid, J. M. Prausnitz and B. E. Poling, *The Properties of Gases and Liquids* (4th Edn). McGraw-Hill, New York (1987).
13. S. G. Kandlikar, C. A. Bijlani and S. P. Sukhatme, Predicting the properties of mixtures of R22 and R12—II. Transport properties, *ASHRAE Trans.* No. 2343 (1975).
14. McAdams, *Heat Transmission* (3rd Edn). McGraw-Hill, New York (1954).
15. P. J. Berenson and R. A. Stone, A photographic study of the mechanism of forced convection vaporization, *Chem. Engng Prog. Symp. Ser.* No. 57, **61**, 213–219 (1965).
16. H. Soumerai, *Practical Thermodynamic Tools for Heat Exchanger Design Engineers*. Wiley, New York (1987).
17. R. W. Lockhart and R. C. Martinelli, Proposed correlation of data for isothermal two-phase, two-component flow in pipes, *Chem. Engng Prog.* **45**, 39–48 (1949).
18. J. G. Collier, *Convective Boiling and Condensation* (2nd Edn). McGraw-Hill, New York (1981).
19. G. B. Wallis, *One Dimensional Two-phase Flow*. McGraw-Hill, New York (1969).
20. M. Cooper, Private communication, Oxford University.
21. *ASHRAE Handbook of Fundamentals*. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, Georgia (1981, 1985).
22. E. G. Porichanskii, O. P. Ponomareva and P. I. Svetlichnyi, Study of the isobaric heat capacity of Freon-152a in a wide range of parameters of state, *Izv. Vyssh. Uchebn. Zaved., Energy* **3**, 122–125 (1982).
23. G. Morrison, Private discussion, Thermophysics Division, Center for Chemical Engineering, NBS, Gaithersburg, Maryland.
24. J. Yata, T. Minamiyama and S. Tanaka, Measurement of thermal conductivity of liquid fluorocarbons, *Int. J. Thermophys.* **5**(2), (1984).
25. L. P. Filippov, *Vest. Mosk. Univ., Ser. Fiz. Mat. Estestv. Nauk.* **8**, 67–69 (1955).
26. C. R. Wilke, A viscosity equation for gas mixtures, *J. Chem. Phys.* **18**, 517–519 (1950).
27. K. Watanabe and M. Okada, Measurements of the surface tension of four halogenated hydrocarbons, CCl_3F, CCl_2F_2, $C_2Cl_3F_3$, and $C_2Cl_2F_4$, *Int. J. Thermophys.* **2**(2), (1981).

APPENDIX. DETERMINATION OF PHYSICAL PROPERTIES OF REFRIGERANTS

This appendix is concerned with the determination of physical properties of both pure and mixed refrigerants. It cannot be overemphasized that physical property values should be as accurate as possible to develop an accurate correlation and compare the results. Even though the ASHRAE handbook [21] provides most of the pure refrigerant data, it was discovered that the transport property data for the fluids considered in this study from ASHRAE is quite often unreliable.

In this study, all thermodynamic properties such as density, composition, enthalpy, and saturation temperature except liquid specific heat are evaluated from the Carhahan–Starling–DeSantis (CSD) equation of state [9]. Since ref. [9] gives the details of the formulation and accuracy of the equation of state, the interested reader is referred to this report for further information.

Also of particular concern are liquid specific heat and transport properties of pure refrigerants and proper mixing rules for mixtures. The unit, source, accuracy if mentioned, and a proper mixing rule will be given for each property.

(1) Specific heat of liquid, C_{pl} (kJ kg^{-1} °C^{-1})

Data from the National Engineering Laboratory of Scotland (NEL) [10], ASHRAE handbook (1981) [21] and Porichanskii *et al.* [22] (for R152a only) are compared.

Table A1. Comparison of specific heat of liquid at 260 K

Fluid	NEL	ASHRAE	Porichanskii *et al.*
R22	1.14	1.54	
R152a	1.572	1.325	1.66

When data for R22 and R152a from ASHRAE were used in the energy balance analysis, the results were not satisfactory. Both NEL and Porichanskii *et al.*'s data, however, yielded good results. Table A1 lists C_{pl} values from different sources at 260 K. As seen in Table A1, ASHRAE data deviates from other sources by 20–30%. Thus, data from NEL is used for the analysis. No measurements were found for the liquid specific heat of refrigerant mixtures, C_{plm}. A linear mass fraction weighting method is used as suggested by Morrison [23]

$$C_{plm} = X_{M1}C_{pl1} + X_{M2}C_{pl2}. \tag{A1}$$

(2) Liquid thermal conductivity, k_l (mW m^{-1} K^{-1})

Data from NEL, ASHRAE, and Yata *et al.* [24] agreed within 5%. No data was found for the liquid thermal conductivity of refrigerant mixtures. The most widely accepted correlation by Filippov [25] was adopted as suggested by Reid *et al.* [12]

$$k_{lm} = X_{M1}k_{l1} + X_{M2}k_{l2} - 0.72X_{M1}X_{M2}(k_{l2} - k_{l1}). \tag{A2}$$

It should be noted that the components were so chosen that $k_{l2} \geqslant k_{l1}$. The accuracy of equation (A2) is within 5%.

(3) Liquid viscosity, μ_l (μPa s)

Data from NEL, ASHRAE, and Phillips and Murphy [11] were compared. Data from ASHRAE deviates from the others by 10%. For instance, at 260 K, ASHRAE data for R22 is 225 μPa s while the data of NEL and Phillips and Murphy are 252 and 255 μPa s, respectively. Phillips and Murphy [11] measured the liquid viscosity of R500, which is an azeotrope, 60%R12/40%R152a by mole. A variety of correlations were compared against the measured data and the method proposed by Kandlikar *et al.* [13] best fitted the data within 4% deviation

$$\ln(\mu_{lm}) = X_1 \ln(\mu_{l1}) + X_2 \ln(\mu_{l2}). \tag{A3}$$

Actually, Kandlikar *et al.* [13] used a mass fraction in equation (A3). It, however, was found that a mole fraction better fitted R500 data. Thus, a mole fraction was used throughout the study.

(4) Vapor viscosity, μ_v (μPa s)

No measurements were found even for pure refrigerants and estimated data from NEL was used. Wilke's [26] correlation was used for mixtures

$$\mu = \frac{Y_1\mu_{v1}}{Y_1 + Y_2P_{12}} + \frac{Y_2\mu_{v2}}{Y_2 + Y_1P_{21}} \tag{A4}$$

where P_{ij} is defined as

$$P_{ij} = \frac{[1 + (\mu_{vi}/\mu_{vj})^{0.5}(M_iM_j)^{0.25}]^2}{[8(1 + M_i/M_j)]^{0.5}}. \tag{A5}$$

(5) Surface tension, σ (mN m^{-1})

Data from NEL and Watanabe and Okada [27] was used. They agreed within 5%. No measured data was available for refrigerant mixtures. A simple linear mole fraction weighting method is used. Although there are other complicated correlations, this method has been proven to be as good as the others

$$\sigma_m = X_1\sigma_1 + X_2\sigma_2. \tag{A6}$$

CALCUL DE LA CHUTE DE PRESSION PENDANT L'EBULLITION DE REFRIGERANTS PURS OU MELANGES, DANS UN ECOULEMENT A L'INTERIEUR D'UN ESPACE ANNULAIRE HORIZONTAL

Résumé—On décrit une étude expérimentale de la chute de pression pendant l'ébullition en écoulement dans un espace annulaire horizontal, pour des réfrigérants purs ou mélangés de R22, R114, R12 et R152a. Plus de 600 pressions ont été considérées pour des pressions réduites entre 0,08 et 0,16 (200–800 kPa). Le domaine de flux de chaleur est 10–45 kW m^{-2} et celui du débit est 16–46 g s^{-1} (correspondant à 230–720 kg m^{-2} s^{-1}). Les résultats sont comparés à des formules bien connues; celle de Bo Pierre ne s'accorde qu'avec la moitié des données tandis que celle de Martinelli et Nelson les surestime de 20%. Les chutes de pression pour les fluides purs ou les mélanges sont bien unifiées par le paramètre X_{tt} de Martinelli. On ne constate pas de dépendance de la chute de pression vis-à-vis de la composition des mélanges. Une formule simple adoptant le principe des états correspondants en thermodynamique est développée afin de faciliter l'estimation de la chute de pression pendant l'ébullition en écoulement. Les prédictions s'accordent aux données des mesures avec une déviation moyenne de 8,4%.

BERECHNUNG DES DRUCKVERLUSTES BEI RINGSTRÖMUNG SIEDENDER REINER KÄLTEMITTEL UND KÄLTEMITTELGEMISCHE IM HORIZONTALEN ROHR

Zusammenfassung—Es wird über experimentelle Untersuchungen zum Druckverlust bei horizontaler Ringströmung reiner Kältemittel und Kältemittelgemische aus R22, R114, R12 und R152a berichtet. Es werden mehr als 600 Meßwerte zum Druckverlust bei Ringströmung für konstante Wärmestromdichte bei einem reduzierten Druck von 0,08–0,16 (200–800 kPa) erhalten. Der Bereich der Wärmestromdichte ist 10–45 kW m^{-2} und des Massenstroms 16–46 g s^{-1} (entsprechend einer Massenstromdichte von 230–720 kg m^{-2} s^{-1}). Die Ergebnisse werden mit bekannten Korrelationen verglichen. Bo Pierre's Korrelation versagt bei der Hälfte der vorhandenen Daten. Martinelli und Nelson's Korrelation berechnet die Werte 20% zu groß. Der Druckverlust bei reinen Kältemitteln und bei Kältemittelgemischen läßt sich mit dem Martinelli-Parameter X_{tt} gut korrelieren. Es wird weiterhin kein Einfluß der Zusammensetzung auf den Druckverlust bei Gemischen gefunden. Es wird eine einfache Korrelation mit einem Diagramm unter Anwendung des Prinzips der korrespondierenden Zustände entwickelt, um die Bestimmung des Druckverlustes beim Strömungssieden zu erleichtern. Die Berechnungen zeigen eine mittlere Abweichung von 8,4% von den Meßwerten.

ОПРЕДЕЛЕНИЕ ПЕРЕПАДА ДАВЛЕНИЯ В ПРОЦЕССЕ КИПЕНИЯ ПРИ ГОРИЗОНТАЛЬНОМ КОЛЬЦЕВОМ ТЕЧЕНИИ ЧИСТЫХ И СМЕШАННЫХ ХЛАДАГЕНТОВ

Аннотация—Экспериментально определяется перепад давления в процессе кипения при горизонтальном течении чистых и смешанных хладагентов R22, R114, R12 и R152a. Получено более 600 данных для случая кольцевого течения в условиях постоянного теплового потока при изменении значений приведенного давления от 0,08 до 0,16 (от 200 до 800 кПа). Тепловой поток и массовый расход жидкости изменяются в диапазонах 10–45 кВ $м^{-2}$ и 16–46 г $с^{-1}$ (соответствующих 230–720 кг $м^{-2}$ $с^{-1}$ в обозначениях массового потока). Результаты сравниваются с известными соотношениями; соотношение Бо Пера не позволяет скоррелировать половину имеющихся данных, в то время как по соотношению Мартинелли и Нельсона получают их значения, завышенные на 20%. Однако перепады давления как для чистых, так и для смешанных хладагентов хорошо коррелируются при помощи параметра Мартинелли X_{tt}. Найдено также, что перепад давления не зависит от состава смесей. На основе термодинамического закона соответственных состояний разработаны простые корреляции, а также диаграмма, позволяющие оценить перепад давления в процессе кипения при течении. Среднее отклонение расчета от измеренных данных составляет 8,4%.