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Two-phase pressure drop of refrigerants during flow boiling in small channels: an experimental investigation and correlation development ☆

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Abstract

Two-phase flow pressure drop measurements were made during a phase-change heat transfer process with three refrigerants (R-134a, R-12, and R-113) at six different pressures ranging from 138 to 856 kPa, and in two sizes of round tubes (2.46 and 2.92 mm inside diameters) and one rectangular channel (4.06 $\times$ 1.7 mm). The data were compared with those from large tubes under similar conditions, and state-of-the-art correlations were evaluated using the R-134a data. The state-of-the-art large-tube correlations failed to satisfactorily predict the experimental data. The data were used to develop a new correlation for two-phase pressure drop during flow boiling in small channels. The correlation was then tested against the experimental data for the three refrigerants; the error was $\pm 20\%$.

Keywords: Two-phase pressure drop; Small channels; Refrigerants; Correlations

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1. Introduction

Only a few studies in the literature report on two-phase fluid flow and heat transfer in compact heat exchangers. Nevertheless, extensive applications exist in the process industries, where phase-change heat transfer allows more compact heat exchanger designs with better performance than those used for single-phase operation. To further the application of compact heat exchangers in the process industries, there is need to understand the fundamental issues of two-phase flow and heat transfer in small channels representative of compact heat exchanger flow passages.

In the refrigeration/air conditioning industry, including automobile environment control, a fundamental understanding of multiphase-flow and heat transfer involving boiling and condensing of refrigerants in small channels is important. In particular, there is need for a validated design correlation for two-phase pressure drop that will facilitate the design and optimization of compact heat exchangers for use with refrigerants.

Jung and Radermacher (1989) conducted experiments with refrigerants R-22, R-114, R-12, and R-152, and refrigerant mixtures in 4-m long, 9.1-mm inside diameter (ID), stainless steel tubes. They proposed a correlation for a two-phase flow total pressure drop of pure and mixed refrigerants. In their correlation, the two-phase multiplier is expressed as a function of quality and reduced pressure.

Souza and Pimenta (1995) tested refrigerants R-134a, R-12, R-22, MP-39, and R-32/125 in a copper tube with two sections; one had an ID of 7.75 mm, and the other of 10.92 mm. They proposed a semi-empirical correlation to calculate the two-phase flow multiplier for pure and mixed refrigerants.

Two-phase flow pressure drop and evaporation heat transfer of refrigerants in small channels have been studied at Argonne National Laboratory. In the Argonne study, two smooth circular tubes and one rectangular channel with hydraulic diameters 2.46, 2.92, and 2.40 mm, respectively, were used to simulate flow passages typical of plate-fin heat exchangers. Heat transfer results have been reported (Wambsganss et al., 1993; Tran et al., 1996), and a correlation for nucleate flow boiling has been developed (Tran et al., 1997). Further objectives of the study were to determine if large-tube correlations can be used to predict two-phase pressure drop of refrigerants in small channels (hydraulic diameter < 3 mm), and, as necessary, to develop an experimentally-validated predictive method that can facilitate the design and optimization of compact heat exchangers for refrigerants, including those that are environmentally acceptable. The two-phase pressure drop data from the Argonne study and the results of correlation evaluation/development are presented in this paper.

2. Test loop description

The test apparatus is shown schematically in Fig. 1. It is a closed-loop system including a gear pump with variable-speed drive, a set of rotameters, a preheater, a condenser, an accumulator, and sight-glasses. The bladder-type accumulator enables stable control of the system pressure. The rotameters were sized to cover the range of flow rates expected in testing small channel flows, and the flow rate was calibrated by a weighing-with-stop-watch technique.

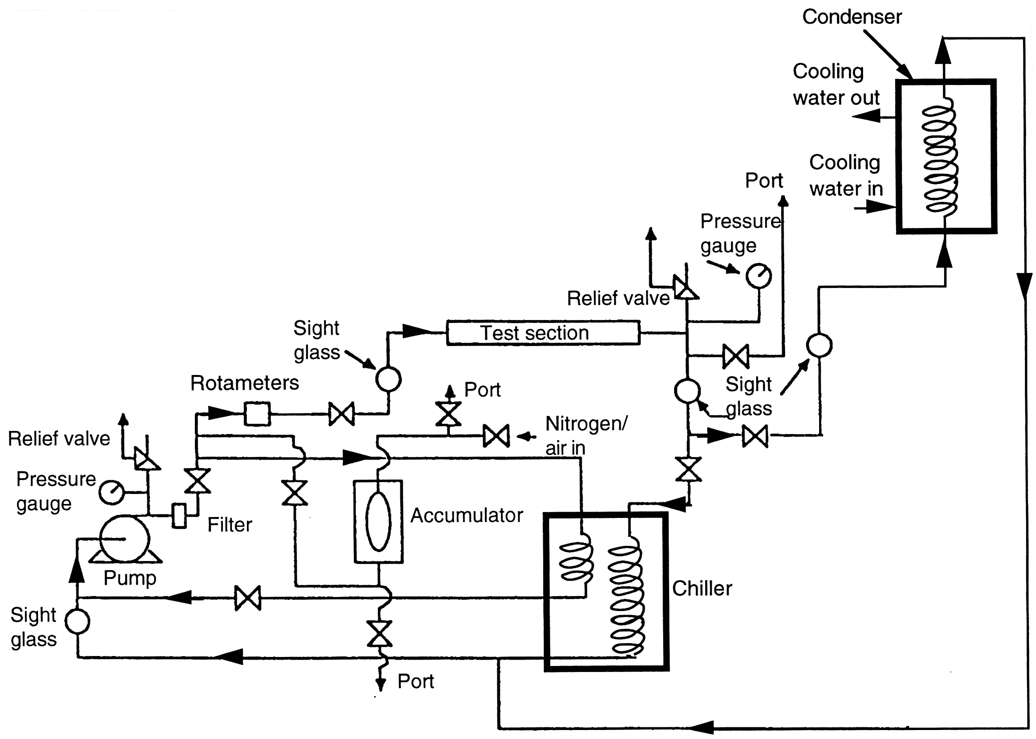

Fig. 1. Schematic diagram of test apparatus.

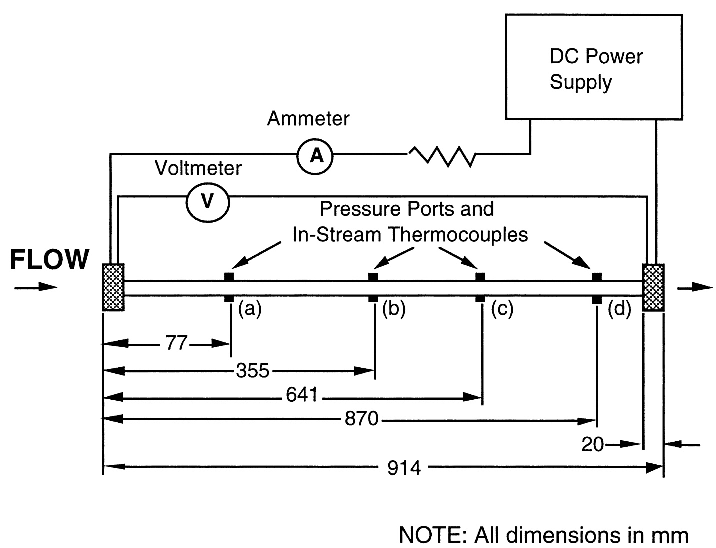

Fig. 2. Test channel showing thermocouple and pressure port locations for R-134a and R-12 tests.

Provisions were made to measure temperature and pressure at various locations, as indicated in Fig. 2. Inlet pressure (p_{in}) was measured with a piezoresistive-type transducer (Endevco Model 8510B-50). Differential pressure across the channel (Δp) was measured with a differential pressure transducer (Validyne Model DP-15-36). The pressure transducers were calibrated to standards traceable to the National Institute of Standards and Technology (NIST). The inlet pressure was measured with an accuracy of ± 2 kPa, and the test section pressure drop was measured with an accuracy of ± 0.7 kPa. The pressure transducers were connected to four pressure ports, as shown in Fig. 2. A typical pressure port consists of a 0.81-mm ID by 25-mm long brass tube brazed onto the test section; a 0.81-mm diameter hole was drilled into the test section.

In-stream temperatures were measured with thermocouples (0.15 mm diameter bead) inserted into the test channels. Dependent on the test channel, either thermocouples or RTDs were used to measure wall temperatures. All thermocouples and RTDs were calibrated, using a temperature-controlled calibration bath and a NIST-traceable, ASTM precision thermometer (Brooklyn Thermometer Company). Temperature measurements were estimated to be accurate to ± 0.2°C.

Temperature and pressure measurements were recorded with a Hewlett-Packard (HP) data acquisition system (DAS) consisting of an HP Vectra micro-computer and a Model 3421 multiplexor. The RTD and thermocouple calibration was done "end-to-end" to include any error that might be associated with the multiplexor or DAS.

For tests with R-134a and R-12, the test channel was a plain circular brass tube with 3.18 mm outside diameter and 2.46 mm inside diameter. The length of the test channel was 914 mm. To minimize heat loss to the surroundings, the test channel was insulated with 4-in. thick fiberglass. The test channel was resistance-heated by passing DC electricity through the tube wall using a high-current-regulated DC power supply. Heat input to the fluid was determined from the electric power input. The test section for R-113 was arranged similar to the ones for R-134a and R-12 except that it was a stainless steel (Type 304) tube, with a total length of 412 mm.

Flow rate was measured with two rotameters for different flow rate ranges. The rotameters were calibrated to an accuracy of better than ± 2% of the reading. Heat was rejected to either the system condenser, using water at 15°C, or to the chiller.

3. Test procedure and single-phase results

Single-phase tests were performed to validate the instrumentation and data reduction method (Tran, 1998). The Fanning friction factor from the single phase test result of R-134a was close to and slightly lower than Blasius' prediction at Reynolds number greater than 3000 in fully developed turbulent flow regime. This result is in agreement with the findings of Olsson (1994) for single phase test in a small circular tube of 2 mm diameter and 300 mm length.

For all two-phase tests, the flow entered the test section in a subcooled condition, and the starting point of saturated boiling ranged from 0.2 to 0.5 mm downstream of the first pressure port location. Since the pressure drop over the subcooled length is negligibly small, the pressure drop measured by the differential pressure transducer was that of the saturated flow boiling section. The total two-phase flow pressure drop consists of two components: an acceleration component and a frictional component. The accelerational pressure drop component results from acceleration of the flow during the evaporation process. This component was calculated, using Zivi's correlation for void fraction (Zivi, 1964), and subtracted from the measured total pressure drop to obtain the frictional component.

Flow boiling heat transfer and two-phase flow pressure drop tests were performed with R-134a at four different averaged saturation pressures 356, 441, 634, and 835 kPa; with R-12 at the two averaged saturation pressures 517 and 824 kPa; and with R-113 at an averaged saturation pressure 169 kPa. The three different refrigerants were tested in four different channels as summarized in Table 1. The exit quality, which was determined from an energy balance over the total heated length, varied from test to test. The experimental pressure drop data were plotted against exit quality for the averaged pressure of the particular test series and various values of mass flux.

The accuracy of the database was assessed by an uncertainty analysis based on the method of sequential perturbations (Moffat, 1988; Holman and Gajda, 1978). This method allows estimation of an overall uncertainty of a set of data by integrating the uncertainty of each source of error into the data base independently, then using a root sum square method to calculate the overall uncertainty. The uncertainty analysis was incorporated into the spreadsheet program, and an uncertainty was calculated for every test run of each data set. The uncertainty in the measurement of the pressure drop was found to be $\pm 10\%$.

Table 1
Summary of test scopes for two-phase flow pressure drop test

Parameter	Test series			
	1[a]	2[b]	3[b]	4[c]
Refrigerant	R-113	R-12	R-12	R-134a
Channel material	Stainless steel	Brass	Brass	Brass
Channel geometry	Circular	Circular	Rectangular	Circular
Hydraulic diameter, D_h (mm)	2.92	2.46	2.40 (1.70 × 4.06)	2.46
Total length (mm)	412	914	881	914
Reduced pressure, P_r	0.04–0.058	0.12–0.21	0.18–0.23	0.08–0.21
Mass flux G (kg/m^2 s)	50–400	63–832	44–505	33–502
Heat flux q'' (kW/m^2)	8.8–90.8	7.5–59.5	7.7–129	2.2–49.8
Exit quality	0.02–0.95	0.21–0.94	0.22–0.82	0.24–0.90
No. of data points	39	137	132	302

[a] Wambsganss et al. (1993).
[b] Tran et al. (1996).
[c] Tran (1998).

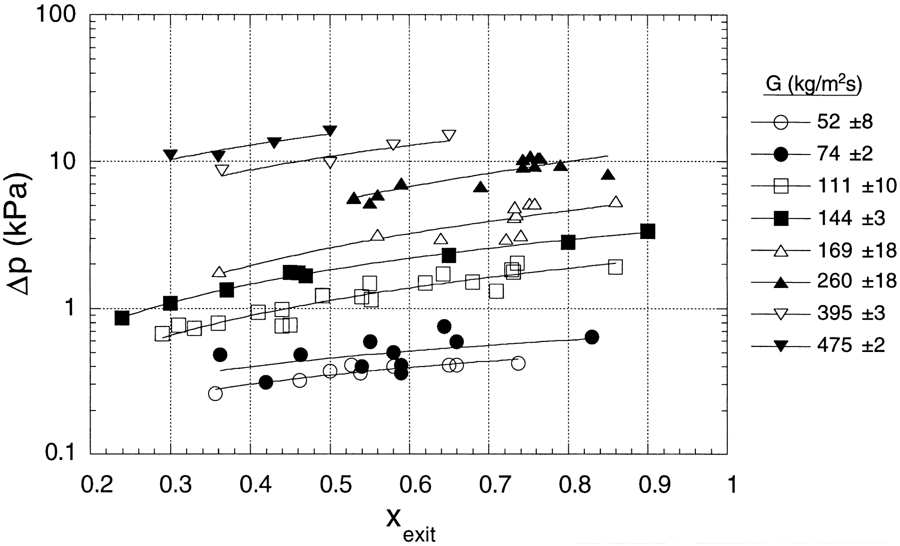

Fig. 3. Pressure drop of R-134a at $p_{sat} = 835$ kPa as a function of exit quality at various values of mass flux.

4. Two-phase pressure drop results

4.1. Refrigerant R-134a

In Fig. 3, the experimental pressure drop data for tests with R-134a are plotted against exit quality for various values of mass flux G and a saturation pressure of 835 kPa. The results show that the two-phase flow pressure drop increases with the increase in exit quality and mass flux; these trends agree with those of a large tube.

The pressure effect on two-phase flow pressure drop was also investigated. A typical result is

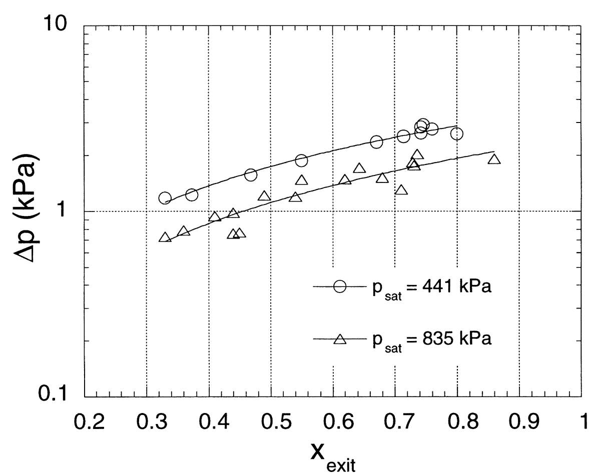

Fig. 4. Pressure effect on two-phase flow pressure drop of R-134a at $p_{sat} = 441$ and 835 kPa, $G = 106 \pm 14$ kg/m^2 s.

shown in Fig. 4, where two-phase flow pressure drop is higher at a lower saturation pressure. For R-134a, a decrease in saturation pressure of 47% causes an increase in two-phase flow pressure drop of about 60%. This pressure effect is in qualitative agreement with large-tube results, where the liquid-to-vapor density ratio is a key parameter.

In an attempt to compare the pressure drop data of R-134a in the present study to that of a large tube, the present data were compared to the test results that Eckels et al. (1994) obtained for R-134a in a large smooth tube with an ID of 8 mm and a test section length of 3.6 m. The Eckels et al. tube diameter was approximately three times that of the tube used in the present study. Due to the difference in the test section length, the comparison is based on pressure gradient instead of pressure drop. The comparison shows that at a saturation pressure of 350 kPa, mass flux of 307 ± 50 kg/m^2 s, and a quality range of 0.68 ± 0.07 (the closest available quality range compared to the 0.82 ± 0.05 in Eckels et al., 1994), the pressure gradient in the present small-channel study is about 3.3 times higher than that of the large-tube data. It is noteworthy that in their pressure drop tests, the inlet quality varied from 7 to 11%, while in the present study, the inlet quality was $<1\%$.

4.2. Refrigerant R-12

The experimental two-phase pressure drop data for R-12 at a saturation pressure of 824 kPa are shown in Fig. 5 as a function of quality for various values of mass flux. Trends are similar to those of R-134a shown in Fig. 3. In Fig. 6, pressure drop data were plotted against quality for saturation pressures of 517 and 824 kPa, and the same mass flux 259 ± 18 kg/m^2 s. Fig. 6 shows that the two-phase flow pressure drop is higher at a lower saturation pressure. This result agrees with that of R-134a (see Fig. 4). Fig. 6 also shows that for R-12 tested in the present study, a decrease in saturation pressure of 37% causes an increase in two-phase flow pressure drop of about 70%.

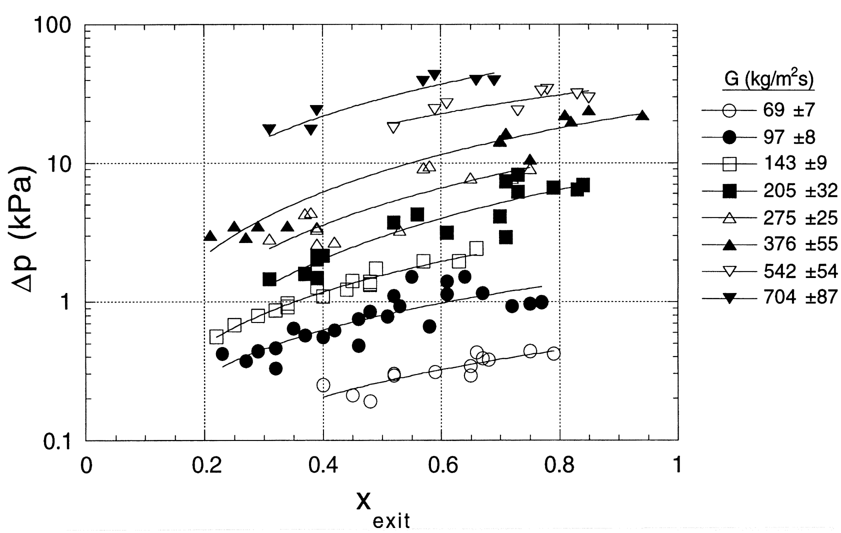

Fig. 5. Pressure drop of R-12 at $p_{sat} = 824$ kPa as a function of exit quality at various values of mass flux.

4.3. Comparison of R-134a and R-12

Refrigerant R-134a is a replacement for R-12 in the refrigeration and air conditioning industry. Therefore, it is of interest and useful to compare the performance of these two refrigerants in terms of pressure drop. Such information is useful to designers in the selection of the appropriate parameters to use with refrigerant R-134a in the design of a new system, or in retrofitting an existing R-12 refrigeration system.

Since both R-134a and R-12 were tested in the same test section under similar test conditions, a direct comparison of two-phase flow pressure drop can be made. Knowing that two-phase flow pressure drop is a function of saturation pressure, exit quality, and mass flux, the comparison can be made based on either the same mass flux or the same heating rate, as suggested by Eckels and Pate (1991). Comparison based on the same heating rate was suggested because R-134a has a higher latent heat of vaporization than R-12, and also because a design is often based on a specified heating or cooling rate.

We compared the pressure drop data of R-134a and R-12 in terms of mass flux for a saturation pressure of 835 kPa and a quality of 0.70 ± 0.05 (Fig. 7). The results show the same trend reported by Eckels and Pate (1991) for a large tube of 8 mm dia. and 3.67 m length, tested at a saturation pressure of ≈ 350–490 kPa (≈ 5–15°C saturation temperature). Fig. 7 shows that the two-phase flow pressure drop of R-134a is about 31% higher than that of R-12 for a range of mass flux from 60 to 300 kg/m^2 s and a quality of 0.70 ± 0.05. This result is comparable to the difference in pressure drop of above 50% reported by Eckels and Pate (1991) for a range of mass flux from 125 to 400 kg/m^2 s at a quality of 0.80–0.88. Again, the tube used by Eckels and Pate was more than three times larger in diameter, the quality was higher, and the saturation pressure was much lower, than those of the present study.

Fig. 8 presents the comparison of pressure drop in terms of heat flux, at the quality of 0.70 $\pm$ 0.05. This comparison indicates that for a given heating rate, the two-phase flow pressure drop of R-134a is 31–44% lower than that of R-12 for the range of heat flux from 5–31 kW/

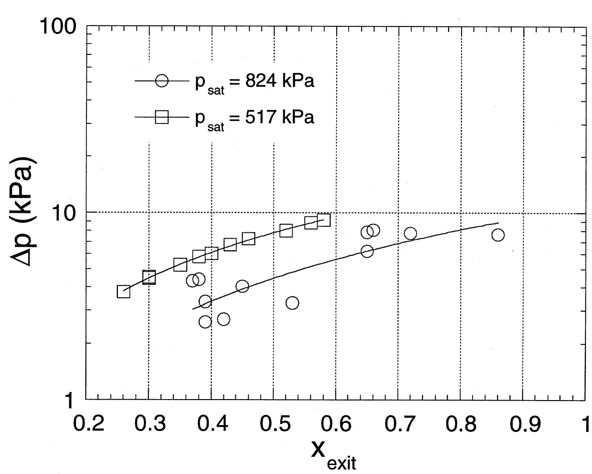

Fig. 6. Pressure effect on two-phase flow pressure drop of R-12 at $p_{sat} = 517$ and 824 kPa for $G = 259 \pm 18$ kg/m^2 s and $x_{exit} = 0.58 \pm 0.28$.

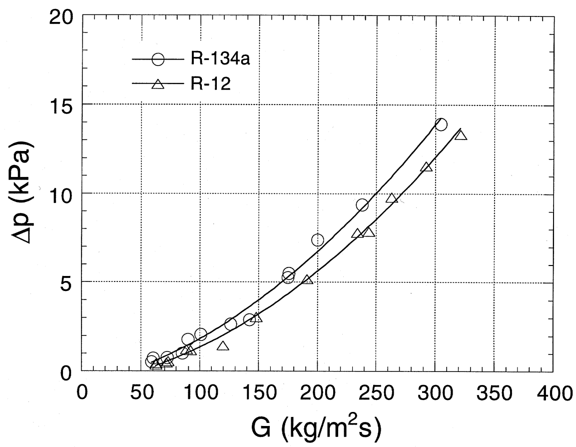

Fig. 7. Comparison of two-phase flow pressure drop between R-134a and R-12 in terms of mass flux for $x_{exit} = 0.7 \pm 0.05$ and $p_{sat} = 835$ kPa.

m^2. This result is comparable with that reported by Eckels and Pate (1991) for a large tube, where the pressure drop of R-134a was 10–30% lower than that of R-12 for the same heat capacity. Again, the difference from Eckels and Pate's result is due to the much smaller tube, the lower quality, and the higher saturation pressure of the present study. The lower pressure drop of R-134a compared to that of R-12 in Fig. 6 is seen because a lower mass flux of R-134a is required to achieve the same heating rate as R-12, due to the higher latent heat of vaporization of R-134a.

4.4. R-113 in round tube and R-12 in rectangular channel

To broaden the database for small-channel two-phase flow pressure drop, the pressure drop

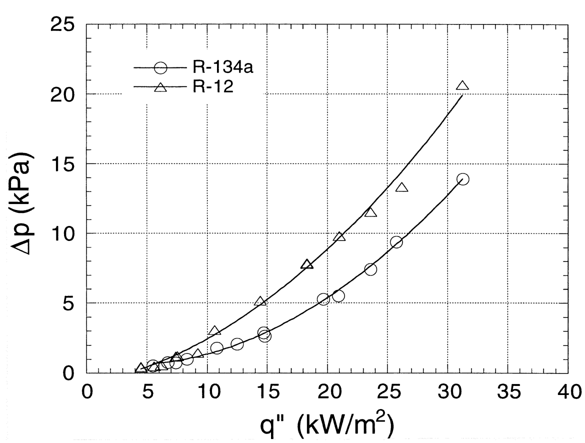

Fig. 8. Comparison of two-phase flow pressure drop between R-134a and R-12 in terms of heat flux for $x = 0.7 \pm 0.05$ and $p_{sat} = 835$ kPa.

results from flow boiling tests with R-113 (Wambsganss et al., 1993) were also included. Unlike R-134a and R-12, which have similar fluid physical properties, R-113 has a higher surface tension, higher liquid viscosity, higher liquid density, and lower gas density than those of R-12 and R-134a at the same saturation pressure or temperature. The two-phase flow pressure drop data of R-113 exhibit the same trends, with quality and mass flux, as in R-134a and R-12 (Tran, 1998).

The hydraulic diameter of the rectangular channel used with R-12, was approximately equal to the diameter of the circular tube. A comparison of the test data from R-12 in round tube, with the data from R-12 in rectangular channel, showed no significant difference that would indicate a geometry effect.

5. Comparison with large tube correlations

The experimental data for R-134a were compared with five state-of-the-art correlations for predicting two-phase frictional pressure drop in large tubes: (1) Friedel's correlation (Friedel, 1979), (2) the *B*-coefficient method (Chisholm, 1983), (3) the *C*-coefficient method (Chisholm, 1983), (4) Jung and Radermacher correlation (Jung and Radermacher, 1989), and (5) Souza and Pimenta correlation (Souza and Pimenta, 1995). These correlations were developed specifically for large tubes with an inside diameter of 8 mm or more. The correlations of Jung and Radermacher and Souza and Pimenta were developed from refrigerant data. Tran, (1998) summarizes the five correlations and presents detailed (i.e., function of quality) comparisons with the experimental data; a typical comparison plot is given in Fig. 9.

A careful review of the frictional two-phase flow pressure drop comparison between

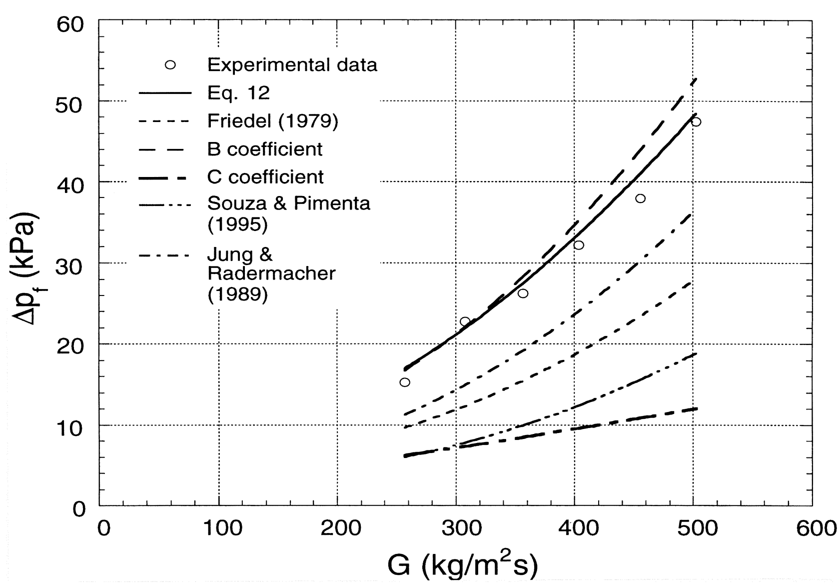

Fig. 9. Comparison of experimental data for R-134a at $p_{sat} = 365$ kPa, and $x = 0.73 \pm 0.02$ with five large-tube correlations.

measured data and the five large-tube correlations led to the conclusions below (Tran 1998). At low mass flux, lower qualities, and certain pressures, some of the correlations satisfactorily predict experimental results, while the others significantly underpredict (as four of the correlations do in Fig. 9) or slightly overpredict the results. For all saturation pressures, none of the large-tube correlations satisfactorily predict the measured frictional two-phase flow pressure drop for mass flux > 150 kg/m^2 s, and qualities larger than ≈ 0.6.

We suggest that the reason the large-tube correlations fail to predict small-channel response is related to the different flow behavior in small channels. The coalesced bubbles are confined, elongated, and slide over a thin liquid film as they flow downstream, compared to the flow boiling in a large tube, where the bubbles grow and flow along the tubes going through different flow regimes without restriction. Therefore, it is reasonable to suggest that the greater pressure drop in a small channel may be due to additional friction related to the deformed/elongated bubble movement. Also, for the case of large tubes, the particular experiments of Souza and Pimenta (1995) showed a flow regime change from annular to mist at a quality of ≈ 0.85, and the pressure drop reached a peak at this point. When the flow regime changes from annular to mist, the tube wall dries out; and the shear stress at the wall falls at a quality of ≈ 0.85, due to a large viscosity ratio of 20 between the liquid refrigerant and its vapor (Souza and Pimenta, 1995). Most of the large-tube correlations predict a much lower pressure drop at this point. However, the comparison in the present study shows that the decreased prediction of pressure drop using large-tube correlations starts at a lower quality of ≈ 0.6, and the decrease is clearly evident at a quality of 0.73 ± 0.02 for all four pressures. This indicates that the large-tube correlations cannot be used in predicting two-phase flow pressure drop in small channels for a quality > 0.6. The present comparison also suggests that the transition from annular flow to mist flow regime does not occur in small channels until the quality is > 0.85; in other words, the dryout point will be extended farther downstream than that in a large tube.

6. Pressure drop correlation

In the following, a new pressure drop correlation is developed on the basis of Chisholm's *B*-coefficient method (Chisholm 1983). For two-phase flow in a smooth tube, the "*B*-coefficient" correlation is defined by

$$\phi_{\mathrm{fLo}}^2 = 1 + (\Gamma^2 - 1)\left[Bx^{0.875}(1-x)^{0.875} + x^{1.75}\right], \tag{1}$$

where ϕ_{fLo}^2 is a dimensionless two-phase multiplier due to friction where the entire flow is liquid only, B is a constant, x is equilibrium mass quality, and Γ^2 is a dimensionless physical property coefficient defined by

$$\Gamma^2 = \frac{(\mathrm{d}p/\mathrm{d}z)_{\mathrm{fGo}}}{(\mathrm{d}p/\mathrm{d}z)_{\mathrm{fLo}}}, \tag{2}$$

where $\mathrm{d}p/\mathrm{d}z$ is the pressure gradient along the test section, subscript fGo is for entire fluid flowing as a gas only, and subscript fLo is for entire fluid flowing as a liquid only. The value of B is determined according to the criteria given in Table 2.

The two-phase pressure drop over a quality range $x = x_a$ to $x = x_b$ can be calculated as

$$\Delta p_f = \bar{\phi}^2_{fLo} \times \Delta p_{fLo} \tag{3}$$

$$\bar{\phi}^2_{fLo} = \frac{1}{(x_b - x_a)} \int_{x_a}^{x_b} \phi^2_{fLo}(x)\,dx, \tag{4}$$

where Δp_f is pressure drop due to friction, and $\bar{\phi}$ is averaged two-phase multiplier.

The B-coefficient method does not contain a parameter that accounts for the effect of tube dimension and fluid surface tension. Surface tension is expected to be an important fluid property that needs to be included in the two-phase flow pressure drop correlation, particularly for refrigerants. In the B-coefficient method, the pressure effect is reflected in the change of density and viscosity of the vapor and liquid phases. The size effect is included in the calculation of single-phase Fanning friction factor f, which relies on the Reynolds number as in the case of frictional pressure drop of single-phase flow. The complexity of two-phase flow, therefore, is not fully described, especially for small channels where the pressure drop is strongly influenced by the dynamics of the growing and flowing bubbles confined in a narrow space with a flowing liquid phase. Additional factors must be considered, including the effects of the interface between vapor and liquid phases, and the wetted surface between liquid and the channel wall.

In developing the new correlation, the definitions of B and Γ^2 were modified to better reflect the physics of flow boiling in small tubes in which the channel size, fluid physical properties, mass flux, pressure, and quality are important factors. A new parameter related to Γ^2 is defined as

$$\psi^2 = C \times \Gamma^2 = C\frac{(dp/dz)_{fGo}}{(dp/dz)_{fLo}}, \tag{5}$$

where ψ is the physical property index and C is a constant. The constant C in Eq. (5) is considered to be a scaling factor that represents the difference in pressure gradient between small tubes and large tubes.

Table 2
Criteria for selecting B

Γ	G (kg/m^2 s)	B
≤ 9.5	≤ 500	4.8
	$500 < G < 1900$	$2400/G$
	≥ 1900	$55/G^{0.5}$
$9.5 < \Gamma < 28$	≤ 600	$520/(\Gamma G^{0.5})$
	> 600	$21/\Gamma$
≥ 28	–	$15{,}000/\Gamma^2 G^{0.5}$

A dimensionless number that can be used to substitute for the current B-coefficient was next sought. Cornwell and Kew (1993) proposed a dimensionless group called the "confinement number", expressed as

$$N_{\text{conf}} = \frac{\left[\frac{\sigma}{g(\rho_L - \rho_G)}\right]^{0.5}}{D}, \tag{6}$$

where N_{conf} is the dimensionless confinement number, σ the surface tension, g the gravitational constant, ρ_L the liquid density, ρ_G the vapor density, and D is the inside diameter of the channel. The numerator of this dimensionless parameter represents the ratio of surface tension force to buoyancy force.

Cornwell and Kew (1993) and Kew and Cornwell (1995) performed boiling heat transfer tests with R-113 in a small channel, with a gap size of 1–3 mm, and reported that the confinement number correlated well with their heat transfer test results. However, in their published papers, they did not use the confinement number to correlate their two-phase flow pressure drop data. In our development of the new correlation, we used the confinement number N_{conf} as a substitute for the B coefficient, because N_{conf} includes surface tension and hydraulic diameter and can thus account for the maximum size of the bubble confined in small channel during a flow boiling process. The confinement number N_{conf}, given by Eq. (6), also shows that a smaller channel gives a higher value of N_{conf}, which leads to a higher pressure drop.

Including the scaling factor C as a multiplier on Γ^2, and using the confinement number N_{conf} to replace B, the new form is given as

$$\phi_{\text{fLo}}^2 = 1 + (C\Gamma^2 - 1)\left[N_{\text{conf}} x^{0.875}(1-x)^{0.875} + x^{1.75}\right], \tag{7}$$

where x is equilibrium mass quality. In application, Eq. (7) is substituted into Eq. (4), and Eq. (4) is used with Eq. (3) to calculate pressure drop. The major difference between the proposed correlation and other correlations, such as the B-coefficient method (Chisholm, 1983) and that of Souza and Pimenta (1995), is the distinction between flow regimes for the entire flow of liquid or gas flowing alone in the channel, and the use of the confinement number.

For the special case of saturated liquid at the entrance, which is the situation for the subject tests, $x = 0$, the integrated result is

$$\Delta p_{\text{f}} = \Delta p_{\text{fLo}}\left\{1 + (C\Gamma^2 - 1)\left[N_{\text{conf}} F_1(x_{\text{exit}}) + F_2(x_{\text{exit}})\right]\right\}, \tag{8}$$

where x_{exit} is the exit quality, and

$$F_1(x_{\text{exit}}) = 0.0056 + 0.6466 x_{\text{exit}} - 0.5343 x_{\text{exit}}^2 + 0.0888 x_{\text{exit}}^3 \tag{9}$$

$$F_2(x_{\text{exit}}) = \frac{(x_{\text{exit}})^{1.75}}{2.75}. \tag{10}$$

The equation for F_1 in (9) was developed by curve-fitting calculated values from Chisholm

(1983) over the range of $x_0 = 0$ to $x = x_{\mathrm{exit}}$. The equation for F_2 in Eq. (10) was obtained by direct integration.

Eq. (8) was used in an optimization procedure to determine the value of the constant C. The optimization process was performed using a statistical analysis package, and was based on 610 data points of the three fluids; see Table 1. After the optimization process, the constant C in Eq. (5) was determined to be 4.3.

The new frictional two-phase multiplier can then be expressed as

$$\bar{\phi}^2_{\mathrm{fLo}} = 1 + (4.3\Gamma^2 - 1)\left[N_{\mathrm{conf}} x^{0.875}(1-x)^{0.875} + x^{1.75}\right] \tag{11}$$

where N_{conf} is given by Eq. (6). Thus, the final form of the proposed correlation will be

$$\Delta p_{\mathrm{f}} = \Delta p_{\mathrm{fLo}}\left\{1 + (4.3\Gamma^2 - 1)\left[N_{\mathrm{conf}} x^{0.875}(1-x)^{0.875} + x^{1.75}\right]\right\}. \tag{12}$$

Eq. (12) is applicable for smooth tubes, for the refrigerants tested in the present study, and for pressure from 138 to 864 kPa, mass flux from 33 to 832 kg/m^2 s, heat flux from 2.2 to 90.8 kW/m^2, and quality from 0 to 0.95. In the final form of frictional two-phase multiplier in Eq. (11), the confinement number N_{conf} is the only parameter that accounts for fluid surface tension and the flow characteristic of two-phase flow boiling in small channels, where bubbles grow, coalesce into large bubbles, are elongated due to the size restriction of the small channel, and slide downstream, as reported by Kasza and Wambsganss (1995) and Kasza et al. (1997). The pressure drop calculated with the new frictional two-phase multiplier, Eq. (11), is compared with all experimental data in Fig. 10.

A data analysis shows that most of the data are within 20% of error of prediction, and 93.8% of data are within 30%. The proposed correlation has a mean deviation of 12.8% in predicting the frictional two-phase flow pressure drop for all three fluids, meanwhile the B-coefficient method predicts the same data with a mean deviation of up to 40%. This represents a significant improvement of the proposed correlation, (12), over the B-coefficient method.

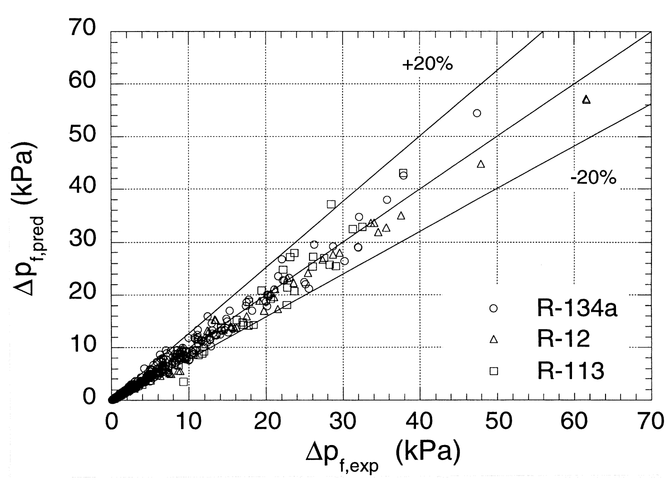

Fig. 10. Predicted frictional two-phase flow pressure drop using Eq. (12) versus experimental frictional two-phase flow pressure drop data.

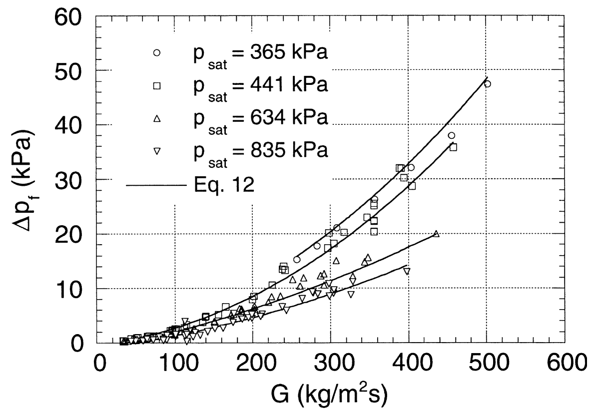

Fig. 11. Comparison between present correlation and experimental data of R-134a.

Predictions with Eq. (12) are also compared with R-134a experimental data in Fig. 11. It is shown that Eq. (12) accurately predicts the frictional two-phase flow pressure drop of R-134a for all R-134a data at four pressures at the quality $x = 0.71 \pm 0.06$.

7. Concluding remarks

- An experimental investigation of flow boiling of three refrigerants in small channels produced an extensive database of two-phase pressure drop information.
- Pressure drop with R-134a was compared with R-12 on the basis of both similar mass flux and heating rate. In both cases, the trends were in reasonable agreement with those observed in comparisons with large-tube data.
- Five state-of-the-art large-tube correlations were evaluated, but they failed to predict the pressure drop of flow boiling in small channels for all test conditions.
- A new correlation for two-phase flow frictional pressure drop in small channels was developed on the basis of the *B*-coefficient method, taking into account the effects of surface tension and channel size. The new correlation is applicable for smooth tubes with hydraulic diameters of 2.40–2.92 mm for the three refrigerants tested, and the range of quality and flux tested.
- The proposed correlation shows a significant improvement over the *B*-coefficient method in predicting the frictional pressure drop of all three fluids.

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