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A new pressure drop correlation for subcooled flow boiling of refrigerants

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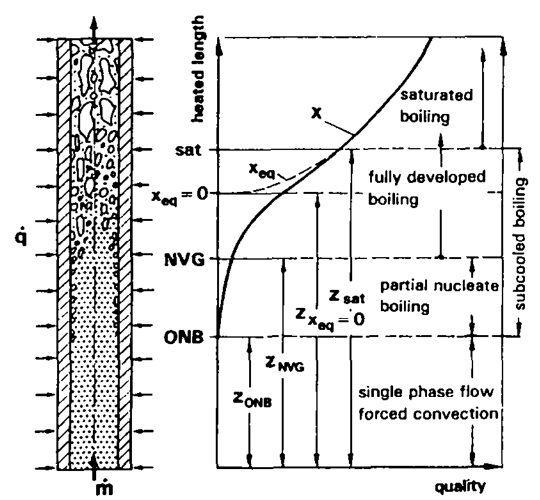
Abstract—Pressure drop data from experiments on subcooled flow boiling of refrigerant R12 and R134a have been correlated. A wide range of pressures (8–20 bar), mass fluxes (750–3000 kg s^{-1} m^{-2}), subcooling temperatures (2–47.6 K) and heat fluxes (up to 207 500 W m^{-2}) has been considered. The new correlation for the reduced pressure drop (sum of frictional and accelerational component) is a function of the boiling number and the Jakob number. The ratio of heated and wetted perimeter of the flow channel is included in the correlation so that it is applicable for both tubes and annuli. With the exception of six out of a total of 326 experimental data points all data lie within ±25% around the correlation.

1. INTRODUCTION

PHENOMENA occurring during subcooled boiling in a vertical tube are presented in Fig. 1. The fluid being subcooled at the tube inlet flows through a uniformly heated channel. Along this path different heat transfer regimes are encountered. A thermal boundary layer begins to form in the liquid flow immediately after entering the heated tube. Heat is transferred to the fluid by single phase forced convection. The wall temperature rises in the flow direction. At a certain point (ONB, onset of nucleate boiling) the wall temperature is sufficiently high to activate the first nucleation sites. Bubbles form at the wall but recondense immediately before entering the still subcooled bulk flow. Consequently the void fraction is very small. The latent heat of the condensing bubbles raises the temperature of the bulk flow so that at a certain point downstream of the ONB-point the rate of vapour generation on the wall becomes larger than the rate of condensation in the core. This is the so called point of net vapour generation (NVG). Further downstream a point is reached, where theoretically saturation is attained (theoretical thermodynamic equilibrium, $x_{eq} = 0$). Actually this saturation state of the bulk liquid occurs still further downstream at point 'sat'. The saturated boiling regime starts here. In the entire subcooled boiling region both the heat transfer coefficient and the pressure drop are larger than in single phase flow.



ONB	onset of nucleate boiling
NVG	net vapour generation
$x_{eq} = 0$	thermodynamic equilibrium (theoretical)
sat	saturated

FIG. 1. Flow patterns, void fraction and heat transfer regimes for flow boiling in a vertical tube.

The subcooled flow boiling region extends from the ONB-point to the point where saturated boiling starts. As in saturated boiling, the subcooled flow boiling region can be subdivided into a partial nucleate boiling region and a fully developed boiling region.

Subcooled flow boiling as a complex physical phenomenon is still not sufficiently well understood to allow reliable predictions of heat transfer and pressure drop with simple and generally applicable correlations. A new correlation for the pressure drop in subcooled flow boiling of refrigerants which employs non-dimensional parameters is suggested here.

2. LITERATURE SURVEY

In literature only few studies of pressure drop in the subcooled flow boiling region can be found. The empirical correlations are summarized in Table 1. All these correlations follow the suggestion of Lockhart and Martinelli [1] to use the ratio of the two-phase

NOMENCLATURE

Bo	boiling number, $\dot{q}/\dot{m}\Delta h_v$ [–]	$\Delta p_{tot}\downarrow$	total pressure drop in downflow [N m^{-2}]
$c_{p,l}$	specific heat capacity [J kg^{-1} K^{-1}]	ΔT_u	subcooling [K]
D_h	heated perimeter [m]	η	dynamic viscosity [kg m^{-1} s^{-1}]
D_w	wetted perimeter [m]	ξ	friction coefficient [–]
Ja	Jakob number, $c_{pl}\Delta T_u/\Delta h_v$ [–]	ρ_l	density of saturated liquid [kg m^{-3}]
$\dot{m}$	mass flux [kg s^{-1} m^{-2}]	ρ_g	density of saturated vapour [kg m^{-3}]
$\dot{q}$	heat flux [W m^{-2}]	τ	shear stress [N m^{-2}]
Re	Reynolds number, $\dot{m}d/\eta$ [–]	τ_w	wall shear stress [N m^{-2}]
x	quality [–]	Φ^2	pressure drop ratio [–].
z	coordinate.		

Greek symbols

α	heat transfer coefficient [W m^{-2} K^{-1}]
Δh_v	latent heat of evaporation [J kg^{-1} K^{-1}]
Δp_a	accelerational pressure drop [N m^{-2}]
Δp_{err}	error in pressure drop [N m^{-2}]
Δp_f	frictional pressure drop [N m^{-2}]
Δp_h	hydrostatic pressure drop [N m^{-2}]
Δp_r	reduced pressure drop [N m^{-2}]
$\Delta p_{tot}\uparrow$	total pressure drop in upflow [N m^{-2}]

Subscripts

acc	accelerational
fric	frictional
ONB	onset of nucleate boiling
scb	subcooled boiling
tot	total
$x_{eq=0}$	theoretical thermodynamic equilibrium (saturation)
1ph	single phase flow
2ph	two-phase flow.

flow pressure drop and the pressure drop found in liquid flowing with the same mass flux. In all cases water was used as working fluid, thus the correlations are valid for water only. One of the first papers published was by Reynolds [2]. The local boiling pressure drop data for forced circulation of distilled water in a uniformly heated, horizontal stainless steel tube were correlated with an empirical equation within $\pm 20\%$. The equation by Reynolds yields the sum of the frictional and the accelerational pressure drop. Owens and Schrock [3] correlated the total pressure drop data, i.e. for frictional, accelerational and hydrostatic effects, of water in a vertical tube with an empirical correlation. The maximum deviations are $+49\%$ and

Table 1. Pressure drop correlations (valid only for water)

Reynolds [2]	$\left[\left(\frac{dp}{dz}\right)_{scb}\Big/\left(\frac{dp}{dz}\right)_{1ph}\right]_{fric+acc} = \cosh\left[\left(1.2+1.458\left(\frac{\dot{q}}{10^6}\right)\right)Z\right]$
Owens/Schrock [3]	$\left[\left(\frac{dp}{dz}\right)_{scb}\Big/\left(\frac{dp}{dz}\right)_{1ph}\right]_{tot} = 0.97+0.028\exp(6.13Z)$
Rohde [4]	$\left[\left(\frac{dp}{dz}\right)_{scb}\Big/\left(\frac{dp}{dz}\right)_{1ph}\right]_{tot} = \exp\left[\frac{26.48}{p}\left(\frac{\dot{q}}{\dot{q}_{1ph,max}}-1\right)\right]$ $\dot{q}_{1ph,max} = \frac{(\Delta T_u)_{in}+\Delta T_s}{[4z/(\dot{m}c_{pl}d)+1/\alpha_{1ph}]}$
Tarasova/Orlov [6]	$\frac{\tau}{\tau_w} = 1+3.09\left(\frac{\rho_l}{\rho_g}Bo\right)^{0.7}\left[7-\left(1+48\frac{x_{eq}(z)}{x_{eq}(z_{ONB})}\right)^{0.5}\right]\frac{D_h}{D_w}$ (frictional + accelerational)
Tarasova *et al.* [7]	$\left[\left(\frac{dp}{dz}\right)_{scb}\Big/\left(\frac{dp}{dz}\right)_{1ph}\right]_{tot} = 1+Bo^{0.7}\left(\frac{\rho_l}{\rho_g}\right)^{0.78}\frac{20Z}{(1.315-Z)}$
scb = subcooled boiling	$Z = \frac{(z-z_{ONB})}{(z_{x_{eq=0}}-z_{ONB})} = \frac{\text{length from ONB}}{\text{length between } x_{eq=0} \text{ and ONB}}$
1ph = single phase flow	$z_{ONB} = z-$coordinate of onset of nucleate boiling

−36%. The authors only used data for which the subcooled' boiling length could be clearly defined. Rohde [4] gave another empirical correlation for the total pressure drop of subcooled boiling water.

In the German VDI-Wärmeatlas [5] an empirical correlation by Tarasova and Orlov [6] is recommended. The equation is based on the general momentum balance for calculating the frictional and the accelerational pressure drop using a fictitious wall shear stress. Another simplified correlation was given later by Tarasova *et al.* [7]. In both correlations the boiling number and the density ratio are included.

Dormer and Bergles [8] described a graphical method to predict the total pressure drop in subcooled boiling water in horizontal tubes with small diameters (from 1.57 to 4.57 mm).

Jens and Lottes [9] also measured the pressure drop in subcooled boiling water but they were not able to correlate their data. They mentioned, that the effect of dissolved gas and scale on pressure drop was unpredictable in most cases.

Tong [10] gave an overview of purely empirical correlations in the order of pressure ranges (from $p = 3$ to 150 bar) valid for water only.

Mayinger *et al.* [11] observed periodical pressure drop oscillations in subcooled boiling water. The frequency and the amplitude of these oscillations increased with increasing heat flux and subcooling. Instabilities and pulsations of the flow may occur. The heater surface can be damaged by burnout.

Stängl [12] measured the total pressure drop of refrigerant R12 together with the void fraction in a vertical annulus. The pressure drop data were corrected by subtracting the hydrostatic component so that the so called reduced pressure drop was obtained. The behaviour of the reduced pressure drop was quite similar to the behaviour of the void fraction. For the pure frictional component Stängl [12] found good agreement within ±25% with the Friedel correlation [13].

In an earlier paper Hahne *et al.* [14] reported on void fraction distribution and pressure drop in subcooled flow boiling of R12 in a vertical tube. A new measurement strategy has been suggested for eliminating the hydrostatic component by subtracting the two values of the pressure drop for upflow and downflow from each other. The so called reduced pressure drop showed hysteresis with respect to changes in the heat flux. The heat transfer coefficient shows a similar hysteresis with increasing and decreasing heat flux.

The use of a pressure drop ratio or a shear stress ratio implies that the correlations predict a value of unity for zero heat flux. The correlations by Reynolds [2] and by Rohde [4] do not fulfill this criterion. The other correlations are of an additive form and provide a value of unity for zero heat flux. Owens and Schrock [3] chose a value of 0.97 instead of unity in their correlation in order to take account of the fact that the two-phase flow pressure drop can actually fall below the value for single phase flow: due to nucleation the wetted area of the wall is reduced. Since the viscosity of the vapour is much smaller than the viscosity of the liquid a decrease of the frictional component of the pressure drop is obtained. In the case of high subcoolings and high mass fluxes the vapour condenses again in the core of the flow, the accelerational component of the pressure drop vanishes and the resulting total pressure drop is smaller than for single phase flow. Although the effect is small, taking it into account in the manner proposed by Owens and Schrock [3], for zero heat flux it would lead to a physically doubtful prediction of a pressure drop being always 97% of the single phase value.

3. EXPERIMENTAL SET-UP

The experiments were conducted in a forced convection loop. A schematic flow diagram including the main components of the experimental apparatus is shown in Fig. 2. Subcooled liquid is pumped by a canned motor pump. The volumetric flow rate is measured by a turbine flow meter. The liquid is heated to a preset degree of subcooling and then passes through a 2 m long inlet section before entering the test section. The flow can be directed upwards or downwards. Sight glasses are mounted on both ends of the test section to observe flow phenomena; they also reduce axial heat losses.

The test section (see Fig. 3) is a 550 mm long vertical copper tube with an inner diameter of 20 mm. The heated length is 500 mm. Four heating wires (Philips Thermocoax) are wound around the tube in four

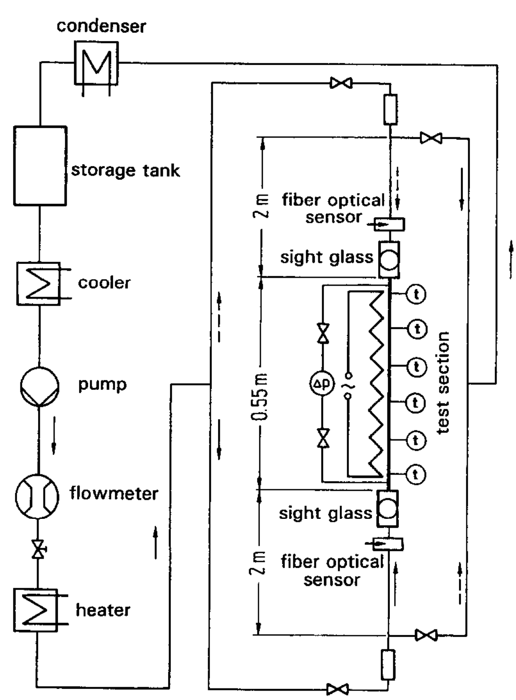


FIG. 2. Experimental set up.

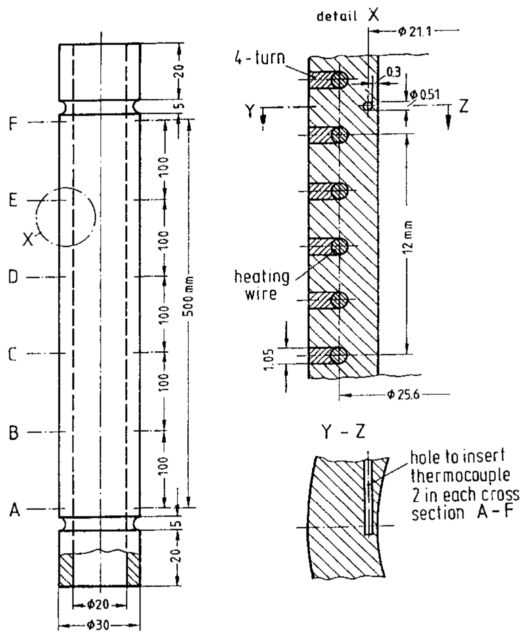


FIG. 3. Test section.

spiral grooves. The maximum heating power is 4×1000 W.

Overall mass flux, temperature, pressure, and subcooling at the entrance of the test section are kept constant during an experimental run.

Pressure taps are situated at the inlet and the outlet of the test section. At these locations four small holes are spaced 90° apart, around the circumference of the test section. A circular tube connects the four pressure taps and the tubing to the differential pressure transducer.

The pressure in the test section is measured with a Bourdon-tube pressure gauge with an accuracy of ± 13 mbar. The pressure drop over the entire length ($\Delta z = 550$ mm) of the test section is measured with a high precision differential pressure transducer. Its accuracy is ± 0.3 mbar.

4. PRESSURE DROP MEASURING STRATEGY

The total pressure drop includes three components: the frictional pressure drop Δp_f, the accelerational pressure drop Δp_a and the hydrostatic pressure drop Δp_h. In addition, a systematic error—Δp_{err}—is introduced by the weight of the liquid in the vertical sections of the pressure tubes. The first two components oppose the motion of the fluid. Their sign depends on the flow direction. The hydrostatic pressure drop and the systematic error, however, depend only on the direction of the acceleration of gravity and not on flow direction.

In upflow all four components act against the fluid motion. The total pressure drop for upflow $\Delta p_{tot}\uparrow$ can therefore be expressed as

$$\Delta p_{tot}\uparrow = \Delta p_f + \Delta p_a + \Delta p_h + \Delta p_{err}. \qquad (1)$$

In case of downflow the first two components change their sign:

$$\Delta p_{tot}\downarrow = -\Delta p_f - \Delta p_a + \Delta p_h + \Delta p_{err}. \qquad (2)$$

The hydrostatic component and the systematic error can be eliminated by means of an appropriate measurement strategy. By subtraction of equation (2) from equation (1), Δp_h and Δp_{err} are eliminated yielding the reduced pressure drop

$$\Delta p_r = 1/2(\Delta p_{tot}\uparrow - \Delta p_{tot}\downarrow) = \Delta p_f + \Delta p_a. \qquad (3)$$

For this measuring strategy two conditions have to be met: all experimental parameters have to remain constant when the flow direction is reversed, and the resulting conditions in the test section, i.e. the heat transfer, the flow regime and the void fraction distribution will not be changed by the flow reversal.

Only for heat transfer is the latter condition exactly fulfilled. Neither the heat transfer coefficient nor the wall superheat are influenced in any measurable way by the flow direction, see ref. [15]. The void fraction distribution, however, differs somewhat between upflow and downflow; this introduces another systematic error which is, however, much smaller than Δp_{err}.

In order to test the measuring strategy it was applied to single phase flow. The results were compared to values predicted by the well-known Blasius correlation for turbulent flow. The measured values correspond well with this correlation.

5. RESULTS AND DISCUSSION

5.1. *Simplified pressure drop correlation*

The experimental results show that the reduced pressure drop depends primarily on the heat flux $\dot{q}$, the total mass flux $\dot{m}$ and the subcooling ΔT_u at the inlet of the test section, see ref. [14]. Combining these three physical quantities into common non-dimensional parameters leads to the boiling number *Bo* and the Jakob number *Ja* as independent parameters:

$$Bo = \frac{\dot{q}}{\dot{m}\Delta h_v}, \quad Ja = \frac{c_{p,l}\Delta T_u}{\Delta h_v}. \qquad (4a,b)$$

These numbers are chosen in analogy to common simple correlations for heat transfer in subcooled boiling. In order to obtain a non-dimensional pressure drop ratio Φ^2 containing the reduced two phase pressure drop per unit length $(\Delta p/\Delta z)_{r,2ph}$, the single phase pressure drop $(\Delta p/\Delta z)_{1ph}$, for liquid flowing with the same mass flux, is chosen as the reference value as suggested by Lockhart and Martinelli [1]. Considering that in single phase flow Φ^2 has to attain unity, a simple additive form was chosen for the correlation.

$$\Phi^2 = \frac{(\Delta p/\Delta z)_{r,2ph}}{(\Delta p/\Delta z)_{1ph}} = 1 + C\,Bo^m Ja^n. \qquad (5)$$

The correlation reaches the asymptotic boundary value $\Phi^2 = 1$ in case of single phase flow with $Bo = 0$.

The reference value for single phase flow is obtained from the Blasius correlation for turbulent flow,

$$\Delta p_{1ph} = \xi \dot{m}^2 L/(2d\rho) \quad \text{with} \quad \xi = 0.3164\, Re^{-0.25}. \tag{6}$$

All fluid properties are taken at the saturation state corresponding to the pressure in the test section. Pressure drop data are considered for decreasing heat flux only.

The following pressure drop correlation was found for our R12 data

$$\Phi^2 = 1 + 13\,000\, Bo^{1.6} Ja^{-1.2}. \tag{7}$$

A correlation for this pressure drop ratio is shown in Fig. 4. The pressure drop ratio Φ^2 is plotted vs the boiling number Bo for different Jakob numbers Ja. With increasing boiling number the pressure drop ratio increases due to an increasing void fraction.

With increasing Jakob number the pressure drop ratio decreases. This is due to a decrease in void fraction with increasing subcooling.

The asymptotic boundary value for the pressure drop ratio $\Phi^2 = 1$ is reached for all Jakob numbers at zero boiling number.

In Fig. 4 the experimental data are subdivided in six classes of Jakob numbers with each Ja being in a range of $\Delta Ja = \pm 0.005$.

Of all experimental results 92% fit within $\pm 20\%$ into the correlation. The root mean square deviation is 10.9%.

For the R12 substitute R134a the following correlation was obtained

$$\Phi^2 = 1 + 15\,000\, Bo^{1.6} Ja^{-1.2}. \tag{8}$$

The root mean square deviation is 10.6%.

The higher constant for R134a may be an effect of pressure. Compared to R12 the pressure for R134a was about 1 bar higher in the experiments.

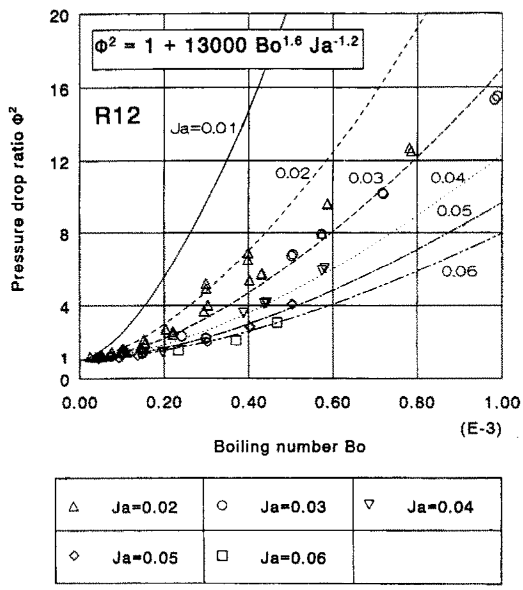


FIG. 4. Pressure drop correlation for R12 (lines) and experimental results (points).

Data by Stängl [12], obtained in a vertical annulus with R12 at elevated pressures of 15–20 bar, were also correlated. The following correlation

$$\Phi^2 = 1 + 80\,000\, Bo^{2.1}\, Ja^{-0.9} \tag{9}$$

gives a root mean square deviation of 7.1% with largest deviations of +18.1% and −17.1%.

The Reynolds-data [2], for water in a horizontal tube at lower pressures than we applied, gives good agreement with

$$\Phi^2 = 1 + 32\,500\, Bo^{1.6}\, Ja^{-1.2}. \tag{10}$$

The root mean square deviation is 6.6%.

In Table 2 various parameters are summarized for our data and those of Reynolds [2] and Stängl [12].

Table 2. Parameters for pressure drop correlations

Fluid	R12 our data	R134a our data	R12 Stängl [12]	H_2O Reynolds [2]
Geometry	Vertical tube	Vertical tube	Vertical annulus	Horizontal tube
Material	Copper	Copper	Stainless steel	Stainless steel
Length [mm]	520	520	1390	1829
Hydr. diameter [mm]	20	20	14	9.53
Data points	71	78	119	58
Pressure [bar]	8.5–9.5	8.8–10.8	15–20	3.0–6.9
Mass flux [kg s^{-1} m^{-2}]	750–1700	950–1950	1000–3000	2100–3200
Subcooling [K]	2.0–7.7	1.2–11.6	2.4–47.6	23.3–88.9
Heat flux [W m^{-2}]	5200–101 500	10 000–101 500	94 200–207 500	411 000–960 000
Boiling number	2.5×10^{-5}–9.9×10^{-4}	4.8×10^{-5}–6.3×10^{-4}	6.0×10^{-4}–1.7×10^{-3}	6.0×10^{-5}–2.2×10^{-4}
Jakob number	0.015–0.06	0.01–0.11	0.028–0.55	0.047–0.18
Reynolds number	71 500–160 800	111 800–227 000	81 500–215 000	99 000–185 000

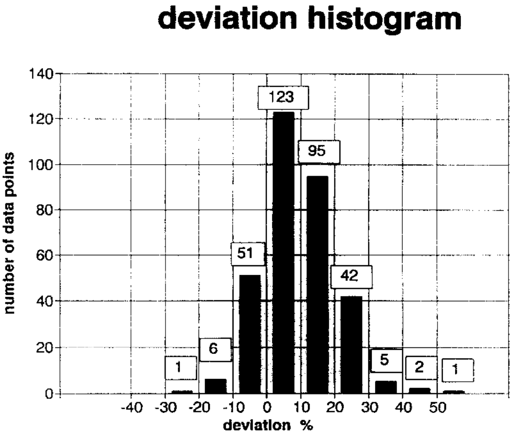


FIG. 5. Deviation of experimental results from equations (11) for R12, R134a (with $C = 500$) and for water ($C = 80$).

5.2. *Extended pressure drop correlation*

The constant C in the presented pressure drop correlation ranges from 13 000 to 80 000 (equations (7)–(10)). The influence of pressure and geometry does not seem to be well represented by the non-dimensional numbers chosen here.

An effective extension could include the density ratio (ρ_l/ρ_g) and the ratio of the heated and wetted perimeter (D_h/D_w). The density ratio of saturated liquid ρ_l and saturated vapour ρ_g would represent the influence of pressure. The perimeter ratio is recommended by Tarasova and Orlov [6] as a correction factor for annular channels, where the heated surface differs from the wetted surface. In the case of tubes the perimeter ratio equals one.

The extended pressure drop correlation can now be written as

$$\Phi^2 = 1 + C\,Bo^m\,Ja^n(\rho_l/\rho_g)(D_h/D_w). \qquad (11)$$

For the regression analysis 268 result points for the refrigerants R12 and R134a were used to obtain the constant $C = 500$ and the exponents, $m = 1.6$ and $n = -1.2$. For the density ratio (ρ_l/ρ_g) and the perimeter ratio (D_h/D_w) exponents are chosen to be one, because of the relatively small range of these parameters with respect to boiling number and Jakob number. This gives the correlation

$$\Phi^2 = 1 + 500\,Bo^{1.6}\,Ja^{-1.2}(\rho_l/\rho_g)(D_h/D_w). \qquad (12)$$

For water [2] 58 results were used and the constant was found to be $C = 80$ with the same exponents $m = 1.6$ and $n = -1.2$ as for the refrigerants.

For the refrigerant- and water-results the deviations from the extended correlation (equation (11)) are shown in Fig. 5. Of all results, 98% fall within the range $\pm 25\%$ deviation. The root mean square deviation for all data is 10.8% with a maximum deviation of $+45.9\%$ and a minimum deviation of -34.6%.

A comparison between the calculated pressure drop ratio and the experimental one is shown in Fig. 6 and for a partial region in Fig. 7. The largest ratio obtained is about 16, most of the ratios are found in the range $1 \leqslant \Phi^2 \leqslant 4$. The deviations between the calculated and the experimental pressure drop ratio in this region are presented in Fig. 8. A strong scattering can be observed for Φ^2 between 1 and 3. The low values of Φ^2 can occur for high values of the boiling number in the case of high subcoolings.

6. CONCLUSIONS

• From two-phase flow experiments in a vertical tube the combined accelerational and frictional pressure drop can be obtained by subtracting pressure drop results for downflow from those for upflow. The hydrostatic pressure drop and a systematic error can thus be eliminated. This method is applicable to two-

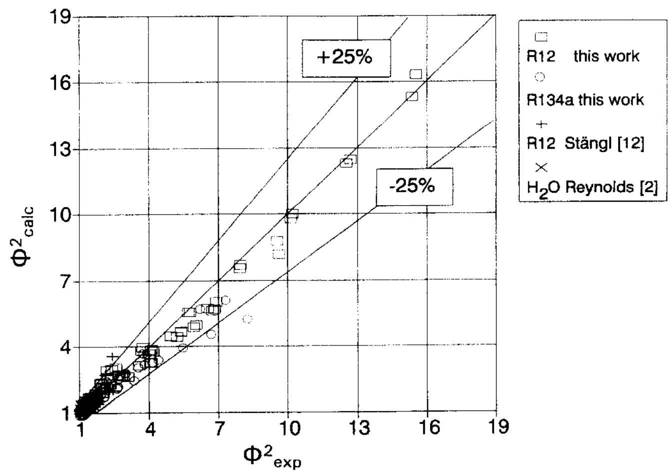


FIG. 6. Comparison between calculated (equation (11)) and experimental pressure drop ratios.

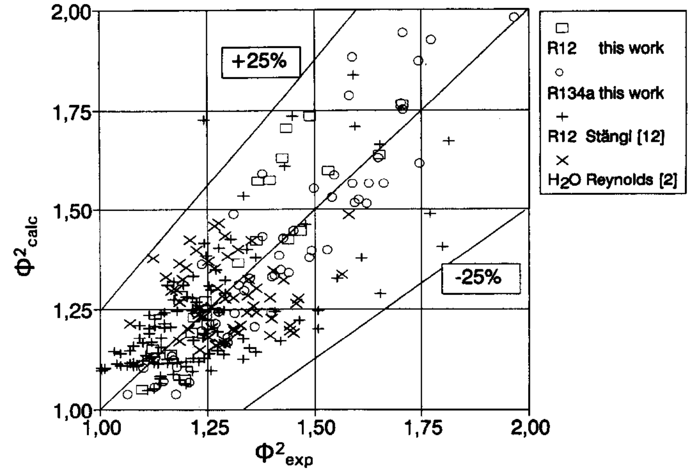


FIG. 7. Comparison between calculated (equation (11)) and experimental pressure drop ratio (enlarged).

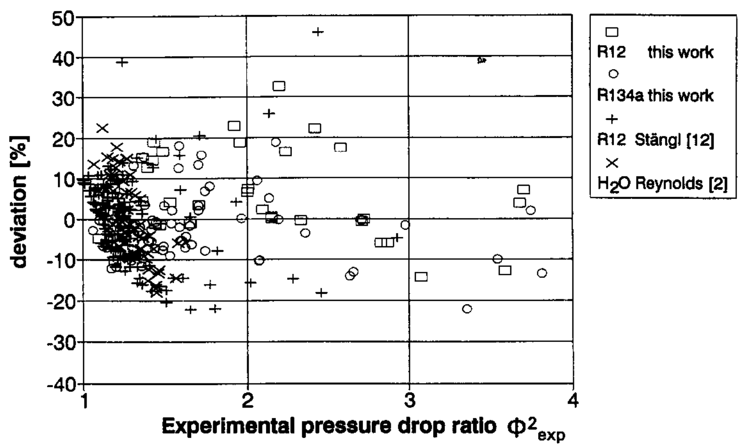


FIG. 8. Deviation between calculated (equation (11)) and experimental pressure drop ratio vs experimental pressure drop ratio.

phase flow in vertical tubes without flow regime changes.

• A good correlation for the reduced pressure drop for subcooled flow boiling of R12 and R134a in tubes and annular channels is:

$$\Phi^2 = 1+500\,Bo^{1.6}\,Ja^{-1.2}(\rho_l/\rho_g)(D_h/D_w).$$

This correlation represents 98% of the measured data within $\pm 25\%$. It gives the accelerational and frictional pressure drop $\Delta p_r = \Delta p_a + \Delta p_f$ in subcooled flow boiling. Thus this correlation allows for the calculation of the total pressure drop in horizontal tubes. In the case of vertical tubes a void fraction model must be used to calculate the hydrostatic component separately.

• The knowledge of the ONB-point is not necessary.

• Further work should be directed to test the new pressure drop correlation with data of other fluids and/or extended parameter ranges.

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