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International Journal of Multiphase Flow 31 (2005) 358–370
International Journal of Multiphase Flow
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Brief communication

An empirical correlation for calculating steam–water two-phase pressure drop in uniformly heated vertical round tubes

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Received 3 December 2003; received in revised form 31 October 2004

1. Introduction

The prediction of the pressure drop in two-phase flows is of critical importance in the design of thermal equipment used in the energy industry. Presently the most commonly used method for calculating the pressure drop consists of the numerical integration of the pressure distribution due to acceleration, gravitation and friction along the channels. Most of the available correlations for calculating the gravitation and acceleration components of the pressure gradient are based on the one-dimensional momentum equation written for the separated two-phase flow model. As a rule, the frictional pressure drop in two-phase flows is referred to the pressure loss of a pure liquid flow having the same total mass flow rate, which define the commonly used two-phase multiplier. The literature contains an innumerable number of relationships and models for calculating the two-phase multiplier. These relationships have, in general, good predictive accuracy for adiabatic flow conditions. In practice, they can also be used to calculate the two-phase pressure drop for non-adiabatic flows. In such a case, besides an additional correlation required for estimating the void fraction, it is still necessary to use supplementary relationships for determining the

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doi:10.1016/j.ijmultiphaseflow.2004.10.003

quality that corresponds to the onset of subcooled boiling conditions as well as the flow quality $\tilde{x}$ for the region where the thermodynamic quality x_{th} is quite low (Fig. 1(a)). Furthermore, some correction factors must be introduced to take into account the effect of the heat flux on both single- and two-phase frictional pressure losses. It must be pointed out, that each of these constitutive

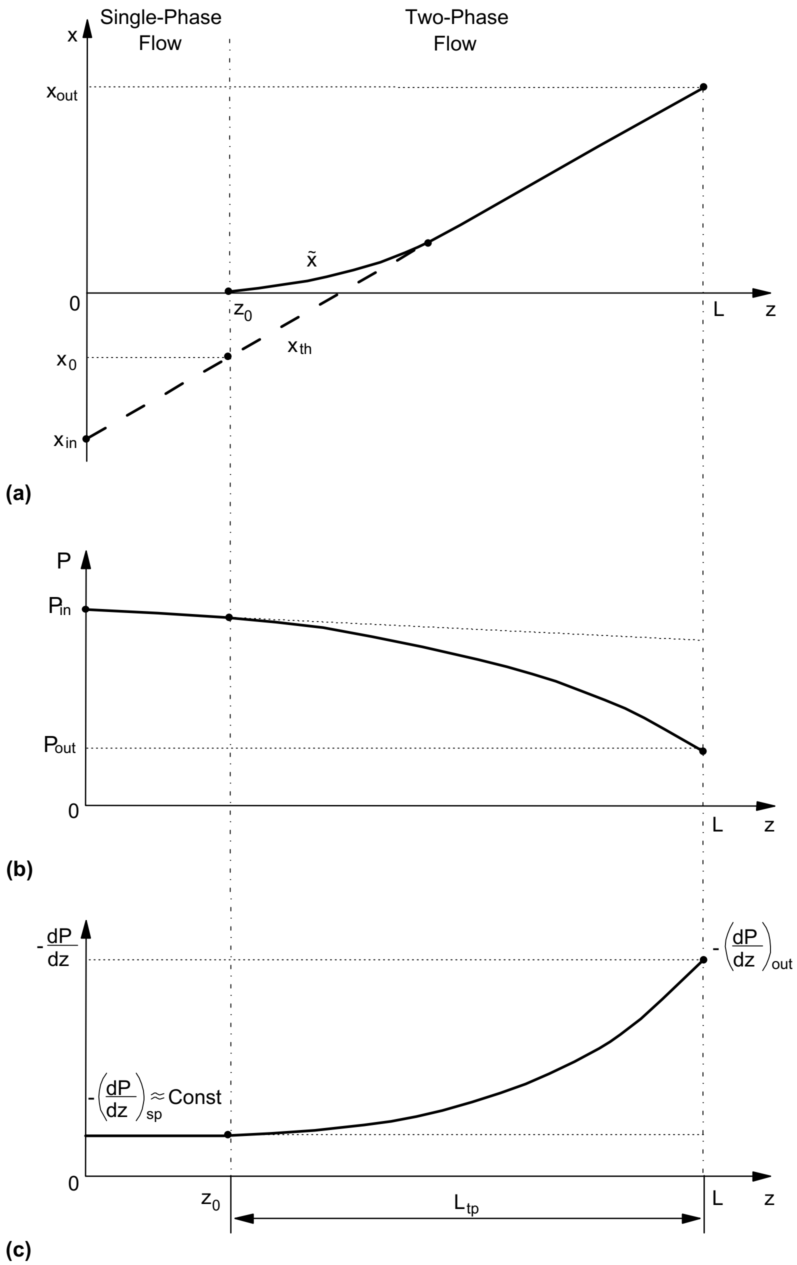

Fig. 1. Variation of the quality, the pressure and the pressure gradient along a heated channel.

relationships have their own range of applicability. In most of the cases, this range is quite difficult to determine, due to nonuniform parametric data distribution and it does not necessarily overlap the range of the correlations used to evaluate the other two-phase flow parameters. Hence, the necessity of simultaneously using several correlations to estimate two-phase pressure drops for non-adiabatic flows, limits the applicability of this approach and can generate huge unexpected errors.

2. Pressure drop in a heated tube

It is possible, however, to consider a different approach for calculating the pressure drop which would consist in a direct determination of ΔP in a heated channel. The possibility of applying such an approach results from the formal analysis of the pressure drop distribution along a channel (Fig. 1(b)). Assuming that:

- the single-phase flow pressure gradient $(\frac{dP}{dz})_{sp}$ does not depend on the heat flux;
- within the range given by the inlet temperature t_{in} and the saturation temperature t_{sat}, the effect of temperature on $(\frac{dP}{dz})_{sp}$ is neglected;
- the two-phase pressure drop distribution along the length L_{tp} in Fig. 1(c) (length over which the two-phase flow takes place) can be represented by the following exponential expression:

$$\left(\frac{dP}{dz}\right)_{tot} = \left(\frac{dP}{dz}\right)_{sp} + \left(\frac{z-z_0}{L_{tp}}\right)^m \left[\left(\frac{dP}{dz}\right)_{out} - \left(\frac{dP}{dz}\right)_{sp}\right], \tag{1}$$

then

$$\begin{aligned}\Delta P &= \int_0^L \left(\frac{dP}{dz}\right)_{tot} dz = \left(\frac{dP}{dz}\right)_{sp} L + \int_{z_0}^L \left[\left(\frac{dP}{dz}\right)_{out} - \left(\frac{dP}{dz}\right)_{sp}\right]\left(\frac{z'}{L_{tp}}\right)^m dz' \\ &= \Delta P_{sp} + \frac{L_{tp}}{m+1}\left[\left(\frac{dP}{dz}\right)_{out} - \left(\frac{dP}{dz}\right)_{sp}\right].\end{aligned} \tag{2}$$

It must be pointed out that if the shape of the real pressure distribution differs from that given by Eq. (1); the general form of Eq. (2) will be the same however it will be necessary to replace the multiplication factor $1/(m+1)$ by a more appropriate coefficient k for this type of distribution. Thus:

$$\Delta P = \Delta P_{sp} + kL_{tp}\Delta\left(\frac{dP}{dz}\right)_{out}. \tag{3}$$

For a round uniformly heated tube, the length L_{tp} can be determined from the equation:

$$L_{tp} = \frac{\dot{m}Dh_{LG}(x-x_0)}{4q''}, \tag{4}$$

where h_{LG} is the latent heat of vaporization and $x \equiv x_{th}$ (note that for the rest of this document the thermodynamic quality will be expressed by x). Thus, if the value of the coefficient k is known, the pressure loss ΔP may be determined using only the values of the pressure gradients at the inlet and at the outlet of the heated channel without integrating the pressure distribution along the length of the channel.

According to Aubé (1996) the profiles of the pressure drop distributions do not change significantly, consequently the range of variation of k must not be very large. A relationship for calculating this coefficient can be obtained by applying two approaches: (a) by analyzing the profiles of the pressure distributions along a heated channel, or (b) by applying available methods for calculating $\Delta(\frac{dP}{dz})_{out} = (\frac{dP}{dz})_{out} - (\frac{dP}{dz})_{sp}$ and developing a correlation for k based on experimental data of ΔP. The second approach is more effective, because it allows the effect of the uncertainty of other relationships necessary to calculate $\Delta(\frac{dP}{dz})_{out}$, to be reduced.

Moreover, the large number of iterations required for determining $(\frac{dP}{dz})_{out}$, will greatly complicate the procedure necessary for obtaining k. It is thus advisable to develop a correlation for obtaining ΔP directly. Compared to the most common methods based on the evaluation of the separate components of the pressure gradient, from the physical viewpoint the aforementioned approach can be less favorable (because a unique correlation must reflect the influence of different mechanisms which determine ΔP), however it allows the computational task to be reduced.

3. Data base and certain aspects of the correlation development

3.1. Data base

Seven different data sets have been used to develop the present correlation (Table 1). All the data have been obtained using water flows in vertical uniformly heated round tubes with some inlet subcooling. In all the cases, the heat was produced by Joule effect by applying an electrical current on the wall of the test section. In general there are differences between the heated length

Table 1
Range of experimental conditions of the pressure drop data bank

Authors	D (mm)	L (m)	P (bar)	$\dot{m}$ (kg/m^{2}s)	x	q'' (kW/m^2)	ΔP (bar)	No. of data
Alessandrini et al. (1963)	15.2; 24.9	2.45	49.9–50.9	1080–3890	0.022–0.515	1680–2710	0.30–1.36	48
Bertoletti et al. (1964)	4.9–9.2	0.40–2.7	48.8–98.7	1060–3940	0.018–0.774	476–4810	0.17–6.86	187
Peterlongo et al. (1964)	15.1	4.1	49.8–50.4	1070–3940	0.145–0.608	1080–2260	0.72–2.79	79
Leung (1994)	5.5	2.5	50.2–97.2	1130–9980	0.010–0.563	113–3220	0.32–15.54	498
Aubé (1996)	13.4; 22.9	1.8	9.9–42.4	1040–9920	0.001–0.217	416–3570	0.16–6.48	89
Olekhnovitch (1997)	8.0	0.75–3.5	5.0–40.4	980–6120	0.047–0.760	523–5550	0.56–17.42	477
Present study	8.0–15.7	1.0–3.0	9.7–40.4	910–6140	0.0004–0.648	57–4663	0.10–9.24	8557
Overall	4.9–24.9	0.40–4.1	5.0–98.7	910–9980	0.0004–0.774	57–5550	0.10–17.42	9935

and the length over which the total pressure drops were measured. It is important to remark, that in all the cases these differences are less than a few millimeters. Taking into consideration that toward the end of the heated length the pressure gradients can be quite high, some corrections to the data were nevertheless introduced. It must be noted that the lack of information on some details concerning the test sections used by different authors and the data treatment that in some cases they have applied, forced us to use specific corrections for each data set.

The test section used to carry out the present study [1] is shown schematically in Fig. 2. The power applied to the test section was simultaneously determined by two different methods. The first one consists of an analog multiplier that gives the power as the product of the voltage drop multiplied by the electric current measured with a 50 mV–5000 A, 1% calibrated shunt (Bach—Simpson, Model H5000-50). The second method consists of a numerical sampling and multiplication of the aforementioned parameters carried out by the computer through the data acquisition system. In this case the electric current passing through the test section was measured with a high accuracy LEM (Model LA-5000T) unit. Before starting a set of experiments, several heat balance test were carried out; the maximum observed uncertainty in the applied power was rarely greater than ±2% of the collected values. In addition, the inlet and outlet flow temperatures were measured with thermocouples calibrated to ±1 °C of the readings; the water flow rate was measured by using "Flow Technology" turbine flow meters with an accuracy better than ±1% of the reading. In both cases, the accuracy of the measurements was corroborated by frequent verifications carried out during the experiments. The internal diameters of the tubes were determined with a precision of ±0.3%; the inaccuracy in the length over which the pressure drop is determined, even if the thermal expansion is taken into account, cannot provoke an error greater than ±0.2%.

The value of the pressure drop in the test section (for $\Delta P < 6.1$ bar) was obtained from the reading of one (the most appropriate) of three differential pressure "Sensotec" transducers with an accuracy better than ±0.25% of the full scale, or from the difference between the inlet and outlet absolute pressures (for $\Delta P > 6.1$ bar). These pressures were measured with 51 bar "Sensotec" pressure transducers with an accuracy of ±0.1% of the full scale. Furthermore, a correction was carried out to take into account the weight of the water columns in the measurement lines from the pressure taps to the transducers. Each pressure line has a long horizontal leg, thus the correction due to the weight of the water column was carried out as a function of the ambient temperature measured closer to the location of the pressure transducers. Finally, the single-phase pressure drop taking place in ΔL_{in} was subtracted from the total pressure drop. The single-phase pressure drop is determined using the following relations:

$$\Delta P_{\text{sp0}} = \rho_{\text{L}}(t_{\text{in}}) g\,\Delta L_{\text{in}} + f\,\frac{\Delta L_{\text{in}}}{D}\,\frac{\dot{m}^2}{2\rho_{\text{L}}(t_{\text{in}})} \tag{5}$$

with

$$f = \frac{1.325}{\left[ln\left(\frac{k_{\text{s}}/D}{3.7} + \frac{5.74}{Re^{0.9}}\right)\right]^2}, \tag{6}$$

[1] The experimental data can be obtained by request from the authors of this paper.

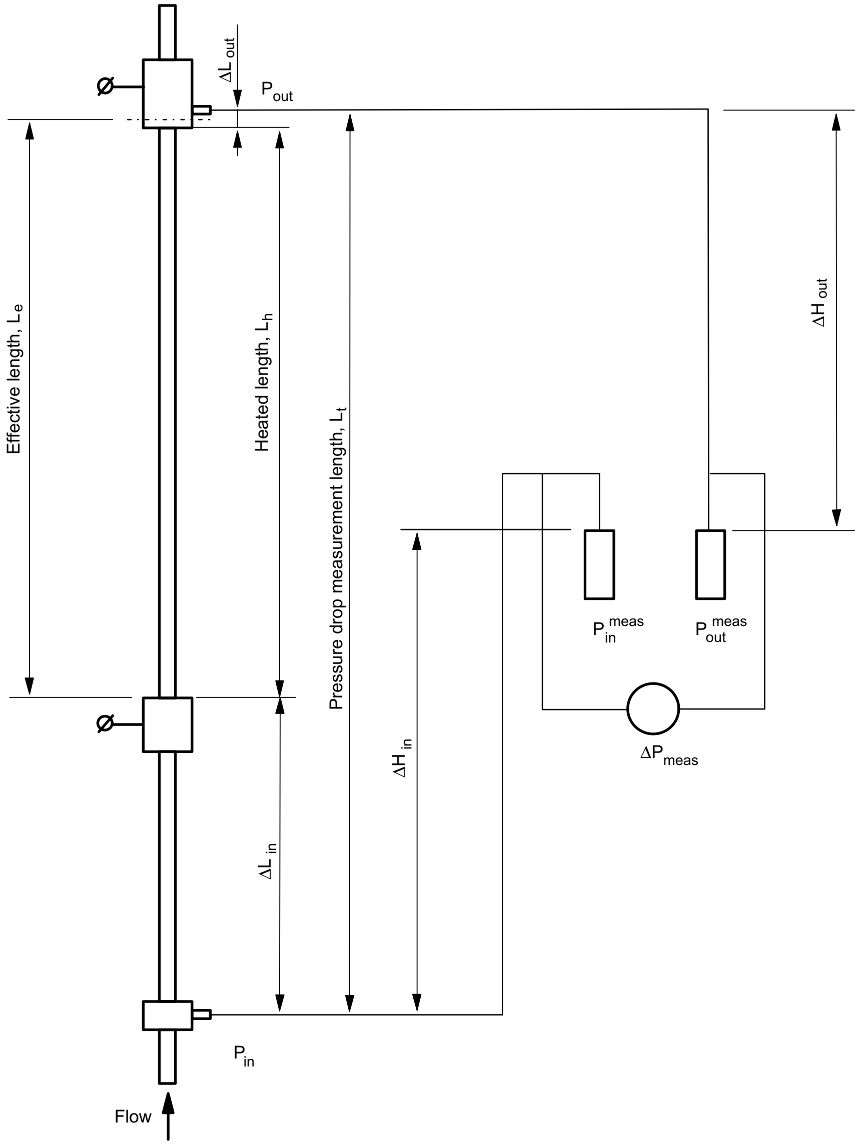

Fig. 2. Schematic view of the test section.

where k_s is the absolute surface roughness of the inner wall of the tube and $Re = \frac{\dot{m}D}{\eta_L(t_{in})}$ is the Reynolds number.

The pressure losses determined in this way have been referred to an effective length $L_e = L_h + \Delta L_{out}/2$, that has also been used to calculate the heat flux (for all the test sections ΔL_{out}'s are between 11 and 12 mm). It must be pointed out that the data have been obtained under low pressure conditions and for heat fluxes increased up to the critical heat flux (CHF), that is within a range where the two-phase pressure losses reach the highest value.

The data of Alessandrini et al. (1963), Bertoletti et al. (1964) and Peterlongo et al. (1964) were obtained only under CHF conditions. In all these experiments the pressure drops were measured over a total length that exceeded the heated length, with $\Delta L_{\rm in} \approx 10$–15 mm. These data, without subtracting the pressure drop in the single-phase flow region, have also been referred to the effective length, while the values of the heat fluxes were gathered without modification. It is important to remark that the precision of these data is probably not very high. A preliminary analysis has shown that in the data of Bertoletti et al. (1964) there exist 14 values, that even though they were collected under similar experimental conditions, are quite different (in some cases up to two times). These values were rejected, thus only 9921 data points were used to develop the proposed correlation.

Instead of studying the overall pressure drops in heated channels, Leung (1994) and Aubé (1996) studied the pressure drop distributions. In the first case, the pressure drops were measured along a total length which was a little bit shorter than the heated length and $L_{\rm h}$ was used to determine ΔP and q''. In the second case, the pressures were also measured upstream and downstream of the heated region and the pressure drops at the inlet and outlet of the heated length were determined by interpolation of the pressure profiles.

Finally, Olekhnovitch (1997) during his study of CHF under low pressure conditions, determined the pressure at the end of the heated length by using Friedel's (1979) correlation and the pressure measured 94 mm downstream of this point.

3.2. Surface roughness

In order to correctly predict the two-phase pressure drop with heat addition, not only it is necessary to know the inner surface roughness of the tube due to its fabrication, but also how it may change due to the formation of deposits caused by long periods of boiling flows. Aubé (1996), for all the tubes he used, applied an average roughness value of $\overline{k_{\rm s}} = 0.6$ mkm that he estimated from preliminary single-phase flow experiments. In the present study, similarly to the work of Leung (1994), the single-phase adiabatic flow pressure drop was measured after changing the flow conditions, just before starting to apply heat to the test section. This procedure permitted us to carry out a continuous control of the roughness of the surface by estimating its value from the measured single-phase pressure loss and the inverse form of Eq. (6):

$$k_{\rm s} = 3.7D\left[\exp\left(-\sqrt{\frac{1.325}{f}}\right) - \frac{5.74}{Re^{0.9}}\right]. \tag{7}$$

It was observed that the calculated roughness values have the following trends: (a) for tubes subjected to boiling flow and critical heat flux conditions, the surface roughness did not show a substantial change; (b) the surface roughness of the tubes used for the present study has almost no dependence on how they were manufactured (the tubes were obtained from different suppliers), neither on the material (tubes having an ID of 8 mm were manufactured from Inconel 600 while the others from stainless steel 316), nor the diameter; (c) an average value of the surface roughness of $\overline{k_{\rm s}} = 0.7$ mkm, observed in the present study, is quite close to that reported in Aubé (1996) whereas the roughness reported in Leung (1994) is substantially higher, $\overline{k_{\rm s}} = 3.9$ mkm; (d) for some cases of the work of Leung the calculated values of the surface roughness are abnormally high.

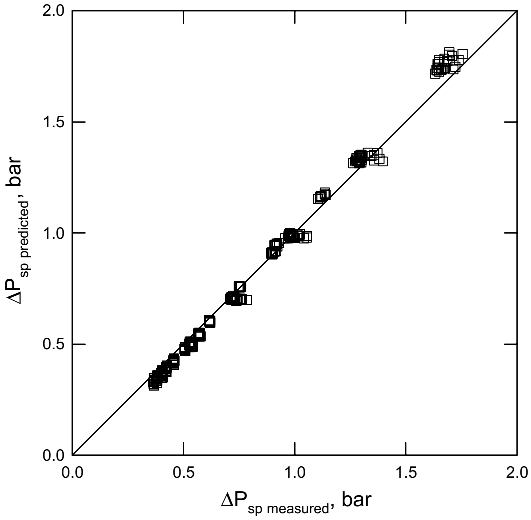

Fig. 3. Comparison of experimental and predicted single-phase pressure drops.

The fact that the surface roughness seems to be independent of some factors upon which it should depend, the unexplained high values of k_s for the data of Leung (1994) and the lack of information of the surface roughness of the tubes used by other researchers, do not permit ΔP_{sp} to be accurately predicted for each test section. Under such a situation a single value of $k_s = 1.5$ mkm, usually recommended for stainless steel seamless tubes has been used. A comparison between the predicted $\Delta P_{sp\,predicted}$ and measured $\Delta P_{sp\,measured}$ pressure losses for the present study is shown in Fig. 3.

3.3. Onset of subcooled flow boiling

There exists a large number of correlations for calculating the point of onset of subcooled flow boiling. For the current study the correlation of Miropolskiy et al. (1971) has been chosen because it presents the advantage of yielding the value of x_0 directly. This correlation is given by the following expression:

$$x_0 = -0.49\left(\frac{q''}{\dot{m}h_{LG}}\right)^{0.3} Re_q^{0.4}\left(\frac{P}{P_{cr}}\right)^{0.15}, \tag{8}$$

where

$$Re_q = \frac{(q''/h_{LG})\sqrt{\sigma/[g(\rho_L - \rho_G)]}}{\eta_L}. \tag{9}$$

Using this relationship can however lead to a difficulty in developing a correlation for ΔP. This difficulty is meanly due to the fact that for relatively low inlet subcooling conditions and quite high heat fluxes, boiling must start immediately at the beginning of the heated length. In such a case the pressure drop profile in this region will be different from that corresponding to a case where boiling starts far downstream from the beginning of the heated region (Fig. 4). From a view point of calculating the pressure drop, this change in the pressure drop profile can be taken into account by replacing $(\frac{dP}{dz})_{sp}$ and L_{tp} by an effective single-phase pressure drop gradient $(\frac{dP}{dz})_{sp\,eff}$, and the length of the tube L. Such a procedure would require specific conditions to determine when this replacement is necessary. It is obvious that this will involve an excessive complexity in determining the correlation's coefficients.

A different and rather approximate approach has been applied. Taking into account that for low inlet subcooling conditions and high heat fluxes L_{tp} obtained from Eq. (4) can be longer than the heated length, it is possible to accommodate the correlation for x_0 in such a way that the surface S_2 will be equal to S_1 (Fig. 4). In this case the calculated pressure losses will be quite close to the real ones. This approach has been taken into account by using a generic form of Eq. (8):

$$x_{0\Delta P} = A\left(\frac{q''}{\dot{m}h_{LG}}\right)^m Re_q^n \left(\frac{P}{P_{cr}}\right)^l \tag{10}$$

with the coefficients A, m, n and l obtained from the regression analysis of the ΔP data.

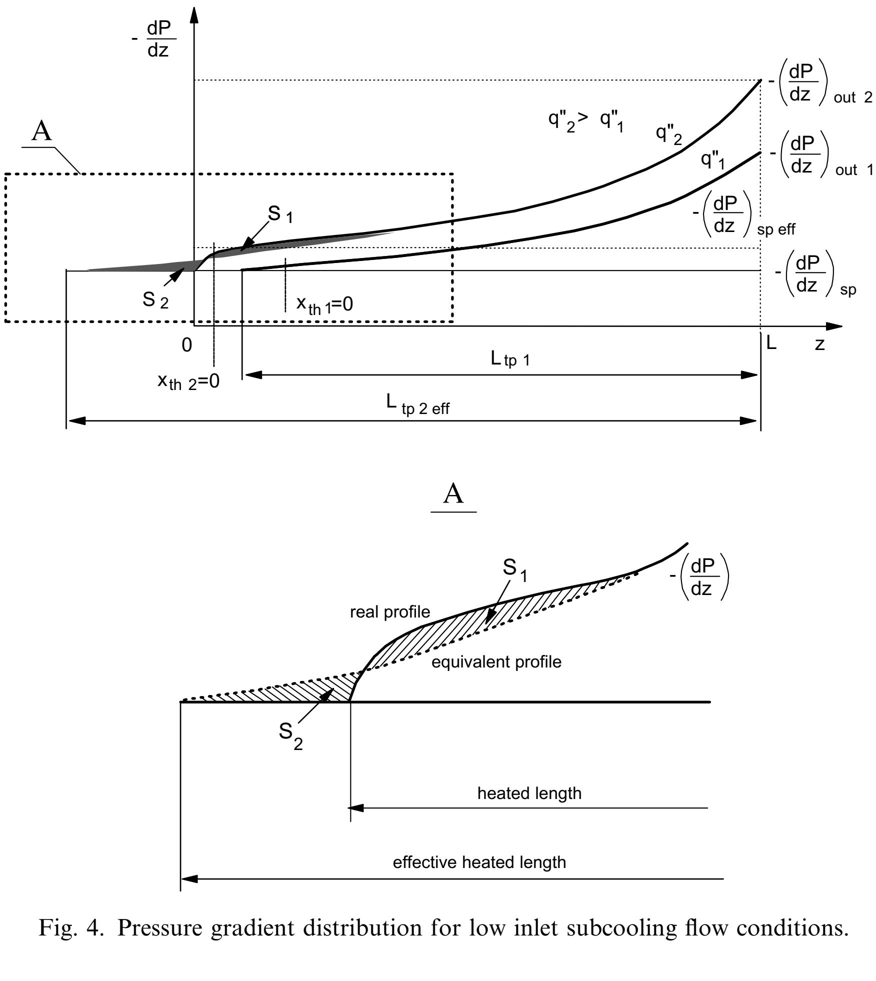

Fig. 4. Pressure gradient distribution for low inlet subcooling flow conditions.

4. Correlation for predicting two-phase pressure drop

A correlation for calculating the total pressure losses must reflect the effect of all the forces (inertia, viscosity and gravitation) exerted on the flow. Therefore, the following non-dimensional parameters have been considered for developing the correlation:

- the Froude number, $Fr_{\mathrm{L}} = v_{\mathrm{L}}^2/gD$, which is a measure of the ratio of the inertial force to the gravitational force; where $v_{\mathrm{L}} = \frac{\dot{m}}{\rho_{\mathrm{L}}}$ is the mean liquid velocity of the single-phase flow and
- the Reynolds number, $Re_{\mathrm{L}} = \dot{m}D/\eta_{\mathrm{L}}$, which is a measure of the ratio of the inertial force to the viscous force,

  with a procedure that allows these non-dimensional numbers, that are valid for single-phase flows, to be adapted to two-phase flows. In addition the following three non-dimensional velocities have also been used:
- the axial vapour velocity referred to the mean liquid velocity of the single-phase liquid flow

$$V_{\mathrm{G}} = \frac{v_{\mathrm{G}}}{v_{\mathrm{L}}} = x\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{G}}}, \tag{11}$$

  where $v_{\mathrm{G}} = \dot{m}x/\rho_{\mathrm{G}}$ is the mean velocity of the vapour flow at the outlet of the heated length assuming that the vapour phase occupies the whole cross-section of the tube;
- the dimensionless vapour generation velocity

$$U_{\mathrm{G}} = \frac{1}{v_{\mathrm{L}}}\frac{q''}{h_{\mathrm{LG}}\rho_{\mathrm{G}}}; \tag{12}$$

- the dimensionless liquid phase transition velocity

$$U_{\mathrm{L}} = \frac{1}{v_{\mathrm{L}}}\frac{q''}{h_{\mathrm{LG}}\rho_{\mathrm{L}}}. \tag{13}$$

It must be noted that a certain number of dimensionless groups containing the ratio $\frac{\eta_{\mathrm{L}}}{\eta_{\mathrm{G}}}$ were also tried, but none of these groups were found to have a significant effect. The final correlation is given by following relationships:

$$\Delta P = \Delta P_{\mathrm{sp}} + \psi_{\mathrm{H}}\frac{L_{\mathrm{tp}}}{D}\frac{\dot{m}^2}{2\rho_{\mathrm{L}}}, \tag{14}$$

$$\psi_{\mathrm{H}} = 6.93\frac{C_k U_{\mathrm{L}}^{0.330}}{Fr_{\mathrm{L}}^{0.067}(Re_{\mathrm{L}}V_{\mathrm{G}})^{0.198}}\left(V_{\mathrm{G}}^{0.721} + V_{\mathrm{G}}U_{\mathrm{G}}^{0.515} - U_{\mathrm{G}}^{0.350}\right), \tag{15}$$

$$C_k = 1 + x^2\frac{\rho_{\mathrm{G}}}{\rho_{\mathrm{L}}}(1 - U_{\mathrm{G}}^{-0.624}), \tag{16}$$

$$L_{\mathrm{tp}} = \frac{\dot{m}Dh_{\mathrm{LG}}(x - x_{0\Delta P})}{4q''}, \tag{17}$$

$$x_{0\Delta P} = -0.367\left(\frac{q''}{\dot{m}h_{\mathrm{LG}}}\right)^{0.470} Re_q^{0.761}. \tag{18}$$

The physical properties of the water and the steam were calculated using the correlations given in Garland and Hand (1989) and Garland et al. (1992). Note that the non-dimensional group $Re_{\mathrm{L}}V_{\mathrm{G}}$ in Eq. (15) represents the Reynolds number for a flow having a velocity equal to that of the steam at the outlet of the heated length, v_{G}, while its viscosity is equal to the kinematic viscosity of the liquid v_{L}:

$$Re_{\mathrm{L}}V_{\mathrm{G}} = Re^* = \frac{v_{\mathrm{G}}D}{v_{\mathrm{L}}}. \tag{19}$$

A comparison between the predicted and measured pressure losses obtained using the aforementioned correlation is given in Fig. 5(a). The absolute and relative errors, and the standard deviation shown in this figure are calculated, respectively as:

$$|\overline{\Delta}| = \frac{1}{N}\sum_{i=1}^{N} |\Delta P_{\mathrm{sp\,predicted}_i} - \Delta P_{\mathrm{sp\,measured}_i}|, \tag{20}$$

$$\overline{\delta} = \frac{1}{N}\sum_{i=1}^{N} \left(\Delta P_{\mathrm{sp\,predicted}_i}/\Delta P_{\mathrm{sp\,measured}_i} - 1\right), \tag{21}$$

$$|\bar{\delta}| = \frac{1}{N}\sum_{i=1}^{N} |\Delta P_{\mathrm{sp\,predicted}_i}/\Delta P_{\mathrm{sp\,measured}_i} - 1|, \tag{22}$$

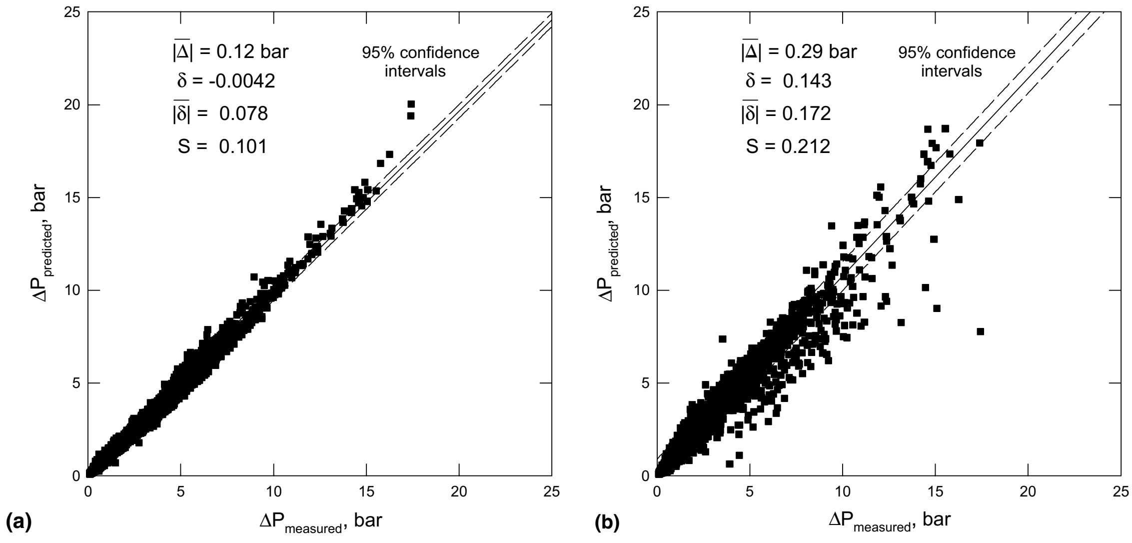

Fig. 5. $\Delta P_{\mathrm{predicted}}$ as a function of $\Delta P_{\mathrm{measured}}$: (a) proposed correlation and (b) correlation of Friedel (1979).

$$S = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(\Delta P_{\mathrm{sp\,predicted}_i}/\Delta P_{\mathrm{sp\,measured}_i} - 1\right)^2}. \tag{23}$$

Fig. 5(b) shows the same data but calculated using the standard method that consists of integrating the pressure distribution along the channel, with Friedel (1979) correlation used for calculating $(\frac{dP}{dz})_f$, and the correlation of Chexal and Lellouche (1986) for determining the void fraction required in $(\frac{dP}{dz})_a$ and $(\frac{dP}{dz})_g$. A comparison of these figures yields the following observations:

- The proposed correlation is able to predict the pressure losses at least twice as well as the conventional method. It must be pointed out that the thermodynamic quality x has been used instead of the mass flow quality $\tilde{x}$ to carry out the calculations based on the conventional method. According to Fig. 5(b), even though x has been used in Friedel's correlation, it overpredicts the pressure losses ($\bar{\delta} = 0.143$). A more physical approach in which $\tilde{x}$ is used to take into account the pressure losses in the subcooled boiling region, will tend to further increase the overpredictions. Then, the comparison will be even more favorable for the present correlation.
- The proposed correlation is characterized by an absolute error of the prediction which is quite small ($|\bar{\Delta}| = 0.12$ bar), even though for low values of ΔP, the relative error can reach several tens of percents. This behavior can be explained by the fact that after reaching an acceptable value of the relative error, only the mean absolute error was minimized during the coefficient adjustment process. It is important to note that most of the data where the correlation shows high relative errors belong to Bertoletti et al. (1964); and as has already been pointed out there are some doubts about their precision. In addition, a large number of the data collected during the present study and having high relative errors, were obtained under very low quality conditions ($x < 0.01$). In such a case a considerable relative error in determining the thermodynamic quality can provoke a huge dispersion in the values of $\Delta P_{\mathrm{predicted}}$. Moreover, for the experiments carried out using short heated lengths under low exit quality conditions, the uncertainties of the surface roughness, i.e., single-phase pressure loss, combined with the method used for determining the two-phase flow pressure drop described in Section 3.1, tend to increase the inaccuracy of the data. However, it was observed that the absolute errors of these data points were always less than ± 0.1 bars which is quite acceptable for any industrial application.
- The data with $\Delta P > 15$ bar for which a slight overestimation of the prediction is observed, are from Olekhnovitch (1997). This overestimation can be related to some lack of precision in the data set itself. It has been mentioned that in this case Friedel's correlation was used to determine the pressure drop at the end of the heated length (a correction term), starting from a pressure measured 94 mm downstream of this region. For a tube of 8 mm ID, a length of 94 mm is rather considerable. In view of the fact that the pressure gradient at the outlet of the test section can be quite high, a relatively small error introduced in this correction term can induce a significant absolute error.

Thus, the proposed correlation represents quite an efficient tool for predicting ΔP, at least within the domain where the maximum number of data points used for its development are concentrated (see Table 1), the precision of the proposed correlation is better than ±20% at 95% confidence. It is obvious that the use of a different correlation for calculating ΔP_{sp}, introducing new dimensionless parameters and enlarging the data base could certainly permit the precision of the correlation to be increased.

5. Conclusions

A new empirical correlation for predicting steam–water two-phase flow pressure drops in uniformly heated vertical round tubes is proposed. For a rather large range of flow parameters, the average precision of the predictions of the proposed correlation is ±8%. It is apparent that this precision is much higher than that obtained by using a conventional method based on the integration of the pressure drop gradient along the channel.

The present correlation allows the total pressure drop to be calculated without the necessity of iterative or integration procedures to be carried out, which is obviously a great advantage. Thus, this correlation can be easily used in applications that require the estimation of the total two-phase pressure drop in real time without requiring a large amount of computational power.

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