Flow Boiling Pressure Drop of R134a in Micro Diameter Tubes: Experimental Results and Assessment of Correlations

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Abstract The experimental results of two phase flow boiling pressure drop of R134a in vertical micro diameter stainless steel tubes are presented in this paper. The tests were conducted using four tubes; one tube with an inner diameter of 0.52 mm and 100 mm heated length and three tubes with an inner diameter of 1.1 mm and different heated lengths (150, 300 and 450 mm). Other experimental conditions include: mass flux range of 200 – 500 kg/m^2 s, system pressure range of 6 – 10 bar, inlet sub-cooling of about 5K and heat flux range of 1 – 140 kW/m^2. The results indicated that the total measured two phase pressure drop increases with increasing mass flux, heat flux (exit quality) and decreasing system pressure and tube inner diameter. The test section heated length was found to have a significant effect on the measured pressure drop per metre length. The total measured two phase pressure drop results were also compared with eighteen macro and micro scale models and correlations.

Keywords: Micro tubes, Flow Boiling, Pressure drop

1. Introduction

The prediction of flow boiling pressure drop in small to micro diameter channels is very crucial for the proper design and operation of compact heat exchangers. These exchangers are required for many applications such as those reported by Venkatesan et al. (2011) which include process control systems, steam power plants, petrochemical plants, refrigeration and air conditioning systems, aircrafts/spacecrafts, air separation plants, chemical process industries, microelectronic cooling systems and research nuclear reactors. Accordingly, many researchers focused on measuring the two phase pressure drop at micro scale and compared their results with the widely used macro scale correlations. However, there are large discrepancies on the comparative performance of these correlations at small scales. For example, some people such as Tran et al. (2000), Zhang and Webb (2001), Kawahara et al. (2002), Qu and Mudawar (2003), Wen et al. (2004), Cavallini et al. (2005), Karayiannis et al. (2008) and Choi et al. (2009) found that the macro scale models and correlations predict poorly the two phase pressure drop. On the contrary, some other researchers such as Zhang and Webb (2001), Lee and Mudawar (2005), Pehlivan et al. (2006), Qi et al. (2007), Agostini et al. (2008) and Kawahara et al. (2009) found that the homogeneous flow model predicts the two phase pressure drop very well.

The success of the homogeneous flow model in some studies and the failure in some others may be attributed to the difference in fluids tested in these studies. Fluid properties, particularly surface tension, may influence the boundaries of isolated bubbles/slug flow regimes in which homogeneous flow model works better. For example, Qi et al. (2007) attributed the success of the homogeneous model in predicting their flow boiling pressure drop data of liquid nitrogen to the effect of surface tension. Liquid nitrogen has a very small surface tension, compared to refrigerants, that allows for the dominance of discrete bubbles up to intermediate quality values and for the dominance of annular mist flow at high qualities.

Another possible reason for the failure of the conventional models and correlations to predict two phase pressure drop at micro scale could be the difference in flow pattern characteristics. For example, Watel (2003) and later Chen et al. (2006) reported that the flow patterns in micro channels are different compared to those observed in macro channels. This difference in flow patterns characteristics may influence the frictional component of pressure drop, which is the largest, and consequently the performance of the macro scale models and correlations. Fukano and Kariyasaki (1993) found that the measured pressure drop in capillary tubes, in the intermittent flow regime (slug/plug), is higher compared to that predicted using large diameter tubes correlations. This was attributed to an additional mechanism other than wall friction that works during intermittent flow in capillary tubes. This mechanism is the sudden expansion occurring when the liquid trapped in the film flows into the liquid slug at the tail of the bubble. The contribution of pressure loss due to this mechanism was found to be more significant in horizontal capillary tubes than vertical ones. Another example that shows the effect of flow patterns on pressure drop characteristics is the experimental study conducted by Wen et al. (2004). They reported high local pressure fluctuations in mini channels due to bubble growth and expansion, which do not exist in macro channels. The local pressure was measured at five axial locations during flow boiling of water in a vertical mini channel (2 × 1 mm). At all locations, the magnitude of fluctuations in the measured pressure was almost similar (5kPa), which was comparable to or greater than the time averaged pressure drop measured across the channel.

Many researchers investigated the effect of parameters such as heat flux (exit quality), mass flux, system pressure, inlet sub-cooling, and channel diameter on the flow boiling pressure drop in small to micro diameter channels. Tong et al. (1997) measured pressure drop of water flow boiling in four tubes of 1.05, 1.38, 1.8 and 2.44 mm diameters. It was found that, the pressure drop increases continuously with increasing heat flux and mass flux due to the enhancement in wall shear stress. Increasing inlet sub-cooling (decreasing liquid inlet temperature) resulted in a decrease in pressure drop due to the stronger condensation in the channel core. Additionally, decreasing tube diameter was found to increase pressure drop. This was attributed to the fact that at a given mass flux the boundary layer becomes thinner as the diameter decreases, which results in higher velocity gradients leading to higher shear stress and in turn higher pressure drop. Yan and Lin (1998) investigated the effect of mass flux, heat flux and saturation pressure on the measured pressure drop of R134a in a 2 mm diameter tube. It was found that the pressure drop increased almost linearly with increasing vapour quality and mass flux. At low mass fluxes, the slope of the ΔP-q curve was very small, i.e. pressure drop increases slowly, while the slope of the curve significantly increased with increasing mass flux. Increasing saturation pressure resulted in a decrease in the measured pressure drop especially for exit qualities greater than 0.65 while the effect was very small for qualities less than this value. Tran et al. (2000) measured flow boiling pressure drop for three refrigerants R134a, R12 and R113 using two circular test sections of 2.46 and 2.92 mm diameters and one rectangular channel of 2.4 mm hydraulic diameter. It was found that the measured two phase pressure drop increases with increasing exit quality, mass flux and decreasing system pressure. For R134a when the system pressure was decreased by about 47 %, the pressure drop increased by about 60 % and also for R12 the pressure drop increased by about 70 % when the system pressure was decreased by about 37 %. Huo et al. (2007) and Karayiannis et al. (2008) investigated flow boiling pressure drop of R134a in five tubes of 4.26, 2.88, 2.01, 1.1 and 0.52 mm diameters. It was found that, the pressure drop increases almost linearly with heat flux and sometimes a decrease in pressure drop was observed at high heat flux values due to the occurrence of intermittent dryout. Also, it was found that increasing mass flux and decreasing system pressure resulted in an increase in the measured pressure drop. The effect of pressure was attributed to the reduction in liquid to vapour density ratio as the pressure increases. Decreasing tube diameter was found to increase pressure drop though the effect was less clear at high exit quality for the three larger tubes (4.26 – 2.01 mm). When the diameter was decreased from 2.01 to 1.1 mm, a significant increase in pressure drop was found while the pressure drop of the 0.52 mm tube was much higher. Note that the single phase friction factor for this smaller tube was found to be very high - more than twice the value predicted by laminar and Blasius equations. Analyzing the inner surface of this tube showed that the cross section was not uniform along the tube, which may explain the very high two phase pressure drop in this tube. This size of tube was to be re-examined in the present paper. Additionally, Karayiannis et al. (2008) reported that the frictional pressure drop component was the dominant and the acceleration term is not negligible; it increases with increasing exit quality.

The above review indicates that there is an agreement on the effect of mass flux, heat flux (exit quality), system pressure, and channel diameter on the measured two phase pressure drop. However, there are still discrepancies about the performance of macro and micro scale pressure drop prediction methods. Accordingly, more experiments are required such as the current experimental study. The current paper presents the experimental results of two phase pressure drop measured in vertical stainless steel tubes with diameters of 0.52 and 1.1 mm. The effect of heat flux, mass flux and system pressure was investigated. Also, the effect of evaporator length was investigated for the 1.1 mm tube, which, to the best of our knowledge, was not studied before or reported in the open literature. Additionally, the results are compared with some of the existing macro and micro scale models and correlations.

2. Pressure Drop Models and Correlations

Generally, the two phase pressure drop consists of three components namely; frictional, gravitational and acceleration components as given by Eq. (1).

$$\left(\frac{dP}{dz}\right)_{tp} = \left(\frac{dP}{dz}\right)_{f} + \left(\frac{dP}{dz}\right)_{g} + \left(\frac{dP}{dz}\right)_{acc} \tag{1}$$

The widely used approaches to calculate the three terms of the right hand side of Eq. (1) are based on either the homogeneous flow model or the separated flow model. The homogeneous flow model assumes that the two phases travel with the same velocity (no slip) and the acceleration, gravitational and frictional components are given for vertical tubes as:

$$\Delta P_{acc} = \frac{G^2 x_e}{\rho_L}\left(\frac{\rho_{Lv}}{\rho_v}\right) \tag{2}$$

$$\Delta P_g = \frac{\rho_L g L_{tp} \rho_v}{\rho_{Lv} x_e} \ln\left[1 + x_e \frac{\rho_{Lv}}{\rho_v}\right] \tag{3}$$

$$\Delta P_f = \frac{2 f_{Lo} L_{tp} G^2}{D \rho_L}\left[1 + \frac{x_e}{2}\left(\frac{\rho_{Lv}}{\rho_v}\right)\right] \tag{4}$$

The separated flow model assumes that the two phases are segregated and each phase travels with its own velocity. The gravitational and acceleration components require the knowledge of void fraction (α) and they are given as:

$$\Delta P_g = g \frac{L_{tp}}{x_e} \int_0^{x_e} \left[\alpha \rho_v + (1-\alpha)\rho_L\right] dx \tag{5}$$

$$\Delta P_{acc} = \frac{G^2}{\rho_L}\left[\frac{x_e^2}{\alpha_e}\frac{\rho_L}{\rho_v} + \frac{(1-x_e)^2}{(1-\alpha_e)} - 1\right] \tag{6}$$

There are many correlations for the void fraction (α) in the open literature, see Woldesemayat and Ghajar (2007). The frictional component is a complex term and is usually determined using the principle of the two phase frictional multiplier proposed by Lockhart and Martinelli (1949). It accounts for the friction coming from the interaction between the phases. Based on that, the two phase frictional pressure gradient was given as:

$$\left(\frac{dP}{dz}\right)_f = \phi_L^2\left(\frac{dP}{dz}\right)_L \tag{7}$$

$$\phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{8}$$

The Chisholm constant (C) in Eq. (8) takes the values from 5 – 20 depending on whether the regimes of the gas and liquid are laminar or turbulent. After the work of Lockhart and Martinelli (1949), several correlations were proposed for predicting two phase pressure drop in large diameter tubes such as those of Chisholm (1973), Friedel (1979) and Muller-Steinhagen and Heck (1986). The details of these correlations can be found in Collier and Thome (1994).

In an attempt to develop new pressure drop correlations for micro scale applications, most researchers modified the Chisholm constant (C) in Eq. (8) to consider the micro scale effects. Mishima and Hibiki (1996) measured the two phase pressure drop for air-water flow in capillary tubes with diameters ranging from 1 to 4 mm. They have found that the Chisholm constant depends on tube inner diameter where it decreases when the diameter decreases. They proposed a correlation for this parameter given by Eq. (9), which is applicable for vertical, horizontal, circular and rectangular channels.

$$C = 21\left(1 - e^{-0.319 d_h}\right) \tag{9}$$

The hydraulic diameter d_h in the above equation should be expressed in mm. Lee and Lee (2001) investigated two phase flow pressure drop of air-water flow through narrow rectangular channels. The dimensions of the channels were 20 mm width, 640 mm length, and different depths of 0.4, 1, 2, and 4 mm. They proposed a correlation for the Chisholm constant as a function of the all liquid Reynolds number and two other dimensionless groups to account for the effects of surface tension, viscosity and liquid slug velocity as given by Eq. (10).

$$C = A\lambda^q \psi^r \mathrm{Re}_{Lo}^s \tag{10}$$

where: $\psi = \dfrac{\mu_L j_L}{\sigma}$, $\lambda = \dfrac{\mu_L^2}{\rho_L \sigma d_h}$, $\mathrm{Re}_{Lo} = \dfrac{GD}{\mu_L}$

The constant A and the exponents in Eq. (10) depend on whether the liquid and gas are laminar or turbulent. A = 6.833 × 10^{-8}, q = -1.317, r = 0.719, s = 0.557 for laminar liquid/laminar gas, A = 0.06185, q = 0, r = 0, s = 0.726 for laminar liquid/turbulent gas, A = 3.627, q = 0, r = 0, s = 0.174 for turbulent liquid/laminar gas, A = 0.408, q = 0, r = 0, s = 0.451 for turbulent liquid/turbulent gas. Warrier et al. (2002) investigated flow boiling of FC-84 in an aluminum multi-channel heat sink with hydraulic diameter of 0.75 mm and heated length of 307.4 mm. They proposed using a fixed value of 38 for the Chisholm constant irrespective of the flow regimes. This was attributed to the rapid development into annular flow in narrow channels with liquid mostly confined to the corners of the rectangular channel. It is worth mentioning that, some of the physical properties of the FC-84 vapour were not known such as density and viscosity. The density of the vapour was calculated from the perfect gas law and the viscosity was taken as that of R113.

Qu and Mudawar (2003) investigated flow boiling pressure drop of water in a multi-channel heat sink with hydraulic diameter of 0.35 mm. They modified the correlation of Mishima and Hibiki (1996) defined by Eq. (9) in order to take into account the mass flux effect as given by:

$$C = 21\left(1 - e^{-0.319 d_h}\right)\left(0.00418G + 0.0613\right) \tag{11}$$

It is worth noting that the units of d_h and G in the above equation are mm and kg/m^2 s respectively. Lee and Mudawar (2005) measured the two phase pressure drop of R134a in a multi-channel heat sink with channel dimensions of 231 × 713 µm. Based on the premise that slug and annular flow regimes are the most dominant regimes at high heat fluxes, they proposed a new correlation following the separated flow model by modifying the Chisholm constant. The new constant takes into account the effects of liquid inertia, liquid viscous force and liquid surface tension and it was expressed as:

$$C = \begin{cases} 2.16\,\mathrm{Re}_{Lo}^{0.047}\,We_{Lo}^{0.6} & laminar\ liquid - laminar\ gas \\ 1.45\,\mathrm{Re}_{Lo}^{0.25}\,We_{Lo}^{0.23} & turbulent\ liquid - laminar\ gas \end{cases} \tag{12}$$

Lee and Garimella (2008) investigated flow boiling heat transfer and pressure drop of deionised water in parallel micro channels with fixed channel depth of 400 μm and channel width ranging from 102 to 997 μm. They proposed a correlation for the Chisholm constant in the laminar liquid/laminar vapour regime such that it takes into account the effect of the hydraulic diameter and mass flux as given by Eq. (13).

$$C = 2566 G^{0.5466} d_h^{0.8819} (1 - e^{-319 d_h}) \tag{13}$$

Contrary to Eq. (9) and Eq. (11), d_h in Eq. (13) is expressed in m. Zhang et al. (2010) conducted an extensive analysis using artificial neural networks (ANN) to find the most important parameters that significantly influence the frictional pressure drop. They have found that mass flux, system pressure and vapour quality have minor effects whereas the hydraulic diameter and the dimensionless Laplace constant (La) defined below have the strongest effects. Based on that, they modified the correlation of Mishima and Hibiki (1996) and proposed the following correlation for the Chisholm constant in the laminar liquid and laminar vapour regimes. This correlation is valid for channels with diameters ranging from 0.014 to 6.25 mm.

$$C = 21\left(1 - \exp\left(-\frac{0.358}{La}\right)\right) \tag{14}$$

where $La = \dfrac{\left[\dfrac{\sigma}{g(\rho_L - \rho_v)}\right]^{0.5}}{D}$

Li and Wu (2010) collected a large number of experimental data from the open literature for adiabatic two phase flow in micro/mini-channels and proposed a pressure drop correlation using regression analysis. They then modified the Chisholm constant in the two phase multiplier using the Bond number and the Reynolds number. The proposed new correlation was given as:

$$C = \begin{cases} 11.9 Bd^{0.45} & Bd \le 1.5 \\ 109.4\left(Bd\,\mathrm{Re}^{0.5}\right)^{-0.56} & 1.5 < Bd \le 11 \end{cases} \tag{15}$$

Lee et al. (2010) collected 484 data points for flow boiling pressure drop in mini/micro channels with diameters < 3 mm and covered seven working fluids; water, n-pentane, ammonia, CO_2, R410a, R134a and R12. It is worth mentioning that, most of the data were for rectangular multi-channel configurations. These data were collected from nine different experimental studies. They found that, the Chisholm constant depends on the Bond number and the exit quality. As the Bond number increases, the Chisholm constant was found to increase until it reaches an asymptotic value of about 30 at Bond number values greater than about 2. They attributed this to the presence of large number of bubbles at high values of the Bond number and consequently high interfacial interactions between the phases. On the other hand, at small values of the Bond number, one confined bubble fills the channel with less interaction between the phases. As the exit quality increases, the Chisholm constant was found to increase rapidly according to a polynomial trend where its value exceeded 100, which is much greater than the original values in the Lockhart-Martinelli model. The new proposed correlation was given as:

$$C = 121.6(1 - e^{-22.7 Bd}) x_e^{1.85} \tag{16}$$

Some researchers proposed correlations for the two phase frictional multiplier rather than modifying the Chisholm constant. Tran et al. (2000) investigated flow boiling of three refrigerants (R134a, R12, R113) in two circular channels with diameters of 2.46 and

2.92 mm and a rectangular channel with hydraulic diameter of 2.4 mm. They have found that the macro scale correlations could not predict their measured two phase pressure drop. They believed that the failure of these correlations was due to an additional frictional component resulting from the deformation of the elongated bubbles in small diameter channels, which makes the pressure drop in micro channels too high. Accordingly, they proposed a correlation for the two phase frictional multiplier based on 610 data points. Their correlation was basically a modified version of the Chisholm (1983) B-coefficient correlation. They replaced the coefficient (*B)* in the original correlation with the confinement number (*Co*) proposed by Kew and Cornwell (1997) to account for the effect of tube diameter and surface tension as given by:

$$\phi_{Lo}^2 = 1 + \left(4.3\Gamma^2 - 1\right)\left[\left(Co x^{0.875}(1-x)^{0.875} + x^{1.75}\right)\right] \quad (17)$$

where $\Gamma^2 = \left[\frac{\left(dP/dz\right)_{go}}{\left(dP/dz\right)_{Lo}}\right]$

Yu et al. (2002) investigated flow boiling heat transfer and pressure drop of water in a 2.98 mm diameter tube with 910 mm heated length. In their study, the flow regimes of the liquid and vapour flowing alone were laminar and turbulent respectively. They compared the experimental two phase frictional multiplier against the classical Lockhart-Martinelli multiplier and found that the correlation over-predicted the experimental values. They attributed this to the occurrence of slug flow over a wide range of qualities in small diameter tubes, which reduces the pressure gradients compared to the annular flow regime that dominates in large diameter tubes. This explanation may be suitable for refrigerants but in their study they used water and the present authors believe that annular flow develops at very small quality values, i.e. annular flow regime dominates. Due to this overestimation, they fitted their experimental two phase frictional multiplier data as a power function given by:

$$\phi_L^2 = \frac{1}{X^{1.9}} \quad (18)$$

Garimella et al. (2002) developed a model for intermittent slug flow during condensation of R134a in horizontal micro channels of hydraulic diameters ranging from 0.5 to 4.91mm. Although this model was developed for condensation, it is included here to see whether there is a big difference or not between pressure drop in slug flow during condensation and evaporation. They selected a control volume as a unit cell consisting of a cylindrical bubble surrounded by a uniform liquid film and followed by a moving liquid slug. In the liquid slug, the frictional pressure gradient was calculated using the Blasius equation for friction factor and using slug velocity. In the bubble/film region, the bubble velocity was calculated using the drift flux model with zero drift velocity and $U_B / U_{LS} = 1.2$. Also, in the bubble/film region, the pressure gradient was calculated using the Blasius equation and the relative velocity between the bubble and the interface. Interface velocity was obtained by solving the momentum equation in the film and the bubble iteratively. Additionally, they accounted for the pressure loss arising from liquid film flowing into the liquid slug, which was called transition pressure losses. These losses were calculated based on the relative velocity between the liquid slug and film interfacial velocity. Finally, the relative lengths of each zone in the cell were calculated using expressions that were given by Fukano et al. (1989). The required frequency was correlated as a function of the liquid slug Reynolds number. This model was found to capture 88 % of the data within 25 % error bands with mean absolute error of 13.4 %. However, they reported that the slug frequency is a weak point that needs further modeling in order to correlate it with flow conditions, channel diameter and fluid properties. Furthermore, the contribution of each zone to the total pressure drop was found to be: 53 % was resulting from the liquid slug, 12 % from the bubble-film region and 35 % from the transition losses.

Shiferaw et al. (2009) developed a semi-mechanistic model for slug flow based on the 3-zone heat transfer model that was proposed by Thome et al. (2004) with the assumption of smooth liquid film. In this model the total pressure drop was calculated by time averaging the pressure drop for single phase liquid, elongated bubble with a thin liquid film and single phase vapour. The model predicted the flow boiling data of R134a in vertical tubes with diameters of 4.26, 2.88, 2.01, 1.1and 0.52 mm with mean errors 23, 20, 16.8, 16.3 and 22 %, respectively. Also, the effect of film waviness for a rough liquid film was assessed by arbitrarily increasing the friction coefficient and showed significant effect on pressure drop and should be taken into consideration.

3. Experimental Facility

The detailed description of the experimental facility used in the present study can be found in Shiferaw et al. (2009). It was designed to investigate flow boiling heat transfer, pressure drop and flow patterns using a wide range of tube diameters and test fluid. In this paper, only pressure drop results are presented for R134a. Four stainless steel tubes were used in the tests described in this paper. The first tube has an inner diameter of 0.52 mm and 100 mm heated length whilst the other three tubes have an inner diameter of 1.1 mm and different heated lengths (150, 300 and 450 mm). All tubes are manufactured by the cold drawn process and each test section consists of an adiabatic calming section, heated section and a borosilicate visualization section. The schematic drawing of the test section can be found in Mahmoud et al. (2011). The heated section was directly heated by passing a DC current through two copper electrodes that were welded at the inlet and outlet. The supplied power was directly measured between the test section electrodes using a Yokogawa power meter WT110 with an accuracy of ± 0.29 %. This is to exclude the voltage drop across the connections between the power supply and the test section electrodes. The outer surface temperature was measured locally by K-type thermocouples with a mean absolute error value of ± 0.22K. The first and last thermocouples were located away from the electrodes to avoid the effect of heat losses at the electrodes. All thermocouples were attached to the surface by using an electrically insulating but thermally conducting epoxy. Fluid temperature and pressure were measured at the test section inlet and outlet using T-type thermocouples with an accuracy of ± 0.1K and pressure transducers with an accuracy of ± 0.32 % respectively. The pressure drop was directly measured between the test section inlet and outlet using a differential pressure transducer (Omega PX771A-025DI) with an accuracy of ± 0.1 %. A Phantom V4 digital high speed camera with 1000 frame/s and resolution of 512 × 512 pixels was used for flow visualization. The data were monitored through a Labview program at frequency of 1Hz using three data loggers: Solartron models SI35951E (two) and SI35351C. All the data were recorded for 90 sec after attaining steady state and the sample of the 90 data points was averaged and used in the calculations.

Single phase validation was conducted first for each tube and the results were published in Mahmoud et al. (2011). The validation indicated that the single phase friction factor agrees well with the laminar flow theory and Blasius equation. It is worth mentioning here that the test sections in the current study do not have any inlet or outlet restrictions. In other words, the total flow boiling pressure drop was directly measured across the heated section without any inlet or outlet pressure losses. Also, since the flow enters the heated section as a sub-cooled liquid, the single phase pressure drop part was subtracted from the total measured value; see Mahmoud et al. (2011). Additionally, the pressure drop results presented here are corresponding to stable boiling conditions; see Karayiannis et al. (2011) and Mahmoud et al. (2011).

4. Experimental Results and Discussion

Figures 1 – 3 depict the effect of heat and mass fluxes, system pressure and tube diameter on the total measured two phase pressure drop for D = 1.1 mm and L= 150 mm. Similar figures were obtained for the other test sections described in this paper. It is obvious from Fig. 1 that the total measured two phase pressure drop increases almost linearly with increasing heat flux (exit quality) due to the increase in the contribution of the acceleration pressure drop component as the vapour quality increases. The same figure demonstrates also that the measured pressure drop increases with increasing mass flux with higher mass flux effect at high heat flux values compared to the low heat flux values. This mass flux effect can be explained based on the momentum equation where the frictional and acceleration pressure gradients are proportional to the term G^2 but depend also on quality; see Eqs. (4) and (6). At a fixed heat flux value, increasing the mass flux, results in a significant reduction in the exit quality value, which leads to a reduction in the frictional and acceleration pressure drop components. Accordingly, at a fixed low heat flux value, the exit quality becomes very small (approaches zero) as mass flux increases which approximately cancel the effect of the G^2 term. This fact may explain the very small mass flux effect at low heat flux values shown in Fig.1. At fixed high heat flux values, the effect of the G^2 term overcomes the effect of quality. Therefore, the mass flux effect becomes obvious at high heat fluxes.

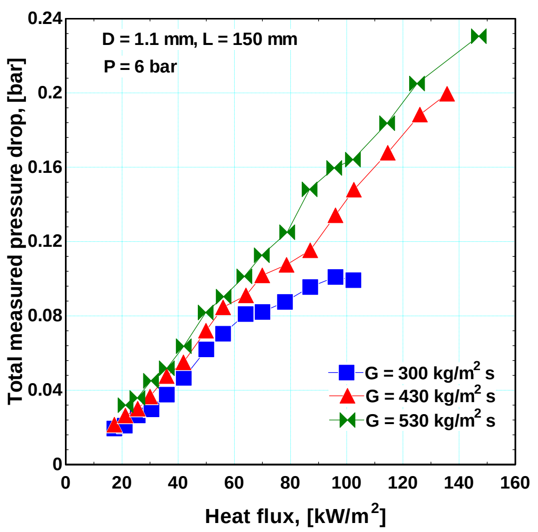


Figure 1 The total measured two phase pressure drop versus heat flux at different mass fluxes for D = 1.1 mm, L = 150 mm.

Figure 2 shows the effect of system pressure on the total measured pressure drop. The figure shows that the measured pressure drop decreases with increasing system pressure due to the effect of pressure on fluid properties. For R134a, as the pressure increases, the liquid density decreases slightly whereas the vapour density increases significantly. For example, increasing the pressure from 6 to 10 bar resulted in 6 % reduction in the liquid density and 69 % increase in the vapour density. This means that at a given mass flux value, the vapour superficial velocity will decrease significantly as the pressure increases resulting in a reduction in the acceleration component of the pressure drop. Additionally, the increase in system pressure reduces the liquid and vapour viscosities and consequently the frictional component of the pressure drop. The effect of fluid properties on pressure drop can be quantified using the homogeneous flow model (for example) at G = 300 kg/m^2 s, D = 1.1 mm, L = 0.15 mm, x = 0.5 when the pressure increases from 6 – 10 bar as follows: As the pressure increased from 6 – 10 bar, the liquid to vapour density ratio decreased by about 44 % and the liquid to vapour viscosity ratio decreased by about 25.5 %. This resulted in a decrease in the acceleration component by about 42 %, a decrease in the frictional component by about 50 % and an increase in the gravitational component by about 40.5 %.

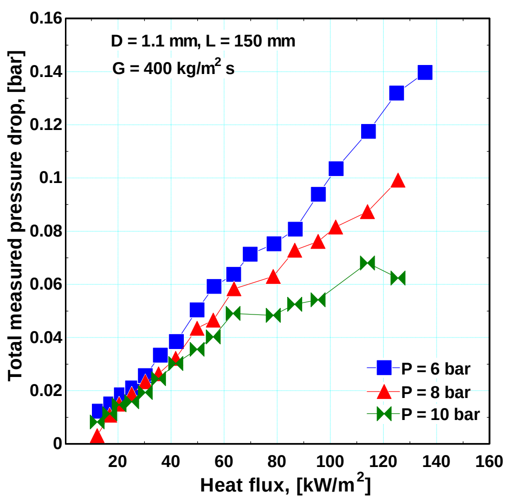


Figure 2 The effect of system pressure on the total measured two phase pressure drop at G = 400 kg/m^2 s for D = 1.1 mm, L = 150 mm.

Figure 3 depicts the effect of tube diameter on the total measured two phase pressure drop per unit length at 6 bar system pressure and 400 kg/m^2 s mass flux. It is obvious that as the diameter was decreased from 1.1 to 0.52 mm (about 50 % reduction), the measured pressure drop increased by about 300 %. The increase in pressure drop as the diameter decreases may be attributed to the large velocity gradient in the boundary layer next to the wall arising from the thinning of the liquid film as the diameter decreases. This may significantly increase the wall shear stress and consequently the frictional pressure drop.

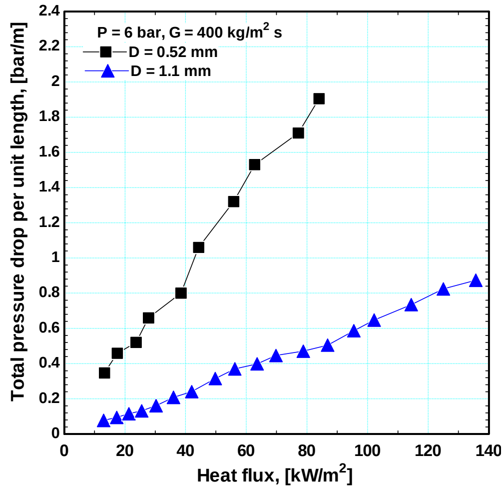
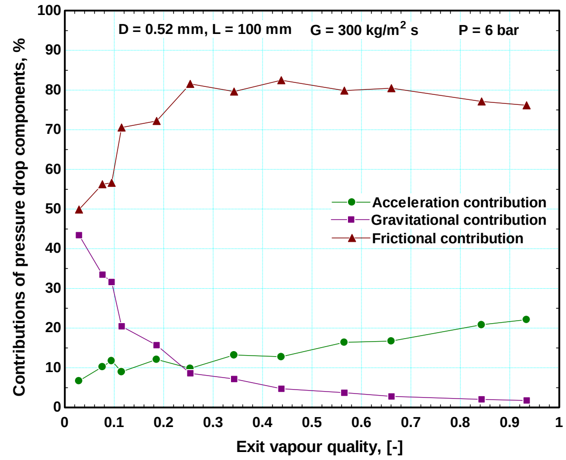
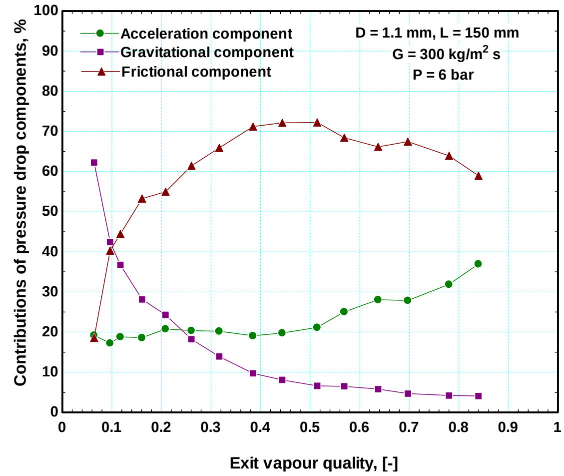


Figure 3 The effect of tube diameter on the total measured pressure drop at G = 400 kg/m^2 s and P = 6 bar.

Figure 4 illustrates the contributions of the frictional, acceleration and gravitational components to the total measured pressure drop at 6 bar system pressure and 300 kg/m^2 s mass flux. The void fraction correlation of Lockhart-Martinelli (1949) was used to calculate the gravitational and acceleration components presented in Fig. 4. The figure indicates that the contribution of the frictional component is the largest. For the 0.52 mm tube, Fig. 4a shows that the contribution of the gravitational component decreased from about 43 to 2 % whereas the contribution of the acceleration component increased from about 7 to 22 %. The contribution of the frictional component increased rapidly from about 50 to about 80 % up to an exit quality value of about 0.25 after which it remained approximately constant with a tendency to decrease at very high exit quality. For the 1.1 mm tube, Fig. 4b shows similar behaviour to that found in the 0.52 mm tube. The gravitational contribution



(a)



(b)

Figure 4 The contributions of the three pressure drop components to the total measured pressure drop at G = 300 kg/m^2 s and p = 6 bar for (a) D = 0.52 mm and (b) D = 1.1 mm.

dropped rapidly from about 60 % to less than 5 % whereas the acceleration contribution increased from about 19 to 37 %. The high gravitational contribution at the lowest exit quality observed in the 1.1 mm tube compared to the 0.52 mm is due to the longer heated length. The heated length of the 0.52 mm tube is 100 mm while that of the 1.1 mm tube is 150 mm. The frictional contribution increased rapidly from about 18 to 70 % up to exit quality value of about 0.4 after which it remained constant over a narrow range of quality before it decreases at high exit quality values. It is obvious also from Figs. 4a and 4b that as the diameter decreases, the contribution of the frictional component increases which may support the previous explanation of the boundary layer thinning.

Figure 5 depicts the effect of the heated length on the total measured pressure drop per unit length for D = 1.1 mm, P = 6 bar and G = 500 kg/m^2 s. Each curve in Fig. 5 represents the slope of the *P(z)* line at each exit quality value. In other words, each point was calculated as the total measured pressured drop divided by the heated length of the tube. It is clear from Fig. 5 that, at the same exit quality value, the total measured pressure drop per unit length increases as the heated length decreases. This means that the slope of the *P(z)* line in the shortest tube is much higher than that in the longer tubes for the same exit quality. This behaviour may be explained as follows: at the same exit quality, the applied

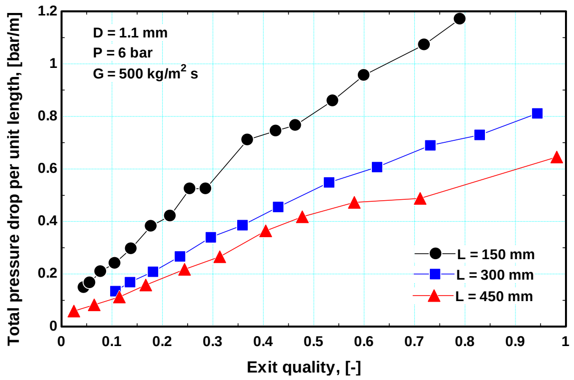


Figure 5 The effect of tube heated length on the total measured pressure drop per unit length for D = 1.1 mm at P = 6 bar and G = 500 kg/m^2 s

heat flux in the shortest tube is much higher than that in the longer tubes. As a result of that high heat flux (high wall superheat), nucleate boiling was the dominant heat transfer mechanism in the shortest tube; see Karayiannis et al. (2011). Accordingly, the possibility of having small nucleating bubbles next to the wall in the shortest tube may increase the frictional pressure drop due to the additional interaction between the phases in the liquid film region next to the wall. This point is clarified further by plotting the frictional pressure drop per unit length versus exit quality in Fig. 6. It is worth mentioning here that the total measured pressure drop (bar) was found to increase with increasing the tube heated length, which is expected.

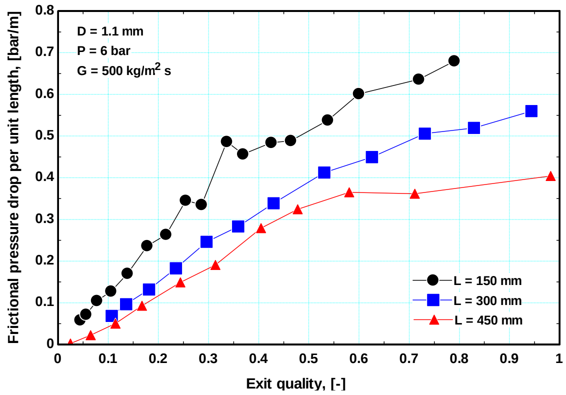


Figure 6 The frictional pressure drop per unit length versus exit quality for the three heated lengths for D = 1.1 mm, G = 500 kg/m^2 s and P = 6 bar.

5. Assessment of Models and Correlations

This section compares 541 experimental data points with the macro and micro scale models and correlations presented in section 2. The data include those of the 0.52 mm and 1.1 mm diameter tubes at system pressure of 6 – 10 bar, mass flux range of 200 – 500 kg/m^2 s and exit quality up to about 0.9. It is worth mentioning that, each time, the void fraction correlation that was used in the original pressure drop model with which the current data are compared was also used here. The assessment was conducted based on the mean absolute error (*MAE*) defined by Eq. (19) and the percentage of data within ± 30 % error bands (β).

$$MAE = \frac{1}{N}\sum_{i=1}^{N}\left[\frac{\left|\Delta P_{measured} - \Delta P_{\Pr edicted}\right|}{\Delta P_{measured}}\right]_i \qquad (19)$$

Table 1 summarizes the *MAE* and β values for each model and correlation. The table demonstrates that the macro scale correlation of Muller-Steinhagen and Heck (1986) predicts the data of all tubes very well with

MAE/β values of 16.9/89.3 % as seen in Fig. 7. The success of the Muller-Steinhagen and Heck (1986) correlation in predicting the current data may be attributed to the fact that the correlation was developed based on a wide range of refrigerants. Additionally, the correlation was formulated such that the two phase pressure drop reduces to single phase liquid pressure drop at x = 0 and single phase gas pressure drop at x = 1. Ribatski et al. (2006) assessed this correlation at micro scale and reported similar performance. The table indicates also that the second ranked model is the homogenous flow model with *MAE/β* values of 19.5/80.6 % as shown in Fig. 8. The reasonable prediction using the homogeneous flow model may indicate that there is a good mixing between the phases and thus the slip ratio (U_v/U_L) in micro tubes approaches unity. Serizawa et al. (2002), Hayashi et al. (2007) and Saisorn and Wongwises (2008) found that the measured void fraction in micro tubes agrees well with the homogeneous void fraction, i.e. slip ratio ≈ 1.

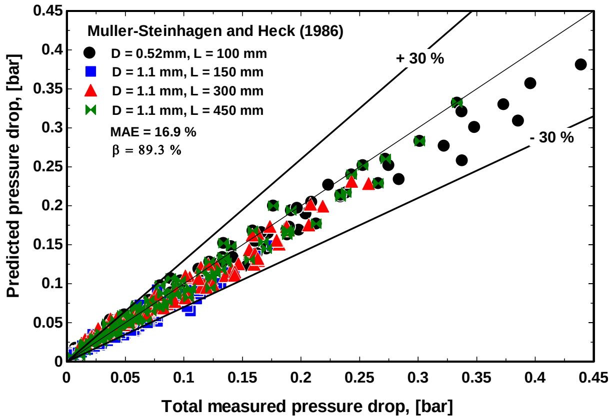


Figure 7 The comparison with the Muller-Steinhagen and Heck correlation (1986).

Table 1 The mean absolute error value (MAE) and the percentage of data within ± 30 % (β) for each model and correlation.

Model/correlation	D = 0.52 mm		D = 1.1 mm					
			L = 150 mm		L = 300 mm		L = 450 mm	
	MAE %	β %	MAE %	β %	MAE %	β %	MAE %	β %
Homogeneous model	18.5	83.4	20.5	80	19.9	79.5	18.7	80.6
Lockhart-Martinelli (1949)	16.5	85.4	32.2	56	51	38.5	60.6	24.5
Chisholm (1973)	67.4	15.9	56	18.8	81.9	3.3	95.8	2
Friedel (1979)	20.6	69	21.5	79	28.5	63	31.9	60
Muller-Steinhagen and Heck (1986)	13.7	93.4	21	88.2	18.2	81.1	20	78.6
Mishima and Hibiki (1996)	24.5	69	24.2	77	21.8	83.6	19.8	83.7
Tran et al. (2000)	168	0.7	68.5	16.5	105.8	0	121	4
Modified Tran et al. (2000)	23.1	74	22.3	86	20.8	80	22.9	76.5
Yu et al. (2000)	45.5	27.8	40.4	38.2	53.7	30.3	57.5	30.6
Lee and Lee (2001)	30.5	52.3	56.9	25.9	75.7	17	78.7	14
Warrier et al. (2002)	198.4	0.7	175	0	251	0	264	1
Qu and Mudawar (2003)	63.2	41.7	45.5	45.9	62	32.8	73.3	16.3
Lee and Mudawar (2005)	43.4	31.8	70.1	8.8	101.4	2.4	111.7	14
Lee and Garimella (2008)	20.3	74.8	51.2	44	64.8	31	78.3	15.3
Zhang et al. (2010)	30.2	45	22.9	81.2	23.8	72	24	71.4
Li and Wu (2010)	15.9	92	26.5	76.5	31.4	64.7	35.2	48
Lee et al. (2010)	152.5	25	130	34	229	23.8	227	17.3
Garimella et al. (2002)	71.8	14	29	69	69.6	17.8	74.8	18
Shiferaw et al. (2009)	22.8	70.8	20.9	81.8	21.6	74.5	24.8	64

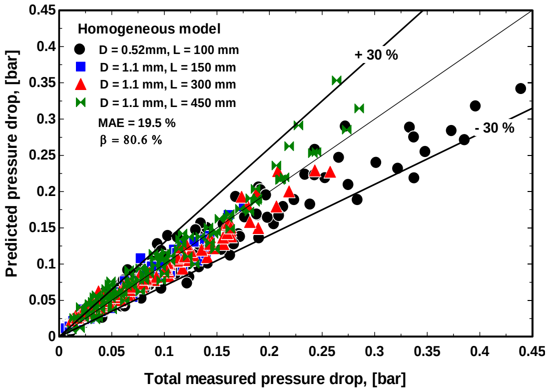


Figure 8 The comparison with the homogeneous flow model.

The third best correlation is the modified Tran et al. (2000) correlation with MAE/β values of 22.3/79.1 %. It is worth noting that, the original correlation of Tran et al. (2000) is the second worst predicting correlation as summarized in Table 1 and shown in Fig. 9. The figure indicates that the Tran et al. (2000) correlation has a fixed bias that makes the correlation predict poorly the current experimental data. The reason why the correlation highly over-predicts the current experimental data may be due to the fact that they did not investigate a wide range of tube diameters to determine the proper exponent of the confinement number and the constant 4.3 in their correlation given by Eq. (17) in this paper. They only investigated one diameter value of 2.92 mm at which the confinement number value is about 0.25 for R134a at 6 bar system pressure. It is interesting to note that, when the constant 4.3 was changed in the current study to 1.75, the correlation performed much better compared to the original version of the correlation. Figure 10 shows the global comparison between the current experimental data and the modified Tran et al. correlation.

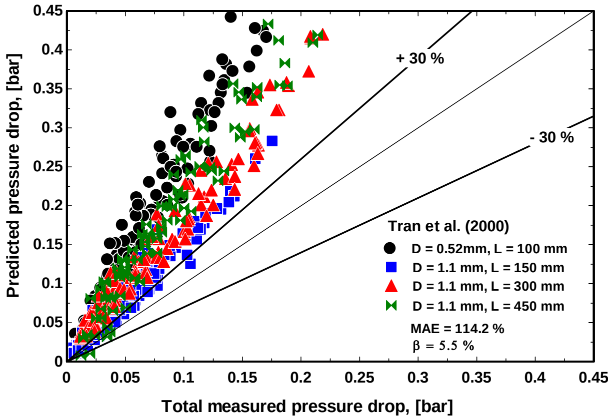


Figure 9 The comparison with the Tran et al. (2000) correlation.

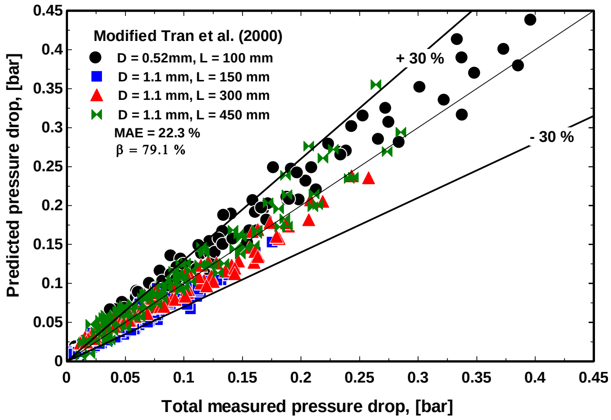


Figure 10 The comparison with the Tran et al. (2000) correlation modified by the present authors by changing the constant c in Eq. (17) in the original paper from 4.3 to 1.75.

The fourth best correlation is the Mishima and Hibiki (1996) correlation with MAE/β values of 22.9/77.3 % as depicted in Fig. 11. Table 1 indicates also that the correlation of Friedel (1979), Li and Wu (2010) and the model of Shiferaw et al. (2009) give reasonable predictions with MAE values less than 30 %. The correlations of Lockhart and Martinelli (1949) and Chisholm (1973) over-predicted the current data with MAE greater than 30 %. It is worth noting from Table 1 that the performance of most micro scale correlations is worse than that of the macro scale correlations.

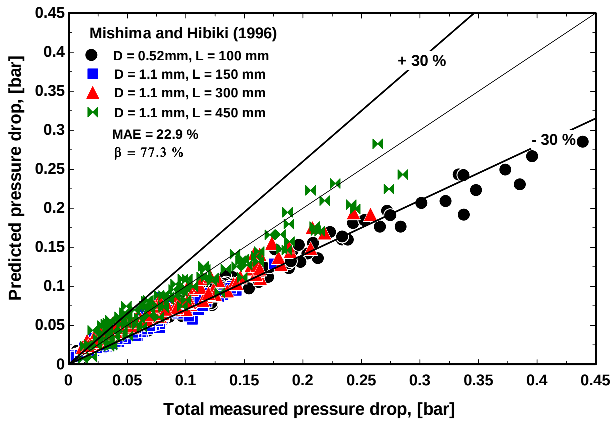


Figure 11 The comparison with the Mishima and Hibiki (1996) correlation.

It can be noted from Table 1 that, some correlations predicted the data of one tube only whereas they failed to predict the data of the other. For example, the correlations of Li and Wu (2010) and Lockhart-Martinelli (1949) predicted the data of the 0.52 mm diameter tube very well while the performance of these correlations is worse in the other tube. Also, it is worth noting that the correlations which were developed for multi-channels configurations such as those proposed by Lee and Lee (2001), Warrier et al. (2002), Qu and Mudawar (2003), Lee and Mudawar (2005) and Lee and Garimella (2008) predicted poorly the current experimental data.

6. Conclusions

The flow boiling pressure drop of R134a was measured in vertical stainless steel tubes with diameters of 0.52 and 1.1 mm at mass flux range of 200 – 500 kg/m^2 s and system pressure 6 – 10 bar. The results indicated that:

1. The two phase pressure drop increases with increasing heat flux (exit quality), increasing mass flux, decreasing system pressure and decreasing tube diameter.
2. The contribution of the frictional pressure drop component to the total measured two phase pressure drop was found to be the largest compared to the contribution of the gravitational and acceleration components. Its value reached about 70 % in the 1.1 mm diameter tube and about 80 % in the 0.52 mm diameter tube. This indicates also that the contribution of the frictional component to the total measured two phase pressure drop increases as the diameter decreases.
3. The contribution of the acceleration component to the total measured pressure drop cannot be ignored in flow boiling studies compared to adiabatic studies. Its value reached about 22 % in the 0.52 mm tube at exit quality value of about 0.9 whilst the corresponding value in the 1.1 mm tube was about 37 %. The higher contribution of the acceleration component in the 1.1 mm tube compared to the 0.52 mm tube is due to the higher values of Reynolds number.
4. The heated length has a significant effect on the measured pressure drop per unit length. At the same exit quality, the shorter the heated length, the higher the measured pressure drop per unit length. At the same exit quality, the applied heat flux to the shortest tube is much higher compared to the longer tubes. This results in the dominance of nucleate boiling in the shortest tube. Accordingly, with the premise that there are small nucleating bubbles next to the wall, the additional interaction between the phases in the liquid film due to nucleation may explain why the pressure drop per unit length in the shortest tube is high. This result is important in explaining why the performance of some correlations gets worse as the heated length increases.
5. The current experimental data were predicted very well by the correlation of Muller-Steinhagen and Heck (1986) followed by the homogenous flow model, a modified form of the Tran et al. (2000) correlation and the Mishima and Hibiki (1996) correlation.

NOMENCLATURE

Bd	Bond number, $\Delta\rho g D^2 / \sigma$
C	Chisholm constant, see Eq. (8)
Co	confinement number, $\left[\sigma / \Delta\rho g D^2\right]^{0.5}$
d_h	hydraulic diameter, m
D	diameter, m

f friction factor, [-]
g gravitational acceleration, m/s^2
G mass flux, kg/m^2 s
j superficial velocity, m/s
L length, m
La Laplace constant, $\left[\sigma / \Delta\rho g D^2\right]^{0.5}$
N number of data points
P pressure, Pa
q heat flux, W/m^2
Re Reynolds number, GD/μ_L
U velocity, m/s
x vapour quality, [-]
X Martinelli parameter, $\left[\frac{(dP/dz)_L}{(dP/dz)_v}\right]^{0.5}$
We Weber number, $G^2 D/\rho\sigma$
z axial distance, m

Greek

α void fraction
β percent of data within ± 30 % error band.
Δ change
μ viscosity, Pa.s
ρ density, kg/m^3
σ surface tension, N/m
ϕ two phase frictional multiplier in Eq. (8)
ψ dimensionless parameter in Eq. (10)

Subscripts

acc acceleration
B bubble
e exit
f friction
g gravitation
go gas only
L liquid
Lo liquid only
LS liquid slug
Lv liquid to vapour change
v vapour
tp two phase

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