BOILING TWO-PHASE PRESSURE DROP IN SMALL DIAMETER TUBES

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ABSTRACT

An experimental study of two-phase pressure drop in small diameter tubes is described in this paper. Stainless steel tubes of internal diameter and length of 4.26 mm, 500 mm and 2.01 mm, 211 mm were used. The working fluid was R134a and the range covered was: mass flux 100 – 500 kg/m^{2}s; system pressure 8-14 bar and exit quality up to 0.9. The heat flux applied to the tubes ranged from 13 – 150 kW/m^2. The effect of diameter on pressure drop is discussed in this paper and a detailed presentation of the results of the comparison with existing pressure drop correlations, some particularly developed for small tubes, is given.

NOMENCLATURE

B	Constant
C	Chisholm parameter in the correlation of Lockhard -Martinelli, dimensionless
D	internal diameter, m
dp/dz	pressure gradient, Pa/m
f	single-phase friction factor, dimensionless
F	function
G	mass flux, kg/m²·s
g	gravitational acceleration, m/s²
L	length, m
P	pressure, bar
Γ	dimensionless physical property
Re	Reynolds number, $Re = G \cdot D/\mu$
x	vapour quality
χ	Lockhard - Martinelli parameter
Greek	
ε	surface roughness
μ	dynamic viscosity, kg/m·s
σ	surface tension, N/m
υ	void fraction
ϕ^2	two-phase multiplier
ν	specific volume, m^3/kg
ρ	density, kg/m³
Subscripts	
a	acceleration component
conf	confinement
e	exit
f	friction
h	hydraulic
l	liquid
lo	liquid only
n	exponenet
tp	two phase
v	vapour
vo	vapour only
z	z direction

INTRODUCTION

Fundamental data on two-phase flow boiling pressure drop are essential for the design and operation of compact and ultra-compact heat exchangers. Advances in high performance electronic chips and the miniaturization of electronic circuits and other compact systems stimulate demand for developing efficient heat removal techniques that may involve such devices. The accurate prediction of two-phase pressure drop will also enable the effective design of refrigeration, air conditioning and heat pump systems. In particular, there is need for a validated correlation for two-phase pressure drop that will facilitate the design and optimization of compact heat exchangers for use with refrigerants.

Two-phase flow pressure drop has been the research subject for several decades. The most widely used analysis method is based on the concept of two-phase multipliers proposed by Lockhart and Martinelli (1949) and empirical correlation of the multipliers from Chisholm (1967, 1973). Ould Didi et al. (2002) extended the study of Kattan et al. (1998) by conducting two-phase pressure drop experiments with five refrigerants (R134a, R123, R402A, R404A and R502) in 10.92 and 12.00 mm tubes. They compared their data against seven two-phase frictional pressure drop prediction methods. They concluded that the method by Grönnerud (1979) and that by Muler-Steinhagen and Heck (1986) provided the most accurate predictions. However, most of the data available in the literature and the prediction methods have been developed for large diameter tubes (above 8 mm). The use of these correlations for compact heat exchanger design needs further study.

In most of the pressure drop studies performed the contribution of friction loss was dominant and the contribution of acceleration and hydraulic loss were small (Saitoh et al. 2005, Coleman and Krause 2004, Ould Didi et al. 2002). However, Kureta et al. (1998) found that the acceleration loss is comparable to the friction loss under certain conditions in small diameter tubes. They compared their experimental results for water with several correlations and models in tubes with inner diameter ranging from 2 and 6 mm. They reported that for the vertical tube of 2 mm inside diameter the pressure drop can be predicted by the Martinelli and Nelson (1948) model for friction pressure drop and the acceleration loss calculated based on the annular flow model. For the 6 mm tube, they recommended the homogeneous flow model for the wall friction. Akagawa et al. (1969) carried out experiments for a 2 mm tube and water. They calculated the acceleration and friction loss. They compared the acceleration component with the homogeneous and annular flow models and found the later predicted their results better. The friction component was predicted using the Martinelli and Nelson model (1948). Tran et al. (2000) analyzed the two-phase pressure drop associated with nucleate boiling in small channels and tubes that was measured in their earlier studies. The objective of their study was to determine if large-tube correlations could be used to predict two-phase pressure drop of refrigerants in small channels (hydraulic diameter < 3 mm). They used two smooth circular tubes and one rectangular channel with hydraulic diameters 2.46, 2.92, and 2.40 mm, respectively. Five state-of-the-art large-tube correlations were evaluated, but they failed to predict the pressure drop of flow boiling in small channels for all test conditions. The divergence was attributed to the difference in the flow patterns that can exist in the small channels. They also developed a new correlation to predict the two-phase pressure drop neglecting the acceleration component. This was based on the B-coefficient method developed earlier by Chisholm (1967, 1973). Yu et al. (2002) studied two-phase pressure drop of water in a small horizontal tube of 2.98 mm inside diameter and 0.91 m heated length. They compared their experimental results at a system pressure of 2 bar with the widely used Chisholm correlation (1967, 1973) for larger channels. The two-phase pressure drop data of the small channel of this study were consistently lower than those expected in larger channels at the same mass flux. Warrier et al. (2002), did experiments in a multi-channel system with hydraulic diameter 0.75 mm and developed a correlation for the two-phase pressure drop. In their prediction they used the Lockhart and Martinelli (1949) correlation to evaluate the void fraction and the separated flow model to determine the two-phase pressure drop. The prediction agreed with their experimental result within $\pm$ 27%. Saitoh et al. (2005) studied the effect of tube diameter on the boiling heat transfer and pressure drop for refrigerant R134a in horizontal small diameter tubes (0.51, 1.12, and 3.1 mm ID). The frictional pressure drop was calculated based on the homogeneous model and also on the Lockhart-Martinelli correlation. The measured pressure drop for the 3.1 mm ID tube agreed well with that predicted by the Lockhart-Martinell correlation. However, with decreasing internal diameter, the measured pressure drop was predicted better by the homogenous model rather than by the Lockhart-Martinelli correlation.

In the reports summarised above, it can be seen that there is a difference in the prediction methods for the small diameter passages in comparison with conventional large diameters. In small diameter passages, some researchers noted that the acceleration pressure loss is comparable to the friction loss and proposed the annular flow model for prediction while others neglected it. The experimental data for two-phase pressure drop for small and micro tubes are scarce especially for a substantial range of variables. There is also still lack of understanding of the mechanism that will allow formulating an accurate design correlation for compact systems. More accurate pressure data are needed in order to have a reliable design method for the prediction of pressure drop in small to micro tubes and eventually in compact heat exchangers.

EXPERIMENTAL FACILITY

An experimental facility was designed and constructed during this study to allow a detailed and accurate investigation of the effect of diameter on flow patterns, pressure drop as well as heat transfer rates. The pressure drop results are discussed in this paper. A schematic diagram of the experimental facility is shown in Figure 1. It consists primarily of (a) the tank (1), which is used to receive liquid refrigerant with a small amount of subcooling although the fluid in this tank is kept at two-phase state; (b) a refrigerant circulating pump (2); (c) two mass flow meters for measuring high (5) and low (6) flow rates respectively, thus ensuring high measurement accuracy; (d) a chiller (7) and preheater (8) to control the refrigerant inlet temperature; (e) test section (10); (f) condenser (13). The system pressure corresponding to the desired saturated temperature can be controlled through heating the two-phase refrigerant in the tank. A gear pump, which can create continuous flow, is used in this system to circulate the refrigerant from the tank through a filter drier (3), sight glass (4), and mass flow meters then chiller followed by preheater. The chiller, which is a tube-in-tube heat exchanger, removes heat from R134a using an R22 cooling system to maintain subcooled conditions. Following the chiller, refrigerant is heated in the preheater to a desired subcooled temperature or saturated vapour quality. Inlet temperature or quality at the entry to the test section is thus controlled by varying the heating power applied on the preheater. A thermocouple and a pressure transducer are installed to record the temperature and pressure at the entry to the test section. The inlet quality is calculated based on the heating power, which is measured by a digital power meter, and the change of enthalpy between the preheater inlet and the inlet to the test section. The two mass flow meters installed in the system measure the subcooled liquid flow rates. At the exit of the pump, the higher pressure, which is created by the pump, results in a few degrees of subcooling from the saturated liquid from the tank. A thermocouple and a pressure transducer are installed before the flow meters to check the fluid state at this point, make sure it is liquid, which ensures high measuring accuracy of the mass flow meters. A by-pass valve after the pump allows part of the refrigerant to return back to the tank after the pump and hence can be used to regulate the flow rate as required. Two-phase and subcooled flow boiling heat transfer in small diameter tubes can be investigated under a uniform heat flux condition by supplying electric current directly through the test section. The quality at the exit of the test section is determined by an energy balance of the heat supplied and the enthalpy change of the refrigerant. The total enthalpy change is calculated based on the flow rate of the refrigerant, the pressure and temperature change measured by a differential pressure transducer and thermocouples, respectively, installed at two ends of the test section. A glass tube is installed immediately after the stainless steel test section to observe the flow patterns. The mass flow rate and inlet quality to the glass tube test section are adjusted according to which part of the flow regime is going to be investigated. After the test section, the two-phase state refrigerant is separated into pure liquid and vapour in a separator. This can reduce the pressure drop that occurs in the connecting pipe and condenser. The liquid refrigerant flows into the tank directly while the vapour refrigerant is condensed in the condenser. The separator can be bypassed to direct all fluid (liquid and vapour) into the condenser. Subcooling in the condenser will then offer the possibility of reducing the system pressure when required. Cooling in the chiller and condenser is provided by the R22 plant. There is a small tube to balance the pressure before the condenser and the pressure in the receiver. This aims to reduce and avoid pressure fluctuations before the condenser. The test rig can use a range of working fluids including refrigerants and water. The current working fluid is R134a. . The test section was made of stainless steel cold drawn tubes, 4.26 mm id with 0.245 mm wall thickness and 521 mm length and 2.01 mm id with 0.19 mm wall thickness and 233 mm length. Other parameters were varied in the range: mass flux 100 – 500 kg/m^{2}s; pressure 8 – 14 bar; exit quality up to 0.9; heat flux 13 - 150 kW/m^2. The uncertainty in temperature measurement was ± 0.16 K, flow rate measurements ± 0.4%, and pressure measurements ± 0.15 %. The differential pressure transducer used to measure the pressure drop has an accuracy of about 0.3 %. The effect of diameter on pressure drop is discussed and a detailed presentation of comparisons with existing state of the art pressure drop models and correlations is included.

DATA REDUCTION

Since the test section was installed vertically, the pressure drop measured by the differential pressure transducer installed across the test section was an overall contribution of three components: friction, acceleration and height, in which friction pressure drop was expected to be the major component. The friction pressure drop was calculated by subtracting the acceleration pressure drop and hydrostatic pressure from the measured value.

$$\frac{dp}{dz} = \frac{dp_f}{dz} + \frac{dp_a}{dz} + \frac{dp_z}{dz} \tag{1}$$

where $\frac{dp_f}{dz}$ is the friction pressure gradient, $\frac{dp_a}{dz}$ acceleration pressure gradient and $\frac{dp_z}{dz}$ hydraulic pressure gradient. The acceleration component is given by Equation (2) as follows (Chisholm 1983):

$$\frac{dp_a}{dz} = -G^2 \frac{d}{dz}\left(\frac{x^2}{\upsilon\rho_v} + \frac{(1-x)^2}{(1-\upsilon)\rho_l} \right) \tag{2}$$

The hydraulic component is described by Equation (3):

$$\frac{dp_z}{dz} = g\left[\upsilon\rho_v + (1-\upsilon)\rho_l\right] \tag{3}$$

As suggested by Bao et al. (1994), the local void fraction can be calculated from the Lockhart-Martinelli multiplier as follows

$$\upsilon = 1 - \frac{1}{\phi} \tag{4}$$

and the Lockhart-Martinelli multiplier is defined as:

$$\phi^2 = \frac{(dp_f / dz)_{tp}}{(dp_f / dz)_l} = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{5}$$

where $X^2 = (dp_f / dz)_l / (dp_f / dz)_v$ and C is a constant that depends on the liquid and vapour Reynolds number Re.

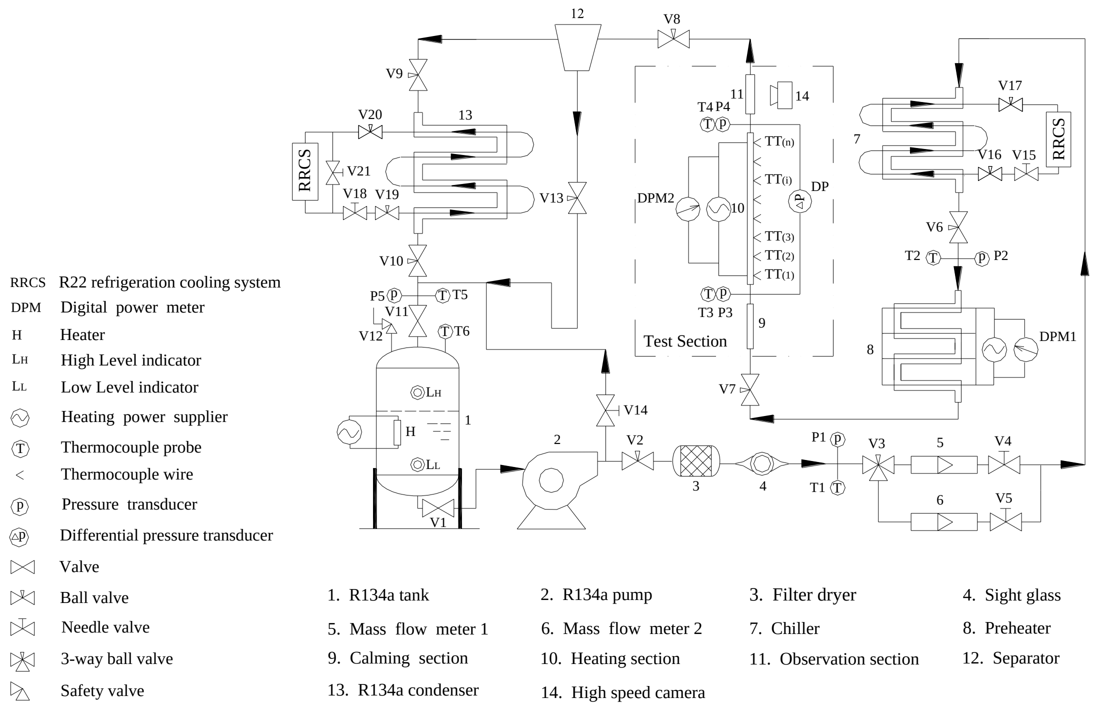

Figure 1 Overall schematic of the experimental facility.

BRIEF BACKGROUND THEORY OF CORRELATIONS

The general approach to predict the two-phase friction drop is to follow Chisholm's method of using the two-phase multiplier. The two-phase multiplier method is widely used to calculate the two-phase friction pressure drop as a product of a single-phase friction pressure drop and a two-phase multiplier (Martinelli and Nelson (1948), Lockhart and Martinelli (1949) and Chisholm (1983)). Chisholm (1983) expressed the friction component as Equation (6):

$$\Delta P = \phi^2 f \frac{L}{D}\frac{G^2}{2\cdot\rho_l}\left[1 + x\left(\frac{\rho_l}{\rho_v} - 1\right)\right] \qquad (6)$$

where ϕ^2 is the two phase friction factor and f single phase friction factor, see equation 6. Since the single-phase pressure drop can be estimated with good accuracy from well-established equations, the two-phase multiplier becomes the dominant factor in determining two-phase friction pressure. The frictional pressure gradient for single-phase flow can be obtained from the correlation as follows (Chisholm 1983),

$$\frac{dp_f}{dz} = -f\frac{G^2}{2D\rho_l}(1-x)^2 \qquad (7)$$

where the friction factor f is a function of Reynolds number (Re) and surface roughness. In smooth pipes, the friction factor for turbulent flow can be expressed by a Blasius type equation

$$f = \frac{a}{Re^n} \qquad (8)$$

where for Re between 2000 and 100000, a = 0.314 and n = 0.25; or for Re between 5000 and 200000, a = 0.186 and n = 0.2. For Re < 1000, where laminar flow occurs, a = 64 and n =1.0. In the present comparison, the correlation proposed by Churchill (1977), which spans all fluid flow regimes, is used:

$$f = 8\left[\left(\frac{8}{Re}\right)^{12} + \frac{1}{(A+B)^{3/2}}\right]^{1/12} \qquad (9)$$

where

$$A = \left\{2.457 ln\left[\frac{1}{(7/Re)^{0.9} + 0.27\varepsilon/D}\right]\right\}^{16} \qquad (10)$$

and

$$B = \left(\frac{37530}{Re}\right)^{16} \qquad (11)$$

Most commonly, two models are used to correlate pressure drop; namely the homogeneous model and the separated-flow model (Whalley, 1986). In the homogeneous model, the liquid and gas phases are assumed to have the same velocity, which is suitable for bubble flow. In this model, the mixture density can be calculated approximately from the thermodynamic equilibrium quality x, obtained from the inlet conditions and the heat input. This leads directly to the acceleration and hydrostatic pressure gradient. In the separated-flow model, the acceleration and hydrostatic pressure drops are calculated based on a one-dimension model, which assumes an ideal smooth interface between the liquid and vapour phases.

Some pressure drop correlations proposed in the literature are listed in Table 1 in the appendix. The experimental results from the present study are compared with these correlations and presented in the paper.

EXPERIMENTAL RESULTS

The experimental data for two-phase frictional pressure drop, deducted from the experimental data using equations (2), (3), in the 2.01 mm and the 4.26 mm tubes are depicted in Figures 2 and 3, respectively, as a function of exit vapour quality for various values of mass flux and system pressure. The results show that the two-phase flow pressure drop increases with increasing exit vapour quality and mass flux but decreases with increasing system pressure. These trends agree with those in large tubes. Note that the single-phase friction pressure drop for the 2.01 mm tube calculated using equation (7) by Churchill (1977) was a value of 0.013 and 0.018 bar for a mass flux 400 and 500 kg/m^{2}s respectively. This is as expected if we were to extend the lines in Figure 2 to the abscissa (quality, x=0). The corresponding values for the 4.26 mm tube are 0.01bar for mass flux of 400 kg/m^{2}s and 0.016 bar for mass flux of 500 kg/m^{2}s. Due to the difference in the test section length, the comparison of pressure drop in these two tubes is based on pressure gradient instead of pressure drop and is shown in Figure 4. As seen in the figure the friction pressure gradients in the 2.01 mm tube are about 3 times higher than that in the 4.26 mm tube. The friction pressure drop and the pressure gradient presented in Figures 2-4 also show the linear dependence of pressure drop with quality.

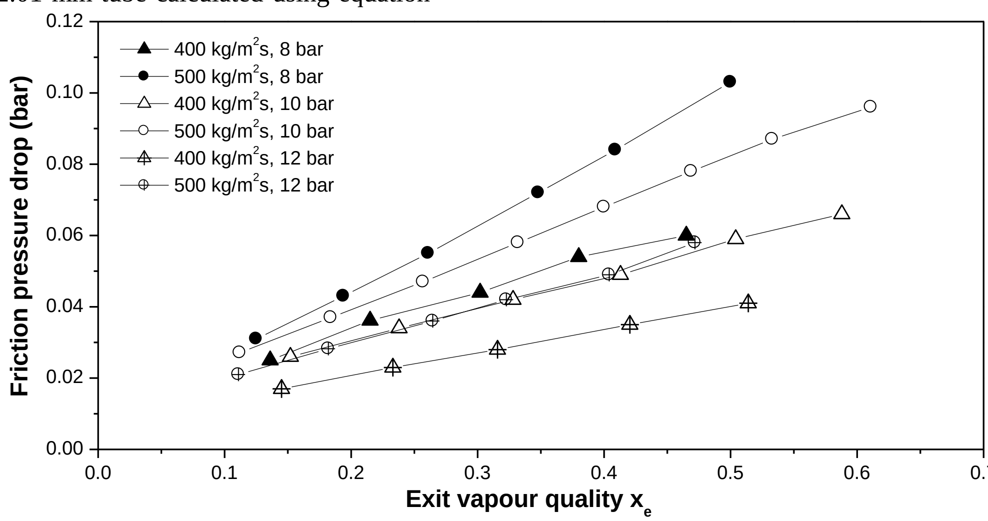

Figure 2. Pressure and mass flux effect on two-phase flow frictional pressure drop, d = 2.01mm.

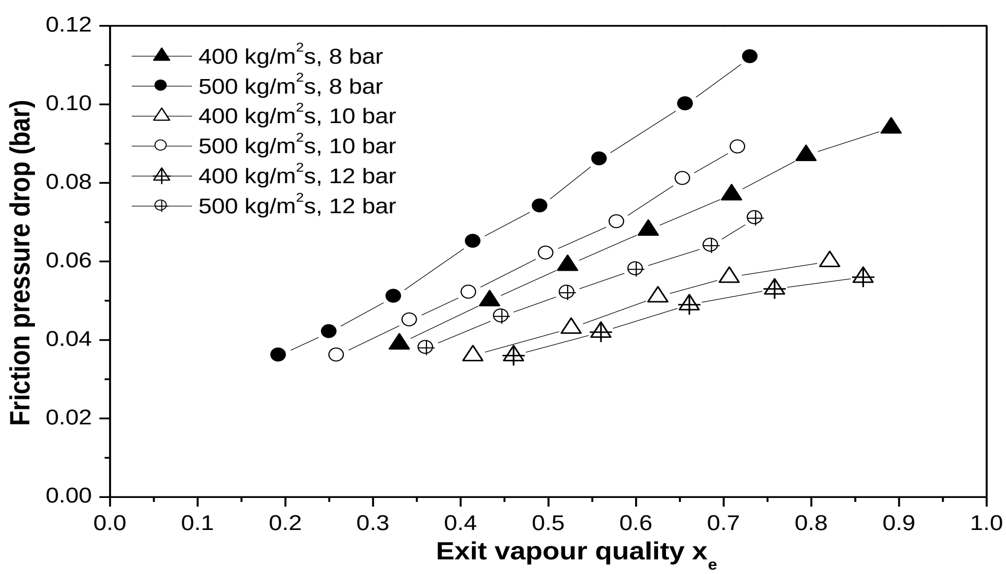

Figure 3. Pressure and mass flux effect on two-phase flow frictional pressure drop, d = 4.26 mm.

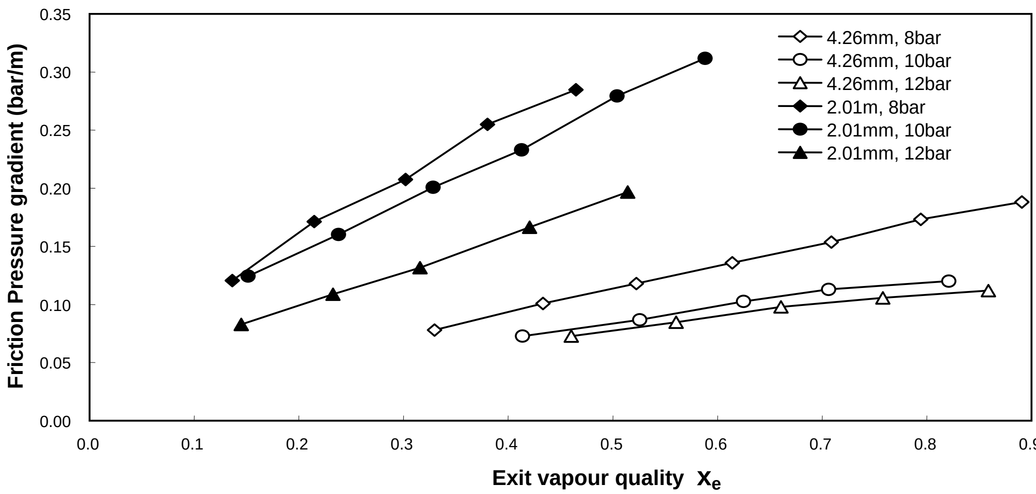

Figure 4. Tube diameter effect on friction pressure gradient, G = 400 kg/m^2s.

COMPARISON

Below the experimental two-phase friction pressure drop data are compared with various correlations.

The Chisholm et al. correlation

The friction multipliers used by Chisholm et al. (1983) correlate the two-phase pressure drop to that which would occur in single-phase flow if the total mass flux were liquid or vapour only. The Chisholm et al. correlation was mainly developed for pressure drop in turbulent flow. the Reynolds number in the experimental data varies from 4000 to 6500 in the 2.01 mm tube and 9500 to 14000 in the 4.26 mm tube. Figure 5 shows the comparisons of experimental pressure drop in the 4.26 mm tube with the Chisholm et al. correlation, which predicts the experimental data within ±30%. However, when the pressure drop is less than 0.08 bar, this correlation overestimates the experimental data, and after this point, it underestimates them. It was found from the experiment results that the bigger the outlet vapour quality, the higher the pressure drop. Since the length of the test sections were fixed in this study, the bigger outlet quality was caused by higher heat flux, which means that heat flux may have some effect on pressure drop. Figure 6 shows the comparison of the experimental pressure drop in 2.01 mm tube with the prediction of the Chisholm et al. correlation. Again, the correlation predicts the experimental data mostly within ±30%, which means that there is no tube diameter effect on the Chisholm et al. correlation prediction for tube diameter down to 2.01 mm.

The Tran et al. correlation

Tran et al. (2000) found that the large tube correlations failed to predict satisfactorily the two-phase pressure drop for three refrigerants (R134a, R12, R113) during flow boiling in channels with hydrodynamic diameters of 2.5 and 2.9 mm. By incorporating the effects of tube dimension and fluid surface tension, they used the Confinement number proposed by Kew and Cornwell (1995) to develop a correlation based on the Chisholm (1983) method.

Figures 7 and 8 show the comparisons of experimental pressure drop in the 4.26 mm and 2.01 mm tubes with the Tran et al. correlation, respectively. It is clearly seen in both figures that the Tran et al. correlation underestimates our experimental data by more than 30%. The predictions of the experimental data in the 2.01 mm tube are even worse than those in the 4.26 mm tube. One would expect the comparison to be better for the 2.01 mm tube since the diameter used by Tran et al. were similar and their fluid included R134a. However, the rest of the parameters in the Tran et al. did not cover our experimental range, i.e. in Tran et al. the L/D was 372, the reduced pressure 0.08 – 0.21 and the exit quality 0.24 – 0.95; they were 100, 0.20 – 0.34 and 0.0 – 0.9 respectively in our experiments. Therefore, it may be inferred that the Tran et al. correlation cannot predict the effect of these parameters on the friction pressure drop very well.

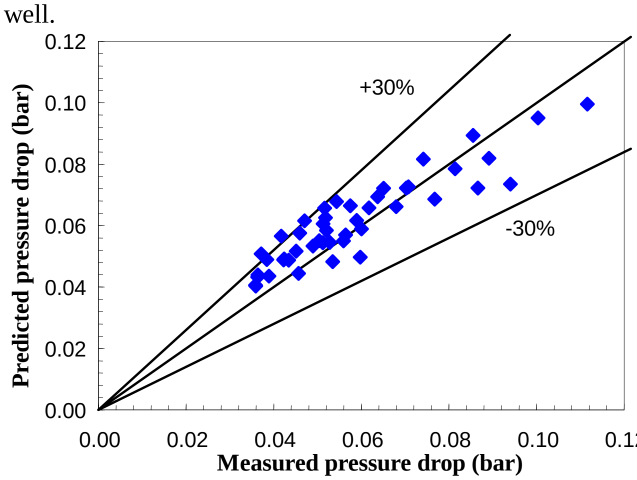

Figure 5. Comparison with the Chisholm et al. (1983) correlation, d = 4.26 mm.

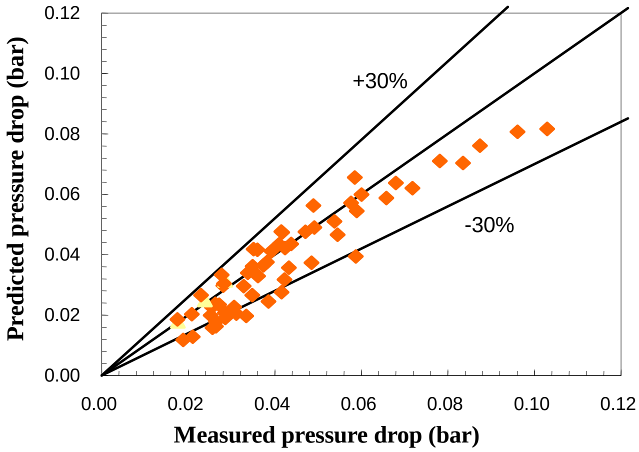

Figure 6. Comparison with the Chisholm et al. (1983) correlation, d = 2.01 mm.

The Yu et al. correlation

Yu et al. (2002) proposed a correlation based on modifications of the Chisholm two-phase multiplier correlation (Chisholm et al. 1983). They found that the Chisholm correlation consistently over predicted their data. They attributed this to the fact that the occurrence of slug flow over a large quality range in small channels reduces the pressure gradients from the annular flow condition found in large tubes upon which the Chisholm correlation is substantially based. Therefore, by looking at the three terms of the Chisholm correlation, they found that the $1/X^2$ term was dominant in their experiment data and these data were then better correlated with a power function, $\phi_{lo}{}^2 = X^{-1.9}$. However, from our flow pattern observations (X. Huo, 2006), the dominant flow pattern is not slug flow but churn flow, which can cause higher friction pressure drop than the slug flow due to a stronger fluctuation at the interface between liquid layer and vapour core. Therefore, it is not surprising that the Yu et al. correlation very much under estimates our experimental data for both 4.26 mm and 2.01 mm tubes, which can be seen in Figures 9 and 10 respectively.

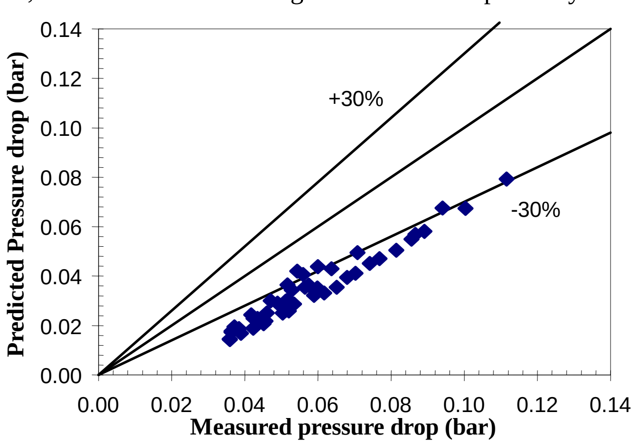

Figure 7. Comparison with the Tran et al. (2000) correlation, d = 4.26 mm.

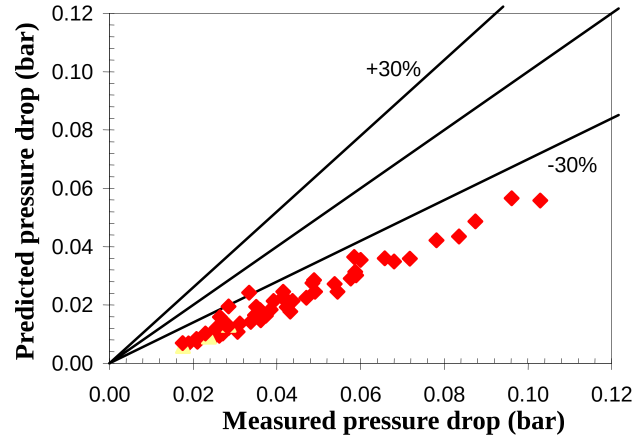

Figure 8. Comparison with the Tran et al. (2000) correlation, d = 2.01 mm.

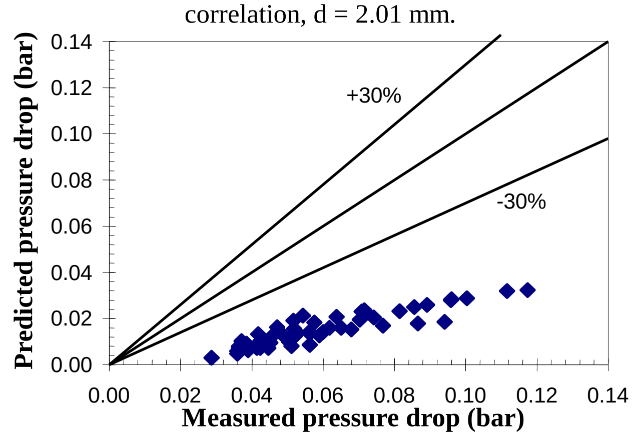

Figure 9. Comparison with the Yu et al. (2002) correlation, d = 4.26 mm.

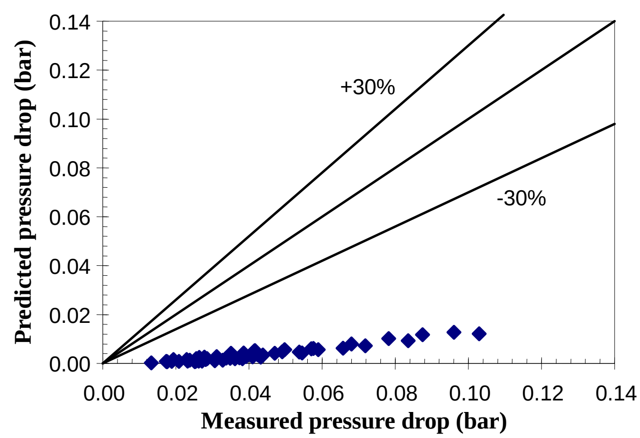

Figure 10. Comparison with the Yu et al. (2002) correlation, d = 2.01 mm.

The Warrier et al. correlation

Warrier et al. (2002) proposed a correlation for pressure drop based on the separated flow model. The two-phase multiplier for only liquid flowing ($\phi_{lo}{}^2$) was given by

$$\phi_{lo}{}^2 = \phi_l{}^2 \left(\frac{f_l}{f_{lo}}\right)(1-x)^2 \qquad (12)$$

where f_l and f_{lo} are the friction factors for liquid flowing alone (mass flux = G(1-x)) and the mixture flowing as liquid (mass flux = G), respectively. The two-phase multiplier for liquid flowing alone (ϕ_l^2) (mass flux = G(1-x)) was given by the Martinelli-Nelson (1948) correlation. Warrier et al. (2002) used a value of 38 for the constant C in the Martinelli-Nelson correlation basing this decision on their experiment data.

It can be seen from the comparison presented in Figures 11 and 12 that the Warrier et al. correlation overpredicts our experimental results. However, the disparity is less for the smaller 2.01 mm tube. The reason for this is because this correlation was proposed for a hydraulic diameter of 0.75 mm with the channels oriented horizontally. The smaller the hydraulic diameter, the less the effect of gravity, therefore, the correlation proposed for horizontal tubes can be applied to vertical flow with increasing accuracy as the diameter decreases.

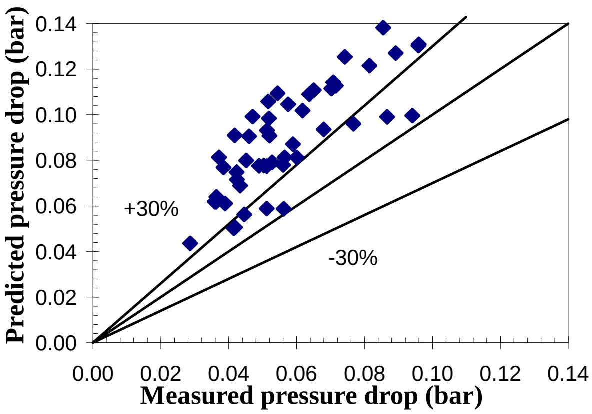

Figure 11. Comparison with the Warrier eta al. (2002) correlation, d= 4.26 mm.

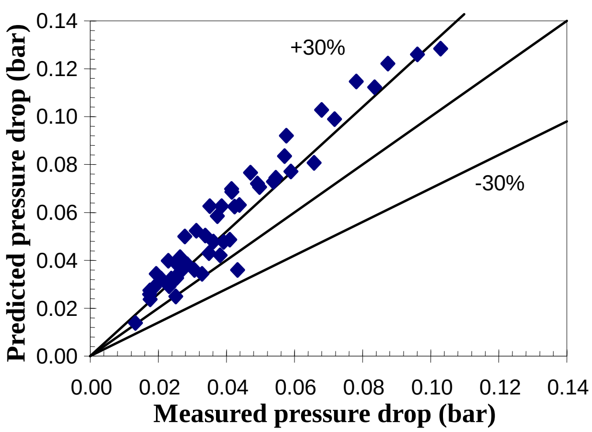

Figure 12. Comparison with the Warrier et al. (2002) correlation, d = 2.01 mm.

The Lee and Lee correlation

Lee and Lee (2001) proposed a correlation based on 305 data points to represent the two-phase pressure drop through a horizontal rectangular channel. The two-phase frictional multiplier was expressed using a Lockhart-Martinelli type correlation with the parameter C modified to take account of the gap size and the flow rates of the gas and liquid. The correlation was valid for the Martinelli parameter (X) and the all-liquid Reynolds number (Re_{lo}) ranges of 0.303 – 79.4 and 175 – 17700, respectively. The correlation represented well the pressure drop through their narrow channels, the gap between the upper and the lower plates of each channel ranged from 0.4 to 4.0 mm with a fixed width of 20 mm, which results in the range of the hydraulic diameter from 0.78 to 6.67 mm. This correlation predicted their measured data within ±10%.

Figures 13 and 14 present the comparison of the prediction by the Lee and Lee correlation and our experiment results in the 4.26 mm and 2.01 mm tubes, respectively. Although the effects of the mass flux and the gap size were taken into account in this correlation, it was based on adiabatic water-air flow at atmospheric pressure, therefore, it does not predict the effect of system pressure on the friction pressure drop very well for both the 2.01 mm tube and 4.26 mm tube. The data which were over predicted by more than 30% were obtained at the higher system pressures of 12 and 14 bar.

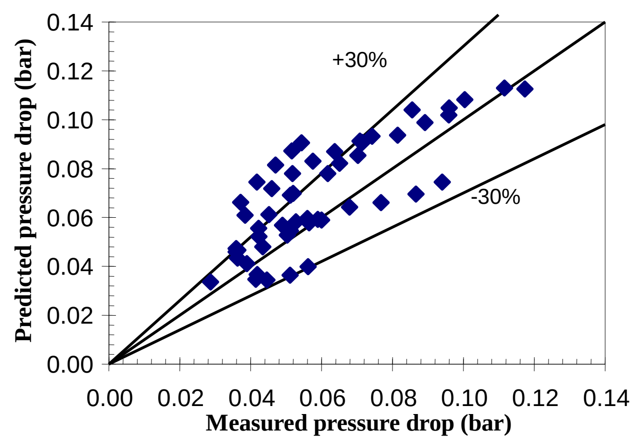

Figure 13. Comparison with Lee and Lee (2001) correlation, d = 4.26 mm.

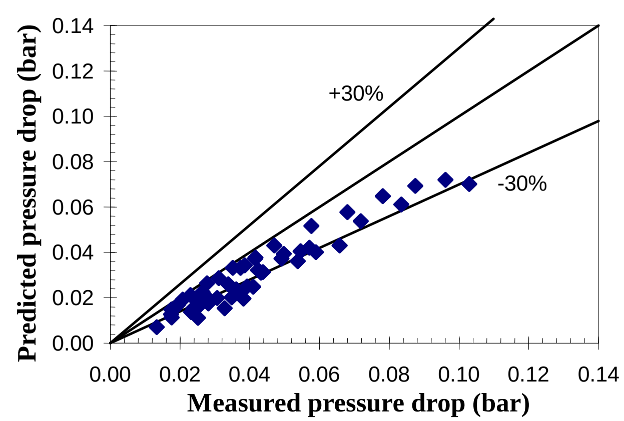

Figure 14. Comparison with the Lee and Lee (2001) correlation, d = 2.01 mm.

The Qu and Mudawar correlation

Qu and Mudawar (2003) developed a correlation based on the combination of laminar liquid and laminar vapour flow and incorporating the effects of both channel size and fluid mass velocity. The comparison of the prediction of this correlation and our experiment results is depicted in Figures 15 and 16. This correlation was based on a mass flux range of 134.9 – 400.1 kg/m^{2}s and a system pressure of 1.17 bar. Furthermore, the exit quality in the work of Qu and Mudawar was only up to 0.2 and this means that the correlation was proposed based on annular two-phase flow model, in which the vapour phase flows along the channel center as a continuous vapour core, a portion of the liquid phase flows as thin film along the channel wall, while the other portion is entrained in the vapour core as liquid droplets. From our observation, the flow pattern was churn flow until a quality up to about 0.18 in the 2.01 mm tube and 0.22 in the 4.26 mm tube. This may explain the fact that the Qu and Mudawar correlation predicts our results at high quality better than those at low quality, as seen in Figure 2&3 (high ΔP occurs at high quality).

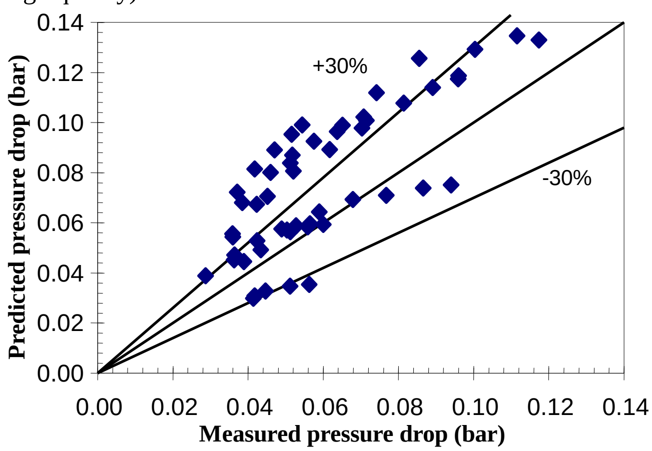

Figure 15. Comparison with Qu and Mudawar (2003) correlation, d = 4.26 mm.

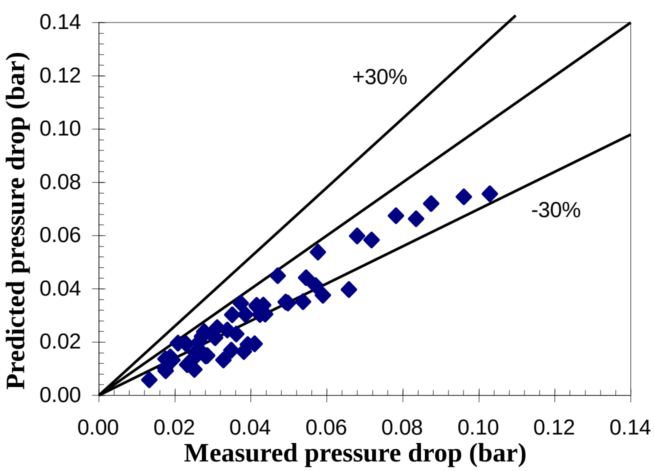

Figure 16. Comparison with Qu and Mudawar (2003) correlation, d = 2.01 mm.

CONCLUSIONS

The experimental pressure drop results of these study show that the pressure drop along the test section increases with mass flux and exit quality but decreases with system pressure. The pressure drop gradients in 2.01 mm tube are about 3 times higher than those in the 4.26 mm tube.

The Chisholm et al. (1983) correlation for friction pressure drop in large tubes predicted our data within ±30%. The agreement with the other correlations for small diameter channel was much worse. A possible reason for this is the limited or different experimental range (e.g. diameter, system pressure, mass flux) on which the correlation was based. Therefore, more fundamental analysis and experiments are needed to get an improved pressure drop correlation for small to micro diameter tubes.

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APPENDIX I

Table 1. Existing two-phase flow pressure drop correlations.

Reference	Pressure drop correlation	Fluids and Range of Application
Chisholm et al. (1983)	$\frac{(dp_f/dz)_{tp}}{(dp_f/dz)_{lo}} = 1 + (\Gamma^2 - 1)\cdot \left[Bx^{(2-n)/2}(1-x)^{(2-n)/2} + x^{2-n}\right]$ $\Gamma = (\frac{\Delta p_{vo}}{\Delta p_{lo}})^{0.5} = (\frac{\rho_l}{\rho_v})^{0.5}(\frac{\mu_v}{\mu_l})^{n/2}$	$G = 500 - 1900$ kg/m^{2}s
Tran et al. (2000)	$\phi_{lo}^2 = 1 + (C\Gamma^2 - 1)\left[N_{conf}\, x^{0.875}(1-x)^{0.875} + x^{1.75}\right]$ $\Delta p_f = \Delta p_{f\,lo}\left\{1 + (C\Gamma^2 - 1)\left[N_{conf} F_1(x_e) + F_2(x_e)\right]\right\}$ $C = 4.3$ $\quad N_{conf} = \frac{\left[\frac{\sigma}{g(\rho_l - \rho_v)}\right]^{0.5}}{D}$	$P = 1.38 - 8.64$ bar $d = 2.40, 2.46, 2.92$ mm $P_r = 0.04 - 0.23$ $G = 33 - 832$ kg/m^{2}s $q = 2.2 - 90.8$ kW/m^2 $x_e = 0 - 0.95$ R134a, R12, R113
Yu et al. (2002)	$\phi_{lo}^2 = X^{-1.9}$ $X = 18.65(\frac{\rho_v}{\rho_l})^{0.5}(\frac{1-x}{x})\frac{Re_v^{0.1}}{Re_l^{0.5}}$	$P = 2$ bar $d = 2.98$ mm $G = 50 - 200$ kg/m^{2}s Water
Warrier et al. (2002)	$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $C = 38$	$d = 0.75$ mm, $L/d = 409.8$ $G = 557 - 1600$ kg/m^{2}s $q = 0 - 59.9$ kW/m^2 $x = 0.03 - 0.55$ FC84
Lee and Lee (2001b)	$\phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ if Re_l, $Re_v < 2000$ $C = 6.833\times10^{-8}\lambda^{-1.317}\psi^{0.719}Re_{Lo}^{0.557}$ $\lambda = \frac{\mu_L^2}{\rho\,\sigma D_h}$, $\psi = \frac{\mu_L U_l}{\sigma}$, where U_L is velocity of liquid if Re_l, $Re_v > 2000$ $\quad C = 0.408 Re_{lo}^{0.451}$ if $Re_l < 2000$, $Re_v > 2000$ $\quad C = 6.185\times10^{-2} Re_{lo}^{0.726}$ if $Re_l > 2000$, $Re_v < 2000$ $\quad C = 3.627 Re_{lo}^{0.174}$	$d = 0.78 - 6.67$ mm Atmospheric pressure Water/air
Qu and Mudawar (2003)	$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $X = (\frac{\mu_l}{\mu_v})^{0.5}(\frac{1-x_e}{x_e})^{0.5}(\frac{v_l}{v_v})^{0.5}$ $C = 21[1 - exp(-0.319\times10^3 d_h)](0.00418G + 0.0613)$	$T_{in} = 30$ and 60 ^oC $G = 134.9 - 400.1$ kg/m^{2}s $P_{out} = 1.17$ bar $d_h = 0.35$ mm Water