Experimental Investigation of Two-Phase Pressure Drop in a Microchannel

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Abstract An experimental study of two-phase pressure drop in a horizontal circular microchannel is presented in this paper. Test tube is made of fused silica having an internal diameter of 781µm with a total length of 261mm and a heated length of 191mm.The external surface of the test tube is coated with an electrically conductive thin layer of ITO (Indium Tin Oxide) for direct heating of test section. Refrigerant R134a was used as the working fluid and mass flux during the experiments was varied from 100 to 650 kg/m^2sec. Experiments were performed at a system pressure of 7.70 bar (corresponding to saturation temperature of 30^oC). Two-phase frictional pressure drop characteristics with different mass flux, vapour fraction and heat flux were explored in detail. Finally, the prediction capability of well known available correlations in the literature, developed for macrochannels and others especially developed for microchannels was also checked. The homogeneous model predicts the data fairly well with a mean absolute deviation (MAD) of 19% and 69% of data within ±20% error band. The Müller-Steinhagen and Heck (1986) correlation developed for macrochannels predicts the data with a MAD of 19% and 61% of data within ±20% error band. The Mishima and Hibiki (1996) correlation, developed for microchannels, also shows fairly good approximation of the data with MAD of 19% and 57% of data within ±20% error band.

Keywords: Micro Flow, Microchannel, Two-Phase, Pressure Drop, Flow Boiling

1. Introduction

Micro/mini channel heat sinks have received much attention, in recent years, from industrial and scientific community due to several advantages they offer such as superior heat transfer performance during high heat flux cooling applications, minimization of the fluid inventory, less overall weight of heat exchange system, compactness and low cost. It is worth mentioning here that reducing diameter will require an increased number of parallel channels to compensate for higher pressure drop.

There could be several application areas for microchannel heat sinks such as cooling of electronics, automotive heat exchangers, commercial refrigeration and air conditioning just to mention a few. Compactness and reduced fluid inventory are very attractive benefits offered by the microchannels which can be exploited for enhancing the safety of heat exchange equipment and for meeting the ever increasing concerns of the environmental pollution. Therefore, it is of utmost importance to be able to predict the two-phase pressure drop of miniaturized heat exchange equipment very accurately in order to use this type of equipment in real applications.

2. Literature review

Looking at the open literature, most of the work done on two-phase pressure drop can be divided into two types, based on either the homogeneous model or the separated flow model. The homogeneous model, which is simpler to apply, assumes the two phase fluid flowing as a single phase with mixture averaged properties while the separated flow model considers two phases flowing separately.

The total pressure drop can be represented as the sum of frictional, gravitational and momentum pressure drop components:

$$\left(\frac{dP}{dz}\right)_t = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_{gr} + \left(\frac{dP}{dz}\right)_m \quad (1)$$

Gravitational pressure drop is zero for horizontal channels.

$$\left(\frac{dP}{dz}\right)_{gr} = 0 \quad (2)$$

Momentum pressure drop is due to change in vapor fraction along the channel and is calculated as:

$$\left(\frac{dp}{dz}\right)_m = \frac{G^2}{\rho_l}\left[\left(\frac{x^2}{\alpha}\right)\left(\frac{\rho_l}{\rho_g}\right) + \left(\frac{(1-x)^2}{1-\alpha}\right) - 1\right] \quad (3)$$

Where α is void fraction and can be calculated using any of several correlations suggested in the literature. In this paper this quantity is calculated using Zivi's (1964) correlation given as:

$$\alpha = \left[1 + \frac{1-x}{x}\left(\frac{\rho_g}{\rho_l}\right)^{\frac{2}{3}}\right]^{-1} \quad (4)$$

Experimental frictional pressure drop is calculated by deducting the momentum pressure drop from total experimental pressure drop. Frictional pressure drop can also be estimated using homogeneous or separated flow model or any other suitable correlation for two phase frictional pressure drop, e.g. by those listed in table 4 and table 5.

Kawahara et al (2002) experimentally investigated the flow patterns, void fraction and two phase pressure drop in a 100μm circular microchannel made of fused silica in which deionized water and nitrogen were used as test fluid. The homogeneous model over predicted their experimental data of two phase frictional pressure drop while they found good approximations with Lockhart-Martinelli correlation (1949).

Revellin et al. (2006) measured adiabatic two phase pressure drop in circular microchannels having inner diameters of 0.509mm and 0.790mm while using R134a and R245fa as working fluids. Similar to single phase friction factor, they distinguished three different zones for laminar (for $Re_{tp} \leq 2000$), transition (for $2000 \geq Re_{tp} \leq 8000$) and turbulent regions ($Re_{tp} \geq 8000$). 62% of their data in the turbulent regime was predicted by Müller-Steinhagen and Heck correlation within ±20% error band while no other tested correlation predicted their data satisfactorily.

Lee and Lee (2000) performed experiments to study the two-phase adiabatic pressure drop in small rectangular channels. Water and air was allowed to pass through channel gap sizes of 0.4, 1, 2 and 4 mm and constant width of 20 mm in the experiments. Pressure drop increased with increase in superficial velocities of liquid and gas, and with the decrease in the gap size. They presented a set of correlations, of the Lockhart-Martinelli type, with modified value of parameter C to take gap size and flow rate into account. A newly developed correlation predicted their results within ±10% error band.

Mishima and Hibiki (1996) measured two-phase frictional pressure drop for air-water in vertical circular microchannels with tube inner diameters from 1 to 4mm. They developed a new correlation with a modified equation for Chisholm parameter C taking into account the inner diameter of tubes. The new equation correlated their data well.

3. Two-phase frictional pressure drop correlations for macro and micro channels

Widely used models for calculating two-phase pressure drop are the homogeneous and the separated flow models. Some other models are either modified forms of the homogeneous model or the separated flow model. Some of the models and correlations used for comparison in this study are given below and the others can be found in the references provided.

3.1. Homogeneous model

Frictional pressure drop is calculated as

$$\left(\frac{dp}{dz}\right)_f = \frac{f_h G^2}{2D\rho_h} \quad (5)$$

$$\rho_h = \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_l} \right)^{-1} \quad (6)$$

$$f_h = \frac{64}{\mathrm{Re}_h} \quad \text{for Re} \le 2300 \quad (7)$$

$$f_h = 0.3164 \times \mathrm{Re}_h^{-0.25} \quad \text{for Re} > 2300 \quad (8)$$

$$\mathrm{Re}_h = \frac{GD}{\mu_h} \quad (9)$$

Where ρ_h and f_h are homogeneous density and friction factor respectively. Homogeneous viscosity for calculating Reynolds number is calculated using any of the following definitions

Mc Adam *et al*'s (1942) definition

$$\mu_h = \left(\frac{x}{\mu_g} + \frac{1-x}{\mu_l} \right)^{-1} \quad (10)$$

Cicchitti *et al*'s (1960) definition

$$\mu_h = x\mu_g + (1-x)\mu_l \quad (11)$$

Dukler *et al*'s (1964) definition

$$\mu_h = \rho_h \left[\frac{x\mu_g}{\rho_g} + \frac{1-x}{\rho_l} \right] \quad (12)$$

Where homogeneous density is calculated using equation 6

Beattie and Whalley's (1982) definition

$$\mu_h = \alpha_h \mu_g + \mu_l (1-\alpha_h)(1+2.5\alpha_h) \quad (13)$$

3.2. Lockhart-Martinelli Model (1949)

$$\left(\frac{dp}{dz} \right)_f = \Phi_l^2 \left(\frac{dp}{dz} \right)_l \quad (14)$$

Where Φ is two-phase multiplier and is calculated as

$$\Phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (15)$$

Criteria for selecting the value of parameter C is given in table 1.

Table 1
Values of parameter C for Lockhart-Martinelli model

Liquid	**Gas**	**C-value**
Laminar	Laminar	5
Laminar	Turbulent	12
Turbulent	Laminar	10
Turbulent	Turbulent	20

Martinelli parameter X is defined as follows

$$X^2 = \frac{\left(\frac{dp}{dz} \right)_l}{\left(\frac{dp}{dz} \right)_g} \quad (16)$$

With

$$\left(\frac{dp}{dz} \right)_l = \frac{f_l G^2 (1-x)^2}{2\rho_l D} \quad (17)$$

and

$$\left(\frac{dp}{dz} \right)_g = \frac{f_g G^2 x}{2\rho_g D} \quad (18)$$

Where f_l and f_g are calculated according to equations 7 and 8 using Re_l and Re_g as

$$\mathrm{Re}_l = \frac{G(1-x)D}{\mu_l} \quad (19)$$

$$\mathrm{Re}_g = \frac{GxD}{\mu_g} \quad (20)$$

3.3. Grönnerud Correlation (1979)

Frictional pressure drop is given as

$$\left(\frac{dp}{dz} \right)_f = \Phi \left(\frac{dp}{dz} \right)_l \quad (21)$$

Where

$\left(\frac{dp}{dz} \right)_l$ is calculated using equation 17

Two-phase multiplier is defined as

$$\Phi = 1 + \left(\frac{dp}{dz}\right)_{Fr} \left[\frac{\frac{\rho_l}{\rho_g}}{\left(\frac{\mu_l}{\mu_g}\right)^{0.25}} - 1 \right] \tag{22}$$

$$\left(\frac{dp}{dz}\right)_{Fr} = f_{Fr}\left[x + 4(x^{1.8} - x^{10} f_{Fr}{}^{0.5})\right] \tag{23}$$

f_{Fr} is based on liquid Froude number as given in equation 25 below. If Fr_l is greater than or equal to 1 then f_{Fr} is set to 1 else it is given as follows

$$f_{Fr} = Fr_l{}^{0.3} + 0.0055 \left(\ln \frac{1}{Fr_l}\right)^2 \tag{24}$$

$$Fr_l = \frac{G^2}{gD\rho_l{}^2} \tag{25}$$

3.4. Müller-Steinhagen and Heck (1986) correlation

$$\left(\frac{dp}{dz}\right)_f = M(1-x)^{\frac{1}{c}} + Bx^c \tag{26}$$

With C=3

$$M = A + 2(B-A)x \tag{27}$$

$$A = \left(\frac{dp}{dz}\right)_{lo} \tag{28}$$

$$B = \left(\frac{dp}{dz}\right)_{go} \tag{29}$$

Pressure gradients in A and B are defined as

$$\left(\frac{dp}{dz}\right)_{lo} = \frac{f_{lo}G^2}{2\rho_l D} \tag{30}$$

$$\left(\frac{dp}{dz}\right)_{go} = \frac{f_{go}G^2}{2\rho_g D} \tag{31}$$

Where f_{lo} and f_{go} are calculated according to equations 7 and 8 using Re_{lo} and Re_{go} as

$$\mathrm{Re}_{lo} = \frac{GD}{\mu_l}$$

$$\mathrm{Re}_{go} = \frac{GD}{\mu_g}$$

3.5. Mishima and Hibiki correlation (1996)

This correlation is specially developed for micro channels taking into account the diameter of the channel. Lockhart-Martinelli model has been used with a modified value of parameter C given as

$$C = 21(1 - e^{-319D}) \tag{32}$$

Where D is the diameter in meter.

3.6. Lee and Mudawar correlation (2005)

This is also a microchannel correlation taking into account the surface tension force and diameter of the channel. Lockhart-Martinelli model is used with modified form of parameter C as given below

$$C_{LT} = 1.45\,\mathrm{Re}_{lo}{}^{0.25}\,We_{lo}{}^{0.23} \tag{33}$$

$$C_{LL} = 2.16\,\mathrm{Re}_{lo}{}^{0.047}\,We_{lo}{}^{0.6} \tag{34}$$

$$We_{lo} = \frac{G^2 D}{\rho_l \sigma} \tag{35}$$

Where equation 33 is used for laminar liquid and turbulent vapour flow condition and equation 34 is used for laminar liquid and laminar vapour flow condition.

3.7. Tran et al (2000) correlation

This correlation was developed for microchannels and is given below:

$$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{lo} \Phi^2{}_{lo} \tag{36}$$

$$\Phi_{lo}{}^2 = 1 + (CY^2 - 1)\left[N_{co}x^{0.875}(1-x)^{0.875} + x^{1.75}\right] \tag{37}$$

Value of parameter C was found to be 4.3

using 610 data points by a statistical analysis package. Y is defined as follows:

$$Y^2 = \frac{\left(\dfrac{dp}{dz}\right)_{go}}{\left(\dfrac{dp}{dz}\right)_{lo}} \tag{38}$$

Confinement number was used for taking into account the microchannel flow characteristics with the following definition:

$$N_{co} = \left(\frac{\sigma}{D^2 g(\rho_l - \rho_g)}\right)^{0.5} \tag{39}$$

3.8. Zhang and Webb (2001) correlation

A microchannel correlation was developed by Zhang and Webb as follows:

$$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{lo} \Phi_{lo}{}^2 \tag{40}$$

Two-phase multiplier is defined as:

$$\Phi_{lo}{}^2 = (1-x)^2 + 2.87x^2\left(\frac{P}{P_{crit}}\right)^{-1} + 1.68x^{0.8}(1-x)^{0.25}\left(\frac{P}{P_{crit}}\right)^{-1.64} \tag{41}$$

4. Experimental facility and data reduction

Schematic diagram of the test facility is shown in figure 1. Refrigerant in subcooled condition from condenser is circulated with a magnetic gear pump with microprocessor control (type MCP-Z standard). Pump can be run in different modes and gives a wide variety of flow rates depending upon the pump head used. Fluid then passes through a Coriolis effect mass flow meter and afterwards through a pre-heater. The accuracy of the mass flow meter was within ±0.2% for most of the tests but for single phase tests and Reynolds number greater than 2000 the accuracy was ±0.5%. A preheater is used to adjust the inlet temperature of the fluid at a desired level. After passing through pre-heater, the fluid comes into a 7µm filter which removes any small particles and also introduces an extra pressure drop at the inlet of the test section. Fluid then passes through the test section, absorbs heat and flows back to the condenser thereby completing the flow loop. The condenser level defines the system pressure which is controlled by the flow rate of the cooling water. Further adjustment of the system pressure within ±0.5% is done by controlling the electrical input to the heater in a refrigerant tank connected to the main loop.

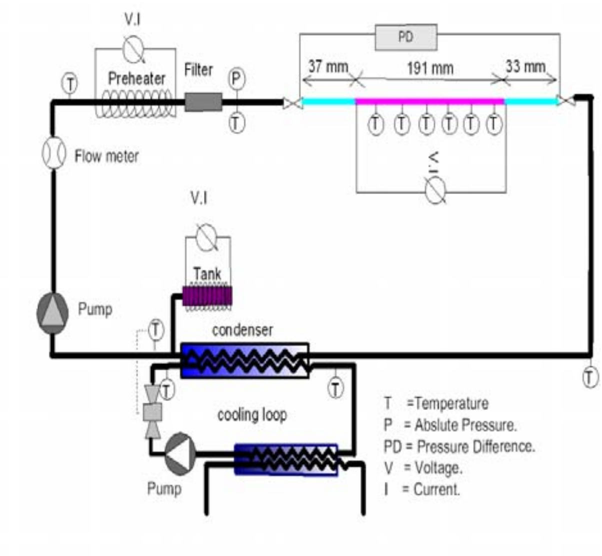

Fig. 1. Schematic diagram of experimental set up

Test facility is instrumented with an absolute pressure transducer (Druck, 25bar) to measure the pressure at the inlet of the test section and a differential pressure transducer (Druck, 350mbar) to measure the pressure drop across the test section. The uncertainty in the system pressure measurements was estimated to be less than ±10mbar. The uncertainty in differential pressure for two-phase flow was estimated to be ±8mbar for extreme case but in most of the points was up to ±5mbar. Six thermocouples were glued on the outer surface of the tube using an electrically insulating and thermally conductive epoxy. Thermocouples were calibrated using an ice bath and were found to be differing up to ±0.1 °C. Apart from the thermocouples attached to the test section, there are some thermocouples inserted at different locations in the experimental setup to measure the fluid bulk temperature. The entire test rig was heavily insulated to minimize the heat losses. For single phase tests the error in the heat balance on the test section was in the worst case up to ±8% but in most of the cases it was less than 5%.

The test tube was made of fused silica with an internal diameter of 781μm and a total length of 261mm with 191mm of heated length. Brass Swage lock connections were used at the two ends of the test section to connect it to the main loop. Two holes on the brass connections just at the inlet and outlet of the test section were drilled for pressure taps.

The single phase friction factor was calculated as

$$f = \frac{64}{\mathrm{Re}} \quad \text{for laminar flow Re<2300} \tag{42}$$

$$f = 0.3164 \times \mathrm{Re}^{-0.25} \quad \text{for turbulent flow Re>2300} \tag{43}$$

where Re is the Reynolds number

For a given test point, the heat flux added to the test section q” was calculated as

$$q'' = \frac{Q - Q_{loss}}{A} \tag{44}$$

$$Q = I \times V \tag{45}$$

where I and V are the current and voltage in ampere and volts respectively and A is the heat transfer area given as $A=\pi DL_{hs}$. Q_{loss} is the heat loss to the ambient, calculated by heating the test section without pumping the fluid: When the wall temperatures became constant, the power was noted. This procedure was repeated and finally the power was correlated with the temperature difference between the wall and the ambient for different values of power and wall temperature. A Yokogawa WT130 power meter was used to measure the voltage and current applied to the test section. This instrument has an accuracy of 0.25%. The overall uncertainty in the effective power (the difference between applied power and power loss) was estimated to be less than 3%.

The thermodynamic vapour quality at any axial position $x_{th}(z)$ was calculated from the heat transferred to the fluid as,

$$x_{th} = \frac{q''\pi D(z - z_{sat})}{A_c G i_{lg}} \tag{46}$$

Where D is the internal diameter of the test tube, z is any axial location, z_{sat} is the location where boiling starts (i.e. saturated conditions are reached), A_c is the cross sectional area of the test tube, G is the mass flux and i_{lg} is the latent heat of vaporization. Axial position where saturated conditions are reached can be calculated as

$$z_{sat} = \frac{mc_p(T_{sat} - T_{in})}{q''\pi D} \tag{47}$$

Where m is the mass flow rate, C_p is the specific heat of the fluid, D is the diameter and T_{in} is the fluid temperature at the inlet of test section. The overall uncertainty in the exit vapor quality was estimated to be up to 3%.

Table 2
Experimental conditions and parameters

Parameter	Operating range
D [mm]	0.781
G [kg/m^{2}sec)	180,270,400,500,650
q” [kW/m^2]	7 to 60
T_{sat} [^{o}C]	30
$\Delta T_{sub,i}$ [^{o}C]	7,8,5,5,5

The total pressure drop in the test section is the sum of the two-phase frictional pressure drop, momentum pressure drop and single phase pressure drop from inlet up to the position where boiling starts i.e. z_{sat}

$$\Delta P_t = \Delta P_{sp} + \Delta P_{tp} + \Delta P_m \tag{48}$$

Momentum pressure drop can’t be neglected in these tests as it is considerable along the test section due to change in vapor fraction from inlet to outlet and is accounted for using equation 3.

5. Experimental results and discussion

Single phase tests were conducted prior to the two-phase tests to validate the instrumentation and experimental method. Single phase friction factor plotted against Re is shown in figure 2. The experimental results confirm that single phase classical correlation $f=64/Re$ in laminar region and $f=0.3164\times Re^{-0.25}$ suggested by Blasius (1913) for turbulent region can still be applied in microchannels.

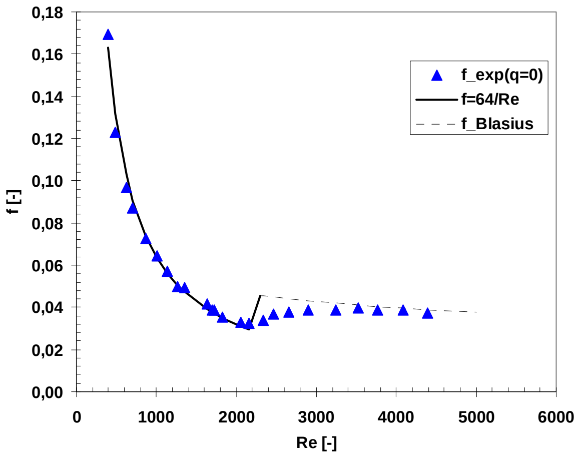

Fig. 2. Single phase friction factor against Reynolds number

Variation of total two-phase pressure drop with heat flux for different mass fluxes is shown in figure 3. It can be observed that for a given mass flux the total pressure drop increases with increase in heat flux. Increase in heat flux increases the vapor fraction and void fraction which makes the pressure drop higher for a higher heat flux. For a given heat flux the total pressure drop is higher for a higher mass flux. This is also an expected trend since increasing mass flux increases the shear stress at the tube wall causing a higher pressure drop. For all mass fluxes the increase in pressure drop is more or less linear.

Effect of mass flux and exit vapor quality on two phase frictional pressure drop is shown in figure 4. For a given mass flux two phase frictional pressure drop increases with increase in vapor quality, as expected. Increased velocity of vapor and liquid due to vapor quality causes the increase in pressure drop. For a given vapor quality, the higher the mass flux, the higher is the pressure drop. This variation of pressure drop with mass flux is in agreement with studies found in literature. Liquid and vapor velocity in two-phase flow are directly proportional to the mass flux of refrigerant which will also be expected to affect the frictional pressure drop.

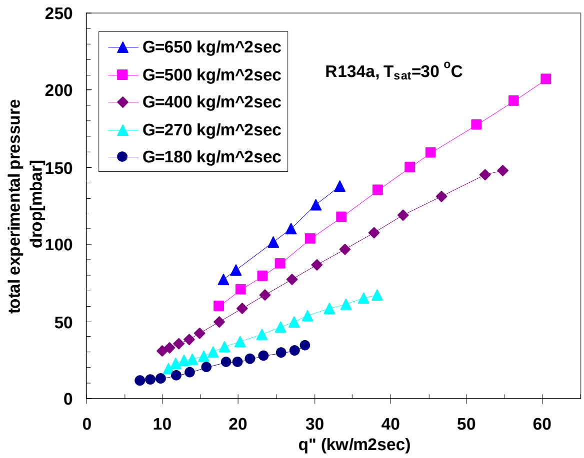

Fig. 3. Variation of total pressure drop with heat flux for different mass fluxes

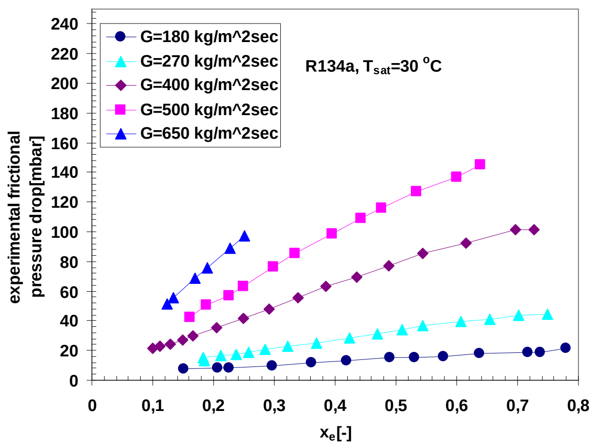

Fig. 4. Variation of frictional pressure drop with exit vapor quality for different mass fluxes

6. Comparison with correlations

6.1 Macro scale correlations

Experimental data have been compared with Macro scale and micro scale correlations to explore the available methods for predicting the two-phase pressure drop. First, the homogeneous model is used with different definitions of the mixture viscosity. Table 3 shows the assessment of the homogeneous model with various definitions of homogeneous viscosity. Cicchitti et al's (1960) viscosity model works better in comparison to other models. It predicts the data with a MAD of 19% with 69% of the data within ±20% error band. Comparison with cichhitti et al's model is shown in figure 5. Similar trends could also be plotted for other viscosity models. It was observed that almost all of the data is predicted within ±50% error band by all the viscosity models with a trend that most of the data is under predicted. The homogeneous model assumes the two phases moving with the same velocity and is believed to work better in bubbly flow regime, while in micochannels due to the limitation of the diameter, bubbly flow doesn't prevail for a long time and soon after departing the bubbles coalesce and form large slugs. Also, the dominant flow regimes are slug and annular, and in microchannels, the transition to annular flow regime is early (at low vapor fractions) as compared to for macrochannels. These reasons could explain why homogeneous model under predicts the experimental data.

Table 3
Assessment of the Homogeneous model with different viscosity definitions

Viscosity Definition	MAD (%)	% of data within ±20%
McAdams (1942)	29	13
Cicchitti (1960)	19	69
Dukler (1964)	33	9
Beattie & Whalley (1981)	22	37

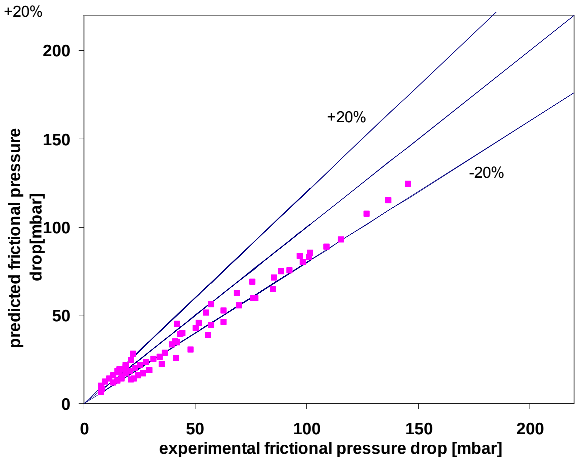

Fig. 5. Homogeneous model using Cicchitti's equation for homogeneous viscosity at T_{sat}=30 °C

Table 4 shows the assessment of all the macroscale correlations used for comparison in this study including the homogeneous model with Cichhitti et al's homogeneous viscosity definition.

The Lockhart-Martinelli model is widely used for predicting the two-phase frictional pressure drop in macro tubes and many microscale pressure drop correlations are modifications of this model. As can be seen in table 4, this model predicts the data with a MAD of 98% with none of the predicted data within ±20% error band. It can also be shown that this model over predicts the data unlike the homogeneous model. The Lockhart-Martinelli model uses a so called two-phase multiplier for correlating the pressure gradient as explained earlier in this study. The two phase multiplier is calculated using the Martinelli parameter X and a parameter C. The value of parameter C ranges from 5 to 20 depending upon the flow regime as shown in table 1. In fact parameter C has a great influence on the two phase multiplier. Martinelli and co-workers suggested C values for macro channels and for their low pressure data and the present experimental data has been taken at higher pressures. Also factors dominant in macrochannels could be less important in microchannels and the C value shall be calculated taking into account parameters having dominant effect in microchannel flow (like surface tension and effect of diameter). These factors could possibly explain the deviation of experimental data from the Lockhart-Martinelli model.

Friedel's (1979) correlation predicts the data with a MAD of 55% with 42% of the data within ±20% error band. It can be shown that this correlation works well for high mass fluxes (G=400 and 500 kg/m^{2}sec) but none of the experimental data at low mass fluxes falls with in ±20% range. Possible reason for deviation could be that this correlation was developed for macrochannels where the flow is almost invariably turbulent even at low mass fluxes and this will rarely be the case for the mass flux range covered in this study. Grönnerud's (1979) correlation, which was developed for macro channels and specifically for refrigerants, predicts the data with a MAD of 31% with 34% of the data within the ±20% error band. It can also be shown that most of the predicted data lies within ±50% error band.

Table 4
Assessment of macroscale correlations

Correlation	MAD (%)	% of data within ±20%
Homogeneous (Cicchitti) (1960)	19	69
Lockhart-Martinelli (1949)	98	0
Müller-Steinhagen and Heck (1986)	19	61
Friedel (1979)	55	42
Grönnerud (1979)	31	34

Müller-Steinhagen and Heck (1986) correlation predicts the data best after the homogeneous model. This correlation predicts the data with a MAD of 19% with 61% of the data within the ±20% error band. . The comparison can be seen in figure 6. This correlation was developed using a vast data bank (9300 data points) containing different fluids and a wide range of diameters as low as 4mm and is quite simple and easier to use than other methods, and predicts the data reasonably well. Revellin and Thome (2007) also found that Müller-Steinhagen and Heck (1986) correlation predicted their microchannel data well. Cavallini et al (2005) also found good match of experimental data and the data predicted by this correlation.

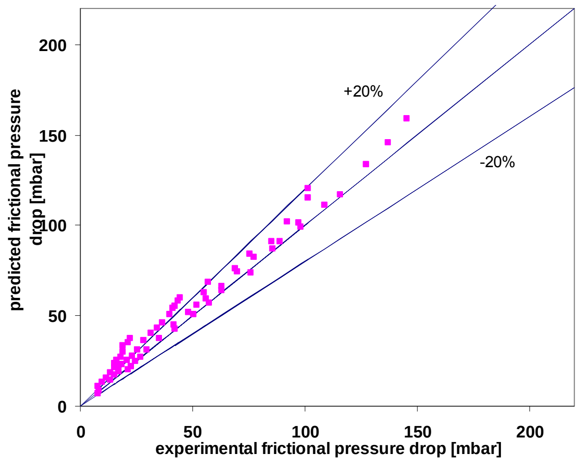

Fig. 6. Comparison of experimental data with Muller-Steinhagen and Heck correlation, T_{sat}=30 ^{o}C

6.2 Microscale correlations

Microscale correlations used for comparison are shown in table 5. The Mishima and Hibiki (1996) correlation well predicts the data as compared to other microscale correlations with a MAD of 19% and 57% of data within ±20% error band. Figure 7 shows the plot of experimental and predicted data using Mishima and Hibiki (1996) correlation. Owhaib et al (2008) also found that Mishima and Hibiki (1996) and Müller-Steinhagen and Heck (1986) correlations predicted their results better than other correlations.

Table 5.
Assessment of microscale correlations

Correlation	MAD (%)	% of data within ±20%
Mishima and Hibiki (1996)	19	57
Tran et al (2000)	40	42
Zhang and Webb (2001)	44	30
Lee and Mudawar (2005)	145	0

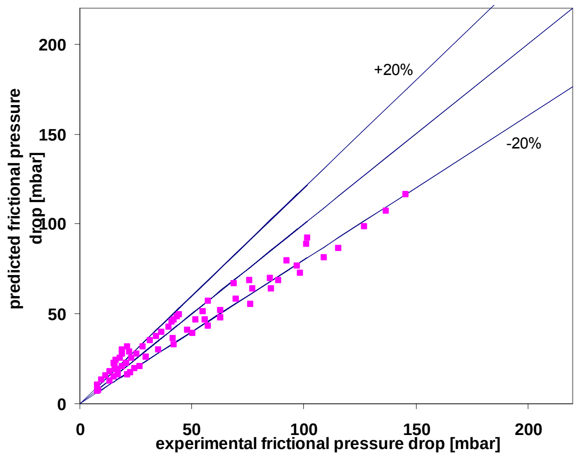

Fig. 7. Comparison of experimental data with Mishima and Hibiki correlation, T_{sat}=30 ^{o}C

Conclusions

An experimental study was conducted to explore the two-phase pressure drop characteristics of a microchannel. Data obtained was also compared with both macroscale and microscale pressure drop correlations. The main findings of the study are summarized below:

- Single phase tests were performed prior to two-phase tests and the results confirm the applicability of single phase classical theory to microchannels.
- Two-phase pressure drop increases approximately linearly with both heat flux and vapor fraction.
- Homogeneous model and Müller-Steinhagen and Heck (1986) correlation are in better agreement with experimental data as compared to other macroscale models used in the study.
- Mishima and Hibiki (1996) correlation developed for microchannels also predicts the data better as compared to other microscale correlations considered in the study.

Nomenclature

A	heat transfer area (m^2)
A_c	cross sectional area (m^2)
C_p	specific heat (J/kg K)
D	diameter (m)
f	friction factor
G	mass flux (kg/m^2s)
I	current (A)
i_{lg}	latent heat of vaporization (J/kg)
MAD	mean absolute deviation, $1/N\ \Sigma\lvert\Delta_{pred} - \Delta P_{exp}\rvert/\Delta_{exp}$ (%)
m	mass flow of refrigerant (kg/s)
P	pressure (bar)
P_{crit}	critical pressure (bar)
Q	power (W)
Q_{loss}	power loss (W)
q''	heat flux (W/m^2)
T	temperature (ºC)
V	voltage (V)
x_{th}	thermodynamic vapor quality (-)
z	any axial position (m)

Greek letters

α	void fraction
μ	dynamic viscosity (Ns/m^2)
ρ	density (kg/m^3)
σ	surface tension (N/m)
ΔP	pressure loss (mbar)
ΔT_{sub}	subcooling degree, T_{sat}-T (ºC)

Subscripts

exp	experimental
f	*frictional*
g	gas
gr	*gravitational*
h	homogeneous
hs	heated section
in	inlet
l	liquid
m	momentum
out	outlet
pred	predicted
sat	saturation
sp	single phase
t	*total*
tp	two phase

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