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TWO-PHASE PRESSURE DROP OF AMMONIA IN A MINI/MICRO-CHANNEL

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ABSTRACT

Experiments have been performed to investigate two-phase pressure drop in a circular vertical mini-channel made of stainless steel (AISI 316) with internal diameter of 1.70 mm and a uniformly heated length of 245 mm using ammonia as working fluid. The experiments are conducted for heat flux range of 15 to 350 kW/m^2 and mass flux range of 100 to 500 kg/m^{2}s. A uniform heat flux is applied to the test section by DC power supply. Two phase frictional pressure drop variation with mass flux, vapour quality and heat flux was determined. The experimental results are compared to predictive methods available in literature for frictional pressure drop. The Homogeneous model and the correlation of Müller Steinhagen et al. [14] are in good agreement with our experimental data with MAD of 27% and 26% respectively.

Keywords: Pressure Drop, Two Phase, Mini channel

1. INTRODUCTION

The two phase pressure drop is an important parameter in designing engineering appliances. Micro and mini channels fluid flow has become an important topic because of its applications in microelectronics, bioengineering, micro turbines, compact refrigeration systems etc. The merits of micro channels are their higher heat transfer capabilities and compactness but at the same time one demerit may be larger pressure drop.

The interest in mini channel heat exchangers during the last decade has resulted in a large number of experimental studies of two-phase pressure drop. The majority of these studies have been performed using HFC refrigerants with fairly similar properties. There are still discrepancies which need to be clear despite great efforts made by researchers to estimate pressure drop during flow boiling in mini and micro channels. It is important to do tests with different fluids so that on the basis of experimental observations, a generalized predictive method can be developed for two phase pressure drop in mini and micro channels.

Due to the need of experimental observations of different fluids, ammonia pressure drop results are reported in this article. To get advantage of the favourable properties of ammonia, pressure drop is a crucial parameter which needs to be studied. One of the objectives of this study is to determine if macro channels correlation still can be used for prediction of frictional pressure drop in mini/micro channels.

2. LITERATURE SURVEY

In literature, experimental studies using ammonia in micro channels are rare. Several studies in macro scale are available. Some of the experimental studies using ammonia and other fluids are mentioned here.

Kabelac et al. [2] performed experiments using ammonia in smooth and finned test sections with 10 mm internal diameter and a length of 450 mm. Test were performed at mass flux range of 50 to 150 kg/m^{2}s, vapor quality range of 0 to 0.9, saturation temperature range of -40 to 4 ^{o}C and heat flux range of 17 to 75 kW/m^2. When compared with correlations, it was observed that Chisholm [3] correlation predicted the data of heated test section quite well.

Triplett et al. [4] investigated two phase pressure drop and void fraction in micro channels using mixtures of water and air as fluid. For bubbly and slug flow, homogeneous model predicted the experimental data well. Homogeneous model and other correlations over predicted the data in annular flow regime. Pamitran et al. [5] performed experiments for two phase pressure drop using natural refrigerant CO_2 as fluid. Results were compared with 13 prediction methods and a new prediction method based on Lockhart-Martinelli [6] correlation was developed which predicted that data with mean deviation of 9.41%.

Hwang et al. [7] conducted experiments using R -134a as test fluid. Existing correlations did not predict their data well. They developed a new correlation based on Lockhart-Martinelli [6] correlation. The new correlation predicted experimental data with absolute average deviation of 8.1%.

Owhaib et al. [8] conducted tests using R-134a in three vertical mini channels of diameters 1.70 mm, 1.224 mm and 0.824 mm in the same test rig used in this study. Results were compared with 5 macro scale and 8 micro scale correlations. Müller Steinhagen and Heck [14] for macro scale and Mishima and Hibiki [16] for micro scale predicted the data with MAD of 30% and 22% respectively. Ali et al. [9] also did experiments in same test facility but for horizontal glass test section of 0.781mm. It was observed that two phase pressure drop increases linearly with increase of heat flux and vapour fraction.

3. FRICTIONAL PRESSURE DROP CORRELATIONS

Two types of models are available for two phase frictional pressure drop in literature. One is homogeneous model which is also called zero slip model. In homogeneous model, the liquid and gas phases are assumed to have the same velocity. In this model, only one phase is considered with average properties. Second is separated flow model also called slip flow model. In this model, liquid and gas are considered to be separate entities. Uniform properties are assumed in each phase. There are many correlations in open literature but some of them which are used in this study are explained in table 1.

Table 1 . Pressure drop correlations of macro and micro scale

Correlation	Equation
Homogeneous model	$\left(\frac{dp}{dz}\right)_f = \frac{2f_{TP}G^2}{D\rho_{TP}}$ $\rho_{TP} = \left(\frac{x}{\rho_V} + \frac{1-x}{\rho_L}\right)^{-1}$ $f_{TP} = \frac{16}{Re_{TP}}\ for\ Re_{TP} < 2000\ or$ $f_{TP} = 0.079\ Re_{TP}{}^{-0.25} for\ Re_{TP} > 2000$ $Re_{TP} = \frac{GD}{\mu_{TP}}$ Three forms of viscosity models are; McAdams et al. [10] $\mu_{TP} = \left(\frac{x}{\mu_V} + \frac{1-x}{\mu_L}\right)^{-1}$ Chichitti et al. [11] $\mu_{TP} = (x\mu_V + (1-x)\mu_L)$ Dukler et al. [12] $\mu_{TP} = \rho_{TP}\left(x\frac{\mu_V}{\rho_V} + (1-x)\frac{\mu_L}{\rho_L}\right)$
Friedel [13]	$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{LO}\Phi_{LO}^2$ $\Phi_{LO}^2 = E + \frac{3{,}24\ FH}{Fr^{0.045}We^{0.035}}$ $Fr = \frac{G^2}{gD\rho_H^2}, F = x^{0.78}(1-x)^{0.224}$ $H = \left(\frac{\rho_L}{\rho_V}\right)^{0.91}\left(\frac{\mu_V}{\mu_L}\right)^{0.19}\left(1-\frac{\mu_V}{\mu_L}\right)^{0.7}$ $We = \frac{G^2D}{\sigma\rho_H}\ with\ \rho_H = \left(\frac{x}{\rho_V} + \frac{1-x}{\rho_L}\right)^{-1}$ $E = (1-x)^2 + x^2\frac{\rho_L f_{VO}}{\rho_V f_{LO}}$ $f_{LO} = \frac{16}{Re_{LO}}\ for\ Re_{LO} < 2000\ or$ $f_{LO} = 0.079\ Re_{LO}{}^{-0.25}$ $for\ Re_{LO} > 2000$ $f_{VO} = \frac{16}{Re_{VO}}\ for\ Re_{VO} < 2000\ or$ $f_{VO} = 0.079\ Re_{VO}{}^{-0.25}\ for\ Re_{VO} > 2000$ $Re_{LO} = \frac{GD}{\mu_L} and\ Re_{VO} = \frac{GD}{\mu_V}$ $\left(\frac{dp}{dz}\right)_{LO} = f_{LO}\frac{2G^2}{D\rho_L}$

Lockhart- Martinelli [6]	$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_L \Phi_L^2$ $\Phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ with C given by: $C_{tt} = 20, C_{Vt} = 12, C_{tV} = 10, C_{VV} = 5$ $X = \left[\frac{\left(\frac{dp}{dz}\right)_L}{\left(\frac{dp}{dz}\right)_V}\right]^{0.5}$ $with \left(\frac{dp}{dz}\right)_L = f_{LO}\frac{2G^2}{D\rho_L}(1-x)^2$ $\left(\frac{dp}{dz}\right)_V = f_V \frac{2G^2}{D\rho_L}(x)^2$ $f_L = \frac{16}{Re_L}\ for\ Re_L < 2000\ or$ $f_L = 0.079\ {Re_L}^{-0.25}\ for\ Re_L > 2000$ $f_V = \frac{16}{Re_V}\ for\ Re_V < 2000\ or$ $f_V = 0.079\ {Re_V}^{-0.25}\ for\ Re_V > 2000$ $Re_L = \frac{GD}{\mu_L}(1-x) and\ Re_V = \frac{GD}{\mu_V}x$
Müller Steinhagen and Heck [14]	$\left(\frac{dp}{dz}\right)_f = \mathrm{F}(1-x)^{1/3} + \left(\frac{dp}{dz}\right)_{VO} x^3$ With $F = \left(\frac{dp}{dz}\right)_{LO} + 2\left[\left(\frac{dp}{dz}\right)_{VO} - \left(\frac{dp}{dz}\right)_{LO}\right]x$
Grönnerud [15]	$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_L \Phi_L^2$ $\left(\frac{dp}{dz}\right)_L = f_{LO}\frac{2G^2}{D\rho_L}(1-x)^2$ $\Phi_L^2 = 1 + \left(\frac{dp}{dz}\right)_{Fr}\left[\frac{\frac{\rho_L}{\rho_V}}{\left(\frac{\mu_L}{\mu_V}\right)^{0.25}} - 1\right]$ $\left(\frac{dp}{dz}\right)_{Fr} = f_{Fr}\left[x + 4(x^{1.8} - x^{10}{f_{Fr}}^{0.5})\right]$ $Fr_L = \frac{\mathrm{G}^2}{\mathrm{D}{\rho_L}^2\mathrm{g}}$ If $Fr_L > 1\ then\ f_{Fr} = 1$ $If\ Fr_L < 1$ $then\ f_{Fr} = {Fr_L}^{0.3} + 0.0055\left(\ln\frac{1}{Fr_L}\right)^2$
Chisholm [3]	$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{LO} \Phi_{LO}^2$ $\Phi_{LO}^2 = 1 + (Y^2-1) * [Bx^{0.875}(1-x)^{0.875} + x^{1.75}]$ $Y^2 = \frac{\left(\frac{dp}{dz}\right)_{VO}}{\left(\frac{dp}{dz}\right)_{LO}}$ $and \left(\frac{dp}{dz}\right)_{VO} = f_{VO}\frac{2G^2}{D\rho_V}$ $If\ 0 < Y < 9.5\ then$ $B = \frac{55}{G^{0.5}} for\ G \geq 1900\frac{kg}{m^2s}$ $B = \frac{2400}{G}\ for\ 500 < G\ < 1900\frac{kg}{m^2s}$ $B = 4.8\ for\ G < 500\ \frac{kg}{m^2s}$ If $9.5 < Y < 28\ then$ $B = \frac{520}{YG^{0.5}} for\ G \leq 600\frac{kg}{m^2s}$ $B = \frac{21}{Y} for\ G > 600\frac{kg}{m^2s}$ For Y > 28 $\mathrm{B} = \frac{15000}{\mathrm{Y}^2\mathrm{G}^{0.5}}$
Mishima and Hibiki [16]	$C = 21(1 - e^{-319D})$ This C value should be used with Lockhart-Martinelli Correlation.
Zhang and Webb [17]	$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{LO} \Phi_{LO}^2$ $\Phi_{LO}^2 = (1-x)^2 + 2.87x^2\left(\frac{P}{P_{crit}}\right)^{-1}$ $+1.68x^{0.8}(1-x)^{0.25}\left(\frac{P}{P_{crit}}\right)^{-1.64}$
Tran et al. [18]	$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_{LO} \Phi_{LO}^2$ $\Phi_{LO}^2 = 1 + (4.3Y^2 - 1)$ $[Cox^{0.875}(1-x)^{0.875} + x^{1.75}]$ $Co = \left(\frac{\sigma}{D^2 g(\rho_L - \rho_V)}\right)^{0.5}$ $and\ Y^2 = \frac{\left(\frac{dp}{dz}\right)_{VO}}{\left(\frac{dp}{dz}\right)_{LO}}$

4. EXPERIMENTAL FACILITY AND DATA REDUCTION

The experimental apparatus is schematically illustrated in Figure 1. The refrigerant coming from the sub cooler was pumped by a magnetic gear pump, type MCP-Z standard, to the test section. This pump allows a wide range of flow rates. The circuit included a Coriolis mass flow meter to measure the flow rate. To adjust the inlet temperature of test section, a pre heater was used. A filter of 7 micro meters was used to restrict any particles to enter the test section.

An absolute pressure transducer (Druck, 25bar) was used to measure the system pressure and the pressure drop across the test section was measured by a differential pressure transducer (Druck, 350mbar). The test section consists of metal (AISI 316 stainless steel) tube with inner diameter of 1.70 mm.

Ten T-type thermocouples were mounted on the surface of the test section to measure the wall temperature. The tip of each thermocouple was electrically insulated and then attached at the outer wall with special epoxy which is thermally conductive and electrically insulating. Temperatures were measured at outer wall then calculation was done to get inner wall temperature. To measure temperature at the inlet of test section, at the outlet of test section and at different system points, T type thermocouples of 0.1mm diameter were installed.

The test section was heated using an electric DC power supply by applying a potential difference over the test tube itself. This direct heating ensured homogeneous heat flux over the test section. After the test section, the fluid was condensed in the condenser and further sub cooled in sub cooler. As a first step, mass flow, pressure and inlet temperature was set then electric power was applied step by step. Data was recorded for each step when steady state conditions were achieved. The temperatures, the mass flow and the system pressure were recorded using a data logger connected to a computer. Thermal and transport properties of ammonia were taken from REFPROP 7.

Heat loss calculations are done. For low heat fluxes up to 50 kW/m^2 , heat loss varies from 1 to 2% of the applied power and above 50 kW/m^2, heat loss is less than 1% of the applied power therefore it is ignored in this article.

For a given test, the heat flux added to the test section is calculated as;

$$q'' = \frac{Q}{A} \qquad (1)$$

Where

$$Q = IV \qquad (2)$$

And

$$A = \pi D L_h \qquad (3)$$

where I and V are the current and voltage, A is the heat transfer area, D is the inner diameter of test section and L_h is the heated length. At the inlet of test section, there is sub cooling of 1K.

The thermodynamic vapor quality at any vertical location (z) is calculated as;

$$x(z) = \frac{q'' P (z - z_\circ)}{A_c G i_{fg}} \qquad (4)$$

Where

$$z_\circ = \frac{\dot{m}_{NH_3} C_p (T_{sat} - T_{in})}{q'' P} \qquad (5)$$

C_p is the specific heat of the fluid, $\dot{m}$ is the mass flow rate of ammonia, T_{sat} is the saturation temperature, T_{in} is the inlet temperature of test section, q'' is the heat flux, A_c is cross-sectional area and P is the perimeter. z_o is the location on the heated section at which saturated conditions would be reached.

The measured pressure drop is the sum of components which are mentioned in the following formula;

$$(\Delta P)_{measured} = (\Delta P)_{ic} + (\Delta P)_{oe} + (\Delta P)_{tp} + (\Delta P)_{sp} \qquad (6)$$

$(\Delta P)_{ic}$ represents single phase pressure drop due to contraction at inlet which can be calculated as [22];

$$(\Delta P)_{ic} = \xi_{ic} \frac{\rho(u_2^2 - u_1^2)}{2} \qquad (7)$$

$\xi_{ic} = f(\frac{lam}{turb}, \frac{A_2}{A_1})$, drag Compression factor

$(\Delta P)_{oe}$ represents pressured drop due to outlet expansion which can be calculated as [22];

$$(\Delta P)_{oe} = \frac{G^2 \sigma (1-\sigma)}{\rho_l} \left[\frac{x_{exit}^2}{\alpha_{exit}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{exit})^2}{(1 - \alpha_{exit})} \right] \qquad (8)$$

Where $\sigma = A_1 / A_2$

$(\Delta P)_{sp}$ represents single phase pressure drop at the inlet of test section.

$$(\Delta P)_{sp} = \left|(\Delta P)_{sp}\right|_{fric} + \left|(\Delta P)_{sp}\right|_{grav} \qquad (9)$$

Where $\left|(\Delta P)_{sp}\right|_{grav} = g L_{sp} \rho_l$

$$\left|(\Delta P)_{sp}\right|_{fric} = f \frac{\rho_l u^2}{2} \frac{L_{sp}}{D_i}$$

If Re<2300 then $f_{lam} = 64/Re$

If Re>2300 then $f_{turb} = 0,316/Re^{0,25}$

$(\Delta P)_{tp}$ represents pressure drop due to two phase flow and it includes frictional, gravitational and acceleration part;

$$(\Delta P)_{tp} = (\Delta P)_{fric} + (\Delta P)_{grav} + (\Delta P)_{acc} \qquad (10)$$

$$\left|(\Delta P)_{tp}\right|_{grav} = g L_{tp} \frac{1}{x_{exit}} \int_{x_{in}}^{x_{out}} \left[\alpha \rho_g + (1 - \alpha) \rho_l \right] dx \qquad (11)$$

$$\left|(\Delta P)_{tp}\right|_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{exit}{}^2}{\alpha_{exit}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{exit})^2}{(1 - \alpha_{exit})} - 1 \right] \qquad (12)$$

Where definition of α is from Zivi's model [19]

$$\alpha_{exit} = \left[1 + \left(\frac{1 - x_{exit}}{x_{exit}} \right) \left(\frac{\rho_g}{\rho_l} \right)^{2/3} \right]^{-1}$$

Frictional two phase pressure drop can be determined by subtracting gravitational and acceleration pressure drop from $(\Delta P)_{tp}$.

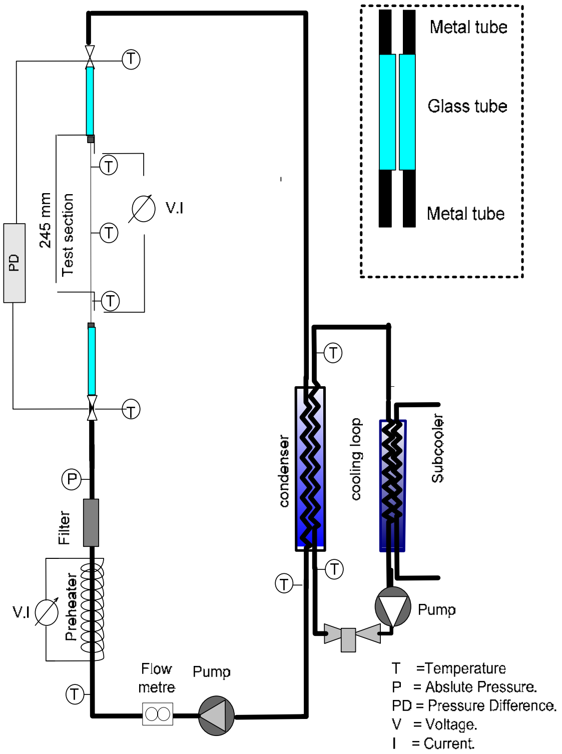

Figure 1. Schematic diagram of experimental test rig

5. RESULTS AND DISCUSSION

Figure 2 shows the two phase frictional pressure drop versus exit vapor quality for several mass fluxes and constant inlet sub cooling of 1K. It can be seen that frictional pressure drop is higher for higher vapor quality and for higher mass flux. For a given outlet quality, pressure drop is higher for higher mass flux as expected. Similar results were reported by Hwang et al. [7] and Lie et al. [20]. It can also be seen that for higher mass flux, frictional pressure drop increase sharply with small increase in quality.

The frictional pressure drop is plotted versus mass flux for a large range of heat fluxes in Figure 3. It can be seen from this figure also that pressure drop is higher for higher mass flux which may be due to higher shear effect on walls. For a given mass flux, pressure drop is higher for higher heat flux which can be explained by increase in quality for increasing heat flux, which ultimately increases pressure drop. This was also observed by Owhaib et al. [8] and Ali et al. [9].

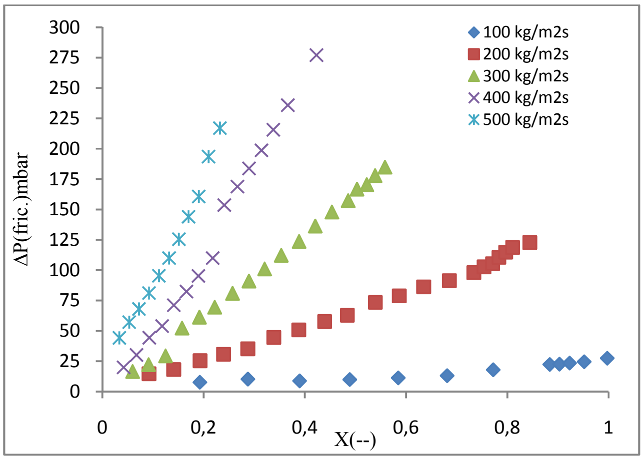

Figure 2. Frictional pressure drop versus exit quality for different mass fluxes.

6. COMPARISON WITH CORRELATIONS

The experimental data was compared with macro and micro scale correlations and results for all correlations are presented in table 2.

An interesting comparison is shown in the right hand column of table 2. First, a second degree polynomial was used to fit the plot of predicted data vs experimental data. Then, the MAD of the polynomial plotted vs the predicted data was calculated. This number then shows the coherence of the predicted data.

Figure 4 shows the experimental two phase frictional pressure drop plotted versus predicted frictional pressure drop by homogeneous model using viscosity definition of Chicchitti et al. [11]. It predicts the data with MAD of 27%.

Figure 5 shows the comparison of experimental two phase frictional pressure drop with Friedel correlation [13]. This correlation predicts the data with MAD OF 29%. This correlation is developed for macro scale and in macro scale, flow regime is turbulent even at lower mass fluxes in contrary to micro scale where flow regime is laminar at lower mass fluxes.

Figure 6 shows the comparison of Müller Steinhagen et al. [14] correlation with experimental frictional pressure drop. It is shown by Tribbe et al. [21] by using large data bank that this correlation is one of the best predicting methods of two phase pressure gradient. This is confirmed by our measurements as the correlation is able to predict our experimental data well with MAD of 26%.

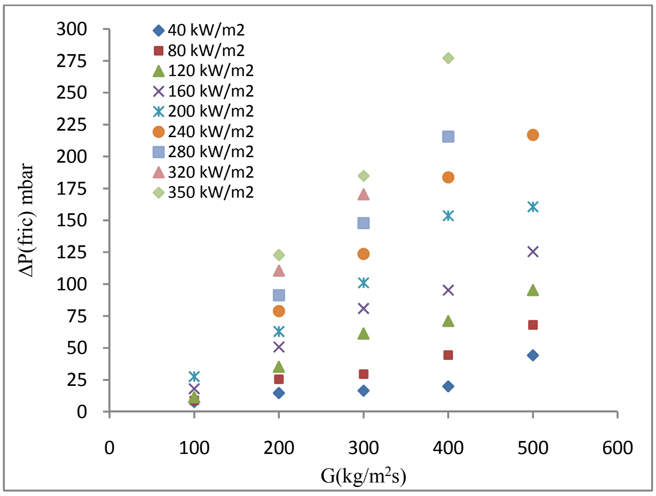

Figure 3. Frictional pressure drop versus mass flux for different heat fluxes at T_{sat}=23°C

Table 2. Assessment of correlations

Correlartion	%MAD	%MAD with itself
Homogeneous	27	11
Friedel [13]	29	13
Lockhart-Martinelli [6]	36	22
Müller Steinhagen and Heck [14]	26	8
Grönnerud [15]	49	16
Chisholm [3]	106	17
Mishima and Hibiki [16]	34	12
Zhang and Webb [17]	39	12
Tran et al. [18]	181	9

Comparison of Lockhart Martinneli [6] with our experimental data is presented in Figure 7. This correlation is not able to predict our data well with MAD of 36%. Most of the micro channel correlations are based on this correlation. Many researchers developed correlations using this correlation but with different definitions of the C value.

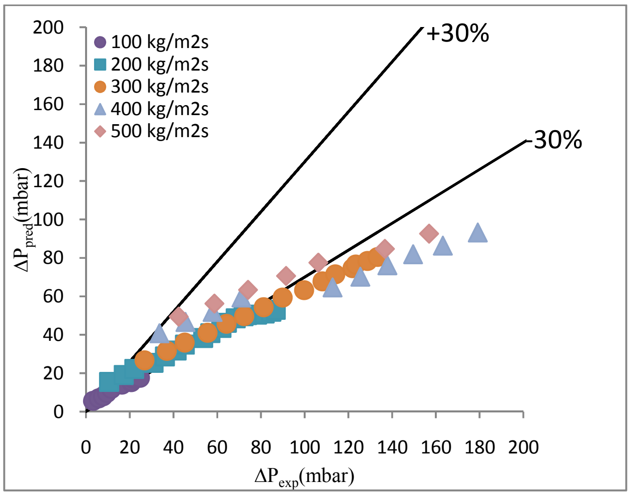

Figure 4. Comparison of two phase frictional pressure drop with homogeneous model.

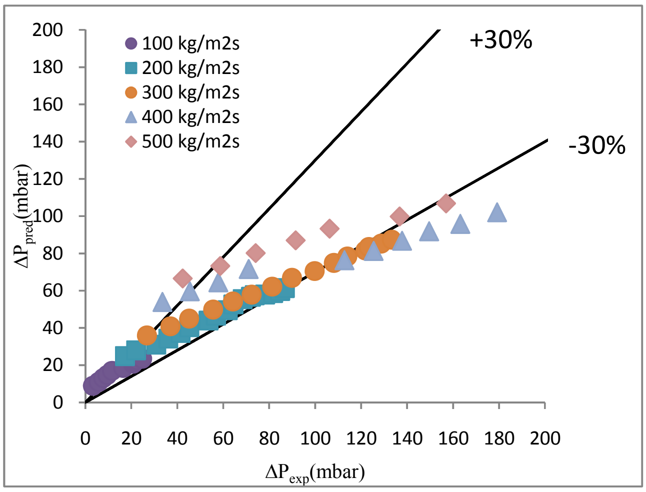

Figure 5. Comparison of two phase frictional pressure drop with Friedel correlation [13]

Predictions of the micro channel correlation suggested by Mishima and Hibiki [16] are shown in Figure 8. In this correlation, the constant C is dependent on diameter. Otherwise, this correlation is identical to the Lockhart Martinelli correlation [6]. The inclusion of diametric effects is the reason that this correlation works well in most pressure drop studies for mini and micro scale studies. In this case, it did not work well with MAD of 34% which is contrary to the results in our previous studies [8, 9] with other fluids.

Figure 9 shows the experimental two phase frictional pressure drop plotted versus predicted frictional pressure drop by the correlation of Zhang and Webb [17]. This correlation predicts our data with MAD of 39%.

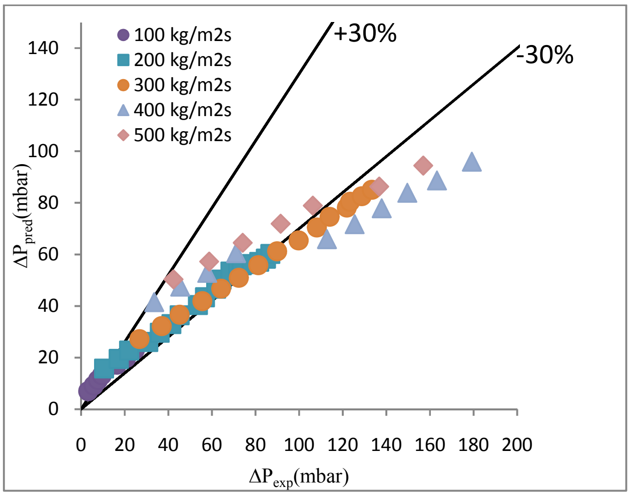

Figure 6. Comparison of two phase frictional pressure drop with Müller Steinhagen et al. [14] correlation

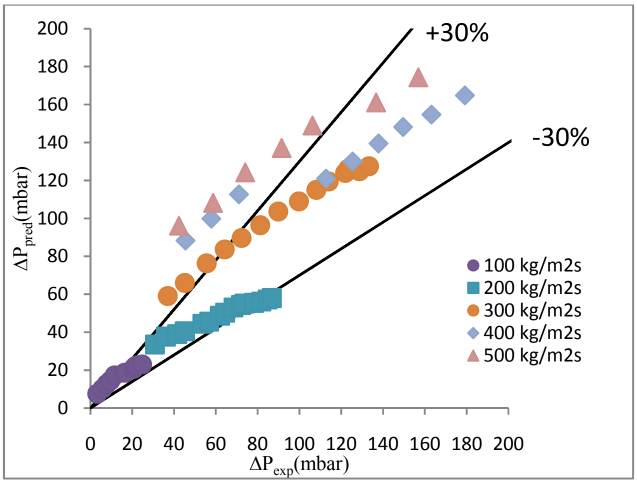

Figure 7. Comparison of two phase frictional pressure drop with Lockhart Martinneli correlation [6]

Only few correlations over predict the experimental data. One example is the correlation by Tran et al. [18] which over predicts the data with MAD of 181% but on the other hand the data is very coherent, with MAD of 9% when the polynomial fit is plotted vs the predicted data. This shows that this correlation, as is also some of the others, is capturing the changing phenomena in two phase flow with the increase of heat flux and mass flux. This may be explained by the fact that this correlation is incorporating the surface tension effects and confinement number. These two factors gain importance when we go from macro to micro scale and are therefore crucial to any predicting method for small diameter channels.. The prediction of Tran et al. [18] is shown in figure 10.

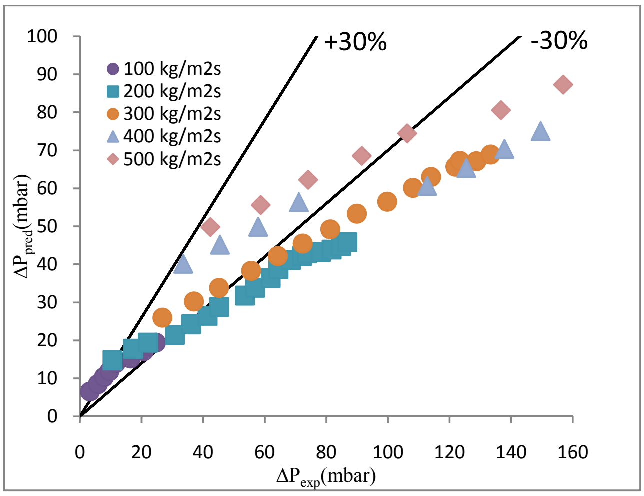

Figure 8. Comparison of two phase frictional pressure drop with Mishima and Hibiki correlation [16]

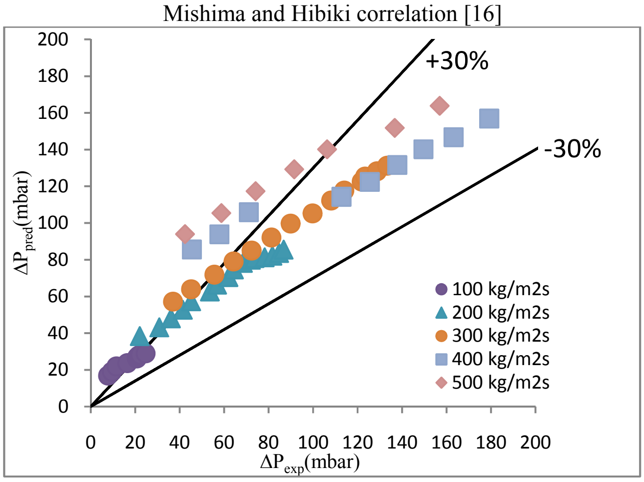

Figure 9. Comparison of two phase frictional pressure drop with Zhang and Webb correlation [17]

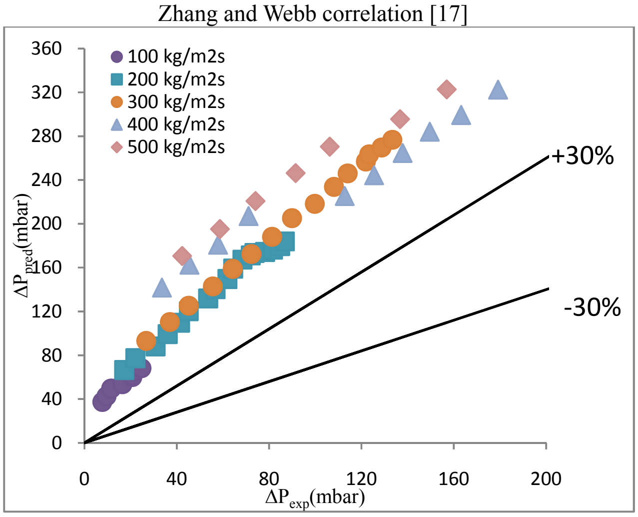

Figure 10. Comparison of two phase frictional pressure drop with Tran et al. [18] correlation

CONCLUSION

The experimental pressure drop for two phase flow of ammonia in vertical circular channel of internal diameter of 1.70 mm are reported in this article. The main conclusions are;

- The experimental data was compared to predictions of several correlations for micro- and macro-channels from the literature. Most correlations under predicts the measured pressure drop, especially at high heat flux or high average vapor fraction.
- For most correlations, the plots of predicted vs experimental pressure drop are not coherent for different mass fluxes.
- The correlations by Müller Steinhagen et al. [14] and homogeneous model for macro channels are in best agreement with the experimental data.
- No micro scale correlation predict our experimental data well.
- Further experimental studies with smaller diameter tubes should be conducted to fully understand the behavior of ammonia in mini and micro channels and to develop a good micro scale correlation.

Nomenclature

A	heat transfer area (m^2)
B	Chisholm parameter, –
C	constant of Lockhart and Martinelli or Chisholm parameter, m
Co	confinement number, $Co = \left(\frac{\sigma}{D^2 g(\rho_L - \rho_V)}\right)^{0.5}$
C_p	specific heat (J/kg K)
E	Friedel parameter, –
F	Friedel parameter, –
F	Muller-Steinhagen parameter, bar/m
f	friction factor, –
Fr	Froude number, $\frac{G^2}{\rho^2 g D_i}$
D	diameter (m)
G	mass flux (kg/m^{2}s)
g	acceleration of gravity, m/s^2
H	Friedel parameter, –
I	current (A)
i_{fg}	latent heat of vaporization (J/kg)
k	thermal conductivity (W/mK)
L	length (m)
MAD	mean absolute deviation, $=1/N\,\Sigma\lvert X_{pred} - X_{exp}\rvert/X_{exp}$ (%)
$\dot{m}$	mass flow of refrigerant (kg/s)
P	pressure (bar)
ΔP	pressure difference (mbar)
Q	power (W)
q''	heat flux (W/m^2)
Re	Reynolds number, $\frac{GD}{\mu}$
T	temperature (ºC)
V	voltage (V)
We	Weber number, $\frac{G^2 D_i}{\rho\sigma}$
x	vapor fraction
X	Lockhart–Martinelli parameter, $X = \left[\frac{\left(\frac{dp}{dz}\right)_L}{\left(\frac{dp}{dz}\right)_V}\right]^{0.5}$
Y	Chisholm parameter, –
z	axial position (m)

Greek Letters

Φ	two-phase flow multiplier
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3
σ	surface tension N/m

Subscripts

c	cross-sectional
exp	experimental
h	heated
i	inside
ic	inlet contraction
in	inlet
L	liquid
V	vapor
o	outside
pred	predicted
sat	saturation
ic	inlet contaction
tp	two phase
sp	single phase
oe	outlet expansion
fric	frictional
grav	gravitational
acc	acceleration
LO	liquid only
Oe	outlet expansion
VO	vapor only

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