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PREDICTION OF PRESSURE DROP IN REFRIGERANT-LUBRICANT OIL FLOWS WITH REFRIGERANT OUTGASSING IN SMALL DIAMETER TUBES

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Abstract. *This paper presents an analysis of the available prediction methodologies for frictional pressure drop in two-phase gas-liquid flows of refrigerant-lubricant oil mixtures in a small diameter pipe. In this particular application, the liquid-vapour phase change is caused by a reduction of the solubility of the refrigerant in the mixture. The very low vapour pressure of the oil causes it to remain in the liquid state throughout the pipe length, whilst the refrigerant progressively evaporates from the liquid mixture (outgassing). Several correlations and methods for the calculation of the frictional two-phase pressure drop were investigated. Some of these correlations are state-of-the-art methods developed based on data for small diameter channels (Mishima and Hibiki, 1996; Tran et al., 2000; Wang et al., 2000; Chen et al., 2001). As will be seen, none of the above methodologies perform satisfactorily over the wide range of conditions of the refrigerant-lubricant oil mixtures flows tested by Lacerda et al. (2000) in a 2.86 mm tube. Reasons for such discrepancies are explored in the manuscript and the best approach to predict the frictional pressure gradient for such flows is outlined.*

Key Words: *Phase change, refrigerant-oil mixtures, small diameter pipes, refrigerant outgassing.*

1. Introduction

Flashing flows take place in a number of industrial applications. In the refrigeration industry, in particular, the most common of these is the capillary tube expansion device. The research on flows in capillary tubes is extensive and has led to numerous advances both at a practical and at a fundamental level (Seixlack, 1996).

In recent years, problems of a new type involving liquid flashing have gained renewed attention in the refrigeration industry (Lacerda et al., 2000). These are flows of refrigerant-lubricant oil mixtures with high contents of oil, whose understanding is crucial to the development of a knowledge basis onto which lubrication models can be built.

An initial step in the construction of such a basis is the study of refrigerant-lubricant oil flows in a simple geometry, i.e., a straight horizontal tube. Due to the very low vapour pressure of the oil, under some pressure-temperature conditions, the solubility of the refrigerant in the liquid mixture is such that it is expelled from the mixture in the form of vapour. As will be seen, the resulting two-phase, two-component flow possesses very peculiar characteristics and requires special attention as far as its modelling is concerned.

The present work attempts to analyse some of these features, namely the behaviour and prediction of the component of the total pressure gradient due to friction. The main objectives of this paper are:

1. To review the current methodologies available for the prediction of two-phase pressure drop in small diameter tubes. Emphasis will be given to methodologies which could be employed or extendend to the flow of refrigerant-lubricant oil mixtures undergoing phase change;
2. To apply some of these available methods to the NRVA-UFSC database on flashing flows of refrigerant-oil mixtures;
3. To evaluate the performance of the methods and correlations in dealing with these flows and to recommend the best approach for practical purposes.

Recently, Yana Motta et al. (2001) thoroughly reviewed the literature on flashing flows of oil-refrigerant mixtures. A similar review will not be repeated here, although it is worthy of note that the majority of works available treat the oil as the contaminant (weight fractions lower than 5%). There is a dearth of studies in which the oil is contaminated by the refrigerant.

This paper is organised as follows. Section 2 reviews in brief the experimental work carried out by Lacerda et al. (2000) on two-phase flow of R12-SUNISO 1GS mixtures in a 2.86 mm ID tube. In Section 3, an analysis of the experimental results is undertaken in the light of existing methods of prediction of frictional pressure drop. The applicability of such methods to refrigerant-oil flows is also discussed. A novel approach for the evaluation of the frictional pressure drop in oil-refrigerant systems is presented in Section 4. Finally, conclusions are drawn in Section 5.

2. A review of the experimental work

2.1. Experimental setup

This section gives a brief overview of the experimental campaign undertaken by Lacerda et al. (2000). The experimental apparatus is shown in Figure 1. During an experimental run, the pressure difference between the high and low pressure vessels (HPV, LPV) is kept constant. Conditions in the HPV are such that an equilibrium liquid mixture at the bottom of the vessel co-exists with the refrigerant vapour at the top. Pressure and temperature sensors monitor the conditions of gas and liquid in both vessels. High and low pressure reservoirs (HPR, LPR) operating at pressures higher and lower than those at HPV and LPV respectively, keep the pressure at constant levels in both vessels. Mass flow rates are measured using a level transducer inside HPV and also through a Coriolis flow meter located at the exit of HPV.

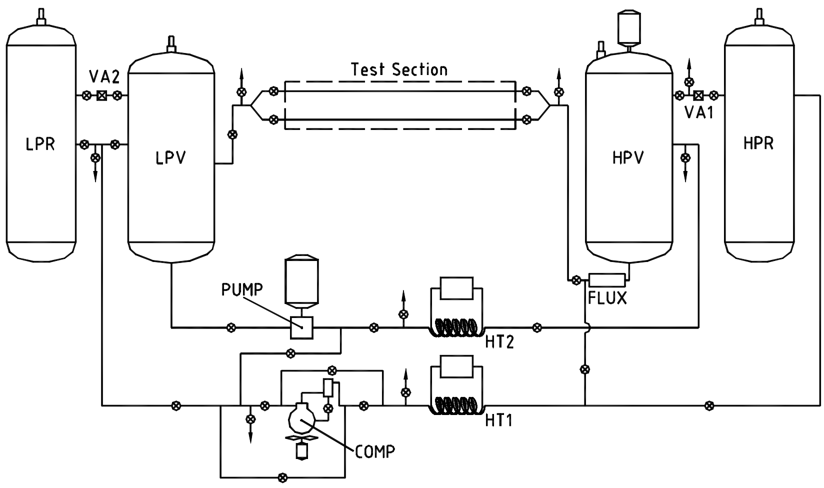


Figure 1: A schematic of the experimental apparatus.

The equilibrium liquid mixture existing in HPV is driven into either of two 5.30 m long horizontal tubes. The first tube is made of borosilicate glass (3.03 mm ID) and allows flow visualisation. The second one is a metallic Bundy type 2.86 mm ID equipped with 12 pressure tappings and 12 thermocouples placed in alternating positions along the tube.

Ancillary equipment consists of a compressor, an oil pump and two heat exchangers. Data acquisition and monitoring were controlled by a desktop computer. The reader is referred to Lacerda et al. (2000) for a detailed description of rig operation and validation and experimental procedure.

2.2. Experimental Results

Numerous experimental runs were carried out with saturation pressures ranging from 2 to 4.5 bara and temperatures around 293 K. Table 1 lists the experimental conditions evaluated in the present study.

In Table 1, G is the total mass flux (flow rate divided by cross-sectional area), *oil* and *ref* subscripts stand for oil and refrigerant, p_{in} is the inlet pressure and x_{exit} is the exit gas mass fraction (quality).

Table 1: Experimental conditions.

Run No.	G (kg m^{-2}s^{-1})	G_{oil} (kg m^{-2}s^{-1})	G_{ref} (kg m^{-2}s^{-1})	p_{in} (bar)	x_{exit}
37	394.8	303.9	90.9	2.70	0.06
22	465.7	375.6	90.1	2.50	0.13
36	486.4	374.4	112.0	2.69	0.09
34	529.7	408.3	121.4	2.70	0.15
21	556.9	423.9	133.0	2.92	0.14
18	583.7	470.9	112.8	2.49	0.12
42	664.0	441.9	222.1	3.43	0.08
40	832.8	553.0	279.8	3.43	0.16
38	896.4	592.7	303.7	3.43	0.22
66	1451.1	772.8	678.3	4.08	0.21

The total pressure gradient at a position intermediate between adjacent pressure tappings was estimated based on the ratio of the difference between the two adjacent pressure readings to the distance between the pressure tappings. Local parameters (physical properties, concentration and quality) were estimated using the average pressure value at the intermediate position and the temperature reading (already obtained experimentally at that distance).

The frictional pressure gradient at an intermediate position was obtained by subtraction of the estimated local accelerational pressure gradient from the total pressure gradient. The local accelerational component was estimated through a homogeneous flow assumption using the calculated intermediate quality.

3. Two-phase refrigerant-oil flows

3.1. Physical Properties Evaluation

From a modelling point of view, the addition of a lubricant oil to a once pure refrigerant system represents a formidable challenge. The difficulties associated with the prediction of physical properties, the thermodynamics of the resulting mixture, the determination of mixture critical parameters, amongst others, are all increased beyond measure due to the presence of the oil.

As far as physical properties prediction is concerned, empiricism is the rule rather than the exception for refrigerant-oil mixtures. Despite many successful attempts to describe the PVT behaviour of such mixtures in a more general way (Elvassore et al., 1999; Bertucco et al., 1999), prediction methods for properties such as enthalpy, viscosity and thermal conductivity nearly always serve a specific mixture sample, for which a batch of experimental data has been made available. This lack of generality is a result of the complicated nature of the oil, for which the chemical structure, molecular weight, critical parameters are not always known and vary greatly from sample to sample.

In the present work, the following properties were available from the oil manufacturer: solubility, liquid mixture viscosity and liquid mixture density. These are described in detail by Lacerda (2000). Mixture surface tension has been evaluated through a methodology which incorporates critical parameters estimation (Mermond et al., 1999) and the method of Sprow and Prausnitz (1967). Details are given by Barbosa (2002).

3.2. Forces Involved in Two-Phase Microchannel Flows

Several mechanisms affect the hydrodynamics of two-phase oil-refrigerant flows in small channels. This section summarises the forces acting on such flows and their approximate order of magnitude. In a recent review, Ghiaasiaan and Abdel-Khalik (2001) enumerated the five basic dimensionless groupings particularly important to the characterisation of two-phase flow in microchannels. These are as follows:

1. The Eötvös number (ratio of buoyancy to surface tension forces)

$$Eo = \frac{(\rho_L - \rho_G)\, g d_T^2}{\sigma} \tag{1}$$

2. The phasic Weber numbers (ratio of inertia to surface tension forces)

$$We_L = \frac{U_{LS}^2 d_T \rho_L}{\sigma} \tag{2}$$

$$We_G = \frac{U_{GS}^2 d_T \rho_G}{\sigma} \tag{3}$$

3. The phasic Reynolds numbers (ratio of inertia to viscous forces)

$$Re_L = \frac{\rho_L U_{LS} d_T}{\eta_L} \tag{4}$$

$$Re_G = \frac{\rho_G U_{GS} d_T}{\eta_G} \tag{5}$$

where g is the acceleration due to gravity, d_T is the tube diameter, σ is the surface tension, ρ is the density, η is the viscosity. Subscripts L and G refer to liquid and gas, respectively. U_{LS} and U_{GS} are the liquid and gas superficial velocities.

In the vast majority of microchannel flows ($d_T \approx 1$ mm), $Eo \ll 1$. Therefore, buoyancy effects can be neglected and the two-phase hydrodynamics is not affected by the channel orientation. In the two-phase flows studied here, calculated values of Eo were such that buoyancy and surface tension effects are about the same order of magnitude. However, in our case, buoyancy and surface tension effects are both overshadowed by inertia forces which were found to be the most significant ones, with Re_L, Re_G, We_L and $We_G \gg 1$ in almost all of the situations investigated. On these simple arguments, it may be that methods devised specifically for predicting flow properties of microchannel flows are not fully applicable to those investigated here, since they would tend to overemphasize the influence of second order effects.

3.3. Prediction of Two-Phase Frictional Pressure Drop

The two-phase frictional pressure drop is calculated using the two-phase multiplier model (Lockhart and Martinelli, 1949). In this model, the two-phase frictional pressure drop is given by one of four possible combinations as shown by Eq. 6.

$$\left.\frac{dp}{dz}\right|_{f,TP} = \Phi_L^2 \left.\frac{dp}{dz}\right|_{f,L} = \Phi_{LO}^2 \left.\frac{dp}{dz}\right|_{f,LO} = \Phi_G^2 \left.\frac{dp}{dz}\right|_{f,G} = \Phi_{GO}^2 \left.\frac{dp}{dz}\right|_{f,GO} \tag{6}$$

where the subscript f denotes friction.

The four different possibilities of representing the two-phase frictional pressure drop are as follows: (i) in terms of the single phase liquid pressure gradient calculated at a mass flux $G\,(1-x)$ (subscript L), (ii) in terms of the single phase liquid pressure gradient calculated at a mass flux G (subscript LO), (iii) in terms of the single phase gas pressure gradient calculated at a mass flux Gx (subscript G), (iv) in terms of the single phase gas pressure gradient calculated at a mass flux G (subscript GO). The single phase pressure drop is correlated using standard relationships for the single phase friction coefficient. This depends on several factors, including the relative wall roughness, and is generally a matter of availability or preference of the correlator.

To gain some insight into the nature of the pressure gradient in such flows, the simplest of the correlations will be used as a starting point. The following expressions for the two-phase multipliers, Φ^2, were proposed by Chisholm (1967) as a fit to the original graphical correlation of Lockhart and Martinelli (1949).

$$\Phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{7}$$

$$\Phi_G^2 = 1 + CX + X^2 \tag{8}$$

where X^2 is the ratio of the liquid to gas single phase frictional pressure gradient

$$X^2 = \left.\frac{dp}{dz}\right|_{f,L} \bigg/ \left.\frac{dp}{dz}\right|_{f,G}. \tag{9}$$

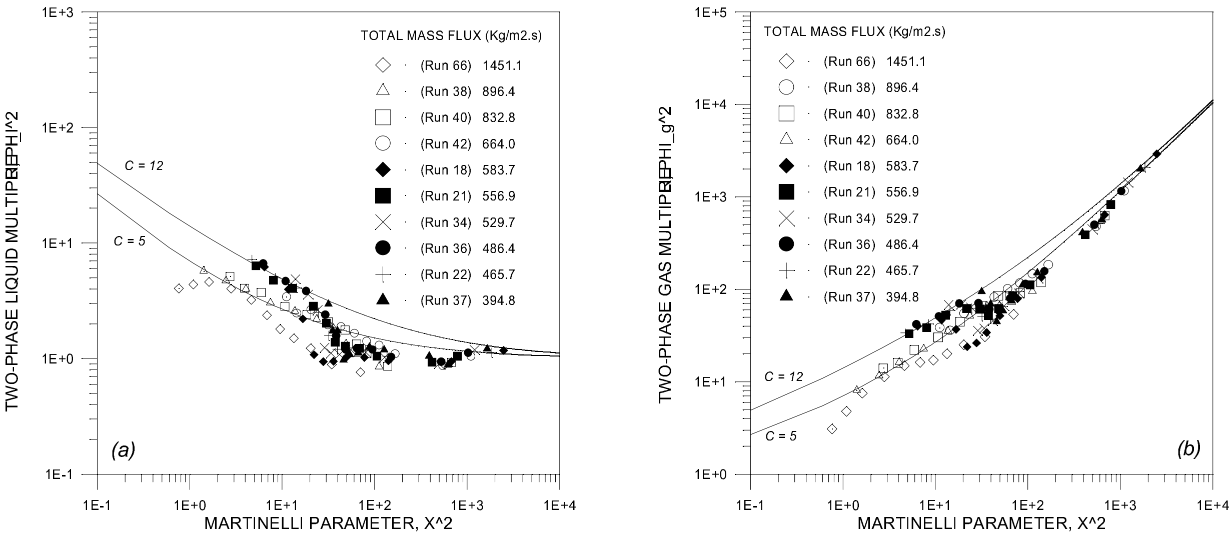


Figure 2: Two-phase frictional pressure gradients in terms of the Marinelli-Chisholm parametrisation (a) liquid, (b) gas.

According to Chisholm (1967), C assumes different values depending on the combination of flow regimes (laminar or turbulent) attained by the different phases if they alone were flowing in the tube. For laminar flow in both phases, $C = 5$; for laminar gas and turbulent liquid, $C = 10$; for laminar liquid and turbulent gas, $C = 12$; for turbulent flow in both phases, $C = 20$.

In the present study, in a hypothetical situation in which each phase was allowed to flow alone in pipe, the liquid phase would always be in laminar flow and the gas flow would be laminar for the lowest qualities, but as it is gradually expelled from the mixture due to the reduction of solubility, it would rapidly reach a state in which it would be in turbulent flow.

Figures 2.a and 2.b illustrate the frictional pressure gradient data in terms of the parameters of the Martinelli-Chisholm formulation of Eqs. 7 and 8. As will be seen, for high values of X^2 (and consequently low qualities) the data collapse into a well defined line. However, as the quality increases ($X^2 \leq 100$), the once ordered distribution starts to break down and there is considerable scatter. Also shown in Figures 2.a and b are the predictions given by the Φ_L^2 and Φ_G^2 correlations of Chisholm (1967) using C values for laminar-laminar and laminar(liquid)-turbulent(gas) flows. The curves do follow the trend of the data in some way, but fail to predict them accurately, specially at high qualities. In fact, as an illustration of the scatter, it was observed that the whole database is encompassed within the envelope formed by curves $C = 0$ and $C = 12$.

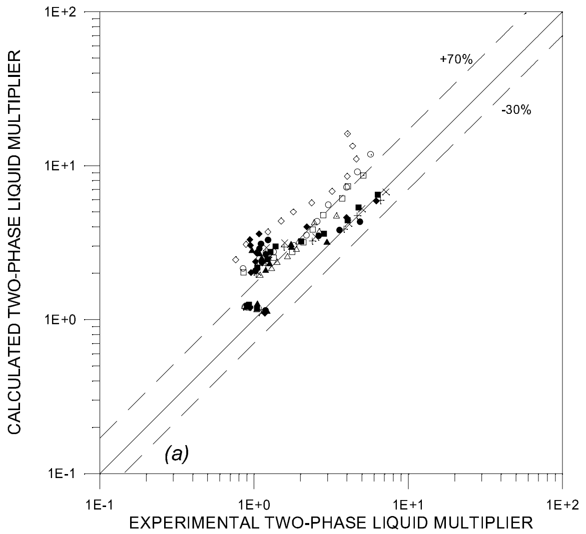
The single phase pressure gradients were estimated through standard friction factor relationships

$$\frac{dp}{dz}_f = 2f\frac{G^2}{\rho d_T} \tag{10}$$

where f is the Fanning friction coefficient. In the present investigation, the Blasius relationship was used for the turbulent region.

The performance of the Chisholm correlation can be better analysed through Figures 3.a and 3.b, that compare experimental and calculated two-phase multipliers. The correlation overpredicts the experimental data with a rms error of 91.4% and 83.9% for the liquid and gas multipliers, respectively.

In some way, one is expected to believe that the Chisholm (1967) correlations will not predict satisfactorily the oil-refrigerant data. The correlations were originally proposed based mainly on air-water adiabatic data in large diameter tubes, a situation far from that presented here. Even for those flows, as pointed out by Hewitt (1992), the Martinelli-type correlations are notably deficient in accounting for the demonstrable influence of mass flux on the pressure drop multipliers, and standard deviations may range up to as much as 100% with order of magnitude difference in some points.



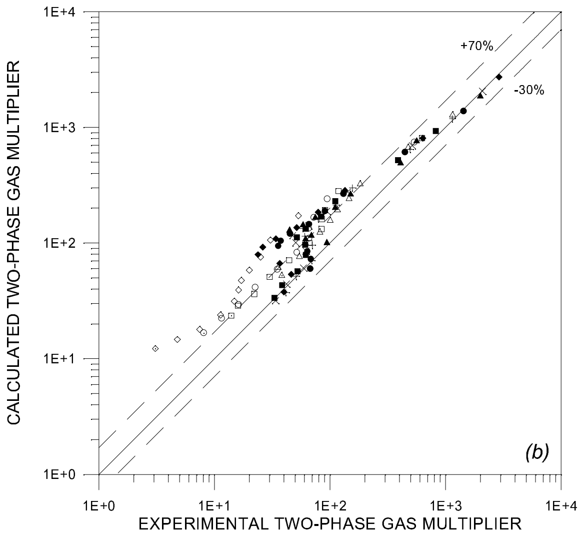


Figure 3: Comparison of experimental and calculated two-phase frictional pressure gradients using Chisholm (1967) correlation (a) liquid, (b) gas (Symbols identical to those in Fig. 2).

In the next section, other methods and correlations for the frictional pressure gradient are studied. Their features and limitations in dealing with refrigerant-oil systems will be discussed qualitatively and quantitatively.

3.4. Other Methods

In recent years, a number of pressure drop correlations have been proposed for two-phase flows in small diameter channels. It is not the purpose of this paper to give a detailed account of all of these works, but rather to critically evaluate the most prominent ones in the light of oil-refrigerant flashing flows.

The effect of tube diameter on two-phase multipliers has been investigated experimentally by Mishima and Hibiki (1996). They proposed an alternative expression for the parameter C given by

$$C = 21\left[1 - \exp\left(-0.319 d_T\right)\right] \tag{11}$$

where d_T is given in mm. Mishima and Hibiki's database comprised air-water, R113-N_2 and ammonia-vapour systems for vertical and horizontal flows. The majority of the tube diameters investigated lies in the range between 1.05 and 4.08 mm. An evaluation of Eq. 11 for the present data tube diameter gives $C = 12.6$.

Air-water data in a 3 mm ID horizontal tube were analysed by Wang et al. (2000). They divided the data into two regions: (i) a region in which the multipliers are dependent on mass flux and (ii) a region in which the multipliers are insensitive to changes in mass flux. A criterion adapted from that of Klimenko and Fyodorov (1990) was adopt as the transition between both regions. All of the present oil-refrigerant database fell in Region (ii) of Wang et al.'s method. For this region, the two-phase multiplier is given by

$$\Phi_L^2 = 1 + \frac{C_1}{X} + \frac{C_2}{X^2} \tag{12}$$

where

$$C_1^2 = \left[5 + \left(\frac{Re_G}{Re_L}\right)^{0.3}\right] + \frac{Re_{LO}}{17} \tag{13}$$

and

$$C_2 = \frac{0.05 Re_L}{Re_G}. \tag{14}$$

A more convenient way of correlating phase change data is through the all-liquid or all-gas formulations presented in Eq. 6. Numerous methods have been proposed over the years and perhaps the most used of them is that due to Friedel (1979)

$$\Phi_{LO}^2 = \mathsf{A} + 3.21x^{0.78}(1-x)^{0.224}\left(\frac{\rho_L}{\rho_G}\right)^{0.91}\left(\frac{\eta_G}{\eta_L}\right)^{0.19}\left(1-\frac{\eta_G}{\eta_L}\right)^{0.7} Fr_H^{-0.0454} We_H^{-0.035} \tag{15}$$

where,

$$\mathsf{A} = (1-x)^2 + x^2\frac{\rho_L f_{GO}}{\rho_G f_{LO}} \tag{16}$$

$$Fr_H = \frac{G^2}{g d_T \rho_H^2} \tag{17}$$

$$We_H = \frac{G^2 d_T}{\sigma \rho_H^2}. \tag{18}$$

According to Friedel, the single phase Fanning friction factors should be calculated from ($j = G, L$)

$$f_{jO} = \frac{0.25}{\left[0.86859\ln\left(\frac{Re_j}{1.964\ln Re_j - 3.8215}\right)\right]}. \tag{19}$$

Modifications of Φ_{LO}^2 correlations to deal with flows of refrigerants in small diameter tubes have been proposed by many investigators. Tran et al. (2000) modified the Φ_{LO}^2 correlation of Chisholm (1973) by incorporating the term $\sqrt{Eo}$ into it

$$\Phi_{LO}^2 = 1 + \left(4.3Y^2 - 1\right)\left[\frac{1}{\sqrt{Eo}}x^{0.875}(1-x)^{0.875} + x^{1.75}\right] \tag{20}$$

where

$$Y^2 = \frac{dp}{dz}_{f,GO} \Big/ \frac{dp}{dz}_{f,LO}. \tag{21}$$

Chen et al. (2001) put forward a correction factor for the Friedel (1979) correlation, which is a function of Eo and of We.

Homogeneous models have also been widely used in small channel flows. In terms of the Fanning friction factor, f, the single phase all-liquid multiplier becomes

$$\Phi_{LO}^2 = \frac{f_H \rho_L}{f_{LO}\rho_H} \tag{22}$$

where ρ_H is the homogeneous density given by

$$\frac{1}{\rho_H} = \frac{x}{\rho_G} + \frac{1-x}{\rho_L} \tag{23}$$

and f_{LO} is the single phase all-liquid friction factor and f_H is the homogeneous friction factor calculated from standard single phase flow relationships using the homogeneous Reynolds number, $Re_H = \frac{G d_T}{\eta_H}$. The successful modelling of homogeneous flows resides in the appropriate choice of the homogeneous viscosity, η_H. Recent methods for refrigerant systems have adopted the model of Beattie and Whalley (1982)

$$\eta_H = \eta_L(1-\varepsilon_H)(1+2.5\varepsilon_H) + \eta_G\varepsilon_H \tag{24}$$

where ε_H is the homogeneous void fraction. An example of such a method is that of Chen et al. (2001), who proposed a correction factor based on the dimensionless parameters of Section 3.2.

Methods specifically devised to deal with refrigerant-oil mixtures are rare. Alofs and Hasan (1990) proposed the following method based on that of Macken et al. (1979)

$$\Phi_G^2 = \left(1 + X^{2/n}\right)^n \tag{25}$$

where

$$n = 1.8 + 3.652 \tanh\left(1.186 \times 10^{-5} G_G\right) \tag{26}$$

and G_G is the gas mass flux given in lbm h^{-1} ft^{-2}.

In this method, the liquid phase is assumed laminar and the gas phase turbulent. The turbulent Fanning friction coefficient is given by

$$\frac{1}{2\sqrt{f}} = -0.8 + 2\log_{10}\left(2Re_G f\right). \tag{27}$$

Alofs and Hasan (1990) tested their method with experimental data obtained in a 6.28 cm ID pipe with R12/150 SUS and R502/150 SUS mixtures.

Figures 4.a to 4.f illustrate the performance of some of the above methods in correlating two-phase refrigerant-oil data. Due to the use of different two-phase multipliers and correlation parameters, the results are presented in terms of the ratio of calculated to experimental frictional pressure gradient and of quality. For the sake of present comparisons, gas phase parameters of Φ^2_{LO} methods — such as those of Tran et al. (2000) and Friedel (1979) — were calculated assuming pure refrigerant properties. Implications resulting from this assumption will be addressed later.

All methods, exceptions being those of Tran et al. (2000) and Alofs and Hasan (1990) at lower mass fluxes, severely overpredict the frictional pressure gradient. In the Mishima and Hibiki (1996) and Wang et al. (2000) methods, the bigger discrepancies take place at intermediate qualities for the lower mass fluxes, whereas for Friedel (1979) and Chen et al. (2001), the discrepancies increase continuously independent of mass flux. At low and moderate mass fluxes, the Tran et al. (2000) method underpredicts the experimental data considerably. The Alofs and Hasan (1990) correlation performs well at moderate qualities and low and moderate mass fluxes. However, it fails at high mass fluxes.

4. A proposed correlation

A detailed analysis of the database and of the performance and limitations of the methods described in Section 3.4 has led to the belief that the best approach for correlating the present data is an adaptation of the gas pressure drop multiplier of Chisholm (1967).

Although correlations based on Φ^2_{LO} proved successful in correlating two-phase flows of pure refrigerants (Tran et al., 2000), those which involve all-gas (subscript GO) parameters present a difficulty for refrigerant-oil mixtures since these cannot be evaluated. Gaseous oil properties are not available due to lack of practical application. Thus, on implementing the Friedel (1979) and Tran et al. (2000) methods using pure refrigerant properties as all-gas properties, an unquantifiable systematic error was brought to the analysis.

When developing the correlation, it was observed that as local quality values are quite small, the term X^2 contributed most significantly to the value of the pressure drop. In addition, it was found that the best form of the correlation for Φ^2_G is the one in which all three terms present in Chisholm's original expression are present. It was decided then to model the first term on the RHS of Eq. 8 and the constant C as a function of parameters that express the relative oil content, the relative vapour content and the dominance of inertia effects.

The following expression provided the best correlation of the experimental data

$$\Phi^2_G = 1.24\frac{x}{(1-x)}(1-w) + \left[1.95\exp\left(6.94 \times 10^{-3}\frac{Re_G}{Re_L}\right)\right]X + X^2 \tag{28}$$

with a r.m.s. error of 26%. w is the local weight fraction of refrigerant in the liquid mixture.

Figure 5 shows a comparison between the experimental and calculated Φ^2_G using Eq. 28.

The remaining scatter in the distribution might be due to several factors which cannot be quantified through means available at present. In what follows, we list those considered more important: (i) *Metastability and vaporisation delay.* On calculating the local quality, it was assumed that this was equal to the equilibrium quality obtained from the solubility curve. However, as in pure refrigerant systems, over a considerable portion of the flow length the local quality may be lower than the equilibrium quality due to the presence of metastable single phase liquid and metastable liquid in the presence of gas. Metastability prediction of pure refrigerants in microchannels is discussed by Meyer and Dunn (1998) and by Melo et al. (1999). (ii) *Mass transfer and selective adsorption.* In mixtures, the resistance to diffusion caused by concentration gradients in the vicinity of growing bubbles reduces the rates of nucleation and may also contribute to the reduction of the local quality in comparison with that at equilibrium. Concentration gradients may also be formed as a consequence of the

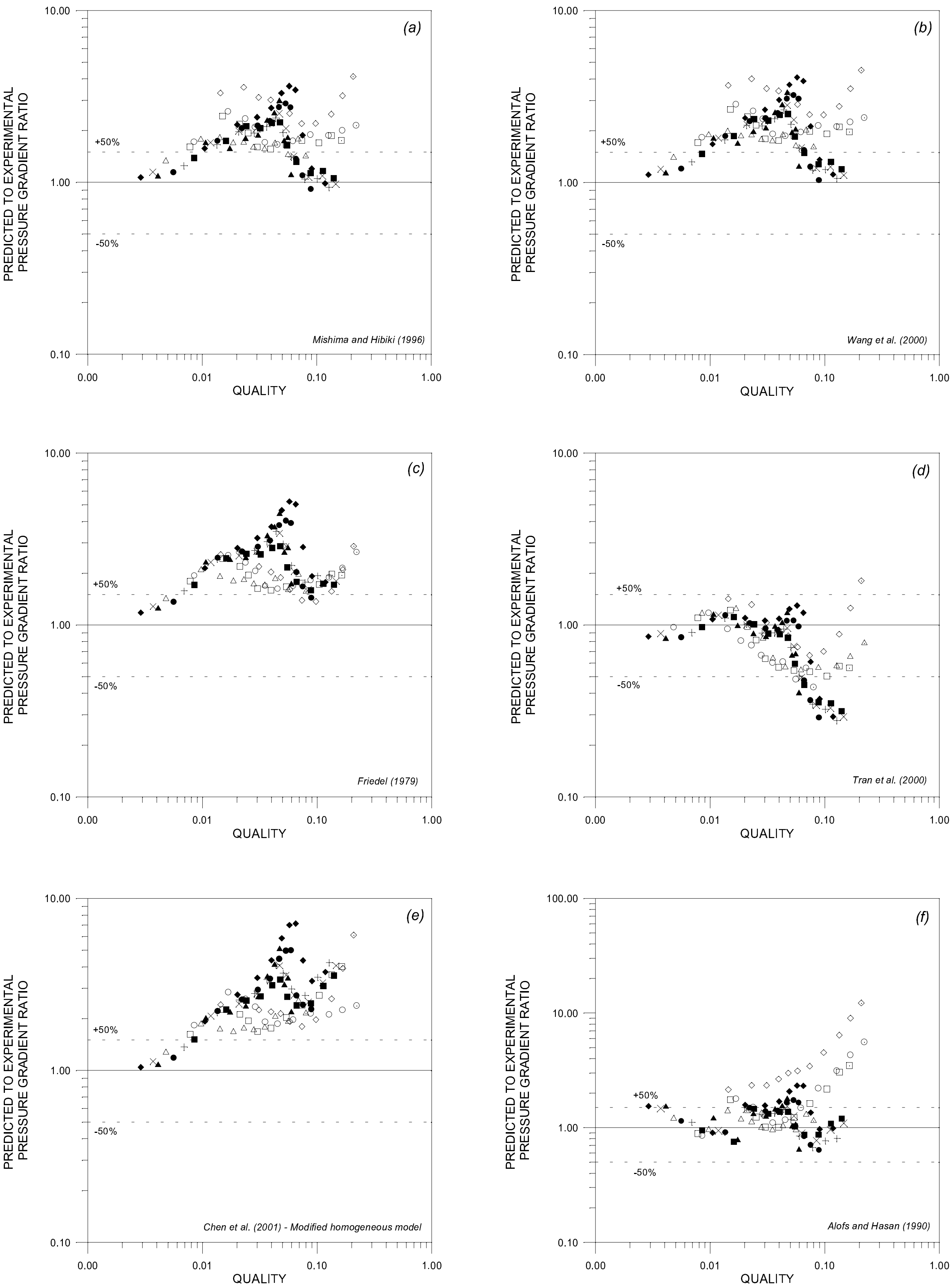


Figure 4: Comparison of experimental and calculated two-phase frictional pressure gradients for different correlations (Symbols identical to those in Fig. 2).

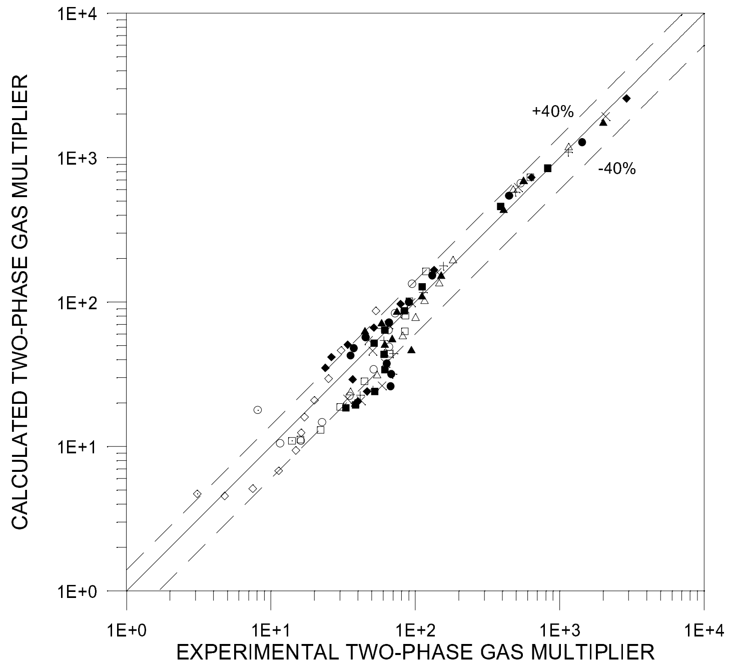


Figure 5: Comparison of experimental and calculated two-phase frictional pressure gradients using Eq. 28 (Symbols identical to those in Fig. 2).

adsorptive ability of the solid surface (Stephan and Mitrović, 1982). If the concentration of the more volatile component is lower at the solid-liquid interface than in the bulk, then the number of active bubble nuclei may be greatly reduced, thus delaying the inception of vaporisation. (iii) *Foam formation*. It has been observed that towards the end of the tube, the flow configuration changes so that a foam-type structure appears. It is well known that foams possess complex rheology. The parameters that govern this change in structure are not well defined and our inability to capture their behaviour and influence on local pressure gradient may also contribute to the scatter observed in Fig. 5.

5. Conclusions

Frictional pressure drop in oil-rich R12-SUNISO 1GS refrigerant-oil two-phase flows with volatile outgassing has been investigated theoretically. The following conclusions can be drawn from this analysis:

1. For the conditions investigated ($d_T = 2.86$ mm), inertia effects were found to be dominant. Buoyancy and surface tension forces are of the same order of magnitude, but of a lower significance when compared to inertia. This result is in contrast with those found in microchannel flows;

2. Two-phase multiplier correlations based on a variety of experimental conditions including oil-refrigerant flows and flows in small diameter tubes failed to predict the present database;

3. An adaptation of the Chisholm (1967) correlation for the two-phase gas multiplier, Φ_G^2, taking into account the relative oil content, the relative vapour content and the dominance of inertia effects in such flows predicts the data with a r.m.s. error of 26%.

6. Acknowledgements

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