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Summary

The adiabatic two-phase frictional multipliers for SUVA, R-134a flowing in a rectangular duct (with $D_H = 4.8$ mm) have been measured for 3 nominal system pressures (0.9 MPa, 1.38 MPa and 2.41 MPa) and 3 nominal mass fluxes (510, 1020 and 2040 kg/m²/s). The data is compared with several classical correlations to assess their predictive capabilities. The Lockhart-Martinelli model gives reasonable results at the lowest pressure and mass flux, near the operating range of most refrigeration systems, but gives increasingly poor comparisons as the pressure and mass flux is increased. The Chisholm B -coefficient model is found to best predict the data over the entire range of test conditions; however, there is significant disagreement at the highest pressure tested (with the model over predicting the data upwards of 100% for some cases). The data shows an increased tendency toward homogenous flow as the pressure and flow rate are increased, and in fact the homogeneous model best predicts the bulk of the data at the highest pressure tested.

1.0 Introduction

The pressure drop in fluid systems is one of the fundamental parameters of interest to design engineers. The pressure drop in two-phase (i.e., gas-liquid) flow can be dramatically higher than pure liquid flow at the same overall mass flux. Two-phase multipliers have been used to account for this and provide a simple means of estimating the relative increase in pressure drop due to the presence of the gas phase.

A number of correlations and analyses have been developed to predict the two-phase multipliers for a variety of two-phase flows. The simplest analysis is perhaps the homogeneous approximation, where both phases are assumed to flow with the same average velocity. This approximation may be useful at high mass flux or high pressure, where the slip ratio (gas velocity relative to liquid velocity) is expected to be low, but in general the homogeneous model will under predict the actual pressure drop in real systems. Martinelli and Nelson [1] and Lockhart and Martinelli [2] were early proponents of a separated flow model, where each phase is assumed to flow at different average velocities. They developed a correlation which performed well for different fluids at low mass flux, but did not allow for sensitivity to mass flux. Baroczy [3] captured the mass flux effect as well as the effect of fluid property by fitting a large set of data taken in different fluids and at different flow rates; however the graphical form of his correlation did not lend itself to easy application. Chisholm [4] combined the results of Lockhart and Martinelli and Baroczy with a new analysis to obtain an analytical expression for pressure drop which is more convenient to use than the Baroczy plots. Friedel later [5] examined a large data base to develop his own correlation. Most of the data used to develop the above correlations were obtained from air-water and steam-water systems.

Recently, there has been interest in understanding the pressure drop for refrigerant fluids, in particular for refrigerant systems where the coolant flows through small tubes or micro-channels [6-10]. Tran, et al. [8] have considered small tubes as those under about 3 mm in diameter. The general operational range of interest in most refrigerant systems encompasses pressures under 1 MPa and mass fluxes

less than $600 \text{ kg/m}^2/\text{s}$. SUVA R-134a, a substitute for Freon used commonly in air conditioning systems, has been investigated by several researchers in this area [8-10]. Knowledge of refrigerant fluid behavior at higher pressures and mass flux is also important for scaling studies between the refrigerant modeling fluid and pertinent steam-water applications [11].

Many of the previous studies comparing frictional pressure drop with correlations involved using analytic expressions for void fraction to obtain acceleration pressure drop which was then subtracted from the measured total pressure drop. This introduces some uncertainty into the measured frictional component. In this work, the adiabatic pressure drop for SUVA is examined to directly determine the frictional multiplier; this data is then compared with several available correlations across a wide range of mass flux and pressure to assess the quality of the predictions.

2.0 Experimental System

A schematic layout of the SUVA R-134a loop is shown in Fig. 1. A circulating canned rotor pump provides the flow, which is passed through a preheater, then split into 3 independent inlet flow zones before entering the test section. Each inlet zone contains a Venturi flow meter, throttling valve and inlet heater. The inlet flow was distributed among the 3 inlet zones depending on the total flow rate. At the highest mass flux (i.e., $G=2040 \text{ kg/m}^2/\text{s}$), the inlet flow was evenly divided among the 3 inlet zones, while for the lower total flow rates, a single center inlet zone was normally used. The use of 3 inlet zones improves the accuracy of the inlet condition, especially at the lower flow rates.

Figure 1. SUVA test loop. Test section details are provided in Fig. 2.

Before entering the test section, the flow enters a plenum consisting of a series of screens, a nozzle and a flow straightener, as shown in Fig. 2. This effectively removes gross maldistribution of flow due to the inlet zones (i.e., the manner in which the flow enters the test section has no discernable effect on the measurements taken at downstream locations). The test section itself is a vertical, 1.22 m long rectangular duct with a width of 57.2 mm and thickness of 2.5 mm. The planar arrangement of the duct allows for the mounting of 8 fused silica windows, each 3.8 cm thick by 7.6 mm wide by 27.9 cm long. The windows are mounted in pairs, four on each side of the test section, to form the flow duct and facilitate good flow visualization when desired. Each window has 3 transparent, conductive indium-tin-oxide films vacuum-deposited on the inside surface to allow for direct heating of the fluid. For the bulk of the data shown here, the voids in the test section were created by using the inlet heaters, with no power applied to the window heaters (referred to as adiabatic testing). For some limited data, the window heaters in the first 3 test section elevations were used to create voids (uniform power among all 18 heater strips), while the last elevation remained unheated (referred to as heated testing). Between the window elevations, as well as at the inlet and exit of the test section, 2.54 cm diameter ports are located, which permit access to the flow for various instruments. For these tests, thermocouple rakes were inserted into the inlet and exit ports, while the remaining ports were left open.

The flow exiting the test section enters an exit plenum, and then is passed through a large CO₂ heat exchanger to remove the voids before entering the suction side of the pump. A pressurizer is used to maintain system pressure. A bypass line passing through a 10 micron particulate filter is left open at low flow rate to constantly remove solid contaminants from the loop. The SUVA itself was processed using a distillation method to assure that the contamination level was minimized. The distillation method was found to be particularly good for removing unwanted oil from the SUVA.

3.1 Loop and Test Section Instrumentation

A summary of the loop and test section instrumentation, along with their associated uncertainties, is given in Table 1. The absolute and differential pressures were measured with standard Rosemount transducers. The system pressure was measured near the test section exit (either X=117.2 cm or 123.3 cm) as illustrated in Fig. 2. Single Type K thermocouples were installed at the inlets to the trim heaters and a rake of up to 9 Type K thermocouples was installed at the test section inlet to measure average inlet temperatures. The system heat losses were determined via single phase liquid data runs flowing through an unheated loop (and test section) by measuring the temperature difference between inlet and exit at various pressures and flow rates. This data was used to develop a correlation of the form:

$$Q_{loss} = \frac{A(\bar{T} - T_{ambient})^B}{Re^C} \quad (1)$$

where $\bar{T}$ is the average temperature between inlet and exit (i.e., $(T_{in} + T_{exi})/2$), Re is the Reynolds number, and A, B and C are coefficients obtained by curve fitting the heat loss data. Similar heat loss equations were established for the trim heaters, the inlet plenum and the test section. The uncertainty in heat loss was estimated to be +/-54 W, +/-22 W and +/- 6W for the test section, inlet heater and inlet plenum, respectively.

Figure 2. Test section schematic and experimental measurement locations. ΔP refers to differential pressure drop and GDS refers to gamma densitometer measurements of average void fraction.

Measurement	Instrument(s)	Uncertainty
Inlet flow rates	Venturi + Rosemount Model 3051C	+/-13.1 kg/m ² /s
Loop pressure	Rosemount Model 1151	+/-19.3 kPa (+/-2.8 psi)
Differential pressure	Rosemount Model 3051C	+/-0.33 kPa/m for <19.5 kPa/m +/-0.42 kPa/m for >19.5 kPa/m
Trim heater inlet temperature	Special grade Type K thermocouple	+/-1.1 °C
Test section inlet temperature	Special grade Type K thermocouples	+/-0.6 °C
Trim heater power	Watt meters	+/-5%
Test section power	Watt meters	+/-130W
Heat loss	Correlated data	+/-82 W maximum
Void fraction	Gamma densitometer	+/-0.02 absolute

Table 1. Summary of Measurement Instrumentation

3.1.1 Gamma Densitometer System (GDS)

The average cross-sectional void fraction in the test section was measured using a gamma densitometer system (GDS). This system includes a 9 curie Cesium-137 gamma source on one side of the test section and a 5.1 cm square sodium-iodide gamma detector on the opposite side. The collimator on the source cask provided a 19 mm high by 4.3 mm wide gamma beam along the width direction of the test section to interrogate the entire cross section of the flow at various stream wise positions. The GDS allows a direct measurement of the density of a two-phase mixture in the path of the gamma beam through the following relationship:

$$\rho_{2\phi} = \frac{\ln\left(\frac{I_o}{I}\right)}{At} \quad (2)$$

where I_o and At are calibration constants obtained from gamma count measurements at each desired measurement position with an empty test section and a sub-cooled liquid filled test section. The count rate I is that measured for the two-phase test condition. The two-phase density is related to the void fraction and vapor and liquid densities through the following relationship:

$$\rho_{2\phi} = (1 - \alpha)\rho_l + \alpha\rho_g \quad (3)$$

where α is the void fraction, ρ_l is the liquid phase density, and ρ_g is the gas phase density. Solving for α yields:

$$\alpha = \frac{\rho_l - \rho_{2\phi}}{\rho_l - \rho_g} \quad (4)$$

The liquid and vapor phase densities were determined based on the saturation properties at the measured test section exit pressure.

Most of the void fraction data was taken at $X=112.6$ cm, corresponding to GDS_3 in Fig. 2. For some of the runs, data was taken at $X=103.3$ cm (GDS_2) instead. Also, some selected data was taken at $X=82.1$ cm (GDS_1) along with the GDS_3 data, which allows for subsequent comparisons in the adiabatic flow cases.

3.1.2 Pressure Drop

Rosemont pressure transducers were used to measure the pressure drop between selected axial positions in the test section. In all cases, the pressure drop in the uppermost window elevation (i.e., from $X=98.6$ to $X=117.3$ cm or ΔP_3) was measured; for some cases, the pressure drop in the third window elevation (from $X=68.2$ to $X=86.8$ cm or ΔP_1) as well as across the third bridge insert (from $X=86.8$ to $X=98.6$ cm or ΔP_2) was also measured. Stainless steel tubing, 6.35 mm inner diameter, connected pressure taps along the test section edge to the Rosemont transducers.

Two Rosemont transducers calibrated at different pressure ranges were available to optimize the measurements. The liquid head in the pressure lines ($\rho'_l gh$) was added to the ΔP measurements to obtain the 2-phase pressure drop within the test section as follows:

$$\Delta P_{\text{Rosemont}} = P_A - (P_B + \rho'_l gh) \quad (5)$$

where P_A and P_B are upstream and downstream pressure positions within the test section. Eq (5) may be rewritten as:

$$\Delta P_{\text{Rosemont}} + \rho'_l gh = \Delta P_f + \Delta P_a + \Delta P_h \quad (6)$$

The pressure drop within the test section includes frictional, acceleration and gravity components, as indicated by the right hand side of Eq (6). For these tests, the pressure drop measurements were taken over unheated lengths (where the void fraction and flow quality gradients were near zero) so that the acceleration component could be neglected. Using the GDS measured average void fraction, $\bar{\alpha}$, the frictional component of the pressure drop was determined as:

$$\Delta P_f = \Delta P_{\text{Rosemont}} + \rho'_l gh - (\rho_l (1 - \bar{\alpha}) gh + \rho_v \bar{\alpha} gh) \quad (7)$$

with ρ'_l taken to be the liquid density at ambient temperature outside the test section and ρ_l, ρ_v taken to be the saturation phase densities corresponding to the test section exit pressure. The average ambient temperature for all the runs was 25.9 °C with a standard deviation of 2.9 °C.

3.1.3 Flow Quality

The exit quality was calculated as follows:

$$x_{\text{out}} = \frac{[Q_{\text{net}} - \dot{m}(h_{l,\text{out}} - h_{l,\text{in}})]}{\dot{m}(h_{v,\text{out}} - h_{l,\text{out}})} \quad (8)$$

where $h_{v,\text{out}}$ = saturation vapor enthalpy at test section exit based on exit pressure
 $h_{l,\text{out}}$ = saturation liquid enthalpy at test section exit based on exit pressure
 $h_{l,\text{in}}$ = subcooled liquid enthalpy at heater inlet based on inlet temperature
 Q_{net} = net heat input to the system, from heater inlet to test section exit

For adiabatic testing, the system included inlet heaters and unheated test section, whereas for heated testing, the system included the test section only. The uncertainty in the calculated quality will be given in Section 4.3.

4.0 Results and Discussion

Data was taken for 3 nominal flow rates and 3 nominal pressures. Table 2 summarizes the actual mean, deviations and number of points for the conditions measured.

Parameter	Nominal Value	Actual Mean Value	Actual Deviation	Data Points
pressure	0.9 MPa	0.89 MPa	12.4 kPa	41
pressure	1.38 MPa	1.36 MPa	23.4 kPa	78
pressure	2.41 MPa	2.40 MPa	39.3 kPa	110
flow	510 kg/m ² /s	510.6 kg/m ² /s	8.9 kg/m ² /s	62
flow	1020 kg/m ² /s	1020.8 kg/m ² /s	17.9 kg/m ² /s	99
flow	2040 kg/m ² /s	2029.6 kg/m ² /s	34.7 kg/m ² /s	68

Table 2. Statistics of Flow Conditions Studied

In the following sections, the two-phase multiplier models used for comparison with the data are presented, the data integrity and uncertainty are discussed and the results of the comparisons between data and predictive models are described. The two-phase multiplier is defined as:

$$\Phi_{lo}^2 = \frac{\Delta P_{2\phi}}{\Delta P_{lo}} \quad (9)$$

where the subscript 2ϕ refers to the two-phase flow condition and the subscript lo indicates the condition with only liquid flowing in the channel (at a mass flow rate = GA).

4.1 Review of ΔP Predictors

4.1.1 Homogeneous model

The homogeneous model is obtained by assuming no slip between the gas and liquid phases (i.e., $V_l = V_g$):

$$x = \frac{\dot{m}_v}{\dot{m}_{total}} = \frac{\rho_v \alpha}{(1 - \alpha)\rho_l + \alpha\rho_v} \quad (10)$$

$$\Phi_{lo}^2 = \frac{\rho_l}{\rho_{2-\phi}} \frac{f_{2-\phi}}{f_{lo}} \quad (11)$$

where

$$\rho_{2-\phi} = (1 - \alpha)\rho_l + \alpha\rho_v$$

Using the Blasius equation for friction factor:

$$f = 0.316(\text{Re})^{-0.25} \quad (12)$$

the two-phase multiplier is:

$$\Phi_{lo}^2 = \frac{\rho_l}{\rho_{2-\phi}} \left(\frac{\mu_{2-\phi}}{\mu_l} \right)^{0.25} \quad (13)$$

The equation proposed by McAdams [12] is used to evaluate $\mu_{2-\phi}$:

$$\frac{1}{\mu_{2-\phi}} = \frac{x}{\mu_v} + \frac{(1-x)}{\mu_l} \quad (14)$$

Note this allows for the proper viscosities at $x = 0$ and $x = 1$. The final 2-phase homogeneous friction multiplier is:

$$\Phi_{lo}^2 = \left[1 + x \left(\frac{(\rho_l - \rho_v)}{\rho_v} \right) \right] \left[1 + x \left(\frac{(\mu_l - \mu_v)}{\mu_v} \right) \right]^{-0.25} \quad (15)$$

4.1.2 Lockhart-Martinelli model

Although the homogeneous model may perform adequately in flow conditions where the slip ratio is low (e.g. high pressure or high mass flux conditions), it is not generally applicable for most 2-phase flows. A better approach has been to allow for a velocity difference between the two phases. This may be accomplished using a separated flow model, where the velocity of each phase is assumed to be uniform at any axial cross section (though not necessarily equal). For steady, one dimensional, adiabatic flow through a channel of constant cross sectional area, the total pressure drop is given as:

$$\frac{dP}{dz} = \Phi_l^2 \left(\frac{dP}{dz} \right)_l - [(1-\alpha)\rho_l + \alpha\rho_v]g \quad (16)$$

where the subscript l indicates the condition with liquid flowing alone in the channel (at a mass flow rate = $G(1-x)A$). Lockhart and Martinelli [2] developed a correlation for Φ_l^2 using a series of data for adiabatic 2-phase flow in horizontal tubes:

$$\Phi_l^2 = \frac{(dP/dz)_{2\phi}}{(dP/dz)_l} = \left[1 + \frac{20}{X_{tt}} + \frac{1}{X_{tt}^2} \right] \quad (17)$$

where X_{tt} is the Martinelli parameter, assuming both the gas and liquid phases are in the turbulent flow regime, defined as:

$$X_{tt} = \left[\frac{(dP/dz)_l}{(dP/dz)_v} \right]^{1/2} \quad (18)$$

For the separated 2-phase flow model, the frictional gradients may be computed as:

$$(dP/dz)_l = -\frac{1}{2} f_l \frac{G^2 (1-x)^2}{\rho_l D} \quad (19)$$

$$(dP/dz)_v = -\frac{1}{2} f_v \frac{G^2 x^2}{\rho_v D} \quad (20)$$

$$f_l = 0.316 \left(\frac{G(1-x)D}{\mu_l} \right)^{-0.25} \quad (21)$$

$$f_v = 0.316 \left(\frac{GxD}{\mu_v} \right)^{-0.25} \quad (22)$$

Substituting:

$$X_{tt} = \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_v} \right)^{0.125} \left(\frac{1-x}{x} \right)^{0.875} \quad (23)$$

This compares well with the expression for X_{tt} developed by Martinelli and Nelson using data for flow boiling inside tubes [1]. The relationship between the multipliers Φ_l^2 and Φ_{lo}^2 may be determined using Equations 19 and 21 and similar expressions where liquid only flows through the duct (i.e. $lo : x = 0$):

$$\Phi_{lo}^2 = \left[1 + \frac{20}{X_{tt}} + \frac{1}{X_{tt}^2} \right] (1-x)^{1.75} \quad (24)$$

4.1.3 Chisholm B-Coefficient model

For fully rough surfaces, where the friction factor is not a function of Reynolds number (i.e., the exponent, $n = 0$ in Equation (12)), Chisholm [4] derived a modified form of Equation (17) as:

$$\frac{\Delta P_{2\phi}}{\Delta P_{lo}} = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2} \quad (25)$$

with

$$C = \frac{1}{S} \sqrt{\frac{\rho_l}{\rho_g}} + S \sqrt{\frac{\rho_g}{\rho_l}} \quad (26)$$

and the slip ration, S defined as:

$$S = \frac{V_g}{V_l} \quad (27)$$

A property coefficient was introduced:

$$\Gamma = \sqrt{\frac{\Delta P_{go}}{\Delta P_{lo}}} = \left(\sqrt{\frac{\rho_l}{\rho_g}} \right) \left(\frac{\mu_g}{\mu_l} \right)^{n/2} \quad \text{for smooth walls} \quad (28)$$

which is similar to the coefficient used by Baroczy [3]:

$$\frac{\rho_g}{\rho_l} \left(\frac{\mu_g}{\mu_l} \right)^{0.2} \quad (29)$$

Chisholm transformed Equation (25) to the general form for smooth tubes (i.e., $n \neq 0$) approximately as:

$$\Phi_{lo}^2 = 1 + (\Gamma^2 - 1) \{ Bx^{(2-n)/2} (1-x)^{(2-n)/2} + x^{2-n} \} \quad (30)$$

$$B = \frac{C\Gamma - 2^{n-2} + 2}{\Gamma^2 - 1} \quad (31)$$

Using the Baroczy correlation as the basis, a plot of C vs. Γ was obtained, then transformed into a plot of B vs. Γ and G. The final recommendation for B was a compromise between Baroczy, Lockart-Martinelli and Chisholm such that the greatest estimate of pressure gradient was obtained. Table 3 summarizes the values of B used in the current application.

G (kg/m ² s)	B
< 500	4.8
500 < G < 1900	2400/G
> 1900	55/G ^{0.5}

Table 3. Values of B for Chisholm correlation for smooth tubes and $\Gamma \leq 9.5$

4.1.4 Friedel model

The last correlation considered is the Friedel correlation [5], given as:

$$\Phi_{lo}^2 = E + \frac{3.24FH}{Fr_h^{0.045} We_L^{0.035}} \quad (32)$$

where

$$Fr_h = \frac{G^2}{gD_H \rho_h^2} \quad (33)$$

$$E = (1-x)^2 + x^2 \frac{\rho_l f_g}{\rho_g f_l} \quad (34)$$

$$F = x^{0.78} (1-x)^{0.224} \quad (35)$$

$$H = \left(\frac{\rho_l}{\rho_g} \right)^{0.91} \left(\frac{\mu_g}{\mu_l} \right)^{0.19} \left(1 - \frac{\mu_g}{\mu_l} \right)^{0.7} \quad (36)$$

The liquid Weber number We_L is defined as:

$$We_L = \frac{G^2 D_H}{\sigma \rho_h} \quad (37)$$

with the homogeneous density ρ_h given as:

$$\frac{1}{\rho_h} = \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_l} \right) \quad (38)$$

4.2 Review of data and uncertainty analysis for two-phase multiplier and flow quality

The two-phase multipliers determined from the pressure drop data will be plotted versus quality, as calculated using Equation (8). The uncertainty in quality was determined via a propagation of errors analysis as follows:

$$\begin{aligned} \varepsilon_x^2 = & \left[\frac{1}{\dot{m} h_{fg}} \right]^2 \left[\varepsilon_Q^2 + \varepsilon_{Q_{loss}}^2 \right] + \left[\frac{(Q - Q_{loss})}{h_{fg} \dot{m}^2} \right]^2 \left[\varepsilon_{\dot{m}}^2 \right] + \left[\frac{C_p}{h_{fg}} \right]^2 \left[\varepsilon_{T_{in}}^2 + \varepsilon_{T_{sat}}^2 \right] + \\ & \left[\frac{x}{h_{fg}} \right]^2 \varepsilon_{h_{fg}}^2 + \left[\frac{T_{in} - T_{sat}}{h_{fg}} \right]^2 \left[\varepsilon_{C_p}^2 \right] \end{aligned} \quad (39)$$

The individual uncertainties for heat input, heat losses, flow rate, and inlet temperature is given in Table 2. The uncertainty in h_{fg} and C_p was estimated to be +/-0.5%, based on the uncertainty in system pressure of +/- 19.3 kPa (+/-2.8 psi). Similarly, the uncertainty in T_{sat} was taken to be +/-0.83 °C.

The uncertainty in two-phase multiplier is determined by examining the uncertainties in the measurements of the two-phase pressure drop as well as in the Blasius correlation used for the single-phase pressure drop. A series of test runs performed in single phase liquid flow were used to assess the accuracy of the Blasius correlation (Equation (12)). The pressure gradients were measured in the upper window elevation in the test section (i.e., ΔP_3 in Fig. 2). The difference between the measured gradients and the predicted gradients using Equation (12) are plotted in Fig. 3. The average difference is 0.05 kPa/m with a standard deviation of 0.38 kPa/m (the 2-phase multiplier and deviation is 1.002 and 0.17 respectively). The small average difference indicates that no clear bias is present in the single phase data compared to the Blasius correlation. The standard deviation

is close to the measurement uncertainty in the ΔP transducers and indicates no large additional random errors are present when measuring single phase liquid flow. The increase in pressure loss due to high aspect ratio geometry, noted by Jones [13], was not observed here, possibly due to the differences in development lengths.

Figure 3. Pressure gradient comparisons for single phase flow (pressure transducer uncertainty = 0.33 – 0.42 kPa/m)

Some two-phase pressure gradient data was available for the upper two window elevations (ΔP_3 and ΔP_1 in Fig. 2) and are examined in Fig. 4. The difference between the measured gradient in the upper window (ΔP_3) and the average between the upper two windows $(\Delta P_1 + \Delta P_3)/2$ is presented, normalized by the single phase frictional gradient (consistent with the definition of two-phase multiplier). The average of the difference is -3.7% with a standard deviation of 13%. This data indicates larger random uncertainty compared to the single phase data; however, there are other factors that may be potentially associated with this data, including flow development affects, spacing variations, and void fraction gradients in addition to potential vapor infiltration of the pressure lines. A very conservative estimate of the uncertainty in measured pressure gradient in the 2-phase condition is taken to be the maximum of either the Rosemount transducer uncertainty or $\pm 2\sigma$, which is $\pm 26\%$ of the single phase pressure gradient at the particular flow condition. It is noted that the frictional pressure drop component was obtained by subtracting the gravity head from the direct pressure drop measurements. A propagation of errors has shown that the uncertainty due to the gravity head magnitude is negligible compared to the uncertainty in the pressure drop measurement and will not be included here for simplicity.

Figure 4. Normalized difference between two-phase ΔP_3 gradient and average gradient of ΔP_1 and ΔP_3 .

Fig. 5 presents a comparison of the pressure gradients ΔP_1 and ΔP_3 as well as ΔP_2 and ΔP_3 . Error bars proportional to the uncertainty magnitudes described above are included with the data. Most of the comparisons between the upper two window elevations (ΔP_1 and ΔP_3) are within the measurement uncertainty. The comparisons between ΔP_2 and ΔP_3 indicate an increased pressure gradient in the bridge insert between window elevations; this effect is especially evident at the higher flow rates where any misalignment between the insert and the window would be magnified. Fig. 6 shows GDS measurement comparisons in the upper two window elevations for the cases where these measurements are available. The agreement for the large majority of the measurements is within the experimental uncertainty of the GDS (i.e., ± 0.02 absolute void as summarized in Table 2).

Figure 5. Pressure gradient comparisons. The mass flux, G , has units of $\text{kg/m}^2/\text{s}$.

The final uncertainty in two-phase multiplier is determined via error propagation analysis:

$$\frac{\varepsilon_{\Phi^2}}{\Phi^2} = \sqrt{\left[\frac{\varepsilon_{\Delta P}^2}{\Delta P^2} + \frac{\varepsilon_{D_H}^2}{D_H^2} + \frac{\varepsilon_f^2}{f^2} + \frac{\varepsilon_\rho^2}{\rho^2} + 4 \frac{\varepsilon_V^2}{V^2} \right]} \quad (40)$$

The relative uncertainty in measured ΔP is estimated as described above. The relative uncertainty for the remaining four terms is between +/-1 and +/-2%.

4.3 Data Comparisons to Reference Models

The uncertainties in quality and two-phase multiplier, given by Equation (39) and Equation (40) respectively, were used to generate error bars for all the data points and are included in the data presented in Figs. 7-9. Each figure includes data for the three nominal flow rates (510 kg/m²/s, 1020 kg/m²/s and 2040 kg/m²/s) at a given nominal pressure (0.9 MPa, 1.38 MPa or 2.41 MPa). Both adiabatic and heated data, defined in Section 3, are included, although the large majority of points are for completely adiabatic runs. Also included in the figures are the predictive correlations discussed in Section 4.1.

The data sets most applicable to refrigeration systems are the low flow, low pressure results shown in Fig. 7, particularly $G=510$ kg/m²/s. This data shows reasonable agreement with both Lockhart-Martinelli and Chisholm, which is consistent with other published results [6, 8]. Overall, the data for 0.9 MPa shows good agreement with either the Baroczy or Chisholm models, both of which do the best job capturing the sensitivity to mass flux as observed in the data. The Friedel correlation does not reflect enough sensitivity to mass flux but does appear to be a reasonable measure of the average multiplier considering all three mass fluxes. The multipliers for 0.9 MPa are all higher and, at times, significantly higher than the homogeneous prediction, perhaps indicating the presence of large interfacial velocity gradients contributing to large interfacial friction and shear. The density ratio between the liquid and gas phases for this pressure is about 28:1, which would lead to high relative velocities between phases throughout all pertinent flow regimes, including bubbly, slug and annular. The results for the heated data are consistent with the adiabatic data, indicating no clear dependence on flow development affects.

The data comparisons for $P=1.38$ MPa are shown in Fig 8. In general, the trends here are similar to $P=0.9$ MPa although the multiplier magnitudes are less. The decrease in magnitude is due primarily to the decrease in density ratio at this pressure (i.e., 16:1 vs. 28:1). Still, there is a wide range of multiplier level as a function of mass flux; at $G=2040$ kg/m²/s, the data is in line with the homogeneous prediction for $x > 0.2$, while it is two to three times the homogeneous prediction at $G=510$ kg/m²/s. The approach to the homogeneous prediction at higher flow rates is intuitively consistent with the expected decrease in relative velocity with increasing flow rate. The Lockhart-Martinelli model greatly over predicts the data, while the Baroczy and Chisholm models again do the best overall job predicting the data for all flow rates. There appears to be a trend in the data where the multiplier dips at intermediate values of quality before rising again. This trend appears more evident as the flow rate increases. Similar behavior has been observed previously (see Vassallo, *et al.* [14]) and was attributed to the flow regime change between the slug and annular flow regimes. In transitional flow, where flooding type waves may be present near the wall, a larger frictional gradient may be experienced compared to the higher quality conditions where annular flow begins to be established. Then, as the liquid film thins near the wall, an increase in quality will lead to an increase in two-phase pressure drop.

Figure 7. Two-phase frictional multipliers, $P=0.9$ MPa; data-model comparisons. The mass flux, G , has units of $\text{kg/m}^2/\text{s}$.

Figure 8. Two-phase frictional multipliers, $P=1.38$ MPa; data-model comparisons. The mass flux, G , has units of $\text{kg/m}^2/\text{s}$.

Figure 9. Two-phase frictional multipliers, $P=2.41$ MPa; data-model comparisons. The mass flux, G , has units of $\text{kg/m}^2/\text{s}$.

The dip in pressure gradient is also evident in the $P=2.41$ MPa data, shown in Fig. 9. Interestingly, the $G=1020$ $\text{kg/m}^2/\text{s}$ and 2040 $\text{kg/m}^2/\text{s}$ data have nearly collapsed on each other and the bulk of this data lies very close to the homogeneous prediction. Some of the data appears to be less than the homogeneous prediction, which is consistent with the results from Miropolskii, *et al* [15] for steam-water flow in tubes at $P=2.85$ MPa. The Chisholm correlation, which was usually within 30% of the data for the lower pressure conditions, is further removed here and is upwards of 100% high. This may be due to the fact that for lower pressures the density difference effect on momentum may dominate the losses and is better captured by the model, while at the higher pressure (where the density ratio is about 7:1), other smaller effects (like surface tension affecting both bubble size and interfacial drag) may play a larger role and not be captured as well. Previous comparisons with higher pressure steam-water data [16] in fact showed that the homogeneous model performed very well and was recommended for general use along with the Baroczy correlation. Similar conclusions can be drawn from this data. The Lockhart-Martinelli model is not a good choice for this pressure.

Local void fraction SUVA data, summarized in Vassallo, *et. al.* [14], indicates that the distribution of the liquid phase in annular type flows is highly dependant on system pressure. For example, at $G=510$ $\text{kg/m}^2/\text{s}$ and $P=0.9$ MPa, there is significant liquid film on the walls, while at $G=510$ $\text{kg/m}^2/\text{s}$ and $P=2.41$ MPa, there is much less film and a larger concentration of liquid droplets in the vapor core. Thus, the lower pressure condition acts more like a separated flow, which is consistent with the Lockhart-Martinelli analysis. This is shown in Fig. 7, including the characteristic of a maximum in the two-phase multiplier at quality $\sim 60\%$ and a downward approach to the all vapor multiplier at 100% quality. On the other hand, the higher pressure condition acts more like a homogeneous vapor-droplet flow, with multipliers generally increasing with quality. This is consistent with the data shown in Fig. 9.

Table 4 summarizes the average difference and average deviation between the predictive models and data for each pressure and mass flux. Two-phase multiplier values for the Lockhart-Martinelli, Chisholm, Friedel and homogeneous models were calculated explicitly at the calculated quality for each data point while a fifth order polynomial was fitted to the Baroczy prediction to obtain those particular values. The highlighted regions in Table 4 indicate conditions where the average difference between model and data was less than about 25%. Overall, it is concluded from these data comparisons that, for R134-a flowing in a thin duct, the Lockhart-Martinelli model is poor, is best applied for the lowest pressure, lowest flow condition and its predictive capability gets progressively worse as the pressure or flow rate is increased. The best models for the entire range of pressures and flows studied are either Baroczy or Chisholm, although the Chisholm model, being an explicit equation, is more readily useful. The homogeneous model does the best job at the highest pressure studied for intermediate to high mass fluxes.

Model	Flow Conditions								
	L, L	L, M	L, H	M, L	M, M	M, H	H, L	H, M	H, H
Chisholm	0.25 (.14)	0.0 (.13)	0.13 (.18)	0.22 (.17)	0.09 (.16)	0.07 (.18)	0.28 (.45)	0.43 (.31)	0.22 (.19)
Martinelli	0.46 (.17)	0.88 (.42)	1.86 (.48)	0.73 (.33)	1.33 (.08)	2.67 (1.17)	2.26 (.62)	3.45 (1.26)	4.0 (1.6)
Homogeneous	-0.57 (.04)	-0.35 (.14)	-0.17 (.12)	-0.44 (.13)	-0.24 (.14)	-0.03 (.15)	-0.26 (.11)	-0.05 (.17)	0.09 (.16)
Friedel	-0.28 (.07)	-0.06 (.16)	0.17 (.17)	-0.18 (.04)	0.07 (.15)	0.37 (.23)	0.24 (.22)	0.48 (.32)	0.53 (.33)
Baroczy	-0.14 (.10)	0.01 (.16)	0.13 (.18)	-0.11 (.08)	0.07 (.16)	0.17 (.19)	0.19 (.28)	0.50 (.33)	0.25 (.16)

Table 4. Average difference and average standard deviations (in parenthesis) for each model vs. data. The flow conditions are summarized as low, medium or high pressures (i.e., L=0.9 MPa, M=1.38 MPa, H=2.41 MPa) followed by low, medium and high mass fluxes (i.e., L=510 kg/m²/s, M=1020 kg/m²/s, H=2040 kg/m²/s). Thus, L,M refers to the condition with P=0.9 MPa and G=1020 kg/m²/s. The red highlights indicate flow conditions where the average difference was less than about 25%.

5.0 References

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