Void Fraction and Pressure Drop in Microchannels

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Abstract

VOID FRACTION AND PRESSURE DROP IN MICROCHANNELS

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University of Illinois at Urbana-Champaign, 2001
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An experimental investigation of pressure drop and void fraction in aluminum microchannel tubes was conducted for a variety of flow conditions. Primary fluids of interest are refrigerants, R134a and R410A. The microchannel tubes are square port geometries with hydraulic diameters of 0.91 and 1.35 mm for test section lengths of 61 and 107 cm (24" and 42"). The test matrix encompasses the entire quality range from 0-1, mass fluxes from 75 kg/m²s to 350 kg/m²s, and 3 temperatures from 10 to 35 °C. Test sections are adiabatic and in a horizontal orientation.

Single-phase and two-phase pressure drop data were taken. No "small-tube" effects due to the hydraulic diameter were found for single-phase frictional pressure drop. Two-phase pressure drop was found to be mass flux dependent. A change in pressure drop behavior as mass flow rates increased beyond a set value is caused by a two-phase flow regime transition. Available pressure drop correlations tend to consistently overpredict the experimental pressure drop.

A new microchannel testing facility was constructed at the University of Illinois at Urbana-Champaign. Details on system design, components, and operation are given.
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<tr>
<td>C</td>
<td>Frictional Flow Constant</td>
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<tr>
<td>D</td>
<td>Tube Diameter</td>
</tr>
<tr>
<td>f</td>
<td>Friction Factor</td>
</tr>
<tr>
<td>$f_D$</td>
<td>Friction Factor [Darcy]</td>
</tr>
<tr>
<td>G</td>
<td>Mass Flux (kg/m$^2$ s)</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational Constant = 9.81 m/s$^2$</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy</td>
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<tr>
<td>L</td>
<td>Test Section Length</td>
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<td>$\dot{m}$</td>
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<td>P</td>
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<tr>
<td>v</td>
<td>Velocity</td>
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<td>x</td>
<td>Quality</td>
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<tr>
<td>z</td>
<td>Distance</td>
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### Greek Symbols

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<tr>
<td>$\alpha$</td>
<td>Void Fraction</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Viscosity (kg/m s)</td>
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<td>$\rho$</td>
<td>Density (kg/m$^3$)</td>
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Dimensionless Parameters

\( \varepsilon/D \)  
Surface Roughness Factor

Fr  
Froude Number

K_{\infty}  
Pressure Drop Number

\( \phi^2 \)  
Two-Phase Multiplier

Re  
Reynolds Number

We  
Weber Number

X, X_{tt}  
Lockhart-Martinelli Parameter

Subscripts

c  
Critical

eff  
Effective

eq  
Equivalent

exp  
Experimental

h  
Hydraulic

ic  
Inscribed-Circumscribed

l  
Liquid

lo  
Liquid Only

sub  
Subcooled

th  
Theoretical

tp  
Two Phase

v  
Vapor
Chapter 1

Introduction

Due to tighter design constraints and a steady demand to continually decrease the total size required for an air-conditioning and refrigeration system, there exists a high demand for a reduction in the size of heat exchangers while still maintaining a high level of performance. Microchannels have been used for many such applications but without a fundamental understanding of their operating characteristics. Several microchannel studies have been performed to examine specific conditions, but few have attempted to develop a fundamental understanding for the fluid mechanics within microchannel tubes.

This study creates a foundation for the complete and general study of microchannel tubes. A thorough literature review, found in Chapter 2, was conducted and recorded in order to bring all aspects of past and present microchannel investigations together. The newly constructed microchannel testing facility in the Air Conditioning and Refrigeration Center (ACRC) at the University of Illinois in Urbana-Champaign is discussed in Chapter 3. This new system is extremely versatile in the variety of tubes, refrigerants, and conditions that it is able to investigate. This versatility will allow for a larger test matrix to be conducted in order to generalize microchannel characteristics. Such characteristics include pressure drop, as discussed in Chapter 4 and void fraction techniques as found in Chapter 5. Future experiments with the microchannel testing facility will utilize data and concepts from this study in order to examine void fraction, possible flow regimes, and heat transfer performance characteristics. This ultimately will allow for a thorough and fundamental understanding of microchannel tubes.
Chapter 2
Literature Review

This chapter explains many of the references that made an impact on the work presented. First, an overview of important results in single-phase pressure drop will be explained followed by information on two-phase flow regimes and pressure drop. Finally, void fraction techniques and studies are discussed.

2.1 Single-Phase Pressure Drop

A common method for characterizing the single-phase flow pressure loss in pipes utilizes the friction factor as a function of the Reynolds number, which can encompass a variety of flow regimes. Laminar flow generally occurs at low Reynolds numbers, \( \text{Re} < 1750 \), whereas turbulent flow is typically assumed for \( \text{Re} > 2300 \). The standard method to calculate the friction factor from the Reynolds number is by a Moody chart. For a smooth, circular tube, the exact solution to the Navier-Stokes equations for laminar flow yields:

\[
\frac{f_D}{\text{Re}} = \frac{64}{\text{Re}}
\]

(2.1)

where \( f_D \) is the Darcy friction factor. The Fanning friction factor, related to the Darcy by \( f_D = 4 \cdot f_F \), is equally acceptable but less commonly used in the literature. For turbulent flows in a smooth tube, the Blasius correlation given by

\[
\frac{f_D}{\text{Re}^{0.25}} = \frac{0.3164}{\text{Re}^{0.25}}
\]

(2.2)

is used to determine the friction factor (valid for all turbulent flows with \( \text{Re} \leq 10^5 \)).
In an air-conditioning and refrigeration condenser, single-phase flow accounts for a small, but important, portion of the total refrigerant flow. Superheated vapor is found at the inlet while subcooled liquid exits the heat exchanger. Thus, it is important to characterize refrigerant pressure drop for these conditions. Microchannels are found with a variety of geometrical configurations, of which a general model for single-phase pressure drop may not predict all cases accurately. Heun [1995] conducted single-phase pressure drop using both R-134a and nitrogen for round, square, triangular, and “enhanced square” microchannel geometries. He found that the Churchill [1977] correlation,

\[
\frac{f_D}{8} = \left[ \frac{8}{\text{Re}} \right]^{12} + \frac{1}{(A + B)^{3/2}} \right]^{1/12}
\]

\[A = \left\{ 2.457 \ln \left[ \left( \frac{7 \text{ Re}^{0.9}}{\text{Re}} + 0.27 \frac{\varepsilon}{D} \right) \right]^{16} \right\}
\]

\[B = \left( \frac{37530}{\text{Re}} \right)^{16} \]

which combines expressions for the laminar, transition and turbulent regimes, accurately predicts the friction factor data for single-phase vapor flow through microchannels. Non-circular geometries exhibit flatter \( f_D \) vs. \( \text{Re}_{\text{eff}} \) trends through the transitional Reynolds number range as compared to the round ports. The small square port microchannel displays a delayed transition to turbulent flow. It was found that the laminar equivalent diameter, \( D_{le} \), defined as:

\[
D_{le} = \frac{C_{f,\text{circ}}}{C_{f,\text{nd}}} D_h
\]
allows circular tube correlations to accurately predict both laminar and turbulent single-phase pressure drop in circular, square, and enhanced square port geometries. $C_f$ is the frictional flow constant, which depends on the cross-sectional geometry, based on an analogous relation of the Navier-Stokes laminar flow solution given by:

$$C_f = f \times Re$$  \hspace{1cm} (2.5)

For triangular ports, $D_{le}$ is appropriate for laminar flow whereas the inscribed-circumscribed diameter, $D_{ic}$, is valid for turbulent flow and defined for squares and equilateral triangles, respectively, with sides of length $a$ as:

$$D_{ic} = (1 + \sqrt{2})a$$
$$D_{le} = \sqrt{3}a$$ \hspace{1cm} (2.6)

Previous studies have found that in order to have complete dynamic similarity between circular ducts and non-circular ducts, both the Reynolds number and friction factor need to be appropriately scaled. Reynolds number scaling is accomplished by equating the critical value of Reynolds number for all geometries to that of circular ducts by defining an effective diameter ($D_{le}$, $D_{ic}$, etc.). Jones [1976] developed the following correlation to be applied as an appropriate Reynolds number length scale for rectangular ducts with aspect ratio, $\alpha$.

$$\frac{D_{eff}}{D_h} = \frac{2}{3} + \frac{11}{24} \alpha(2 - \alpha)$$  \hspace{1cm} (2.7)

Alternatively, a "scaling" for the friction factor must be considered due to the influence of the hydrodynamic entrance region, the increase in wall shear stresses, and a change in the momentum flux due to a developing velocity profile. Graham [1995] presents the
previously studied concept of a pressure drop number, \( K_\infty \) in such a fashion that the overall momentum balance becomes:

\[
\frac{\Delta P}{\frac{1}{2} \rho v^2} = f_D \frac{L}{D_h} + K_\infty
\]  

(2.8)

Pressure drop numbers are determined by solving the Navier-Stokes equations in the hydrodynamic entrance region. Due to the non-linearity of the equations, few closed-form solutions exist. Additionally, approximate analytical methods can be used to calculate the pressure drop number. Lundgren, Sparrow and Starr [1964] utilized boundary layer approximations and a uniform inlet velocity profile to develop an expression for ducts of arbitrary cross-section, given by:

\[
K_\infty = \frac{2}{A_c} \int_A \left[ \left( \frac{u_{fd}}{u_m} \right)^3 - \left( \frac{u_{fd}}{u_m} \right)^2 \right] dA_c
\]  

(2.9)

\( A_c \) is the cross-sectional area of the duct; \( u_{fd} \) is the fully developed axial velocity and \( u_m \) is the mean axial velocity. Other investigations by Shah and London [1978], Shah and Bhatti [1987], Miller and Han [1971], and Chen [1973] further explain duct properties and their influence.

Yang and Webb [1996] correlated R-12 subcooled liquid flow data for both plain (smooth) and micro-fin extruded aluminum microchannels in an adiabatic test section 560mm in length for \( 2500 < \text{Re}_D < 23,000 \). The pressure drop for the micro-fin tube is nearly double of that for the plain tube with Fanning friction factors experimentally determined to be \( f_m = 0.0814 \text{Re}_D^{-0.22} \) and \( f_p = 0.0676 \text{Re}_D^{-0.22} \), respectively. These results are 14% higher than the friction factor predicted by the Blasius equation for the plain and 36% higher for the micro-fin.
McAdams [1954] suggested that the friction factor equation, valid for smooth tubes with \(5000 < \text{Re} < 200,000\), should take the form, \(f_f = 0.046 \text{Re}^{-0.2}\). Even with a wide array of friction factor equations available, the most widely used formula was presented in Fox and McDonald [1992] and derived by Colebrook:

\[
\frac{1}{f_f^{0.5}} = -2.0 \log\left(\frac{e}{D} + \frac{2.51}{\text{Re}f_f^{0.5}}\right) \quad (2.10)
\]

\[
f_f = 0.25\left[\log\left(\frac{e}{D} + 5.74\right)\right]^{-2} \quad (2.10)
\]

The above equation is transcendental and thus requires iterative methods in order to evaluate values of \(f_f\). The relative roughness for a given type of pipe can be found from using the Moody chart (e.g., see Fox and McDonald [1992]). Most experiments assumed that the tube was smooth, even for the extruded aluminum microchannels.

Chang and Ro [1996] determined that due to the small diameter of a capillary tube, the relative roughness had a considerable effect on the shear stress at the wall even though the absolute roughness is practically negligible. They determined friction factors using Haaland's equation:

\[
f_f = \left\{-1.8 \log\left[\frac{6.9}{\text{Re}} + \left(\frac{e}{3.7D}\right)^{1.11}\right]\right\}^{-2} \quad (2.11)
\]

Peng et al. [1994a] investigated the flow characteristics of water flowing through rectangular microchannels having hydraulic diameters of \(0.133 - 0.367\) mm and \(H/W\) ratios of \(0.33 - 1\). The size of the transition to turbulent range diminished with decreasing hydraulic diameter and occurred at low Reynolds numbers (200-700).

It also should be noted that a considerable amount of work has been conducted for flow characteristics of microtubes. These tubes differ from microchannels in that the
overall dimensions are on the order of 50 - 350 μm. A typical application includes forced convection cooling of electronic chips in MicroElectronic Mechanical Systems (MEMS). Peng et al. [1994a, b] conducted experiments of forced convection of water in rectangular microchannels with hydraulic diameters ranging from 0.133 to 0.367 mm and aspect ratios from 0.333 to 1. The friction factor, $f$, was found to be proportional to $Re^{-1.98}$ (rather than $Re$) for laminar flow, and proportional to $Re^{-1.72}$ for turbulent flow (as compared to $Re^{1/4}$). Geometric parameters such as hydraulic diameter and aspect ratios were found to be most influential in flow characteristics such that an increase in $H/W$ increases the friction factor whereas for constant $H/W$, decreasing the hydraulic diameter causes a significant decrease in the friction factor. It was also found that the transitional Reynolds number diminishes and the transition range becomes smaller as the microchannel dimensions decrease. Mala and Li [1999] present data for water flow in microtubes with diameters ranging from 50 to 254 μm. A friction constant ratio, $C^*$, is introduced as:

$$C^* = \frac{f_{exp} \cdot Re}{f_{th} \cdot Re} = \frac{f_{exp}}{f_{th}}$$  (2.12)

It was determined that for flow in microtubes, conventional flow theory ($C=f\cdot Re$) does not apply and the frictional flow constant, $C$, is not equal to a constant. From experimental data, the frictional constant ratio, $C^*$, was found to vary with Reynolds number and to be always larger than 1. It was also found that the presence of surface roughness affects the laminar velocity profile and thus decreases the transitional Reynolds number. Other studies include Tso and Mahulikar [1999] who examined flow transitions, Harms et al. [1999] investigated developing convective heat transfer, and Arkilic et al. [1994] examined gaseous flow in microtubes.
After utilizing any of the aforementioned friction factor equations, the frictional pressure drop for the flow can then be calculated using the following relation:

\[ \Delta p = \frac{2fG^2L}{\rho D_h} \]  

(2.13)

2.2 Two-Phase Flow Regimes

A fundamental knowledge of two-phase flow regimes in small passages is required to thoroughly understand and characterize the mechanisms associated with heat transfer and pressure drop. Many studies have been conducted in order to accurately characterize and predict two-phase flow regimes in small diameter tubes. Identification of flow patterns by visual observation is subjective, and thus must be characterized in detail. Dobson [1994] presented a general classification for common flow regimes in horizontal round tubes as seen in Figure 2.1.

Stratified flow is seen at very low vapor velocities. Condensate collects in a liquid pool at the bottom of the tube where the velocity is primarily in the mean flow direction. Due to the low vapor velocity, the liquid-vapor interface is smooth. For an increasing vapor flow, this interface becomes unstable and gives rise to surface waves, which is termed wavy flow. Increasing the vapor velocity further, two different transitions can occur. For high liquid fractions, slug flow is observed. Low liquid fractions produce a transition from wavy-annular to annular flow. At even higher vapor velocities, the crests of the waves are sheared off and annular-mist flow is produced.
Figure 2.1 Two-Phase Flow Regimes

- Stratified Flow
- Wavy Flow
- Wavy-Annular Flow
- Annular Flow
- Annular-mist Flow

Flow Regimes with High Void Fraction ($\alpha > 0.5$)
Flow Regimes with Low Void Fraction ($\alpha < 0.5$)

- Slug Flow
- Plug Flow
- Bubbly Flow
The other category of flow regimes seen in Figure 2.1 is for low void fractions, or high liquid fractions. Slug flow forms from interfacial waves that grow sufficiently in amplitude to block the entire cross-section at various locations in the tube. These slugs tend to produce pressure spikes due to rapid deceleration of the vapor flow. Pseudo-slug flow is visually similar to slug flow but does not produce the large pressure spikes. Condensation forces these slugs to coalesce into bubbles in the presence of a predominantly liquid flow. This is classified as the plug flow regime. The plugs eventually break down into smaller vapor bubbles that disperse throughout the liquid creating a bubbly flow. This category of regimes, slug, plug and bubbly, occupies only a small percentage of the overall quality range.

Most two-phase studies have used air and water as the working fluids in larger diameter tubes. An early study of small-tube flow regimes by Damianides and Westwater [1988] utilized air-water flow in a compact aluminum heat exchanger with a 1.74 mm hydraulic diameter, and 1-5mm ID glass tubes. For the heat exchanger, an absence of smooth stratified and wavy stratified flow regimes was seen. High liquid velocities promoted bubble flow, of which suspended bubbles pass freely. For intermediate liquid velocities, intermittent flow was observed (which encompasses plug, slug, and pseudoslug flow regimes). At low liquid velocities through all tested vapor velocities, annular flow was seen. The experiment was conducted using superficial liquid and gas velocities in the ranges 0.0838-8.62 m/s and 1.05-101.2 m/s, respectively. Maldistribution was noted to be a significant problem with liquid tending to favor the outer channels and vapor the inner channels. Alternatively, the results for the 1mm and 2 mm glass tubes are not able to accurately predict the flow regimes present in the compact
heat exchanger. Slug flow dominates the majority of air-water flow in the 1mm tube with no sign of separated flow. For a 2mm tube, intermittent flows were mostly observed with zones of wavy stratified to annular flow for increasing gas velocities. It was found that the available predictive flow regime models worked well for the 5mm but failed for the 1mm tubes.

Wambsganss et al. [1990] tested air/water mixtures for flow pattern identification and pressure drop in a small, rectangular channel of 9.52 x 1.59mm for a wide range of qualities (0.0002 - 1) and mass fluxes (50 - 2000 kg/m²s). Two additional flow patterns, slow-slugging and churn, were observed with no appearance of bubble flow. Wambsganss states that the difference in flow patterns may be due to surface tension effects, which are more substantial in small channels. It was also found that the small round tube flow regime maps developed by Damianides do not match well with maps developed for the rectangular channels as tested.

Wang, Chiang and Lu [1997] conducted experiments adiabatically using 3 refrigerants in a round tube test section (test section length =1200mm) with an inner diameter of 6.5mm for mass fluxes between 50 and 700 kg/m²s. It was found that plug, slug, and stratified flow patterns exist at the lower mass fluxes (100 kg/m²s) with the appearance of wavy and annular flows at higher mass fluxes (200-400 kg/m²s) for R-22. Annular flow becomes the dominant flow pattern for mass fluxes above 400. R-134a was noted to display consistently similar flow regimes to R-22 through the range of operating conditions. Flow pattern progression versus quality is considerably different, however, for R-22 and R-407C at a given mass flux. R-407C experiences a delay in the flow-pattern transition due to the variation in density and viscosity of each component. The
least volatile component, R-134a, tends to increase in concentration of the liquid phase in both low and high quality regions (lowering the mean liquid velocity) throughout evaporation conditions whereas the more volatile components, R-32 and R-125, tend to dominate the vapor phase which increases the mean vapor density and thus lowers the mean vapor velocity.

Fukano and Kariyasaki [1993] investigated isothermal flow of air/water in capillary tubes with inner diameters of 1, 2.4, and 4.9mm for vertical upward, vertical downward, and horizontal flow orientations. After comparison of the generated flow maps from vertical and horizontal flow, it was found that orientation has no effect on flow patterns. Thus, it was concluded that the surface tension forces have a much larger effect on the flow pattern than gravitational forces for gas-liquid flow in capillary tubes with diameters less than 5mm.

Other notable references with respect gas-liquid two-phase flow pattern classification in small tubes include Triplett et al. [1999a,b] and Xu et al. [1999]. Triplett examined circular microchannels with diameters of 1.1 and 1.45mm and triangular microchannels with hydraulic diameters of 1.09 and 1.49mm utilizing air and water. The range of gas and liquid superficial velocities include 0.02-80 and 0.02-8 m/s, respectively. Xu tested air and water adiabatically in vertical rectangular channels (12 x 260mm) with narrow gaps of 0.3 and 0.6-1.0mm. The smallest channel, 0.3mm, displayed different flow structures than the larger channels. Bubbly flow was not observed. At lower liquid flow rates, increased friction shear is seen that attaches the liquid droplets to the walls while being pushed along by the gas phase. Both Xu and
Triplett noticed essentially the same flow behaviors and transitions of bubbly, churn, slug, slug-annular, and annular.

Since flow patterns strongly influence key design characteristics, it is important for the designer to be able to predict the flow pattern based on operating conditions. Many methods exist for flow regime prediction in smooth tubes, with considerably less data available for small tubes or microchannels. One of the first attempts at a flow regime map was performed by Baker [1954] in large diameter tubes. Using a larger database, Mandhane [1974] developed a similar map whereas Taitel and Dukler [1976] created a more theoretically based flow regime map. Soliman [1982] and Steiner [1993] have also contributed much work to the large tube regime map. Work by Barnea et al. [1983], Damianides and Westwater [1988], Fukano and Kariyasaki [1993], and Triplett et al. [1999a] created some of the current small tube two-phase flow pattern maps.

2.3 Two-Phase Pressure Drop

A major design consideration for condenser/evaporator operation in air-conditioning and refrigeration applications involves characterizing the two-phase frictional pressure loss. Due to the demand for decreasing the total refrigerant charge required in a refrigeration system, many of the tubes for these heat exchangers are approaching hydraulic diameters on the order of, and less than, 1mm. A very limited amount of work has been performed in the area of two-phase pressure drop modeling in small diameter extruded aluminum microchannels. Thus, several large diameter (\(D_i > 5\)mm), along with existing small tube, correlations will be compared to data produced from this study.
Several of these correlations are based on the separated flow model developed by
Lockhart and Martinelli [1949]. Lockhart and Martinelli defined two-phase multipliers,
which relate the actual two-phase frictional pressure gradient to the single-phase pressure
gradient. This multiplier should depend on the a ratio of liquid to vapor phase pressure
gradients, commonly referred to as the Lockhart-Martinelli parameter \( X \), given as
follows:

\[
X = \left[ \left( \frac{\Delta P}{\Delta z} \right)_l \right]^{0.5} \left[ \left( \frac{\Delta P}{\Delta z} \right)_v \right]^{0.5} \tag{2.14}
\]

Assuming that both the liquid and vapor flows are turbulent, the Lockhart-
Martinelli parameter is defined as:

\[
X_{II} = \left( \frac{1-x_{ave}}{x_{ave}} \right)^{0.9} \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \tag{2.15}
\]

Two-phase multipliers, as found in correlations, commonly utilize the above
"turbulent-turbulent" Lockhart-Martinelli parameter and take the form:

\[
\Phi^2 = \frac{(\Delta P)_p}{(\Delta P)_lo} \tag{2.16}
\]

where \( tp \) and \( lo \) denote the two-phase and liquid only components, respectively.

Souza [1993] conducted experiments using R-12 and R-134a in a 10.9 mm
diameter smooth copper tube and determined the following correlation for the two-phase
multiplier:

\[
\Phi^2 = (1.376 + c_1 X_{II}^{-c_2})(1-x)^{1.75} \tag{2.17}
\]

For \( 0 < Fr_l \leq 0.7 \)

\[
c_1 = 4.172 + 5.480Fr_l - 1.564Fr_l^2
\]
where $Fr_I$ is defined as:

\[
Fr_I = \frac{G^2}{\rho_l^2 g D}
\] (2.18)

The total pressure drop during two-phase flow was separated into the friction and acceleration components while utilizing a Froude rate effect to determine the multiplier.

An additional separated flow model developed by Jung and Radermacher [1989] yields the following two-phase multiplier:

\[
\Phi^2 = 12.82X_r^{-1.47}(1 - x)^{1.8}
\] (2.19)

This correlation was developed from experiments using horizontal annular flow boiling of pure and mixed refrigerants. Pressure drop with refrigerant mixtures was found to be independent of composition but correlate well with the Lockhart-Martinelli parameter, $X_r$.

Friedel [1979] developed a general correlation based on a database of 25,000 points including horizontal and vertical flow orientations in large tubes (the smallest diameter in the database was 4mm). Friedel found that the two-phase multiplier should take the following form:

\[
\Phi^2 = C_{F1} + \frac{3.24C_{F2}}{Fr^{0.045}We^{0.035}}
\] (2.20)

where the constants $C_{F1}$ and $C_{F2}$ are defined as:
\[ C_{F1} = (1-x)^2 + x^2 \left( \frac{\rho_l}{\rho_v} \right) \left( \frac{f_{x0}}{f_{lo}} \right) \]

\[ C_{F2} = x^{0.78} (1-x)^{0.24} \left( \frac{\rho_l}{\rho_v} \right)^{0.911} \left( \frac{\mu_v}{\mu_l} \right)^{0.19} \left( 1 - \frac{\mu_v}{\mu_l} \right) \] (2.21)

and We and Fr are the Weber and Froude numbers, respectively, given by:

\[ We = \frac{G^2 D}{\rho_p \sigma}, Fr = \frac{G^2}{gD\rho_p^2}, \rho_p = \left( \frac{x}{\rho_v} - \frac{1-x}{\rho_l} \right)^{-1} \] (2.22)

In a recent microchannel study, Zhang and Kwon [1999] modified the Friedel correlation to fit their two-phase friction data for a round port geometry (D = 2.13mm) using R-134a, R-22, and R-404A. Applying theoretical limits for qualities of 0 and 1, and replacing property groupings with a reduced pressure, they developed the following correlation:

\[ \Phi^2 = (1-x)^2 + 2.87 x^2 \left( \frac{P}{P_c} \right)^{-1} + 1.68 x^{0.8} (1-x)^{0.25} \left( \frac{P}{P_c} \right)^{-1.64} \] (2.23)

Several comparisons of this relation to previous data and other correlations were made. Agreement was found to be within 15% for most of the two-phase pressure drop data in small diameter tubes.

The small tube studies of Yang and Webb [1996a], Wambsganss et al. [1992], and Damianides [1988] all determined that the Lockhart-Martinelli parameter did not characterize their two-phase pressure drop data well. Yang and Webb found that the equivalent mass velocity concept as proposed by Akers [1960] tended to correlate their data much better for both their plain tube and micro-fin extruded aluminum microchannels (rectangular port with Dh of 2.64 and 1.56 mm, respectively). A two-
phase friction factor is defined in terms of an equivalent all liquid flow that will give the same frictional pressure drop as the two-phase flow, given by:

\[
f_{wp} = \frac{\Delta p}{G^2 / 2 \rho_l / 4L} = \frac{\Delta p}{\text{Re}_{eq}^2 \mu_l / 2 \rho_l / 4L}
\]

with the Akers equivalent mass velocity, \( G_{eq} \), defined as:

\[
G_{eq} = G \left[ (1 - x) + x \left( \frac{\rho_l}{\rho_v} \right)^{1/2} \right]
\]

and the equivalent all liquid Reynolds number is defined as:

\[
\text{Re}_{eq} = \frac{G_{eq} D_h}{\mu_l}
\]

In order to estimate the frictional pressure gradient, Yang and Webb empirically developed the following relation, which relates the two-phase and single-phase friction factor by:

\[
\frac{f_{wp}}{f_l} = 0.435 \text{Re}_{eq}^{0.12}
\]

The procedure to predict the two-phase frictional pressure drop is given as follows. First, calculate the liquid only Reynolds number and determine the liquid only frictional component (as discussed in section 2.1). Next, find \( \text{Re}_{eq} \) using equation (2.26) and then \( f \) from equation (2.27). The two-phase frictional pressure gradient can then be estimated using equation (2.24).

Yan and Lin [1999a,b] conducted evaporation and condensation pressure drop experiments of R-134a in an arrangement of 28 copper pipes with an inside diameter of 2mm and test section length of 0.2m. Utilizing the equivalent Reynolds number and the
Akers equivalent mass velocity, empirical correlations for the two-phase friction factor under evaporation and condensation conditions, respectively, are given by:

\[
\begin{align*}
  f_{vp} &= 0.11 \text{Re}_{eq}^{-0.1} \\
  f_{vp} &= 498.3 \text{Re}_{eq}^{-1.074}
\end{align*}
\]

The average deviations between these correlations and the experimental data presented for mass fluxes in the range of 100 – 200 kg/m²s were about 17% for both conditions.

### 2.4 Void Fraction

The investigation of void fraction in aluminum microchannel tubes will provide a basis to determine refrigerant charge requirements and, when combined with simultaneous pressure drop information, to evaluate probable and improbable flow configurations. Many large tube void fraction experiments have been conducted over the last half-century for applications in system simulation, heat transfer, and pressure drop correlations. Usually, the experiments were system specific and thus the correlations developed have not shown good agreement for refrigerants under typical operating conditions. A detailed description of existing void fraction correlations and techniques can be found in Graham [1998] and Wilson [1998].

Microchannel void fraction has not been thoroughly examined, however. Current techniques rely on flow visualization and extensive geometrical approximations while working with air and water or similar fluids.
A new microchannel test facility, which consists of both a refrigerant and air-water flow system, was constructed and completed in the Fall of 1999 at the University of Illinois at Urbana-Champaign. This chapter describes experimental apparatus, instrumentation, and the data collection techniques. An overview of the refrigerant flow system and the air-water flow system will be given followed by an examination of the main components in each. Procedures on the operation of the new systems will also be presented.

3.1 Experimental Test Facility

The facility consists of a refrigerant flow system, an air-water flow system, test sections, and a void fraction trapping mechanism. Each part of the test facility will be described in the following sections.

3.1.1 Refrigerant Flow System

A refrigerant flow system has been designed and constructed in order to simulate a wide range of typical air-conditioning and refrigerant operating conditions. Key design parameters include the ability to accurately control the test section quality, mass flux, and saturation temperature. To utilize a wide variety of refrigerants and refrigerant-oil combinations, a “once-through” flow system was developed as shown in Figure 3.1. The major components of the system are the high and low temperature refrigerant reservoirs,
pressure transducers, thermocouples, mass flow sensors, and the void fraction trapping system. The system is operated adiabatically in order to establish a reference for future research in which void fraction and pressure drop with heat transfer is evaluated.

Two reservoirs on each side of the test section hold 13.6 kg (30 lb) refrigerant tanks, each of which hold a liquid and vapor valve. One tank is heated in a water reservoir by a 1500 Watt immersion heater, whereas the other tank is submerged in an ice bath. This pressure difference, due to the different saturation temperatures of each tank, drives the flow. Saturated vapor and liquid flow separately out of the heated tank and into flow conditioners. The liquid line is sub-cooled by using a cross-flow heat exchanger connected to the campus chilled water system in order to eliminate the possibility of the liquid flashing to vapor while flowing through other system components. Similarly, to avoid condensing, the vapor line is superheated by one 1 x 12 inch, 76.6 ohm Minco® electrical heating strip connected to a 115 V Variac. After the thermocouples and pressure transducers measure the conditions of both lines, each line flows through a separate Micro-Motion® coriolis-type mass flow meter. Mass flow rates of both the liquid and vapor lines are controlled by simultaneously adjusting the needle valves directly after the flow meters in conjunction with the needle valve directly before the low temperature refrigerant tank. These valves are used to set the pressure level and pressure difference driving the flow by essentially “floating” the test section pressure towards either refrigerant reservoir tank. A series of ball valves on either side of the test sections control the flow direction to either test section or through a test section bypass. A vacuum pump is connected to evacuate the system after changing refrigerant tanks.
3.1.2 Test Sections

A variety of different test sections can be investigated in the microchannel experimental facility. The test section is comprised of two major components: extruded aluminum microchannel and end pieces. Two types of microchannel tubes have been studied and are shown in Figure 3.2. Hydraulic diameters as shown in the figure are directly measured on mounted and polished tube samples using a high-powered microscope. All tubes are taken from the same extrusion "push" so as to minimize port variability due to tool wear. The next major test section components are the end pieces, or transition sections, as seen in Figures 3.3 and 3.4. One transition section is required on each side of the microchannel. On the inlet side, the end piece must properly condition the flow, allow for property measurement, as well as provide a connection to the 3/8" copper tube. The end piece is able to obtain uniform port velocity and distribution by first expanding and then contracting the flow. An equal mix of liquid and vapor was shown flowing into the ports throughout the various flow regimes after observation through the sight glass in one of the inlet transition sections. Property measurement is accomplished by machining a pressure tap in the expansion block. An O-ring seal (Buna-N O23 - 1.5mm) seals the expansion block to the contraction block. A smooth flow transition is accomplished by a 14° contraction down to the exact microchannel size. Another O-ring seal (Buna-N O12 - 2, 2.5 mm for medium and high pressure applications, respectively) is placed over the microchannel and compressed by the next plate. In order to minimize thermal effects, which may cause leakage between the tube and transition section, all components are machined from aluminum. Round tube connections are made with brass NPT fittings and teflon thread seal.
3.1.3 Air-Water System

To extend the property range, an additional system was designed and completed at the University of Illinois at Urbana-Champaign in the Summer 2000. In the present state, the system uses air and water as the working fluids. Other fluids, such as glycol or oil, can be easily incorporated. The system, as shown in Figure 3.5, is designed to operate in a "once-through" fashion at adiabatic temperatures. Shop air near 100 psi is filtered and regulated by a Wilkerson® R18 Regulator to the desired pressure. Since the shop air is also used for the void fraction measurement system (described in Chapter 5), a reservoir tank is used to dampen airflow fluctuations due to pneumatic cylinder operation. To meter the flow rates of both liquid and gas, two needle valves located on either side of the various flow meters are used. In order to assure an accurate flow rate measurement for a wide range of airflows, three different Cole-Parmer® flow meters are arranged in parallel. For the water, a 1-7 GPM Cole-Parmer® water flow meter is used. After the air and water lines are combined, the temperature of the mixture is measured to determine fluid properties and sent to the test section. The test section is identical in design to those used in the refrigerant flow system as described in the previous section. Flow visualization can be incorporated easily by interchanging the test sections.

3.2 Instrumentation and Measurements

This section will describe the major components found in the flow systems and techniques used in measuring the different parameters of the experiment. These parameters include temperature, pressure, and mass flow rate.
3.2.1 Temperature Measurements

Type T thermocouples (Copper-Constanan) are used for all temperature measurements. Thermocouple wires are connected to a Cole-Parmer® 89500-00, -10 temperature panel meter. This instrument allows direct measurement and display for a variety of thermocouple types. For the Type T, temperatures in the range of -133.0 to 400.0 °C are quantified with a resolution of 0.1 °C and a manufacturer specified accuracy at 23°C ±5°C of ±0.05% ±0.9 °C.

3.2.2 Pressure Measurements

The microchannel test facility uses four pressure transducers. Three of the transducers are absolute pressure transducers, which are found both in the pure vapor and pure liquid refrigerant lines before the mass flow meters, and at the inlet to the test section. These variable reluctance pressure transducers are made by Validyne Engineering Corp. (DP15-54T1S4A) and have a range from 0-2100 kPa (0-300 psia) with an accuracy of ±0.25% of full scale. The last transducer is a variable reluctance differential pressure transducer, which is also manufactured by Validyne (DP15-44T1S4A), and measures the differential pressure drop across the test section with a range of 0-220 kPa (0-32 psid) and an accuracy of ±0.25% of full scale. A dead weight balance was used to calibrate all transducers to the desired range and accuracy. Pressure transducer lead wires are connected to the Validyne CD280-4, four channel carrier demodulator. It provides regulated 5 V rms, 5 kHz transducer power with a 1 kHz frequency response. The ±10 Vdc outputs from the demodulator are sent to a panel of Omega® DP24-E process meters. These meters are calibrated for the 0-10 Vdc input to a
4-digit display with a resolution of 0.1 and 0.01 for absolute pressure and differential pressure measurements, respectively. Differential pressure measurement error was experimentally determined to be ±0.42 kPa (0.061 psi) by comparing the transducer measurement with a water manometer. Barometric pressure was measured by using a Robert E. White Instruments M111 Marine Barometer.

3.2.3 Mass Flow Measurements

Two flow meters are used for the refrigerant system. For both the liquid and vapor lines, a Micromotion® Elite® (CMF010M323NU) mass flow sensor is used. This meter uses the coriolis flow metering system in which the vibrational frequency of the internal U-tube is measured and the meter delivers a specific current based on this frequency. The signal is sent to the Micromotion® (RFT9739D4SUA) field-mount transmitter with a digital display. This combination produces an accuracy of ±0.10% and ±0.50% of the rate for the liquid and vapor flow rates, respectively. The nominal flow range is 0-82 kg/h (0-3 lbm/min) with a maximum flow rate of 108 kg/h (4 lbm/min).

3.3 Calculated Parameters

Parameters that cannot be measured directly from instrumentation need to be calculated during the data reduction. These include the liquid and vapor line enthalpies, mass flux, enthalpy entering the test section, and the test section quality.
3.3.1 Mass Flux

For each microchannel tube under investigation, the port area needs to be carefully evaluated to determine the mass flux. Mass flux is determined by summing the liquid and vapor mass flow rates recorded by the Micromotion® flow meters and then divided by the appropriate cross-sectional area as given in Figure 3.2. The uncertainty in mass flux measurement is a combination of flow meter error with microchannel port area measurement error and determined to be ±5%.

3.3.2 Test Section Inlet Properties

Since the refrigerant system flows saturated liquid and vapor separately prior to the test section inlet, a mass and energy balance must be used to calculate the test section properties. Engineering Equation Solver (EES) v6.036, developed by S.A. Klein and F.L. Alvarado [1992-2000], was used to determine the property curve fits of both R134a and R410A. The program as seen in Appendix A calculated the various test section properties. Enthalpies of both the liquid and vapor lines are calculated initially by using the temperature and then later verified to be within saturation conditions by the finding the saturation pressure associated with the measured temperature and comparing. The enthalpy in the adiabatic test section is determined by using the energy balance, as seen in Equation 3.1 below:

\[ \dot{m}_{ls} h_{ls} = \dot{m}_v h_v + m_l h_l \]  

(3.1)

where

\[ \dot{m}_{ls} = \dot{m}_v + m_l \]  

(3.2)
and the subscripts \(ts\), \(v\), and \(l\) denote locations in the test section, vapor line, and liquid line, respectively. Using the calculated test section enthalpy combined with measured test section properties, including pressure and temperature, determine the test section quality. EES also calculates the density and viscosity of both liquid and vapor component of the two-phase fluid.

### 3.4 Uncertainty Analysis

The total uncertainty inherent in a calculation is a combination of the bias and precision components of error associated with the measurements. Bias error, also referred to as the fixed error, generally exist due to instrument calibration, data acquisition, and data reduction. Precision error, also called the random or repeatability error, may be caused by random electronic fluctuations or the inability to maintain true steady state conditions. The errors associated with the measurement of experimental quantities are shown in Table 3.1.

#### Table 3.1 Bias and Precision Errors in Measured Quantities

<table>
<thead>
<tr>
<th>Measured Quantity</th>
<th>Bias Error</th>
<th>Precision Error</th>
<th>Total Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow Liquid line [kg/s]</td>
<td>0.10%</td>
<td>±0.001</td>
<td>±0.0014</td>
</tr>
<tr>
<td>Mass flow Vapor line [kg/s]</td>
<td>0.50%</td>
<td>±0.001</td>
<td>±0.0051</td>
</tr>
<tr>
<td>Pressure Vapor line [kPa]</td>
<td>±8.6</td>
<td>±0.7</td>
<td>±8.628</td>
</tr>
<tr>
<td>Temperature Vapor line [°C]</td>
<td>±0.25</td>
<td>±0.15</td>
<td>±0.292</td>
</tr>
<tr>
<td>Pressure Liquid line [kPa]</td>
<td>±8.6</td>
<td>±0.7</td>
<td>±8.628</td>
</tr>
<tr>
<td>Temperature Liquid line [°C]</td>
<td>±0.25</td>
<td>±0.15</td>
<td>±0.292</td>
</tr>
<tr>
<td>Pressure Test section inlet [kPa]</td>
<td>±8.6</td>
<td>±0.7</td>
<td>±8.628</td>
</tr>
<tr>
<td>Differential Pressure Test section [kPa]</td>
<td>±0.57</td>
<td>±0.07</td>
<td>±0.575</td>
</tr>
<tr>
<td>Temperature Test section outlet [°C]</td>
<td>±0.25</td>
<td>±0.15</td>
<td>±0.292</td>
</tr>
<tr>
<td>Pressure Atmospheric [mbar]</td>
<td>±0.5</td>
<td>±0.5</td>
<td>±0.707</td>
</tr>
</tbody>
</table>
The uncertainty for the calculated parameters depends on the initial measurement error, which propagates through the calculation. To determine the propagation of error, the uncertainty propagation feature within EES was used. All sources of error were entered into EES and the total uncertainty for the desired calculation was determined. The quality calculation includes the largest amount of error propagation since it is dependent upon enthalpy, temperature, pressure, and mass flow rate either directly or indirectly. It was found that low qualities have the largest error whereas high qualities have an error of approximately 1%. The average error associated with a quality of 0.10 is about 10%.

3.5 Refrigerant System Preparation and Operation

This section discusses the various steps that were taken to prepare the refrigerant flow system for operation. The manner in which the facility is operated for data acquisition will also be discussed in this section.

3.5.1 Refrigerant System Preparation

Once the refrigerant flow system was constructed, several steps needed to be taken prior to operation. The system was first checked for leaks in all sections. First, the system is pressurized to 1000 kPa (145 psi) and spray tested with Big Blu® micro leak detector on all fittings. If no leak is detected, the system is sectioned off and continuous pressure and temperature measurements are taken. If the mass remains constant for a given amount of time, by using the ideal gas law, the system is leak free. More accurate
leak detecting utilizes the Yokogawa universal service leak detector (H-10G) with the system charged with R134a vapor.

Once the refrigerant system is free of leaks, it is ready for operation. First, the refrigerant tanks need to be prepared. One empty tank is vacuumed and submerged in an ice bath to be charged with liquid and vapor refrigerant. The other tank is vacuumed. This process is only required for the first time the system is use the tanks. After initial operation, the tanks can just be switched without further charging or evacuation. The evacuated tank is then placed in the ice-water reservoir of the refrigerant flow system and connected with a charging hose. The charged tank is submerged in the hot water reservoir. Both tanks are clamped down to resist floating as the tanks empty. Hot water is turned on to fill the reservoir even with the neck of the tank. Next, the immersion heater is switched on. At this point, it is advised to turn on all the pressure transducers, thermocouples, and mass flow meters to continuously monitor set-up conditions. A thermocouple is placed in the hot water reservoir to ensure that the temperature does not exceed a value for which the saturation pressure is larger than 2070 kPa (300 psi) and the burst valve actuated. Charging hoses for the liquid and vapor lines can be connected to the refrigerant-filled tank. All valves on the tanks are to remain closed until the system is completely evacuated and the hot water reservoir reaches the desired temperature (about 35°C for R134a and around 30°C for R410A).

The test sections need to be installed and leak tested before a series of experiments can be conducted. As mentioned previously, two different O-ring seals are used depending on the pressure level. For R134a, the 2mm O-ring seal should be used and the O-ring plate tightened down completely. R410A, on the other hand, requires a
slightly larger O-ring seal due to the higher pressures. The plate is not to be tightened down completely, however, since compression of the microchannel test section can occur. The test section is completely isolated by a series of ball valves and connected to the compressed nitrogen supply for leak testing. Big Blu® is used at all transition section plates while successively increasing the nitrogen supply pressure to near operating conditions. If a leak is detected, the O-ring plate either needs to be tightened further or the O-ring itself needs to be repositioned or replaced.

After the test section is able to hold pressure, all valves in the flow system are to be opened with the exception of valves on the refrigerant tanks themselves. The vacuum pump is activated to evacuate the entire system. Pressure transducers should accurately reflect the vacuum conditions and the differential pressure transducer should be zeroed. Next, all valves should be closed and sections isolated. At this point, the chilled water loop and 115V Variac should be turned on (for additional subcooling of liquid and superheating of vapor, respectively).

3.5.2 Refrigerant System Operation

After proper preparation of the refrigerant flow system, it is ready for operation and data acquisition. Valves on the charged tank can be opened gradually and the liquid and vapor lines filled. The valve on the evacuated tank can also be opened. Initial refrigerant flow should bypass the test section and flow directly to the cold tank by opening the needle valves slightly as well as opening the test section bypass ball valve. The variac should be adjusted so that condensation is avoided in the vapor line. Bypass of refrigerant flow should be continued until condensate is no longer present in the vapor
line in addition to the lack of flashing or evaporation in the liquid line. This is required in order to achieve the most accurate mass flow and property measurements.

Once the condensation and evaporation in the lines are clear, the bypass valve is shut and flow is directed through the test section by opening the other ball valves. Since key properties within the test section are calculated after data acquisition, an estimate for these properties needs to be determined to obtain the desired testing conditions. A mass balance is used for a rough estimate for the test section quality based on the separate liquid and vapor mass flow rates. This method was found to be relatively accurate since a nearly equal amount of vapor superheating and liquid subcooling is used. Mass flux is estimated in conjunction with quality where the total mass flow rate from both lines must sum to meet the desired mass flow which corresponds to a given mass flux based on the test section geometry. Test section saturation temperature is measured directly and controlled by the needle valves. A fully open valve near the cold reservoir brings the test section temperature near this condition. Alternatively, a mostly closed needle valve near the cold reservoir with fully open control valves raises the test section saturation temperature.

The refrigerant flow system is very versatile in the microchannel flow conditions it is able to investigate from a variety of refrigerants, test sections, and operation conditions. Its single pass design eliminates pumps and many other components, which may cause added turbulence and variation within the test section. Two test sections allow for continuous data acquisition at a particular operating condition.
3.6 Air-Water System Operation

The following steps should be taken to start-up and operate the air-water facility. Shop air supply line should be fully opened and the regulator valve set to 340 kPa (50 psi). Next, the test section needs to be prepared in the same fashion as previously described. Differential pressure measurement across the test section is determined by using a water manometer. The mass flow is set by opening the ball valve associated with the desired air flow meter while metering both the air and water flow rates with needle valves. Two drain valves, located after the test section, are opened to initiate flow.

3.7 Figures

A schematic of the refrigerant flow system, the microchannels under investigation, the transition section end pieces, and the air-water system are shown here.
Figure 3.1 Refrigerant System Schematic
6-Port Microchannel
\[ D_h = 1.35 \text{ mm} \]
\[ A_{\text{Cross-Section}} = 1.269 \times 10^{-5} \text{ m}^2 \]

14-Port Microchannel
\[ D_h = 0.91 \text{ mm} \]
\[ A_{\text{Cross-Section}} = 1.151 \times 10^{-5} \text{ m}^2 \]

Figure 3.2 Microchannels Under Investigation
Figure 3.3 Aluminum Transition Pieces
Figure 3.4 Orientation of Transition Pieces

1/4"-20 x 3/4 Bolts (in 4 Locations)

3/8" OD Copper

Pressure Tap

O-ring Seals
Hot and Cold water source

Water Flow meters

1-7 GPH

To Pneumatic Cylinders

2-10 CFH

1-8 CFM
10-70 CFH

1/2" PVC to Drain.

Air Flow meters

Pressure Regulator

100 psi Air line

Figure 3.5 Air-Water System Schematic
Chapter 4
Pressure Drop in Microchannels

This chapter discusses the experimental results obtained from the microchannel research facility. First, the losses in the end pieces of the test section will be presented followed by the single-phase and two-phase pressure drop results isolated to the specific microchannel tube. Lastly, a comparison of experimental data with existing pressure drop correlations will be discussed.

4.1 Transition Section Pressure Losses

In order to accurately describe the characteristics of the microchannel tube, pressure drop measurement through each component should be carefully examined. As described in Chapter 3, the pressure taps for the test section are contained in the expansion chamber of the end pieces. Thus, the differential pressure measurement includes both a flow contraction into the tube and an expansion out, and is referred to as the total pressure drop across the test section. This quantity was directly measured experimentally for the full operating range of the microchannel facility.

Due to the versatility of the microchannel facility, two transition sections could be placed directly together with a short piece of microchannel tube. Joined by a 7.62 cm (3") microchannel, the entrance and exit pressure losses dominate the total pressure drop. Utilizing the homogeneous flow model parameters for contraction and expansion devices suggested by Collier and Thome [1994], a correlation for transition section pressure drop
for both types of microchannel tubes is developed. The pressure difference across a
sudden contraction for homogeneous flow is given by:

\[ p_2 - p_1 = \frac{G^2}{2 \rho_l} \left[ \left( \frac{1}{C_c} - 1 \right)^2 + \left( 1 - \frac{1}{\sigma^2} \right) \right] \left[ 1 + x \left( \frac{1}{\rho_v} - \frac{1}{\rho_l} \right) \frac{\rho_l}{\rho_v} \right] \]  

(4.1)

and the pressure difference across an expansion is:

\[ p_2 - p_1 = \frac{G^2 \sigma(1-\sigma)}{\rho_l} \left[ 1 + x \left( \frac{1}{\rho_v} - \frac{1}{\rho_l} \right) \frac{\rho_l}{\rho_v} \right] \]  

(4.2)

where \( C_c \) is a coefficient of contraction and \( \sigma \) is the expansion to contraction area ratio.

These equations are similar in that a constant factor, which is area dependent, is
multiplied by the average fluid kinetic energy,

\[ \frac{G^2}{2 \rho_l} \left[ 1 + x \left( \frac{1}{\rho_v} - \frac{1}{\rho_l} \right) \frac{\rho_l}{\rho_v} \right] \]  

(4.3)

in order to obtain the desired inlet and exit pressure drop. For the development of a more
accurate correlation, a data reduction method was used to determine the area dependent
coefficient specific to the transition pieces. A comparison of the measured transition
section pressure drop with this average kinetic energy term is shown in Figure 4.1. For
both R-134a and R-410A, two slopes are shown that are dependent on the number of
ports in the microchannel tube. This slope is the transition section specific coefficient in
which both the inlet contraction and exit expansion are described. Thus, using this value
for a given microchannel combined with the average kinetic energy of the flow yields the
pressure loss in the end pieces. Application of the transition section specific area
coefficient and kinetic energy term in comparison to measured data shows very close
agreement. Figure 4.2 displays the agreement of the transition section prediction within ±10% of the experimentally measured pressure drops.

It is important to evaluate the overall impact of the transition section pressure drop in comparison to the total pressure drop across the test section. Direct application of the sudden contraction and expansion pressure loss equations yields a significantly lower pressure loss than measured. The modified correlation is used to predict the transition section pressure drop and compared to the total pressure drop, as seen in Figure 4.3. This plot shows the influence of the transition section effects as compared to the total test section for two different microchannel tube lengths and over a large fluid property range. On average, the transition section pressure drop accounts for 10% of the total pressure drop. Using the proposed equations from Collier and Thome, the transition section pressure drop only accounts for 1-2% of the total pressure drop.

Transition section pressure drops becomes increasingly significant as the test section length is decreased and therefore, cannot be neglected. Ultimately, inlet and exit pressure loss information allows the microchannel test section pressure drop to be isolated from the total measured pressure drop.

4.2 Single-Phase Pressure Drop Results

There are two very important results that are derived from this study of single-phase refrigerant pressure drop in microchannels. First, it provides a theoretical limit for the two-phase pressure drop as the test section qualities approach pure liquid and vapor characteristics. Secondly, it will help determine if small length scale or surface
roughness effects are seen in these microchannel tubes with respect to single-phase refrigerant flow.

A common method for presentation of single-phase pressure drop results is by obtaining the representative friction factor. This provides a non-dimensional basis for comparisons among a wide variety of parameters. As discussed in Chapter 2, the Churchill equation, which provides a smooth transition of laminar, transitional, and turbulent friction factor equations, can be used to predict the frictional behavior throughout the entire Reynolds number range. Reynolds numbers, for all data, were calculated using the measured hydraulic diameter. Inaccuracy in the hydraulic diameter can greatly affect the overall trend of friction factor versus Reynolds number.

Visual measurement of microchannel ports was utilized in order to determine the hydraulic diameters for both the 6-port and 14-port microchannel tubes. An initial value was calculated using the manufacturer's specifications based on the extrusion die design. Each microchannel type was directly measured using a high-powered microscope with an accuracy of ±0.05mm. Sample tube cross sections were filled with epoxy and polished. Representative ports were then measured, and the hydraulic diameters recorded. It should be noted that the nominal port dimensions and measured dimensions were very different. Nominal hydraulic diameters were found to be 1.64mm and 1.26mm (compared to 1.35mm and 0.91mm as optically measured) for the 6-port and 14-port microchannel tubes, respectively. This difference may be attributed to manufacturing variability (die wear) and measurement error. Since this hydraulic diameter difference greatly influences the overall frictional loss characteristics, the behavior with respect to both values will be evaluated.
Several experimenters have utilized a variable length scale to account for the difference between the measured and predicted friction factor. Using the optically measured value for the hydraulic diameter, Figure 4.4 shows the Darcy friction factor for both liquid and vapor R134a flowing through the 6-port microchannel tube at a variety of Reynolds numbers. The Churchill equation tends to predict the behavior in the true laminar and turbulent regimes. It can be seen that a delayed liquid transition to turbulence occurs. This is believed to be due to the steady flow nature of the experimental system. Due to the absence of pressure fluctuations from pumps, motors, or other reciprocating equipment in the refrigerant flow system, laminar flow may be sustained at higher Reynolds numbers. Near the transition region, however, a difference is seen between the liquid and vapor data points. This behavior is subject to several different explanations. First, it may be due to an inaccuracy in the vapor mass flow measurements. The low Reynolds number vapor flow is located in an increasingly less accurate regime of the Mircomotion® mass flow meters. Errors in this region may be as large as 0.73% of full scale [kg/min] for the lowest vapor flow rates. Secondly, this may be due to pressure transducer error. Pressure transducer lines were checked to determine whether the difference between vapor and liquid measurement might be due to liquid trapped in the sensor lines. The sensor lines were heated to ensure vapor only when liquid refrigerant passed through the test section. No indication of transducer error was indicated. However, since measured vapor pressure drops across the test section for low Reynolds numbers are relatively small, experimental error is a larger percentage of the total measured pressure drop. Another possible explanation for the liquid to vapor friction factor discrepancy could be due to condensing vapor in the test section. This
would create a two-phase effect and essentially give the originally smooth microchannel an added surface roughness $(e/D)$ factor. A further study utilizing a non-condensable gas in addition to minimizing the mass flow and pressure transducer measurement inaccuracies needs to be conducted.

Figure 4.5 is the single-phase friction factor behavior using the optically determined hydraulic diameter for both a change in refrigerant and test section type. A 14-port microchannel tube is used with R410A. The same trends as discussed for the 6-port, R134a friction factor shown in Figure 4.4 are observed in Figure 4.5. Thus, it is concluded that the Churchill equation can be used to characterize the single-phase pressure drop for a variety of refrigerants and tube designs. Changing the hydraulic diameter to the nominal value does not show a better agreement with the Churchill friction factor prediction equation as seen in Figures 4.6 and 4.7. This change to a larger value increases the total friction factor and also influences the Reynolds number calculation. Due to the significant variation of this measured value, a more careful representation is needed in order to determine an effective microchannel hydraulic diameter. Direct measurement of the hydraulic diameter is important for a general representation of the microchannel tubes investigated. Correlations tend to show that a further study be conducted in order to effectively determine the physical representation for a microchannel tube hydraulic diameter. From this investigation, it is assumed that the single-phase pressure drop can be predicted in the laminar and turbulent Reynolds number flow regimes for the two geometrical tube designs and fluids by determining the tube hydraulic diameter and using the basic laminar and turbulent flow models.
4.3 Two-Phase Pressure Drop Results

Two-phase flow conditions are generated within the test sections by the refrigerant flow system as discussed in Chapter 3. In reference to Figure 2.1, which shows possible two-phase flow regimes, the influence on pressure drop due to the variety of flow configurations should be considered. Distinct pressure drop characteristics are to be expected as flow transitions are encountered. Thus, the pressure drop behavior for the microchannel tubes in question will aid in the prediction of possible two-phase flow regimes and the overall in-tube refrigerant characteristics for a set of system operating conditions. The two-phase pressure drop experiment was conducted according to the following test matrix:

<table>
<thead>
<tr>
<th>Quality, x [-]</th>
<th>R134a</th>
<th>R410A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass Flux, G [kg/m²s]</td>
<td>75, 150, 300</td>
<td>75, 150, 300</td>
</tr>
<tr>
<td>Temperature [°C]</td>
<td>10, 20, 35</td>
<td>10, 20</td>
</tr>
<tr>
<td>Tube Geometry [# ports]</td>
<td>6, 14</td>
<td>6, 14</td>
</tr>
<tr>
<td>Heat Flux [W/m²]</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 4.8 shows several important trends. First, the pressure drop increases with quality. With increasing qualities, a higher vapor mass proportion is found in comparison to lower qualities. This causes the vapor velocity to increase for a constant temperature and mass flux. Because the frictional pressure drop is directly related to the square of the velocity, a higher overall pressure drop is found for increasing quality. An additional trend observed in Figure 4.8 is with respect to mass flux dependence. A significant change in the observed pressure drop is found for mass fluxes above 150 kg/m²s, on average. Over the range of temperatures, there is little to no change in the pressure drop.
values for all mass fluxes below this 150 kg/m$^2$s value. For mass fluxes above this value, however, the pressure drop increases aggressively with increasing mass flux. The general trend for the pressure drop is semi-quadratic, where a maximum pressure drop is reached at a quality in the 80-90% range. For larger tubes, this trend was common for annular flow regimes. Possibly, the flow regime transitions from an intermittent (slug-plug) flow to an annular type flow at a mass flux of approximately 150 kg/m$^2$s.

Figure 4.9 displays the temperature independence of pressure drop for a lower mass flux, assumed intermittent, flow. For refrigerant flow in larger tubes, a decreasing temperature yields a higher pressure drop for a given quality and mass flux. This is due to a decrease in the vapor density as temperature decreases. For a fixed mass flux ($G = \rho v$), this causes an increase in the vapor velocity and thus a higher pressure drop. At low mass fluxes, however, this trend is not clearly seen. Figure 4.9 shows refrigerant temperature variations for a mass flux of 150 kg/m$^2$s with the 6-port microchannel tube. Low temperature data points tend to be above the higher temperature points, but no distinct difference can be found. Little to no variance with temperature was found for the higher two temperatures. Therefore, temperature has a secondary effect on refrigerant pressure drop in microchannels, with little to no influence at the low mass flux.

Figure 4.10 is a plot of the pressure drop versus quality for both tube geometries at 3 different mass fluxes. It is seen that there is no distinct difference between the two tube geometries at the lowest mass flux, 75 kg/m$^2$s. At 150, however, a slight change can be seen for the 14-port tube in comparison to the 6-port. As expected, due to the increased surface area of the 14-port tube geometry, it exhibits a higher pressure drop. Because mass flux is the primary factor, it is believed that at this value, a flow transition
to annular type flow occurs within the 14-port tube while the 6-port tube remains in the intermittent regime. At higher mass fluxes, where both tubes are believed to be in the annular type flow regime, the 14-port tube exhibits a significantly higher pressure drop.

Figure 4.11 displays the pressure drop trends versus quality for R410A at various mass fluxes. It is shown that R410A exhibits similar quality, temperature, and mass flux trends as found with R134a. The assumed flow regime transition implied by the aggressive increase in observed pressure drops appears to occur at a higher mass flow rate than found with R134a. As shown on the figure, for mass fluxes below 180 kg/m²s there is little to no change in the pressure drop behavior with large changes in temperature, mass flow, or quality. Also, for a higher mass flux where annular type flow is assumed, a maximum pressure drop is observed at a lower quality as compared to R134a. R410A tends to reach a maximum at a quality between 60 – 70% whereas the R134a maximum was found to be between the 80-90% range.

Overall, R410A should display a slightly lower pressure drop than R134a due to the difference in the fluid density. R410A vapor has a higher density than does R134a, thus for a given mass flow rate, R134a vapor exhibits a higher velocity than R410A vapor and thus a higher frictional pressure drop. Figure 4.12 shows this difference in pressure drop between R134a and R410A in the 14-port tube at 20 °C and mass fluxes near 350 kg/m²s. The difference in the qualities where the maximum pressure drop occurs is easily shown. Additional pressure drop versus quality plots for a wide range of operating conditions with both refrigerants and tube geometries can be found in Appendix B.
4.4 Correlation Comparisons

This section will compare data taken from this investigation to several correlations as described in Chapter 2, Literature Review. Both large tube and microchannel correlations are discussed with respect to both the optically measured and nominal hydraulic diameters. Figure 4.13 shows that the large tube pressure drop prediction developed by Friedel [1979] is in poor agreement with the microchannel tube data for the optical hydraulic diameter. 14-port data collapses well for both R134a and R410A data. On the other hand, there appears to be a distinct difference between refrigerants for the 6-port data. R410A data seems to be predicted well whereas R134a data is overpredicted by the Friedel correlation. This may be due to the data reduction techniques used for the 6-port, R410A experiment. Because a larger O-ring needed to be used, this can influence the total pressure drop across the test section. Due to the high pressure levels of R410A, a larger compression O-ring was required in the transition sections. This caused a small deformation of the 6-port microchannel tube if excessive levels of compression were obtained. An additional pressure drop may have occurred across these O-ring depressions. Further investigation is required to determine if the larger O-ring influences the overall pressure drop behavior or if a flow phenomenon is encountered.

Figure 4.14 is a comparison of data with a microchannel correlation developed by Yang and Webb [1996 a]. Microchannel tubes as used in the current investigation are considerably smaller than the microchannels that were used to develop the correlation. It is seen from the figure that the data is neither correlated nor predicted with a similar magnitude of pressure drop when using the optically measured hydraulic diameter.
Initially this difference was believed to be due to an inaccurate classification of the entrance and exit effects. Those losses appear to have been empirically determined. After applying the derived transition section correlation, the data is somewhat overpredicted but continues to display a lack of correlation. Thus, due to the amount of discrepancy between data in the comparison, the microchannel pressure drop correlation as described may not incorporate the appropriate microchannel flow characteristics.

Because the large tube Friedel correlation tended to describe the flow properties of the microchannel, other large tube pressure drop correlations were applied. Figure 4.15 shows the Friedel, Souza [1993], and Jung [1989] correlations in comparison to experimental pressure drop data. All three predict a higher than measured pressure drop for the optical hydraulic diameter. However, good correlation is seen with all three models. It is believed that this is due to the nature of the correlations. These large tube correlations were developed for annular or mostly annular flow regimes. It was found that mass flow rate is the most influential parameter with respect to microchannel pressure drop characteristics and a flow transition to annular-like flow is believed to exist at the higher flow rates. These correlations tend to correlate the pressure drop behavior at these high flow rates due to this similarity in flow regimes. It was assumed that the microchannel prediction failed to classify the flow properties with respect to pressure drop.

An additional microchannel correlation as developed by Zhang [1999] was also examined. This correlation is compared to the microchannel relation presented by Yang and Webb, as shown in Figure 4.16, and displays a similar difficulty in correlating the data from this investigation for the optical hydraulic diameter.
Since hydraulic diameter was found to significantly influence the prediction capabilities for single-phase flow, a similar analysis was conducted for two-phase flow correlations. Figure 4.17 shows the Friedel [1979] prediction using the nominal hydraulic diameter. When comparing to Figure 4.13, the previous Friedel correlation using the optically determined hydraulic diameter, a significant difference is seen. First, the data moves in magnitude from overpredicted to underpredicted as compared to the pressure drop prediction. This is consistent with many other microchannel investigations where current correlations are used. General agreement with the correlation does not change and the flow physics incorporated into the correlation still describes pressure drop behavior. The discrepancy with the 6-port, R410A data is still seen, however. This is believed to be due to an O-ring depression, which may cause an additional pressure drop, but further investigation is required. 14-port data is now on the same level of magnitude as the 6-port, R410A data. This is due to the order of magnitude difference caused by the change in hydraulic diameter and cross-sectional area for each microchannel tube.

Figure 4.18 shows the effect of the change to the nominal hydraulic diameter for the Yang and Webb [1996 a] correlation. Data is still in poor agreement with the pressure drop prediction. However, the change of approximately 0.3 mm between the two hydraulic diameters changed the total magnitude of prediction by nearly a factor of 5, from overpredicting to underpredicting the microchannel pressure drop. These same trends are once again shown for all large tube and microchannel correlation comparisons with the nominal hydraulic diameter as found in Figure 4.19 and 4.20, respectively. The general grouping of the large tube correlations remains constant. Thus, it is seen that due to the change in measured to nominal hydraulic diameter, microchannel tubes should be
characterized more carefully with respect to the actual physical behavior the microchannel tube exhibits in relation with tube hydraulic diameter.

In general, this large discrepancy should lead investigators to carefully determine the actual tube hydraulic diameter. It is believed that the actual microchannel tube hydraulic diameter in this investigation is between the optically measured and nominal value. Due to the significant change in prediction behavior with respect to tube hydraulic diameter and possible small tube depressions, heat exchanger design should be more carefully examined. Brazing, welding, and forming processes each may change the overall microchannel structure and create a large discrepancy between the design and actual operation parameters. This investigation utilized a single microchannel tube, which is free of additional heat exchanger manufacturing processes. Thus, in order to insure accurate representation with respect to two-phase pressure drop correlations in microchannels, a further study that accounts for manufacturing variability and the flow physics required to characterize the tube hydraulic diameter needs to be conducted.

4.5 Figures

Several figures are included here that contain plots for the transition section losses, single- and two-phase pressure drop, and a comparison of experimental two-phase pressure drop to existing correlations. Additional plots for two-phase pressure drop are found in Appendix B. Additional figures for both large tube and microchannel correlations compared separately for both the optically measured and nominal hydraulic diameters are found in Appendix C.
Figure 4.1 Transition Section Pressure Drop versus Average Kinetic Energy

\[
\frac{G^2}{2\rho_i} \left[ 1 + \rho_i \left( \frac{1}{\rho_v} - \frac{1}{\rho_i} \right) x \right]
\]

Figure 4.2 Transition Section Predicted versus Measured Pressure Drop

50
Figure 4.3  Influence of Transition Section Pressure Drop

Figure 4.4  Single-Phase R134a, Friction Factor versus Reynolds Number using Optical $D_h$
Figure 4.5 Single-Phase R410A, Friction Factor versus Reynolds Number using Optical $D_h$

Figure 4.6 Single-Phase R134a, Friction Factor versus Reynolds Number using Nominal $D_h$
Figure 4.7  Single-Phase R410A, Friction Factor versus Reynolds Number using Nominal $D_h$

![Friction Factor versus Reynolds Number](image1)

Figure 4.8  Pressure Drop versus Quality for 6-port, R134a at 20 °C

![Pressure Drop versus Quality](image2)
Figure 4.9  Pressure Drop versus Quality for 6-port, R134a at Constant Mass Flux

Figure 4.10  Pressure Drop versus Quality for R134a and Both Tube Geometries
Figure 4.11  Pressure Drop versus Quality for 14-port, R410A at 20 °C

Figure 4.12  Pressure Drop versus Quality for 14-port, R134a and R410A at 20 °C
Figure 4.13 Friedel [1979] Prediction versus Pressure Drop using Optical $D_h$

Figure 4.14 Webb [1996 a] Prediction versus Pressure Drop using Optical $D_h$
Figure 4.15  Large Tube Correlations versus Pressure Drop using Optical $D_h$

Figure 4.16  Microchannel Correlations versus Pressure Drop using Optical $D_h$
Figure 4.17 Friedel [1979] Prediction versus Pressure Drop using Nominal $D_h$

Figure 4.18 Webb [1996 a] Prediction versus Pressure Drop using Nominal $D_h$
Figure 4.19  Large Tube Correlations versus Pressure Drop using Nominal $D_h$

Figure 4.20  Microchannel Correlations versus Pressure Drop using Nominal $D_h$
Chapter 5
Void Fraction Techniques

This chapter explains the development of the refrigerant trapping system to be used in a later phase of the project. Success of the system is critical for determination of the void fraction in microchannels for a variety of flow conditions.

5.1 Refrigerant Trapping System Concepts

In large tube experiments, a simple linked-bar mechanism could be used to simultaneously close ball valves at either end of a test section. As test section volumes are decreased, however, the volume enclosed in the valve body becomes a significant percentage of the total test section volume. Thus, void fraction measurements become increasingly less accurate as test section volumes decrease.

Because the microchannel volume is considerably less than the large tubes of previous void fraction investigations, a new refrigerant trapping technique needed to be designed. This new technique was to effectively crimp and seal the microchannel at the two ends of the test section. One main method has been used to perform this task. Pneumatic cylinders equipped with a blade and connected to 80 psi shop air were chosen. This included several variations in blade design, pneumatic cylinder design, and overall trapping system operating parameters. This method should prove to be accurate for microchannel void fraction determination. Visualization and geometric approximation void fraction techniques leave room for human error and interpretation. Therefore, the trapping system as designed will provide more accurate microchannel void fraction data.


5.2 Refrigerant Trapping System Design

The refrigerant trapping system houses three major components. First, a steady supply of compressed air must be available. As described in the air-water system, the compressed air is filtered, regulated and stored in a reservoir. From here, it can either be used for the air-water system or for the pneumatic cylinders. The second major component distributes and actuates the compressed air to both cylinders for simultaneous sealing of the microchannel at the two test section ends. Two air solenoid valves are used and actuated by a switch on the control board. These valves distribute air flow to the two pneumatic cylinders equipped with blades, the third major component. Various designs contained different configurations with respect to the size, number and orientation of cylinders. The latest design uses a 4” bore, Bimba Flat Line Series, cylinder securely attached to a lever arm as shown in Figure 5.1. The lever arm houses the upper blade assembly. It was found that the most probable configuration for the blades was to have one on each side of the microchannel applying pressure to the same area. Thus, the lower blade assembly, fixed to the lower strut braces, was included. It features an interchangeable blade so that many different blade designs can be tested.

As of the current date of publication, the refrigerant trapping system is unable to seal the microchannel test section 100% of the time. Gradual sharpening of the blades will decrease the contact area and increase the total pressure applied to the microchannel. Thus, it is believed that after further evaluating the blade design, the microchannel refrigerant trapping system will be able to seal any given test section and microchannel void fraction measurements will be made available.
Figure 5.1 Refrigerant Trapping System Schematic
Chapter 6

Conclusion

The initial investigation of void fraction and pressure drop in aluminum microchannels has been conducted. This included the design and construction of the new microchannel testing facility, evaluation of the microchannel pressure drop for a variety of flow conditions, and the development of the refrigerant trapping system.

Pressure drop studies were conducted adiabatically over the entire range of qualities (0-1), mass fluxes from 75 to 350 kg/m²s, 3 temperatures from 10 to 35 °C for R134a and R410A using 2 square-port microchannels with hydraulic diameters of 0.91 and 1.35mm. The single-phase frictional pressure drop showed that both laminar and turbulent friction factor equations are able to predict the frictional losses in microchannels. Thus, no "small-tube" effects need to be compensated in the analysis. However, a further study needs to be conducted, which utilizes a non-condensable gas in addition to minimizing the mass flow and differential pressure measurement errors, to resolve the single-phase behavior as described in this investigation. Two-phase pressure drop was found to be mass flux dependent. For mass fluxes greater than 150 kg/m²s, a flow regime transition was believed to occur. Pressure drops increased significantly with mass flow while displaying annular-like behavior. Below this value, the pressure drop has little dependence on mass flux. Temperature was a secondary parameter and showed little to no pressure drop change for the entire range.

The new microchannel testing facility was constructed. The refrigerant trapping system has been developed and tested so that microchannel void fraction experiments
may be conducted. A complimentary study investigates the pressure drop for other fluids and develops microchannel fluid flow visualization. Future work involves determining the probable microchannel flow configurations for building physical fluid flow models.

It is important to note the significant change when comparing data with two-phase pressure drop correlations due to hydraulic diameter or other tube variations. This may lead to a more thorough investigation of heat exchanger design and manufacturing in order to understand the impact from tube variation due to brazing processes.
Bibliography


Appendix A

EES Code for Calculation of Test Section Properties

{A lookup table is used which contains the following measured quantities which are displayed in columns and order specific: liquid mass flow rate (m_dot_liq), vapor mass flow rate (m_dot_vap), vapor line pressure (P_vap), vapor line temperature (T_vap), liquid line pressure (P_liq), liquid line temperature (T_liq), pressure at test section inlet (P_TS), test section pressure drop (dP), temperature at test section outlet (T_TS), and atmospheric pressure [mbars]. Flow rates are input to the lookup table in kg/min. Temperatures and pressures are input in °C and psi, respectively.}

{The number of calculations (Nrow), number of microchannel ports (ports), and the refrigerant (refrig$) are input by the user prior to calculations.}

{Function to automatically specify the appropriate test section cross-sectional area}
Function AREA(ports)
    IF (ports=6) THEN
        AREA := 1.26867E-05
    ELSE
        AREA := 1.15094E-05
    ENDIF
END

{Variable input}
    ports = 14;
    refrig$ = 'R134a'
    Nrow = 1
    A_port = AREA(ports)

Duplicate i=1,Nrow

{Inlet Conditions}
{Liquid Line}
P_liq[i] = lookup(i,5)*6.8947573+lookup(i,10)*0.1
T_liq[i] = lookup(i,6)
m_dot_liq[i] = lookup(i,1)
h_liq[i] = enthalpy( refrig$, x=0, T=T_liq[i])
{Vapor Line}
P_vap[i] = lookup(i,3)*6.8947573+lookup(i,10)*0.1
T_vap[i] = lookup(i,4)
m_dot_vap[i] = lookup(i,2)
h_vap[i] = enthalpy( refrig$, P=P_vap[i], T=T_vap[i])
\{Mass Flux Calculation\}

\[
\text{MFlux[i]} = \frac{\text{m\_dot\_liq[i]} + \text{m\_dot\_vap[i]}}{(60 \ast \text{A\_port})}
\]

\{Conditions at Test Section\}

\{Temperature\}

\[
\text{T\_TS[i]} = \text{lookup}(i,9)
\]

\{Pressure Drop\}

\[
\begin{align*}
\text{P\_TS[i]} &= \text{lookup}(i,7) \ast 6.8947573 + \text{lookup}(i,10) \ast 0.1 \\
\text{dP[i]} &= \text{lookup}(i,8) \ast 6.8947573
\end{align*}
\]

\{Enthalpy\}

\[
\text{h\_mix[i]} = \frac{\text{h\_liq[i]} \ast \text{m\_dot\_liq[i]} + \text{h\_vap[i]} \ast \text{m\_dot\_vap[i]}}{(\text{m\_dot\_liq[i]} + \text{m\_dot\_vap[i]})}
\]

\{Quality\}

\[
\text{x[i]} = \text{quality( refrig$\$, P=\text{P\_TS[i]}, h=\text{h\_mix[i]})}
\]

\{Properties\}

\[
\begin{align*}
\text{rho\_liq[i]} &= \text{Density( refrig$\$, T=\text{T\_TS[i]}, x=0)}; \\
\text{rho\_vap[i]} &= \text{Density( refrig$\$, T=\text{T\_TS[i]}, x=1)}; \\
\text{mu\_liq[i]} &= \text{Viscosity( refrig$\$, T=\text{T\_TS[i]}, x=0)}; \\
\text{mu\_vap[i]} &= \text{Viscosity( refrig$\$, T=\text{T\_TS[i]}, x=1)};
\end{align*}
\]

\{Analysis of Conditions at equilibrium\}

\[
\text{P\_2[i]} = \text{P\_TS[i]} - \text{dP[i]}
\]

\[
\text{T\_sat[i]} = \text{Temperature( refrig$\$, P=\text{P\_2[i]}, x=0.5)};
\]

\[
\text{Error[i]} = \text{Abs(\text{T\_TS[i]} - \text{T\_sat[i])}
\]

\text{end}
Appendix B

Additional Two-Phase Pressure Drop Figures

This appendix presents additional plots from the microchannel two-phase pressure drop data. Testing was performed throughout the entire quality range for three mass fluxes and using two microchannel geometries with R134a and R410A.

Figure B.1  Pressure Drop versus Quality for 6-port, R134a at 10 °C
Figure B.2 Pressure Drop versus Quality for 6-port, R134a at 35 °C

Figure B.3 Pressure Drop versus Quality for 14-port, R134a at 10 °C
Figure B.4 Pressure Drop versus Quality for 14-port, R134a at 20 °C

Figure B.5 Pressure Drop versus Quality for 14-port, R134a at 35 °C
Figure B.6  Pressure Drop versus Quality for 14-port, R134a at $G = 170\ \text{kg/m}^2\text{s}$

Figure B.7  Pressure Drop versus Quality for 6-port, R410A at 10 $^\circ\text{C}$
Figure B.8  Pressure Drop versus Quality for 6-port, R410A at 20 °C

Figure B.9  Pressure Drop versus Quality for 14-port, R410A at 10 °C
Appendix C

Two-Phase Pressure Drop Comparison with Correlations

This appendix presents additional plots for the comparison of microchannel two-phase pressure drop data with existing correlations. Each plot contains data for both microchannel geometries with R134a and R410A refrigerants.

Figure C.1 Jung [1989] Prediction versus Pressure Drop using Optical $D_h$
Figure C.2  Souza [1993] Prediction Versus Pressure Drop using Optical $D_h$

Figure C.3  Zhang [1999] Prediction versus Pressure Drop using Optical $D_h$
Figure C.4  Jung [1989] Prediction versus Pressure Drop using Nominal $D_h$

Figure C.5  Souza [1993] Prediction Versus Pressure Drop using Nominal $D_h$
Figure C.6  Zhang [1999] Prediction versus Pressure Drop using Nominal D_h