Two-Phase Pressure Drop and Flow Regime of Refrigerants and Refrigerant-Oil Mixtures in Small Channels

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Abstract

As microchannel heat exchangers have become more sophisticated in their design, more exact understanding of the flow inside them is necessary. A decrease in diameter enhances the heat transfer (which takes place at the inner walls of the tubes), but also increases the pressure drop (as the diameter decreases, it becomes like drinking a milkshake through a coffee stirrer). The inclusion of even small amounts of oil in circulation can have a significant effect as well. Historical correlations and studies of two-phase flow have been shown to be insufficient for predicting pressure drops in the smaller channels, due to the different fluid physics that are relevant in flows of small diameter. This study is aimed at understanding the fluid property effects that contribute to pressure drop and flow regime. Two-phase pressure drop data for four refrigerants (R134a, R410A, R290 and R717) were measured in a channel with hydraulic diameter of 148 µm. These data were combined with previous two-phase data of R134a in small channels (hydraulic diameters ranging from 70 to 300 µm) to generate a separated flow model that spans a wide variety of fluid properties. Refrigerant was then mixed with two different viscosities of oil at concentrations ranging from 0.5 to 5% oil, and two-phase pressure drop measurements were taken of those mixtures. Flow visualizations of three of these refrigerants (R134a, R290 and R717) and several concentrations of a R134a-oil mixture were made in a channel with 500 µm hydraulic diameter, and flow regime classifications and comparisons with previous flow maps were made. Finally, a mechanistic description of the two-phase flow that occurs in small channels is put forth, based on the pressure drop measurements and the flow visualizations.
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List of Symbols

Roman symbols

$A$ Area

$A, B, a, b$ Correlation constants

$a_{1-9}$ Fluid propert correlation coefficients

$Bo$ Bond number

$C$ Frictional interaction parameter

$C_0$ Correlation constant in Zhao and Hu (2000) regime map

$C_f$ Laminar friction coefficient

$Ca$ Capillary number

$d$ Diameter

$d_h$ Hydraulic diameter

$Eo$ Eötvös number

$EP$ Energy product

$f$ Friction factor for pipe flow

$G$ Mass flux ($\frac{kg}{m^2s}$)

$h_f$ Frictional head loss in a pipe flow

$h_{fg}$ Enthalpy of formation

$KE$ Kinetic energy

$j$ Superficial velocity

$L$ Length

$m$ Mass flow rate ($\frac{kg}{s}$)

$OCR$ Oil circulation rate
Pressure
$P_{\text{red}}$ Reduced pressure
$\Delta P'$ Excess pressure drop in mechanistic model
$p, q, r, s$ Correlation constants
$\dot{Q}$ Volume flow rate
$\dot{q}$ Heat flux
$r$ Radius
$r_h$ Hydraulic radius
$Re$ Reynolds number
$S$ Slip ratio
$s$ Aspect ratio of rectangular channel
$T$ Temperature
$U_B$ Bubble velocity
$u(r)$ Velocity profile in mechanistic model
$u_i$ Interface velocity in mechanistic model
$u_o$ Centerline velocity in mechanistic model
$\overline{V}$ Average velocity
$V$ Volume
$V_{\text{char}}$ Characteristic velocity
$W_v$ Parameter in Zhao and Hu (2000) regime map
$We$ Weber number
$x$ Vapor quality
$x_{IA}$ Transitional vapor quality between intermittent and annular flow
$x_{psh}$ Pseudo-superheat vapor quality
$X$ Lockhart-Martinelli parameter

Greek symbols
$\alpha$ Void fraction
$\beta$ Volumetric quality
$\beta_{1,2,3}$ Correlation constants for separated flow model
δ Film thickness in mechanistic model
θ Contact angle
κ Correlation constant in Zhao and Hu (2000) regime map
λ Dimensionless group equivalent to $Re/Ca$
µ Fluid viscosity
ρ Fluid density
σ Surface tension
τ Wall shear
φ Two-phase multiplier in separated flow models
ψ Dimensionless group equivalent to $Ca$
ω Local oil concentration

Subscripts

avg Averaged quantity
ds Downstream of bubble in mechanistic model
t Liquid phase quantity (superficial)
to Liquid taken alone
meas Measured quantity
o,oily Oil quantity
pred Predicted quantity
ref Refrigerant
tot Total quantity
tt Turbulent-turbulent flow regime
us Upstream of bubble in mechanistic model
v Vapor phase quantity (superficial)
v0 Vapor taken alone
vt Laminar-turbulent flow regime
vv Laminar-laminar flow regime
Chapter 1

Introduction

With the increased popularity of microchannel heat exchangers for use in the air-conditioning and refrigeration industry, the use of two-phase refrigerant flows in small channels is of interest. While there have been many notable studies of two-phase flow in microchannels, and numerous studies of refrigerant flow in regular sized tubes, the combination of refrigerant with microchannels has been considered in only a small number of studies. Even fewer are studies that have examined the effects of oil on the two-phase flow of refrigerant.

It has been established that the physics that govern microchannel flow are no different than those in large tubes. For single-phase flows, this means that as long as the flow is properly characterized by dimensions and properties, the frictional losses are known. For two-phase flows, the relative influence of physical parameters of the flow can be sharply different, depending on the channel dimensions, and this will produce different effects in small channels. For example in a microchannel the wall wetting effect, which produces menisci in a large tube, will pull the fluid all the way up the wall, eliminating the stratified flow regime from existence.

The addition of oil into a refrigerant flow will modify the fluid properties. As long as the oil and refrigerant is a fully miscible pair, the oil will stay dissolved in the liquid refrigerant and the combination will remain homogeneous. Although the amount of oil in circulation is typically a small fraction of the refrigerant in circulation, when the refrigerant is mostly vapor there will always be a small quantity of liquid in circulation that is oil-rich. In conventional-sized tubes, this will probably not be significant, but inside a microchannel heat exchanger this could produce large contributions to the pressure drop. In addition, the dissolved mixture will affect the thermodynamic properties, so that there exist conditions where a temperature measurement will indicate that the flow should be superheated vapor and yet there will still be liquid refrigerant stuck in the oil. This “apparent superheat” will result in errors in system performance measurements, especially if only
temperatures are used to characterize the state points of the cycle.

The purpose of this study is to examine in detail the physics of two-phase flow in microchannels, by varying the fluid properties by use of different pure refrigerants and by the addition of varying concentrations of oil. Channel sizes below the typical industrial heat exchangers have been used. This magnifies the fluid property effects at work in small channels in order to better understand the fundamental effects at work in two-phase flows in small channels.

Pressure drop measurements are made in a channel of hydraulic diameter of 148 µm, of the pure refrigerants R134a, R410A, propane and ammonia, and also refrigerant-oil mixture of R134a with two viscosities of POE oil of varying oil concentration rates. In addition, flow visualizations are made in a glass channel of 0.5 mm diameter, of R134a, propane, ammonia and multiple concentration rates of R134a and one POE oil. From these, a mechanistic model for frictional pressure loss is developed which models the flow regimes directly.
Chapter 2

Background and Literature Review

2.1 Two-phase flow nomenclature

The nomenclature in two-phase flow is only marginally consistent throughout the vast literature. This first section of this chapter is devoted to laying out the various terminology and nomenclature that will be used in future sections.

2.1.1 Fluid properties

The fluid properties that are most relevant to two-phase flow are density, viscosity, and surface tension.

Density and viscosity

Density is denoted by $\rho$ with units of $\text{kg m}^{-3}$, viscosity by $\mu$ with units $\text{kg m s}^{-1}$. Since and liquid and vapor are both considered, the subscript $l$ will denote the liquid phase and the subscript $v$ will denote the gas phase. In this way, $\rho_l$ is the liquid phase density, and $\mu_v$ is the vapor phase viscosity. Kinematic viscosity, which is the ratio of viscosity to density, will not be used in this work.

Surface tension

The liquid-vapor surface tension, $\sigma$, has units $\text{N m}^{-1}$, or force per meter. This can be viewed as the contracting force per unit length around the perimeter. However, this is dimensionally equivalent to energy per unit area ($\text{J m}^{-2}$) and can be equivalently considered as a measurement of the cohesive energy present at an interface, or the excess free energy at the interface between two phases.

\[ \sigma = \left( \frac{\partial E}{\partial A} \right)_{T, V, n} \quad (2.1) \]
where $E$ is the Helmholtz free energy of the system, $A$ is the area of the interface, $T$ is temperature, $V$ is the volume, and $n$ is the number of molecules.

This surface energy can contribute as a difference in pressure when the interface is curved. This pressure difference across a curved interface is given by the Young-Laplace equation:

$$\Delta P = \sigma \left( \frac{1}{r_a} + \frac{1}{r_b} \right)$$  \hspace{1cm} (2.2)

where $r_a$ and $r_b$ are the radii of curvature in the two dimensions that describe the surface. For a spherical interface, $r_a = r_b = r$, and Equation 2.2 reduces to: $\Delta P = 2\sigma / r$.

There is another aspect of surface energy in two-phase flow: the contact angle, $\theta$, that the liquid-vapor interface makes with a solid. The contact angle is related to the free energies of the liquid-solid, liquid-vapor and vapor-solid combinations. This is only pertinent when there is a liquid-solid-vapor contact line, which is quite common in the nucleate boiling mode of heat transfer, but does not enter into the annular or slug flow regimes when there are no dry-out conditions.

### 2.1.2 Geometry of two-phase flow

The types of measurements that are available and the final application of the flow are what govern the framework of a two-phase flow study. In most refrigerant applications of two-phase flow, the overall refrigerant mass flow rate, $\dot{m}$ (with units of kg/s), is of primary interest. This can be expressed as a mass flux, $G$, which is the mass flow rate per unit area, with units of kg/m$^2$s:

$$G = \frac{\dot{m}}{A}$$  \hspace{1cm} (2.3)

The variable $A$ is the cross sectional area through which the flow is moving so mass flux is frequently used when considering flows in heat exchangers, with constant areas. Since in single-phase flow mass flow rate can be written as $\dot{m} = \rho \overline{V} A$, with $\overline{V}$ being an velocity averaged over the cross-section, mass flux is equivalent to $G = \rho \overline{V}$, and is sometimes called “mass velocity.” Since across an entire system, the mass flow rate, $\dot{m}$, is constant, mass flux, $G$, is only meaningful in sections of the system where the cross-sectional area is constant such as heat exchangers.

In circular channels, the characteristic length scale is diameter, $d$. Area is then calculated as
\[ A = \frac{\pi d^2}{4} \]. For channels that are not circular, the calculation of area depends upon the geometry of the cross-section, and the characteristic length scale typically used is the hydraulic diameter, \( d_h \):

\[ d_h = \frac{4A}{WP} \quad (2.4) \]

where \( WP \) is the wetted perimeter of the channel. A feature of hydraulic diameter is that a number of identical channels in parallel will have a hydraulic diameter equal to the hydraulic diameter of one of the individual channels.

**Mass, area and volume fractions**

When the flow of refrigerant is two-phase, the total mass flow rate, \( \dot{m} \), is divided into liquid, \( \dot{m}_l \), and vapor, \( \dot{m}_v \). These two must sum to the total mass flow rate:

\[ \dot{m} = \dot{m}_l + \dot{m}_v \quad (2.5) \]

The vapor mass fraction, \( x \), is defined as the ratio mass flow of vapor to the total mass flow rate of refrigerant:

\[ x = \frac{\dot{m}_v}{\dot{m}} \quad (2.6) \]

and \( x \) is usually taken to be equal to the thermodynamic quality of the flow. The value of \( (1 - x) \) is the liquid mass fraction:

\[ 1 - x = \frac{\dot{m}_l}{\dot{m}} \quad (2.7) \]

Mass fluxes of the individual phases can be noted as:

\[ G_v = G_x = \frac{\dot{m}_v}{A} \quad (2.8) \]

\[ G_l = G(1 - x) = \frac{\dot{m}_l}{A} \quad (2.9) \]

In the flow, the total cross-sectional area, \( A \), of the pipe is divided between liquid and vapor.
The void fraction, \( \alpha \), is the ratio of vapor area to total area, and thus, \( 1 - \alpha \) is the liquid area ratio:

\[
\alpha = \frac{A_v}{A} \quad \text{(2.10)}
\]

\[
1 - \alpha = \frac{A_l}{A} \quad \text{(2.11)}
\]

It can be seen that void fraction is defined locally at a specific cross-section of the channel. However, for flows that are not uniform along the channel (e.g. plug/slug flows) a volume fraction, \( \frac{V_v}{V} \), can be determined for a representative length of the channel that is equivalent to void fraction. Alternately, a time-average of the flow can be used to determine void fraction in cases when a limited section of the flow is visible instead of a certain length.

Another fraction called volumetric quality, \( \beta \), is the ratio of the volume flow rates. The volume flow rates, \( \dot{Q}_l \) and \( \dot{Q}_v \) are given by:

\[
\dot{Q}_l = \frac{\dot{m}_l}{\rho_l} \quad \text{(2.12)}
\]

\[
\dot{Q}_v = \frac{\dot{m}_v}{\rho_v} \quad \text{(2.13)}
\]

and the volumetric quality is the ratio of the volume flow rate of vapor to the total volume flow rate:

\[
\beta = \frac{\dot{Q}_v}{\dot{Q}_v + \dot{Q}_l} \quad \text{(2.14)}
\]

The volumetric quality is also called the homogeneous void fraction, because a homogeneous flow with a slip ratio of one will yield \( \alpha = \beta \) for void fraction.

**Velocities**

The superficial velocities are the velocity that each phase would have if the mass flow of that phase were flowing through the channel by itself. These will be denoted \( j_v \) and \( j_l \) for the vapor and liquid superficial velocity and are calculated as:

\[
j_v = \frac{Gx}{\rho_v} \quad \text{(2.15)}
\]

\[
j_l = \frac{G(1 - x)}{\rho_l} \quad \text{(2.16)}
\]
Substituting these superficial velocities into the Equation 2.14, the volumetric quality can be written as:

\[ \beta = \frac{j_v}{j_v + j_l} \]  

(2.17)

It is apparent that the actual velocities that would exist in the flow will always higher than the superficial velocities, unless the flow is single phase, since the area for each individual phase is reduced. The actual velocities for the two phases, the phase velocities, are denoted \( \nabla_v \) and \( \nabla_l \). These are an average across the cross-section for the phase, and they can be calculated if the void fraction is known:

\[ \nabla_v = \frac{j_v}{\alpha} = \frac{Gx}{\rho_v\alpha} \]  

(2.18)

\[ \nabla_l = \frac{j_l}{1-\alpha} = \frac{G(1-x)}{\rho_l(1-\alpha)} \]  

(2.19)

So for void fractions close to unity (which is common for flows in evaporators), \( j_v \) and \( \nabla_v \) are almost equal but \( j_l \) deviates strongly from \( \nabla_l \).

The slip ratio is the ratio of the phase velocities:

\[ S = \frac{\nabla_v}{\nabla_l} \]  

(2.20)

Performing a mass balance on the flow, with the mass flow rates given as:

\[ \dot{m}_v = \rho_v \nabla_v A_v = \rho_v \nabla_v \alpha A \]  

(2.21)

\[ \dot{m}_l = \rho_l \nabla_l A_l = \rho_l \nabla_l (1 - \alpha) A \]  

(2.22)

the slip ratio can be represented in the following manner:

\[ S = \left( \frac{1}{\alpha} - 1 \right) \frac{x}{1 - x} \frac{\rho_l}{\rho_v} \]  

(2.23)
Solving Equation 2.23 in terms of void fraction yields:

\[ \alpha = \frac{1}{1 + \left( \frac{V_v}{V_l} \right) \left( \frac{1-x}{x} \right) \left( \frac{\rho_v}{\rho_l} \right)} \]  

(2.24)

The homogeneous flow assumption is that the slip ratio between the two phases is unity: \( S = 1 \), i.e. the two phases are moving with the same velocity. The types of flow regimes for which this assumption is valid would include misty flow, where the liquid phase is being carried by the vapor phase in droplets, bubbly flow, where the vapor phase is being carried by the liquid phase in bubbles, and some forms of intermittent flow, when the slugs and plugs follow one another through the channel. However in slug/plug flows where there is a liquid film surrounding the vapor slug, the average liquid and vapor flow rates are not equal.

### 2.1.3 Dimensionless groups

#### Two-phase Reynolds numbers

The Reynolds number, which represents the dimensionless ratio of inertial forces to viscous forces, can take several forms in two-phase flow. Reynolds number is defined as the product of a fluid density, \( \rho \), a velocity scale, \( V \), and a length scale, \( L \), divided by a fluid viscosity, \( \mu \):

\[ Re = \frac{\rho V L}{\mu} \]

The various forms of the Reynolds number arise from the various velocity scales and fluid properties that appear in the two-phase flow. For channel flow, the length scale is typically taken to be the hydraulic diameter of the channel, \( d_h \). A Reynolds number based on the superficial liquid velocity, \( j_l \) is then denoted \( Re_l \):

\[ Re_l = \frac{\rho_l j_l d_h}{\mu_l} = \frac{G(1-x)d_h}{\mu_l} \]  

(2.25)

and the relevant fluid properties are taken as the saturated properties of the liquid, denoted by subscript \( l \). Equivalently, there is a superficial vapor Reynolds number, defined with characteristic velocity taken to be the superficial vapor velocity, \( j_v \), and the fluid properties taken from saturated
vapor properties:

\[
Re_v = \frac{\rho_v j_v d_h}{\mu_v} = \frac{G x d_h}{\mu_v} \tag{2.26}
\]

An alternate way to form a Reynolds number is to take the total mass flux and calculate a characteristic velocity which assumes that the entire flow is all liquid. This gives a liquid-only Reynolds number, \(Re_{lo}\):

\[
Re_{lo} = \frac{G d_h}{\mu_l} \tag{2.27}
\]

Parallel to this is the vapor-only Reynolds number, \(Re_{vo}\):

\[
Re_{vo} = \frac{G d_h}{\mu_v} \tag{2.28}
\]

Notice that this definition of Reynolds number is based on a completely non-physical characteristic velocity; the \(V_{char}\) that \(Re_{lo}\) is based on would be equal to \(\dot{m}/(\rho_l A)\). A refrigerant flow with a vapor quality of \(x = 0.2\), which corresponds in microchannel flow to a void fraction of perhaps \(\alpha = 0.8\), the characteristic velocity, \(V_{lo}\) would be roughly four times less that of the phase velocity, \(\overline{V}_l\). However, this characteristic velocity requires no knowledge of void fraction and is simple to calculate.

If void fraction is known, the phase velocities can be determined, and Reynolds numbers based on \(\overline{V}_l\) and \(\overline{V}_v\) can be formulated:

\[
Re_{\overline{V}_l} = \frac{\rho_l \overline{V}_l d_h}{\mu_l} = \frac{G (1 - x) d_h}{\mu_l (1 - \alpha)} = \frac{Re_l}{1 - \alpha} \tag{2.29}
\]

\[
Re_{\overline{V}_v} = \frac{\rho_v \overline{V}_v d_h}{\mu_v} = \frac{G x d_h}{\mu_v \alpha} = \frac{Re_v}{\alpha} \tag{2.30}
\]

The phase velocities produce the most meaningful Reynolds numbers that can be formulated, but they require a good correlation for – or direct knowledge of – the void fraction of the flow.

**Two-phase Weber numbers**

The Weber number is the ratio of inertial forces in a flow to surface tension forces:

\[
We = \frac{\rho V^2 L}{\sigma} \tag{2.31}
\]
Again, in channel flow, the characteristic length is typically taken as the hydraulic diameter, \( d_h \). Superficial Weber numbers, \( We_l \) and \( We_v \) are defined with the superficial velocities:

\[
We_l = \frac{\rho_l j_l^2 d_h}{\sigma} = \frac{G^2 (1 - x)^2 d_h}{\sigma \rho_l}
\]

(2.32)

\[
We_v = \frac{\rho_v j_v^2 d_h}{\sigma} = \frac{G^2 x^2 d_h}{\sigma \rho_v}
\]

(2.33)

Weber numbers based on the total mass flux taken to be a specific phase, \( We_{lo} \) and \( We_{vo} \), are defined in like manner to Reynolds numbers:

\[
We_{lo} = \frac{G^2 d_h}{\sigma \rho_l}
\]

(2.34)

\[
We_{vo} = \frac{G^2 d_h}{\sigma \rho_v}
\]

(2.35)

Analogous to the Reynolds number, the Weber number can be defined with a phase velocity as the characteristic velocity, but only if the void fraction is known:

\[
We_{V_i} = \frac{\rho_l V_i^2 d_h}{\sigma} = \frac{G^2 (1 - x)^2 d_h}{\sigma \rho_l (1 - \alpha)^2} = \frac{We_l}{1 - \alpha}
\]

(2.36)

\[
We_{V_v} = \frac{\rho_v V_v^2 d_h}{\sigma} = \frac{G^2 x^2 d_h}{\sigma \rho_v \alpha} = \frac{We_v}{\alpha}
\]

(2.37)

**Capillary flow: Suo and Griffith (1964)**

A dimensional analysis of intermittent flow was performed by Suo and Griffith (1964). They found that velocities in two-phase capillary slug flow can be described by seven dimensionless parameters, four of which were ratios of: phase densities, phase viscosities, slug length to radius of the tube and bubble length to radius of the tube. These were disregarded as being not relevant to the velocities
of the phases. The other three dimensionless groups were represented as:

\[
\lambda = \frac{\mu l^2}{\rho_l \sigma d_h} \quad \text{(2.38)}
\]

\[
\psi = \frac{\mu l V}{\sigma} \quad \text{(2.39)}
\]

\[
\Omega = \frac{\rho_l g d_h^2}{\sigma} \quad \text{(2.40)}
\]

The group \( \lambda \) in Equation 2.38 is constant for a given fluid and channel diameter, being only a function of physical and geometrical parameters.

The group \( \psi \) in Equation 2.39 represents the ratio of the viscous to surface tension effects, given in terms of a characteristic velocity, \( V \). The characteristic velocity used by Suo and Griffith was \( U_B \), the bubble velocity, which they calculated from photographic images. They related the slug velocity, \( U_S \), to the bubble velocity by means of volumetric flow rates, so it was the only velocity scale in their formulation. Here, the characteristic velocity used in \( \psi \) is simply given as \( V \), because different characteristic velocities can be chosen by different people.

The group \( \psi \) in a single-phase flow would be known as the Capillary number, since the characteristic velocity will then be just the average velocity, \( V \). The Capillary number, \( Ca \), of a single-phase flow is found as the ratio:

\[
Ca = \frac{\mu V}{\sigma} = \frac{We}{Re} \quad \text{(2.41)}
\]

For a two-phase flow, there are multiple velocity scales that could be defined, meaning that there can be multiple Capillary numbers that could be defined, depending on which velocity scales used. If the Weber and Reynolds numbers were calculated on different velocity scales, even more Capillary numbers could be computed.

The group \( \Omega \) in Equation 2.40 represents the ratio of gravity to surface tension effects. This group is almost identical to the Bond number, which is the ratio of the buoyancy forces to the surface tension forces and is defined as:

\[
Bo = \frac{g(\rho_l - \rho_v) d^2}{\sigma} \quad \text{(2.42)}
\]

The relative importance of surface tension and gravity effects has been cast into various param-
eters throughout the literature. The Eötvös number is analogous to the Bond number, but without the constant of gravity (which gives $Eo$ units of $s^2/m$): $Eo = \frac{(\rho_l - \rho_v) d^2}{\sigma}$. The Confinement number, $Co = 1/\sqrt{Bo}$, is another parameter with an analogous meaning as the Bond number. The Laplace constant, which has value of $\sqrt{\sigma/(g(\rho_l - \rho_v))}$ and is then compared to the diameter of the channel, also serves the same purpose. All five of these parameters represent the relative importance of buoyant (gravitational) forces to surface tension forces. When considering flows in small channels, or capillary slug flows (as was the original analysis by Suo and Griffith), the effects of gravity are negligible compared to surface tension, and thus $\Omega$, $Bo$, $Eo$, $Co$ and the Laplace constant can all be neglected.

The two groups $\lambda$ and $\psi$ can be combined to form liquid Reynolds and Weber numbers. For example, if the superficial liquid velocity, $j_l$, is used in $\psi$:

$$Re_l = \frac{\rho_l j_l d_h}{\mu_l} = \frac{\psi}{\lambda}$$

$$We_l = \frac{\rho_l j_l^2 d_h}{\sigma} = \frac{\psi^2}{\lambda}$$

### 2.2 Pressure drop

#### 2.2.1 Single-phase pressure drop

Pressure drop in single-phase flow is a well-studied phenomenon. As described by White (2008, Chapter 6), the frictional head loss in a pipe, $h_f$, is correlated by the Darcy friction factor, $f$, which is a function of the Reynolds number of the pipe flow, the roughness of the inside of the pipe relative to the internal diameter, $\frac{\varepsilon}{d}$, and the shape of the pipe. The correlation, first proposed by Weisbach in 1850, is:

$$h_f = f \frac{L V^2}{d_h 2g}$$

(2.43)

For laminar flow, the friction factor is only dependent on $Re$, in the form:

$$f = \frac{C_f}{Re}$$

(2.44)

where the parameter $C_f$ a constant, which depends only on the channel geometry. For circular
geometry, $C_f$ is equal to 64, but many geometries can produce analytical solutions for that constant; Shah and London (1978) have a correlation for rectangular geometries as a function of aspect ratio, $s$:

$$C_f = 96(1 - 1.3553s + 1.9467s^2 - 1.7012s^3 + 0.9564s^4 - 0.2537s^5)$$  \hfill (2.45)

Churchill (1977) formulated an explicit equation that spans the laminar, transitional and turbulent regimes. For the turbulent regime he combined Colebrook’s smooth pipe equation from with the rough pipe equation of Nikuradse. For the transitional regime he used the intermittancy of turbulence method described by Wilson and Azad (1975) to generate values for friction factors in the transitional regime and correlated them himself into the following equation:

$$f_{\text{trans}} = 56.8 \times 10^{-10} Re^2$$  \hfill (2.46)

and for the laminar regime, he used Equation 2.44. Combining these equations with the method described by Churchill and Usagi (1972), the final equation for friction factor is given by:

$$f = 8 \left[ \left( \frac{8}{Re} \right)^{12} + \frac{1}{(A^{16} + B^{16})^{3/2}} \right]^{1/12}$$  \hfill (2.47)

where:

\begin{align*}
A &= -2.457 \ln \left[ \left( \frac{7}{Re} \right)^{0.9} + 0.27 \frac{f}{d_h} \right] \\
B &= \frac{37530}{Re}
\end{align*}

This correlation has been used in this work to represent the frictional losses in single-phase flow.

### 2.2.2 Two-phase pressure drop

Two-phase pressure drop models have traditionally been divided into two categories: homogeneous flow models and separated flow models. The homogeneous flow model assumes that the liquid and vapor phases are flowing at the same velocity; in particular that they can be treated as a single
fluid with averaged properties for density, viscosity, etc. As such, they are typically valid for flow regimes such as mist flow or bubbly flow where one phase is evenly dispersed into the other and this concept of averaged properties is valid. The separated flow model on the other hand, allows the two phases to be distinct and models an interaction between them. Both the homogeneous and separated flow models have been developed through long histories of two-phase flow studies. There have been some recent attempts to model two-phase pressure drop that could be termed “mechanistic models”, because they attempt to model portions of the fluid mechanics of the two phase flow. This results in a comprehensive look at the flow that, for certain regimes, can yield a better model.

**Homogeneous models**

To use a homogeneous model, the two-phase mixture is treated as a single-phase flow. Then a two-phase Reynolds number, $Re_{avg}$, is calculated based on average fluid properties:

$$Re_{avg} = \frac{Gd_h}{\mu_{avg}}$$ (2.48)

and the friction factor based on that Reynolds number is calculated from the single-phase equations, such as Equation 2.47, which in turn gives you the pressure drop of the flow.

The key to a homogeneous two-phase model is the choice of average properties. With the liquid and vapor are flowing together in the channel, the density of the mixture in any section of channel will be governed by a mass average. This density is called $\rho_{avg}$, and the mass average is given in the following manner:

$$\frac{1}{\rho_{avg}} = \frac{x}{\rho_v} + \frac{1-x}{\rho_l}$$ (2.49)

The choice of average viscosity, $\mu_{avg}$ is a little less transparent. Bittle and Weis (2002) describe the concept of average viscosity as an ideal-damping element in a dynamic system. This method considers the viscous dissipation of the shearing force moving the flow along the channel, and the arrangement of the phases (parallel or series) in relation to this dissipation. This is illustrated in Figures 2.1 and 2.2. In Figure 2.1, the phases act between the shearing force moving to the right and the wall. In Figure 2.2, the arrangement of parallel and series dampers are shown in an
additive combination and reciprocal additive combination. In the case of annular flow, for example, the viscosities of each phase is in a series configuration if you consider the shearing force between the flow and the wall. However, in the slug/plug flow flow regime, the viscosities of the two phases are in parallel damping configuration between the flow and the wall. The phase viscosities can be weighted by mass (as is always the case with the phase densities above) or by volume. This conceptual framework – parallel vs. series and mass vs. volume – will be used to understand the following viscosity models that have been proposed.

![Figure 2.1: Schematic of shearing force acting on a two-phase mixture in series and parallel](image)

![Figure 2.2: Schematic of dampers acting in series and parallel](image)

One average viscosity model that has been traditionally attributed to McAdams et al. (1942),
calculates the average viscosity in the following manner:

\[
\frac{1}{\mu_{\text{avg}}} = \frac{x}{\mu_v} + \frac{1 - x}{\mu_l}
\]

(2.50)

This is a series combination of the two phase viscosities, weighted by mass.

Dukler et al. (1964) developed a model that is also a series combination of the phase viscosities, but weighting their contributions by volume:

\[
\mu_{\text{avg}} = \alpha \mu_v + (1 - \alpha) \mu_l
\]

(2.51)

With the homogeneous flow assumption that the slip velocity ratio is 1, i.e. the two phases are moving at the same velocity, we can make the algebraic substitution for \(\alpha\) from Equation 2.24, and Dukler’s homogeneous viscosity is more commonly seen in the following form:

\[
\mu_{\text{avg}} = \rho_{\text{avg}} \left( \frac{x}{\rho_v} \mu_v + \frac{1 - x}{\rho_l} \mu_l \right)
\]

(2.52)

where \(\rho_{\text{avg}}\) is given in Equation 2.49.

Cicchitti et al. (1960) formulated the mixture viscosity based on a parallel combination with mass weighting:

\[
\mu_{\text{avg}} = x \mu_v + (1 - x) \mu_l
\]

(2.53)

Following this framework, Bittle and Weis put forth a new model that they called Model IV by combining the viscosities in series and weighting them by volume fraction:

\[
\frac{1}{\mu_{\text{avg}}} = \frac{\alpha}{\mu_v} + \frac{1 - \alpha}{\mu_l}
\]

(2.54)

Other homogeneous models have been proposed that deviate from this ideal-damper framework. Beattie and Whalley (1982) combined the Dukler model with a viscosity model based on the
theoretical work of Einstein to produce a hybrid model of the following form:

\[ \mu_{\text{avg}} = \alpha \mu_v + (1 - \alpha)(1 + 2.5\alpha) \mu_l \] (2.55)

Empirical constants have been included in this model. This formulation has the unfortunate property of yielding average viscosities that are larger than the pure liquid viscosity at low \( \alpha \).

Another homogeneous model by Lin et al. (1991) starts from the McAdams model, and is customized by an empirical exponent on the quality to their own R-12 experimental data. Since this is a specialized model with little relevance to the current work, it is included here only to demonstrate the variety of homogeneous models that could be developed:

\[ \mu_{\text{avg}} = \frac{\mu_v \mu_l}{\mu_v + x^{1.4}(\mu_l - \mu_g)} \] (2.56)

However, from the damper analogue of viscosity, it is believed that the first four models are representative of the generic types of average viscosity that can be developed for use in a homogeneous model. The use of empirical constants from curve fitted data has been done in many cases to correct for the non-homogeneity of the real flows.

**Separated flow models**

With their restricting assumptions, the homogeneous models fall short of applying to the entire range of two-phase flow regimes. Separated flow models have been developed that allow the two phases to move at different velocities, have different properties, and interact with one another. The basic formulation of a separated flow model is that of a two-phase multiplier, usually designated \( \phi^2 \). A single-phase flow pressure gradient based on some fluid properties is calculated from the flow properties and is multiplied by the corresponding two-phase multiplier to yield the two-phase pressure gradient. For example, \( \left( \frac{\Delta P}{\Delta L} \right)_l \) would be the single-phase pressure gradient based on the liquid flow rate in the tube. Then the pressure gradient of the two-phase mixture, \( \left( \frac{\Delta P}{\Delta L} \right)_{tp} \), is found by:

\[ \left( \frac{\Delta P}{\Delta L} \right)_{tp} = \phi^2 \left( \frac{\Delta P}{\Delta L} \right)_l \] (2.57)

This can be done for the vapor phase as well, using the multiplier \( \phi^2_v \) and calculating the
pressure gradient \( \left( \frac{\Delta P}{\Delta L} \right)_v \). It can also been done by assuming that the entire flow is liquid, and calculating \( \left( \frac{\Delta P}{\Delta L} \right)_{lo} \) based on \( Re_{lo} \) (see Equation 2.27), and the two-phase multiplier to use is called \( \phi_{lo}^2 \). Likewise, if you calculate the pressure gradient by assuming that the entire flow is vapor, you use \( Re_{vo} \), and the two-phase multiplier is \( \phi_{vo}^2 \).

Friedel (1980) developed a popular correlation using the multiplier \( \phi_{lo}^2 \).

\[
\phi_{lo}^2 = C_1 + \frac{3.24 C_2}{Fr_{avg}^{0.045} We_{avg}^{0.035}}
\]  
(2.58)

where:

\[
C_1 = (1 - x)^2 + x^2 \frac{\rho_l}{\rho_v} \frac{f_{vo}}{f_{lo}}
\]

\[
C_2 = x^{0.78} (1 - x)^{0.24} \left( \frac{\rho_l}{\rho_v} \right)^{0.91} \left( \frac{\mu_v}{\mu_l} \right)^{0.19} \left( 1 - \frac{\mu_v}{\mu_l} \right)^{0.7}
\]

\[
Fr_{avg} = \frac{G^2}{gd_h \rho_{avg}^2}
\]

\[
We_{avg} = \frac{G^2 d_h}{\sigma \rho_{avg}}
\]

and \( g \) is gravitational acceleration, and \( \rho_{avg} \) is the homogeneous density, given in Equation 2.49. This correlation has been used with good agreement in conventional sized channels, and even down to smaller (1 mm) channels with CO\(_2\). Is known not to work well when the viscosity ratio of the phases \( (\mu_l/\mu_v) \) exceeds 1000.

Laying the historic groundwork for the separated flow approach, Lockhart and Martinelli (1949) developed a parameter, \( X \), that is the ratio of the the liquid to vapor pressure gradients:

\[
X^2 = \frac{\left( \frac{\Delta P}{\Delta L} \right)_l}{\left( \frac{\Delta P}{\Delta L} \right)_v}
\]  
(2.59)

This is known as the Lockhart-Martinelli parameter, and from multiple plots of data from various sources, they demonstrated that it could be used to correlate pressure drop and void fraction of flows. Lockhart and Martinelli developed correlations for this parameter by differentiating between the laminar ("viscous") and turbulent flow regimes of the different phases. To determine this, they looked at the superficial phase Reynolds numbers, \( Re_l \) and \( Re_v \). When \( Re \) was greater than
Table 2.1: Chisholm’s values for $C$

<table>
<thead>
<tr>
<th>liquid</th>
<th>vapor</th>
<th>$C$</th>
</tr>
</thead>
<tbody>
<tr>
<td>turbulent</td>
<td>turbulent</td>
<td>20</td>
</tr>
<tr>
<td>viscous</td>
<td>turbulent</td>
<td>12</td>
</tr>
<tr>
<td>turbulent</td>
<td>viscous</td>
<td>10</td>
</tr>
<tr>
<td>viscous</td>
<td>viscous</td>
<td>5</td>
</tr>
</tbody>
</table>

2000, that phase was considered turbulent, and then $Re$ was below 1000 that phase was considered laminar.

Later, Chisholm (1967) developed a theoretical basis for why $X$ would correlate pressure drop and void fraction (holdup), and recommended a simplified equation of:

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$$ \hspace{1cm} (2.60)

Chisholm’s values for $C$, based on the flow regime separations of Lockhart-Martinelli, for engineering purposes, found in Table 2.1.

This formulation can be combined with Equations 2.57 and 2.59 to elucidate the parameter $C$:

$$\left(\frac{\Delta P}{\Delta L}\right)_{tp} = \left(\frac{\Delta P}{\Delta L}\right)_l + C \sqrt{\left(\frac{\Delta P}{\Delta L}\right)_l \left(\frac{\Delta P}{\Delta L}\right)_v} + \left(\frac{\Delta P}{\Delta L}\right)_v$$ \hspace{1cm} (2.61)

Seen in this way, $C$ can be considered as an interaction parameter between the liquid and vapor phases. Some of the studies that have been directed at determining $C$ based on flow properties will be discussed below.

Following Lockhart and Martinelli, Chisholm defined the lines of demarcation between laminar and turbulent for each phase were defined based on the values of $Re_l$ and $Re_v$: “viscous” flow was $Re < 1000$ and turbulent flow was $Re > 2000$. These values for $C$ have historically had good agreement in conventional sized tubes, but in microchannels and capillary tubes they over-predict measured results.

Efforts at correlating $C$ for smaller tubes include Mishima and Hibiki (1996), who tested air-water flow in 1 to 4 mm diameter tubes and came up with an empirical fit for $C$ that is a function
of diameter:

\[ C = 21(1 - e^{-0.319d_h}) \]  

(2.62)

where \( d_h \) is measured in millimeters. Using Equation 2.62, however, neglects all the properties of the flow except the diameter of the tubes, by making \( C \) only a function of \( d_h \). Mishima and Hibiki considered three different data sets, including different geometry channels and one data set involving ammonia-vapor flow. This study was one of the earliest attempts to develop a separated flow model for microchannels, and the only one at the time to explicitly take diameter into account. Unfortunately, from the work performed here of testing different fluids in the same channel, it was easily observed that \( C \) does not depend on only the tube diameter.

English and Kandlikar (2005) measured two-phase pressure drop in an almost-square channel, with a reported hydraulic diameter of 1.018 mm, and fluids representing a wide variation in surface tension values. They correlated the pressure drop measurements by modifying Mishima and Hibiki’s correlation for \( C \) to be:

\[ C = 5(1 - e^{-0.319d_h}) \]  

(2.63)

which essentially neglected to take into account the surface tension variation in their experimental fluids, at least in terms of a correlation for \( C \).

Another effort at correlating \( C \) for smaller channels was done by Lee and Lee (2001). Their channels were rectangular, with \( d_h \) varying from 0.78 to 6.7 mm, and their working fluids were air and water. They followed the dimensional analysis of Suo and Griffith (1964) and considered the dimensionless parameters, \( \lambda \), \( \psi \) and \( \Omega \), shown above in Equations 2.38, 2.39, and 2.40. Following Suo and Griffith, they neglected \( \Omega \) (the ratio between gravity and surface tension forces), but departing from the Suo and Griffith analysis, they used \( V = j_l \) in place of \( V = U_B \) for the characteristic velocity in formulating \( \psi \) (Equation 2.39). Then they argued by reference to Wambsganss et al. (1992) that a correlation for \( C \) should take the mass velocity into account by means of the parameter \( Re_{lo} \). They correlated \( C \) in the following manner:

\[ C = A\lambda^q\psi^r Re_{lo}^s \]  

(2.64)

The constants \( A, q, r, \) and \( s \) were determined by regressing the measured pressure drop data for
Table 2.2: Lee and Lee’s coefficients for $C$ from Equation 2.64

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Vapor</th>
<th>$A$</th>
<th>$q$</th>
<th>$r$</th>
<th>$s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>laminar</td>
<td>laminar</td>
<td>$6.83 \times 10^{-8}$</td>
<td>-1.317</td>
<td>0.719</td>
<td>0.557</td>
</tr>
<tr>
<td>laminar</td>
<td>turbulent</td>
<td>$6.185 \times 10^{-2}$</td>
<td>0</td>
<td>0</td>
<td>0.726</td>
</tr>
<tr>
<td>turbulent</td>
<td>laminar</td>
<td>3.627</td>
<td>0</td>
<td>0</td>
<td>0.174</td>
</tr>
<tr>
<td>turbulent</td>
<td>turbulent</td>
<td>0.408</td>
<td>0</td>
<td>0</td>
<td>0.451</td>
</tr>
</tbody>
</table>

each of the four flow regimes corresponding to Chisholm’s divisions of laminar-laminar, laminar-turbulent, turbulent-laminar, and turbulent-turbulent (where the term “viscous” has fallen out of use in recent years and has been replaced by “laminar”). These values are seen in Table 2.2.

The presence of zeros in the exponents for $q$ and $r$ in every regime except laminar-laminar flow means that the parameters $\lambda$ and $\psi$ were not found to be contributing to the flow except in this one flow regime. However, the parameter $Re_{lo}$ (defined in Equation 2.27) assumes a velocity based on an entirely liquid flow, and is therefore constant with varying quality. Thus $C$ calculated from these parameters will be constant with increasing quality once either phase is in the turbulent flow regime. This has not prevented good agreement between their model and several published experiments, including flows of refrigerants, across multiple regimes of flow.

Xiao Tu (2004) used Lee and Lee’s parameters for the separated flow model in his work. He tested adiabatic R134a two-phase flows in five rectangular channels of varying $d_h$: 304.7, 150.0, 141.0, 104.1, and 69.5 $\mu$m. The aspect ratios of these channels varied from 0.09 to 0.24. After comparing many different models for two-phase pressure drop to his experimental data, Tu found that Dukler’s homogeneous model and Lee and Lee’s separated model had the best agreement with the measured data. Tu used Lee and Lee’s formulation of $C$ (Equation 2.64) to develop a two-phase pressure drop model of his own, except instead of $j_l$ as the characteristic velocity of the parameter $\psi$, Tu used $U_B$ (i.e. $\psi = \frac{\mu U_B}{\sigma}$). To determine $U_B$ in the absence of actual bubble velocity measurements, Tu assumed a homogeneous model of void fraction, the Armand expression, to yield:

$$U_B = 1.2(j_l + j_v)$$  \hspace{1cm} (2.65)

The Armand expression for void fraction is explained by Chung and Kawaji (2004).

Rather than assume the superficial Reynolds number was a valid way to separate flow regimes,
Table 2.3: Tu’s coefficients for $C$ from Equation 2.64

<table>
<thead>
<tr>
<th>Dominant Regime</th>
<th>Condition</th>
<th>$A$</th>
<th>$q$</th>
<th>$r$</th>
<th>$s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surf. Tens.</td>
<td>$W_e_v \leq 11.0 \sqrt{W_e_l}$</td>
<td>$1.279 \times 10^{-9}$</td>
<td>-1.9558</td>
<td>0.9971</td>
<td>0.3997</td>
</tr>
<tr>
<td>Inertial</td>
<td>$W_e_v &gt; 11.0 \sqrt{W_e_l}$</td>
<td>$1.386 \times 10^{-4}$</td>
<td>-0.654</td>
<td>0.203</td>
<td>0.519</td>
</tr>
</tbody>
</table>

Tu regressed his data to determine the coefficients $A$, $q$, $r$, and $s$ based on flow regime divisions proposed in the flow map developed by Akbar et al. (2003). This flow map, discussed below in Section 2.3.3, uses the superficial Weber numbers of the two phases as axes to separate the flow regimes. The relevant section of the map for microchannel flow is the division between surface tension dominated and inertial dominated flows, and according to the findings of Akbar that transition line is given by $W_e_v = 11.0 \sqrt{W_e_l}$. The regime $W_e_v > 11.0 \sqrt{W_e_l}$ is dominated by inertial forces yielding annular flow, and when $W_e_v \leq 11.0 \sqrt{W_e_l}$ the flow is dominated by surface tension forces, yielding slug/plug or bubble flow. The coefficients that Tu determined from his data are shown in Table 2.3. Since none of the coefficients vanish for either regime, the parameters are all equally considered over the whole scope of the flow regimes.

Lee and Mudawar (2005) experimented with pressure drop across microchannel evaporators of $d_h = 350 \mu m$. They developed a correlation for $C$ based on the parameters, $Re_{lo}$ and $W_e_{lo}$ for two different flow regimes corresponding to laminar or turbulent vapor flow (the liquid phase was always laminar in their study):

$$C_{vv} = 2.16 Re_{lo}^{0.047} W_e_{lo}^{0.60}$$  \hspace{1cm} (2.66)
$$C_{vt} = 1.45 Re_{lo}^{0.25} W_e_{lo}^{0.23}$$  \hspace{1cm} (2.67)

However, since $Re_{lo}$ and $W_e_{lo}$ both are calculated with a characteristic velocity equivalent to a flow completely in the liquid state, neither $Re_{lo}$ nor $W_e_{lo}$ depend for example on the quality, $x$.

Niño (2002) correlated two-phase pressure gradient of refrigerants in multi-port channels of 1.02 and 1.54 mm diameter by dividing the flow into intermittent and annular regimes and applying different techniques to the different regimes. In the intermittent flow region, he correlated the two-phase pressure gradient by the average kinetic energy of the flow. For the annular flow region, he developed a new separated flow model that was based on the vapor-only multiplier, $\phi_v^2$. 

22
Niño’s average kinetic energy model for pressure gradient of intermittent flow – which was similar in form to the corrections used to account for two-phase entrance and exit effects of the microchannels – compares the pressure gradient per unit length to the average kinetic energy in the flow per hydraulic diameter of the channels. This collapsed the data from the fluids tested (air-water, R134a and R410A) in both the 6-port and 14-port microchannels onto a single line. The average kinetic energy is calculated as:

\[ KE_{avg} = \frac{\rho_{avg} G^2}{2} \]  

(2.68)

where \( \rho_{avg} \) is the same as that in Equation 2.49. It should be noted that this is a homogeneous model of pressure gradient. A representative plot of Niño’s data, showing the linear collapse of intermittent flow pressure gradient data for his three fluids and two different sized channels is seen in Figure 2.3.

![Figure 2.3: Two-phase pressure gradient vs. Average Kinetic Energy divided by hydraulic diameter, from Niño (2002)](image_url)

For the annular flow regime, Niño took the separated flow model approach and correlated the
vapor-only two-phase multiplier, \( \phi^2_{vo} \), to the combined parameter \( (X_{tt} + \frac{1}{We}) \left( \frac{\rho_l}{\rho_v} \right)^q \), where \( p = 1.3 \) and \( q = 0.9 \) were selected for curve fitting. The parameter \( X_{tt} \) is the Lockhart-Martinelli parameter from Equation 2.59, for the turbulent-turbulent flow regimes. This allowed for correlation across a wide range of fluids for his work and the subsequent work of Adams (2003), although it was found to be slightly less successful at correlating the data for large pressure gradients in subsequent work. It should be mentioned that the parameter selected has inertial effects combined with the surface tension effects, and although the precise reason for the success of the collapse of the data is not clear, there is some physical insight gained by the combination of these dimensionless groups in this manner.

**Mechanistic pressure drop models**

The term “mechanistic model” will be used here to indicate any sort of pressure drop that is not homogeneous – meaning that it does not average the fluid properties in any way – but is also not a separated flow model – meaning that it is not based on the interaction parameter, \( C \). Separated flow models, which enjoy a lot of success for conventional tubes, fall short of representing the exact fluid physics within the tubes. To take an example, in annular flow the length scale used in the calculation of \( Re_l \) is the hydraulic diameter. However, a more correct physical length scale would be thickness of the liquid film. Mechanistic models attempt to predict pressure drop by more directly considering the fluid mechanics at work.

One mechanistic model was developed by Garimella et al. (2002). They considered intermittent flow of condensing refrigerant in parallel microchannels with hydraulic diameters down to 0.5 mm, and assume a unit cell model for the flow which has two regions: the slug region that is all liquid and the bubble/film region that has a film of liquid surrounding a bubble. A sketch of the geometry they assumed is shown in Figure 2.4. A homogeneous void fraction model was applied by asserting that the bubble velocity is 1.2 times the average liquid velocity, (which is consistent with the formulation of the liquid in the film moving more slowly than the liquid in the slug. The total pressure drop was calculated by assuming a power law turbulent velocity profile in the slug and a laminar velocity profile for the bubble/film region. Also considered were losses between the film and slug, where the slow moving film liquid is caught up in the liquid slug at the end of each
bubble. These losses were reported to be a significant portion of the overall losses, and were used to close the model, by correlating how many of these bubble(slug) cells were located per unit length of the channel. It was not discussed how a transfer of momentum within the control volume can contribute to overall pressure loss, because the actual effect is that the disturbance of the flow resulting from this transition between the film and slug lead to higher wall shear in these regions. The total pressure drop was calculated by the sum of the pressure drop from the two regions.

Figure 2.4: Unit cell geometry assumed by Garimella et al. (2002)

Chung and Kawaji (2004) developed a mechanistic model for pressure drop based on their observations of air-water flow in tubes of diameter 50 and 100 µm. Their flow visualizations revealed that the only flow patterns that existed in the small channels were slug flow and annular flow (with other flow regimes that they named but which are essentially either one of these two, such as “gas core with wavy liquid film”, and “gas core with thin-smooth liquid film”). Their model was developed by considering the same unit cell as Garimella et al.: a slug region and a bubble region made up of a bubble and film that uniformly surrounded the bubble. Dry-out conditions were not considered, neither were they observed in their flow visualizations which for adiabatic air-water flow is expected. The total pressure drop was computed by the sum of the pressure drops in each region. The pressure drop in the slug region was computed from the friction factor, calculated on the phase Reynolds number of the slug, i.e. the characteristic velocity was the liquid phase velocity, $V_l$, which was calculated by dividing the superficial velocity, $j_l$, by the liquid fraction, $(1 - \alpha)$, which was measured in their visualization experiments. The void fraction was used to
determine the relative lengths of the regions within the unit cell, and with this information they successfully predicted the measured pressure drop. Interesting to note from their study was that larger 250 and 530 µm channels behaved homogeneously in terms of void fraction whereas the 50 and 100 µm channels departed from homogeneous void fraction behavior. But the 50 and 100 µm smaller channels behaved homogeneously in terms of matching the Dukler pressure drop model, where the larger channels did not.

Figure 2.5: Geometry of unit cell assumed by the model of Chung and Kawaji (2004)

Jacobi (2005) provides a discussion of an elongated bubble model for microchannel heat transfer, which had been developed and subsequently refined in a series of articles: Jacobi and Thome (2002), Thome et al. (2004) and Dupont et al. (2004). This model, a schematic of which is shown in Figure 2.6, also includes the microchannel flow of slugs of liquid and annular rings of liquid surrounding bubbles, but it makes the additional allowance for dry-out with a third region that is a vapor slug. Jacobi (2005) points out that a weakness of Garimella’s model is that the assumed turbulent velocity profiles are not realistic for microchannel flow. While the elongated bubble flow shows the most promise in terms of predicting pressure drops, he concludes that fluid properties such as surface tension need to be properly incorporated into such models before they will have general applicability.

The geometry of the elongated bubble is quite similar to the flow pattern known as a Taylor Bubble. A common theme among the two-phase flow literature, the Taylor bubble is a portion of vapor moving within a tube filled with viscous liquid. The outer constraining walls of the tube
Figure 2.6: Unit cell for the Jacobi and Thome elongated bubble model for evaporating heat transfer

are important, as some of the earliest questions investigated were related to the effect the shape of the bubble had on the measured pressure drop as it passed (Barnea, 1990), the thickness of the film between the vapor and wall and the flow patterns in the liquid in front of the bubble (Taylor, 1961), or the shape of the interface and the speed of the bubble relative to the fluid (Bretherton, 1961). Even to the present day Taylor bubbles remain to be of interest, and have been the subject of high speed video and PIV measurements (Polonsky et al., 1999; Thulasidas et al., 1997), scaling techniques with various viscosity fluids (Aussillous and Quéré, 2000), and CFD analysis (Kreutzer et al., 2005). In particular interest to the slug flow regime of microchannel refrigerant is the pressure drop of the passing bubble, and the velocity profiles of the liquid, at least as compared to the bubble. One significant difference between classical Taylor bubble and the slug flow observed here is that the classical Taylor bubbles are typically vertically moving bubbles.

2.3 Two-phase flow visualization

2.3.1 Flow visualization in large channels

There are many configurations the phases in a vapor-liquid system can take. The configurations, called flow patterns or flow regimes, of fully-developed two-phase flow are determined by the various
parameters of the flow, including fluid properties, flow rates, tube diameter, and others. Although there have been several notable analyses of the physical properties within the flow that can successfully predict transition between certain regimes, the complexity of the systems means that visualizations of the configurations must be made. By these visualizations, flow maps have been generated to enable prediction of flow regime based several parameters. The parameters determined to be the most relevant to the flow are typically selected for the axes of the graphs. The various flow regimes are indicated by regions on the graph, which is called a “flow map” when the transition lines are drawn in. The transition lines are determined from experiments or by analysis. One of the earliest flow regime maps was generated by Bergelin and Gazley (1949) for air-water flows in a 25.4 mm (1 inch) pipe. An early success at selecting axes was Alves (1954), who chose the superficial velocities as axes which allowed air-water and air-oil flow regimes in a 25.4 mm (1 inch) pipe to be represented on a single map. Baker (1954) had success combining data sets from several sources involving mostly air-water in several diameters onto a single map by selecting coordinates in such a way to provide common transition boundaries. Mandhane et al. (1974) used an existing database of nearly 6000 data points, covering pipe diameters from 12.7 to 165 mm (0.5 to 6.5 inches) and a variety of fluids and fluid properties, to produce a flow map that has had success in the range of the fluids studied, shown in Figure 2.7.

Dukler and Hubbard (1975) and then Taitel and Dukler (1976) performed geometrical and physical analyses of the phases and determined some theoretical transition lines between flow regimes. When drawn on the same axes as the Mandhane et al. flow map, Figure 2.8, the theoretical lines compare well in the range of pipe diameters used in developing the Mandhane et al. map.

Since the pipe diameter plays a role in the physics that are relevant to the two-phase flows, Barnea et al. (1983) looked at air-water two-phase flows in pipe diameters that varied from 4 to 12 mm and incorporated surface tension forces into the Taitel-Dukler analysis. This allowed for a better prediction of the regime transition in smaller channels. Their analysis resulted in a criteria for the existence of the stratified flow regime as a function of pipe diameter. Stratified flow is where the lower density phase remains on top of the higher density phase, and does not exist in smaller pipes as a result of the surface tension forces. The criteria Barnea et al. (1983) developed was based on the Bond number, Equation 2.42, and the fluid properties.
Figure 2.7: Flow map developed by Mandhane et al. (1974)

Figure 2.8: Taitel-Dukler flow map compared to Mandhane et al. flowmap
Concurrently to the work of Barnea et al. (1983), and also located at Tel Aviv University, Brauner and Maron (1992) performed a stability analysis on the stratified flow regime to include the stabilizing effect of surface tension on a wide variety of pipe diameters. Surface tension becomes the dominant stabilizing term as the pipe diameter decreases, since the wavelengths that can exist at the interface of the flow are limited by channel geometry. They identified the Eötvös number, which is related to the Bond number as is described in Section 2.1.3, as the relevant criteria for the existence of the stratified flow regime. They compare this criteria with the air-water data of Luninski (1981), and although the figure quality suffers in the archived copy of the article, their analysis is validated within the 4 to 25.2 mm range they examined.

Under the direction of Prof. Westwater, on the campus of the University of Illinois, Graska (1986) and Damianides (1987) studied the effect of decreasing diameters on flow regimes. Graska (1986) observed flow regimes in channels of 5, 12.7, and 16.5 mm using air-water and air-acetic acid flows and observed that none of the existing flow maps would predict the flow regimes seen in his 5 mm pipes. The predictions of the Taitel-Dukler map would not predict correctly the flow regimes observed in the 12.7 mm pipes. Graska concluded that surface tension effects were not significant for 12.7 mm pipes, but were significant for 5 mm. Damianides (1987) studied flow regimes of air-water in 1, 2, 3, 4, and 5 mm pipes, observing that smooth, stratified flow could not be obtained for any pipe size, and that any form of separated flow could not be obtained in 1 mm diameter. These observed flow maps were presented on axes of superficial liquid and vapor velocities (Damianides and Westwater, 1988).

Flow visualizations of refrigerant liquid-vapor flows have revealed that the same flow pattern maps that apply to air-water regime boundaries do not apply to two-phase refrigerant flows. The Steiner map was a modification to the Taitel-Dukler map to match the observed flow regimes of R12 and R22 vapor-liquid flows. Kattan et al. (1998) improved upon the Steiner map for more contemporary refrigerants. They first translated the Steiner flow map to the coordinates of mass flux, $G$, vs. vapor quality, $x$. These axes are well-suited to refrigeration systems which involve only one fixed set of fluid properties, but they are not suited to collapsing the flow regimes of multiple fluids, since the transition lines need to be recalculated based on fluid properties. Kattan et al. (1998) performed flow visualizations of diabatic refrigerant vapor-liquid in a 12.0 mm diameter
glass tube using R123, R134a, R402A, R404C, and R502. From these visualizations, they modified
the Steiner map transition lines for the stratified-wavy/annular transition and also addressed the
transitions to partial dryout and mist flow regimes from the annular regime. Subsequent modifi-
cations to this map have been published, including Hajal et al. (2003) and Wojtan et al. (2005),
which have refined the transition lines and also identified transition curves between dryout and
mist flow regimes, and also the slug and stratified-wavy regimes. The intermittent-to-annular tran-
sition, which is of primary interest in small channels, has remained unmodified from the original
Steiner map. Transformed to $x$-$G$ coordinates, which makes the transition occur at a constant
vapor quality, $x_{IA}$, the intermittent-annular transition is given by:

$$
x_{IA} = \left\{ 0.34^{1/0.875} \left( \frac{\rho_v}{\rho_l} \right)^{-1/1.75} \left( \frac{\mu_l}{\mu_v} \right)^{-1/7} + 1 \right\}^{-1}
$$

(2.69)

Coleman and Garimella (2003) visualized vapor-liquid flows of condensing R134a in nine dif-
ferent channels with round, square and rectangular cross-sections and varying hydraulic diameters
from 1 to 4.91 mm, including multi-port tubes. The flow regimes they observed were intermit-
tent, wavy, annular and dispersed. Wavy was a form of annular flow with a thicker liquid film on
the bottom of the pipe and a wavy vapor-liquid interface, caused by shear between the phases.
Dispersed flow included bubbly flow, when the liquid phase was turbulent and the vapor phase
disperses throughout the liquid. Between the square, rectangular and round channels, at hydraulic
diameter of approximately 4 mm, the intermittent flow regime was found to be largest in the round
channels, extending to higher vapor qualities and mass fluxes. They explanation was given that
square and rectangular geometries retain liquid in the corners of the channels and therefore transition
to annular flow more easily (at lower qualities and mass fluxes). The flow map, which was first
developed by Coleman and Garimella (1999) and then subsequently extended by Garimella et al.
(2002) and Coleman and Garimella (2003), identified the flow to be in the intermittent regime by
the following function of vapor quality, $x$:

$$
x \leq \frac{a}{G + b}
$$

(2.70)

where $G$ was the mass flux, in $\text{kg m}^{-2}\text{s}$, and the coefficients $a$ and $b$ are given as function of $d_h$, measured
in millimeters:

\[ a = 69.5673 + 22.595 \exp(0.2586 \cdot d_h) \]  \hspace{1cm} (2.71) \\
\[ b = -59.9899 + 176.8137 \exp(0.3826 \cdot d_h) \]  \hspace{1cm} (2.72) \\

Wang et al. (1997) observed two-phase vapor-liquid flow of R22, R134a, and R407C in a 6.5 mm glass pipe. The observed flow transition boundaries were also considerably different than air-water flows, although the modifications to the Baker map by Hashizume (1983) which reduced the surface tension correction factor matched the visualization data of Wang et al. (1997) well. The flows of R407C were seen to be different than those of R22 or R134a, and this difference was attributed to the blend refrigerant evaporating its components unequally, resulting in a unique variation of fluid properties across the vapor quality. R407C is a blend of R32, R125 and R134a, while R22 and R134a are single-component fluids.

Yang and Shieh (2001) performed flow visualization of air-water in pipes of 1, 2, and 3 mm for R134a liquid and vapor in pipes of 2 and 3 mm. The flow regime transitions for air-water flows matched well with those observed by Damianides and Westwater (1988) and no stratified flow was observed in the 1 mm pipe. However, the flow regime transitions were different for the R134a flows. The slug to annular regime transition occurred at lower values of superficial vapor velocity for R134a than for air-water, which was attributed to the lower surface tension of the refrigerant.

Wongwises et al. (2002) visualized two-phase flow regimes of R134a with a 5% mixture of PAG oil in pipes of 7.8 mm diameter. New flow regimes that were attributed to the oil were observed, all involving froth on the liquid-vapor interface. The froth consisted of tiny bubbles of vapor, which were observed to take up varying percentages of the pipe. In “froth-wavy” flow, the bubbles took up a small portion of the pipe, elevating the apparent liquid level a little. In “froth flow”, the channel was observed to be completely filled with tiny bubbles. The flow regimes observed were compared unsuccessfully to the Mandhane et al. (1974) flow map. The grouping of the observed data points onto the axes of superficial liquid and vapor velocities make it appear that the choice of superficial velocities is not an appropriate one to make, since the observed flow regimes overlap one another on these axes.
Tabatabai and Faghri (2001), using a balance of forces on the liquid bridges that make the difference between slug and annular flow, develop a criteria for the transition between flow regimes that includes surface tension forces that occur in the bridges of the flows. This transition is favorably compared criteria to a database of flow visualization points, which included multiple refrigerants in pipe diameters ranging from 4.6 mm, as well as air-water flows in pipe diameters down to 1 mm.

Some investigations have been directed at the fluid-wall interactions related to the wetability of the pipe walls. Barajas and Panton (1993) visualized air-water flows inside four pipes, all with 1.6 mm inner diameters but with varying wall-water-air contact angles ($\theta$) of 34°, 61°, 74°, and 106°. The observed flow regime maps were similar for all of the wetting ($\theta < 90^\circ$) pipes, and the flow regime map of the $\theta=34^\circ$ pipe they say matched well with the map of Damianides and Westwater (1988). The significant difference as the contact angle of the wall increased was the appearance of a “rivulet” and “multiple rivulet” flow regimes, where the liquid flowed in one or more streams on the wall of the tube. It was not confined to the bottom being found on the sides and top as well, and resembled the flow of a river. The rivulet regime did not occur at all on the $34^\circ$ pipe, and took up an increasing amount of the region that was classified as “wavy” flow in the pipes of increasing contact angles. The non-wetting pipe, $\theta=106^\circ$, had a sizable portion of the annular regime being taken up by multi-rivulet flow, in addition to a shifting of the slug-annular transition line earlier into the slug region. The comparison of the flow maps developed for the different pipes is shown in Figure 2.9.

Iguchi and Terauchi (2001) studied two-phase air-water flows in wetting and non-wetting pipes with diameters of 5, 10, and 15 mm and contact angles of 36°, 77°, and 104°. The flow direction was vertically upward, which is different between this and the other studies mentioned here which all involved horizontal flows. The flow regimes observed for the $36^\circ$ and $77^\circ$ pipes were almost identical, indicating that as long as the pipe was wetted, no difference was observed. Even in the non-wetted pipe of $\theta = 104^\circ$, no major differences were noted, although the bubbly flow in the non-wetting pipe was seen to have bubbles attach and detach themselves from the walls as they moved up the channel. It is important to note that the only transition boundary that was examined in this study was that between the bubbly and the slug flow regimes; the slug to annular flow regime was not investigated.
It seems from these studies that while the issue of fluid-wall energy is one of the elements involved in two-phase flow, the contact angle of the wall-liquid-vapor combination is not a significant contributor to flow regime.

2.3.2 Flow visualization in microgravity

The Bond number, Equation 2.42, can be used as a criteria for distinguishing small channels from large channels. The ratio of buoyancy forces to surface tension forces present makes it possible to tell when stratified conditions will exist in the flow, and the concept of “small” channel can be defined as one that does not exhibit stratified conditions (Barnea et al., 1983; Brauner and Maron, 1992). One way to eliminate stratified flow conditions is to reduce the diameter of the channel, thus increasing the relative magnitude of the surface tension forces. Another way to eliminate stratified flow is to reduce the buoyancy forces by means of a reduced gravity field. Two-phase flows in microgravity conditions have similar fluid physics to microchannel flows, and flow maps that were developed in microgravity have brought insight to microchannel flows.
Dukler et al. (1988) observed microgravity flow patterns of air-water in a 9.52 mm diameter pipe and then proposed a transition criteria between the slug and annular regimes based on slug voids, the percentage of the liquid slug observed to be filled with vapor bubbles. Unfortunately for the universal application of this criteria to microchannel flows, this slug void criteria is dependent on the visualization of the tiny bubbles that were present in their liquid slugs. Without images of the flow, it is impossible to guess what fraction of the slug is bubbles and high density of bubbles in the slugs have not been observed in microchannel flows. Therefore, this transition criteria is not one that can be adopted in the present microchannel flows.

Zhao and Rezkallah (1993), Rezkallah and Zhao (1995), Rezkallah (1996), and Lowe and Rezkallah (1999) made a series of microgravity visualizations of two-phase flow in a parabolic-trajectory jet in pipes of diameters of 9.5, 12.7, and 25.4 mm and then later in a series of 6–40 mm pipes. The working vapor phase was air, and the working liquid phase was water and a glycerin-water mixture to modify the fluid properties. The most significant variation was to surface tension. In this series of experiments and analysis, they developed a flow map that took superficial Weber numbers (Equations 2.33 and 2.32) for the axes with transition criteria to account for the various flow conditions. The Weber number axes were selected and initially the transitions were thought to be constant values of $W_{el}$, but in subsequent developments a slanted transition line was found.

These Weber number axes were also adopted by Bousman et al. (1996) to attempt to explain their microgravity flow visualizations in 12.7 and 25.4 mm pipes. Air was the vapor phase, and water mixed with varying concentrations of Zonyl to provide a range of liquid viscosities and surface tension values. They noted that the Weber number axes provided clean transitions at low values of $W_{el}$, but not for higher values. They also noted that the bubble-to-slug transition was sensitive to diameter, viscosity and surface tension, but the slug-to-annular transition was not observed to be affected by any of these things. The slug-to-annular transition is the most significant boundary in microchannel flows, since it is the transition between surface tension dominated and inertial dominated flow regimes.

Parang and Chao (1999) took collections of flow visualization data that were available from microgravity experiments. The diameters of the pipes included ranged from 9.5 to 40 mm, and the fluids were primarily air-water, although the Zonyl and glycerine mixtures were used as well.
In order to determine which parameters were relevant for flow regime map axes, they plotted only the data points that were near or on the flow regime boundaries. This made for easy identification of parameters were selected for axes when the transition boundaries between different data sets collapsed onto a single line. They found that the axes \( \frac{We_v}{Re_l} \) vs. \( We_l \) gave a transition from slug to annular flow at a constant value of \( \frac{We_v}{Re_l} = 0.03 \).

Zhao and Hu (2000) developed a criteria for transition between the slug and annular flow regimes of microgravity two-phase air-water flow based on the idea that the inertia of the vapor must break through the liquid bridge in the slug. The criteria was:

\[
\rho_v j_v (V_v - V_l) = O \left( \frac{2\sigma}{r} \right) \tag{2.73}
\]

where \( \rho_v j_v (V_v - V_l) \) is the impulsive force of the vapor relative to the liquid bridge, which must be on the same order of magnitude as the surface tension force in the nose of the vapor bubble, \( 2\sigma/r \), to break through and transition to annular flow. They introduce the empirical constant, \( \kappa \), which is order 1 to indicate order of magnitude, \( \kappa = 0.8 \) was the value that caused this criteria to successfully match the experimental data. The following relation of void fraction was used:

\[
\frac{j_v}{j_v + j_l} = \alpha C_0 \tag{2.74}
\]

which brought in a second empirical constant, \( C_0 \). The right-hand-side of Equation 2.74 was introduced earlier in Equation 2.14 as the volumetric quality, \( \beta \). Zhao and Hu (2000) gave \( C_0 \) the constant value of 1.16, which makes Equation 2.74 into a homogeneous relation for void fraction. The Armand expression, explained in Chung and Kawaji (2004), uses the value \( C_0 = 1.2 \).

Zhao and Hu’s transition criteria for the flow to be in annular flow is then given by:

\[
\sqrt{We_v} \geq W_v \tag{2.75}
\]

where:

\[
W_v = \sqrt{\frac{4\kappa C_0 \sqrt{\alpha(1 - \alpha)}}{C_0 - 1}} \tag{2.76}
\]

This transition criteria was compared to multiple microgravity flow visualizations and a capillary
flow visualization with success. The use of empirical constants for $\kappa$ and $C_0$ make this less than purely mechanistic, but the balance of inertial forces to surface tension forces appeal to the physical elements of making a flow regime map. What is neglected, however, is a consideration of the length of the liquid slug that must be overcome; the analysis as it is presumes only surface tension forces on either side of a short liquid bridge and never accounts for the additional surface energy that must be created to elongate the nose of a bubble to stretch through liquid slug of considerable length.

2.3.3 Flow visualization in small channels

In regular gravity, the dominance of surface tension forces over gravity comes at smaller channel diameters. Some attempts at grouping channels by size, e.g. Kandlikar (2002), have been made however the Bond number criteria are to be preferred since the channel classification is clearly dependent on the fluid properties of the flowing liquid in addition to the overall size.

Damianides (1987) observed that between 1 and 2 mm, separated flow in air-water flows were no longer observable. As flow visualization studies have been observing smaller and smaller flows, there has been agreement (Kandlikar, 2002) that only three general two-phase flow patterns are observed in small channels: isolated bubble, confined bubble or plug/slug, and annular. Many studies have classifications with increased refinement on these three categories, but these are the only three categories that are seen.

Akbar et al. (2003), in an effort to develop a flow map that could be used in two-phase and slurry bubble transport (Akbar, 2004), compiled a collection of two-phase air-water visualization data from the literature of flow in circular channels with diameters around 1 mm. They adopted the Weber number axes from Lowe and Rezkallah (1999) but modified slightly the transition lines to provide a rough best-fit of the compiled data sets. Figure 2.10 shows the transition lines from the compiled data as well as their proposed map. Of interest in the work of Tu and Hrnjak (2004) is the transition line in the upper left, which separates the inertia dominated flow regime from the transition regime leading to the surface tension regime. The equation of that line, which was discussed above, is given by:

\[
We_v = 11.0 We_l^{0.14} \tag{2.77}
\]
Figure 2.10: Flow map developed by Akbar et al. (2003) as a compilation of visualization data compiled from other sources
Triplett et al. (1999) visualized air-water flows in circular microchannels of \( d = 1.1 \) and \( 1.45 \) mm, and semi-triangular microchannels with \( d_h = 1.09 \) and \( 1.49 \) mm. They observed the flow regimes bubbly, slug, churn, slug-annular, and annular. Churn flow was a regime of elongated slugs, with tail breakup and bubbles entrained in the annular film of liquid. Slug-annular flow had bulges of liquid in the annular film. The flow regime maps observed for the four channel geometries were similar to one another but did not match with any of the flow pattern maps developed in larger channels (Suo and Griffith, 1964; Taitel and Dukler, 1976; Barnea et al., 1983). The regime transitions of Damianides and Westwater (1988) provided a reasonable match.

Zhao and Bi (2001) observed upward flowing two-phase air-water flows through triangular channels of reported \( d_h = 0.866, 1.443, \) and \( 2.886 \) mm. The flow regimes observed consisted of: dispersed bubble, slug, churn, and annular. In the smallest channel diameter low flow rates produced a capillary bubble flow pattern, which was influenced by the combination of the sharp corners and the confining walls of the channel present in the triangular crosssection. The previous flow regime maps that had been developed in round microchannels did not sufficiently predict the observed regime boundaries. The flow transition boundaries observed by Triplett et al. (1999), which were made in a triangular channel, were the best match to the observed data, although Triplett et al. (1999) observed dispersed bubble flow instead of the capillary bubble regime.

Chen et al. (2002) performed flow visualization of nitrogen-water flows in glass channels of \( d_h = 1.0 \) and \( 1.5 \) mm. They classified their observations into the categories: bubbly, slug, bubble-train slug, churn, and annular flows. Representative photographs from their visualizations are shown here in Figure 2.11. They only observed bubbly flow in the 1.5 mm channels. This is in minor contrast to the observations of Triplett et al. (1999), who presented photos of bubbly flow in 1.09 mm channels. However, this discrepancy is not troubling because the flow patterns that are achieved in microchannels are highly dependent on the inlet geometry of the test section, especially with regard to flows of low vapor content. The regime that was named bubble-train flow consisted of one or more vapor bubbles that were separated only by a “clear interface between the connecting bubbles”. The “clear interface” was perhaps more clear in the actual visualizations because it was not completely clear in the photos reproduced in the article. From the provided photo the third regime shown in Figure 2.11, this regime looks like it also might be a bulge of liquid in the annular
film that goes completely around the channel. They say that this flow regime has been reported by others, it was called “frothy slug flow” by Zhao and Rezkallah (1993) and that the regime occurs when the inertial force of the trailing bubble is large enough to overcome surface tension and break through the liquid slug, but is not large enough to rupture the interface. Flow patterns that appear similar to this regime have been observed in the present study when oil was added to the refrigerant, in Chapter 5.

Serizawa et al. (2002) visualized air-water flows in circular channels of diameters of 20, 25, and 100 µm and for a steam-water flow in a circular channel of 50 µm diameter. A microscope and camera arrangement was used to observe the flow. Some new titles were given for flow regimes observed: liquid ring flow is a form of annular flow with regularly-spaced and symmetric bulges of liquid in the annular film and liquid lump flow has irregularly spaced bulges. Bubbly flow and slug flow were observed at low vapor qualities, as well as droplet flow at high vapor qualities. Of note were slug flows with wall dryout, where small droplets could be seen sticking to the walls and an annular film front could also be seen. They made a comparison to the flow map of Mandhane.
et al. (1974), and claim that the predictions of that map were good. However, the dispersed flow regime of Mandhane’s map was left off of the comparison, even though some of the data fell into that region. No stratified flow was observed, and in the stratified regime of Mandhane’s map were observed slug and bubble flow. Mandhane’s region of annular flow was populated by the liquid ring and liquid lump flow observations, which they were, although some of the liquid ring observations fell into the predicted slug region. The effect of surface wettability was investigated by cleaning the channel by a series of steps which increased the wettability of the inside of the glass. Stable annular flow was observed only when the channel was cleaned, as well as a flow regime they named skewered flow because the vapor bubbles resembled a skewered Japanese barbecue.

Chung and Kawaji (2004) performed flow visualization in circular microchannels of diameters: 50, 100, 250, and 500 µm using air and purified water. In the 250 and 500 µm channels, they observed flow pattern typical of other visualization studies of 1 mm channels, such as Triplett et al. (1999). However, in the 50 and 100 µm channels, only slug flow was observed, even annular flow was not observed. Again, since the existence of a particular flow pattern in a microchannel is dependent on the inlet geometry, it can not be concluded that annular flow does not exist in channels of that size from its reported absence. It was Prof. Kawaji who pointed out, in a personal conversation, the dependence on inlet configuration in microchannel flows. In this work, they calculated void fractions from their visualizations and found that the two larger channels acted homogeneous with respect to void fraction, but that the smaller channels did not, even though the only observed flow pattern was slug flow. They postulated that perhaps because these channels had such a higher pressure gradient associated with them, the vapor bubbles were traveling faster through the liquid leading to a lower observed void fraction and also the potential of missing them in the camera.

Qu et al. (2004) conducted flow visualization with nitrogen-water flows in rectangular microchannels with reported dimensions of 0.406 × 2.032 mm (yielding a $d_h$ of 0.677 mm). They observed the typical regimes of slug and annular flow, as well as: bubbly/slug flow, which consisted of slugs interspersed with smaller bubbles; liquid/slug flow, which was slug flow oscillating with sections of single-phase liquid; and liquid/annular flow, which was annular flow oscillating with single-phase liquid. The flow regime map, shown in Figure 2.12, did not match any previous flow map in large channels.
Pettersen (2004) observed the flow patterns of near-critical, evaporating CO$_2$ flow in a channel of diameter 0.98 mm. The observed flow patterns did not match any of the known flow pattern maps developed for large tubes. Entrainment of the liquid into the vapor flow was observed here where other microchannel studies do not mention such a flow characteristic. Another interesting observation was that with no heat flux, at 0°C temperature, a stratified flow condition was observed. At that condition with heat flux, the flow regime of that point was observed as annular. The heat flux conditions were achieved by a transparent resistive film coating on the outside of the glass visualization section, so the visualization was being done in the heated section. Since CO$_2$ was near its critical point ($T_{\text{critical}}=31.1^\circ\text{C}$) at the working temperatures, the value the surface tension of the flow is significantly lower than other working fluids. This is what allowed for the stratified conditions at such small diameters, and also allows for smaller droplets that could be entrained into the vapor flow. It is also clear that the flow regime maps for diabatic and adiabatic conditions will not be the same, as the evaporation will affect the flow regime.
Revellin et al. (2006) visualized adiabatic flows of R134a in a glass channel of 0.5 mm. The flow patterns observed were: bubbly, slug, semi-annular, and annular flow. The flow regimes observed did not match any of the existing flow pattern maps for large channel refrigerant or microchannel air-water flows. Although no generalized flow map was developed in this work, experimental transition criteria were sketched onto the plots, and an optical measurement and classification technique was developed indicating that a map consisting of a much wider data set to be coming soon from the EPFL.

2.4 Refrigerant-oil mixtures

2.4.1 Refrigerant-oil studies

The effect of oil on refrigeration systems has a long history, however for much of that history, the primary concern had been for heat transfer effects of oil. Even recent literature finds the effects of oil on heat transfer to be a big interest (Dang et al., 2007). Increases in pressure drop have been noted in evaporators, and attributed to the thick film of oil resulting from immiscible refrigerant-oil combinations (Shah, 1975), and the increased pressure drop have been accounted for in correlations (Pierre, 1964). The immiscible refrigerant-oil combinations are still a topic of present interest (Yun et al., 2007), especially with regard to microchannels.

Studies have also been done for visualization of the various oil flows. As mentioned above, Wongwises et al. (2002) observed a “frothy” flow pattern in flows of R134a with oil, and the increased effective liquid height was thought to be the cause of a shift in the transition between stratified flow and annular flow. Motta et al. (2002) performed visualizations of flows of R410A and oil in capillary tubes.

One disadvantage of all the studies cited above is that the precise circulation of oil is not easy to control. Furthermore, the experimental results are typically reported based on the refrigerant-oil pair and the estimated circulation rate, and not in terms of the quantity of oil present in the channel and thus the fluid properties of the mixture.

In the following sections, the basic fluid and mixture relations for miscible refrigerant-oil combinations will be laid out. It will be this framework that will be used in Chapter 5 to describe the
actual content of the flows that are studied.

2.4.2 Equations of two-phase refrigerant-oil flows

Oil that circulates with the refrigerant flow is quantified by oil circulation rate, \( OCR \), which is the ratio of oil flow to total flow in the refrigerant-oil mixture. In an evaporating flow, the oil will not vaporize, so as the refrigerant-oil mixture is heated only liquid refrigerant boils into vapor, and the remaining liquid phase will have an increasing local oil concentration in the direction of the flow. The opposite happens in a condensing flow: the oil concentration in the liquid phase decreases along the condensing length. Since it is the local concentration of refrigerant and oil that affects the local fluid properties, it is the local oil concentration that is critical to considering a flow with constant \( OCR \) and changing quality. Considered in this work are only miscible refrigerant-oil combinations, in which the liquid refrigerant and oil constitute a single mixed phase and the liquid phase is herein considered to be a homogeneous mixture of the liquid refrigerant and oil.

Refrigerant-oil liquid mixture properties as function of local oil concentration have been measured by Seeton and Hrnjak (2006) for multiple combinations of refrigerants and oils. An example mixture is the combination of R134a-POE32, for which a complete correlation for density and viscosity of the mixtures is known for temperature and refrigerant composition.

Figure 2.13 is an example plot of the viscosity of the mixtures as a function of temperature that was measured. The mass concentration of refrigerant in oil is shown on the right side of the plot, with pure R134a at the lines of lowest viscosity, and pure oil at the highest. The curved dotted lines are isobars, lines of constant mixture pressure. An evaporating or condensing two-phase flow will follow an isobar, as shown by the yellow dots for 3 bar, resulting in a gliding temperature for the mixture.

From this, the local fluid properties of a refrigerant-oil mixture can be determined from the local oil concentration. Defining the local concentration of refrigerant in the liquid phase or the mass fraction of refrigerant, \( \omega \), based on the flow rates yields:

\[
\omega = \frac{\dot{m}_l}{\dot{m}_o + \dot{m}_l} \tag{2.78}
\]

On the other hand, \( OCR \), the oil circulation rate on a “sample basis”, given by ASHRAE Standard

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41.4 is the ratio of oil flow rate to total flow rate, liquid, vapor and oil $\dot{m}_\text{tot} = \dot{m}_\text{v} + \dot{m}_\text{l} + \dot{m}_\text{o}$:

$$OCR = \frac{\dot{m}_\text{o}}{\dot{m}_\text{tot}}$$  \hspace{1cm} (2.79)

and the vapor quality of oily flow is considered equivalent to the thermodynamic quality and defined as:

$$x_{\text{oily}} = \frac{\dot{m}_\text{v}}{\dot{m}_\text{tot}} = x$$  \hspace{1cm} (2.80)

Combining Equations 2.78 – 2.80, we can determine the local concentration of refrigerant as a function of quality and $OCR$:

$$\omega = 1 - \frac{OCR}{1 - x}$$  \hspace{1cm} (2.81)

The local concentration of oil in the refrigerant-oil mixture is given by $1 - \omega$.

Note that from these definitions the quality of the refrigerant flow, $x_{\text{oily}}$, is not allowed to go
from 0 to 1, but rather from 0 to \((1 - OCR)\), since there will always be some liquid oil in circulation. If \(x_{oily} = (1 - OCR)\), the refrigerant fraction \(\omega\) would be 0, meaning all the liquid phase would be oil. It is important to note that in practice this does not happen: governed by the solubility of the refrigerant-oil combination, some of the refrigerant remains dissolved in the oil and is “stuck” in liquid form. Thus, the liquid phase will be made up of refrigerant and oil, the limiting concentration of which is the solubility limit of the refrigerant/oil mixture at the temperature and pressure of the flow. As energy is input to the mixture, an increase in temperature and pressure will result which will change the solubility level of the mixture and boil refrigerant out of solution.

In calculations involving flowing refrigerant, it is convenient to calculate quality based on heat input instead of liquid fractions, since the heat input is easily calculated. This gives rise to “apparent superheat” conditions, in which the outlet of the evaporator appears to be superheated by temperature measurements but in fact still has an amount of liquid refrigerant trapped in the oil. This “pseudo-superheated quality”, \(x_{psh}\), can be greater than one and is determined as the ratio of the heat added to a saturated liquid flow to the latent heat of the mixture:

\[
x_{psh} = \frac{q_{input}}{h_{fg}(P)}
\]

(2.82)

where \(h_{fg}\) is the latent heat of pure refrigerant which is a function of pressure. Local saturation pressure is preferred over temperature, because of solubility effects of the refrigerant-oil mixture. It is important to note that \(x_{psh}\) is what is calculated in practice, and it is always assumed to be the vapor quality. In reality, \(x_{psh}\) will be slightly higher than \(x\), and is allowed to be greater than one.

### 2.4.3 Properties of evaporating refrigerant-oil flows

The local concentration given in Equation 2.81 can be plotted for various OCRs. In Figure 2.14 the local oil concentration, \((1 - \omega)\) is plotted as a function of quality, \(x\), which is considered equal to \(x_{oily}\) and \(x_{psh}\). Note that the local oil fraction for pure liquid \((x=0)\) is equal to the OCR, and then it increases as refrigerant is boiled out of the liquid phase. A 5 °C apparent superheat was assumed at the exit of the evaporating flow, which meant that highest vapor quality attained was not equal to 1, but rather determined by the solubility limit of the refrigerant-oil mixture at that
temperature. This particular solubility limit is the only feature of this plot that is unique to the R134a-POE32 combination; other refrigerant-oil combinations will have the same shape as those shown here, and the curves will terminate before quality 1.

From the liquid composition, the viscosity and density of the liquid phase can be calculated for a given quality and initial OCR. The variation of these fluid properties with evaporating quality are shown in Figures 2.15 and 2.16.

It can be seen in Figure 2.15 that the viscosity of the mixture shows the greatest increase at high vapor qualities. Since the flow through a typical evaporator will spend the majority of the length of the flow in these higher-quality regions, the high viscosities have an importance that is not apparent strictly from the vapor quality axis. The liquid mixture density, shown in Figure 2.16, has an even larger variation along the evaporating flow, and it is at the high quality regions of the flow that the steepest changes in fluid property occur.
Figure 2.15: Viscosity of R134a-POE32 mixture for varying quality

Figure 2.16: Density of R134a-POE32 mixture for varying quality
Chapter 3

Pressure Drop

3.1 Experimental facility

A schematic of the test facility for measuring pressure gradient is shown in Figure 3.1. It is arranged in a once-through system driven by a temperature-induced pressure differential. The refrigerant is heated in the reservoir tank to provide this driving differential. Liquid refrigerant is removed from the bottom of the tank and is subcooled in the piping to room temperature before it reaches the mass flow meter. The mass flow meter (Rheotherm model, TU1/16) measures liquid flow rate based on an energy balance and must be supplied with subcooled liquid. The flow meter was calibrated for each fluid by weighing the refrigerant moving through and averaging over a five-minute interval. Each experimental run had from 10 to 15 °C of subcooling at the mass flow meter. The temperature and pressure of the refrigerant were measured before the expansion/metering valve. Immediately after the metering valve, the refrigerant piping entered a vacuum dome, which minimized the convective losses. A electric heater supplied a heat flux, $\dot{q}$, to the flow to control the vapor quality of the refrigerant entering the channel. The current supplied to the heater was measured by the datalogger. Saturation pressure as well as differential pressure were measured inside the channel via two pressure taps. Outlet temperature and pressure were measured, and then the refrigerant was collected in a receiver which was maintained at ambient temperature and placed on a digital balance (Sartorius model BP6100). The balance reading was recorded by the data logger and was used as a redundant check of the mass flow rate of the refrigerant. A photograph of the portion of the test facility under the vacuum dome can be seen in Figure 3.2.

The absolute pressure measurements were made with transducers (Setra, model 206, range 0-1723 kPa), with an accuracy of 0.2% of full scale. The differential pressure measurements were made with a differential transducer, (Sensotec, model number Z/5556-01, range 0-34.5 kPa), with
Figure 3.1: Test facility schematic. Vacuum chamber was used to eliminate convective losses in the heater and piping.

Table 3.1: Experimental Uncertainties

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{m}$</td>
<td>±5%</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>±1%</td>
</tr>
<tr>
<td>Temperature</td>
<td>±0.2 °C</td>
</tr>
<tr>
<td>Absolute Pressure</td>
<td>±3.5 kPa</td>
</tr>
<tr>
<td>Differential Pressure</td>
<td>±0.34 kPa</td>
</tr>
<tr>
<td>Quality ($x$)</td>
<td>±5.25%</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>±0.17 kPa</td>
</tr>
</tbody>
</table>

an accuracy of 1% of full scale. Temperature measurements were made with type-T thermocouples, with an accuracy of 0.2 °C. A propagation of error analysis was preformed, with the measurement and calculated uncertainties listed in Table 3.1.

3.2 Test section

The pressure drop test section was made in two halves of a piece of aluminum that were bolted together. The channel was milled into the bottom half, using a flat-tipped mill. Inlet and outlet ports were drilled into both ends of the channel and two pressure tap holes positioned in the middle of the channel. The top half was attached by means of bolts to form the rest of the channel. A photograph of the bottom piece containing the milled channel is shown in Figure 3.3. The holes
Figure 3.2: The portion of the facility under the vacuum dome
around the perimeter of the piece are threaded for the bolts from the top of the channel. The test section was made in this way to allow for a detailed measurement of the channel size. It was noted by the present author experiments that as much as a 10% deviation in the measurement of the channel dimensions resulted in a 50% error in single-phase frictional pressure drop relations. It was also noted in these same experiments by the same author that the PVC channels used previously were not chemically compatible with R410A or ammonia; both fluids were absorbed into the plastic causing swelling of the flat top surface of as much as 10% of the channel height. The dimensions of the aluminium channel were measured by a Sloan Dektak stylus surface profilometer. Measurements were made at eleven crossectional locations and averaged to determine overall channel dimensions. The maximum variation in measurement of these depths was 2.5 \( \mu \text{m} \), and maximum variation in width between these crossectional locations was 14 \( \mu \text{m} \). A representative plot of the channel profile can be seen in Figure 3.4. Note that the side walls of the channel are actually vertical, but the stylus tip on the profilometer produces the angle seen in the figure. Thus, the width was measured from uppermost start of the downward slope. The channel depth was measured to be 100.4 \( \mu \text{m} \) and the width was measured to be 281.1 \( \mu \text{m} \), giving a hydraulic diameter of 148.0 \( \mu \text{m} \).

Figure 3.3: Photograph of milled channel

Two-phase pressure drop data from R134a that was collected in five PVC test sections by Tu (2004) was combined with the two-phase pressure drop data from the aluminum test section collected here. The five PVC channels, which ranged in size from 70 \( \mu \text{m} \) to 305 \( \mu \text{m} \) with aspect ratios ranging from 0.09 to 0.24, are described by Tu and Hrnjak (2002), Tu (2004), and Tu and Hrnjak (2004). The aluminum channel was manufactured with an identical design as the PVC ones, but with a higher aspect ratio (aspect ratio of 0.36). The milled finish of the aluminum surface was then polished to be smooth enough to form a seal between the top and bottom halves of the
Figure 3.4: Profile of channel. The angled side walls are an artifact of the measurement stylus rather than the channel.

3.3 Experimental conditions

The experiments were carried out at ambient room temperature, which ranged from 22 to 25 °C. The pressure measured at the test section was the saturation pressure for the respective fluid, which corresponded to reduced pressures of: 0.09 for Ammonia, 0.17 for R134a, 0.23 for Propane, and 0.36 for R410A. However, reduced pressure (which is the dimensionless ratio of pressure to critical pressure) is not the most fundamental way of considering the flow. The most relevant fluid property that is related to reduced pressure is surface tension. Table 3.2 shows the variation of fluid properties by fluid in the experiments conducted. Note that for these fluids and at these conditions, an increase in surface tension correlates to a decrease in reduced pressure but that between multiple fluids reduced pressure is insufficient to characterize the variations in surface tension.
Table 3.2: Comparison of fluid properties between different refrigerants at current experimental conditions.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>$\rho_l$ [kg/m$^3$]</th>
<th>$\rho_{ratio}$ [-]</th>
<th>$\mu_l$ [mP]</th>
<th>$\sigma$ [dyne/cm]</th>
<th>$P_{red}$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>R410A</td>
<td>1050</td>
<td>14.8</td>
<td>1.16</td>
<td>5.2</td>
<td>0.36</td>
</tr>
<tr>
<td>Propane</td>
<td>491</td>
<td>23.2</td>
<td>0.975</td>
<td>6.8</td>
<td>0.23</td>
</tr>
<tr>
<td>R134a</td>
<td>1200</td>
<td>36.3</td>
<td>1.92</td>
<td>8.0</td>
<td>0.17</td>
</tr>
<tr>
<td>Ammonia</td>
<td>601</td>
<td>74.2</td>
<td>1.3</td>
<td>21</td>
<td>0.09</td>
</tr>
</tbody>
</table>

3.4 Pressure gradients

3.4.1 Single-phase friction factor

Single-phase experiments were carried out in the channel, and the measured laminar friction factors were found to match the laminar flow theory. Laminar flow through a rectangular channel has a constant $C_f$, which depends upon the aspect ratio in a manner described by Hartnett and Kostic (1989) as:

$$C_f = 96(1 - 1.3553s + 1.9467s^2 - 1.7012s^3 + 0.9564s^4 - 0.2537s^5)$$ (2.45)

where $0 \leq s \leq 1$ is the aspect ratio of the rectangular channel. Transition to turbulent flow was also found to follow traditional correlations, and for the turbulent flow regime, the Churchill equations (Churchill, 1977) described the observed friction factors and were used. Figure 3.5 shows the measured laminar liquid refrigerant friction factor compared to Equation 2.45 in the channel used.

3.4.2 Two-phase pressure gradient

Two-phase pressure gradient experiments were conducted with four fluids, R134a, R410A, Propane (R290) and Ammonia (R717). The range of mass fluxes approximately spanned $G = 300-700$ kg/m$^2$s, where the range of mass fluxes for each fluid was limited by the lower limit of the mass flow meter. These data were combined with the aforementioned set of R134a two-phase pressure gradient data from Tu and Hrnjak (2002). The combined data set totaled 393 points.

Figures 3.6 – 3.9 show representative plots of pressure drop per unit length as a function of vapor quality. Presentation of the pressure gradient data in this fashion is a little misleading,
Figure 3.5: Single phase friction factor for laminar liquid refrigerant

because the mass fluxes were not held precisely constant for the data run, and in the numerical analysis the exact values of mass flow were used. The value of mass flux presented in these figures is an average of the data set. For comparison, several predictions from two-phase models are shown as well. In the figures, the solid points are the measured data, and hollow points are predictions from: (1) Dukler’s homogeneous model, given previously in Eq. 2.52, and the separated flow models of: (2) Tu and Hrnjak, Eq. 2.64, with $V = U_B$ for $\psi$, (3) Mishima and Hibiki, Eq. 2.62, (4) Lee and Lee, Eq. 2.64, and (5) Lee and Mudawar, Eqs. 2.66 and 2.67. Several other models were compared to the present data and are discussed below. Refrigerant flow in channels of this size has been compared to many other models (Tu and Hrnjak, 2002), and these were the models found to have the best agreement; many other models were not considered here because of their poor predictions in similar size channels from previous tests.

A composite of all the data taken is shown in Figure 3.10, with uncertainty propagation in both the $x$ and $y$ direction indicated. The uncertainty in measured pressure drop is smaller than the graph point, but the uncertainty in quality measurement can be seen to increase as the quality increases.
Figure 3.6: Two-phase pressure gradient data for R134a, with $G \approx 290 \frac{\text{kg}}{\text{m}^2\cdot\text{s}}$.

Figure 3.7: Two-phase pressure gradient data for Propane, with $G \approx 330 \frac{\text{kg}}{\text{m}^2\cdot\text{s}}$. 
Figure 3.8: Two-phase pressure gradient data for R410A, with $G \approx 450 \text{ kg/m}^2\cdot\text{sec}$.

Figure 3.9: Two-phase pressure gradient data for Ammonia, with $G \approx 440 \text{ kg/m}^2\cdot\text{sec}$.
3.4.3 Comparison to other pressure gradient models

To quantify how well a given model predicts two-phase pressure gradient over the data set of \( N \) values, the mean deviation of the model prediction from the measured pressure gradient is calculated by the formula:

\[
\text{Mean Deviation} = \frac{1}{N} \sum \frac{|\Delta P_{\text{meas}} - \Delta P_{\text{pred}}|}{\Delta P_{\text{meas}}} \quad (3.1)
\]

A summary of the mean deviation of some of the pressure gradient models is shown in Table 3.3, as well as what percent of those predictions were within \( \pm 20\% \) of the measured pressure gradient. The lowest mean deviation was found from the homogeneous viscosity model of Dukler et al. (1964), and largest amount of data within the \( \pm 20\% \) range was found from Tu and Hrnjak (2004). The other homogeneous viscosity models tested, the ones by McAdams et al. (1942) and Beattie and Whalley (1982), also showed reasonable agreement on a mean-deviation basis. The predictive ability of homogeneous models has been found repeatedly in microchannel studies, and is assumed to be related to the high frequency of intermittent-type flow inside small tubes. However, visualizations
Table 3.3: Comparison of predictions made by other models.

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Mean dev. (Eq. 3.1)</th>
<th>Percent data in ±20% range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dukler et al. (1964)</td>
<td>18.2%</td>
<td>65.1%</td>
</tr>
<tr>
<td>Tu and Hrnjak (2004)</td>
<td>18.9%</td>
<td>71.7%</td>
</tr>
<tr>
<td>Mishima and Hibiki (1996)</td>
<td>20.6%</td>
<td>49.9%</td>
</tr>
<tr>
<td>McAdams et al. (1942)</td>
<td>21.0%</td>
<td>60.8%</td>
</tr>
<tr>
<td>Beattie and Whalley (1982)</td>
<td>34.4%</td>
<td>50.6%</td>
</tr>
<tr>
<td>Lee and Lee (2001)</td>
<td>44.0%</td>
<td>26.7%</td>
</tr>
<tr>
<td>Lee and Mudawar (2005)</td>
<td>122%</td>
<td>8.4%</td>
</tr>
<tr>
<td>Friedel (1980)</td>
<td>252%</td>
<td>0%</td>
</tr>
</tbody>
</table>

of small channel flow seem to reveal high slip velocities invalidating the homogeneous assumptions.

Of the separated flow models, the one developed in channels of this size by Tu and Hrnjak had the best agreement with the measured data. With the exception of the ammonia data (Figure 3.9), the predictions of this model are better than any other. This is not a surprise, since the converse has certainly been demonstrated by the inability of models developed with air-water to predict refrigerant flows accurately. The predictive ability of this model for this data set is also not a surprise, since a portion of this data set was originally used to formulate the model of Tu and Hrnjak (2004).

The separated flow model of Mishima and Hibiki (1996) was also found to have close agreement with the measured data, in spite of only computing the value of $C$ from the channel diameter and neglecting any other fluid properties. The remainder of the separated flow models, however, predicted the results poorly. In the case of the Friedel model, which has been widely used in larger channels, the model predictions were considerably different from the measured data and it is only included here to demonstrate the difficulty in using traditional correlations to predict microchannel flows.

Mean deviation is not the only way to gage “goodness” of a model prediction, since it effectively reduces all of the variations possible to a single number. Many other techniques were used to compare the various models in addition comparing mean deviations in order to understand where the models predict well or poorly. For example, the ratio of model prediction to measured pressure gradient was plotted vs. vapor quality in order to see if particular models were better suited in
different regions of quality, which are coarsely related to flow regime. In addition, the data points were separated by fluid and channel size in order to determine if particular models were better suited to different fluids. Figures 3.11, 3.12, and 3.13 show this for the Dukler et al. (1964) homogeneous model, the Tu and Hrnjak (2004) separated flow model and the Lee and Lee (2001) separated flow model. The new measurements points are shown in solid objects, and the R134a data from Tu (2004) is shown as hollow data points. The sharp change in predictive ability of the Lee and Lee model (Figure 3.13) that happens around quality 0.25 is a result of the flow regime changing from ll to lv, and a different set of coefficients being used in their model. A similar phenomenon can be seen with the ammonia data in the Tu and Hrnjak model (Figure 3.12), where the model for the intermittent regime of ammonia can be seen to result in severe over-prediction of pressure gradient. Also from this figure, the Tu and Hrnjak (2004) separated flow model is seen to do well with R134a, and with low quality flows of R410A, but seen to over predict all the other cases. Overall, the homogeneous models demonstrated better predictive ability in the high quality regimes than the low quality regimes. This is surprising because the higher qualities correspond to the annular flow regime, which is not considered a homogeneous configuration.

Figure 3.11: Ratio of predicted two-phase pressure gradient by Dukler homogeneous model to measured pressure gradient.
Figure 3.12: Ratio of predicted two-phase pressure gradient by Tu and Hrnjak (2004) separated flow model to measured pressure gradient.

Similar plots were produced for variables other than quality, such as $Re_t$, $Re_v$, and $We_v$, in order to gain some small insight into what conditions are predicted well by the various models. It was possible to pick out the “different” fluids from these plots: ammonia typically would be poorly predicted even in cases that the other fluids were well-predicted, and the PVC channel with the smallest aspect ratio was more poorly predicted by at least one model than the other channels. By looking at the pressure gradient data – and the calculated quantities derived from that data – compared to the different flow variables, a new separated flow model was developed.

3.5 Separated flow model

3.5.1 Development of separated flow model

The separated flow methodology chosen to produce a pressure gradient model was to produce a correlation for the Chisholm interaction parameter, $C$. Following, Lee and Lee (2001), Tu and Hrnjak (2004), and Lee and Mudawar (2005), the form of the correlation was taken to be a product combination of the dimensionless parameters. However, to pick the relevant dimensionless parameters,
a extensive review of the data was undertaken. Since a correlation for the Chisholm interaction parameter, $C$, was desired, the values of $C$ calculated from the measured pressure gradient were plotted against various dimensionless parameters to try to determine which parameters exhibit the strongest effect on $C$. Recall that the dimensionless parameters that are based solely on fluid properties and/or geometry do not vary as the quality, $x$, changes. For example, the parameter $\lambda$, defined previously as:

$$\lambda = \frac{\mu^2}{\rho \sigma d_h} \quad (2.38)$$

is only dependent on the properties of the fluid and the channel geometry. The dependence of $C$ for the measured data on $\lambda$, shown in Figure 3.14, appears to be non-existent. The liquid-only parameters $Re_{lo}$ and $We_{lo}$ also have the property of remaining constant with increasing quality as long as the total mass flux, $G$, remains constant. A plot of $C$ vs. $Re_{lo}$ would then appear the same as Figure 3.14 (the data falling in vertical lines), as long as a the data set were taken at constant mass flux. As before, the solid data points are the new data with multiple fluids, the hollow data
points are Tu’s R134a data in multiple size channels.

Figure 3.14: The dimensionless parameter $\lambda$ has no variation with quality or mass flux

This observation is worth mentioning because several of the studies cited above correlated a functional dependence of $C$ on one or more of these parameters. When only these non-varying parameters are selected for the correlation, then that correlation could not have come up with a successful functional form for $C$. An example of this would be the correlation of Lee and Mudawar (2005) which took the general form:

$$C = a_1 Re_{lo}^{a_2} We_{lo}^{a_3}$$  \hspace{1cm} (2.66)

Including one or two of these constant parameters, along with other parameters that does have variation could work, since what is being sought in these correlations is a three or four dimensional surface. For example, if $\lambda$ and $Re_v$, the Reynolds number based on superficial vapor velocity, are plotted, the constant values of $\lambda$ spread the surface, as shown in Figure 3.15

The plots of $C$ vs. $Re_v$ and $C$ vs. $Re_l$ can be seen in Figures 3.16 and 3.17, respectively. From comparing these two plots, there appears to be a stronger dependence on the vapor Reynolds
number than on the liquid.

The characteristic velocity in the $\psi$ parameter (defined in Equation 2.39) can also be either based on the liquid or the vapor. In this case, the choices include $j_l$, the superficial liquid velocity, or following Suo and Griffith (1964), the bubble velocity, $U_B$. The plots of $\psi_{U_B}$ and $\psi_j$ variation on $C$ are shown in Figures 3.18 and 3.19, respectively. From appearances, the vapor velocity is again the more significant variable in the flow, since $\psi_{U_B}$. This could be explained by the fact that the vapor phase has higher velocities, and therefore could be considered to contribute more to the frictional losses than the liquid phase.

The three-dimensional plotting capabilities of Matlab to produce graphs that could be spun in the user interface allowed for many other parameters to be considered, in pairs. Resulting from this type of data review was the observation that the two parameters mentioned above, $Re_v$ and $\psi_{U_B}$, exhibited the strongest functional influence on the two-phase interaction parameter $C$. The
Figure 3.16: $C$ dependence on superficial vapor Reynolds number.

Table 3.4: Coefficients for new separated flow model (Equation 3.2).

<table>
<thead>
<tr>
<th>Flow regime</th>
<th>$\beta_1$</th>
<th>$\beta_2$</th>
<th>$\beta_3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface Tension</td>
<td>1.0077e-5</td>
<td>1.4591</td>
<td>-0.6428</td>
</tr>
<tr>
<td>$We_{vs} \leq 11.0 We_{ls}^{0.14}$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inertial</td>
<td>0.0146</td>
<td>0.4794</td>
<td>-0.6888</td>
</tr>
<tr>
<td>$We_{vs} &gt; 11.0 We_{ls}^{0.14}$</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A linear form of the correlation chosen:

$$C = \beta_1 Re^\beta_2 \psi^{\beta_3}$$  \hspace{1cm} (3.2)

The data were separated into inertial dominated and surface tension dominated flow regimes following the flow regime criteria of Akbar et al. (2003), which was the same flow regime criteria used by Tu and Hrnjak (2004). Regressing the data for the coefficients $\beta_1$, $\beta_2$, and $\beta_3$, resulted in the values shown in Table 3.4.
3.5.2 Testing the separated flow model

It should be noted that the uncertainty for the calculated $C$ values were not uniform across the board. Figure 3.20 is a plot of $C$ vs. $\psi_{U_B}$ with uncertainty in the calculation of $C$ indicated. The highest uncertainties correspond to the qualities that are near one. The one point with extremely large uncertainty on the far right of the plot corresponds to a quality of $x = 0.981 \pm 0.05$. For the pure refrigerant tested here, this is close to dryout.

When applying this new model to the measured data, the mean deviation (computed by Equation 3.1) was 12.6%, with 82.4% of the data falling ±20%. Figure 3.21 shows the percent difference between measured and predicted values using the new model as a function of quality. The overall predictive ability appears good although the higher qualities, corresponding to the inertia dominated flow regime, appear to be better predicted. Figure 3.22 shows the predicted pressure gradient with respect to the measured, in a form that does not convey the quality range limitations clearly.

Figure 3.17: $C$ dependence on superficial liquid Reynolds number.
Figure 3.18: Bubble velocity as characteristic velocity of $\psi$

3.6 Conclusions

A model for the Chisholm interaction parameter of two-phase flow for refrigerant in channels of hydraulic diameter 70 to 305 $\mu$m has been developed based on the Reynolds number of the vapor phase and the non-dimensional group $\psi U_B$. This model matches the measured data with a 12.6% mean deviation, across a wide variation of fluid properties, including density, viscosity, and surface tensions.

In contrast to previous models that have been developed for small channels, the parameters for the current model do not produce constant values for a given fluid and channel geometry. However, this model is still far from universal in application, and at the heart of this model, it is a correlation of known data based on the observed dependence of that data of the parameters. For this reason, it is certainly limited to the regime of microchannel flows where the only competing regimes are inertial and surface tension forces, and possibly even further limited to the ranges of fluids and channel sizes used in its development.
Figure 3.19: Superficial liquid velocity as characteristic velocity of $\psi$

Figure 3.20: Uncertainty in the calculated values of $C$
Figure 3.21: Ratio of predicted two-phase pressure gradient by the new separated flow model to measured pressure gradient.

Figure 3.22: New model prediction compared to measured pressure gradient
Chapter 4

Flow Visualization

4.1 Flow visualization facility

The facility for pressure drop was modified to accommodate flow visualization. Compression fittings were modified by countersinking 1.58 mm (1/16") pipes to the side of the fitting with small pressure tap holes drilled into the fitting. The fittings were used to attach a Pyrex glass tube with an inner diameter of 0.5 mm, using nylon ferrules to seal the edges. This glass tube with pressure taps directly replaced the aluminum channel that had been used for pressure drop measurements, with the same differential pressure transducer being used with the pressure taps. Unfortunately, the pressure taps were prior to the entry of the glass, and so the measured pressure difference across the glass tube included entrance and exit effects. To minimize the pressure drop due to entrance and exit effects, the diameters of the inlet pipe and the glass channel were matched, and the compression fittings were drilled to allow the end of the glass tube to come close to the inlet and outlet pipes. The visualization section is pictured in Figure 4.1. Everything else that made up the pressure drop facility, discussed in Chapter 3, was left the same.

A Sony Handicam was affixed with a macro lens and used to visualize flows inside the glass tube. A sheet of paper was placed behind the tube for contrast, and a stroboscope flashing at a high frequency (approximately 6000 Hz) aimed at the paper to illuminate the test section by reflection was used to freeze the images for the camera. Because of the image encoding of the Handicam, which was designed to produce a television signal, it was necessary to “deinterlace” the captured videos. This was done by means of a freely available software program called AviSynth, version 2.5.

Because of the low frame rate of the Handicam (24 fps) and the high speeds of the flows, successive frames from the captured videos were not able to capture fluid motion. As a consequence,
each of the frames was a separate photo that was unrelated to the others. Under some of the flow conditions, however, the images were clearly not statistically independent.

Flow visualizations were made at ambient temperature, which ranged from 21.6 to 24.0 °C over all of the tests. The mass flux for most of the refrigerant flow visualized was approximately \(335 \frac{\text{kg}}{\text{m}^2\text{s}}\), although one run with propane and one run with ammonia were visualized at approximately \(160 \frac{\text{kg}}{\text{m}^2\text{s}}\). This provided a limited range of superficial velocities and Weber numbers.

### 4.2 Flow regimes observed

From examination of the flow visualization videos, there were only four major flow regimes that could be identified: bubble-slug, slug, slug-annular and annular. The images of the different flow regimes match in appearance and description of the visualizations of R134a presented by Revellin et al. (2006), which were also performed in 0.5 mm channels. However, here distinctions are not made between “wavy annular” and “smooth annular”, “bubbly” and “bubbly/slug”, or “slug/semi-annular” and “semi-annular” as the were by Revellin et al. (2006). In the case of “slug/semi-
annular” and “semi-annular”, regimes demonstrating these characteristics were simply classified as “slug-annular”. While the visualization data herein represent only a limited range of flow conditions, the regimes observed include all of the regimes that have been seen in small channels.

A problem exists in the presentation of flow visualization data in such a way that can present the specifics and overall happenings of the flow at the same time. In order for the presentation to be exhaustive all of the unique features that were observed, no matter how infrequently, ought to be shown. However, this runs the danger that the presentation will not be representative of the overall flow, since the abnormal features that were observed infrequently are given as much visibility as the typical features. In an attempt to reduce the subjectivity of the flow regime classification, a weak statistical analysis was used to distinguish the regimes of slug, slug-annular and annular. A typical video from a given flow condition was around 5-6 seconds, or 300 de-interlaced images. In some of the images, the elongated vapor bubbles were longer than the field of view, which gave the temporary appearance of annular flow. For a flow condition to be classified as slug flow, less than 50% of the images could appear to be annular. For a flow condition to be classified as the slug-annular regime, between 50% and 95% of the images appeared annular, and if more than 95% of the images appeared annular, the condition was classified as annular.

For presentation in this section, four images were selected from each fluid for each flow regime. Within these images, an exhaustive presentation of the flow phenomena was attempted rather than a representative one. This is most important in the slug-annular flow regime, where at least one liquid slug was shown for each fluid even though the presence of slugs was infrequent in that flow regime, sometimes as few as one frame in three hundred for a given flow condition. In all of the images, gravity is downward and the flow is moving from left to right.

The descriptions of the flow regimes and representative photos for the flow regimes identified here for the three refrigerants are following:

**Bubble-slug flow** was observed at extremely low vapor qualities ($x < 0.1$). This is a surface tension dominated flow regime. Little bubbles, with diameters approximately half the channel diameter, trains of these little bubbles, and larger bubbles of vapor in the classical Taylor “bullet-shape” were observed. The liquid films on the sides of the bubbles were all smooth, although some instabilities were seen toward the back of the vapor bubbles, such as those on the left of
Figure 4.3(c). Occasionally, “lumps” or rings of liquid were seen within the vapor bubbles. The surface of the vapor-liquid around these lumps was always laminar in appearance. Although instabilities were sometimes seen on the sides of the bubbles there was no tail breakdown observed; the tails of the bubbles were flat although sometimes at an angle to the vertical. Some of the vapor bubbles were long, but they rarely took up the entire video frame, which was roughly 20-30 times the channel diameter. Some of the half-channel-diameter small bubbles, and even the medium-sized vapor bubbles, exhibited “spikes” on their trailing side. The half-channel-diameter bubbles had the spike on the side near the wall of the channel, and the bubbles that were equal in diameter to the channel had them visible on both the top and bottom. An illustration of these spikes can be seen in Figures 4.3(a) and (b), and a pair of bubbles with their spike pointing toward the nearest channel wall (up or down) is shown in Figure 4.4(d). These spikes on the tail end of the bubbles are also shown in the refrigerant visualization presented in Revellin et al. (2006) and also the air-water visualization of Triplett et al. (1999). In ammonia flow at extremely low quality ($x=0.022$, seen in Figure 4.5) almost-spherical bubbles were observed that were not deformed by moving through the liquid. This indicates that the difference between the bubble velocity and the bulk liquid velocity was small.

![Figure 4.2](image_url)

Figure 4.2: Four representative photos of Bubble-Slug flow of R134a, $x=0.081$, $G=336 \frac{kg}{m^2s}$
Figure 4.3: Four representative photos of Bubble-Slug flow of propane, $x=0.09$, $G=161 \ \text{kg/m}^2\text{s}$

Figure 4.4: Four representative photos of Bubble-Slug flow of ammonia, $x=0.076$, $G=166 \ \text{kg/m}^2\text{s}$

Figure 4.5: Four representative photos of Bubble flow of ammonia, $x=0.022$, $G=164 \ \text{kg/m}^2\text{s}$
**Slug flow** was the next flow regime that the refrigerant-vapor mixture assumed, as vapor quality increased. The slug flow regime is considered a surface tension dominated regime, because the vapor phase is impeded in the forward direction by vapor-liquid surfaces. Elongated bubbles were the primary flow pattern visible, with rounded noses and flattened tails. The vapor bubbles were mostly observed to be shorter than the length of the video camera frame, with less than 50% of the images from any given condition showing a bubble taking up the entire field of view. Figures 4.6(b) and 4.7(b) are examples of these elongated bubbles. Lumps or rings of liquid were observed within the vapor bubbles, such as those visible in Figures 4.6(b) and 4.8(a) and (b); a rippled vapor-liquid interface was observed within and around them, but only minimal disturbance of the surfaces of the vapor-liquid interface were visible elsewhere in the flow. No regularity or periodicity was observed with regard to the spacing of these rings. Various breakdowns of the tails were observed, such as that seen in Figure 4.7(c), including the phenomenon known as “skirting.” Some of the tails were still flat. For ammonia vapor and liquid, the tail breakdown was delayed further into the slug flow regime than for propane or R134a. No trains of small bubbles were observed in propane or R134a, but the little bubbles were seen preceding longer vapor bubbles, such as in Figures 4.6(d) and 4.7(d). In the ammonia flows, little bubbles were still observed, but trains of little bubbles were not observed in the slug flow regime.

![Figure 4.6: Four representative photos of Slug flow of R134a, x=0.140, G=333 kg/m²s](image)

Figure 4.6: Four representative photos of Slug flow of R134a, $x=0.140$, $G=333 \text{ kg/m}^2\text{s}$
Figure 4.7: Four representative photos of Slug flow of propane, $x=0.15$, $G=331 \text{ kg m}^2\text{s}^{-1}$

Figure 4.8: Four representative photos of Slug flow of ammonia, $x=0.028$, $G=337 \text{ kg m}^2\text{s}^{-1}$
**Slug-annular flow** was the next distinct configuration that the two-phase flow took as vapor quality increased. Slug-annular flow is considered to be in the inertial dominated regime of the flow, because of the predominance of vapor velocity that was evident from the liquid-vapor surface in the pictures. More than half of the images from the videos classified as slug-annular appeared to be annular, such as Figure 4.9(b). However more than 5% (and less than 50%) of the images contained some portion of a slug structure – noses or tails or combinations of the two – indicating that the liquid slugs did exist on occasion. Images similar to this have been classified “churn” flow by other researchers. Figures 4.9(a), 4.10(d) and 4.11(d) show the liquid slugs with nose and tail combination that was typical of this flow regime. There were frequently rings of thicker liquid observed on the walls, and the liquid-vapor surface was frequently rough, indicating that turbulence was likely present in the vapor phase. However, no particular regularity or periodicity was observed with regard to the spacing of these rings. Short vapor bubbles were rare but still present occasionally, such as the one seen in Figure 4.10(c). Tiny bubbles, much smaller than the channel diameter, were frequently observed in the wake of the longer bubbles for all of the refrigerants and can be seen in Figures 4.9(a), 4.10(c) and 4.11(d). There were also “breakthrough” lumps, which appeared to be locations where the nose of one vapor bubble caught up with the tail of the bubble ahead of it, breaking through the liquid slug that separated them and forming the appearance of a rivulet of vapor. One such a breakthrough lump can be seen half way along Figure 4.10(a). However, since the fluid motion was not captured in the current visualization technique, it can not be certain that this is what is occurring. The breakthrough lumps in particular have been observed in what was called the churn flow regime by other flow visualization studies.

![Figure 4.9: Four representative photos of Slug-Annular flow of R134a, x=0.231, G=336 kg/m²s](image)

Figure 4.9: Four representative photos of Slug-Annular flow of R134a, $x=0.231$, $G=336$ kg/m²s
Figure 4.10: Four representative photos of Slug-Annular flow of propane, $x=0.27$, $G=332 \ \text{kg/m}^2\text{s}$

Figure 4.11: Four representative photos of Slug-Annular flow of ammonia, $x=0.109$, $G=336 \ \text{kg/m}^2\text{s}$
**Annular flow** was the flow regime observed at the highest qualities, where the vapor region was continuous in the center of the channel and the liquid was a film around the wall of the channel. Annular flow is also an inertial dominated flow regime. Less than 5% of the frames from the video of conditions classified as annular exhibited noses or tails of bubbles, and the surface of the liquid film on the wall is rough most of the time indicating that the vapor flow was turbulent. Occasionally short smooth sections of liquid-vapor interface were be observed. When the entire liquid film had a rough surface, the data point was classified as “annular, turbulent”. There were occasional lumps of liquid that ring the wall, so the liquid film was not a constant thickness. No conclusive regularity or frequency could be observed in the spacing of those lumps in either the annular or the slug-annular regimes. Representative images from the annular flow regime can be seen in Figures 4.12, 4.13, and 4.14.

![Annular flow images](image1)

**Figure 4.12:** Four representative photos of Annular flow of R134a, $x=0.59$, $G=332 \ \text{kg/m}^2\text{s}$

![Annular flow images](image2)

**Figure 4.13:** Four representative photos of Annular flow of propane, $x=0.63$, $G=159 \ \text{kg/m}^2\text{s}$
4.3 Regime transitions observed

Since flow visualizations were only taken at a limited range of mass fluxes, approximately 160 and 335 kg/m²s, it can be illuminating to show constant mass flux for the various fluids. Figures 4.15 and 4.16 show representations of the flow regimes observed as vapor quality increases for the 160 and 335 kg/m²s flow rates, respectively. Note that Figure 4.19, which is discussed below, indicates the actual mass fluxes of the individual data points as well as the exact vapor quality by displaying them on mass flux vs. quality coordinates. The representations shown in Figures 4.15 and 4.16 are shown to make it easier to see where the regime boundaries are.

Figure 4.15 shows the propane and ammonia flow regimes at approximately 160 kg/m²s. No flow visualization at qualities higher than \( x = 0.43 \) was taken for ammonia, so the full-blown turbulent-annular flow regime was not observed. All of the flow regime transitions were found to occur at much earlier vapor qualities for ammonia than for propane. Although the surface tension of ammonia is much higher than that of propane, and thus it might be suspected that the surface tension dominated regimes would be extended, the liquid-to-vapor density ratio of ammonia is twice that of propane. This means that for a given vapor quality, there will be twice as much vapor volume present in an ammonia flow, and therefore the flow regime transitions could be expected at lower vapor qualities.

Figure 4.16 shows propane, R134a and ammonia at the higher mass flux, approximately 335 kg/m²s. The transition points for R134a and propane are roughly the same at this mass flux, and again
the transition points for ammonia occur at lower vapor qualities. Compared to the lower mass flux of Figure 4.15, the regime transitions for the higher mass flux occur at lower vapor qualities for both propane and ammonia, which is expected. Again, the entire range of vapor qualities was not investigated for ammonia, this was due to the size of the heater that controlled the vapor quality. The amount of heat required to fully vaporize this flow rate of ammonia was much more than the heater was able to provide.

4.4 Comparison to available flow maps

With these four flow regimes identified and then further classified into surface tension or inertial dominated flow regimes, a comparison was made to previously published flow maps that might be successful for predicting microchannel flow regimes. The flow maps developed in large channels, such as Baker, Mandhane, and Taitel-Dukler, which have been shown previously to be unsuccessful when applied to flows of refrigerant or flows in microchannels, were not considered here.

The flow regime transition of Akbar et al. (2003) between intermittent and annular type flows, which was used in the separated flow model from Chapter 3 and also by Tu and Hrnjak (2004) is shown overlaid on the data in Figure 4.17. This map uses as its axes the superficial Weber numbers,
$W_{ev}$ vs. $W_{el}$. The observed flow regimes from all three fluids occur earlier in the intermittent regime than predicted by Akbar’s map, with the most drastic difference being seen in the ammonia. From looking at the observed transitions of R134a and propane only, it appears that the superficial Weber number axes may be appropriate, if the transition is shifted downward. However, including ammonia into the comparison, the use of superficial Weber numbers for axes does not seem to work to collapse the multiple fluids into a single transition criteria.

Although Revellin et al. (2006) also observed that microchannel flows of refrigerant did not match the Kattan-Thome flow map, the present data is plotted in Figure 4.18 on $G$-$x$ axes. The transition lines for $x_{IA}$ from Hajal et al. (2003) for the vapor quality for transition between intermittent and annular flow, which was given in Equation 2.69. Since the transition lines are a function of fluid properties, each fluid has its own transition line, which is drawn in the matching color of the data points. The intermittent-annular transition is the relevant part of the Kattan-Thome-Favrat flow map for these flows, and the present data for none of the three fluids match the predictions. Note that for clarity only the 350 \( \frac{\text{kg}}{\text{m}^2\text{s}} \) data are shown, but the 160 \( \frac{\text{kg}}{\text{m}^2\text{s}} \) data do not match any better.
The transition criteria for intermittent to annular flow observed by Coleman and Garimella (2003), which was also cast in $G$-$x$ coordinates, is shown in Figure 4.19. Recall that this map was developed from visualizations of R134a, it can be observed that the R134a flow patterns match the predicted transition well. The propane flow transitions also match the predicted transitions, however the two-phase ammonia was observed to transition to annular flow at much lower vapor quality than Coleman and Garimella’s map predicts. Since the transition criteria of this map was determined by observation of only a single fluid, it is not surprising that the map does not predict well the regime transitions when applied to other fluids.

The flow map of Qu et al. (2004) was developed in small channels ($d_h = 0.677$ mm) with nitrogen-water flows. To develop their flow map took visualization data that were evenly spaced data points when using axes of superficial velocity. The transition lines they drew were halfway between the data points that were classified to be in different regimes. Comparing only the transition line from Qu et al. to the present data did not to satisfactorily predict transition. However when the data points observed by Qu et al. were overlaid onto the current data, the present data fell within the previously observed boundaries. Figure 4.20 shows the present data compared to
Figure 4.19: Present data compared to Coleman and Garimella (2000) flow map transition

the boundary points observed by Qu et al. (2004) from their reported map. The boundary points
that are here titled “intermittent” were originally classified into two regimes by Qu et al.: slug and
bubbly/slug.

The flow transition criteria of Zhao and Hu (2000) is compared to the present data in Figure 4.21.
The axes used for comparison are the parameter $W_v$, defined in Equation 2.76, and the square root
of the superficial liquid Weber number, which makes the transition line $y = x$, a curve on the
semi-log plot. This transition criteria was discussed previously and given in Equation 2.75. The
predicted transition is a bit early than the observed transitions for the flows of R134a and propane
and a bit late for the ammonia liquid-vapor flow. However, this model does have a reasonable
ability to predict the majority of the data points observed. The predictive ability of this map and
the mechanistic approach from which it was developed makes it an appealing candidate for flow
mapping.

Modification of the empirical parameters $\kappa$ and $C_0$, which were used in the development of
this map, was found to shift all of the data points in the same direction, so modification to these
constants can not produce a better fit with these transition criteria.
Qu et al. (2004) observed intermittent
Qu et al. (2004) observed annular

Figure 4.20: Present data compared to Qu et al. (2004) flow map transition

Zhao and Hu transition

Figure 4.21: Present data compared to Zhao and Hu (2000) flow map transition
4.5 Flow map analysis

While the flow map of Zhao and Hu (2000) demonstrated predictive ability for the observed data points, an analysis of other parameters was carried out in an attempt to see if parameters that had not been considered in other studies could be found to predict flow regime transition. Particular interest was taken in the energies involved in the flows, and both kinetic and surface tension energies were seen to be related to flow regimes.

Capillary numbers of the data points, defined in Equation 2.41, were used as map axes. The Capillary numbers were computed using both the liquid and vapor superficial velocities for the characteristic velocities of the Reynolds and Weber numbers, producing four combinations of $Ca$, which were plotted against: each other, the two superficial Reynolds numbers, and the two superficial Weber numbers. This technique covers the map produced in the analysis of Parang and Chao (1999). However, none of these parameters were able to separate the observed flow regimes into distinct regions in a way that would make them successful as flow map axes.

Next, consideration was made of the energies present in the phases. Kinetic energies, which were based on the superficial phase velocities, $j_v$ and $j_l$, were compared. The superficial kinetic energies in the phases are given by:

\[
KE_v = \frac{1}{2} \rho_v j_v^2 \\
KE_l = \frac{1}{2} \rho_l j_l^2
\]

Figure 4.22 shows the present flow regime data plotted with kinetic energies. Indicated with a solid line is $y = x$, where the kinetic energies in the vapor and liquid phases are equal, and indicated in a dotted line is $y = 3x$, the point at which the superficial kinetic energy in the vapor phase is 3 times larger than that of the liquid phase. Apparent from the graph is that the intermittent-to-annular transition occurs approximately when the superficial phase energies are equal, and certainly before the vapor phase has ten times the superficial kinetic energy of the liquid phase.

In examining various forms of energies present in the flow, another combination was found to be interesting. The energy product, $EP$, is the product of the specific superficial vapor kinetic
energy multiplied by the specific surface energy:

\[ EP = \rho v j^2 (\sigma/d_h) \quad (4.3) \]

The constant one half was left out of the kinetic energy. \( EP \) is shown plotted against the superficial liquid Weber number, \( We_l \), in Figure 4.23. Using these axes, it was possible to draw a line through the data to cleanly separate the data points that were observed to be in the intermittent and annular regimes. However, the energy product, \( EP \), is not dimensionless, having units of specific energy squared. It also failed to compare favorably with flow visualizations taken of CO\(_2\) in a related project. It is therefore not clear that the energy product is a universal flow regime identifying parameter.

### 4.6 Conclusions

Two-phase flows of refrigerants in a glass 0.5 mm channel have been visualized. Four distinct flow regimes have been identified: Bubble-Slug, Slug, Slug-Annular, and Annular. The first two of these
The observed data were compared to previous flow maps from the literature were, and the map by Qu et al. (2004) of observed N₂-water flows in small channels as well as the mechanistic analysis of micro-gravity flows presented by Zhao and Hu (2000) were both able to reasonably match the observed data. Others, including the map of Akbar et al. (2003), were not able to sufficiently predict the observed transitions. Of these flow maps that did reasonably match the observed data, Zhao and Hu (2000) was preferred, because of its mechanistic approach to the regime division. However, when compared to visualizations made of near-critical CO₂ flows in 0.5 mm glass tubes, this criteria does not predict as well as it does for the refrigerants studied here.

Analysis of the data was also undertaken, with the goal of identifying parameters that could be used for a new map. Many relevant dimensionless and dimensional parameters and combinations thereof were examined, and the energies present in the flows were found to provide the best matching of the flow regimes observed. The superficial kinetic energies of the two phases as well as the $EP$, the product of vapor kinetic energy and surface energy, were two parameters that seemed to contain
information important to the regime transition. Of these two options, the phase kinetic energies is preferred, because of the dimensional aspect of $EP$.

Given the limited selection of data to compare to, the classification of flow regime seems to be best matched with the observational criteria that the superficial vapor kinetic energy is 1 to 3 times greater than the superficial liquid kinetic energy. While it is not known that this criteria is fully universal, it has also been matched to the near-critical two-phase CO$_2$ visualization data successfully, and it contains some of the same mechanistic meaning that the analysis of Zhao and Hu (2000) does. Further work would need to include a wider span of flow visualization data in order to make more general statements about the universality of these proposed parameters.
Chapter 5

Refrigerant-Oil Flow

5.1 Property data of refrigerant-oil mixtures

The fluid properties and solubility of R134 and two POE oils were measured by Seeton and Hrnjak (2006). One oil was a POE32 and the second was a POE68. The density and viscosity are described in terms of the temperature of the mixture and the bulk composition, $\omega$, which was defined in Equation 2.78. The measured fluid properties were then correlated to the temperature and composition of the mixture in the manner described by Seeton and Hrnjak (2006) and Seeton (2006). The form of the density correlation is shown in Equation 5.1 with units on $T$ of Kelvin and units on $\rho$ of grams per cc. The form of the viscosity equation that was used for the R134a-POE32 combination is shown in Equation 5.2, with $T$ in Kelvin again and $\mu$ in centiPoise. For viscosity of the R134a-POE68 mixture, the kinematic viscosity, $\nu$, which is equal to $\mu/\rho$ was correlated, with the form of the equation shown in Equation 5.3. The nine coefficients that are needed for each correlation are shown in Table 5.1, for both the R134a-POE32 mixture and the R134a-POE68 mixture. These correlations were used in to quantify the the density and viscosity of the refrigerant-oil mixtures in this work.

$$\rho = a_1 + a_2 T + a_3 T^2 + \omega (a_4 + a_5 T + a_6 T^2) + \omega^2 (a_7 + a_8 T + a_9 T^2) \quad (5.1)$$

$$\log_{10} (\log_{10} (\mu)) = a_1 + a_2 \ln T + a_3 (\ln T)^2 + \omega (a_4 + a_5 \ln T + a_6 (\ln T)^2) + \omega^2 (a_7 + a_8 \ln T + a_9 (\ln T)^2) \quad (5.2)$$
Table 5.1: Coefficients in R134a-POE oil density and viscosity correlations, Equations 5.1, 5.2 and 5.3 for $\rho$ in g/cc $\mu$ in cP and $\nu$ in cSt

<table>
<thead>
<tr>
<th>Coefficients</th>
<th>$\mu$, POE32 in Eq. 5.2</th>
<th>$\rho$, POE32 in Eq. 5.1</th>
<th>$\nu$, POE68 in Eq. 5.3</th>
<th>$\rho$, POE68 in Eq. 5.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_1$</td>
<td>8.46137e00</td>
<td>1.24165e00</td>
<td>2.28818e01</td>
<td>-9.98306e-01</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-3.13824e00</td>
<td>-9.44674e-04</td>
<td>-3.66648e00</td>
<td>1.53781e-02</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-7.53085e-02</td>
<td>2.95688e-07</td>
<td>-1.13506e-02</td>
<td>-3.04969e-05</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-1.33290e02</td>
<td>3.64988e-01</td>
<td>-5.62250e00</td>
<td>8.55045e00</td>
</tr>
<tr>
<td>$a_5$</td>
<td>1.10267e02</td>
<td>-5.79118e-04</td>
<td>2.14571e01</td>
<td>-6.05100e-02</td>
</tr>
<tr>
<td>$a_6$</td>
<td>-2.29319e01</td>
<td>7.33590e-07</td>
<td>-2.11160e00</td>
<td>1.12643e-04</td>
</tr>
<tr>
<td>$a_7$</td>
<td>1.41890e02</td>
<td>9.35208e-02</td>
<td>2.38924e01</td>
<td>-6.61713e00</td>
</tr>
<tr>
<td>$a_8$</td>
<td>-1.15240e02</td>
<td>1.68089e-03</td>
<td>-1.18034e01</td>
<td>5.08400e-02</td>
</tr>
<tr>
<td>$a_9$</td>
<td>2.32502e02</td>
<td>-7.05531e-06</td>
<td>1.27920e00</td>
<td>-9.82240e-05</td>
</tr>
</tbody>
</table>

\[
\ln(\ln(\nu + 0.7 + e^{-\nu}K_0(\nu + 1.244068))) = a_1 + a_2 \ln T + a_3(\ln T)^2 
+ \omega (a_4 + a_5 \ln T + a_6(\ln T)^2)
+ \omega^2 (a_7 + a_8 \ln T + a_9(\ln T)^2)
\] (5.3)

In order to quantify the effect of oil concentration on the surface tension of the refrigerant-oil mixture, the surface tension test rig was used. The measurements of surface tension for the R134a-POE32 mixture were preformed by Chris Seeton, and the measurements of surface tension for the R134a-POE68 mixture was made by the present author. A description of the surface tension test facility and measurement technique can be found in Seeton and Hrnjak (2006). The surface tension was only measured in the room temperature region, because that was where the relevant pressure gradient and flow visualization experiments were conducted. A quadratic curve fit was made for $\sigma$ at room temperature, in units of milliNewtons per meter, as a function of composition, $\omega$, the form of which is shown in Equation 5.4. The coefficients to use in the correlation are shown in Table 5.2 for the two oils.

\[
\sigma = a_1 + a_2\omega + a_3\omega^2
\] (5.4)
Table 5.2: Coefficient values in the R134a-POE68 room temperature surface tension correlation, Equation 5.4 with $\sigma$ measured in mN/m

<table>
<thead>
<tr>
<th>Coefficient</th>
<th>$\sigma$, POE32</th>
<th>$\sigma$, POE68</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_1$</td>
<td>34.62</td>
<td>21.66</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-101.7</td>
<td>-24.14</td>
</tr>
<tr>
<td>$a_3$</td>
<td>83.46</td>
<td>11.58</td>
</tr>
</tbody>
</table>

Table 5.3: Oil concentrations used in R134a-POE32 tests

<table>
<thead>
<tr>
<th>$m_{oil}$ [g]</th>
<th>$m_{ref}$ [g]</th>
<th>OCR</th>
<th>Mass used [g]</th>
<th>Variation of OCR</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.82</td>
<td>808.17</td>
<td>0.47%</td>
<td>595</td>
<td>4.6%</td>
</tr>
<tr>
<td>12.40</td>
<td>798.04</td>
<td>1.53%</td>
<td>346</td>
<td>2.1%</td>
</tr>
<tr>
<td>21.22</td>
<td>777.09</td>
<td>2.66%</td>
<td>340</td>
<td>2.1%</td>
</tr>
<tr>
<td>43.59</td>
<td>797.13</td>
<td>5.18%</td>
<td>352</td>
<td>2.1%</td>
</tr>
</tbody>
</table>

5.1.1 Experimental facility and technique

For the measurement of two-phase pressure gradients of refrigerant and oil mixtures, the same system described in Section 3.1 was used, although the large reservoir tank was replaced by a smaller tank. In order to generate oil circulation, the refrigerant and oil were premixed in the reservoir tank to the appropriate proportions. The refrigerant-oil combinations were fully miscible at the test conditions, so it was assumed that the mixture leaving the reservoir tank was fully homogeneous. The internal volume of the reservoir tank was measured to be 863.8 mL, and the volume of the mixture was constantly recorded during any time the mixture was flowing so the vapor space at any point in the test could be calculated to correct the concentration if necessary. Tables 5.3 and 5.4 show the initial mass of oil and refrigerant that was mixed into the reservoir tank and the corresponding OCR value. In addition, the total mass of mixture used in all the tests is shown, along with the variation of OCR from the beginning to the end of the test. The variation is minimal for all of the cases, and thus no vapor space correction was deemed necessary.

It is worth mentioning that the calculation of vapor quality used to reduce data taken in the facility corresponds to the pseudo-superheated quality, $x_{psh}$, defined in Equation 2.82 and calculated from heat input and pure refrigerant properties. This is assumed to be equal to the thermodynamic quality.
Table 5.4: Oil concentrations used in R134a-POE68 tests

<table>
<thead>
<tr>
<th>$m_{oil}$ [g]</th>
<th>$m_{ref}$ [g]</th>
<th>OCR</th>
<th>Mass used [g]</th>
<th>Variation of OCR</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.98</td>
<td>819.80</td>
<td>0.48%</td>
<td>357.5</td>
<td>2.1%</td>
</tr>
<tr>
<td>11.39</td>
<td>738.05</td>
<td>1.52%</td>
<td>312.9</td>
<td>2.0%</td>
</tr>
<tr>
<td>23.17</td>
<td>751.00</td>
<td>2.99%</td>
<td>297.4</td>
<td>1.8%</td>
</tr>
<tr>
<td>39.44</td>
<td>726.51</td>
<td>5.15%</td>
<td>317.4</td>
<td>2.1%</td>
</tr>
</tbody>
</table>

5.1.2 Pressure gradient with low viscosity oil

The pressure gradient of the low viscosity oil combination, R134a-POE32, was tested first. The three mass fluxes tested were approximately 335, 450, and 600 $\text{kg/m}^2\text{s}$. As expected, an increase in mass flux caused an increase in pressure gradient. Compared in this section are the variation of pressure gradient with oil concentration for constant mass flux.

Figure 5.1, the data from the $G \approx 450 \text{ kg/m}^2\text{s}$ experiment, shows exactly what might be expected: an increasing pressure gradient with increasing oil concentration, with a greater effect seen at higher qualities. Higher quality implies a higher local oil concentration at the same OCR, as shown in Figure 2.14.
Figure 5.2: Two-phase pressure gradient for R134a-POE32, $G \approx 335 \, \frac{kg}{m^2\text{s}}$

Figure 5.2 shows the pressure gradient data from the $G \approx 335 \, \frac{kg}{m^2\text{s}}$ experiment. The same trend of increasing pressure gradient with increasing oil concentration can be seen at higher qualities ($x > 0.6$), but in the lower quality range, it is not as apparent.

Figure 5.3 shows the pressure gradient data from the $G \approx 600 \, \frac{kg}{m^2\text{s}}$ experiment. A weak dependence of pressure gradient on oil concentration can be observed, which is the strongest for the highest oil concentration, $OCR=5.18\%$, data.

When viewing these plots, it is necessary to keep in mind that the mass flux, $G$, was not entirely the constant value for every point. Thus, there are fluctuations in pressure gradients that result from fluctuations of mass flux. With the flow rates so small ($G = 450 \, \frac{kg}{m^2\text{s}}$ corresponds to a mass flow rate in this channel of $\dot{m}= 0.76$ grams per minute), exact control of the mass flux was a difficult task. Figure 5.4 shows the variations in the mass flux for the data shown. Since mass flux is a more significant contributor to the two-phase pressure gradient than the $OCR$, the variations in mass flux should be kept in mind when considering the pressure gradient as a function of quality plots. In particular, the $G \approx 450 \, \frac{kg}{m^2\text{s}}$ data with 0.47% $OCR$ was repeated twice with slightly different mass fluxes, which can be seen in Figure 5.1 as slightly higher pressure gradients.
Figure 5.3: Two-phase pressure gradient for R134a-POE32, $G \approx 600 \frac{kg}{m^2 \cdot s}$

Figure 5.4: Variation of $G$ in the experiments with R134a-POE32
It is also important to note that the tests were run in order of OCR, from least to greatest. The order of the test is significant because the oil from the previous experiments was not extensively cleaned out of the channel and system between runs with the same oil. That oil was left behind is be demonstrated by the pressure gradient plot in Figure 5.5 where after all of the POE32 data were collected, but before the channel was cleaned, an OCR of 0.59% was run. These data are shown as inverted purple triangles, which seem to fall closer in line with the 2.66% OCR pressure gradients when the 0.47%. It is apparent from these measurements that the high oil concentration mixture flowing in the channel contaminated the channel.

5.1.3 Pressure gradient with high viscosity oil

Next, after cleaning the channel and the entire system with several flushes of acetone, the higher viscosity POE68 oil was tested with R134a. The mass fluxes tested were approximately the same as before: 300, 450, and 600 $\text{kg/m}^2\text{sec}$. The same general trends for oil concentration and quality were seen for the higher viscosity oil as for the lower, so the results are compared across viscosities. In Figures 5.6 – 5.8 solid symbols are used for the R134a-POE68 pressure gradient data of higher
viscosity and hollow symbols are used for the lower viscosity R134a-POE32 data that were presented in the previous section, in Figures 5.1 – 5.3. Again, the fluctuations in mass flux between the individual points should be noted as these plots are examined; these variations are shown in Figure 5.9.

Figure 5.6 shows the two-phase pressure gradient for the low mass flux data. It is evident – and unsurprising – that the higher viscosity oil caused a higher pressure gradient than the lower viscosity oil. There also appeared to be an increased variation of pressure gradient with oil concentration than for the lower viscosity mixture.

Figures 5.7 and 5.8 show that at the medium and high mass fluxes that were tested, the two-phase pressure gradient is again higher for the higher viscosity oil. There is also a clear increase in two-phase pressure gradient with increasing OCR for the entire range of vapor quality. The mass flux was more consistent with the high viscosity oil, and this is perhaps the reason that the variation in pressure gradient with increasing OCR is more significant at the higher mass flux condition.
Figure 5.7: Two-phase pressure gradient for R134a-POE68, $G \approx 450 \text{ kg/m}^2\text{-sec}$

Figure 5.8: Two-phase pressure gradient for R134a-POE68, $G \approx 600 \text{ kg/m}^2\text{-sec}$
5.2 Comparison to separated flow model

Since the fluid properties of the refrigerant-oil mixtures were known as a function of oil concentration, the actual fluid properties could be applied into the separated flow model of Section 3.5 and compared to the measured two-phase pressure gradient. Equation 2.81 was applied in order to determine the local oil concentration, $\omega$, from the quality and $OCR$, and then the correlations for density, viscosity and surface tension were applied to the model parameters. The ratio of model prediction to measured pressure gradient can be seen for varying quality in Figure 5.10.

The separated flow model predicts the pressure gradient with 19.1% mean deviation, and 69.0% of the data are within $\pm25\%$. From examining the data, the higher $OCR$ flows seem to be most frequently under-predicted, and the lower $OCR$ flows seem to be most frequently over-predicted. This may indicate some sort of systematic error at work in applying the separated flow model to flows with refrigerant-oil mixtures. In addition, since the solubility of the refrigerant-oil mixture was not taken into account, there were two data points at high heat input and high $OCR$ that could not have fluid properties calculated because the calculation called for dry-out conditions. An improvement of the fluid property calculation routines would include correcting for the saturation
Figure 5.10: Ratio of predicted pressure gradient by separated flow model compared to measured pressure gradient in R134a-POE flows

Table 5.5: Oil concentrations used in R134a-POE68 flow visualization tests

<table>
<thead>
<tr>
<th>$m_{oil}$ [g]</th>
<th>$m_{ref}$ [g]</th>
<th>OCR</th>
<th>Mass used [g]</th>
<th>Variation of OCR</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.38</td>
<td>881.81</td>
<td>0.606%</td>
<td>586.6</td>
<td>3.8%</td>
</tr>
<tr>
<td>10.89</td>
<td>709.63</td>
<td>1.51%</td>
<td>518.1</td>
<td>4.4%</td>
</tr>
<tr>
<td>22.64</td>
<td>747.94</td>
<td>2.94%</td>
<td>453.0</td>
<td>3.2%</td>
</tr>
</tbody>
</table>

pressures of the refrigerant-oil mixture.

5.3 Flow visualization of refrigerant-oil mixtures

Refrigerant-oil mixtures were visualized in the same way that the flow visualizations of pure refrigerant were made. The refrigerant bottles were charged with a mixture of R134a and POE68 that corresponded to an OCR of 0.606%, 1.51%, and 2.94%. The same method of monitoring the variation in initial OCR was used, and the variation in the values of the oil circulation across the experiments can be seen to be minimal in Table 5.5.

The flow visualizations were made in the manner described in Section 4.1 and the regimes
observed were classified according to the method described in Section 4.2.

The flow visualizations of R134a were all done at approximately $330 \text{ kg m}^{-2} \text{s}^{-1}$, and with the larger diameter glass channel ($d_h=0.5 \text{ mm}$) this mass flux rate was not difficult to maintain constant, so all variation on flow regime boundaries were due to the oil concentration.

### 5.3.1 Flow regime transitions observed

Table 5.6 shows the quality ranges that were seen to be the transition boundaries in R134a and R134a-POE68. Figure 5.12 shows these same quality ranges in a graphical representation. The addition of oil shifts the bubble-slug to slug transition to earlier qualities, and oil in the refrigerant flow also seems to shift the slug to slug-annular transition to slightly lower qualities. An increase in OCR does not seem to have a significant effect on the flow regime transitions, and the slug-annular to annular transition was not affected either. The regime termed “turbulent-annular”, for which there were no longer smooth annular film observed on the images, seems to be affected by the increasing oil concentration. This could result from the increasing viscosity of the annular film leading to higher slip velocities between the vapor and liquid, and therefore more wavy structures.
Figure 5.12: Representation of the flow regimes for R134a-POE mixtures at $G \approx 330 \text{ kg m}^{-2}\text{s}$

Table 5.6: Quality ranges of regime transition boundaries for R134a/POE68, $G \approx 330 \text{ kg m}^{-2}\text{s}$

<table>
<thead>
<tr>
<th>Concentration</th>
<th>Bubble-slug/Slug</th>
<th>Slug/Slug-annular</th>
<th>Slug-annular/Annular</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pure R134a</td>
<td>0.098-0.103</td>
<td>0.160-0.195</td>
<td>0.260-0.340</td>
</tr>
<tr>
<td>0.606% oil</td>
<td>0.008-0.053</td>
<td>0.111-0.161</td>
<td>0.285-0.391</td>
</tr>
<tr>
<td>1.51% oil</td>
<td>0.008-0.044</td>
<td>0.111-0.133</td>
<td>&gt; 0.233</td>
</tr>
<tr>
<td>2.94% oil</td>
<td>0.023-0.063</td>
<td>0.128-0.155</td>
<td>0.267-0.351</td>
</tr>
</tbody>
</table>

on the surface of the liquid film, and therefore is less significant than the other transition regimes.

The flow visualization study of Bousman et al. (1996) found that variations in surface tension and viscosity affect the bubble to slug transition boundaries but not the slug to annular transition boundaries. This trend was confirmed in the data observed here.

The annular and turbulent annular flow regimes were similar in visual appearance to the flows of pure refrigerant. Figure 5.14, for example, shows an annular flow with an $x_{psh}$ of 0.507. However, dry-out was never observed in the refrigerant-oil flows. The 0.606% oil concentration was heated to a pseudo-superheated quality, $x_{psb}$, of 1.1 and still the flow regime was turbulent annular. The liquid film got thinner, from $x_{psb} = 1.044$ to $x_{psh} = 1.1$, but there is still clearly a liquid layer, in Figure 5.13. As discussed previously, some of the refrigerant is absorbed into the oil, and the saturation temperature of the mixture is dependent on $\omega$, the refrigerant-oil concentration. Therefore, as refrigerant is boiled out of the mixture, the temperature of the mixture is changed, and the pseudo-superheated condition results.

In the slug region, the inclusion of oil made a visible difference. The tails of the bubbles still
Figure 5.13: Turbulent-annular flow with apparent superheat: 0.606% oil in circulation, with $x_{psh} = 1.1$.

Figure 5.14: Turbulent-annular flow: 0.606% OCR, with $x_{psh} = 0.507$, same in appearance to pure refrigerant flows.

break down, and Figure 5.15 for an OCR of 1.51% and $x_{psh}$ of 0.200, with tiny bubbles in the wake of the slug getting caught up in the nose of the following slug could be a picture of pure refrigerant flow. There appeared to be higher density of small bubbles with refrigerant-oil mixtures than were apparent in the pure refrigerant flows. In addition some unique features were observed in the slug flow regime. The long bubbles were frequently divided by large annular lumps in the liquid annulus, sometimes even appearing to be a think film of liquid that separated the bubble into several bubbles. It is not clear in some of the images whether there are several little bubbles that are coming together or if the longer bubble is about to break into many smaller bubbles, because trains of small bubbles were also much more frequently observed with refrigerant-oil flows than with pure refrigerant. Four representative photos are seen in Figure 5.16, although the majority of that flow condition looks like typical slug flow.

The bubble-slug flow regime visualizations appear to be the same with oil as the pure refrigerant. The shifting of the bubble-slug/slug regime transition boundary is believed to be related to the

Figure 5.15: Slug flow: 1.51% OCR, with $x_{psh} = 0.200$, similar in appearance to pure refrigerant flows.

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strange phenomena of bubbles divided into segments. The mechanism behind this segmented structures is not apparent. Similar structures were observed in the water-N$_2$ flow visualizations of Chen et al. (2002), who classified them as “bubble-train slug flow”, and noted that: “typically two four bubbles, and possibly up to 10 or more, are in contact, like a train, with a clear interface between the connecting bubbles.” They note that this pattern was observed previously by Mishima and Hibiki (1996) who did not classify it separately from slug flow. Chen et al. (2002) also note that Zhao and Rezkallah (1993) observed a similar flow structure in micro-gravity visualizations, and classified the pattern “frothy slug slug.

5.3.2 Comparison to flow maps

An examination of the flow regime maps for the refrigerant-oil flows was made with the fluid properties of the liquid phase calculated from the mixture property correlations of Equations 5.1 through 5.4. Figure 5.17 shows the refrigerant-oil flows compared to the transitions observed by Qu et al. (2004). Just as Figure 4.20 showed that the regime transitions for the pure refrigerants fell within the observed transitions of the Nitrogen flows observed by Qu et al. (2004), the transitions of the refrigerant-oil mixtures also all fall within the range of the transitions.

From the pure refrigerant flow visualizations, the criteria of Zhao and Hu (2000) was found to
Figure 5.17: Refrigerant-oil observed flow regimes compared to Qu et al. (2004) flow map

have good predictive ability for regime transitions. Figure 5.18 shows the refrigerant-oil flows in relation to this criteria, which was defined previously in Equation 2.75. As it does for the pure refrigerant data, which was shown in Figure 4.21, this mechanistic criteria shows good predictive ability for the transition between the surface tension dominated and inertial dominated regimes.

The superficial phase kinetic energies, $KE_v$ and $KE_l$, defined in Equations 4.1 and 4.2, were used in Figure 4.22 to identify regime transitions for pure refrigerants. What was shown what that most of the pure refrigerants transitioned flow regime in the region where $KE_v$ was between 1 and 3 times that of $KE_l$. Figure 5.19 shows that the same holds true for the flows of refrigerant and oil.

From the analysis of the pure refrigerant flow maps, it was shown in Figure 4.23 that the energy product, $EP$, defined in Equation 4.3, could be plotted against the superficial liquid Weber number to distinguish the flow regimes. However, in Figure 5.20, these same quantities are plotted, and the observed line of transition from the pure refrigerant data does not match the refrigerant-oil flow regime transitions.
Figure 5.18: Refrigerant-oil observed flow regimes compared to Zhao and Hu (2000) criteria

Figure 5.19: Superficial kinetic energies of the refrigerant-oil phases used for flow map
5.4 Conclusions

Refrigerant R134a and POE oil were tested in small channels. Pressure gradient measurements and flow visualizations were made of the mixtures. The oil, however, has the most significant effect on pressure gradient at higher vapor qualities. Correlations for the mixture properties were applied to the refrigerant-oil flows, in order to calculate the local fluid properties of the data. The separated flow pressure gradient model was applied to the mixtures with reasonable success. Flow visualization showed that the oil affects the transition between the bubble-slug and slug regimes at low vapor qualities, and has a slight effect on the slug to slug-annular regime transition. The mechanism of the transition boundary shifting is thought to be related to the segmented flow structures that were observed in the slug flow regime. The presence of tiny bubbles in the wake of large vapor bubbles seems to be much more prevalent in the refrigerant-oil flows than the flows of pure refrigerant. Examination of the flow regime maps reveals that the same criteria which had success for pure refrigerants also appear successful for predicting the transition of the refrigerant-oil flows.
Chapter 6
Mechanistic Model

6.1 Model description

The relative simplicity of the flow regimes present in small channel two-phase flow makes appealing a mechanistic approach to modeling the pressure drop. Since the only major flow regimes present are slug and annular flow, a simplified model considers these to be the only regimes present and neglects certain phenomena – such as tail break-up and tiny bubbles entrained in the slugs – that were observed in the experiments. In this model, assumptions were made about the geometry and velocity profiles that allow the determination of pressure drop in the channel. By balancing the flow rates and pressure drops with the assumed geometry and velocity profiles, a model of the pressure drop has been developed based on the two flow regimes. No empirical correlation parameters were used in the model that would make the predictions fit the data.

The model is developed in the geometry of a round channel, and then it is applied to the data taken in the non-round channels. Non-geometric parameters such as hydraulic diameter were used in the development to increase the applicability of the model, however this assumption may limit the model applicability in channels of high aspect ratios. Subsequent developments would be to incorporate the effects of channel shape and aspect ratio into account.

6.1.1 Annular regime

The simpler of the two flow regimes is the annular regime, because it is uniform in the streamwise direction. There are two regions of the flow, the liquid annulus, which is considered to be of uniform thickness, and the vapor core. No vapor bubbles are allowed to be within the liquid film, and the liquid-vapor interface is considered smooth. The pressure at each streamwise cross-section is assumed to be uniform, meaning there is no radial dependence on pressure.
The schematic representation of the annular flow regime is shown in Figure 6.1.

The velocity profile in the liquid annulus is assumed to be that of an one-dimensional, annular Couette-Poiseuille profile, since the vapor shear and the overall pressure drop both drive the flow. The form of the annular velocity profile is given as:

\[
\frac{u_l(r)}{u_i} = \frac{\ln \frac{r}{r_v}}{\ln \frac{r_v}{r_v}} - \frac{dP}{dz} \left( \frac{r_h^2 - r_v^2}{4 \mu_l} + \left( r_h^2 - r_v^2 \right) \frac{\ln \frac{r_h}{r_v}}{\ln \frac{r_v}{r_v}} \right)
\] (6.1)

where \( u_i \) is the velocity of the liquid-vapor interface.

By balancing the frictional losses in the vapor phase with the shear in the liquid phase at the interface, \( \tau = \mu_l \frac{du}{dr} \bigg|_{r_v} \), the interface velocity could be determined to be:

\[
u_i = -\frac{dP}{dz} \left( r_h^2 - r_v^2 \right)
\] (6.2)

which, when substituted into Equation 6.1, simplifies the expression for the velocity in the annulus:

\[
u_l(r) = \frac{dP}{dz} \left( r_h^2 - r^2 \right)
\] (6.3)

The vapor phase, which was bounded by the liquid-vapor interface, was assumed to have a turbulent velocity profile:

\[
u_v(r) = u_o \left( 1 - \frac{r}{r_v} \right)^{1/7} + u_i
\] (6.4)

where \( u_o \) is the peak centerline velocity of the turbulent vapor.

The volume flow rates of liquid and vapor, \( \dot{Q}_l \) and \( \dot{Q}_v \), are determined from the mass flow rates, as given in Equations 2.12 and 2.13. The liquid and vapor mass flow rates, \( \dot{m}_l \) and \( \dot{m}_v \), in
turn can be determined from the vapor quality, $x$, and total refrigerant flow rate, $\dot{m}$, as given in Equations 2.6 and 2.7. By dividing the volume flow rates for each phase by the cross-sectional area that the phase takes up, an average velocity for each phase can be determined:

$$\dot{Q}_l = \frac{V_l}{\pi (r_h^2 - r_v^2)}$$ (6.5)

$$\dot{Q}_v = \frac{V_v}{\pi r_v^2}$$ (6.6)

At the same time, the volumetric flow rates can be determined from an area integration of the assumed velocity profile. In cylindrical coordinates the volume flow rate of liquid, $\dot{Q}_l$, takes the following form:

$$\dot{Q}_l = \int_{r_v}^{r_h} u_l(r) 2\pi r dr = -\frac{dP}{dz} \frac{\pi}{8\mu_l} \left( r_h^4 - 2r_h^2 r_v^2 + r_v^4 \right)$$ (6.7)

The only unknowns in this equation are the overall pressure gradient, $-\frac{dP}{dz}$, and the radial location of the liquid-vapor interface, $r_v$. Note that the film thickness, $\delta$, can be related to the given geometry by: $\delta = r_h - r_v$.

A similar integration for the vapor can be performed, but for the single-phase pressure drop of a turbulent flow the Blasius equation can be used to determine the pressure gradient:

$$f_{Blasius} = \frac{0.316}{Re^{1/4}}$$ (6.8)

which requires a characteristic velocity for both the Reynolds number and the friction factor. Since the velocity of the vapor is relative to the liquid layer surrounding it, this characteristic velocity is taken as: $V_{char} = V_v - V_l$. This results in an expression for the pressure gradient in the vapor core of:

$$\frac{dP}{dz} = \frac{0.3164 \rho_v (V_v - V_l)^2}{4 r_v}$$ Re_{vap}^{1/4}$$ (6.9)

where:

$$Re_{vap} = \frac{\rho_v (V_v - V_l)(2r_v)}{\mu_l}$$ (6.10)

Between Equations 6.7 and 6.9, the only unknowns are $r_v$ and $-\frac{dP}{dz}$. These two equations, along with the other supporting expressions, can be solved simultaneously in EES to determine the
6.1.2 Slug regime

Slug flow is a flow regime that presents unique challenges to model. Since it is not uniform in the streamwise direction, the analyses of two-phase flow that assumed uniform streamwise geometry (Lockhart and Martinelli, 1949) are not strictly valid. The standard way of treating this non-uniformity is to assume periodicity in the geometry, where the flow consists of a repeating series of “unit cells” made of a vapor bubble and a liquid plug paired together. This approach is the one taken in all the models previously described in Section 2.2.2. In particular, this means that each cell contains the appropriate mass flows of vapor and liquid to make up the vapor quality, $x$, of the flow. It is further assumed that the vapor bubble is surrounded by a liquid film and the walls remain wetted at all times. Since the flows tested here were adiabatic, this was confirmed by the flow visualizations: dry walls were never observed around a vapor bubble.

The schematic of the assumed slug flow geometry and the unit cell is shown in Figure 6.2. The liquid slug was assumed to have a fully developed, one-dimensional laminar velocity profile far from the bubble, denoted by $u_{\text{slug}}(r)$. This is consistent with the Reynolds numbers of the liquid phase, which all indicate laminar liquid flow for the experimental conditions. The peak centerline velocity of the laminar profile is considered $u_o$, and the mean velocity of the liquid slug is half of that:

$$V_{\text{slug}} = \frac{u_o}{2}$$  \hspace{1cm} (6.11)

The vapor bubble is assumed to move at a uniform velocity, $u_v$. It is important to note that this does still allow for circulation of vapor within the bubble, a phenomenon that commonly observed
in rising bubbles, but that the interface of the bubble is moving together at a fixed velocity, \(u_v\), and therefore the shape of the bubble is not changing in time as the bubble moves down the channel.

The length of the unit cell is designated by \(L_{\text{unit}}\), and the length of the bubble is designated by \(L_{\text{bubble}}\). According to the computational work of Kreutzer et al. (2005), once the slug length is 50 times the channel diameter, the effect of the bubble on the overall pressure gradient can be ignored. Accordingly, the unit cell length was set in this model to be \(50d_h\), since when the liquid slug is longer, a single-phase pressure gradient can be assumed.

The length of the bubble, \(L_{\text{bubble}}\), is determined based on the quantity of vapor and liquid that is flowing in the channel. The hydraulic radius, \(r_h\), is equal to one half of the hydraulic diameter. The film thickness, \(\delta\), which is again the difference between the hydraulic radius and the bubble radius, \(\delta = r_h - r_v\), unlike the annular regime however the film only surrounds the bubble. The bubble radius, \(r_v\), was calculated in the model by Garimella et al. (2002), where it was found to vary only minimally; throughout the entire range, \(r_v\) remained within the range \(0.899 \leq r_v/r_h \leq 0.911\). Chung and Kawaji (2004) in their model assumed that the radius of the bubble was 90% of the channel radius. Following those two studies, the present model assumes \(r_v/r_h = 0.90\), or \(\delta = 0.10r_h\) for every condition.

Because the vapor phase has a much lower density and viscosity than the liquid, the pressure within the bubble can be considered constant. This implies that the streamwise pressure gradient within the liquid film is zero, and that the liquid film is not moving as the bubble moves past. This is consistent with computational study of Kreutzer et al. (2005) which shows the detailed pressure distribution along a horizontally flowing bubble-slug combination. It is also indicated by the experimental studies that have directly measured wall shear stress of upward flowing Taylor bubbles and slug flows, (Nakoryakov et al., 1986; Mao and Dukler, 1991) that showed a location of zero shear stress shortly after the nose region of the passing bubble. Since a gravitational field exists opposing the upward flow, the film region becomes a falling film once the upward pressure gradient is removed.

A control volume has been sketched onto the unit cell, and the control volume is considered to be moving at the vapor velocity, \(u_v\). This means that the bubble remains fixed with respect to the control volume while the walls move to the left at a velocity \(-u_v\). The liquid in the film moves
backward at \(-u_v\), and the liquid in the slug moves backward with a velocity profile \(u_{\text{slug}}(r) - u_v\).

A sketch of just the moving control volume is shown in Figure 6.3.

The centerline velocity of the slug is now equal to \(u_o - u_v\), which in principle can be greater or less than zero, depending on the amount of liquid that bypasses the bubble.

A momentum balance can be performed on the control volume given in Figure 6.3; since the velocity profiles on either side of the control volume are identical, the only surviving terms are the pressure on both sides and the wall shear along the channel walls. This can be reduced to:

\[
\frac{\Delta P}{L_{\text{unit}}} = -\frac{dP}{dz} = \frac{2\tau_w}{r_h}
\]

(6.12)

The wall stress represented as \(\tau_w\) is actually an integrated value along the entire interior wall of the control volume, and the negative sign simply means that the positive pressure gradient is in the upstream direction. There are three distinct regions of the liquid flow within the control volume that all have different quantities of wall shear: the region upstream (to the right) of the nose of the bubble, the region of the bubble and film and the region downstream (to the left) of the tail of the bubble. The wall shear can be split into three components:

\[
\Delta P = \frac{2\tau_{us}}{r_h}L_{us} + \frac{2\tau_{film}}{r_h}L_{film} + \frac{2\tau_{ds}}{r_h}L_{ds}
\]

(6.13)

In the film region, if the velocity is uniform and equal to the wall velocity, \(-u_v\), the wall shear is zero:

\[
\tau_{film} = 0
\]

(6.14)
Therefore, the regions before the nose and after the tail of the bubble produce the entirety of the wall shear and the resulting pressure gradient along the unit cell.

At both ends of the control volume, the velocity profile is laminar and therefore the wall shear opposing the flow is known:

\[ \tau_{\text{lam}} = \frac{2u_o \mu_l}{r_h} \]  

(6.15)

For some length before and after, the wall shear must be thus. Between these laminar regions and the bubble, there are transition regions in which the velocity profiles are not known, and in these regions the local wall shear must be considerably higher than the laminar shear given in Equation 6.15. The upstream and downstream wall shears can be split into laminar and transitional regions:

\[ \Delta P = \frac{2\tau_{\text{lam}}}{r_h} L_{\text{lam},us} + \frac{2\tau_{\text{trans},us}}{r_h} L_{\text{trans},us} + \frac{2\tau_{\text{lam}}}{r_h} L_{\text{lam},ds} + \frac{2\tau_{\text{trans},ds}}{r_h} L_{\text{trans},ds} \]  

(6.16)

The lengths \( L_{\text{lam},us}, L_{\text{trans},us}, L_{\text{lam},ds}, \) and \( L_{\text{trans},ds} \) are not known, but it is known that the overall wall shear in the transitional region upstream and downstream must be greater than the laminar wall shear. Therefore, the overall pressure drop can be simplified by representing it as the sum of pressure drop from a laminar pipe flow plus the “excess” pressure drops arising from the transitional velocity profiles in the nose and tail regions:

\[ \Delta P_{\text{cv}} = \frac{2\tau_{\text{lam}}}{r_h} L_{\text{slug}} + \Delta P'_{\text{nose}} + \Delta P'_{\text{tail}} \]  

(6.17)

A control volume of region upstream of the nose of the bubble is shown in Figure 6.4. Performing a mass balance on this control volume, the relation between \( u_o \) and \( u_v \) can be determined to be:

\[ u_v \left( r_h^2 - \frac{r_n^2}{2} \right) - \frac{3}{4} u_o r_n^2 = 0 \]  

(6.18)

which, for \( r_v = 0.90r_h \), reduces to \( u_v = 1.26u_o \). So, \( u_o - u_v \) is less than zero, and the centerline velocity is in the negative direction as indicated in Figure 6.3.

A momentum balance on this control volume will include momentum terms from the velocity
The momentum terms on the right hand side of Equation 6.19 will be equal and opposite to the momentum terms that arise from a control volume analysis of the tail region (shown below in Figure 6.5), and therefore do not contribute to $\tau_w$, the integrated wall stress for the entire unit cell control volume. Velocity profiles of the nose region of upward-moving Taylor bubbles that have been reported in numerous experimental and computational studies, (Nogueira et al., 2006; Kreutzer et al., 2005; Polonsky et al., 1999; Thulasidas et al., 1997), indicate that the nose region is not highly disturbed flow. It is known that within the control volume in Figure 6.4, the pressure difference across the curved interface of the nose of the bubble on the left which is given by the Young-Laplace equation, Equation 2.2. For small radii of curvature, this pressure difference can become significant, and since the curvature is set up by the flowing liquid, it must be balanced by the wall shear in the region. Therefore, the excess pressure difference (in excess of the laminar pipe flow) in the nose region is approximated as:

$$\Delta P'_{nose} = \frac{2\sigma}{r_v}$$  

(6.20)
Figure 6.5: Control volume of the region downstream of the bubble

and the average shear stress in the region before the nose is then given by:

\[
\frac{2\tau_{us}}{r_h} L_{us} = \frac{2\sigma}{r_v} + \frac{2\tau_{lam}}{r_h} L_{us} + \rho_l \left( u_v^2 \frac{r_v^2}{r_h^2} - u_o u_v + \frac{1}{3} u_o^2 \right)
\]  

(6.21)

Since the liquid in the film is not moving with respect to the wall, the shear losses in that region are zero, and the final region to consider is the region downstream of the tail of the bubble. A control volume surrounding the region after the tail is shown in Figure 6.5. A momentum balance on this region yields:

\[
P_1 - P_B - \frac{2\tau_{ds}}{r_h} L_{ds} = -\rho_l \left( u_v^2 \frac{r_v^2}{r_h^2} - u_o u_v + \frac{1}{3} u_o^2 \right)
\]  

(6.22)

Since the right hand sides of Equations 6.19 and 6.22 are equal and opposite, when added back into Equation 6.13 they will subtract from each other.

The flow in the region behind the tail of the bubble much more highly disturbed than the flow in the nose region. The wake region of the bubble can cause strong recirculation, as observed in experimental and computational studies (Kreutzer et al., 2005; Giavedoni and Saita, 1999). Although the exact form of the average wall shear in this region is not known, a consideration of the geometry of the flow can yield an approximation of the average wall shear in this region.

This geometric consideration of the flow in this region starts from an examination of the flow in the stationary reference frame. Figure 6.6(a) shows the velocity profiles at the beginning and end
Figure 6.6: (a) Velocity profiles for the region following the bubble in the stationary reference frame. (b) Velocity profiles for a developing flow entrance region of this region in the stationary reference frame. Immediately following the bubble, on the right of the figure, the velocity profile is uniform in the center and zero in the film region surrounding the bubble. On the left side of the figure, the velocity profile is that of a fully-developed laminar flow. Figure 6.6(b) shows these velocity profiles reversed. Since the velocity profiles are simply reversed, a momentum balance on either region will result in equal wall stresses for both configurations. The purpose of this reversal is that the configuration represented in Figure 6.6(b) strongly resembles that of an entrance length of pipe flow. Strictly speaking there is a small expansion and then an entrance length, however, the expansion ratio is 10% which contributes only a negligible amount to the minor losses from a pipe flow, and therefore will be neglected here.

If the wall shear in the wake region of the bubble can be considered to be equal to the wall shear in a sharp entrance of a pipe flow, the framework of wall shear that is in excess of that of laminar flow is a helpful one, because the developing region of a pipe flow is given in terms of a minor loss coefficient, which has no length associated with it, but is the pressure drop due to the developing flow in excess of the frictional losses in that pipe. The minor loss coefficient given by White (2008) for a sharp entrance length is 0.5, which is given in terms of the mean velocity of the
flow. This leads to an excess pressure difference across this control volume is given as:

$$\Delta P'_{tail} = 0.5 \frac{1}{2} \rho_l V_{slug}^2$$

(6.23)

This pressure difference is then applied to Equation 6.22, yielding:

$$\frac{2\tau_{ds}}{r_h} L_{ds} = 0.25 \rho_l V_{slug}^2 + \frac{2\tau_{lam}}{r_h} L_{ds} - \rho_l \left( u_{v}^2 \frac{r_v^2}{r_h} - u_o u_v + \frac{1}{3} u_o^2 \right)$$

(6.24)

When Equations 6.14, 6.21 and 6.24 are applied back into the respective wall shears expressed in Equation 6.13 and then divided by the overall length to produce a pressure gradient, the result is:

$$\frac{\Delta P}{L_{unit}} = -\frac{dP}{dz} = \frac{1}{L_{unit}} \left( \frac{2\tau_{lam}}{r_h} L_{slug} + \frac{2\sigma}{r_v} + 0.25 \rho_l V_{slug}^2 \right)$$

(6.25)

The mean slug velocity, $V_{slug}$ is unknown. Equation 6.11 relates this to the peak slug velocity, which is in turn related to the average vapor velocity, $u_v$, by Equation 6.18. The average vapor velocity is related to the volume flow rate of vapor by the fraction of the unit cell that the vapor bubble takes up:

$$\dot{Q}_v = u_v \pi r_o^2 \frac{L_{bubble}}{L_{unit}}$$

(6.26)

The liquid flow rate is given by the flow in the slug times the fraction of the unit cell that the slug occupies:

$$\dot{Q}_l = Q_{slug} \frac{L_{slug}}{L_{unit}} = \frac{\pi}{2} u_o r_h^2 \frac{L_{slug}}{L_{unit}}$$

(6.27)

Again $\dot{Q}_v$ and $\dot{Q}_l$ are known from the overall mass flow rate and vapor quality, as given in Equations 2.12 and 2.13. This closes the system, and allows Equation 6.25 to predict the pressure gradient across the entire unit cell for the slug regime.

### 6.1.3 Flow regime determination

The geometries and pressure drop model for the two regimes being thusly defined, it is important to determine where each is applicable. From the flow visualizations of Chapter 4, it was determined that the regime map of Zhao and Hu (2000) best predicted the separation between surface
tension dominated flows and inertia dominated flows. This model, which was discussed in detail in Section 2.3.2, predicts transition to annular flow for:

$$\sqrt{We} v \geq W_v$$  \hspace{1cm} (2.75)

where:

$$W_v = \sqrt{\frac{4\kappa C_0 \sqrt{\alpha(1-\alpha)}}{C_0 - 1}}$$  \hspace{1cm} (2.76)

and void fraction was calculated from:

$$\frac{j_v}{j_v + j_l} = \alpha C_0$$  \hspace{1cm} (2.74)

with $C_0 = 1.16$ and $\kappa = 0.8$.

This was the transition criteria employed to determine the transition between the slug and annular flow regimes.

### 6.2 Model results compared to experiment

#### 6.2.1 Pure refrigerant

The pressure gradient prediction in the various regimes being defined and the flow regimes being determined by the model, the model was applied to the data discussed in Chapter 3. Figure 6.7 shows the ratio of the model predictions to the measured data.

The overall mean deviation of the mechanistic model was 18.1%, and 74.6% of the data were found within $\pm 25\%$. This is not outstanding in terms of accuracy, in fact, the separated flow model from Chapter 3 had higher accuracy. However, this model does not rely on the specification of empirical parameters by curve fits in order to match the measured data, and therefore has the potential of more universal application.

#### 6.2.2 Refrigerant-oil mixtures

The mechanistic model was used to predict the pressure gradients in the refrigerant-oil flows that were discussed in Chapter 5. Using the fluid property correlations given in Equations 5.1 – 5.4,
the fluid properties for the data were computed and used in the equations of the model. Figure 6.8 shows the predictive ability of the two-phase flows of refrigerant and oil.

The overall mean deviation of the refrigerant-oil predictions was 18.9%, with 78.8% of the data falling within ±25%. The predictive ability of the mechanistic model for the refrigerant-oil flows is the same as it was the pure refrigerant, and slightly better than the predictive ability of the separated flow model, which was shown in Figure 5.10.

### 6.3 Conclusions

A mechanistic model based on assumed geometries and consisting only of slug/plug or annular flow regimes was developed. The same data set as before, was used to compare to this model. No empirical correlation parameters were used in the construction of this model, but simplified geometries and velocity profiles were assumed. The model matched the measured pure refrigerant data with an 18.1% mean deviation and 74.6% of the data within ±25%, and the refrigerant-oil data with an 18.9% mean deviation and 78.8% of the data falling within ±25%. While this is not
Figure 6.8: Ratio of predicted two-phase pressure gradient of refrigerant-oil mixture by mechanistic model to measured pressure gradient

astoundingly accurate, this is a reasonable predictive ability.

Since this model has the potential to be applicable in microchannel flows, the continuation of this model will be to try to apply it to pressure drop data that has been taken in other facilities. It is unknown whether or not the maldistribution effects arising from parallel channels and/or diabatic conditions will be well-represented by the simplifications presented in this model.

An improvement to this model would be to account for the difference between the circular and rectangular geometries. The model was developed with circular geometry, but the data were collected in rectangular geometries. Both slug and annular flow in rectangular channels are slightly more complex than in circular channels, and it would be instructive to make the necessary modifications to the equations to correct for this difference.

A final improvement to this model could be made with regard to the determination of flow regime in the small channels. The criteria from Zhao and Hu (2000) could either be improved upon, or replaced, if better flow regime transition criteria can be developed.
Chapter 7

Conclusions

In this work, flows of refrigerant and refrigerant-oil mixtures in small channels have been considered. Pressure gradients of multiple refrigerants in a small channel have been measured and represent a wide span of fluid properties. These data were combined with two-phase pressure gradient measurements that were made of a single refrigerant in small channels of different sizes. This combined data set was used to produce a new two-phase pressure gradient correlation based on the separated flow construction.

Two-phase pressure gradients of flows of R134a and two different POE oils were measured in the same channel that was used for the multiple fluids. The observed effects of increased liquid viscosity was as expected namely, that the increases in viscosity, quality, and mass flux increased the two-phase pressure gradient. Using correlations for the fluid properties of the refrigerant-oil mixtures that were based on local oil concentration, the new separated flow model was applied to these pressure drop measurements. This model did not evidence great predictive ability of the oil mixture flows, and an examination of the prediction error seems to indicate a systematic offset based on oil content.

Flow visualizations of multiple pure refrigerants and refrigerant-oil mixtures were also run. The two-phase flow regime map that was developed by Zhao and Hu (2000) matched most closely with the observed flow regime transitions. The ratio of superficial kinetic energy within the phases of the flow, as well as a comparison of the kinetic energy multiplied by an approximation of the surface energy within the flow both somewhat matched the observed transition criteria, but a wider span of fluid properties and flow data would be needed to make definitive conclusions about those parameters.

There were two significant differences observed in the flow visualizations between the pure refrigerant and the refrigerant-oil mixtures. The first was that the inclusion of oil did not allow
for a dry-out, even at vapor qualities that were calculated to be higher than 1. This was because the saturation pressure and temperature change as the oil content changes, and is also well known as “apparent superheat.” The other difference when oil was added was in the appearance of the slugs in the refrigerant-oil flows. The slugs and bubbles appeared to be either breaking apart or coalescing in a manner not seen in pure refrigerant. This has been observed in other studies, but never fully explained. It lead to a shifting of the bubble-slug to slug transition for flows with oil.

Finally, a mechanistic model for describing pressure drop was developed. Based on a series of geometrical simplifications for the two main flow regimes, slug and annular, it determined flow regime by the aforementioned criteria of Zhao and Hu (2000). Then, it assumed simplified velocity profiles for the two regimes and used those velocity profiles to determine the pressure gradient within the channel. Compared to the measured pressure gradient of pure refrigerant flows, the mechanistic model predictions had a mean deviation of 18.1%, with 74.6% of the data being found within ±25% of the prediction. Using the fluid properties calculated for the refrigerant-oil mixtures, the mechanistic model had a mean deviation of 18.9%, with 78.8% of the data being found within ±25%. This is not taken as a resounding success for the mechanistic model and there are several improvements that should be made, including a better flow regime transition model and also a correction in the velocity profiles for rectangular channels.
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