Flow and Heat Transfer in Microchannels
30 to 300 Microns in Hydraulic Diameter

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Abstract

Fluid flow and heat transfer in microchannels are very important in various fields, including miniature thermal systems, compact heat exchangers and electronics cooling. The purpose of this investigation is to contribute to this new research area through carefully designed and well-controlled experimental work. Experiments were performed to characterize single-phase and two-phase flow pressure drop in rectangular microchannels with \( D_h = 69.5 - 304.7 \ \mu m \). In general, the measured single-phase friction factors agree with the analytical solution based on the Navier-Stokes equations, and the critical Reynolds numbers approach the conventional values. Based on the adiabatic two-phase flow pressure drop data, a new correlation was developed, in which the parameter \( C \) in Lockhart-Martinelli type correlation was correlated as a function of three nondimensional parameters in two flow regimes. Adiabatic flow was visualized in a microchannel, and the result suggests a wider range of surface tension dominated flow in microchannels than in minichannels. R134a liquid superheats before the onset of nucleate boiling and the local evaporation heat transfer coefficient were measured in \( 75 \times 811.94 \ \mu m \) microchannels. No indication of deviation from the classic nucleation theory was observed, but the analysis showed that the lack of large range of various sized ‘active’ nucleation sites in microchannels resulted in higher than usual liquid superheats. The heat transfer data suggest a combined effect of nucleation and convection, as well as an earlier dry-out in microchannels. The comparison of the data with existing evaporation heat transfer correlations identified a need for developing new correlations in microchannels. Based on the present data, a semi-empirical correlation was proposed. Microscale orifice-tubes with inner diameter \( D = 31 - 52 \ \mu m \) were tested with R134a flowing with and without a phase change. No indication of choking was observed for a wide range of experimental conditions, and this phenomenon was attributed to the strengthening of metastable effect in microscale tubes. A semi-empirical correlation was proposed based on the experimental data.
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pool boiling heat transfer coefficient, W/m²°C

two-phase flow heat transfer coefficient, W/m²°C

two-phase heat transfer coefficient in the convective boiling dominant region, Equation (7-20), W/m²°C

two-phase heat transfer coefficient in the nucleate boiling dominant region, Equation (7-19), W/m²°C

current, Amps

latent heat of vaporization, J/kg

liquid slug velocity, m/s

Boltzmann’s constant (1.380662×10⁻²³), J/K

thermal conductivity, W/m°C

liquid thermal conductivity, W/m°C

parameter in Equation (5-35)

length, m

orifice-tube length over diameter ratio

entrance length, m

exit length, m

total channel length, m

mass flow rate, kg/s

number of measurements

molecular weight

number of molecular per unit volume, /m³

number of data points

confinement number, defined in Equation (5-20)

heat flux correction term in Gorenflo (1990, 1993) correlation, Equation (7-28)

constant in Equation (5-17)

Nusselt number

pressure, N/m²

critical pressure, N/m²

vapor pressure inside the nucleus, Pa

pressure in liquid space, Pa

reduced pressure p/pₚcrit

Prandtl number for liquid, Equation (7-14)

saturation pressure, N/m²

evaporation pressure, N/m²

the total heat generation the heater, W

heat flux, W/m²

minimum heat flux for ONB, W/m²

reference heat flux in Gorenflo (1990, 1993) correlation, Equation (7-28), W/m²

orifice radius

constant in Equation (5-31)

arithmetic average surface roughness, m

reference surface roughness in Gorenflo (1990, 1993) correlation, Equation(7-28), m

critical bubble radius, m

Reynolds number
Re orifice flow Reynolds number, Equation (8-2)
Re_{eqq} equivalent mass flux, Equation (5-23)
Re_{GS} Reynolds number based on U_{GS}, Equation (4-5)
Re_{lo} Reynolds number assuming total flow to be liquid, Equation (7-21)
Re_{LS} Reynolds number based on U_{LS}, Equation (4-4)
Re_{tp} two-phase flow Reynolds number, Equation (5-39)
Re_{tp} two-phase flow Reynolds number in Chen (1966) correlation, Equation (7-7)
Rp maximum peak value of roughness, m
Rv minimum valley value of roughness, m
s constant in Equation (5-31)
S suppression factor for nucleate boiling
T temperature, °C
T_g superheated liquid temperature, K
T_{sat} saturation temperature in Equation (4-9), K
T_{sh} limit of superheat of the liquid, °C
T_w inner wall temperature, °C
U, u fluid velocity, m/s
UB velocity of the gas bubbles, Equation (5-51), m/s
U_{GS} superficial velocity of vapor (gas) phase, m/s
U_{LS} superficial velocity of liquid phase, m/s
V voltage, Volts
v fluid velocity, m/s
v_l, v_v liquid specific volume at saturation, m^3/kg
v_v, v_l vapor specific volume at saturation, m^3/kg
v_{lv} difference in specific volumes of saturated liquid and vapor, m^3/kg
W channel width, m
We Weber number defined in Equation (5-18)
We_{GS} Weber number defined in Equation (4-3)
We_{LS} Weber number defined in Equation (4-2)
x vapor quality
X Martinelli parameter defined in Equation (5-3)
x_{crit} dry-out quality
z/Dh hydraulic entrance length, dimensionless

**Greek Symbols**

α channel aspect ratio, H/W
α void fraction
β ratio of orifice diameter to conduit diameter in Equation (8-1)
β area ratio in Equation(2-17)
ΔG(r*) maximum free energy, J
ΔT_{ONB} wall superheat for Onset of Nucleate Boiling (ONB), °C
ΔT_{sub} subcooling, °C
ΔT_{sat} wall superheat, °C
Δx change of vapor quality
ΔP, Δp pressure drop, N/m²
ΔP_{sat} change in vapor pressure corresponding to temperature change ΔT_{sat}, N/m²
δ_{wall}  
wall thickness, m
γ  
liquid-only-vapor-only Martinelli parameter, defined in Equation (5-8)
φ_t  
two-phase frictional multiplier in Equation(5-1)
φ_o  
two-phase frictional multiplier in Equation (5-6)
λ  
nondimensional number, Equation (5-33)
λ_c  
collision frequency, defined in Equation (6-4), /s
μ  
viscosity, kg/m·s
μ_{avg}  
average viscosity of homogeneous fluid, kg/m·s
ρ  
density, kg/m³
ρ_{avg}  
average density of homogeneous fluid, kg/m³
σ  
standard deviation
σ  
surface tension, N/m
σ  
the area ratio, defined in Equation (2-14)
ψ  
nondimensional number, Equation (5-49)
ψ  
nondimensional number, Equation (5-32)
ψ_s  
separated flow multiplier

**Subscripts**
A  
acceleration
a  
aluminum
b  
between, web
bottom  
channel bottom
bulk  
bulk fluid
c  
critical
ch1  
channel #1
channel  
of or relating to channel
con  
contraction
con  
convection
conv  
conventional results
e  
ext
eff  
effective
eh  
main heating section
ei  
test section inlet
eo  
test section outlet
exp  
expansion
f  
frictional
H  
height
header  
of or relating to header
in, i  
inlet
l  
liquid phase
lo  
all flow as liquid
m  
mean
nb  
nucleate boiling
oi  
orifice inlet (upstream)
oo  
orifice outlet (downstream)
out  
outlet, exit
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>pb</td>
<td>pool boiling</td>
</tr>
<tr>
<td>pre</td>
<td>pre-evaporator</td>
</tr>
<tr>
<td>r</td>
<td>refrigerant</td>
</tr>
<tr>
<td>s</td>
<td>saturation</td>
</tr>
<tr>
<td>sat</td>
<td>saturation</td>
</tr>
<tr>
<td>sub</td>
<td>subcooling</td>
</tr>
<tr>
<td>tot, t</td>
<td>total</td>
</tr>
<tr>
<td>tp</td>
<td>two-phase</td>
</tr>
<tr>
<td>v</td>
<td>vaporization</td>
</tr>
<tr>
<td>v</td>
<td>saturated vapor</td>
</tr>
<tr>
<td>W</td>
<td>width</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
<tr>
<td>wi</td>
<td>inner wall</td>
</tr>
<tr>
<td>wo</td>
<td>outer wall</td>
</tr>
</tbody>
</table>
Chapter 1. Introduction

1.1 Thermal System Miniaturization

During the past decades, there has been a growing interest to develop miniature thermal systems/devices. For example, Ashraf et al. (1999) explored the design of a mesoscale refrigerator to be fabricated from layers of silicon wafers through techniques of microelectronics fabrication. The primary application was for the cooling of electronics under hot ambient conditions. The outside diameter of the overall system was limited to 75 mm. Brooks et al. (1999) developed a mesoscale combustor/evaporator that provides a lightweight and compact source of heating, cooling, or energy generation for both man-portable and stationary applications. The microchannels in the combustor and evaporator had dimensions from 50 to 200 microns across. Drost and Friedrich (1997) designed a miniature absorption heat pump that had dimensions of 9 cm × 9 cm × 6 cm and cooling capacity of 350 W. The heat pump used microchannel heat exchangers with channel widths between 100 and 300 microns and channel depths less than 1 mm.

Ashraf et al. (1999) summarized the advantages of miniaturizing thermal systems, which are listed as follows. (1) Both thermal and chemical transport processes rely on surface area of heat exchangers or reactors. Hence, higher surface-to-volume ratio of a miniature system helps to make a more compact system with a higher volumetric transport coefficient. (2) Smaller size typically leads to better safety. (3) Use of micro-fabrication technology is expected to provide better cost efficiency. (4) Miniaturization makes newer engineering systems possible. (5) Smaller modular size leads to more options in usage.

1.2 Mesoscopic Cooling System Project at UIUC

At the University of Illinois at Urbana-Champaign, Shannon et al. (1999) were working on the Integrated Mesoscopic Cooler Circuits (IMCCs) project. The goal of the IMCCs project was to develop small, mesoscale, vapor-compression refrigeration systems at the scale of 100 mm square and 2.5 mm thick. As shown in Figure 1-1, the idea was to put many of these inter-connectable IMCC patches together and make an IMCC vest, which can be used for the cooling of military personnel on active duty in hot desert climates or other hazardous environments where personal microclimate control is required.

Figure 1-1 Integrated IMCC vest and a single IMCC patch
Each patch has been designed to produce 3 Watts of cooling capacity while operating between 20 °C (evaporation temperature) and 50 °C (condensation temperature). Each IMCC patch used two flexible microchannel heat exchangers with 54 parallel channels as the condenser and the evaporator, as shown in Figure 1-2, and an orifice-tube with a diameter of 30 ~ 50 µm as the expansion device. The microchannels have cross-section geometry of 400 ~ 800 µm wide and 50 ~ 75 µm high. The manufacturing process of this type of heat exchangers has been described in Selby et al. (2001). Refrigerant 134a was chosen as the working fluid for the system.

![Figure 1-2 IMCC heat exchanger](image)

**Figure 1-2 IMCC heat exchanger**

### 1.3 Object of the Current Study

There is no widely accepted unique hydraulic diameter separating microscale ducts from the conventional scale. Even more, there are ongoing debates on physical rationale for classification based on tube hydraulic diameter. Industry uses the term microchannel for anything smaller than 1 mm in hydraulic diameter, Kandlikar (2003) suggested using “minichannel” for $200 < D_h < 3,000$ µm, and “microchannel” for $10 < D_h < 200$ µm while Mehendale et al., 2000 proposed $1 < D_h < 6$ mm, $100 < D_h < 1,000$ µm and $1 < D_h < 100$ µm for “compact heat exchanger”, “meso- heat exchanger” and “micro- heat exchanger”, respectively. We will adopt the following nomenclature in this document: microchannel with $D_h$ close to or less than 100 µm; mini-channel with $D_h \approx 1$ mm; and macrochannel with $Dh >> 1$ mm.

The design and optimization of miniaturized thermal systems require the clarification of fluid dynamics and heat transfer issues in ducts with inner diameters close to or less than 100 µm. In addition, fluid flow and heat transfer in micro geometries are very important in other fields, such as compact heat exchangers, electronic cooling, bioengineering and biotechnology, aerospace, MEMS (Micro Electro-Mechanical System), material processing and thin film deposition technologies, etc. (Peng and Wang 1993, Stanley et al. 1997, Tuckerman 1984, Mehendale et al. 2000, and Ho and Tai 1996).
Complete review of research on microscale fluid flow and heat transfer can be referred to Ho and Tai (1996), Gad-el-Hak (1999), Abramson and Tien (1999), Mehendale et al. (2000), Ghiaasiaan and Abdel-Khalik (2001) and Kandlikar (2002). Figure 1-3 summarizes the fluid flow and heat transfer studies in microchannels and minichannels, according to the author’s incomplete collection of references. The literature reviews will be introduced in Chapter 3 to Chapter 8. The figure demonstrates that there are growing interests of research in this new area during the past decades. In April 2003, the First International Conference on Microchannels and Minichannels was held in Rochester, New York, USA. Nevertheless, there is still a lack of work in this area. For the existing work, large scatters and even contradictions still exist.

The main goal of the current work is to contribute to this new research area and to resolve some of the issues through carefully designed and controlled experiments. The work started as part of the IMCCs project. Eventually, the design and optimization of the mesoscopic refrigeration system motivated us to go further and to conduct series of experiments to characterize refrigerant flow and heat transfer through microchannels. The specific objectives of the current work were to:

- Investigate whether correlations developed in macroscale extend to the size of channels in the range (30 – 300 µm).
- Explore the effect of aspect ratio (H/W) and channel diameter on fully developed single-phase flow friction factors in rectangular microchannels with hydraulic diameters close to or less than 100 microns over a wide range of Reynolds numbers (laminar and turbulent). The experimental data will be compared with the conventional equations.
• Develop a new correlation for adiabatic two-phase flow pressure drop in microchannels, using experimental measurements of adiabatic two-phase flow pressure drop in microchannels with hydraulic diameter close to or less than 100 microns for a wide range of mass fluxes and vapor qualities. The experimental data will be compared with the existing correlations. The study will be assisted with visualization of two-phase flow in microchannels to understand the flow behavior.

• Investigate the effect of mass flux, vapor quality and heat flux on evaporation heat transfer in a rectangular microchannel with a hydraulic diameter close to 100 microns. Compare the experimental results with existing correlations and suggest usage in microchannels.

• Investigate the incipience of nucleation conditions for subcooled liquid flow boiling in microchannels.

• Characterize R134a flow though microscale orifice-tubes (D = 30 ~ 52 µm) with and without phase changes. The experimental results will be used to develop a correlation for micro-scale orifice-tubes.

1.4 Structure of the Documentation

In Chapter 2, the experimental methods for the current study are discussed. It gives a detailed description of the test sections, experimental facilities, instrumentation, data reduction methods and uncertainty analysis for single-phase pressure drop, adiabatic two-phase pressure drop, evaporation heat transfer, and microscale orifice-tube studies.

The present study is divided into the following six topics: (1) single-phase pressure drop, (2) adiabatic two-phase flow visualization, (3) adiabatic two-phase pressure drop, (4) liquid flow boiling, (5) evaporation heat transfer and (6) flow though orifice-tube. These topics are discussed in Chapter 3 through Chapter 8. Each chapter presents the relevant literature on the topic, analysis of the experimental results and the comparison of the results with existing correlations. New correlations for two-phase flow pressure drop in microchannels, evaporation heat transfer in microchannels and refrigerant flow though microscale orifice-tubes are developed and presented in Chapter 5, Chapter 7, and Chapter 8, respectively.

References


Chapter 2. Experimental Methods

2.1 Chapter Overview
This Chapter describes the experimental methods for the current study. It is divided into four sections. Each section presents the experimental facility, test section, instrumentation, data reduction and uncertainty analysis for different topics in the current study. The single-phase pressure drop part, section 2.2, has the most complete descriptions of the R134a flow system and instrumentation.

2.2 Single-phase Flow Pressure Drop

2.2.1 Test Section
Five microchannel test sections with hydraulic diameters varying from 69.5 to 304.7 µm and with aspect ratios changing from 0.09 to 0.24 were designed and manufactured for single-phase investigation. These test sections were also used in the adiabatic two-phase pressure drop studies. In order to avoid the problem of maldistribution that is typically associated with a multiple-channel heat exchanger, single-channel test sections were designed and employed in the current study. The cross-section of such a channel is illustrated in Figure 2-1. The test section consisted of two parts, a substrate on which a channel was machined and a cover plate. Both parts used a 6.4 mm thick clear PVC block. Using the transparent material enabled the same test section to be used for flow visualization in the future.

![Figure 2-1 Cross-section drawing of a microchannel test section used in pressure drop studies](image1)

The surfaces of the cover plate and the substrate were very smooth (Ra < 20 nm) and flat. As a result, the test sections could be sealed easily by just bolting two surfaces together with balanced forces. As shown in Figure 2-2, the spacing between the bolts was about 6 mm. This design is better than other methods of sealing such as O-ring...
or gasket, because it eliminated the possibilities of introducing an unknown gap between two surfaces. It also provided a well-controlled surface structure that is missed in other methods such as bonding. In addition, this design allowed easy disassembling and re-assembling of the test section; thus, the channel can be cleaned and the geometry can be measured anytime without damage.

As can be seen from Figure 2-2, refrigerant flowed in and out of the channel through circular ducts with diameters equal to the channel width. These tubes were perpendicular to the channel and the entrance to the channel was abrupt. Two identical pressure tap holes, each with a diameter of less than 200 µm, were drilled in the center of the channel bottom, enabling direct measurement of the frictional pressure drops. The holes were checked under a microscope to make sure there were no burs around the edges.

Table 2-1 Geometric parameters of the test sections

<table>
<thead>
<tr>
<th>Test section</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>H (µm)</td>
<td>167.9</td>
<td>93.2</td>
<td>76.6</td>
<td>60.9</td>
<td>37.9</td>
</tr>
<tr>
<td>W (µm)</td>
<td>1646.7</td>
<td>383.5</td>
<td>889.2</td>
<td>359.2</td>
<td>416.5</td>
</tr>
<tr>
<td>H/W</td>
<td>0.10</td>
<td>0.24</td>
<td>0.09</td>
<td>0.17</td>
<td>0.09</td>
</tr>
<tr>
<td>Dh (µm)</td>
<td>304.7</td>
<td>150.0</td>
<td>141.1</td>
<td>104.1</td>
<td>69.5</td>
</tr>
<tr>
<td>L (mm)</td>
<td>96.0</td>
<td>48.0</td>
<td>48.0</td>
<td>48.0</td>
<td>48.0</td>
</tr>
<tr>
<td>L_e (mm)</td>
<td>28.0</td>
<td>14.0</td>
<td>14.0</td>
<td>14.0</td>
<td>14.0</td>
</tr>
<tr>
<td>L (mm)</td>
<td>40.0</td>
<td>20.0</td>
<td>20.0</td>
<td>20.0</td>
<td>20.0</td>
</tr>
<tr>
<td>L_e/Dh</td>
<td>92</td>
<td>93</td>
<td>99</td>
<td>135</td>
<td>203</td>
</tr>
<tr>
<td>L/L_e</td>
<td>315</td>
<td>320</td>
<td>340</td>
<td>460</td>
<td>691</td>
</tr>
</tbody>
</table>

The geometric parameters of the test sections are listed in Table 2-1. The length between the inlet and the first pressure tap hole, the entrance length (L_e/Dh), was selected such that the flow between the two pressure tap holes was fully developed. Since no guidance is available with regard to a proper entrance length for microchannels, the conventional results were used as a reference. For a fully developed turbulent flow, Obot (1988) suggests an entrance length, L_e/Dh, of 60. For laminar flow, the hydraulic entrance length (z/Dh) has been determined analytically and is found to be a linear function of the Reynolds number.

\[
\frac{z}{D_h} = C_e \text{Re} \tag{2-1}
\]

where C_e is a constant. For Reynolds numbers less than 2,000, Hartnett et al. (1962) found that C_e to be 0.033 and 0.046, for rectangular channels of aspect ratio of 0.1 and 0.2, respectively. For Reynolds numbers larger than 2,000, the flow is not completely laminar and the hydraulic entrance is less than the peak value at Re = 2,000. Thus, for an aspect ratio of 0.2, the hydraulic entrance length over the entire Reynolds number is z/Dh ≤ 92, which suggests that an entrance length (L_e/Dh) of about 92 would suffice for the current investigation. As seen from Table 2-1, this requirement was met for all the five test sections.
The channel lengths were measured with a Nikon MM-11 measurescope. The channel height and width were determined with a Sloan Dektak ST stylus surface profilometer. Cross-section profile scans were performed across the microchannel at different locations along the length. Each scan generated 8,000 data points at an interval of 0.25 µm, which created plots as shown in Figure 2-3. The figure shows that the channel sidewalls are not perpendicular to the bottom surface. This is not the real shape of the channel; instead, the reason for the slant is that the stylus tip is installed on a triangular block. In reality, it is taken as a right angle according to the manufacturing technique used.

Table 2-2 Channel height measurement

<table>
<thead>
<tr>
<th>Test section</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean ( H ) (µm)</td>
<td>167.9</td>
<td>93.2</td>
<td>76.6</td>
<td>60.9</td>
<td>37.9</td>
</tr>
<tr>
<td>Std. Dev., ( \sigma_H ) (µm)</td>
<td>1.1</td>
<td>1.3</td>
<td>0.4</td>
<td>0.4</td>
<td>1.0</td>
</tr>
<tr>
<td>Maximum ( H ) (µm)</td>
<td>169.8</td>
<td>95.7</td>
<td>77.4</td>
<td>61.4</td>
<td>39.4</td>
</tr>
<tr>
<td>Minimum ( H ) (µm)</td>
<td>166.2</td>
<td>91.4</td>
<td>76.1</td>
<td>60.4</td>
<td>36.4</td>
</tr>
<tr>
<td>No. of data</td>
<td>19</td>
<td>19</td>
<td>10</td>
<td>7</td>
<td>10</td>
</tr>
</tbody>
</table>

Table 2-3 Channel width measurement

<table>
<thead>
<tr>
<th>Test section</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean ( W ) (µm)</td>
<td>1646.7</td>
<td>383.5</td>
<td>889.2</td>
<td>359.2</td>
<td>416.5</td>
</tr>
<tr>
<td>Std. Dev., ( \sigma_W ) (µm)</td>
<td>13.6</td>
<td>5.6</td>
<td>5.9</td>
<td>3.4</td>
<td>3.6</td>
</tr>
<tr>
<td>Maximum ( W ) (µm)</td>
<td>1676</td>
<td>392</td>
<td>898</td>
<td>364</td>
<td>420</td>
</tr>
<tr>
<td>Minimum ( W ) (µm)</td>
<td>1624</td>
<td>375</td>
<td>881</td>
<td>354</td>
<td>413</td>
</tr>
<tr>
<td>No. of data</td>
<td>19</td>
<td>11</td>
<td>21</td>
<td>7</td>
<td>10</td>
</tr>
</tbody>
</table>

The surface profilometer has the capability of measuring step height down to a few nm. Hence, each scan could generate highly accurate data of the channel geometry at a specific location, and the channel height and width could be determined very accurately. However, the channel topography may not be the same at different locations,
therefore, several measurements were taken and the averaged values were used. Such a result as well as the statistical data is listed in Table 2-2 and Table 2-3. It shows that the microchannel cross-section geometry was quite consistent at different locations, and the channel could be considered as a straight duct. In order to check the repeatability of the average height (H) and average width (W), two separate groups of measurements were performed for test section #2 and #3 after the pressure drop experiment. The results were compared with the values in Table 2-2 and Table 2-3, and the differences were within ± 0.7%, for both H and W.

The Sloan Dektak\textsuperscript{3} ST stylus surface profilometer was also used to characterize the channel surface roughness. As shown in Figure 2-4, the channel cover plates were very smooth with a mean roughness of \( \text{Ra} = 0.02 \mu\text{m} \), but the channel bottom surfaces were much rougher, as shown in Figure 2-5. Therefore, the microchannels had non-symmetric surface characteristics. The channel bottom roughness is reported as the arithmetic average surface roughness (Ra), the maximum peak value (Rp), as well as the minimum valley value (Rv) in Table 2-4

Table 2-4 Channel bottom roughness measurement

<table>
<thead>
<tr>
<th>Test section</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra ((\mu\text{m}))</td>
<td>0.48</td>
<td>0.21</td>
<td>0.48</td>
<td>0.30</td>
<td>0.21</td>
</tr>
<tr>
<td>Rp ((\mu\text{m}))</td>
<td>2.3</td>
<td>0.8</td>
<td>2.4</td>
<td>1.5</td>
<td>0.77</td>
</tr>
<tr>
<td>Rv ((\mu\text{m}))</td>
<td>2.4</td>
<td>0.7</td>
<td>2.0</td>
<td>1.1</td>
<td>1.10</td>
</tr>
<tr>
<td>Ra/(D_h)</td>
<td>0.16%</td>
<td>0.14%</td>
<td>0.35%</td>
<td>0.29%</td>
<td>0.30%</td>
</tr>
<tr>
<td>2Ra/H</td>
<td>0.57%</td>
<td>0.45%</td>
<td>1.25%</td>
<td>0.98%</td>
<td>1.11%</td>
</tr>
</tbody>
</table>

Figure 2-4 Cover plate surface profile
Figure 2-5 Channel bottom surface profiles

2.2.2 Experimental Facility
The facility for the single-phase pressure drop study is shown in Figure 2-6. Because the flow rate in microchannels is inherently small, the apparatus was designed as a once-through system for simplicity and flow stability. A tank containing refrigerant was heated with an electric resistance heater. A variable transformer was used to control the heating power, which in turn determined the saturation pressure of the refrigerant. The high-pressure refrigerant was driven from the bottom of the tank into the test loop as saturated liquid, which was cooled down to room temperature while flowing through the tubing to the test section.
Figure 2-6 Experimental facility for single-phase test

Once leaving the refrigerant tank, the refrigerant flowed through two mass flow meters, one for larger flow rates (Micromotion model CMF010, \( m_2 \) in Figure 2-6), and another for smaller flow rates (Rheotherm® model TU1/16, \( m_1 \) in Figure 2-6). A filter with a mesh size of 0.5 \( \mu \text{m} \) was installed before the flow meters to protect them and the test section from dust particles. Because the Rheotherm® flow meter is based on liquid flow energy balance, a subcooler and a sight-glass was installed to make sure subcooled liquid entered the flow meter.

For the vapor tests, a receiver tank was placed into an ice-water bath, which provided a stable lower pressure. After the flow meters, the refrigerant flowed through a metering valve to control the flow rate. The refrigerant expanded in the metering valve, and turned into low pressure (or low temperature) two-phase flow. For most of the cases in the current study, the flow rate was so low that the refrigerant was brought to room-temperature vapor without the assistance of the heater. For some large flow rates, an electric heater was used to evaporate the refrigerant. After passing through the test section, the vapor was condensed in the receiver tank. The receiver ice-water bath was in a Dewar so that there was no condensation of water vapor outside of the bath that could decrease accuracy.

For liquid refrigerant tests, the receiver tank was exposed to room temperature. Metering valve #1 and heater #2 were by-passed and liquid R134a flowed directly to the test section. After passing through the test section, the refrigerant expanded to the pressure of receiver tank in metering valve #2.

2.2.3 Instrumentation

The receiver tank was placed above a digital balance (Sartorius model BP6100), which provides a third way to measure the mass flow rate in addition to the two flow meters mentioned above. The flow rate was measured by weighing the liquid accumulation during a period of stable state. The digital balance was connected to a computer with an RS-232 cable, which allowed the computer to retrieve the balance reading at intervals of three seconds. The
slope function in Microsoft Excel, which returns the slope of the linear regression line through two groups of data points, was used to calculate the mass flow rate.

The mass flow rate in the current investigation spanned a range between 0.12 – 45.2 g/min. Mass flow rates larger than 8.5 g/min were measured using the Micromotion flow rate meter with an uncertainty of 0.7% according to the calibration curve supplied by the manufacture of the flow meter. The digital balance weighting method has been used to verify this uncertainty range, as shown in Figure 1.

The flow rates below 8.5 g/min were measured with the digital balance. The uncertainties for flow rates below 8.5 g/min and above 0.36 g/min were within ±1.0 %. For flow rate less than 0.36 g/min, at least 30 minutes of data were taken for a stable state, and the uncertainties were determined to be within ± 2.0 %.

The Rheotherm® flow meter operated in the range of 0.36 to 8.5 g/min. It was calibrated with the digital balance, as shown in Figure 2, with an accuracy of ± 3.0 %. The experimental data within the operational region of Rheotherm were crosschecked with the flow rates determined from the balance reading, and the discrepancies were within ± 3.0%. In addition, the continuous readings of this instrument provided an important means, in addition to the pressure and temperature versus time curves, to check whether a stable state had been reached.

The fluid temperatures at the inlet and outlet of the test section were measured with two type-T thermocouples. The test section inlet and outlet pressures were measured with two pressure transducers (Setra model 206, 0 ~ 1724 kPa). The pressure drop across the test section was measured with three differential pressure transducers (Setra model C230, 0 ~ 6.8 kPa, 0 ~ 68kPa, and 0 ~ 172 kPa). All three transducers were used new and calibrated within an accuracy of 0.25% full scale by the manufacture. Before the experiments, these three differential transducers were crosschecked in their overlapping region of operation. As shown in Figure 3 and Figure 4, the crosschecking results confirmed the calibration results.

Measurements were recorded with a Hewlett-Packard (HP) data logger and a microcomputer. The sampling frequency was 20 times per minute, and the sampling time was no less than five minutes for each data point. All the data were recorded in Microsoft Excel for future processing.

2.2.4 Experimental Procedure

The flow system was evacuated before it was charged with R134a. Prior to each experiment, air was bled from the test section, with particular attention being paid to remove air from the pressure tap ports since trapped air may induce noise in the measured pressure drop.

For liquid test, in order to make sure that the refrigerant was in a pure liquid state, most of the data points had inlet/outlet subcooling no less than 5 °C with the remaining no less than 0.5 °C. Similarly, most of the vapor data had inlet/outlet superheat larger than 5 °C with the remaining no less than 0.5 °C.

Each channel was tested with several runs of experiments. For a certain run, the Reynolds number changed either from smaller flow rates to larger ones or vice versa. Since the Reynolds number is a similarity parameter, it is expected that the $f$ versus Re relationship should not be a function of experimental conditions such as absolute pressure of fluids, fluid state (liquid or vapor), flow direction (since the channel was symmetric), as well as whether the experiment is running from larger Reynolds numbers to smaller ones or vice versa. In addition, it should not matter if the channel is taken apart and put together again. These experimental conditions were kept unchanged in a
certain experimental run, but varied extensively between different runs. No systematic differences observed between different groups of data suggest proper design of the test section, the experimental methods, as well as correct data reduction techniques.

The microchannel surface may have been deflected during the experiments because of high test-section pressure. Since the height of the microchannel was as small as 37.9 µm, even a deflection of 1 µm in height may introduce noticeable differences (about 10%) in measured friction factors. The deflection is a strong function of the fluid pressure, i.e., the higher the pressure the larger the deflection. The current test section was designed such that the deflection was expected to be negligible. During the current experiment, the averaged test section pressure for liquid data varied between 655 kPa to 1,065 kPa for different runs. No indications of friction factor dependence on the test section pressure were observed.

For \( m < 0.36 \, \text{g/min} \), the digital balance is used to measure the mass flow rate. The accuracy of this approach depends greatly on the stability of the flow. When the experiments were performed with increasing \( \text{Re} \) or decreasing \( \text{Re} \) (closing or opening valve), the flow rates would have a tendency of increasing or decreasing before a stable state has been established. If the data were taken at this stage, systematic errors could be observed. Different runs of data were compared for the current investigation and no systematic errors were observed.

If the fine channel was contaminated with particles, unexpected experimental results may have appeared. Test section #3 and #4 were uncovered and cleaned between different runs, and no systematic errors were observed.

All these efforts verified our design of experiments and greatly improved the quality of the experimental data.

### 2.2.5 Data Reduction and Uncertainties

The friction factor \( f \), Reynolds number \( \text{Re} \), and friction factor constant \( f \times \text{Re} \) were determined for each data point. The Reynolds numbers were calculated based on the hydraulic diameter of the channel:

\[
\text{Re} = \frac{G D_h}{\mu} \quad (2-2)
\]

The mass flux \( G \) and the hydraulic diameter \( D_h \) were calculated with the following equations:

\[
G = \frac{m}{H \times W} \quad (2-3)
\]

\[
D_h = \frac{2HW}{H + W} \quad (2-4)
\]

Liquid flow was assumed incompressible, and the friction factor was calculated as,

\[
f = \frac{\Delta P D_h}{L G^2} \quad (2-5)
\]

Substituting equations (2-3) and (2-4), Equation (2-5) becomes

\[
f = \frac{4\rho \times \Delta P \times H^3 W^5}{L(H + W)m^2} \quad (2-6)
\]

\[
f = f \times \text{Re} = \frac{8\rho \Delta P H^3 W^5}{L\mu(H + W)^2 m} \quad (2-7)
\]
The vapor flow in so small a channel cannot be taken as incompressible. Based on the momentum theorem, the pressure drop while gas flows in a straight constant area channel is

\[
\frac{dp}{dx} = -\frac{f}{D_h} \frac{\rho u^2}{2} - \rho \frac{du}{dx} \tag{2-8}
\]

The above equation can be integrated assuming isothermal, ideal gas flow, that is,

\[
p_1 - p_2 = \frac{G^2}{2\rho_m} \left( \frac{fL}{D_h} + 2 \ln \frac{p_1}{p_2} \right) \tag{2-9}
\]

where \(\rho_m\) is the density based on the average pressure \(\frac{p_1 + p_2}{2}\).

### Table 2-5 Experimental uncertainties of single-phase pressure drop test

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>(m) (0.2 ~ 0.36 g/min)</td>
<td>± 2.0 %</td>
</tr>
<tr>
<td>(m) (0.36 ~ 8.5 g/min)</td>
<td>± 1.0 %</td>
</tr>
<tr>
<td>(m) (&gt; 8.5 g/min)</td>
<td>± 0.7 %</td>
</tr>
<tr>
<td>Absolute pressure</td>
<td>± 3.5 kPa</td>
</tr>
<tr>
<td>Temperature</td>
<td>± 0.2 °C</td>
</tr>
<tr>
<td>Differential pressure (0 ~ 6.8 kPa)</td>
<td>± 0.017 kPa</td>
</tr>
<tr>
<td>Differential pressure (6.8 ~ 172 kPa)</td>
<td>± 0.43 kPa</td>
</tr>
</tbody>
</table>

### Table 2-6 Error propagation of single-phase pressure drop test

<table>
<thead>
<tr>
<th>Test section</th>
<th>Re</th>
<th>(f)</th>
<th>(C_f)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>± 1.0%</td>
<td>± 5.1%</td>
<td>± 4.8%</td>
</tr>
<tr>
<td>2</td>
<td>± 2.0%</td>
<td>± 5.9%</td>
<td>± 5.6%</td>
</tr>
<tr>
<td>3</td>
<td>± 2.0%</td>
<td>± 6.3%</td>
<td>± 6.0%</td>
</tr>
<tr>
<td>4</td>
<td>± 2.0%</td>
<td>± 6.0%</td>
<td>± 5.8%</td>
</tr>
<tr>
<td>5</td>
<td>± 2.0%</td>
<td>± 4.5%</td>
<td>± 3.2%</td>
</tr>
</tbody>
</table>

The uncertainties for the measured parameters and the calculated results are listed in Table 2-5 and Table 2-6, respectively. The uncertainty propagation functions described in Taylor and Kuyatt (1994) were used to estimate uncertainties for calculated results. Although each data point has an associated uncertainty, only the maximum values of the uncertainties are presented for each test section.

The above uncertainty analysis did not consider the errors introduced from the channel geometry (length, height, and width) measurements. The geometric parameter errors, when propagated into calculated results, are systematic. For example, a positive error in channel height would increase the entire calculated friction factors, \(f\), of that channel for the same percentage. In the \(f\) versus Re curves discussed below, it will shift the entire data up/down and left/right, without changing the shape of the curve. As a result, the uncertainty propagations of these parameters are analyzed separately.
The error associated with the channel length was neglected since it could be measured very accurately (within ± 0.1% error). The repeatability of H and W were within ± 0.7% and the maximum errors for Re, f and fRe associated with these two parameters were ± 0.7%, ± 3.5%, and ± 2.8%, respectively.

2.3 Adiabatic Two-phase Flow Pressure Drop and Visualization

2.3.1 Test Section

2.3.1.1 Two-phase pressure drop test section
The five test sections that had been tested for single-phase flow pressure drop were used in adiabatic two-phase flow pressure drop studies. Please refer to section 2.2.1 for detailed description and characterization of the test sections.

2.3.1.2 Two-phase visualization test section
The test section for two-phase flow visualizations is shown in Figure 2-7. It is based on the same design as that is used for the pressure drop test sections. The only difference is that the cover plate and the substrate were bolted together, and sealed with a very thin layer of vacuum grease.

![Figure 2-7 Two-phase flow visualization microchannel test section](image)

Table 2-7 Geometric parameters of the flow visualization test section

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Average</th>
<th>Maximum</th>
<th>Minimum</th>
</tr>
</thead>
<tbody>
<tr>
<td>H (µm)</td>
<td>92.0</td>
<td>93.3</td>
<td>90.1</td>
</tr>
<tr>
<td>W (µm)</td>
<td>951.6</td>
<td>940.0</td>
<td>960.0</td>
</tr>
<tr>
<td>Dh (µm)</td>
<td>167.8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lt (mm)</td>
<td>44.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Li (mm)</td>
<td>20.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L (mm)</td>
<td>20.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Le (mm)</td>
<td>4.7</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The geometric parameters of the test section are listed in Table 2-7. The total length of the channel, Lt, was 266 hydraulic diameters. The distance between the inlet and the first pressure tap hole, Li, was 119 hydraulic diameters. According to Hartnett et al. (1962), approximately 70 hydraulic diameters were required to arrive at a fully developed flow in a rectangular channel with a aspect ratio of 0.1, for both laminar and turbulent flow. Hence, the single-phase flow between the two pressure tap holes can be taken as fully developed.
The same techniques that were described in section 2.2.1 are used to measure the channel length, width, height and surface roughness. These results are shown in Figure 2-8 and Table 2-7. The surface roughness was reported as the arithmetic average surface roughness, Ra, the maximum peak value, Rp, as well as the minimum valley value, Rv. The channel bottom had an Ra of about 0.35 µm, and Rv and Rp values of about 0.9 ~ 1.6 µm. The channel cover was much smoother, with a Ra value of about 0.12 µm.

2.3.2 Experimental Facility

The experimental loop is shown schematically in Figure 2-9. It is modified from the experimental facility for the single-phase pressure drop experiments, as shown in Figure 2-6. For two-phase experiments, the refrigerant flowed through a metering valve and an electrical heater after passing through the flow meters. The refrigerant was heated to a certain vapor quality before entering the test section. The two-phase refrigerant was then condensed in a condenser, which was a section of circular tube inserted in an ice-water bath.
2.3.3 Experimental Conditions

The test section outlet refrigerant temperatures in two-phase experiments were kept at room temperatures (22.4°C ~ 27.6°C), and the inlet temperatures (23.8°C ~ 32.4°C) were higher than room temperature with the values dependent on the total pressure drop. The test section was made from a low thermal conductivity material (PVC, $k = 0.13$ W/m°C). In addition, it was insulated with two-centimeter thick fiberglass. The heat loss to the environment was estimated to be negligible, thus the flow can be taken as adiabatic.

Since the flow was adiabatic, the acceleration pressure drop in two-phase flow could be neglected. Therefore, the measured pressure drop across the two pressure tap holes was taken as the frictional pressure drop.

2.3.4 Instrumentation and Experimental Uncertainties

The flow rate for two-phase experiment ranged between 0.54 – 5.08 g/min. The occurrence of two-phase flow between the test section and receiver, as well as the use of a condenser, makes the balance-weighting method undesirable for flow rate measurement due to the unsteady nature of the condenser draining into the receiver as it undergoes vapor lock during condensation. As a result, the Rheotherm® readings were used for two-phase experiments.

Table 2-8 Experimental uncertainties of two-phase flow pressure drop

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m$</td>
<td>± 3.0 %</td>
</tr>
<tr>
<td>absolute pressure</td>
<td>± 3.5 kPa</td>
</tr>
<tr>
<td>temperature</td>
<td>± 0.2 °C</td>
</tr>
<tr>
<td>differential pressure (0 ~ 6.8 kPa)</td>
<td>± 0.017 kPa</td>
</tr>
<tr>
<td>differential pressure (0 ~ 68 kPa)</td>
<td>± 0.17 kPa</td>
</tr>
<tr>
<td>differential pressure (0 ~172 kPa)</td>
<td>± 0.43 kPa</td>
</tr>
<tr>
<td>heating power</td>
<td>± 2.0 %</td>
</tr>
<tr>
<td>mass flux, G</td>
<td>± 3.0 %</td>
</tr>
<tr>
<td>pressure gradient, dp/dz</td>
<td>± (0.3% ~ 4.2%)</td>
</tr>
<tr>
<td>vapor quality, x</td>
<td>± (0.003 ~ 0.036)</td>
</tr>
</tbody>
</table>

The total heat generated by the preheater was calculated by multiplying the voltage and current across the heater. The heater was insulated with two layers of blown urethane pipe insulation ($k = 0.04$ w/m°C) with a total thickness of one centimeter, and two layers of fiberglass ($k = 0.026$ w/m°C) with a total thickness of four centimeters. The heater surface temperature was recorded during the experiment, which was used to estimate the heat losses to the environment. The heat losses were estimated to be within 2.0% - 10.0 % of the heating power for 97 % of the experimental data. The calculated heat losses were subtracted from the heat generation when calculating the vapor quality at the test section.

The instrumentation for fluid temperature, absolute pressure, and differential pressure, as well as data logging were the same as that in the single-phase experiments, which are described in section 2.2.3. The uncertainties associated with all the parameters are listed in Table 2-8. The uncertainty propagation function described in Taylor and Kuyatt (1994) was used to estimate uncertainties for calculated parameters.
2.3.5 Visualization Equipment

A Sony DCR-TRV130 camcorder was used to visualize the two-phase flow in the microchannel. The camcorder had a 20x optical zoom, and when used in combination with the Sony VCL-MT4037s 4x macro lens, an 80x zoom capability was achieved. A stroboscope was used to help capture the motion of the fluid flow. The strob light was reflected against a white background into the apparatus. The strobe was set to a rate of approximately 5000 flashes/min. The video was then transferred to the PC and still pictures were captured from the video.

2.4 Liquid Boiling and Evaporation Heat Transfer

2.4.1 Experimental Facility

The microchannel test facility is shown schematically in Figure 2-10. Please refer to Section 2.2.2 for the description of the refrigerant flow system. The preheater was used to adjust the metering valve (expansion device) inlet subcooling. Pre-evaporator #1 was used in the adiabatic two-phase pressure drop experiments for adjusting the vapor quality at the test section. The two-phase flow after the test section was condensed to a liquid state in the condenser, which was by-passed for liquid only pressure drop experiments.

![Figure 2-10 Liquid boiling and evaporation heat transfer test facility](image_url)

2.4.2 Test Section

2.4.2.1 Evaporation heat transfer

As shown in Figure 2-11, the test section consisted of a microchannel heat exchanger that had four parallel channels of 85 mm long each, two heaters and four fine thermocouples. The heat exchanger was fabricated from four layers of Kapton® films with a total thickness of 250 µm. Two inner layers were cut with channels and distribution headers, respectively. The two outer layers were 50 µm thick each. The webs between the channels were 1000 µm wide. The fabrication process of such a heat exchanger has been reported by Selby et al (2001).
Two Kapton® insulated heaters were used in the test section. One of them was installed 10 mm downstream of the inlet header and was used as the pre-evaporator (pre-evaporator #2 in Figure 2-11) to adjust the vapor quality at the inlet of the main heating section. The other was used as the heat source for the evaporation heat transfer experiment. By evaporating the refrigerant with a pre-evaporator after the inlet distribution header, the effect of maldistribution on heat transfer measurement was alleviated. Each of the two heaters was mounted above a three mm thick aluminum block using a high thermal conductive epoxy. The aluminum blocks were used because the high thermal conductivity of aluminum could help in providing a uniform heating condition to each of the four channels. The heater assemblies were attached to the microchannel surface with a layer of thermal paste that had a high thermal conductivity.

The distance between two heaters was 10 mm. Because the test section was very thin (250 µm), this distance was estimated to be long enough to isolate the two heaters. Therefore, the heating effects of the heaters were limited to their own heating zones only. The heaters were supplied with DC power. Two power controllers were used to control the heating powers. The whole test section was insulated with five mm thick fiberglass (k = 0.026 W/m-K). The heater surface temperatures were recorded during the experiment, which was used to estimate the heat losses to the environment. For all the experimental data reported, the heat losses were estimated to be less
than 3.0% of the heating powers from both the pre-evaporator and the main heater. The calculated heat losses were subtracted from the heat generation when calculating the vapor qualities at the test section and the wall heat fluxes.

2.4.2.2 Liquid Flow Boiling

The test section for the liquid flow boiling study is shown in Figure 2-12. The main heater in Figure 2-10 was removed and the four thermocouples were insulated with five mm thick fiberglass.

The heat leakage upstream and downstream was estimated to be negligible, because the thermal conductivity of Kapton is low (k = 0.15 W/m °C) and the test section is very thin (250 µm). Therefore, the heating effect was limited to the section mounted with a heater only. This design is an improvement over others, where materials with high thermal conductivity were used; because of axial conduction, the boiling in the microchannels was mixed with that in the large inlet and exit headers (Jiang et al. 1999, 2000). Because the channel wall was very thin (50 µm) and it was insulated, the readings of the four thermocouples could be taken as the bulk fluid temperature inside the specific channel. This new design for the test section permitted the bulk fluid temperature inside the microchannels to be measured non-intrusively.

![Flow boiling test section](image)

Figure 2-12 Flow boiling test section

2.4.2.3 Characterization of the Microchannels

The geometric parameters of the test section and the channels are listed in Table 2-9 and Table 2-10. The channel width was measured with a Nikon MM-11 measurescope. Forty measurements were taken at different locations along the length of the test section, and the average value was 811.94 µm. The channels were uncovered after the experiments, which allowed the measurement of the height and surface roughness with a Sloan Dektak ST stylus surface profilometer. An example of an open channel profile is shown in Figure 2-13 (a). Six measurements were taken at different locations and the averaged height was calculated to be 68.5 µm. The channel surface roughness was measured and reported as the average roughness (Ra), which was 0.1 µm.

![Microchannel parameters](image)

Table 2-9 Geometric parameters of the test section

<table>
<thead>
<tr>
<th>Wt (mm)</th>
<th>Wb (mm)</th>
<th>Ha (mm)</th>
<th>Li (mm)</th>
<th>Lpre (mm)</th>
<th>Le (mm)</th>
<th>Leh (mm)</th>
<th>Lc (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.2</td>
<td>1.0</td>
<td>3.0</td>
<td>10</td>
<td>22</td>
<td>10</td>
<td>22</td>
<td>21</td>
</tr>
</tbody>
</table>
Table 2-10 Geometric parameters of the channel

<table>
<thead>
<tr>
<th>δ_{wall} (µm)</th>
<th>W (µm)</th>
<th>H (µm)</th>
<th>EH (µm) at 330 kPa</th>
<th>EH (µm) at 650 kPa</th>
<th>Ra (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50.0</td>
<td>811.94</td>
<td>68.5</td>
<td>2.06</td>
<td>6.50</td>
<td>0.1</td>
</tr>
</tbody>
</table>

The heated wall of the channel was only 50 µm thick, the shape of which was not flat when under high pressure. The effect of fluid pressure on the channel outer wall shape was characterized with the profilometer. Figure 2-13 (c) illustrates the outer wall profile under a pressure of 650 kPa, which was the typical pressure for the two-phase flow experiment. Figure 2-13 (d) shows the channel surface profile under the typical pressure for the vapor only pressure drop test. Since the wall is very thin and only miniscule compression could be envisioned, the inner wall profile can be taken as the same as the measured outer wall profile.

Figure 2-13 Characterization of microchannels used in heat transfer study

The channel height was corrected by adding a correction factor, EH, which is a function of the system pressure. EH was estimated as

\[
EH = \sum_{j=1}^{m} EH_j
\]

\[
EH_j = \frac{\sum_{i=1}^{n} (y_i - y_0)}{n}
\]
where m is the total number of measurements. For measurement \(j\) (\(1 \leq j \leq m\)), the correction factor, \(EH_j\), was calculated as the averaged value of the bulge \((y_i - y_0)\), \(1 \leq i \leq n\), along the channel width direction. The \(EH\) value for a specific pressure was averaged over \(m\) different measurements. In this study, \(n = 200\) and \(m = 45\). As summarized in Table 2-10, at working pressure of 650 kPa, the channel was approximated as height of \(68.5 + 6.5 = 75.0 \mu m\); and at 330 kPa as height of \(68.5 + 2.06 = 70.56 \mu m\).

2.4.3 Instrumentation

Four type-T fine gage thermocouples (75 \(\mu m\) wire diameter, \(T_{w1}\) in Figure 2-10 and Figure 2-11) were mounted in the center of the microchannels using a high thermal conductive epoxy, one above each channel, to measure the outer-wall surface temperatures. The measurement techniques for fluid temperature, absolute pressure, differential pressure, mass flow rate, and data acquisition system were the same as the pressure drop experiments, as have been discussed in section 2.2.3.

The thermocouples were calibrated with errors within \(\pm 0.1^\circ C\) and the calibration results are shown in Figure A-5. The uncertainty in flow rate measurements was estimated to be less than \(\pm 3\%\). The uncertainty in pressure measurements was \(\pm 0.2\%\); and in differential pressure measurements, less than \(\pm 0.3\%\).

2.4.4 Energy Balance and Single-phase Flow Distribution

Auxiliary tests with water were preformed to evaluate the distribution, energy balance, and temperature measurements. After the heating section, the water bulk temperature was measured \((T_{ch1} - T_{ch4}\) in Figure 2-12). The high thermal conductivity of the aluminum permitted the assumption of uniform heating to the four channels. Therefore, the differences between \(T_{ch1} - T_{ch4}\) were attributed to maldistribution. The fraction of flow in each channel can be calculated, given the heating power, inlet temperature, as well as mass flow rate of the water. As shown in Figure 2-14, for Reynolds numbers in the range of 42 - 112, the percentage of total flow in each of the four channels was within \(25\% \pm 1.5\%\), indicating good liquid flow distribution. Since the liquid refrigerant was supplied to the inlet distribution header during the flow boiling test, a good liquid flow distribution also indicated a good distribution for the flow boiling experiments before the onset of nucleate boiling (ONB). The situation after ONB is quite complex. An analysis of that is given in Chapter 6.
Figure 2-14 Water flow distribution in the four-channel test section

The energy balance of the test section was determined to be within ±5% for all the six runs (Figure 2-14). Thus, the heat loss from the test section was rather small, and the test section was appropriate for our measurements.

The effectiveness of thermocouples $T_{ch1}$~$T_{ch4}$ in measuring bulk fluid temperatures in the flow boiling test was also evaluated by comparing the measurements of $T_{ch1}$~$T_{ch4}$ with the exit temperature $T_{eo}$. As shown in Figure 2-15, $T_{ch1}$~$T_{ch4}$ under-represent the bulk fluid temperature $T_{eo}$ within 0~1°C. This difference was attributed to heat loss from through the metal T-junction that was used to install the thermocouple to measure the outlet temperature.
2.4.5 Experimental Procedures and Conditions

2.4.5.1 Evaporation heat transfer

For evaporation heat transfer experiment, a good liquid flow distribution can be maintained by keeping the inlet of the test section at or close to a liquid condition. Nevertheless, it is still very difficult to evaluate quantitatively the effect of maldistribution on the measured adiabatic two-phase pressure drop data. For all the following data analysis, the mass flux, \( G \), was calculated by assuming uniform distribution among each of the four channels.

The single-phase friction factor was obtained prior to performing the two-phase experiment, in order to check the reliability of the experimental methods and the correctness of channel geometry measurement. For liquid flow experiments, in order to make sure that the refrigerant was in a pure liquid state, the test section inlet/outlet subcooling was kept no less than 1.3 °C. Similarly, for the vapor data, the test section inlet/outlet superheat was no less than 13 °C.

The adiabatic two-phase flow pressure drop experiments were performed before the evaporation heat transfer experiments. The data were compared with the adiabatic two-phase pressure correlation that is developed in Chapter 5.

For all the experiments, the refrigerant temperatures were kept at or close to the room temperature in order to keep the heat losses to the environment to a minimum.

2.4.5.2 Liquid flow boiling

The liquid was supplied at room temperature to the test section. The liquid subcooling varied from 0.5 to 7°C, by changing the test section pressures. The liquid mass flux ranged from 97 to 310 kg/m²s. The first test procedure was to adjust the control valves in order to get the desired mass flux and pressure. When these values were stable, the heater was turned on and the heating power was increased step by step. At each power level, at least 6 minutes of data were taken, with the time depending on how long it took to reach a stable state. Data were taken continuously until liquid superheat disappeared, i.e., the readings of \( T_{ch1} - T_{ch4} \) were the same as saturation temperature corresponding to the test section pressure. During the experiments, the variation of mass flux was kept within ±5%; inlet subcooling, within ±0.2°C; and saturation temperature, within ±1°C.

The total heat generated by the electrical heater was calculated by multiplying the voltage and current across the heater. The test section was insulated with 5 mm-thick rubber insulation. Heat losses to the environment were estimated as negligible. Axial heat conduction along the length of the tube was also negligible. The heat flux value was calculated by assuming that the same amount of heat had been added to each of the four channels. When subcooled liquid is heated, and especially when liquid superheat occurs, more heat will be added to the front part. Therefore, the calculated heat flux is the average over the entire heating section. The mass flux \( G \) was calculated assuming a uniform distribution among the four channels.

2.4.6 Data Reduction and Uncertainty Analysis

2.4.6.1 Single-phase friction factor

The frictional pressure drop was calculated by subtracting the entrance and exit losses from the measured total pressure drop. The entrance loss and exit loss were estimated using correlations from Kays and London (1964). The sum of the two losses was estimated to be less than 5% of the total pressure drop for all the data. The liquid
flow was assumed incompressible, and the friction factor was calculated from Equation (2-5). The vapor flow friction factor was calculated with Equation (2-9) considering the compressible effect.

2.4.6.2 Two-phase frictional pressure drop

For adiabatic two-phase flow pressure drop, the frictional pressure drop was calculated by subtracting the pressure losses at the test section inlet (sudden contraction, \( \Delta P_{\text{con}} \)) and exit (sudden enlargement, \( \Delta P_{\text{exp}} \)) from the measured total pressure drop \( \Delta P_{\text{tot}} \).

\[
\Delta P_f = \Delta P_{\text{tot}} - \Delta P_{\text{con}} + \Delta P_{\text{exp}}
\]

(2-12)

Note that the acceleration pressure drop was neglected, since the experiments were carried out under adiabatic conditions. The two-phase flow pressure drop for sudden contraction was estimated with the homogeneous model, as suggested by Collier and Thome (1996)

\[
\Delta P_{\text{con}} = \frac{G^2}{2 \rho_{\text{avg}}} \left[ \left( \frac{1}{C_c} - 1 \right)^2 + \left( 1 - \frac{1}{\sigma} \right)^2 \right]
\]

(2-13)

\[
\sigma = \frac{A_{\text{header}}}{A_{\text{channel}}}
\]

(2-14)

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

(2-15)

where the coefficient of contraction \( C_c \) is a function of \( \sigma \) and \( x \) is the vapor quality. The vapor quality at the inlet of the test section was evaluated from the energy balance of the pre-evaporator #1.

For a sudden enlargement (exit effects), the pressure drop is given by Hewitt \textit{et al.} (1993) as

\[
\Delta P_{\text{exp}} = \frac{G^2 \beta (1 - \beta) \Psi_s}{\rho_l}
\]

(2-16)

\[
\beta = \frac{A_{\text{channel}}}{A_{\text{header}}}
\]

(2-17)

The separated flow multiplier \( \Psi_s \) is given by

\[
\Psi_s = 1 + \left( \frac{\rho_l}{\rho_v} - 1 \right) \left( \frac{x}{4} (1-x) + x^2 \right)
\]

(2-18)

Coleman (2000) demonstrated experimentally that the above correlation predicted the minor losses in microchannels with inner diameter of 0.76 mm very well. The estimated pressure losses in the current experiment were rather small and the frictional pressure drop ranged from 98 to 99.5% of the total pressure drop.

2.4.6.3 Evaporation heat transfer

The vapor quality at the inlet of the test section, \( x_{ei} \), was determined by assuming adiabatic flow through the metering valve. The main heating section inlet quality, \( x_{ch, in} \), was calculated based on the energy balance of pre-evaporator #2. Similarly, the test section outlet quality, \( x_{ch, out} \), was evaluated based on the energy balance of the
main heater. The mean quality, \( x_{\text{avg}} \), and the change of quality across the main heating section, \( \Delta x \), can be expressed as

\[
x_{\text{avg}} = \frac{x_{\text{eh,in}} + x_{\text{eh,out}}}{2}
\]

\( \Delta x = x_{\text{eh,out}} - x_{\text{eh,in}} \) \hspace{1cm} (2-20)

The heat transfer coefficient for the evaporation of R-134a was determined from the equation

\[
h_{\text{lp}} = \frac{Q - Q_{\text{loss}} - Q_{\text{bottom}}}{A_{\text{eff}} (T_{\text{wi}} - T_r)}
\] \hspace{1cm} (2-21)

where \( Q \) is the total heat generated by the main heater, \( Q_{\text{loss}} \) is the estimated heat loss to the environment, \( A_{\text{eff}} \) is the effective heat transfer area, \( T_{\text{wi}} \) is the inner wall temperature, and \( T_r \) is the refrigerant temperature. \( Q \) was calculated by multiplying the measured voltage \((V_2)\) and corresponding current \((I_2)\) across the main heater. Note that only the top of the microchannel was heated directly, and the bottom surface was insulated. Numerical analysis demonstrated that about 10% of the total heat was added to the refrigerant from the bottom side (unheated side). This part of the heat was denoted as \( Q_{\text{bottom}} \) in Equation (2-21) and was subtracted from the total heat. Therefore, only three sides of the channel were considered in calculating the effective heat transfer area.

The inner wall temperature, \( T_{\text{wi}} \), is given by

\[
T_{\text{wi}} = T_{\text{wo}} - \frac{(Q - Q_{\text{loss}}) \delta_{\text{wall}}}{A_{\text{eff}} k_w}
\] \hspace{1cm} (2-22)

where \( T_{\text{wo}} \) is the outer wall temperature, which is the average of the four fine-gage thermocouples readings, \( \delta_{\text{wall}} \) is the thickness, and \( k_w \) is the thermal conductivity of the wall. The value of \( k_w \) is 0.15 W/m-K was applied, based on the manufacturer (DuPont) provided data.

The total pressure drop across the test section was as high as 16 kPa in the current experiments, which corresponds to about 1 °C of change in the saturation temperature. As a result, the change of local refrigerant temperature along the channel cannot be neglected. The total pressure drop is expressed as

\[
\Delta P_{\text{tot,calt}} = \Delta P_{\text{Li}} + \Delta P_{\text{Lpre}} + \Delta P_{\text{Lb}} + \Delta P_{\text{Leh}} + \Delta P_{\text{Le}} + \Delta P_{A_{\text{Lpre}}} + \Delta P_{A_{\text{Leh}}} + \Delta P_{\text{con}} - \Delta P_{\text{exp}}
\] \hspace{1cm} (2-23)

where \( \Delta P_{\text{Li}}, \Delta P_{\text{Lb}} \) and \( \Delta P_{\text{Le}} \) are frictional pressure drop in the inlet section \( Li \), the section between two heaters \( Lb \) and the exit section \( Le \), respectively. The flow through these three sections can be taken as adiabatic, hence, only frictional pressure drop needs to be considered. The model developed in Chapter 5 for adiabatic two-phase flow pressure drop in microchannels, Equation (5-52), is used to calculate the frictional pressure drop.

However, the variation of quality across the pre-evaporator and the main heating section must be taken into account. As a result, the frictional pressure drops across these two heating sections are integrated along the length of the channel.

The acceleration pressure drop across the pre-evaporator and the main heating section can be estimated with the correlation expressed by Tran (1998) as,
\[
\Delta P_A = \frac{G^2}{\rho}(\frac{x_{out}^2}{\rho_x\alpha_{out}} + \frac{(1-x_{out})^2}{\rho_y(1-\alpha_{out})}) - \left[ \frac{x_{in}^2}{\rho_x\alpha_{in}} + \frac{(1-x_{in})^2}{\rho_y(1-\alpha_{in})} \right]
\]  
(2-24)

where \(\alpha\) denotes the void fraction, which was expressed by Zivi (1964) as

\[
\alpha = \left[ 1 + \left( \frac{1-x}{x} \frac{\rho_x}{\rho_y} \right)^{0.67} \right]^{-1}
\]  
(2-25)

The refrigerant saturation pressure at the center of the main heating section was estimated as

\[
\Delta P_{ch} = P_{eo} + \frac{\Delta P_{Leh} + \Delta P_{A,Leh}}{2} + \Delta P_{Le} - \Delta P_{exp}
\]  
(2-26)

where \(P_{eo}\) is the test section exit pressure. \(P_{ch}\) was used to evaluate the saturation refrigerant temperature \(T_r\).

2.4.6.4 Uncertainty analysis

Uncertainties of the experimental results were analyzed using the uncertainty propagation function proposed by Taylor B.N. and Kuyatt (1994). The detailed results for the parameters and the estimated uncertainties are given in Table 2-11.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, width and thickness</td>
<td>± 1.0 %</td>
</tr>
<tr>
<td>Temperature, ( T )</td>
<td>± 0.1 °C</td>
</tr>
<tr>
<td>Pressure, ( P )</td>
<td>± 0.2%</td>
</tr>
<tr>
<td>Pressure drop, ( \Delta P )</td>
<td>± 3 kPa</td>
</tr>
<tr>
<td>Voltage, ( V )</td>
<td>± 0.1%</td>
</tr>
<tr>
<td>Current, ( I )</td>
<td>± 1.0%</td>
</tr>
<tr>
<td>Mass flow rate, ( m )</td>
<td>± 3 % (for ( m &lt; 8.5 ) g/min);</td>
</tr>
<tr>
<td></td>
<td>± 0.7 % (for ( m &gt; 8.5 ) g/min)</td>
</tr>
<tr>
<td>Surface heat flux, ( q'' )</td>
<td>± 1.0%</td>
</tr>
<tr>
<td>Mass flux, ( G )</td>
<td>± 4.0%</td>
</tr>
<tr>
<td>Vapor quality, ( x )</td>
<td>± 0.04</td>
</tr>
<tr>
<td>Single-phase friction factor, ( f )</td>
<td>± 6.0%</td>
</tr>
<tr>
<td>Heat transfer coefficient, ( h_{tp} )</td>
<td>± 2.8% - ±12.8% for ( h_{tp} \leq 10 ) kW/m²°C</td>
</tr>
<tr>
<td></td>
<td>± 12.8% - ±19.0% for ( h_{tp} &gt; 10 ) kW/m²°C</td>
</tr>
</tbody>
</table>

2.5 Microscale Orifice-tube

2.5.1 Experimental Facility

The micro-orifice experimental facility is shown schematically in Figure 2-16. Because the flow rate was very small, the apparatus was designed as a once-through system for stability of the flow and simplicity. The system consisted of a refrigerant supply tank, control valve, test section, and receiver tank for adjusting the experimental conditions. The refrigerant tank contained saturated R134a, which was maintained at a desired pressure (or temperature) using a variable transformer (variac) and an electric resistance heater. Liquid refrigerant was driven into the test loop by placing the supply pipe at the bottom of the reservoir. The receiver tank was exposed to room temperature or placed in an ice bath, which provided a stable lower pressure. The orifice downstream pressure, \( P_{oo} \),
was adjusted by a control valve. The orifice upstream subcooling was controlled by adjusting the heating power of a rope heater, which was wrapped around the tube upstream of the test section.

![Experimental apparatus for orifice test](image)

The receiver tank was placed on a digital balance (Sartorius model BP6100A), and the flow rate was measured by weighing the liquid accumulation during a long period of a stable state. A Rheotherm® mass flow rate meter was installed before the test section, as a way to crosscheck the flow rate. In fact, the mass flow rate meter could only guarantee accuracy in a very limited region, and most of the flow rates were out of this range. In addition, the balance was more accurate than the mass flow rate meter if the flow was stable. Therefore, the balance weighting results was used for data analysis, and the flow rate meter could help to make sure a steady state had been reached since its reading was continuous. A filter with a mesh size of 0.5 µm was installed just before the flow meter to protect it from minute contamination particles. Because the Rheotherm® flow rate meter is based on liquid flow energy balance, a subcooler and a sight-glass were used to make sure a subcooled liquid was entering the flow meter. The uncertainty in mass flow rate measurement was estimated to be within ± 1.0% in the current study.
2.5.2 Test Section

The test section was shown in Figure 2-17. It consists of the orifice, orifice holder, filter, thermocouples, and pressure transducers. The orifice holder was manufactured from Delrin®, and has two parts (A and B in Figure 2-17). The orifice was sandwiched between the two parts, and sealed with a very thin layer of epoxy. A filter with a mesh size of 0.5 µm was inserted into the inlet of the orifice holder to avoid clogging of the orifice.

The orifice upstream and downstream fluid temperatures were measured using two type-T thermocouples ($T_{in}$ and $T_{oo}$ in Figure 2-17), both of which were inserted into the center of the flow stream with distances of five millimeters from the orifice. A pressure transducer (Setra model 206, 0 ~ 1724 kPa) was used to measure the orifice upstream pressure, $P_{in}$ in Figure 2-17. The pressure tap was drilled after the filter, since the pressure drop across the filter may not be negligible. A T-compression-fitting, which was connected directly to the test section, was used for downstream pressure measurements (not shown in Figure 2-17). The temperatures and pressures were monitored using a computer data acquisition system. The estimated accuracy of the temperature measurements was ± 0.2 °C.

The pressure transducer was calibrated to ± 0.13% full-scale accuracy.
The orifice is a pinhole drilled by laser on a 130 µm thick, 9.5 mm diameter stainless steel foil. The dimensions of the orifices are shown in scale in Figure 2-18. An Olympus microscope with a magnification of 1,000 times was used to measure the orifice size and observe the entrance conditions. When a front-lighted mode was used, the orifice surface conditions observed are shown in Figure 2-19 (a) and (c). Burrs were observed at the periphery of the hole, which are about 20% of the orifice diameter in size and less than five micrometers in height. A backlighted mode was used to measure the orifice size, as shown in Figure 2-19 (b) and (d). The two orifices were measured to be 52.0 and 31.0 µm in diameter, with errors within ± 0.5 µm.

![Micro-orifice characterization using 1000 times magnification microscope](image)

2.5.3 Instrumentation and Error Analysis

All experimental data were recorded in steady condition for at least ten minutes. It was assumed that steady state was reached when the change in upstream temperature, $T_{in}$, was within ± 0.2 °C, upstream pressure change was within ± 3.0 kPa, and downstream pressure change was within ± 5.0 kPa for a minimum of five minutes preceding data collection.

References


Chapter 3. Single-Phase Flow Pressure Drop

3.1 Chapter Overview

This chapter focuses on investigating fully developed liquid and vapor flow through rectangular microchannels with hydraulic diameters varying from 69.5 to 304.7 µm and with aspect ratios changing from 0.09 to 0.24. R134a liquid and vapor were used as the testing fluids. During the experiments, the Reynolds numbers were varied between 112 and 9,180. Pressure drop data are used to characterize the friction factor in the laminar region, the transition region and the turbulent region. When the channel surface roughness was low, both the laminar friction factor and the critical Reynolds number approached the conventional values, even for the smallest channel tested. Hence, there was no indication of deviation from the Naviers-Stokes flow theory for rectangular microchannels. The friction factor data in the turbulent region were larger than the predictions from the Churchill’s (1977) equation for smooth tubes, even for the smoothest channel tested ($Ra/D_h = 0.14\%$). In addition, it was likely that surface roughness was responsible for higher laminar flow friction and earlier transition to turbulent flow in one of the channels tested.

3.2 Literature Review

A summary of experimental studies of single-phase pressure drop in microchannels is listed in Table 3-1. Wu and Little (1983) measured the friction factor for the flow of gases in microchannels with hydraulic diameter ranging from 55.8 to 83.1 µm. They found that the friction factors for both the laminar and turbulent regimes were larger than predictions from the conventional equations. The transitional Reynolds numbers ranged from 350 to 900. The trend discovered was that rougher channels led to an earlier transition from laminar to turbulent flow. However, Wu and Little (1983) did not measure the surface roughness directly, instead, they estimated it using the Karman equation for the complete turbulent zone.

Peng et al. (1994) investigated experimentally the flow characteristics of water flowing through rectangular ducts having a hydraulic diameter of 133 to 367 µm and an aspect ratio of 0.333 to 1.0. Their experimental results indicated that the flow transition occurred between Reynolds numbers of 200 to 700. In addition, the flow friction behavior for both the laminar and turbulent flow dramatically deviated from the classical equations.

Choi et al. (1991) and Yu et al. (1995) measured friction factors for nitrogen and water flowing through micro tubes having inside diameters between 3 and 102 µm. The inner tube surfaces were described as “molecularly smooth”. The measured friction factors in both laminar and turbulent region were found to be less than those predicted from the conventional correlations. The transitional Reynolds numbers were reported to be approximately 2,000.
### Table 3-1 Summary of single-phase studies in microchannels

<table>
<thead>
<tr>
<th>Reference</th>
<th>Geometry</th>
<th>$D_h$ (µm)</th>
<th>$Ra/D_h$ (%)</th>
<th>Material</th>
<th>Re range</th>
<th>$Re_c$</th>
<th>Testing Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wu and Little (1983)</td>
<td>Trapezoidal</td>
<td>55.8 – 83.1</td>
<td>0.5 – 30</td>
<td>Silicon, Glass</td>
<td>100 – 15,000</td>
<td>350 - 900</td>
<td>N₂, H₂, Ar</td>
</tr>
<tr>
<td>Peng et al. (1994)</td>
<td>Rectangular ($α$: 0.333 – 1)</td>
<td>133 – 367</td>
<td>0.6 – 1</td>
<td>SS</td>
<td>50 – 4,000</td>
<td>200 – 700</td>
<td>Water</td>
</tr>
<tr>
<td>Choi et al. (1991)</td>
<td>Circular</td>
<td>3.0 – 81.2</td>
<td>0.01 – 0.8</td>
<td>Silica</td>
<td>30 - 20,000</td>
<td>2300</td>
<td>N₂ gas</td>
</tr>
<tr>
<td>Yu et al. (1995)</td>
<td>Circular</td>
<td>19 – 102</td>
<td>0.03</td>
<td>Silica</td>
<td>250 – 20,000</td>
<td>2000</td>
<td>N₂, gas, water</td>
</tr>
<tr>
<td>Flockhart and Dhariwal (1998)</td>
<td>Trapezoidal</td>
<td>50 – 120</td>
<td>N/A</td>
<td>Silicon</td>
<td>≤ 600</td>
<td>N/A</td>
<td>Water</td>
</tr>
<tr>
<td>Mala and Li (1999)</td>
<td>Circular</td>
<td>50 – 254</td>
<td>0.7 – 3.5</td>
<td>Fused silica, SS</td>
<td>80 – 2,100</td>
<td>300 - 900</td>
<td>Water</td>
</tr>
<tr>
<td>Qu et al. (2000)</td>
<td>Trapezoidal</td>
<td>51 – 169</td>
<td>1.1 – 1.7</td>
<td>Silicon</td>
<td>0 – 1,500</td>
<td>N/A</td>
<td>Water</td>
</tr>
<tr>
<td>Pfund et al. (2000)</td>
<td>Rectangular ($α$: 0.01 ~ 0.05)</td>
<td>252 - 973</td>
<td>0.02 – 0.4</td>
<td>Polycarbonate/polyimide</td>
<td>40 – 4,000</td>
<td>1700-2200</td>
<td>Water</td>
</tr>
<tr>
<td>Judy et al. (2002)</td>
<td>Round, square</td>
<td>15 – 150</td>
<td>N/A</td>
<td>Fused silica, SS</td>
<td>8 – 2,300</td>
<td>N/A</td>
<td>Water, methanol, isopropanol</td>
</tr>
<tr>
<td>Wu and Cheng (2003)</td>
<td>Trapezoidal</td>
<td>25.9 – 291.0</td>
<td>&lt;0.12%</td>
<td>Silicon</td>
<td>10 – 3,000</td>
<td>1,500 – 2,000</td>
<td>Water</td>
</tr>
</tbody>
</table>
Flockhart and Dhariwal (1998) described the flow characteristics of distilled water flowing through trapezoidal channels with hydraulic diameters ranging from 50 to 120 µm. The Reynolds numbers were lower than 600 and the flow was kept well within the laminar flow regime. The experimental results were compared with numerical analysis results based on conventional fluid mechanics. The conventional theory was found to be able to predict the flow in microchannels; however, the authors did not report the surface roughness values.

Mala and Li (1999) studied flow characteristics of water flowing through stainless steel and fused silica microtubes with diameters ranging from 50 to 254 µm. The mean roughness height was ± 1.75 µm, which was provided by the manufacturers. The authors did not provide the shape and the distribution of the roughness elements. For small Reynolds numbers, i.e., Re < 500, the experimental data were in rough agreement with the classical equation predictions. However, as the Reynolds number increased, the friction factor was significantly higher than the predictions by the conventional theory. The authors proposed two possible explanations. One is earlier transition from laminar to turbulent flow; and the other is the surface roughness effect.

Qu et al. (2000) conducted experiments to investigate frictional pressure drop of water flowing through trapezoidal microchannels with hydraulic diameters ranging from 51 to 169 µm. The channels were fabricated on silicon plates and covered with Pyrex glass covers. The cover was very smooth with an average surface roughness on the order of 10 nm, but the silicon channel had an averaged roughness ranging from 0.8 to 2.0 µm. The measured friction factors in the microchannels were higher (8 – 38 %) than those given by the conventional flow theory. A roughness-viscosity model was proposed to interpret the experimental data.

Pfund et al. (2000) experimentally investigated the friction factors of water flowing through two parallel plates with a depth ranging from 128 to 521 µm and a fixed width of 1-cm. Pressure drops were measured within the channel itself to exclude entrance and exit losses, and transitions to turbulence were observed with flow visualization. The experimental results suggest higher friction factors in laminar flow than the classical values. The transitions from laminar to turbulent flow occurred at Reynolds numbers that were lower than the critical Reynolds number for conventional ducts.

Judy et al. (2002) used the pressure drop data to characterize the friction factor for channel diameters in the range 15 – 150 µm and over a Reynolds number range 8-2,300. The microchannels had round and square cross-section geometries. The authors found that error bounds are dominated by measurement of the diameter. The $fRe$ data revealed no distinguishable deviation from macroscale Stokes flow theory. However, no roughness information about the channel surfaces was reported.

Recently, Wu and Cheng (2003) conducted experiments to measure the friction factor of laminar flow of deionized water in silicon microchannels of trapezoidal cross-section with hydraulic diameters in the range of 25.9-291.0 µm. The relative roughness of all the channels was measured to be no more than 0.12%. The experimental data agreed within ± 11% of the analytical solution based on the Stokes flow theory. The authors also reported that transition from laminar to turbulent flow occurred at Re = 1,500 – 2,000 in microchannels having triangular or trapezoidal cross-section with $D_h = 103.4 – 291.0$ µm.

The review of the literature exhibits large scatter and even contradictions in the experimental results for flow friction in microchannels. As has been pointed out by Pfund et al. (2000), the inconsistency in the reported
results can be attributed to several factors such as channel size, geometry, and relative roughness, which were not measured or were possibly incorrectly determined. Since the bonding of silicon and glass is the principal method for microchannel fabrication (Wu and Little 1983, Flockhart and Dhariwal 1998, Qu et al. 2000), part of the discrepancy may be attributed to the lack of well-controlled surface structure during the bonding process. In addition, most of the studies did not measure pressures within the microchannel, but instead measured the pressure upstream and downstream of the channel, and applied conventional corrections for the inlet and exit losses. Unfortunately, it is still unknown whether these correlations can be used in microchannels. Furthermore, some investigators measured friction factors over a channel length of only about a hundred hydraulic diameters, which may not be sufficient length to allow for a fully developed flow.

Nevertheless, there is a general agreement in the literature that surface roughness is a very important factor for microchannel flow. When the channel was smooth, the flow transition from laminar to turbulent occurred at almost the same Reynolds number as the conventional results, according to Choi et al. (1991) and Yu et al. (1995). However, they reported friction factors in both the laminar and the turbulent region lower than the conventional results. The lower friction factor in the laminar and turbulent region may be due to the errors in tube diameter measurement, considering the smallest tube is only 3 µm in diameter and the measured friction factor is proportional to $D^5$. When the channel is rough enough, earlier transition and discrepancies from the conventional laminar friction factors were observed (Wu and Little 1983, Mala and Li 1999, Pfund et al. 2000).

Therefore, it is expected that the conventional correlations could be reproduced in microchannels providing the channel is smooth and the experiments are well controlled. Flockhart and Dhariwal (1998) reported such a result for channels as small as $D_h = 50$ µm and Reynolds number less than 600, however, they did not report the surface roughness. In order to better understand the flow behavior in microchannels, there is a need to perform extensive experiments with a larger range of Reynolds numbers, channel size and shape, and most importantly, with a wide range of surface roughness. The experimental results in the smoothest channels will provide a basis from which other work can be undertaken. The experimental data for rougher channels can be used to predict where the results move away from the basis.

Kandlikar et al. (2001) investigated the effect of channel roughness on flow friction in 2 tubes of 1.031 mm and 0.62 mm in diameter. The roughness of the inside tube surface was changed by etching it with an acid solution. They found that Ra/D of 0.3%, which may be considered smooth for tubes larger than 1 mm, increased the friction factor and heat transfer. The transition to turbulence also was affected by changing the roughness values above this limit.

The literature review also reveals that there is a lack of single-phase friction factor studies in microchannels for the following two topics: (1) Friction factors in rectangular microchannels. Peng et al. (1994) used rectangular microchannels but their results were dramatically different from other data as well as the conventional correlation predictions. It is not clear from the open literature whether the conventional correlations can be used in rectangular microchannels with hydraulic diameters close to or less than 100 microns. It also not clear what is the effect of aspect ratio, $H/W$, on the laminar flow friction factor and the critical Reynolds number. (2) Most of the studies, such as Flockhart and Dhariwal (1998), Mala and Li (1999), Qu et al. (2000), Judy et al. (2002) and Wu and Cheng
(2003), focused most on the flow friction factor in the laminar region. There is a lack of study for flow friction in the turbulent region in microchannels.

In this work, attempts have been made to clarify the above two issues by measuring the frictional pressure drop of liquid and vapor R134a flowing through five rectangular channels with hydraulic diameters varying from 69.5 to 304.7 µm and aspect ratios changing from 0.09 to 0.24. The frictional pressure drops were measured within the channel, away from the entrance and the exit. The same test section was tested under both liquid and vapor state, which provided a way to check the reliability of the experiment as well as to extend the experimental Reynolds number range to \( Re = 112 - 9,180 \).

### 3.3 Experimental Results

The experimental facility, instrumentation, experimental procedure, data reduction and uncertainties, as well as the characterization of the microchannel test sections were described in section 2.2 and in Tu and Hrnjak (2003).

Hartnett and Kostic (1989) give the following simple equation, which is within 0.05% of the analytical solution of a Newtonian fluid in fully developed laminar flow through rectangular ducts.

\[
C_f = f \cdot Re = 96\left(1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9546\alpha^4 - 0.2537\alpha^5\right) (3-1)
\]

The experimental results for all the test sections are shown in Figure 3-1 to Figure 3-5. In each figure, part (a) represents the friction factor, \( f \), for the entire Reynolds number range in the form of \( f \) versus \( Re \) log-log plot; and part (b) presents the product \( f \cdot Re \) as a function of the Reynolds number for data with \( Re < 3,000 \). The circles and the crosses represent the experimental data for the liquid and vapor state, respectively. The dashed lines in part (a) show the theoretical results for fully developed laminar flow in rectangular channels, Equation (3-1). The solid lines are the Churchill (1977) equations for round tubes with different relative roughness (\( e/D \)).

As seen from Figure 3-1 to Figure 3-5, the liquid data and the vapor data collapse on the same curve. Considering two data points with the same Reynolds number, one in the liquid state and the other in the vapor state, the mass flow rate for the liquid state is typically 20 times that of the vapor one. Additionally, the measured frictional pressure drop for the liquid state is normally about 3 times larger than that of the vapor data. In addition, the test section pressure for the liquid state is much higher than that of the vapor state. Thus, the consistency of the data for liquid and vapor state verified the soundness of the experimental methods.
Figure 3-1 Friction factor for test section #1 (Dh = 305 µm, H/W = 0.1)
Laminar theory: $f = \frac{73.4}{Re}$
Hartnett & Kostic (1989)
Churchill (1977) round tube equations

Liquid R134a data
Vapor R134a data

Figure 3-2 Friction factor for test section #2 (D_h = 150 µm, H/W = 0.24)
Laminar theory: $f = \frac{86.1}{Re}$

Hartnett & Kostic (1989)

Churchill (1977) round tube equations

Liquid R134a data

Vapor R134a data

Figure 3-3 Friction factor for test section #3 ($D_h = 141 \mu m$, $H/W = 0.09$)
Figure 3-4 Friction factor for test section #4 ($D_h = 104 \mu m$, $H/W = 0.17$)
Among the 468 data points shown in Figure 3-1 to Figure 3-5, seventy-five pairs have the same Reynolds number. It should be noted that two Reynolds numbers with differences within ± 1% were considered equal. The difference of the measured friction factors was calculated for each pair, and shown as a function of Reynolds number in Figure 3-6. The data label liquid-liquid and vapor-vapor represent both data points in liquid state or vapor state, respectively. Liquid-vapor means two data points, one in liquid and another in vapor state, have the same Reynolds number. Note that the pairs of data are from different runs, as addressed in the experimental procedure section 2.2.4.
For data pairs with the same state (liquid-liquid or vapor-vapor) in Figure 3-6, 91.0% of the points are within ± 8%. For liquid-vapor pairs, 88% of the points lay within ± 10%. Considering the maximum uncertainty of measured friction factor (± 6.2 %) and the variety of experimental conditions that has been changed for different runs, as well as the ± 1.0% error for the Reynolds number, this is in very good agreement. In addition, most of the large errors reside in the region of 1500 < Re < 2500 (see Figure 3-6). This is the region where the flow transition from laminar to turbulent occurs, which may be the reason for the scatter of the data points.

3.4 Analysis and Discussions

3.4.1 Flow Regions

Test section #1 has the largest hydraulic diameter among all the channels. As seen in Figure 3-1 (a), at low Reynolds number range (Re < 2,190), the measured friction factors decrease linearly with Reynolds number on a log-log plot. This is the laminar flow region. At Re = 2,190, the measured friction factors reach the local minimum value ($f \approx 0.04$) and start to deviate from the laminar data. After that, the friction factor rises with increasing Reynolds number in the intermediate Reynolds number range (2,190 < Re < 3,000), following a trend that has been reported for large pipes and channels. This is the critical region. The Reynolds number (2,190) at which the friction factor starts to break from the laminar flow line, is called the critical Reynolds number. When the Reynolds number is larger than 3,000, the friction factors decrease slowly with Reynolds number. This is the turbulent flow region.

Similarly, the $f$ versus Re results for test section #2 and #4 can be divided into the three regions, the laminar region, the critical region and the turbulent region (see Figure 3-2 and Figure 3-4). For test section #5, the current experimental facility and methods limit the experimental Re to less than 3,272, and only the laminar region and the critical region can be seen from Figure 3-5 (a). There is no clear indication of the critical region for test section #3, as seen in Figure 3-3, which suggests a sudden transition from laminar to turbulent flow.
3.4.2 Laminar Flow Friction Factor

Figure 3-1 (b) to Figure 3-5 (b) show the product $fRe$ versus Re for data with Reynolds number less than 3,500 in linear coordinates. In these figures, the solid lines mark the theoretical predictions based on Equation (3-1). For each test section, the average of all the laminar data was calculated, and multiplied by one plus the uncertainty of $fRe$ (see Table 2-6). The results are represented in Figure 3-1 (b) to Figure 3-5 (b) as two dashed lines. Most of the laminar flow data nicely lay between the two dashed lines, with a few data points close to the critical point as exceptions. Therefore, to the degree of the current experimental uncertainties, the product $fRe$ is a constant value in the laminar flow region, which is consistent with the conventional results.

The mean values of $fRe$ in the laminar region were taken as the experimental friction factor constant and compared with the theoretical predictions based on Equation (3-1), as shown in Figure 3-7 and Table 3-2. In section 2.2.5, the error of $fRe$ associated with the channel geometry measurements were determined to be within ± 2.8% for all the five test sections. This is shown in Figure 3-7 as error bars.

The measured values of the friction constants were higher than the classical results for all the five channels. However, the differences for the four test sections other than #3 were small (less than 4%) and were attributed to the uncertainties associated with height and width measurement. Figure 3-7 indicates that the effect of aspect ratio, $H/W$, on the laminar flow friction constant, $fRe$, in rectangular microchannels follows that of the conventional results, Equation (3-1). The only exception is for test section #3 that has an $fRe$ value about 9% higher than the prediction of Equation (3-1).

Table 3-2 Single-phase friction factor experimental results vs. conventional values

<table>
<thead>
<tr>
<th>Test</th>
<th>$D_n$</th>
<th>$H/W$</th>
<th>$(fRe)/(fRe)_{conv}$</th>
<th>$Re_c/Re_{c,conv}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>304.7</td>
<td>0.10</td>
<td>88.0/84.5</td>
<td>2,190/2,470</td>
</tr>
<tr>
<td>2</td>
<td>150.0</td>
<td>0.24</td>
<td>75.2/73.4</td>
<td>2,150/2,315</td>
</tr>
<tr>
<td>3</td>
<td>141.1</td>
<td>0.09</td>
<td>94.0/86.1</td>
<td>1,570/2,470</td>
</tr>
<tr>
<td>4</td>
<td>104.1</td>
<td>0.17</td>
<td>80.0/78.6</td>
<td>2,290/2,315~2,470</td>
</tr>
<tr>
<td>5</td>
<td>69.5</td>
<td>0.09</td>
<td>88.0/85.6</td>
<td>2,200/2,470</td>
</tr>
</tbody>
</table>

Figure 3-7 Measured friction factor constants vs. theoretical predictions
3.4.3 Critical Reynolds Number

The experimental critical Reynolds numbers are compared with those in conventional channels having abrupt entrances (Obot, 1988), as shown in Table 3-2. It should be noted here that a matching conventional result is not available for an aspect ratio of 0.17. However, according to Obot (1988), the tendency is that the smaller the aspect ratio, the larger the critical Reynolds number. Therefore, a first estimate of the conventional value for aspect ratio of 0.17 would be within the range of 2,315 < Re_c < 2,470.

The current experiments suggest slightly earlier transition from laminar to turbulent for all the test sections. Although this tendency is consistent with the majority of the previous research results in microchannels (Wu and Little 1983, Peng et al. 1994, Mala and Li 1999, Pfund et al. 2000), the present critical Reynolds numbers approach the macroscale results. This is true especially for test section #1, #2, #4, and #5. Critical Reynolds numbers in the range of 200 < Re_c < 900 for microchannels, as has been reported in Wu and Little (1983), Peng et al. (1994), and Mala and Li (1999), were not observed in the current study.

3.4.4 Turbulent Flow Friction Factor

Hartnett et al. (1962) found that the circular-tube correlation accurately predicts the friction coefficient for flow through smooth rectangular ducts of any aspect ratio for Re = 6×10^3 - 5×10^5. For rough tubes, it is well known that the relative roughness e/D, where e is the sand grain size, is a major parameter on the flow characteristics. The relative roughness, e/D, for a specific channel can be determined indirectly from the value of the friction factor in the completely turbulent zone. In this zone, the curve of f appears as a set of horizontal lines. The values of f depend only on the relative roughness, and the Karman equation can be used to estimate the e/D, as has been done by Wu and Little (1983). Obviously, such a region has not been reached in the current experiment, since the friction factor decreases with Reynolds number even for the largest Reynolds number tested (Re = 9,180).

As shown in Figure 3-1 to Figure 3-5, the friction factors in the turbulent region do not follow the smooth tube correlation (Churchill, 1977). Instead, in the regions of Re > 3,000, they follow the Churchill’s equation with relative roughness e/D of about 0.7%, 0.5%, 2.0% and 0.3% for test section #1, #2, #3 and #4, respectively.

Even for the smoothest channel, test section #2 (Ra/D_h = 0.14%), the turbulent friction factors are up to 30% larger than the Churchill’s equation with e/D = 0. In conventional channels, however, a roughness of Ra/D_h = 0.14% can be considered as smooth. The results show that the condition of hydraulic smoothness is more difficult to satisfy in microchannels. A similar trend has been described in Acosta et al. (1985) for microchannels with a hydraulic diameter of 368.9 µm. They reported that the condition of hydraulic smoothness in the turbulent regime could be satisfied only when the walls were “optically smooth”. Thus, the present result suggests that it may not be proper to call a microchannel with relative roughness of 0.12% as “smooth” channel, as has been done by Wu and Cheng (2003).

3.4.5 Discussions on the Effect of Roughness

As seen in Table 2-4 the Ra/D_h value for test section #3 (0.35%) is larger than that of test section #2 (0.14%) and #1 (0.16%). This difference may not be significant in conventional channels, but it can have a dramatic effect on the turbulent flow friction factor in microchannels. As it can be seen from Figure 3-1 to Figure 3-3, in the regime of Re > 3,000, the friction factor of channel #3 is about 20 ~ 30% larger than that of channel #1 and #2.
In addition to this, channel #3 demonstrates unusual behavior in the laminar regime and the transition region when compared with other channels and the conventional results; i.e., the laminar friction factor is larger than theoretical predictions and the transition to turbulent is much earlier than other channels. Extensive efforts have been made to eliminate the possibilities that this is due to some unconsidered experimental errors. For example, the channel has been opened and cleaned three times between different runs of experiments. The flow direction has been reversed for one group of data. The 125 experimental data shown in Figure 3-3 contain six different runs with two runs conducted four months after the first run. As has been analyzed previously, these experimental runs report repeatable $f$ versus Re relationship. In addition, as shown in Table 2-2 and Table 2-3, this channel has well-controlled and consistent topographies at different locations. The channel entrance and the pressure tap holes were observed carefully under a microscope, and no burs were observed. Therefore, we have confidence in the experimental results for channel #3.

Channels #3 and #1 can be considered geometrically similar; that is to say, they have almost the same value of $L_e/D_h$, $L_t/D_h$, $H/W$ and even contraction ratio (supplying pipe area divided by the channel cross-section area). The only differences are the channel sizes and the surface roughness. In large channels, the critical Reynolds number depends on many factors including the channel shape (round or rectangular, aspect ratio), the entrance condition (smooth or abrupt), initial flow condition, surface roughness, as well as the entrance length, $L_e/D_h$ (Obot, 1988). The fact that these two channels are geometrically similar indicates that the difference in surface roughness may be the reason for the disagreement between the two critical Reynolds numbers. It is well known that transition can be delayed to larger Re if the necessary precautions (no disturbance at inlet, no pipe vibration, etc.) are taken. During the current experiment, no special precautions were taken to delay the transition. Therefore, the earlier transition from laminar to turbulent flow in channel #3 than in channel #1 could be attributed to a larger relative roughness, $Ra/D_h$, in channel #3. Wu and Little (1983) and Mala and Li (1999) reported earlier transition to turbulent flow in rough microchannels. However, as show in Table 3-1, their observation was based on larger values of $Ra/D_h$ (>0.5%). The current results indicate that the transition behavior in microchannels could be different when the value of $Ra/D_h$ is equal to or larger than 0.35%.

Channel #6 has nearly the same aspect ratio as channels #1 and #3. The critical Reynolds numbers of channels #1 and #6 are almost the same, but are larger than channel #3. The difference (or similarity) in flow transition among these three channels could be attributed to the combined effect of surface roughness and entrance length. The $Ra/D_h$ value of channel #6 (0.3%) is larger than that of channel #1 (0.16%) but less than that of channel #3 (0.35%). Channel #6 is not geometrically similar to channel #1 and #3. As seen in Table 2-1, the entrance length ($L_e/D_h$) of channel #6 is more than twice the values of channel #1 and #3. In conventional channels, Obot (1988) summarized many examples of entrance length on critical Reynolds number. Therefore, even though the $Ra/D_h$ value of channel #6 is only slightly less than channel #3, the critical Reynolds number is very different.

The parameter $Ra/D_h$ may not be the best measure of surface roughness in microchannels. As seen in Figure 2-5, although channels #3 and #5 have very close values of $Ra/D_h$, the distribution and shape of the roughness elements are very different. For example, in the same 100µm distance, the surface of channel #3 has more occurrences of peaks and valleys. Channel #3 has one peak value of about 5µm height, which is about ten times the
value of $Ra$ for this channel. Considering the channel height of 77 $\mu$m, this roughness element is about 6.5% of the channel height. At several other locations, roughness elements with close to ten times $Ra$ were evident in the profilometry results of channel #3, but not observed in other channels. For this reason, channel #3 is “rougher” than the $Ra/D_h$ value indicates when comparing with other channels. Another example is that channels #2 and #5 have the same value of $Ra$ (0.21). However, they look different in the distribution and shape of their roughness elements.

The effect of roughness in microchannels is not only limited to the flow transition and turbulent flow region. As has been reported by Wu and Little (1983), Pfund et al. (2000), and Qu et al. (2000), the laminar friction factors may be larger than the theoretical predictions in rough microchannels. For channel #3, the mean value of the friction factor in the laminar region is about 9% higher than the classical predictions, which is greater than the possible experimental uncertainties. The higher friction factor may be attributed to a large roughness in this channel. Guo and Li (2003) also pointed out that due to the fact that the microchannels have a large surface to volume ratio, factors related to surface effects have more impact to the flow at small scales and surface roughness is likely responsible for the early transition from laminar to turbulent flow and the increased friction factor in microchannels.

3.5 Summary and Conclusions

Fully developed flow frictional pressure drops have been measured over a Reynolds number range of $112 \leq Re \leq 9,180$ for rectangular microchannels in the hydraulic diameter range of $69.5 \mu m \leq D_h \leq 304.7 \mu m$, in the height-to-width ratio range of $0.09 \leq \alpha \leq 0.24$, and in the relative roughness range of $0.14\% – 0.35\%$

The experimental methods are summarized as follows: (1) The test sections were sealed by pressing two smooth surfaces gently with bolts; thereby the channel geometry being tested could be considered the same as when it is open for measurements. (2) The frictional pressure drop was measured directly with pressure ports inside the channel. (3) The fully developed flow was established for the entire Reynolds number range with an adequate entrance length. (4) In order to check the repeatability of the measured friction factor versus Reynolds number relationship, the experimental conditions (test section pressure, fluid state, approaches to the desired flow rate, uncovering channel, etc.) that are believed to be unrelated to the $f$ versus $Re$ curve were varied extensively in different runs.

The following conclusions are obtained from this experimental investigation. (1) In the laminar region, the experimental data for frictional constants, $f/Re$, of both liquid and vapor R134a flow in four microchannels with smoother surfaces ($Ra/D_h < 0.3\%$) agree with the analytical solution based on the Navier-Stokes equation, and the effect of aspect ratio presented in the correlation by Hartnet and Kostic (1989) works for small channels (Figure 3-7). (2) The critical Reynolds numbers of the above four smoother microchannels were in the range of $2,150 \leq Re_c \leq 2,290$, which are only marginally earlier than the conventional results for rectangular channels with the same aspect ratios. (3) In the turbulent region, the friction factors in all the microchannels tested are considerably larger than that predicted by the Churchill’s (1977) equations for smooth tubes. Even for the smoothest channel with relative roughness of 0.14% the turbulent friction factors are up to 30% larger than the Churchill’s equation with $e/D = 0$. (4) For one channel with the greatest surface roughness, $Ra/D_h = 0.35\%$, but intermediate hydraulic diameter, $D_h = 141.1 \mu m$, the friction factor data showed different behaviors in the entire range of experiments. The laminar friction was about 9% higher than the theoretical predictions; the critical Reynolds number, $Re_c = 1,570$, was earlier than the
conventional results; and the turbulent friction was higher than other channels. The unusual behavior was an
indication of the effect of surface roughness in microchannels.

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Chapter 4. Adiabatic Two-Phase Flow Visualization

4.1 Chapter Overview

The chapter presents the flow visualization results for adiabatic two-phase flow of R134a in a rectangular microchannel with cross-section geometries of 92.0 µm high and 951.6 µm wide. The visualizations were performed in the range of mass flux from 279 to 461 kg/m²-s and vapor quality from 0.05 to 0.95. Bubbly flow, plug/slug flow and annular flow were identified. The results demonstrate that the flow patterns in the microchannel of cross-section geometries considered in the current study are similar to those in minichannels (Dh in the order of 1 mm). The slug/plug flow and annular flow were the predominant flow regimes, and the bubbly flow only occurred when quality was very low. The flow pattern transitions were compared to the experimental results of Coleman (2000) and the correlation of Akbar et al. (2003).

4.2 Literature Review

Table 4-1 summarizes flow regime studies in mini/micro channels. Large tube flow regime transition models have been demonstrated to be inapplicable in minichannels with diameter ranging from 1 to 5 mm (Damianides and Westwater 1988, Lin et al. 1999, Triplett et al. 1999, Coleman 2000, and Coleman and Garimella 1999, Yang and Shieh 2001). In addition, they also identified that hydraulic diameter is a significant variable in the determination of flow patterns for two-phase flow through small pipes. Coleman (2000) demonstrated that aspect ratio was less significant in affecting flow regimes than the tube hydraulic diameter; and the effect of the aspect ratio is more pronounced in smaller hydraulic diameter tubes. The experiment of Wolk et al. (2000) also indicated that the flow regimes of non-circular minichannels are different from that of circular tubes.

Nino (2002) visualized R134a, R410A, and air-water mixtures flowing through a multi-port microchannel tube with a hydraulic diameter of 1.58 mm under adiabatic conditions. The results indicated that several flow configurations might exist in multi-port microchannel tubes at the same time while constant mass flux and quality flow conditions were maintained. The flow mapping of the fluid regimes in this multi-port microchannel was accomplished by developing functions that describe the fraction of time or the probability that the fluid exists in an observed flow configuration.

Wambsganss et al. (1991) studied air-water flow in rectangular mini channels (Dh = 5.45 mm, α = 6). A comparison of existing flow pattern maps for circular pipes, capillary tubes and larger rectangular channels leaded to the conclusion that, while qualitative agreement exists, these maps were not generally applicable to rectangular microchannels on a quantitative basis. They suggested that flow pattern transitions in small rectangular channels should be determined independently.
Table 4-1 Summary of flow regime studies in mini/micro channels

<table>
<thead>
<tr>
<th>Reference</th>
<th>Geometry</th>
<th>Hydraulic diameter (mm)</th>
<th>Experimental conditions</th>
<th>Testing Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wolk et al. (2000)</td>
<td>Circular, rhombic, equilateral triangular and rectangular ($\alpha$: 0.7 ~ 0.97)</td>
<td>5.9 ~ 6.3</td>
<td>upward vertical</td>
<td>Air-water</td>
</tr>
<tr>
<td>Wambsganss et al. (1991)</td>
<td>Rectangular ($\alpha$: 6)</td>
<td>5.45</td>
<td>G = 50 ~ 2000 kg/m²s: x = 0 ~ 1</td>
<td>Air-water</td>
</tr>
<tr>
<td>Coleman and Garimella, 1999</td>
<td>Circular and rectangular ($\alpha$: 0.725)</td>
<td>1.3 ~ 5.5</td>
<td>$U_{GS} = 0.1 ~ 100$ m/s $U_{LS} = 0.01 ~ 10$ m/s</td>
<td>Air-water</td>
</tr>
<tr>
<td>Nino (2002)</td>
<td>Rectangular</td>
<td>1.58</td>
<td>G = 50 to 300 kg/s.m²: x = 0.1 ~ 0.9</td>
<td>R134a, R410A, air-water</td>
</tr>
<tr>
<td>Triplett et al. (1999b)</td>
<td>Circular</td>
<td>1.1 ~ 1.45, 1.09 ~ 1.49</td>
<td>$U_{GS} = 0.02 ~ 80$ m/s $U_{LS} = 0.02 ~ 8$ m/s</td>
<td>Air-water</td>
</tr>
<tr>
<td>Coleman, 2000</td>
<td>Circular and rectangular ($\alpha$: 0.5 ~ 2)</td>
<td>1.0 ~ 4.91</td>
<td>G = 150 ~ 750 kg/m²s: x = 0 ~ 1 Condensation</td>
<td>R134a</td>
</tr>
<tr>
<td>Damianides and Westwater (1987, 1988)</td>
<td>Circular</td>
<td>1.0 ~ 5.0</td>
<td>$U_{GS} = 1.05 ~ 101.2$ m/s $U_{LS} = 0.0084 ~ 8.62$ m/s</td>
<td>Air-water</td>
</tr>
<tr>
<td>Yang and Shieh, 2001</td>
<td>Circular</td>
<td>1.0 ~ 3.0</td>
<td>$U_{GS} = 0.01 ~ 100$ m/s $U_{LS} = 0.005 ~ 5$ m/s</td>
<td>R134a, Air-water</td>
</tr>
<tr>
<td>Lin et al. (1999)</td>
<td>Circular</td>
<td>1.0 ~ 2.36.0</td>
<td>Vertical</td>
<td>Air-water</td>
</tr>
<tr>
<td>Xu et al. (1999)</td>
<td>Rectangular ($\alpha$:12 ~ 40)</td>
<td>0.585 ~ 1.846</td>
<td>Vertical</td>
<td>Air-water</td>
</tr>
<tr>
<td>Serizawa et al. (2002)</td>
<td>Circular</td>
<td>0.020 ~ 0.100</td>
<td>$U_{GS} = 0.0012 ~ 295.3$ m/s $U_{LS} = 0.003 ~ 17.52$ m/s</td>
<td>Steam-water, air-water</td>
</tr>
<tr>
<td>Kawahara et al. (2002)</td>
<td>Circular</td>
<td>0.100</td>
<td>$U_{GS} = 0.1 ~ 60$ m/s $U_{LS} = 0.02 ~ 4$ m/s</td>
<td>Water-nitrogen</td>
</tr>
</tbody>
</table>

Xu et al. (1999) investigated adiabatic air-water flow in vertical rectangular channels 12 mm wide with narrow gaps of 0.3, 0.6, 1.0 mm ($D_h = 0.585 ~ 1.846$ mm, $\alpha \geq 12$). The flow regimes for gaps of 1.0 and 0.6 mm were found to be similar to those in the existing literature, which can be classified into bubbly flow, slug flow, churn-turbulent flow and annular flow. With the decrease of the channel gap, the transition from one flow regime to another occurred at smaller gas flow rates. However, flow regimes for micro-gap of 0.3 mm were quite different from the previous studies: bubbly flow was never observed even at very low gas flow rates, and the flow regimes were classified into cap-bubble flow, slug-droplet flow, churn flow, and annular-droplet flow.

Kawahara et al. (2002) observed the flow patterns for nitrogen-water flow in a 100 µm diameter circular tube made of fused silica. The two-phase flow patterns observed were mainly intermittent and semi-annular flows. Bubbly and churn flow patterns were not observed. A flow pattern map was developed based on the probability of appearance of each flow type.

Serizawa et al. (2002) visualized the flow patterns of air-water flow in circular tubes of 20, 25 and 100 µm i.d. and of steam-water flow in a 50 µm i.d. circular tube. Several distinctive flow patterns, namely, dispersed bubbly flow, gas slug flow, liquid ring flow, liquid lump flow, annular flow, frothy or wispy annular flow, rivulet flow and liquid droplets flow were identified. The general trend of flow transition followed the Mandhane’s (1974) prediction. The result showed that two-phase flow patterns were sensitive to the surface conditions of the inner wall of the test tube.
The literature review reveals that only a small number of literatures are available so far and the current knowledge is still limited for two-phase flow regime in microchannels. One of the questions is whether the flow patterns in microchannels are different from those found in conventional size tubes. One of the biggest differences between the ordinary size channel and mini/micro channel regimes is the role of surface tension. In large channels, gravity force dominates over surface tension force even though they both play important roles. In the mini/micro channels, however, surface tension forces are not only dominant but gravity force has been shown not to be a factor (Suo and Griffith 1964). The predominance of surface tension force on buoyancy leads to the insensitivity of two-phase hydrodynamics to channel orientation, and leads to the suppression of velocity difference between the two phases in the absence of significant acceleration in the low vapor quality region (Ghiaasiaan and Abdel-Khalik, 2001).

4.3 Results and Discussions

The experimental facility, instrumentation, experimental procedure, data reduction and uncertainties, as well as the characterization of the microchannel test sections were described in section 2.3 and in Tu and Hrnjak (2003).

4.3.1 Single-phase Validation

In section Chapter 3, it has been concluded that the single-phase flow friction factor in microchannels approaches the theoretical predictions in the laminar region. Hence, the friction factor data for liquid flow were obtained before the two-phase flow experiments in order to check the reliability of the experimental methods, as well as the correctness of channel geometry measurement.

As seen in Figure 4-1, the measured friction factor was predicted with theoretical results of Hartnett and Kostic (1989), Equation (3-1), with deviations within ±7%, for all the five data points presented. The error-bars associated with the friction factors are also shown in Figure 4-1. From this it was concluded that the experimental facility and test section was sufficient for further study in two-phase.

Figure 4-1: Single-phase friction factor in a 92.0 µm × 951.6 µm microchannel
4.3.2 Flow Patterns and Regimes

Some of the two-phase flow pictures are shown in Figure 4-2. The flow visualization results suggest that the flow patterns in the microchannel are similar to that in minichannels. A description of the flow regimes/patterns labels used here can be found in Coleman and Garimella (1999). Three major flow regimes were identified: including dispersed, intermittent, and annular. The intermittent flow is further subdivided into two flow patterns: the plug flow pattern and the slug flow pattern. Examples of the flow patterns observed are shown in Figure 4-3.

(a) $G = 279 \text{ kg/m}^2\text{-s}$

(b) $G = 461 \text{ kg/m}^2\text{-s}$

Figure 4-2 Flow visualization pictures in a 92.0 $\mu$m × 951.6 $\mu$m microchannel
(a) Dispersed regime: Bubble flow pattern

(b) Intermittent regime: Plug flow pattern

(c) Intermittent regime: Slug flow pattern

(d) Annular flow

Figure 4-3 Flow regimes and patterns in a 92.0 µm × 951.6 µm microchannel

Figure 4-4 shows the flow regime map for the present study. It is to note that the data points on top of one another in Figure 4-4 illustrate the co-existence of two flow patterns at the same test condition. The bubble flow and slug/plug flow co-exist at very low vapor qualities (less than 7%). The slug/plug regime extends from quality of about 10% to 40%. After that, the annular flow coexists with the slug/plug flow. At higher vapor qualities (greater than 60%), the intermittent flow disappears and the annular flow is the only flow pattern. The flow regime transitions for two mass fluxes are almost the same. The only difference is that the transition to annular flow occurs a little earlier for the high mass flux flow. Also shown in Figure 4-4 is the comparison with the result of Coleman’s (2000) for R134a two-phase flow in a 1×1 mm minichannel. It appears that the intermittent flow (slug/plug) occurred in a much wider range of vapor quality in the current study than that in Coleman (2000), which indicates the larger effect of surface tension in microchannels than in minichannels.

The following dimensionless parameters are important for adiabatic gas-liquid two-phase flow in a pipe,

\[ Eo = \frac{(\rho_l - \rho_v)gD_h^2}{\sigma} \quad (4-1) \]

\[ We_{LS} = \frac{U_{LS}^2D_h\rho_l}{\sigma} \quad (4-2) \]

\[ We_{GS} = \frac{U_{GS}^2D_h\rho_v}{\sigma} \quad (4-3) \]

\[ Re_{LS} = \frac{U_{LS}D_h}{\nu_l} \quad (4-4) \]

\[ Re_{GS} = \frac{U_{GS}D_h}{\nu_v} \quad (4-5) \]

where \( Eo \), \( We_{GS} \), \( We_{LS} \), \( Re_{GS} \) and \( Re_{LS} \) stand for Eotvos number, Weber number based on gas superficial velocity, Weber based on liquid superficial velocity, Reynolds number based on gas superficial velocity, and Reynolds number based on liquid superficial velocity, respectively. The \( Eo \) number is the ratio of gravity over the surface tension forces, and it is negligible for flows in microchannels.

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Figure 4-4 Flow regime in a 92.0 × 951.6 µm microchannel vs. Coleman’s (2000) results in a 1×1 mm channel

Akbar et al. (2003) developed a simple two-phase flow regime map based on the Weber numbers following the methodology previously applied in microgravity. The flow regime map was based on air-water flow visualization results in circular and near-circular microchannels with $D_h \approx 1$ mm and the entire flow regime map was divided into four regions:

(a) The surface-tension-dominated region, including bubbly, plug, and slug

\[ \text{For } We_{LS} \leq 3.0, \quad We_{GS} \leq 0.11We_{LS}^{0.315} \]
\[ \text{For } We_{LS} > 3.0, \quad We_{GS} \leq 1.0 \]  
(4-6)

(b) Inertia-dominated region I, including annular and wavy-annular regimes

\[ We_{GS} \geq 11.0We_{LS}^{0.14} \]
\[ We_{LS} \leq 3.0 \]  
(4-7)

(c) Inertia-dominated region II, including the dispersed flow regimes

\[ We_{GS} \geq 1.0 \]
\[ We_{LS} > 3.0 \]  
(4-8)

(d) Transition region

The above transition lines are plotted in Figure 4-5 and compared with the current results. The data shows that the transition from the surface tension dominated flow (bubbly, plug/slug) occurred at larger $We_{GS}$ number in the current study and the transition region is much smaller than the predictions. The result suggests a wider range of surface tension dominated flows in microchannels than that in minichannels. The transition to annular regime line $We_{GS} \geq 11.0We_{LS}^{0.14}$ approximately represents the transition from surface tension dominated flow to the
transition/Annular flow. The difference between the current flow regime and the prediction of Akbar et al. (2003) can be attributed to the difference in the channel size, the geometries and the fluid.

Figure 4-5 Flow pattern transition in a 92.0 × 951.6 µm microchannel vs. Akbar et al. (2003) correlation

4.4 Summary and Conclusions
Experiments were performed to visualize adiabatic R134a flow in a single-channel test section with cross-section of 92.0 × 951.6 µm and total length of 48 mm. During the experiment, the mass flux ranged from 279 to 461 kg/m²-s and vapor quality changed from 0.05 to 0.95.

The flow regimes were found to be similar to those in minichannels that can be classified into bubble flow, slug/plug flow, and annular flow. The bubble flow only existed when the vapor quality was very low.

Compared with the result of Coleman (2000) for R134a flow in a 1×1 mm channel, the intermittent flow (slug/plug) occurred in a much wider range of vapor quality, which indicates the larger effect of surface tension in microchannels than in minichannels. The flow regime transition was compared with the Akbar et al. (2003) correlation based on the Weber numbers. The result suggests wider range of surface tension dominated flow in microchannels than in minichannels. The $W_{e_{GS}} \geq 11.0W_{e_{LS}}^{0.14}$ line in the Akbar et al. (2003) correlation gives a qualitative differentiation of the inertia dominated regime and the surface tension dominated regime in the current study.

References


Chapter 5. Adiabatic Two-Phase Flow Pressure Drop

5.1 Chapter Overview
This chapter describes the experimental results for adiabatic two-phase flow pressure drop of R134a in five rectangular channels with hydraulic diameters varying from 69.5 to 304.7 µm and with aspect ratios changing from 0.09 to 0.24. The parameter ranges examined for two-phase flow are mass flux $G = 101.7 - 793.8$ kg/m$^2$-s; vapor quality $x = 0.02 - 0.98$; and saturation temperature $T_{sat} = 23.2 - 28.9$ °C. The experimental data were compared with twelve existing correlations. The homogeneous model and the Mishima and Hibiki (1996) correlation gave reasonable predictions with mean deviation in the range of 16.5% – 18.8%. Based on the current experimental data, a new correlation, Equation (5-52), was developed for two-phase flow pressure drop in microchannels, in which the parameter $C$ in Lockhart-Martinelli type correlation was correlated as a function of three nondimensional parameters considering the forces and flow regimes in microchannels.

5.2 Literature Review
Table 5-1 summarizes the previous investigations on two-phase flow pressure drop in minichannel and microchannels.


Nino et al. (2001, 2002) compared their experimental data ($D_h = 1 ~ 1.5$ mm) with minichannel correlations. They found that Zhang and Kwon (1999) correlation gave the best prediction. The Tran et al. (2000) correlation overpredicted the pressure drop, but gave a reasonable trend.

Coleman (2000) measured two-phase flow pressure drops in a set of circular and non-circular tubes (triangular, square, rectangular, barrel, and “N” shaped) with hydraulic diameters ranging from 0.424 mm to 4.91 mm. The author compared their experimental data of round tubes ($D_h = 0.5 ~ 4.9$ mm) with popular large tube correlations as well as mini channel correlations. For the micro tubes ($D_h = 0.762, 0.508$ mm), it appears that the Friedel (1979) correlation predicted the experimental data better than other models.

Stanley et al. (1997) showed that the available semi-empirical relations substantially overpredicted the measured pressure drops in rectangular aluminum channels ($D_h = 56 - 256$ µm, $\alpha = 0.5 - 1.5$). The calculated pressure drops based on the homogeneous flow model gave reasonable predictions, while the separated flow model gave very poor results, particularly for the high pressure drops presented. However, the experimental range was very limited, since most of the data were within $x < 0.10$. 
Table 5-1 Summary of two-phase pressure drop studies in mini/micro channels

<table>
<thead>
<tr>
<th>Reference</th>
<th>Geometry</th>
<th>$D_h$ ($\mu m$)</th>
<th>Experimental conditions</th>
<th>Heat source/sink</th>
<th>Testing Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wambgsan et al., 1992</td>
<td>Rectangular ($\alpha = 6, 0.17$)</td>
<td>5450</td>
<td>$G = 50 \sim 2000 , \text{kg/m}^2\text{s}$ $x = 0 \sim 1$</td>
<td>Adiabatic</td>
<td>Air-water</td>
</tr>
<tr>
<td>Tran et al., 2000</td>
<td>Circular</td>
<td>2400 - 2920</td>
<td>$G = 33 \sim 832 , \text{kg/m}^2\text{s}$ $x = 0 \sim 1$</td>
<td>Evaporation</td>
<td>R12, R113, R134a</td>
</tr>
<tr>
<td>Zhang and Kwon, 1999</td>
<td>Circular</td>
<td>2130 - 6200</td>
<td>$G = 200 \sim 1000 , \text{kg/m}^2\text{s}$ $x = 0.2 \sim 0.9$</td>
<td>Adiabatic</td>
<td>R134a, R22, R404A</td>
</tr>
<tr>
<td>Yan and Lin, 1998</td>
<td>Circular</td>
<td>2000</td>
<td>$G = 50 \sim 200 , \text{kg/m}^2\text{s}$ $x = 0.1 \sim 0.9$</td>
<td>Condensation</td>
<td>Evaporation</td>
</tr>
<tr>
<td>Yang and Webb, 1996</td>
<td>Rectangular</td>
<td>1560 - 2640</td>
<td>$G = 400 \sim 1400 , \text{kg/m}^2\text{s}$ $x = 0.1 \sim 0.9$</td>
<td>Adiabatic</td>
<td>R12</td>
</tr>
<tr>
<td>Nino et al., 2001, 2002</td>
<td>Rectangular</td>
<td>1020 - 1540</td>
<td>$G = 50 \sim 300 , \text{kg/m}^2\text{s}$ $x = 0 \sim 1$</td>
<td>Adiabatic</td>
<td>Air-water</td>
</tr>
<tr>
<td>Mishima and Hibiki, 1996</td>
<td>Circular</td>
<td>1000 - 4000</td>
<td>$U_{GS} = 0.05 \sim 18.7 , \text{m/s}$ $U_{LS} = 0.03 \sim 2.39 , \text{m/s}$</td>
<td>Adiabatic</td>
<td>Air-water</td>
</tr>
<tr>
<td>Lee and Lee, 2001a</td>
<td>Rectangular ($\alpha = 0.02 \sim 0.2$)</td>
<td>800 - 6667</td>
<td>$U_{GS} = 0.05 \sim 18.7 , \text{m/s}$ $U_{LS} = 0.03 \sim 2.39 , \text{m/s}$</td>
<td>Adiabatic</td>
<td>Air-water</td>
</tr>
<tr>
<td>Coleman, 2000</td>
<td>Circular, N-shaped, barrel, triangular, rectangular ($\alpha = 0.5 \sim 1$)</td>
<td>424 - 4910</td>
<td>$G = 150 \sim 750 , \text{kg/m}^2\text{s}$ $x = 0 \sim 1$</td>
<td>Adiabatic</td>
<td>Evaporation</td>
</tr>
<tr>
<td>Stanley et al., 1997</td>
<td>Rectangular ($\alpha = 0.5 \sim 1.5$)</td>
<td>56 - 256.9</td>
<td>$Re = 30 \sim 10,000$ $x = 0 \sim 0.45$</td>
<td>Adiabatic</td>
<td>Water-gas</td>
</tr>
<tr>
<td>Moriyama et al., 1992</td>
<td>Rectangular ($\alpha = 0.17 \times 10^{-3} \sim 3.3 \times 10^{-3}$)</td>
<td>10 - 200</td>
<td>$U_{GS} = 0.1 \sim 7 , \text{m/s}$ $U_{LS} = 0.002 \sim 0.22 , \text{m/s}$</td>
<td>Adiabatic</td>
<td>N2-R113</td>
</tr>
<tr>
<td>Kawahara et al., 2002</td>
<td>Circular</td>
<td>100</td>
<td>$U_{GS} = 0.1 \sim 60 , \text{m/s}$ $U_{LS} = 0.002 \sim 4 , \text{m/s}$</td>
<td>Adiabatic</td>
<td>Water-Nitrogen</td>
</tr>
</tbody>
</table>

Moriyama et al. (1992) investigated an N$_2$-R113 two-phase flow in microchannels of 30 mm in width and 5 - 100 $\mu m$ in thickness. They observed that for a gap size less than 25 $\mu m$, the Lockhart-Martinelli equation with $C = 0$ agreed well with the experiments. An empirical correlation was developed based on the experimental results.

Recently, Kawahara et al. (2002) characterized water-nitrogen two-phase flow in a 100 $\mu m$ diameter circular tube. The homogeneous model with average viscosity defined by Dukler et al. (1964) gave reasonable predictions ($\pm 20\%$) to the experimental data. Good predictions (within $\pm 10\%$) were obtained with the Lee and Lee (2001) correlation.

The literature review indicates that there have been a considerable number of two-phase flow pressure drop studies in minichannels, but very few studies are available for channels with a diameter close to or less than 100 $\mu m$. It is not clear from the open literature what correlation(s) shall be used with confidence for refrigerant two-phase flow in microchannels.
5.3 Experimental Results and Analysis

The experimental facility, instrumentation, experimental procedure, data reduction and uncertainties, as well as the characterization of the microchannel test sections were described in section 2.3 and in Tu and Hrnjak (2002, 2003).

The experimental results for test section #1 ($D_h = 304.7 \, \mu m$, $H/W = 0.10$) is shown in Figure 5-1. For lower mass fluxes ($G = 102 \, kg/m^2-s$ and $G = 149 \, kg/m^2-s$), the pressure drop is not a very strong function of vapor quality. When the mass fluxes are higher ($G = 303 \, kg/m^2-s$), the pressure drop increases sharply as the vapor quality increases.

![Figure 5-1](image1)

**Figure 5-1 Two-phase flow pressure gradient for test section #1**

![Figure 5-2](image2)

**Figure 5-2 Two-phase flow pressure gradient for test section #2**

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Figure 5-2 shows the experimental results for two-phase flow pressure drop in test section #2 \((D_h = 150.0 \mu m, H/W = 0.24)\) for three different mass fluxes, \(G = 302 \text{ kg/m}^2\cdot\text{s}, G = 449 \text{ kg/m}^2\cdot\text{s}\) and \(G = 599 \text{ kg/m}^2\cdot\text{s}\). All the experimental data demonstrate the tendency that the pressure drop increases as the vapor quality or mass flux increase. In addition, it is indicated that when the vapor quality is lower \((x < 0.3)\), the pressure drop increases moderately with the increase of vapor quality. However, further increase in vapor quality corresponds to a sharp increase of pressure drop until it reaches a vapor quality of about 0.8. At that point, the pressure drop remains almost constant and varies slightly with the increase of vapor quality.

![Figure 5-2 Experimental results for two-phase flow pressure drop in test section #2](image)

Figure 5-3 Two-phase flow pressure gradient for test section #3

Figure 5-3 shows the experimental results for test section #3 \((D_h = 141.1 \mu m, H/W = 0.09)\) for five different mass fluxes, \(G = 158 \text{ kg/m}^2\cdot\text{s}, G = 315 \text{ kg/m}^2\cdot\text{s}\) and \(G = 471 \text{ kg/m}^2\cdot\text{s}, G = 624 \text{ kg/m}^2\cdot\text{s}\) and \(G = 785 \text{ kg/m}^2\cdot\text{s}\). When the mass flux is low \((G \leq 158 \text{ kg/m}^2\cdot\text{s})\), the two-phase flow pressure drop is not a strong function of vapor quality. For higher mass fluxes \((G \geq 315 \text{ kg/m}^2\cdot\text{s})\), the two-phase flow pressure drop increases sharply when the vapor quality increased. Overall, the pressure drop is a strong function of mass fluxes and the higher the mass flux, the higher the pressure drop.

The experimental results for the two test sections with the smallest hydraulic diameters, test section #4 \((D_h = 104 \mu m, H/W = 0.17)\) and test section #5 \((D_h = 69 \mu m, H/W = 0.09)\), are shown in Figure 5-4 and Figure 5-5, respectively. The low mass flux experimental data are not available for these two test sections because such data correspond to extremely low mass flow rates, which are very difficult to achieve with the current experimental facilities. The results for both test sections show similar trends that have been observed in larger test sections, that is, the pressure drop increases as the vapor quality and the mass flux increase. When the vapor quality is lower \((x < 0.2)\), the pressure drop increases moderately with the increase of vapor quality. However, a further increase in vapor quality corresponds to a sharp increase of the pressure drop.

![Figure 5-3 Experimental results for test section #3](image)
5.4 Comparison with Correlations

In this section, the present experimental data are compared with twelve existing correlations. These correlations are used for comparison due to either their frequent usage in conventional channels, or the fact that they were developed based on the experimental data in mini/micro channels.

The homogeneous model and the separated flow model are the most important approaches for developing two-phase flow pressure drop correlations. The homogeneous model considers the two phases to flow as a single phase possessing means fluid properties. The separated flow model considers the phases to be artificially segregated.
into two streams. Both approaches rely entirely on single-phase friction factors. Hence, this section starts with developing empirical equations for single-phase friction factor in the microchannel test sections.

5.4.1 Empirical Equations for Single-phase Friction Factor

As shown in Figure 5-6 to Figure 5-10, empirical correlations were developed for the five microchannels that have been characterized for single-phase flow pressure drops in Chapter 3. In these correlations, the single-phase flow was divided into three regions: (1) the laminar region, where the friction factor was expressed in the form of \( f = \frac{\text{const}}{Re} \), (2) the transition region, \( f = C_1Re^\text{const} \), and (3) the turbulent flow region, \( f = C_2Re^{-\text{const}} \). All the coefficients are constant values determined from the regression of the experimental data.

![Figure 5-6 Single-phase friction factor empirical correlations for test section #1](image1)

![Figure 5-7 Single-phase friction factor empirical correlations for test section #2](image2)
Reynolds Number, Re
Friction Factor, f

Re ≥ 1,570
f = 0.2036Re^{-0.1441}

Re < 1,570
f = 92.82/Re

Test section #3: Dh = 141.1 µm, H/W = 0.09

Figure 5-8 Single-phase friction factor empirical correlation for test section #3

Reynolds Number, Re
Friction Factor, f

Re ≥ 3,000
f = 0.1692Re^{-0.1671}

2290 ≤ Re < 3000
f = 0.0169Re^{0.1176}

2290 ≤ Re < 3000
f = 76.0/Re

Test section #4: Dh = 104.1 µm, H/W = 0.17

Figure 5-9 Single-phase friction factor empirical correlations for test section #4
5.4.2 Conventional Separated Flow Models

5.4.2.1 Lockhart-Martinelli (1949)

The two-phase multiplier method is the most common technique for correlating two-phase frictional pressure drop. The Lockhart-Martinelli (1949) equation is the most typical of this type of correlations, in which the two-phase flow pressure drop is expressed as

\[
\left( \frac{dp}{dz} F \right)_{zp} = \phi_t^2 \left( \frac{dp}{dz} F \right)_{z_l}
\]

(5-1)

where \( \phi_t^2 \) is the two-phase flow frictional multiplier. A widely used correlation for the frictional multiplier is

\[
\phi_t^2 = 1 + \frac{C}{X} + \frac{1}{X^2}
\]

(5-2)

The Martinelli parameter, \( X \), is defined as a function of the frictional pressure gradient assuming liquid alone flow, \( \left( \frac{dp}{dz} F \right)_{z_l} \), and the frictional pressure gradient assuming gas alone flow, \( \left( \frac{dp}{dz} F \right)_{z_v} \).

\[
X = \left[ \frac{\left( \frac{dp}{dz} F \right)_{z_l}}{\left( \frac{dp}{dz} F \right)_{z_v}} \right]^{1/2}
\]

(5-3)

\[
\left( \frac{dp}{dz} F \right)_{z_l} = f_l \frac{1}{D_k} \frac{G^2 (1-x)^2}{2 \rho_l}
\]

(5-4)

\[
\left( \frac{dp}{dz} F \right)_{z_v} = f_v \frac{1}{D_k} \frac{G^2 x^2}{2 \rho_v}
\]

(5-5)
The parameter $C$ in the Lockhart-Martinelli (1949) equation has the following values, as listed in Table 5-2.

<table>
<thead>
<tr>
<th>Liquid phase</th>
<th>Gas phase</th>
<th>$C$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>20</td>
</tr>
<tr>
<td>Laminar</td>
<td>Turbulent</td>
<td>12</td>
</tr>
<tr>
<td>Turbulent</td>
<td>Laminar</td>
<td>10</td>
</tr>
<tr>
<td>Laminar</td>
<td>Laminar</td>
<td>5</td>
</tr>
</tbody>
</table>

As shown in Figure 5-11, the Lockhart-Martinelli correlation significantly overpredicts almost all the experimental data for the entire vapor quality region. The correlation gives better prediction for the largest channel, test section #1, than other smaller channels.

![Figure 5-11 Two-phase pressure-drop data vs. Lockhart-Martinelli (1949) predictions](image)

5.4.2.2 Chisholm (1973)

The Chisholm (1973) correlation is an adaptation of the Lockhart-Martinelli (1949) correlation. It is expressed as

$$\left( \frac{dp}{dz} \right)_{tr} = \phi_{lo}^2 \left( \frac{dp}{dz} \right)_{lo}$$  \hspace{1cm} (5-6)

where $\left( \frac{dp}{dz} \right)_{lo}$ is the frictional pressure gradient assuming total flow to be liquid. The multiplier $\phi_{lo}^2$ is expressed as

$$\phi_{lo}^2 = 1 + \left( \gamma^2 - 1 \right) \left( B \cdot x^{\frac{1-n}{2}} \left( \left( 1-x \right)^{2-n} + x^{2-n} \right) \right)$$  \hspace{1cm} (5-7)

In Equation (5-7), the liquid-only-vapor-only Martinelli parameter, $\gamma$, is the ratio of frictional pressure gradient assuming total flow to be gas (or vapor) and the friction pressure gradient assuming total flow to be liquid.
\[
\gamma^2 = \left( \frac{\frac{dp}{dz}}{\frac{dp}{dz}} \right)_{lo}
\]  
(5-8)

\[
\left( \frac{dp}{dz} \right)_{lo} = f \frac{1}{D_h} \frac{G^2}{2 \rho_f}
\]  
(5-9)

\[
\left( \frac{dp}{dz} \right)_{vo} = f \frac{1}{D_h} \frac{G^2}{2 \rho_v}
\]  
(5-10)

The values of B and n₁ in Equation (5-7) are calculated using the following equations:

\[
B = \begin{cases} 
\frac{55}{\gamma^{0.5}} & \gamma \leq 9.5 \\
\frac{520}{\gamma \cdot G^{0.5}} & 9.5 < \gamma \leq 28 \\
\frac{15000}{\gamma^2 \cdot G^{0.5}} & \gamma > 28 
\end{cases}
\]  
(5-11)

\[
n_1 = \begin{cases} 
1 & \text{Re}_{lo} \leq 2,300 \\
0.25 & 2300 < \text{Re}_{lo} \leq 20,000 \\
0.2 & \text{Re}_{lo} > 20,000 
\end{cases}
\]  
(5-12)

As shown in Figure 5-12, the Chisholm (1973) correlation significantly overpredicts almost all the experimental data for all the test sections. The error is larger for data with a low vapor quality and smaller when the quality approaches one. The error is not a strong function of mass fluxes or channel diameters.

![Figure 5-12 Two-phase pressure-drop data vs. Chisholm (1973) predictions](image)

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5.4.2.3 Friedel (1979)
Another widely used correlation, which was proposed by Friedel (1979), is based on vast database covering
an extensive parameter range. The multiplier in Equation (5-6) is defined as follows:

\[
\phi_{lo}^2 = A_1 + \frac{3.24A_2A_3}{Fr^{0.045} We^{0.035}}
\] (5-13)

\[
A_1 = (1 - x)^2 + x^2 \frac{\rho_{l} f_w}{\rho_{v} f_{lo}}
\] (5-14)

\[
A_2 = x^{0.78} (1 - x)^{0.224}
\] (5-15)

\[
A_3 = \left( \frac{\rho_{l}}{\rho_{v}} \right)^{0.91} \left( \frac{\mu_{v}}{\mu_{l}} \right)^{0.19} \left( 1 - \frac{\mu_{v}}{\mu_{l}} \right)^{0.7}
\] (5-16)

In Equation (5-13), the Froude (Fr) and Weber (We) numbers are defined as follows:

\[
Fr = \frac{G^2}{gD\rho_{avg}^2}
\] (5-17)

\[
We = \frac{G^2 D}{\rho_{avg} \sigma}
\] (5-18)

where the two-phase average density is defined in Equation (2-15) and \( \sigma \) is the surface tension.

As shown in Figure 5-13, the Friedel (1979) correlation significantly overpredicts all the experimental data. When used in conventional size channels, this correlation has been taken as one of the most accurate two-phase pressure drop correlations since it was obtained by optimizing the equation based on a large data base of two-phase pressure drop measurements. Obviously, the current application is out of the range of the data base, which is the main reason why this correlation does not work.

![Figure 5-13 Two-phase pressure-drop data vs. Friedel (1979) predictions](image-url)
5.4.3 Minichannel Correlations

5.4.3.1 Tran et al. (2000)

Tran et al. (2000) measured refrigerant (R-134a, R-12, and R-113) flow in minichannels with hydraulic diameters in the range 2.4 – 2.92 mm. They developed a new correlation based on the B-coefficient method, taking into account the effect of surface tension and channel size. In this correlation, the multiplier in Equation (5-7) is defined as follows.

\[
\phi_{lo}^2 = 1 + \left(4.3 \cdot \gamma^2 - 1\right) \cdot \left(N_{conf} \cdot x^{0.875} \cdot (1-x)^{0.875} + x^{1.75}\right)
\]  

(5-19)

In Equation (5-19), the confinement number is expressed as

\[
N_{conf} = \left(\frac{\sigma}{g(\rho_f - \rho_v)}\right)^{0.5} \cdot \frac{1}{D_h}
\]  

(5-20)

where \(\sigma\) and \(g\) denote the surface tension and the gravitational constant, respectively.

As shown in Figure 5-14, the Tran et al. (2000) correlation drastically overpredicts the experimental data and the trend is the smaller the channel the larger the error. This trend indicates that this correlation does not consider the effect of channel size on the multiplier very well.

5.4.3.2 Zhang and Kwon (1999)

Zhang and Kwon (1999) proposed a new correlation for two-phase friction in small diameter tubes by modifying the Friedel’s correlation. The multiplier in Equation (5-6) is defined as

\[
\phi_{lo}^2 = \left(1-x\right)^2 + 2.87 x^2 \left(\frac{P}{P_{crit}}\right)^{-1} + 1.68 x^{0.8} \left(1-x\right)^{0.875} \left(\frac{P}{P_{crit}}\right)^{-1.64}
\]  

(5-21)

where \(P\) is the saturation pressure and \(P_{crit}\) is the critical pressure.
As shown in Figure 5-15, the Zhang and Kwon (1999) correlation demonstrates very similar results as those of the Freidel’s correlation. The correlation significantly overpredicts the experimental data and the errors are not a strong function of channel diameters.

![Figure 5-15 Two-phase pressure-drop data vs. Zhang and Kwon (1999) predictions](image)

5.4.3.3 Yang and Webb (1996)

Yang and Webb (1996) provided a correlation based on the equivalent mass velocity concept proposed by Akers et al. (1959). The multiplier in Equation (5-6) is defined as

\[ \phi_{lo}^2 = 0.435 \text{Re}_{eq}^{0.12} \left(1 - x + x \left(\frac{\rho_l}{\rho_v}\right)^{0.5}\right)^2 \]  (5-22)

\[ \text{Re}_{eq} = \frac{G_{eq} D_h}{\mu_l} \]  (5-23)

\[ G_{eq} = G \left(1 - x + x \left(\frac{\rho_l}{\rho_v}\right)^{0.5}\right) \]  (5-24)

where \( G_{eq} \) is the equivalent mass flux of liquid proposed by Akers et al. (1959).

The experimental results are compared with the Yan and Webb (1996) correlation predictions, as shown in Figure 5-16. The correlation overpredicts the experimental data except for a few data points with very low vapor qualities. The trend is that the larger the vapor quality, the larger the error is, and the error is not a strong function of the channel size.
5.4.3.4 Yan and Lin (1998)

Yan and Lin (1998) proposed an empirical correlation for the frictional pressure drop as a function of the equivalent two-phase Reynolds number.

\[
\Delta P_{fp} = f_{fp} \frac{2L G^2}{\rho_{avg} D_h} 
\]

(5-25)

\[
f_{fp} = 0.11 \text{Re}^{0.1}_{eq} 
\]

(5-26)

As shown in Figure 5-17, the Yan and Lin (1998) correlation overpredicts almost all the experimental data. It is indicated that the error for a larger channel is smaller than that for a smaller channel.
Niño et al. (2002) tested adiabatic two-phase flow pressure drop of R410A, R134a, and air-water mixtures flowing in microchannels with hydraulic diameter of 1.54 mm and 1.02 mm. They found that the average kinetic energy method successfully predicted the pressure drop in the microchannels except in the region where fully annular flow regime is expected. The correlation can be expressed as

\[
\Delta P_{tp} = 0.045 \frac{L G^2}{2 \rho_{avg} D_h} 
\]

(5-27)

As shown in Figure 5-18 (a), this method overpredicts the current data points with vapor quality larger than 0.3, and underpredict the data points with vapor quality less than 0.2. In addition, it predicts the data in the largest channel (test section #1) better than that in the smaller channels.

Niño et al. (2002) also developed a new correlation for the annular flow region in microchannels, which is expressed as

\[
\phi_{vo} = \exp\left(-0.0083 A^3 - 0.047 A^2 - 0.0028 A + 0.1661\right)
\]

(5-29)

\[
A = \ln\left(\frac{X_t + \frac{1}{We^{1/3} \rho_v}}{\rho_i}\right)^{0.9}
\]

(5-30)

Figure 5-18 Two-phase pressure-drop data vs. Niño et al. (2002) predictions

As shown in Figure 5-18 (b), the correlation reasonably predicts the data in the largest channel (test section #1) with vapor quality larger than 0.3, but overpredicts the smaller test sections in this quality region. In addition, the correlation tends to underpredict the data points in the low quality region (\(x < 0.2\)).
5.4.3.6 Lee and Lee (2001)

Lee and Lee (2001) proposed a new correlation for two-phase flow pressure drop through rectangular channels \((D_h = 0.8 \sim 6.7 \text{ mm}, \alpha = 0.02 \sim 0.2)\). The parameter \(C\) in the Lockhart-Martinelli (1949) correlation was expressed in terms of three dimensionless parameters \(\lambda\), \(\psi\) and \(\text{Re}_{lo}\) as

\[
C = A \psi^q \text{Re}_{lo}^r
\]

(5-31)

where the constant, \(A\), and the exponent’s \(q\), \(r\) and \(s\) were determined through a data regression, and the values are listed in Table 5-3. The dimensionless number \(\psi\) represents the relative importance of the viscous and surface tension effects, which is expressed as

\[
\psi = \frac{\mu_j j}{\sigma}
\]

(5-32)

where \(j\) is the liquid slug velocity. The dimensionless parameter \(\lambda\) was defined as

\[
\lambda = \frac{\mu_j^2}{\rho_l \sigma D_h}
\]

(5-33)

Table 5-3 Constant and exponents in Lee and Lee (2001), Equation (5-31)

<table>
<thead>
<tr>
<th>Liquid phase</th>
<th>Gas phase</th>
<th>A</th>
<th>q</th>
<th>r</th>
<th>s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar</td>
<td>Laminar</td>
<td>6.833\times10^{-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\times10^{-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>

As shown in Figure 5-19 (a), it seems that the data points fall into two distinct groups. For data with lower vapor qualities or lower mass fluxes, the Lee and Lee (2001) correlation gave reasonable predictions to the experimental data, and most of the data were within ±40% of the model predictions. However, the data with higher vapor qualities or mass fluxes were predicted with an error between 50% and 100%.

In Figure 5-19 (b), the model predictions over the experimental data are plotted as a function of Reynolds number of gaseous phase alone, \(\text{Re}_g\). It is clear from Figure 5-19 (b) that the first group is associated with the region of \(\text{Re}_g < 2,000\), which lies in the Laminar-Laminar flow regime in Table 5-3. The Lee and Lee (2001) correlation gives reasonable predictions for this regime, but tends to underpredict the experimental data. Kawahara et al. (2002) reported similar results for water-nitrogen two-phase flow in a 100 \(\mu\text{m}\) diameter circular tube. They showed that good predictions (within ±10%) were obtained with the Lee and Lee (2001) correlation for the Laminar-Laminar flow regime. However, for the Liquid-Laminar-Gas-Turbulent region (\(\text{Re}_g > 2,000\)), the Lee and Lee (2001) correlation tends to overpredict the current experimental data. In this region, according to Table 5-3, Equation (5-31) is reduced to

\[
C = 0.06185 \text{Re}_{lo}^{0.557}
\]

(5-34)

The equation above indicates that the parameter \(C\) is only dependent on the liquid flow Reynolds number, \(\text{Re}_{lo}\), in the Liquid-Laminar-Gas-Turbulent region (\(\text{Re}_g > 2,000\)).
5.4.4 Microchannel Correlations

5.4.4.1 Moriyama et al. (1992)

In Moriyama et al. (1992), the multiplier in Equation (5-1) is expressed as a function of liquid phase Reynolds number, $\text{Re}_l$, and the Martinelli parameter, $X$.

\[
\phi_l^2 = 1 + \frac{K_M}{X^2}
\]  

\[
K_M = \begin{cases} 
0.9\text{Re}_l^{0.3}, & \text{Re}_l > 1.3 \\
1, & \text{Re}_l \leq 1.3
\end{cases}
\]

The hydraulic diameter of Moriyama et al. (1992) covers the current study, but they used extremely narrow channels, i.e. $\alpha << 1$ and different fluids (N$_2$ - R113). As shown in Figure 5-20, although this correlation performs better than most of the conventional and minichannel correlations mentioned previously, it still overpredicts the experimental data by about 10% ~ 200%.
5.4.4.2 Mishima and Hibiki (1996)

Mishima and Hibiki (1996) incorporated the results of Moriyama et al. (1992) and other investigators’ data in their data base, and observed that parameter $C$ in the Lockhart-Martinelli (1949) correlation decreases with decreasing channel sizes. They proposed a new correlation with $C$ as a function of the channel size

$$C = 21(1 - e^{-0.319D_h})$$

(5-37)

where the unit of $D_h$ is in millimeter.

As shown in Figure 5-21, this correlation gives reasonable predictions to the experimental data for all the test sections, and 96.6% of the data falls into ± 40% of the correlation predictions. The correlation tends to overpredict the data with low vapor qualities. Kawahara et al. (2002) reported that the Mishima and Hibiki (1996) correlation slightly overpredicted their experimental data of water-nitrogen two-phase flow in a 100 µm diameter circular tube.
Figure 5-21 Two-phase pressure-drop data vs. Mishima and Hibiki (1996) predictions

It can be also seen from Figure 5-21 that the error of Mishima and Hibiki (1996) correlation is not dependent on the channel size, which is expected since this correlation considered the effect of channel size on the parameter $C$.

5.4.5 Homogeneous Models

The homogeneous models assume equal vapor and liquid velocities, and the mixture is considered as homogeneous single-phase fluid with average fluid properties. In a homogeneous model, the frictional pressure drop is expressed as,

$$\left( \frac{dp}{dz} \right)_F = f^v_p \frac{1}{D_h} \frac{G^2}{2 \rho_{avg}}$$

where $(dp/dz)_F$ is the frictional pressure gradient, $D_h$ is the hydraulic diameter of the tube, $G$ is the mass flux of the two-phase flow. The mean density of the two-phase mixture, $\rho_{avg}$, is calculated assuming a homogeneous void fraction using Equation (2-15).

The two-phase friction factor, $f^v_p$, is calculated using a normal single-phase friction factor correlation with the Reynolds number, $Re$, replaced by the two-phase flow Reynolds number, $Re^p$, that is defined in Equation (5-39).

$$Re^p = \frac{GD_h}{\mu_{avg}}$$

The average viscosity of the two-phase mixture, $\mu_{avg}$, is a function of the vapor quality and the properties of each phase. Numerous correlations have been developed for the average viscosity calculation, among which the equations proposed by McAdams et al. (1942) and Dukler et al. (1964) are the most frequently used.
In conventional channels, the homogeneous flow assumption is only suitable for bubble flow and mist flow where the slip ratio is close to one. Therefore, the foregoing method for calculating two-phase frictional pressure drop is inaccurate in most applications.

5.4.5.1 McAdams et al. (1942)

The average viscosity of the two-phase mixture proposed by McAdams et al. (1942) is expressed as

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

(5-40)

The prediction of this correlation is compared with the experimental data in Figure 5-22 (a). The correlation gives reasonable predictions and 87.2% of the data falls into ± 40% of the correlation predictions. However, it gives much higher (>50%) predictions to some of the data with low vapor qualities. It also tends to underpredict the data in the channel with a larger hydraulic diameter (test section #1).

![Figure 5-22 Two-phase pressure-drop data vs. homogeneous model predictions](image)

(a) McAdams et al. (1942)  
(b) Dukler et al. (1964)

5.4.5.2 Dukler et al. (1964)

The average viscosity of the two-phase mixture proposed by Dukler et al. (1964) is expressed as

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

(5-41)

The correlation gives reasonably good predictions, as shown in Figure 5-22 (b), with 84.2% of the data fall into ± 30% of the correlation predictions. Kawahara et al. (2002) reported similar results that the Dukler et al.’s (1964) model for the two-phase mixture predicted their experimental data of water-nitrogen two-phase flow in a 100 μm diameter circular tube within ± 20%.

It is also indicated in Figure 5-22 (b) that the homogeneous model with Dukler et al.’s (1964) definition of the two-phase mixture underpredicts the experimental data in a larger channel, test section #1.
5.4.6 Discussion of Correlation Predictions

The performance of all the twelve models evaluated above is summarized in Table 5-4, where the mean deviation and percentage of data within ± 30% range is based on all the 266 data points.

Table 5-4 Comparison of correlations with two-phase pressure drop results

<table>
<thead>
<tr>
<th>Correlations</th>
<th>Mean deviation (%)</th>
<th>Percentage of data within ± 30% range (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Homogeneous (Dukler et al., 1964)</td>
<td>16.5</td>
<td>84.2</td>
</tr>
<tr>
<td>Homogeneous (McAdams et al., 1942)</td>
<td>20.5</td>
<td>77.8</td>
</tr>
<tr>
<td>Mishima and Hibiki (1996)</td>
<td>18.8</td>
<td>83.8</td>
</tr>
<tr>
<td>Lee and Lee (2001)</td>
<td>28.5</td>
<td>62.4</td>
</tr>
<tr>
<td>Moriyama et al. (1992a)</td>
<td>77.4</td>
<td>10.5</td>
</tr>
<tr>
<td>Lockhart-Martinelli (1949)</td>
<td>150.1</td>
<td>0.75</td>
</tr>
<tr>
<td>Yan and Lin (1998)</td>
<td>253.5</td>
<td>2.6</td>
</tr>
<tr>
<td>Chisholm (1973)</td>
<td>197.1</td>
<td>3.4</td>
</tr>
<tr>
<td>Yang and Webb (1996)</td>
<td>227.2</td>
<td>7.9</td>
</tr>
<tr>
<td>Friedel (1979)</td>
<td>318.9</td>
<td>0</td>
</tr>
<tr>
<td>Zhang and Kwon (1999)</td>
<td>416.0</td>
<td>0</td>
</tr>
<tr>
<td>Tran et al. (2000)</td>
<td>1260.1</td>
<td>0</td>
</tr>
</tbody>
</table>

*aMean deviation = \( \frac{1}{N} \sum \frac{\Delta P_{\text{meas}} - \Delta P_{\text{pred}}}{\Delta P_{\text{meas}}} \times 100\% \)

As has been discussed earlier, the conventional correlations, the Chisholm (1973), the Friedel (1979), and the Lockhart-Martinelli (1949) predict the experimental data in microchannels with mean error more than 150.0 %. Similarly, the newly developed models for minichannels, Tran et al. (2000), Zhang and Kwon (1999), Yang and Webb (1996), and Yan and Lin (1998) failed in predicting the experimental data. The common feature of all these correlations is that they have empirically determined coefficients that are based on experimental data. The databases on which these coefficients were based typically consist of experimental data in channels with hydraulic diameter ten times larger than the current study. Thus, the extrapolation of those correlations to the microchannels creates large discrepancies.

On the other hand, the database for the Mishima and Hibiki (1996) correlation include experimental data in channels with hydraulic diameters less than 100 µm, and it considered the effect of channel size on the parameter \( C \). As a result, the Mishima and Hibiki (1996) correlation gives reasonably good predictions for the current experimental data.

The homogeneous models do not employ any empirical constants, but assume equal vapor and liquid velocities. This assumption generally does not hold for two-phase flow in large channels, thus, the application of homogeneous models is very limited in conventional size tubes. However, as has been pointed out by Ghiaasiaan and Abdel-Khalik (2001), the predominance of the surface tension force on buoyancy in microchannels leads to the dominance of nonseparated two-phase flow patterns (bubbly, slug/plug flow), which in turn leads to the suppression of velocity difference between the two phases. This observation explains why the homogeneous models give reasonable predictions to the data in microchannels.
Nevertheless, Kawahara et al. (2002) obtained the time-averaged void fraction data from the analysis of two-phase flow images. The results show that the time-averaged void fraction remained low even at high gas flow rates, indicating significantly larger slip ratios and a weaker momentum coupling between the phases, compared to the flows in minichannels. In addition, their flow visualization also demonstrated that the predominant occurrence of separated flow patterns in their microchannels. Their observation indicates that the separated model would be a better approach in modeling two-phase flow in microchannels. The Mishima and Hibiki (1996) correlation, which gives reasonably good predictions to the current data, is based on the separated model.

At first glance, it seems strange that two completely different approaches, the separated model and the homogeneous model, both work for correlating two-phase pressure drop data in microchannels. However, it is well known that the homogeneous model actually is a special case of the separated model – when the two-phases travel at the same speed the separated model reduces to the homogeneous model. For example, the following derivations show that for laminar-laminar flow with parameter C equals to zero, the Lockhart-Martinelli (1949) correlation that is based on the separated model reduces to the homogeneous model with Dukler’s (1964) definition of average viscosity.

The Lockhart-Martinelli (1949) correlation, Equation (5-1) and (5-2), can be rearranged as

\[
\left( \frac{dp}{dz} F \right)_{\text{q}} = \left( \frac{dp}{dz} F \right)_{\text{l}} + C \left[ \left( \frac{dp}{dz} F \right)_{\text{l}} \left( \frac{dp}{dz} F \right)_{\text{v}} \right]^{0.5} + \left( \frac{dp}{dz} F \right)_{\text{v}} \tag{5-42}
\]

The physical meaning of the Lockhart-Martinelli correlation is that the two-phase flow pressure drop is the sum of the pressure drop of liquid phase alone, \( \left( \frac{dp}{dz} F \right)_{\text{l}} \), the pressure drop of gaseous phase alone, \( \left( \frac{dp}{dz} F \right)_{\text{v}} \), and pressure drop caused by the phase interaction \( C \left[ \left( \frac{dp}{dz} F \right)_{\text{l}} \left( \frac{dp}{dz} F \right)_{\text{v}} \right]^{0.5} \). The parameter C indicates the extent of the interaction. For instance, the value of C, is larger (C = 20) in turbulent-turbulent flow region and lower (C = 5) in laminar-laminar region, as shown in Table 5-2.

For an extreme case where parameter C = 0, Equation (5-42) is reduced to

\[
\left( \frac{dp}{dz} F \right)_{\text{q}} = \left( \frac{dp}{dz} F \right)_{\text{l}} + \left( \frac{dp}{dz} F \right)_{\text{v}} \tag{5-43}
\]

Substituting Equation (5-4) and (5-5) into the above equation, we have

\[
\left( \frac{dp}{dz} F \right)_{\text{q}} = f_v \frac{1}{D_h} \frac{G^2 x^2}{2 \rho_v} + f_i \frac{1}{D_h} \frac{G^2 (1-x)^2}{2 \rho_i} \tag{5-44}
\]

Assuming laminar-laminar flow, the friction factors can be expressed as

\[
f_v = C_f \frac{C_f}{Re_v} \frac{1}{D_h} \mu_v \tag{5-45}
\]
After rearrangement, Equation (5-43) becomes

\[
\frac{dp}{dz} = \frac{C_f G}{2D_h^2} \left( \frac{x}{\rho_v} \mu_v + \frac{1-x}{\rho_l} \mu_l \right)
\]

\[
= \frac{C_f G}{GD_h} \times \frac{1}{D_h^2} \frac{G^2}{2\rho_{avg}}
\]

If the Dukler’s (1964) definition of average viscosity, Equation (5-41), and the two-phase flow Reynolds number \( Re_{tp} \), Equation (5-39), are used, the above equation can be further reduced to

\[
\frac{dp}{dz} = \frac{C_f}{Re_{tp}} \times \frac{1}{D_h} \frac{G^2}{2\rho_{avg}}
\]

This is exactly the homogeneous model, Equation (5-38), when the two-phase mixture is flowing in the laminar region.

The Mishima and Hibiki (1996) correlation, Equation (5-37), shows that the parameter \( C \) decreases with decreasing channel size. When the channel size is less than 100 \( \mu \)m, Equation (5-37) gives \( C \) value close to zero. Hence, the Mishima and Hibiki (1996) correlation approaches the homogeneous model with the Dukler’s (1964) definition of average viscosity, according to the above derivations. In addition, the observation of Moriyama et al. (1992) that the parameter \( C \) was zero for the microchannel with a gap size less than 25 \( \mu \)m is exactly the same as saying that the homogeneous model with the Dukler’s (1964) definition of average viscosity can be used to correlate the data.

The data sets used to develop the Lee and Lee (2001) correlation is in minichannels with hydraulic diameters in the range of 0.8 ~ 6.7 mm, which is about ten times the size of the channels in the current study. However, the correlation gives better predictions than the other minichannel correlations. In this method, the experimental data in the laminar-laminar region were correlated with parameter \( C \) as a function of three non-dimensional numbers \( \lambda, \psi \) and \( Re_{lo} \). For the experimental data in the other regions, the parameter \( C \) was correlated as a function of \( Re_{lo} \) only, as seen from Table 5-3. The reason behind this is that the flow patterns in the laminar-laminar flow region of Lee and Lee (2001) were mostly the surface tension dominated flows (plug/slug flow), but the flow patterns in other regions were not.

Nevertheless, the classification of flow regions based on whether the liquid phase and the vapor phase are laminar or turbulent does not necessarily represent the two-phase flow regimes, especially for microchannels ten times smaller than those of Lee and Lee (2001). Hence, in the laminar-turbulent region for the current study, the parameter \( C \) may still be a function of parameters \( \lambda \) and \( \psi \). It is likely that the missing of the effect of these two parameters on \( C \) in Lee and Lee (2001) for the laminar-turbulent flow was responsible for the large errors of predictions, as shown in Figure 5-19 (b).
5.5 Correlation Development

The Mishima and Hibiki (1996) correlation and the homogeneous model with the average viscosity defined by Dukler et al. (1964) predict the data with mean deviation less than 20%. This is not good enough for the design of miniature thermal systems. This section focuses on developing a new correlation for adiabatic two-phase flow frictional pressure drop in microchannels.

In order to identify the effect of mass flux and vapor quality on the parameter $C$ in the Lockhart-Martinelli type correlation, the measured two-phase pressure drop data are used to calculate $C$ and the results are plotted as a function of vapor quality for different mass fluxes, as shown in Figure 5-23 and Figure 5-24. It seems that a constant value of for each channel does not count the effect of the mass flux and vapor quality. All the measured $C$ values are a strong function of the vapor quality and the mass flux. The general trend is the higher the mass flux and the vapor quality, the higher the $C$ values. Thus, the pressure-drop components corresponding to the phase interaction increase with the increasing of vapor quality and mass flux. A similar dependence of the parameter $C$ on the liquid and vapor flow rate was reported by Lee and Lee (2001) for two-phase flow in minichannels.

![Figure 5-23 Effect of $G$ and $x$ on parameter $C$ for test section #1 and #2](image)

The effect of channel size on parameter $C$ is demonstrated in Figure 5-25. As shown in Figure 5-25 (a), for two test sections with similar aspect ratios ($H/W \sim 0.1$) and similar mass fluxes ($G \sim 300$ kg/m$^2$-s), but different hydraulic diameters, the channel with a larger hydraulic diameter (test section #1) has larger parameter $C$ values. The same trend can be seen from Figure 5-25 (b), where the larger channel (test section #3) has larger $C$ values than the smaller channel (test section #5). This trend is consistent with the observation of Mishima and Hibiki (1996), Equation (5-37).
Figure 5-24 Effect of $G$ and $x$ on parameter $C$ for test section #3, #4 and #5
Figure 5-25 Effect of channel size on parameter \( C \) in Lockhart-Martinelli type correlations

(a) Test section #1 (\( D_h = 305 \, \mu m, \, H/W = 0.1 \))
(b) Test section #3 (\( D_h = 141 \, \mu m, \, H/W = 0.09 \))
(c) Test section #5 (\( D_h = 69 \, \mu m, \, H/W = 0.09 \))

Figure 5-26 shows the effect of aspect ratio, \( \alpha = H/W \), on parameter \( C \) in microchannels. Test section #2 and #3 have almost the same hydraulic diameter (\( D_h \approx 150 \, \mu m \)) but different aspect ratios of 0.24 and 0.09, respectively. For three different mass fluxes, \( G \approx 310 \, kg/m^2-s, 460 \, kg/m^2-s \) and \( 610 \, kg/m^2-s \), the channel with a larger aspect ratio (#2) has slightly larger \( C \) values when the vapor quality is low \((x < 0.2)\), and has slightly smaller \( C \) values when the vapor quality is high \((x > 0.2)\). It seems that the effect of aspect ratio on the measured parameter \( C \) is small and can be neglected.

Figure 5-26 Effect of aspect ratio on \( C \) parameter in Lockhart-Martinelli type correlations
The parameter $C$ is a measure of the interaction between the liquid and vapor phases, and the extent of the interaction depends on the flow regimes or patterns. The flow regimes and patterns, as discussed in Chapter 4, are dependent on the mass flux, vapor quality, channel size and other factors. Thus, the effect of mass flux, vapor quality, and channel size on parameter $C$ actually reflects the effect of these factors on flow regimes. When the channel gets smaller, the mass flux is lower or the vapor quality is small, the surface tension force dominates and the flow pattern is typically intermittent flow (slug/plug). The interaction between the two-phases in intermittent flow is normally smaller, compared with that in the annular flow regime, and this is just the reason why the parameter $C$ is smaller for these cases. Therefore, a good correlation shall take the flow regime and the effect of surface tension force into account.

![Figure 5-27 Flow regimes of the experimental data](image)

As has been discussed in Chapter 4, the transition line, $We_{GS} = 11We_{LS}^{0.14}$, in the Akbar et al. (2003) correlation approximately divided the two-phase flow in a $92.0 \times 951.6 \mu m$ into the surface tension dominated flow regime and the inertia dominated flow regime. The same method is used to classify the present experimental data and the result is shown in Figure 5-27. It looks like approximately half of the current data are in the surface tension dominated region.

Suo and Griffith (1964) studied the surface-tension-force dominant flow through the microchannels. They found that the following two dimensionless parameters are important for this type of flow

$$\psi = \frac{\mu U_b}{\sigma}$$

$$\lambda = \frac{\mu_i^2}{\rho_i \sigma D_h}$$

where $U_b$ is velocity of the gas bubbles and it is defined as:
\[ U_g = 1.2(U_{GS} + U_{LS}) \] (5-51)

The similar two parameters have been used by Lee and Lee (2001) in correlating the parameter \( C \) for laminar-laminar flow. In addition, Lee and Lee (2001) suggest using the liquid only Reynolds number, \( \text{Re}_{lo} \), to take account of the effect of the mass velocity on parameter \( C \).

A new correlation was developed for adiabatic two-phase flow pressure drop in microchannels based on the present experimental data. The correlation is expressed as

\[
C = \begin{cases} 
1.279 \times 10^{-9} \lambda_{GS}^{-1.09} \psi \text{Re}_{lo}^{0.40} , & W_{e_{GS}} \leq 11.0 W_{e_{LS}}^{0.14} \\
1.386 \times 10^{-4} \lambda_{LS}^{-0.65} \psi^{0.2} \text{Re}_{lo}^{0.52} , & W_{e_{GS}} > 11.0 W_{e_{LS}}^{0.14} 
\end{cases}
\] (5-52)

Figure 5-28 Two-phase pressure gradient data vs. the present correlation predictions
Figure 5-29 Two-phase pressure gradient data vs. present correlation predictions for each test section

The parameter ranges used to develop the correlation are $1.25 \times 10^{-5} \leq \lambda \leq 5.57 \times 10^{-5}$, $0.0067 \leq \psi \leq 0.6$, and $113 \leq Re_{lo} \leq 597$. 

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It should be noted that in Equation (5-52), the exponents for the two parameters relating to surface tension force, $\lambda$ and $\psi$, have larger absolute values in the surface tension dominated region, $We_{GS} \leq 11.0We_{LS}^{0.14}$, than those in the inertia dominated region, $We_{GS} > 11.0We_{LS}^{0.14}$. This observation indicates a weaker effect of surface tension force in the inertia dominated region, which is an expected result. On the other hand, the exponent for $Re_{lo}$ in the surface tension dominated region is smaller than that in the inertia dominated region, which is also an expected result. As shown in Figure 5-28, the new correlation predicts 91.0% of all the current experimental data within ±15% with a mean deviation of 7.6%.

In Figure 5-29, the pressure gradient calculated from the new correlation is compared with the experimental data for each test section as a function of mass flux and vapor quality. The correlation represents the trend of the data with respect to vapor quality and mass flux very well, and no systematic errors were observed.

5.6 Summary and Conclusions

Experimental study was performed for adiabatic R134a two-phase flow in rectangular microchannels with hydraulic diameters varying from 69.5 to 304.7 $\mu$m and with aspect ratios changing from 0.09 to 0.24. The experimental data were compared with twelve existing correlations. Most of the correlations, such as Lockhart-Martinelli (1949), Chisholm (1973), and Friedel (1979) made significant errors when compared with the experimental data. However, the homogeneous models and the Mishima and Hibiki (1996) correlation predicted the experimental data reasonably well with mean deviations less than 21%. For the homogeneous models, the definition of mean two-phase viscosity suggested by Dukler et al. (1964) gave a better prediction than that by McAdams et al. (1942).

A new correlation has been developed for adiabatic two-phase flow pressure drop in microchannels, in which the parameter $C$ in Lockhart-Martinelli type correlation was correlated as a function of three nondimensional parameters in two flow regimes, the surface tension dominated regime and the inertia dominated regime. The new correlation is expressed as

$$C = \begin{cases} 1.279 \times 10^{-9} \lambda^{-1.96} \psi Re_{lo}^{0.40}, & \text{if } We_{GS} \leq 11.0We_{LS}^{0.14} \\ 1.386 \times 10^{-4} \lambda^{-0.65} \psi^{0.2} Re_{lo}^{0.52}, & \text{if } We_{GS} > 11.0We_{LS}^{0.14} \end{cases}$$

The proposed correlation predicts 91.0% of all the current experimental data within ±15%, with a mean deviation of 7.6%.

References


presented at AIChE meeting held in Chicago December 2-6, 1962, also AICHe Journal, Vol. 10, No. 1, pp. 38-51


Tu, X. and P. Hrnjak, 2003, “Pressure Drop and Visualization of R134a Two-Phase Flow in a Rectangular Microchannel”, ASHRAE Transaction, Vol. 109, No. 1


Chapter 6. Liquid Superheat during Flow Boiling in Microchannels

6.1 Chapter Overview

This chapter presents the experimental results for liquid flow boiling of R134 in a microchannel test section having four parallel channels, each with a cross-section geometry of $75 \times 811.94 \, \mu m$. A non-intrusive technique was used to measure the bulk liquid superheat after the heating section. The measured liquid superheat was taken as a conservative estimate of the wall superheat before the Onset of Nucleate Boiling (ONB), $(\Delta T_{\text{sat}})_{\text{ONB}}$. In the experiment, the mass flux was changed in the range $G = 97 – 310 \, \text{kg/m}^2\text{-s}$, and the inlet subcooling was changed between 0 °C and 7.0 °C. The results were compared with the classic nucleation criterion for flow boiling. The effects of mass flux, subcooling, and cavity size, as well as the channel diameter on the boiling incipience in microchannels are discussed.

6.2 Literature Review

Recently, in various branches of industry, there has been a growing interest in developing miniaturized thermal systems and microscale devices. The increasing technological demands in various new areas require a comprehensive understanding of the fundamental phenomena that govern thermal transport in small scales. The boiling of liquids in microchannels is one of the most important topics, because of the potential applications in electronic cooling, micro thermal systems, and the like.

Lin et al. (1993) could not generate bubble with static water, methanol, and FC43 liquids in microchannels that are $7.5 \, \mu m$ deep, 30 and 60 $\mu m$ wide, until they raised the heater wall temperature to the proximity of the critical temperature. Jiang et al. (1999, 2000) observed that during the boiling process of water in microchannels with hydraulic diameters of 40 and 80 $\mu m$, the wall temperature increased almost linearly with the wall heat flux until the onset of critical heat flux (CHF) condition. No boiling plateau, which is associated with saturated nucleate boiling state, was observed in the boiling curves.

Peng and Wang (1994a, 1994b) experimentally investigated water and methanol flow boiling in microchannels of cross-section from $200 \times 700 \, \mu m$ to $800 \times 700 \, \mu m$. They noted that no vapor bubbles could be observed even in fully developed nucleate boiling regime. This behavior was termed as “bubble extinction”. Two hypothetical concepts, “evaporating space” and “fictitious boiling”, were proposed to explain this new behavior in microchannels.

Peng et al. (1998) proposed semi-empirical correlations to predict the boiling conditions in the microchannels. The correlations demonstrated that decreasing the size of the microchannels resulted in dramatically high wall superheats for nucleation. Peng et al. (2001) demonstrated experimentally that the correlation described the superheats for nucleate boiling in microchannels. Peng et al. (2000) investigated the role of perturbations on the dynamics of clusters, and developed the physical interpretation of “fictitious boiling”. They also proposed a criterion for the occurrence of the “fictitious boiling”.

The aforementioned experimental data and theoretical work indicated that the flow boiling process in microchannels might not be the same as that in normal scales. Nevertheless, Kandlikar (2002) argued that Peng and his co-workers did not use a proper microscope and high-speed video techniques, and what called “fictitious boiling” is unacceptable. In addition, Kandlikar (2002) pointed out that it is expected that the nucleation criterion for
flow boiling established for large-diameter tubes will hold true unless the tube diameter approaches the cavity dimensions, which may be the case only for a submicrometer-sized tubes. Kandlikar (2003a) used high-speed camera to observe flow patterns during flow boiling of water in a single rectangular microchannel of height 197 µm and width 1054 µm, and the nucleate boiling was observed by the author.

The Peng et al. (1998) correlation is based on the assumption that a bubble nucleus completely fills the microtube, which requires the ‘active’ cavity mouth diameters to be about the same size as the channel diameter. This assumption is unlikely a practical case for most of the microchannel applications. For a microchannel with diameter in the order of 100 µm, most likely the cavity sizes for active nucleation are on the order of a few micrometers or smaller.

Zhang et al. (2002) studied the nucleate boiling conditions and mechanisms in plasma etched silicon microchannels below 150 µm hydraulic diameter. The experiments showed that the high wall superheat in microchannels is primarily due to the lack of ‘active’ nucleation sites rather than limited channel space or a high liquid surface tension. By creating small cavities in the channel walls, the wall superheat can be eliminated from silicon channels as small as 28 µm in hydraulic diameter. The result of Zhang et al. (2002) indicated that the unusual boiling behavior in microchannels reported in Lin et al. (1993) and Jiang et al. (1999, 2000) could be attributed to the lack of ‘active’ nucleation sites in their studies.

The above literature review revealed some contradictions regarding flow boiling mechanisms in microchannels. The recent results of Kandlikar (2002, 2003) and Zhang et al. (2002) indicated that the normal nucleate boiling still exists in microchannels. Nevertheless, it is still not clear from the open literature whether the nucleation criterion for flow boiling established for large-diameter tubes can be used in microchannels. It is also not clear what are the effects of mass flux and subcooling on the wall superheat for the Onset of Nucleate Boiling (ONB), \( (\Delta T_{sat})_{ONB} \) in microchannels.

### 6.3 Experimental Results

The experimental facility, instrumentation, experimental procedure, data reduction and uncertainties, as well as the characterization of the microchannel test sections were described in section 2.4, as well as in Tu and Hrnjak (2001).

As described in section 2.4.5.2, the power level was increased from zero and data were taken continuously until the liquid superheat disappeared. Figure 6-1 illustrates the experiment with a mass flux of 164 ± 5 kg/m²s, inlet temperature of 22.7 ± 0.2 °C (inlet subcooling of 5 °C), and a saturation temperature of 28 ± 1 °C. The experimental curves in Figure 6-1 were divided into four stages that are explained in detail in the following sections.
Figure 6-1 R134a liquid flow boiling in microchannels of $75 \times 811.94 \mu m$, $G = 164 \ kg/m^2s$, $T_{ei} = 22.7 \ ^\circ C$, $T_{sat} = 28 \ ^\circ C$

The first 700 seconds of data in Figure 6-1 were in stage I, in which the heater was off and the fluid in and out of the channels was in a liquid phase. Consequently, all the temperatures, including the bulk fluid temperatures,
$T_{ch1} - T_{ch4},$ the fluid inlet temperature and the fluid outlet temperature, were at the same value of 22.6 °C. The total pressure drop was low and corresponded to the total pressure drop value for liquid only flow.

In stage II, when the heating power was adjusted to 0.5W ($q'' \approx 5.5 \text{ kW/m}^2$), the bulk fluid temperatures, $T_{ch1}-T_{ch4}$, increased to values higher than the saturation temperature. The outlet temperature, $T_{eo}$, was still slightly lower than the saturation temperature, which was attributed to the heat losses through the metal T-junction that was used to install the thermocouple.

When the heater power level was increased to 0.7 W ($q'' \approx 8 \text{ kW/m}^2$), the bulk fluid temperatures in all four channels were brought to about 32 °C; while the outlet temperature reached and remained at the saturation temperature of about 28°C. At the same time, vapor slugs (or plugs) were observed at the exit of the test section. Obviously, the fluid remained in a metastable superheated liquid state inside the microchannel because the boiling initiation condition was not met. When the metastable superheated liquid reached the macroscale exit tube, boiling occurred and the superheat disappeared.

The stage II continued for two additional power levels of 0.9 W and 1.3 W that corresponded to heat fluxes of 10.8 kW/m² and 15.5 kW/m², respectively. Figure 6-1(c) shows that the total pressure drop remained almost the same in stage I and stage II, which is another proof of liquid superheat state in stage II.

When the power level was adjusted to 1.7 W ($q'' \approx 20 \text{ kW/m}^2$), the bulk fluid temperatures in all the four channels increased from 42°C to 46°C. Suddenly the temperatures dropped: channel four, to 29°C; channel three, to 33°C; and channel one and two, to 39 °C. These dropping temperatures were accompanied by a sudden increase in the total pressure drop, as shown in Figure 6-1 (c). After that, the measured bulk temperatures in stage III exhibited a very strange behavior. The bulk temperatures increased and decreased alternatively, with enormous differences between various channels. In the mean time, the measured total pressure drop increased monotonically.

The process that occurred in stage III can be explained as follows. For a certain reason, boiling occurred earlier in channel 4 than in the other channels, and the bulk fluid temperature in that channel dropped to approximately the saturation temperature. The decrease of the bulk fluid temperature introduced more heat into this channel and increased the wall heat flux. Higher heat flux also, in turn, helped nucleation in this channel. The boiling introduced a larger pressure drop to channel 4, which reduced the flow rate and caused the total flow to be redistributed among the channels. The channels with no or less boiling must have had a higher mass flow rate in order to balance the pressure drop. Therefore, the bulk fluid temperatures dropped to different values, and the total pressure drop went up. Due to the fact that liquid superheat is a non-equilibrium metastable state, the boiling process may also be unstable and related to other factors such as heating time, history, etc. All these along with flow redistribution in parallel microchannels made the whole process a very complicated one, resulting in the unusual behavior observed in stage III.

Finally, in stage IV, after the heater power was adjusted to 4.3 W ($q'' = 50 \text{ kW/m}^2$) for 2 to 3 minutes, the bulk fluid temperatures in all four channels dropped to and remained at the saturation temperature. This process was also accompanied by a sudden jump in pressure drop. In this stage, boiling occurred in all four channels, and a stable two-phase flow was obtained. Therefore, energy balance could be used to estimate the vapor quality at this point, which was determined to be about 70%.
It should be noted that the liquid superheat phenomenon observed in stage II and II was hysteresis — it only occurs before the initiation of the nucleation. After stage IV was reached, the power level was reduced, following the same steps as shown in Figure 6-1 (b), the bulk liquid superheat was never observed and the temperatures, $T_{ch1} - T_{ch4}$, were the same as the saturation temperature.

Experiments were performed for different mass fluxes ranging from 97 to 310 kg/m²s, and inlet subcooling ranging from 0 °C to 7 °C. A similar liquid superheat (and the hysteresis) phenomenon was recorded, and all four stages, as shown in Figure 6-1, were identified for each experiment. However, the extents of superheat were different for different mass fluxes and inlet subcooling.

The effects of mass flux, inlet subcooling, as well as saturation on the measured maximum superheats of each channel are listed in Table 6-1. The data points in Table 6-1 suggest that the maximum superheats for different channels were different in the same test, as can be seen for test #2, #3 and #8. However, channel #2 always has the highest maximum-superheat among the four channels, which indicates that this channel may lack active nucleate sites. In addition, it seems that the maximum superheat was a strong function of inlet subcooling but less a function of mass flux.

When designing microchannel heat sinks with subcooled liquid flow boiling, great attention must be paid to the phenomenon of bulk liquid superheat. Normally, heat exchangers with phase changes are preferred because they provide a relatively uniform wall temperature and a higher heat transfer coefficient. However, when liquid superheat occurs, the evaporator performs in a fashion similar to single phase flow heat exchangers. Even worse, when bubbles form in the superheated liquid, the growth rate is very rapid; and the explosive formation of vapor is often a source of instability (Collier and Thome, 1996), as indicated in stage III of Figure 6-1. Therefore, it is very important to understand the mechanism behind the bulk-liquid superheat phenomenon in the microchannels.

In macrochannels, a high liquid superheat before ONB typically occurs in the glass apparatus, where the range of active cavity sizes is severely restricted, or in well wetting fluids at low reduced pressures, such as nitrogen, R-12, and R-114 (Collier and Thome 1996, Celata et al. 1992). For subcooled R134a flow boiling in macroscale annular ducts, Yin et al. (2000) reported wall superheat as high as 18°C before ONB, but this is limited to a very low saturation temperature ($T_{sat} = -2 \, ^\circ C$) and a high subcooling ($\Delta T_{sub} = 10 \, ^\circ C$), and the wall superheat decreased rapidly when saturation temperature increased.

### Table 6-1 Summary of flow boiling experimental results

<table>
<thead>
<tr>
<th>Test</th>
<th>$G$ (kg/m²s)</th>
<th>$T_{sat}$ (°C)</th>
<th>Inlet subcooling (°C)</th>
<th>Maximum superheat for each channel (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>1</td>
<td>163</td>
<td>24.8</td>
<td>0.7</td>
<td>13.1</td>
</tr>
<tr>
<td>2</td>
<td>169</td>
<td>26.3</td>
<td>0.6</td>
<td>3.0</td>
</tr>
<tr>
<td>3</td>
<td>97</td>
<td>26.5</td>
<td>0.5</td>
<td>7.2</td>
</tr>
<tr>
<td>4</td>
<td>229</td>
<td>26.5</td>
<td>0.5</td>
<td>5.8</td>
</tr>
<tr>
<td>5</td>
<td>310</td>
<td>22.8</td>
<td>0.3</td>
<td>6.1</td>
</tr>
<tr>
<td>6</td>
<td>164</td>
<td>28.0</td>
<td>5.0</td>
<td>21.7</td>
</tr>
<tr>
<td>7</td>
<td>169</td>
<td>22.7</td>
<td>0</td>
<td>6.2</td>
</tr>
<tr>
<td>8</td>
<td>156</td>
<td>30.0</td>
<td>7.0</td>
<td>20.2</td>
</tr>
</tbody>
</table>
6.4 Analysis and Discussions

In the classical nucleation theory, the superheated liquid state owes its existence to an energy barrier that causes the vapor embryo to collapse, rather than lead to nucleation, if it is less than a critical size (Das et al. 2000). However, a liquid cannot be superheated up to the critical temperature, and there is a limit to the maximum attainable temperature for any given liquid without boiling according to the homogeneous nucleation theory. Numerous investigators have attempted to measure this limit, which is termed as “limit of superheat of the liquid”, $T_{sh}$, in Das et al. (2000).

According the classic homogeneous theory, the rate of nucleation, $dn/dt$, in a metastable liquid at temperature $T_g$ is given by the product of the number of equilibrium nuclei per unit volume and a collision frequency $\lambda$,

$$\frac{dn}{dt} = \lambda N(r^*) = \lambda N e^{-\Delta G(r^*)/kT_g}$$

(6-1)

$$\Delta G(r^*) = \frac{16}{3} \left( \frac{\pi \sigma^3}{(p_g - p_l)^2} \right)$$

(6-2)

where $\Delta G(r^*)$, $k$, $p_l$, and $\sigma$ are the maximum free energy, the Boltzmann constant, the fluid pressure, and the surface tension of the fluid at $T_g$, respectively. The vapor pressure inside a bubble in equilibrium, $P_g$, is expressed as

$$P_g = P_{sat}(T_g) \exp \left( \frac{(p_g - p_l)v_l M}{RT_g} \right)$$

(6-3)

The collision frequency, $\lambda$, has been given by Westwater (1958) as

$$\lambda = \frac{kT_g}{h}$$

(6-4)

where $h$ is Planck’s constant.

Collier and Thome (1996) recommended that the significant nucleation occur for values of $dn/dt$ between $10^9$ and $10^{13}$ m$^{-3}$s$^{-1}$. For R134a at $p_l = 727$ kPa ($T_{sat} = 28.0 \degree C$), these values correspond to temperature ($T_g$) from 66.8 $\degree C$ to 67.2 $\degree C$, respectively. This gives a superheat of $\Delta T_{sat} = 39 \degree C$. Therefore, the highest superheat measured (Channel #2 of test 6 in Table 6-1), $\Delta T_{sat} = 22.5 \degree C$, is lower than the superheat needed for homogeneous nucleation. Hence, although unusually high liquid superheat temperatures were recorded in the current investigation, they were still within the kinetic limit of superheat based on the classic nucleation theory.

It should be noted that the kinetic limit of superheat was extremely difficult to reach in macroscale spaces even under very carefully controlled conditions (Das et al. 2000). This is due to the fact that foreign bodies and container surfaces normally provide ample nuclei to act as centers of vapor formation, and this method of vapor generation is termed ‘heterogeneous nucleation’. In this method of nucleation, the wall superheat in the heating section, $T_{wi} - T_{sat}$, is of more interest for nucleation initiation than the bulk liquid superheat. In the current study, the measured liquid superheat, $T_{bulk} - T_{sat}$, was taken as a conservative estimation of the wall superheat, $T_{wi} - T_{sat}$. The actual values of $T_{wi} - T_{sat}$ should be higher since the wall is heated from outside. Therefore, it is safe to say that Onset of Nucleate Boiling (ONB) did not start until the wall superheat is higher than 17.5$\degree C$ for the experiment shown in...
Figure 6-1. This wall superheat is much higher than what were normally observed for heterogeneous nucleation of R134a in macrochannels.

Assuming sufficiently wide range of ‘active’ cavity sizes are available, the following equations are given by Collier and Thome (1996) to predict the wall heat flux and superheat for ONB in macrochannels.

\[
(T_w - T_{sat})_{ONB} = \frac{B}{r_{crit}^\alpha} + \frac{q_{ONB}^\alpha r_{crit}}{k_l} \tag{6-5}
\]

\[
r_{crit} = \sqrt{\frac{Bk_l}{q''}} \tag{6-6}
\]

\[
B = \frac{2\sigma T_{sat}^\nu}{i_{vy}} \tag{6-7}
\]

In above equations, \(r_{crit}, l_{v}, \) and \(T_{sat}\) are the critical bubble size, the latent heat of vaporization, and the saturation temperature (Kelvin), respectively. After rearrangement, the above equations become

\[
q_{ONB}^\alpha = \frac{k_l}{4B} (\Delta T_{sat})_{ONB}^2 \tag{6-8}
\]

In the subcooled boiling region, the heat transfer equation is

\[
q'' = h_{lo} (\Delta T_{sat} + \Delta T_{sub}) \tag{6-9}
\]

Solving Equation (6-8) and (6-9) together, the conditions for ONB are expressed as

\[
(\Delta T_{sat})_{ONB} = \frac{2Bh_{lo}}{k_l} \left( 1 + \sqrt{1 + \frac{k_l\Delta T_{sub}}{Bh_{lo}}} \right) \tag{6-10}
\]

\[
q_{ONB}^\alpha = h_{lo} (\Delta T_{sat})_{ONB} + \Delta T_{sub} \tag{6-11}
\]

where \(h_{lo}\) and \(\Delta T_{sub}\) are liquid only flow heat transfer coefficient and subcooling, respectively. The single-phase flow in microchannels is in the laminar region, and the heat transfer coefficient is

\[
Nu = \frac{h_{lo} D_h}{k_l} = \text{const} \tag{6-12}
\]

The comparison of the measured superheat with the predictions of Equation (6-10) is shown in Figure 6-2. The Nusselt number was calculated based on the fully developed laminar flow for one or more walls heated in rectangular ducts suggest by Rohsenow et al. (1985), as shown in Equation (7-2). It is clear from Figure 6-2 that this method significantly underpredicts the experimental data. The calculated wall superheats for ONB, \(\Delta T_{ONB}\), based on Equation (6-10) for the current experimental conditions are in the range of 0.11 – 0.83 °C, which were much smaller than the values listed in Table 6-1.
Due to the fact that a heated surface may not have a complete range of ‘active’ cavity sizes, Equation (6-10) and (6-11) represents the lower bound of the wall heat flux and superheat required to initiate nucleation. Davis and Anderson (1966) pointed out that at low fluxes and low pressures the values of $r_{crt}$ predicted from Equation (6-6) may be so large that no ‘active’ sites of this size are present on the heating surface. In this case, the estimated largest ‘active’ cavity size shall be used and substituted into Equation (6-5).

For the calculated data points in Figure 6-2, the corresponding critical bubble sizes, $r_{crt}$, predicted from Equation (6-6) are in the range of 1.6 to 15 $\mu$m. As has been described in section 2.4.2.3, the surface of the channel...
The wall has a relative roughness, $R_a$, of about 0.1 $\mu$m. The roughness measurement also reported the maximum peak value, $R_p = 0.5$ $\mu$m, as well as the minimum valley value, $R_v = 0.5$ $\mu$m. It is very unlikely that ‘active’ nucleate sites of size above 1.6 microns are available for the present heated surface, according to the surface characteristic and the manufacturing techniques used. Hence, the actual largest ‘active’ cavity size shall be used in calculating $\Delta T_{ONB}$, according to Davis and Anderson (1966).

It is very difficult to measure the largest ‘active’ cavity size directly. As shown in Figure 6-3 (a), Equation (6-5) with the largest ‘active’ cavity size of 0.1 $\mu$m radius seems to predict the current data reasonably well. Nevertheless, it underpredicts the data points with higher subcoolings by a factor of 2 to 3. In Figure 6-3 (b), the effect of the size of the largest ‘active’ cavity size on the calculated superheat is illustrated, using a saturation temperature $T_{sat} = 28$ °C and subcooling $\Delta T_{sub} = 5$ °C. It shows that the superheat for ONB increases when the largest ‘active’ cavity size decreases.

![Figure 6-4](image_url)

**Figure 6-4** Effect of mass flux on measured superheat

The liquid flow in the microchannels is in the laminar region and the heat transfer coefficient of liquid flow is not a function of mass fluxes. Therefore, the mass flux is not a factor in determining the predicted liquid superheat of ONB shown in Figure 6-2 and Figure 6-3. Figure 6-4 demonstrates the effect of mass flux on the measured superheat for saturation temperature of 26.4 °C and inlet subcooling of 0.5 °C for three different mass fluxes, $G = 97$ kg/m²s, 169 kg/m²s, and 229 kg/m²s. It is clear that the measured superheat is not a strong function of the mass flux, which is an expected result.

The above analysis shows that there is no indication of deviation from the classic nucleation theory in the microchannel geometries of the present study. The experimental work of Zhang et al. (2002) also showed the existence of the heterogeneous nucleation in microchannels as small as 28 $\mu$m in hydraulic diameter, by creating small cavities in the channel walls. The high wall superheat in microchannels is primarily due to the lack of large range of various sized ‘active’ nucleation sites rather than a limited channel space.
Figure 6-5 Wall superheat for initiation of nucleation as a function of channel hydraulic diameter assuming cavities of all sizes are present

The remaining section will discuss the effect of microchannel diameter on \( (\Delta T_{sat})_{ONB} \) using the heterogeneous nucleation theories. Take a special case of saturated boiling, i.e. \( \Delta T_{sub} = 0 \), as an example. Substituting Equation (6-7) and (6-12) into Equation (6-10), the condition of ONB for a microchannel can be expressed as

\[
(\Delta T_{sat})_{ONB} = \frac{8\sigma T_{sat} v_{lv} Nu}{i_{lv}} \frac{1}{D_h} \tag{6-13}
\]

For boiling of saturated R134a at temperature of 23 °C, the effect of diameter on wall superheat for initiation of nucleation in microchannels is plotted in Figure 6-5. The plot shows the tendency that \( (\Delta T_{sub})_{ONB} \) increases when the channel diameter is reduced, assuming that cavities of all sizes are present on the heating surface. A similar plot and the same equation have been described in Kandlikar (2003b). The equation may suggest that a higher superheat is required for smaller channels, but this is based on the assumption that cavities of all sizes are available. However, this assumption generally does not hold for microchannels, as the refined techniques that are applied to manufacture these channels tend to produce smoother surfaces. Therefore, the lack of cavities of all sizes, rather than the channel size, is expected to be a dominant factor in determining the conditions of nucleation initiation in microchannels.

No contact angle information is available for R134a and the microchannel material (Kapton). It should be recognized that the contact angle play an important role for boiling incipience in macrochannels.
6.5 Summary and Conclusions

A metastable liquid superheat phenomenon of R134a flow boiling in $75 \times 811.94 \, \mu m$ microchannels was investigated experimentally. The specially designed test section provided a way to measure the bulk fluid temperature in microchannels non-intrusively. A liquid superheat as high as $22.5 \, ^\circ C$ was measured in a microchannel for a saturation temperature of $28 \, ^\circ C$, an inlet subcooling of $5 \, ^\circ C$, and a mass flux of $164 \, kg/m^2s$.

The measured bulk liquid superheat represented a conservative estimation of the more commonly used wall superheat. The experimental results showed that the maximum liquid superheat was not a function of the mass flux, but is a strong function of the inlet subcooling.

The maximum liquid superheat for boiling incipience was calculated making use of the classic theory of nucleation and the laminar flow heat transfer relationship. The calculated values were compared with the experiments and reasonable agreements were achieved assuming the largest ‘active’ cavity size of $0.1 \, \mu m$ radius.

No indication of deviation from the classic nucleation theory was observed in the present study for R134a flow boiling in $75 \times 811.94 \, \mu m$ microchannels. The unusually high liquid superheat measured was attributed to the lack of large range of various sized ‘active’ nucleation sites in the microchannels, rather than a limited channel space.

References


Chapter 7. Evaporation Heat Transfer

7.1 Chapter Overview
This chapter presents the experimental results for evaporation heat transfer of R134a flowing in rectangular microchannels of cross-section $75 \times 811.94 \, \mu m$. The parameter ranges examined were mass flux from 87 to 165 kg/m$^2$/s, heat flux from 10 to 20 kW/m$^2$; vapor quality from 0.1 to 0.9; and saturation temperature at 23 °C. It was found that the heat transfer coefficients were a strong function of heat flux at low vapor qualities, and of mass flux and vapor quality at higher vapor qualities. The present experimental data were compared with ten existing correlations. Based on the present data, a semi-empirical correlation was proposed for the evaporation heat transfer coefficients in microchannels.

7.2 Literature Review
Recently, the growing interest in developing microscale thermal systems and compact heat exchangers requires the characterization of evaporation heat transfer in microchannels with hydraulic diameters close to 100 μm. Evaporation heat transfer studies in small channels are summarized in Table 7-1, which indicates that little data are available in this new area.

Lazarek and Black (1982) reported the experimental results of R113 flow boiling in a 3.1 mm diameter tube. They concluded that the nucleate boiling was the dominant heat transfer mechanism, that is, heat transfer coefficients were a strong function of the heat flux and the system pressure, while the effects of mass flux and vapor quality were very small.

Table 7-1 Summary of evaporation heat transfer studies in mini/micro channels

<table>
<thead>
<tr>
<th>Reference</th>
<th>Geometry</th>
<th>$D_h$ (μm)</th>
<th>$G$ (kg/m$^2$/s)</th>
<th>$q^*$ (kW/m$^2$)</th>
<th>$x$</th>
<th>Testing Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lazarek and Black (1982)</td>
<td>Single circular</td>
<td>3100</td>
<td>125 ~ 750</td>
<td>14 ~ 380</td>
<td>0 ~ 0.7</td>
<td>R113</td>
</tr>
<tr>
<td>Wambsganss et al. (1993)</td>
<td>Single circular</td>
<td>2920</td>
<td>50 ~ 300</td>
<td>8 ~ 91</td>
<td>0 ~ 0.9</td>
<td>R113</td>
</tr>
<tr>
<td>Tran et al. (1993)</td>
<td>Single circular</td>
<td>2460</td>
<td>44 ~ 832</td>
<td>3.6 ~ 129</td>
<td>0 ~ 0.94</td>
<td>R12</td>
</tr>
<tr>
<td>Yan and Lin (1998)</td>
<td>Single rectangular (α = 0.42)</td>
<td>50 ~ 200</td>
<td>5 ~ 20</td>
<td>0.1 ~ 0.9</td>
<td>R134a</td>
<td></td>
</tr>
<tr>
<td>Bao et al. (2000)</td>
<td>Single circular</td>
<td>2000</td>
<td>50 ~ 1800</td>
<td>5 ~ 200</td>
<td>0 ~ 0.9</td>
<td>R11 HCFC123</td>
</tr>
<tr>
<td>Lin et al. (2001)</td>
<td>Single circular</td>
<td>1950</td>
<td>50 ~ 1800</td>
<td>5 ~ 200</td>
<td>0 ~ 0.9</td>
<td>R11 HCFC123</td>
</tr>
<tr>
<td>Lee and Lee (2001a)</td>
<td>Single rectangular (α = 0.02 ~ 0.1)</td>
<td>784 ~ 3640</td>
<td>50 ~ 200</td>
<td>3 ~ 16</td>
<td>0.15 ~ 0.75</td>
<td>R113</td>
</tr>
<tr>
<td>Oh et al. (1998)</td>
<td>Single circular</td>
<td>750 ~ 2000</td>
<td>240 ~ 720</td>
<td>10 ~ 20</td>
<td>0.1 ~ 1.0</td>
<td>R134a</td>
</tr>
<tr>
<td>Ravigururajan (1998)</td>
<td>54 rectangular (α = 0.27)</td>
<td>425</td>
<td>52 ~ 449</td>
<td>7 ~ 700</td>
<td>0 ~ 0.9</td>
<td>R124</td>
</tr>
<tr>
<td>Moriyama et al. (1992)</td>
<td>Rectangular (α = 0.17 ×10$^{-3}$ ~ 3.7×10$^{-3}$)</td>
<td>70 ~ 220</td>
<td>200 ~ 1000</td>
<td>5 ~ 40</td>
<td>0 ~ 0.8</td>
<td>R113</td>
</tr>
</tbody>
</table>

Wambsganss et al. (1993) performed experimental study on boiling heat transfer of R-113 in a 2.92 mm tube and compared the experimental results with nine existing correlations. The Lazarek and Black (1982) model
correlated the data with the lowest mean deviation. The authors identified the nucleate boiling as the dominant heat transfer mechanism. The dominance of nucleation in minichannels was attributed to two reasons: intermittent flow pattern (slug and plug flow) and high boiling number.

Tran et al. (1993) measured the boiling heat transfer coefficient for R-12 flowing in a rectangular channel with cross-section geometries of 4.06 × 1.7 mm and in a circular tube of 2.46 mm in diameter. The authors pointed out that the results corresponded to the nucleate boiling dominant region and proposed a new correlation.

Bao et al. (2000) tested R11 and HCFC123 flow boiling in a smaller diameter tubes with inner diameter of 1.95 mm. They concluded that the nucleate boiling was the dominant mechanism over a wide range of flow conditions and the convective effects were less important because of the relatively low Reynolds numbers and liquid conductivity.

The experimental results of Lazarek and Black (1982), Wambsganss et al. (1993), Tran et al. (1993) and Bao et al. (2000) show that the evaporative convection heat transfer coefficient in minichannels exhibit heat flux dependence and a relative independence from quality and mass flux. This has been ascribed to the nucleate-boiling mechanism that is well known from macrochannel studies. However, Jacobi and Thome (2002) put forward an alternative explanation of this behavior using a thin-film evaporation mechanism.

Yan and Lin (1998) carried out experiments to investigate evaporation heat transfer and pressure drop for R134a flowing in a horizontal tube with diameter of 2.0 mm. They compared the experimental results with the data for larger pipes (\(D_h > 8.0\) mm), and found that the evaporation heat transfer coefficients in their study were about 30 – 80% higher for most situations. They noted that the evaporation heat transfer coefficient was higher at a higher imposed wall heat flux except in the high vapor quality region and at a higher mass flux when the imposed heat flux was low. Their experimental results showed that the heat transfer coefficient was a strong function of the vapor quality for most situations. The authors proposed a correlation similar to that of Kandlikar (1990).

Lin et al. (2001) characterized evaporation heat transfer of R141b flowing in a small tube of 1.0 mm internal diameter. The heat transfer coefficient was demonstrated to be a strong function of the vapor quality and heat flux. They also reported that the pressure drop through small tubes during boiling was significant and the local fluid saturation temperature dropped in the flow direction.

Lee and Lee (2001a) presented the evaporation heat transfer test results for Refrigerant 113 flowing in rectangular channels with low aspect ratios. The gaps between the upper and low plates of each channel were varied from 0.4 to 2 mm while the channel width being fixed to 20 mm. In the quality range of 0.15 – 0.75, the heat transfer coefficient increased with the mass flux and the local quality; however, the effect of heat flux appeared to be minor. For a smaller gap-size and a low mass flow rate, the heat transfer coefficient was primarily controlled by the film thickness. The Kandlikar (1990) correlation covered the high mass flux range (\(Re_c > 200\)) with mean deviation of 10.7%. In the range of \(Re_c \leq 200\), the experimental data were well correlated by a modified form of the enhancement factor \(F\) for the heat transfer coefficient.

Oh et al. (1998) investigated the boiling heat transfer of R134a flowing through horizontal tubes with 0.75 – 2 mm in diameters. They demonstrated that under the experimental conditions investigated, the heat transfer mechanism belonged to the two-phase forced convection, and no nucleate boiling region was observed. In the low quality
region, the measured coefficient increased rather slowly with quality, whereas in the higher quality region there was a relatively rapid increase in the coefficient with quality. At very high quality, the heat transfer coefficient decreased with increasing quality, which is associated with the local dry-out.

As has been pointed out by Kandlikar (2002), there are very few quantitative studies available for flow boiling in microchannels \( (D_h \sim 0.1 \text{ mm}) \).

Ravigururajan (1998) studied flow boiling of R-124 in 54 parallel microchannels with each channel 270 \( \mu \text{m} \times 1000 \mu \text{m} \) and 2.05 cm long. The heat transfer coefficient was a strong function of mass flux, heat flux, wall superheat and vapor quality. It decreased with vapor quality from a value of around 11,000 W/m\(^2\)-K at \( x = 0.01 \) to a value of 8,000 W/m\(^2\)-K at \( x = 0.65 \). The author suspected that one possible reason for this phenomenon was the choking of the channel by the bubbles released from the surface. However, Kandlikar (2002) pointed out that this behavior might be the results of the two trends: (1) the nucleate boiling heat transfer was dominant, leading to its suppression at higher qualities; or (2) the higher vapor fraction leaded to flow oscillations in multichannels with a consequent change in the heat transfer coefficient.

Moriyama \textit{et al.} (1992) investigated the evaporation heat transfer of R113 flowing in extremely narrow channels with thickness of 35-110 \( \mu \text{m} \) and width of 30 mm. The heat transfer coefficient was found to be 3-20 times as large as that of the liquid single-phase flow. The heat transfer coefficient increased quickly from the value of liquid single-phase flow in the region of \( x = 0.0 - 0.1 \), and remained at nearly constant values for \( x = 0.1 - 0.7 \). Dry-out occurs at about \( x = 0.7 - 0.8 \), where the heat transfer coefficient decreased abruptly to the value of vapor flow. The authors explained the experimental results by a slug-flow model for the low-quality region and a film-flow model for the high quality region.

In summary, experimental data with evaporation heat transfer in microchannels are very scarce, especially for local heat transfer data as a function of quality, heat flux and mass flux. It is still unknown from the open literature which correlation(s) can be extended to flow boiling in microchannels. In this paper, attempts were made to investigate experimentally the effect of mass flux, quality, and heat flux on evaporation heat transfer coefficient of R134a flowing through 75.0 \( \times \) 811.94 \( \mu \text{m} \) microchannels, and the experimental results were compared with some existing correlations. Based on the current experimental data, a new correlation was proposed.

7.3 Experimental Results

7.3.1 Single-phase Friction Factor

The measured friction factor, \( f \), as well as the conventional correlation predictions, is plotted as a function of the Reynolds number in Figure 7-1. The laminar flow theory indicates that \( C_f = fRe \) for a rectangular channel is a constant, with the value dependent on the aspect ratio. The simplified equation by Hartnett and Kostic (1989), Equation (3-1) that fits the exact analytical solution within an accuracy of 0.05% was used.

For the current study, the aspect ratio, \( \alpha \), is 0.092 and the equation for friction factor is expressed as

\[
    f = \frac{85.4}{Re} \tag{7-1}
\]
As seen in Figure 7-1, the laminar flow theory predicts all the data with Reynolds number less than 2,000 with mean deviation of 14%. Equation (7-1) tends to underpredict the experimental data. One possible explanation for this trend is the fact that the shape of the channel was not exactly rectangular during the experiment, as has been characterized in section 2.4.2. The transition from laminar flow to turbulent flow occurred at Reynolds number of around 2,000. The friction factor for turbulent flow was higher than the Churchill equation for smooth tubes. All these results are consistent with the single-phase friction factor measured in single-channel test sections that are presented in Chapter 3.

7.3.2 Two-phase Pressure Drop

7.3.2.1 Adiabatic two-phase pressure drop

Two-phase pressure drop experiments were carried out with two mass fluxes, \( G = 124 \text{ kg/m}^2\cdot\text{s} \) and \( G = 164 \text{ kg/m}^2\cdot\text{s} \). For each mass flux, the vapor qualities varied between 0.05 and 0.95. The saturation pressures were kept at about 650 kPa.

As seen in Figure 7-2, the frictional pressure drop was higher when the vapor quality and mass flux increased. The calculated frictional pressure gradient was compared with the two-phase flow pressure drop correlation developed in Chapter 5, Equation (5-52). As shown in Figure 7-3, the correlation predicts most of the data points within ±20%. In addition, there is a tendency for the correlation to overpredict the experimental data points, which can be attributed to the two-phase flow maldistribution in multiple channels.
7.3.2.2 Pressure drop with flow boiling

During the evaporation heat transfer test, the total pressure drop across the whole microchannel test section was measured and the results were checked with the calculation of Equation (2-23). As shown in Figure 7-4, the model represents the present experimental results within deviation of ± 30% except for a few low pressure drop data points, which indicates that the method describe in Equation (2-26) can be used to calculate the local refrigerant
pressure (temperature). However, the model systematically underpredicts the experimental data. One possible explanation is that the heating effect in microchannels is not negligible since the model is based on adiabatic experiments.

![Figure 7-4 Measured total pressure drops in heat transfer experiment vs. predictions](image)

### Figure 7-4 Measured total pressure drops in heat transfer experiment vs. predictions

#### 7.3.3 Evaporation heat transfer

The evaporation heat transfer experiments were performed with a saturation temperature of 23°C, heat flux of 10, 15 and 20 kW/m², as well as mass flux of 87, 123, and 165 kg/m²-s. The effect of mass flux, vapor quality and heat flux was presented in Figure 7-5.

It should be noted that although the measured data were taken as “local heat transfer coefficients”, the change of quality (see Table 7-2) may be large, especially for the low flow rate and high heat flux data.

<table>
<thead>
<tr>
<th>G</th>
<th>q'' = 10 kW/m²</th>
<th>q'' = 15 kW/m²</th>
<th>q'' = 20 kW/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>87 kg/m²-s</td>
<td>0.22</td>
<td>0.33</td>
<td>0.44</td>
</tr>
<tr>
<td>123 kg/m²-s</td>
<td>0.16</td>
<td>0.23</td>
<td>0.30</td>
</tr>
<tr>
<td>165 kg/m²-s</td>
<td>0.12</td>
<td>0.18</td>
<td>0.23</td>
</tr>
</tbody>
</table>

Table 7-2 Change of vapor quality, Δv, across the main heating section

104
For all the nine curves, it seems that when $x$ is less than about 0.3, $h$ is not a strong function of quality; but when $x$ is greater than 0.3, $h$ increases very rapidly with increasing of quality and reaches a maximum value. Thereafter the heat transfer coefficient stays more or less constant for a certain quality range, and starts to drop. For quality greater than about 0.6, the increase in quality results in a sharp decrease in heat transfer coefficient. The current data points with $G = 165$ kg/m$^2$-s and $q'' = 10$ kW/m$^2$ are compared with the experimental results of Oh et al. (1998) of R134a flowing in a 1 mm tube with the same heat flux but a different mass flux ($G = 480$ kg/m$^2$-s), as shown in Figure 7-6. The current data shows very similar trend of the effect of vapor quality on the evaporation heat transfer as the results in a minichannel with a higher mass flux.
Figure 7-6 Comparison of the current data with that of Oh et al. (1998) in a 1 mm tube

All three plots demonstrated that $h$ is a strong function of mass flux for vapor quality greater than about 0.4. For $x < 0.3$, the effect of mass flux on heat transfer is rather small. Figure 7-5 also shows that the point of maximum heat transfer coefficient shifts to lower qualities when the mass flux is reduced, but this phenomenon may be related to the fact that at lower mass fluxes, $h$ is an averaged value over wider range of vapor quality.

Although not shown directly, the effect of heat flux can be observed in Figure 7-5 by comparing the curves under the same heat flux but different mass fluxes. For $x < 0.3$, the heat transfer coefficient is higher when the wall heat flux increases. For data points with $x > 0.3$, higher wall heat flux still comes with a little higher heat transfer coefficient, but the trend is not so obvious as in the low quality region.

The all-liquid Reynolds number, $Re_{lo}$, in the current study varied from 61 to 115. Therefore, both the heat transfer coefficient for total flow assumed liquid, $h_{lo}$ and for liquid component flow, $h_l$, can be evaluated with the laminar flow heat transfer equation. The fully developed laminar flow Nu number for one or more walls heated in rectangular ducts suggested by Rohsenow et al. (1985) was used. For the current study, it is expressed as

$$h_{lo} = h_l = 4.48 \frac{k_l}{D_h}$$  \hspace{1cm} (7-2)

The ratio of the measured two-phase flow heat transfer and the calculated all-liquid heat transfer is presented as a function of mean quality $x_{avg}$ in Figure 7-7. The measured two-phase heat transfer coefficients were about 1 ~ 8 times as large as that of the liquid flow.

Yan and Lin (1998) compared their data in 2 mm tubes with that in larger pipes ($D_h \geq 8$ mm), and the results show that heat transfer coefficient for minichannel is about 30-80% higher for most situations. Yan and Lin (1998) used the same refrigerant (R134a) as in the current work; moreover, the current experimental conditions (mass flux, heat flux, saturation temperature, and vapor quality) are within those measured by Yan and Lin (1998).
The only differences are the channel size and geometry. Since it is difficult to find same test conditions for comparison, Yan and Lin’s (1998) model was used over the current test conditions with $D_h = 2$ mm, and calculated results were compared with the present experimental data. As shown in Figure 7-8, the current experimental results are about 4 ~ 5 times larger, on average, than the evaporation heat transfer coefficients in a tube of 2 mm in diameter.

![Figure 7-7 Measured heat transfer coefficient vs. calculated liquid flow heat transfer coefficient](image1)

![Figure 7-8 Comparison of the present data with Yan and Lin’s (1988) prediction for a 2 mm tube](image2)

### 7.4 Comparison with Existing Correlations

Ten existing correlations were compared with the experimental data. Four correlations based on minichannel experimental results, Lazarek and Black (1982), Yan and Lin (1998), Tran (1998), as well as Lee and Lee (2001a), is used for comparison. The others are large tube correlations, but based on a large amount of data and widely used. In addition, these large tube correlations have been shown to be able to reasonably correlate the
minichannel experimental results (Wambgsanss et al. 1993, Bao et al. 2000, Pettersen et al. 2000). These correlations are Chen (1966), Kandlikar (1990), Liu and Winterton (1988), Chaddock and Brunemann (1967). In addition, the modified version of Kandlikar’s correlation (Kandlikar and Steinke 2003, Kandlikar and Balasubramanian 2003) for mini/micro channels was also used for comparison. When using these correlations, the single-phase heat transfer was calculated using Equation (7-2).

7.4.1 Chen (1966)

The Chen (1966) correlation was the first to use the superposition principle of nucleate and convection heat transfer, in which the heat transfer coefficient is given as the sum of the two heat transfer contributions.

\[ h = h_{con} + h_{np} = Fh_f + Sh_{pb} \]  (7-3)

where

\[ h_{pb} = 0.00122 \frac{k_f^{0.79} c_p f^{0.45} \rho_f^{0.49}}{\sigma^{0.5} \mu_f^{0.29} \rho_v^{0.24} \Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75}} \]  (7-4)

\[ F = 2.35 \left( \frac{1}{X} + 2.13 \right)^{0.736} \]  for \( X < 10 \)  (7-5)

\[ S = (1 + 0.12 Re_{tp}^{1.14})^{-1} \]  for \( Re_{tp} < 32.5 \)  (7-6)

\[ Re_{tp} = \frac{G(1-x)D_h}{\mu_l} F^{1.25} 10^{-4} \]  (7-7)

where the Martinelli parameter, \( X \), is defined in Equation (5-3).

Figure 7-9 compares the experimental data with the predictions of the Chen (1966) correlation. The predicted heat transfer coefficient increases monotonically with the vapor quality and overpredicts most of the present data. In addition, it identified two-phase forced convection as the dominant heat transfer mechanism.

![Figure 7-9 Comparison of measured h_{tp} with predictions of Chen (1966)](image-url)
7.4.2 Chaddock and Brunemann (1967)

The Chaddock and Brunemann (1967) correlation can be expressed as

\[
h_p = 1.91 \left( Bo \times 10^4 + 1.5 \left( \frac{1}{X} \right)^{0.67} \right)^{0.66} h_t
\]  

(7-8)

where the Martinelli parameter, \( X \), is defined in Equation (5-3) and the boiling number is expressed as

\[
Bo = \frac{q''}{Gi_{h_v}}
\]

(7-9)

As shown in Figure 7-10, the Chaddock and Brunemann (1967) correlation overpredicts most of the experimental data. The nucleation contribution predicted by the model is much larger than the predicted convection part. Consequently, the overprediction of the model can be attributed to overpredicted nucleate boiling contribution.

7.4.3 Liu and Winterton (1988)

The Liu and Winterton (1988) correlation combines the two contributions of the heat transfer asymptotically

\[
h_p = (F h_t + S h_{pb})^{1/2}
\]

(7-10)

\[
F = \left[ 1 + x \cdot Pr_t \left( \frac{\rho_t}{\rho_v} - 1 \right) \right]^{0.35}
\]

(7-11)

\[
h_{pb} = 55 \left( \frac{P}{P_{\text{crit}}} \right)^{0.12} \left[ -\log_{10} \left( \frac{P}{P_{\text{crit}}} \right) \right]^{-0.55} \text{Mo}^{-0.5} q^{0.67}
\]

(7-12)

\[
S = \left[ 1 + 0.055F^{0.1}(Re_t)^{0.16} \right]^{-1}
\]

(7-13)
where \( M_0 \) is the molecular weight. The Prandtl number is expressed as

\[
Pr_l = \frac{C_{pl} \mu_l}{k_l}
\] (7-14)

![Figure 7-11 Comparison of measured \( h_{tp} \) with predictions of Liu and Winterton (1988)](image)

The above equation predicts the present experimental data better than the previous correlations, with mean deviation of 77% for the whole range and 35% for data with \( x < 0.5 \). This method identifies convection part as the major mechanism of heat transfer. Although the correlation is based on experimental data in channels with hydraulic diameter larger than 2.95 mm, it has some data in laminar flow region, i.e., the lowest \( Re_l \) is 568.9. This explains why this correlation works better than other conventional correlations.

### 7.4.4 Kandlikar (1990), Kandlikar and Steinke (2003), Kandlikar and Balasubramanian (2003)

In the Kandlikar (1990) correlation, the two-phase flow boiling heat transfer coefficient was expressed as the sum of the convective and nucleate boiling terms, given by

\[
h_{tp} = \left( C_1 C_{o} \left( 25 Fr_{lo} \right)^{C_2} + C_3 Bo^{C_4} Fr_{pl} \right) h_l
\] (7-15)

where \( h_l \) is the liquid-alone flow heat transfer coefficient and \( C_1 \sim C_4 \) are constants given in Table 7-3. The convection number \( C_o \) is expressed as

\[
C_o \equiv \left( \frac{1-x}{x} \right)^{0.8} \left( \frac{\rho_v}{\rho_l} \right)^{0.5}
\] (7-16)

The Froude number with all flow as liquid, \( Fr_{lo} \) is defined as

\[
Fr_{lo} \equiv \frac{G^2}{\rho_l^2 g D_h}
\] (7-17)
The fluid-dependent parameter $F_{fl} = 1.63$ for R134a, as has been suggested by Kandlikar and Steinke (2003). The value of constant $C_3$ is zero for $F_{fl} > 0.04$. The two sets of values given in Table 7-3 correspond to the convective boiling and nucleate boiling dominated regions, respectively. The heat transfer coefficient at any given condition is evaluated using the two sets of constants for the two regions, and the higher of the two heat transfer coefficient values represents the predicted value.

Table 7-3 Constants in Kandlikar (1990) correlation

<table>
<thead>
<tr>
<th></th>
<th>Convective dominated region</th>
<th>Nucleate boiling dominated region</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_1$</td>
<td>1.1360</td>
<td>0.6683</td>
</tr>
<tr>
<td>$C_2$</td>
<td>-0.9</td>
<td>-0.2</td>
</tr>
<tr>
<td>$C_3$</td>
<td>667.2</td>
<td>1058</td>
</tr>
<tr>
<td>$C_4$</td>
<td>0.7</td>
<td>0.7</td>
</tr>
</tbody>
</table>

The predictions of the Kandlikar (1990) correlation are compared with the present experimental data, as shown in Figure 7-12. The correlation indicates that for $x < 0.3$, the contributions of nucleation and convection is equally important. This method also predicts the trend of $h$ as a function of $x$ in the region of $x < 0.5$ well, but overpredicts the data about 1 – 3 times larger.

![Figure 7-12 Comparison of measured $h_p$ with predictions of Kandlikar (1990)](image)

Kandlikar and Steinke (2003) extended the range of the correlation by Kandlikar (1990, 1991) to flow in minichannels and microchannels. The correlations is expressed as

\[
h_p = \text{Larger of } \begin{cases} 
    h_{p,NBD} \\
    h_{p,CBD}
\end{cases}
\]

\[
h_{p,NBD} = \left(0.6683Co^{-0.2} \left(1 - x\right)^{0.8} + 1058.0Bo^{0.7} \left(1 - x\right)^{0.8} F_{fl}\right)h_i
\]

\[
h_{p,CBD} = \left(1.136Co^{-0.9} \left(1 - x\right)^{0.8} + 667.2Bo^{0.7} \left(1 - x\right)^{0.8} F_{fl}\right)h_i
\]
As shown in Figure 7-13, the correlation of Kandlikar and Steinke (2003) predicts the present data better than the original Kandlikar (1990) correlation, but it still overpredicts the data especially in the lower quality region.

Figure 7-13 Comparison of measured \( h_{tp} \) with predictions of Kandlikar and Steinke (2003)

The liquid only flow Reynolds number for the current study is within the range of \( 61 \leq \text{Re}_{lo} \leq 115 \), where

\[
\text{Re}_{lo} = \frac{GD_{l} \rho_{l}}{\mu_{l}}
\]  

(7-21)

Kandlikar and Steinke (2003) expected that additional changes might have to be incorporated in the correlation of equations (7-18) through (7-20) for very low Reynolds number flows in microchannels.

Kandlikar and Balasubramanian (2003) suggested using the NBD region of the Kandlikar and Steinke (2003) correlation, Equation (7-19), for all-liquid flow Reynolds number close to or less than 100. The experimental results are compared with the predictions of Equation (7-19) for three different all-liquid flow Reynolds number, \( \text{Re}_{lo} = 61, 84 \) and 115 in Figure 7-14 through Figure 7-16. For \( \text{Re}_{lo} = 115 \), the correlation predicts the data with \( x > 0.4 \) very well but considerably overpredict the data with \( x < 0.4 \). For \( \text{Re}_{lo} = 84 \), the correlation slightly overpredicts the data with \( x > 0.4 \) but significantly overpredicts the data with \( x < 0.4 \). For \( \text{Re}_{lo} = 84 \), the correlation overpredicts the data in the entire quality region but represents the trend of data with \( x > 0.4 \).
Figure 7-14 Measured $h_{tp}$ vs. predictions of Kandlikar and Steinke (2003) $h_{tp,NBD}$ of $Re_0 = 61$

Figure 7-15 Measured $h_{tp}$ vs. predictions of Kandlikar and Steinke (2003) $h_{tp,NBD}$ of $Re_0 = 84$
Figure 7-16 Measured $h_{tp}$ vs. predictions of Kandlikar and Steinke (2003) $h_{tp, NBD}$ of $Re_{fo} = 115$

7.4.5 Lazarek and Black (1982)

Lazarek and Black (1982) proposed a new heat transfer correlation for flow boiling in minichannels.

$$Nu = 30 \ Re_{fo}^{0.857} \ Bo^{0.714}$$  \hspace{1cm} (7-22)

where the dimensionless parameters Nusselt number, $Nu$, is defined as

$$Nu = \frac{h_{tp} D_h}{k_l}$$  \hspace{1cm} (7-23)

where $h_{tp}, D_h, k_l$ denote the evaporation heat transfer coefficient, hydraulic diameter of the channel, and liquid conductivity, respectively.

Figure 7-17 Comparison of measured $h_{tp}$ with predictions of Lazarek and Black (1982)
As seen in Figure 7-17, the Lazarek and Black (1982) correlation significantly underpredicts most of the data except for a few data points in the high vapor quality region and low vapor quality region. This correlation identified nucleate boiling as the dominant mechanism in evaporation heat transfer. For data points with \( x < 0.3 \), the correlation underpredict the experimental data by about 40~50%.

\[ Nu = 770 \left( Re_{lo} N_{conf} Bo \right)^{0.62} \left( \frac{\rho_{v}}{\rho_{l}} \right)^{0.297} \]  
\[ (7-24) \]

where the Confinement number, \( N_{conf} \), is defined in Equation (5-20).

As shown in Figure 7-17, this correlation significantly underpredicts the present experimental data, and does not represent the trend of the data.

7.4.7 Yan and Lin (1998)

Based on the evaporation heat transfer data for R134a flowing in a horizontal tube with diameter of 2.0 mm, Yan and Lin (1998) proposed a correlation similar to the Kandlikar (1990, 1991) correlation for flow boiling in minichannels, which is expressed as

\[ h_{tp} = \left( C_1 Co^{C_2} + C_3 Bo^{C_4} Fr_{lo} \right) (1 - x)^{0.8} h_1 \]  
\[ (7-25) \]

where \( h_1 \) is the liquid-alone flow heat transfer coefficient, \( x \) is the vapor quality, and \( C_1 \sim C_4 \) are constants that are function of the all-liquid Reynolds number, \( Re_{lo} \), and the reduced temperature.

As shown in Figure 7-19, the Yan and Lin (1998) correlation dramatically overpredicts (more than an order of magnitude) the data with \( x < 0.5 \), however, it reasonably represents some of the data in higher quality region. As has been mentioned previously, the only difference between the current study and that of Yan and Lin (1998) is the channel size and shape. Yan and Lin (1998) reported the decrease of heat transfer with increasing vapor quality for
some test conditions for $x > 0.2$, which explains why this correlation correlates the current data in higher quality regions better than other models.

![Figure 7-19 Comparison of measured $h_p$ with predictions of Yan and Lin (1998)](image)

7.4.8 Lee and Lee (2001a)

Lee and Lee (2001a) identified two-phase convection as the dominant mechanism for $0.2 < x < 0.7$ in their experiments. In conventional channels, Chen (1966) applied the Reynolds analogy to both the single-phase flow and the two-phase flow and reported that the enhancement factor, $F$ (the ratio of the boiling heat transfer to the single-phase heat transfer coefficient), could be represented by the Martinelli parameter in the turbulent film-flow region. Lee and Lee (2001a) extended Chen’s (1966) approach to the laminar film flows, and correlated their experimental data for the range of $Re_l \leq 200$ as

$$h_p = Fh_l$$  \hspace{1cm} (7-26)

$$F = 10.3\alpha^{0.398} \phi_l^{0.598}$$  \hspace{1cm} (7-27)

where $\alpha$ is the aspect ratio, $h_l$ is the single-phase heat transfer coefficient based on the liquid component flow, and $\phi_l$ is the two-phase frictional multiplier, which is evaluated based on the correlation suggested by Lee and Lee (2001b).

As seen in Figure 7-20, the original Lee and Lee (2001a) correlation, which evaluates the two-phase frictional multiplier $\phi_l$ using the model proposed by Lee and Lee (2001b), overpredicts most of the current data points. However, as has been addressed in Chapter 5, the difference may be introduced from the error in calculating the multiplier, $\phi_l$. The adiabatic two-phase flow correlation developed in Chapter 5, Equation (5-52), has been used to calculate the multiplier $\phi_l$ in Equation (7-27). As can be seen in Figure 7-20, this method gives almost the same results, which indicates that the difference is not attributed to the error in calculating the multiplier, $\phi_l$. 

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7.4.9 Summary of Comparison with Correlation Predictions

None of the nine correlations evaluated above could predict the present experimental data over the entire quality region. Some of them, such as Liu and Winterton (1988), Kandlikar and Balasubramanian (2003) and Lee and Lee (2001a) give reasonably good predictions for data points in certain quality ranges.

The conventional correlations, Chen (1966), Chaddock and Brunemann (1967), Liu and Winterton (1988) and Kandlikar (1990), typically use empirical parameters that are determined by correlating data points in tubes tens or hundreds times larger than the current study. The typical flow region in these large tubes is turbulent flow. As a result, it is not surprising that these correlations failed when used in microchannels.

The correlations of Tran (1998) as well as Lazarek and Black (1982) are based on tube diameters about ten times that of the current study. Both of them are nucleate coiling dominant correlations. As discussed previously, these correlations significantly underpredict the majority of the present data points. In addition, the current data points have also been compared with the pool boiling equation, Stephan and Abdelsalam (1980), as well as the pool boiling heat transfer coefficient calculated from Equation (7-12). Both calculations give values much smaller than the measured results over the entire quality region, which indicates that the contribution of convection heat transfer may be large.

7.5 Discussions and Correlation Development

The behavior for the data with $x < 0.3$ is similar to what is typically found in the nucleate boiling dominant region in conventional/mini-size channels, where the heat transfer coefficient is a weak function of vapor quality and mass flux but a strong function of heat flux (Lazarek and black 1982, Tran et al. 1993, Bao et al. 2000). These data points are plotted as a function of heat flux for different mass fluxes, as shown in Figure 7-21.
Figure 7-21 Experimental $h_p$ data with $x < 0.3$ against the heat flux

Also shown in Figure 7-21 are the predictions of the Gorenflo (1990, 1993) correlation for nucleate pool boiling coefficients, which is expressed as

$$h_{pb} = h_0 F_{PF} \left( \frac{q^n}{q_0^n} \right)^{nf} \left( \frac{R_a}{R_{a0}} \right)^{0.133}$$

where $F_{PF}$ is the pressure correction factor correlated as

$$F_{PF} = 1.2 P_r^{0.27} + 2.5 P_r + \frac{P_r}{1-P_r}$$

The heat flux correction term has an exponent $nf$ that is given as

$$nf = 0.9 - 0.3 P_r^{0.3}$$

where $P_r = P / P_{cr}$ is the reduced pressure. According to Gorenflo (1993), the reference heat transfer coefficient, $h_0$, for R134a is 4,500 W/m² °C under a reference heat flux $q_0 = 20,000$ W/m². The reference surface roughness $R_a0 = 0.40$ µm and the surface roughness of the channel wall was determined to be 0.1 µm as has been described in section 2.4.2.3.

As seen in Figure 7-21, the nucleate pool boiling coefficient represents the trend of the data points as a function of heat flux very well, but underpredicts all the data. The recent work of Haynes and Fletcher (2003) presented the experimental investigation of saturated and subcooled boiling heat transfer of R11 and HCFC123 in smooth copper tubes with diameters of 0.92 and 1.95 mm. Their data in the subcooled and saturated regions were well represented by the simple addition of convective and nucleate boiling contributions. Thus, both the enhancement factor ($F$) and the suppression factor ($S$) can be taken as one for the nucleate boiling dominant region in small channels, and Equation (7-3) is reduced to
\[ h_{sp} = h_t + h_{pb} \quad (7-31) \]

The liquid heat transfer coefficient for laminar flow, Equation (7-2), and the Gorenflo pool boiling correlation are used. As shown in Figure 7-21, this simple method predicts all the data points with \( x < 0.3 \) very well (within \( \pm 23\% \)).

When the vapor quality is greater than 0.3, the heat transfer coefficient increases with the increase of the vapor quality. This qualitative observation suggests that the two-phase forced convection is the predominant mechanism for heat transfer in this region.

The heat transfer coefficient reaches the maximum at vapor quality about 0.5 and then decreases sharply when the quality is greater than about 0.5 – 0.7. The decrease of heat transfer coefficient with increasing quality is similar to the “dry-out” phenomenon in macrochannels, which is also sometimes referred to as the liquid-deficient region. The region is the most commonly encountered in the “direct expansion” evaporators used in air-conditioning and refrigeration systems, where the two-phase flow leaves the unit usually near 100% quality or slightly superheated.

Oh et al. (1998) compared their experimental results of R134a flow evaporation heat transfer in minichannels of 0.75 – 2 mm and found out that the dry-out quality, \( x_{crit} \), was lower as the tube diameter got smaller, the heat flux became larger, and the mass flux became smaller. The current study shows the same effect of mass flux on the dry-out quality, i.e., \( x_{crit} \) increases as the mass flux gets larger, as can be seen from Figure 7-5. In addition, the dry-outs in the current work occur at much lower vapor qualities (\( x = 0.5 – 0.7 \)) than that observed by Oh et al. (1998) (\( x = 0.7 – 0.9 \)). This is thought to be because the mass flux in Oh et al. (1998), \( G = 240 – 720 \text{ kg/m}^2\text{-s} \), was larger that that in the current work, \( G = 87 – 165 \text{ kg/m}^2\text{-s} \), as well as the effect of smaller channel size in the current work. However, there is no indication of \( x_{crit} \) dependence on heat flux in the current experimental range.

In macrochannels, Carey (1992) recommended the following correlation to correct the effect of tube diameter on the dry-out quality

\[ x_{crit} = (x_{crit})_{8mm} \left( \frac{8}{D} \right)^{0.15} \quad (7-32) \]

where \( D \) is the tube diameter in mm.

The dry-out qualities in the present work is correlated as a simple equation to consider the effect of mass flux, which is expressed as

\[ U_{LS} = \frac{G(1-x)}{\rho_l} = 0.039 \quad (7-33) \]

Based on the present data for evaporation heat transfer in microchannels, a semi-empirical correlation is proposed in the form

\[ h_{sp} = h_t + h_{pb} \]
\[ h_p = \begin{cases} 
0.985 \phi_1^{1.719} h_t + h_{pb}, & \text{for } U_{LS} \geq 0.039 \\
18.42 C_{o}^{1.356} \phi_{t,x=\text{crit}}^{1.719} h_t + \frac{x - x_{\text{crit}}}{1 - x_{\text{crit}}} h_v, & \text{for } U_{LS} < 0.039 
\end{cases} \] 

where \( \phi_1^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \) 

\( C = \begin{cases} 
1.279 \times 10^{-9} \lambda^{-0.96} \psi \Re_{\text{po}}^{0.40}, & \text{We}_{GS} \leq 11.0 \We_{LS}^{0.14} \\
1.386 \times 10^{-4} \lambda^{-0.65} \psi^{0.2} \Re_{\text{po}}^{0.52}, & \text{We}_{GS} > 11.0 \We_{LS}^{0.14} 
\end{cases} \) 

In Equation (7-34), the multiplier \( \phi_1 \) in Equation (7-34) is calculated with the adiabatic two-phase flow correlation developed in Chapter 5. \( \phi_{t,\text{crit}} \) is the multiplier at the dry-out point \( x_{\text{crit}} \). The convection number (\( C_{o} \)), which is defined in Equation (7-16), is used to represent the effect of quality on the heat transfer coefficient in the dry-out region. The nucleation term, \( h_{pb} \), is calculated from the Gorenflo pool boiling correlation, Equation (7-28). In addition, we assume that the observation of Haynes and Fletcher (2003) that convection does not suppress the nucleation term (\( S = 1 \)) for subcooled and saturated boiling, can be extended to the quality region before the dry-out. The suppression factors, \( S \), in the Chen (1966) correlation as well as in the Liu and Winterton (1988) correlation were determined to be in the range of \( S = 0.9 \sim 1 \) for the current experimental conditions, which justified the assumption.

As shown in Figure 7-22, the proposed correlation predicts 84% of the data points within deviation of \( \pm 30\% \) and the mean deviation is 17.0%.

![Figure 7-22 Comparison of the proposed heat transfer correlation with the measured data](image)

**7.6 Summary and Conclusions**

This chapter presents the experimental results of evaporation heat transfer of R134a flowing through horizontal, rectangular microchannels of 75 x 812 \( \mu \text{m} \). During the experiments, the mass flux, heat flux, the vapor
quality varied in the range of $87 \sim 165 \text{ kg/m}^2\text{s}$, $10 \sim 20 \text{ kW/m}^2$, $0.1 \sim 0.9$, respectively; and the saturation temperature was maintained at $23 \degree C$.

Ten existing correlations were compared with the current experimental data, and none of them gave a reasonable prediction over the entire vapor quality range. The Liu and Winterton (1988), the Lee and Lee (2001a) correlations reasonably predict the current data for quality less than about 0.7. The NBD region of the Kandlikar’s correlation (Kandlikar and Steinke, 2003) predicts the data with $x > 0.5$ reasonably well. The other correlations scatter in a very large range ($10^{-1}\sim 10^{2}$ times of the measured data). Therefore, great care must be taken when applying conventional correlations to microchannel evaporators.

The results of this study showed that the heat transfer coefficient was a strong function of heat flux when the quality was smaller than 0.3; and the effect of vapor quality and mass flux became more important when the quality was greater than 0.3. Attempts were made to correlate and explain the current data with the conventional nucleation boiling and forced convection mechanism, as well as the dry-out effect. The data points in the low quality region ($x < 0.3$) are well represented (within $\pm 23\%$) by the simple addition of liquid laminar flow heat transfer coefficient and the nucleate boiling contributions calculated from the Gorenflo pool boiling correlation. The analysis suggests that evaporation heat transfer in this study is the combined effect of nucleation and convection.

Based on the present data, a semi-empirical correlation, Equation (7-34), is proposed for the evaporation heat transfer coefficients in microchannels. The proposed correlation predicts 84.0\% of the data points within deviation of $\pm 30\%$ and the mean deviation is 17.0\%.

References


Chapter 8. Microscale Short Tube Orifices

8.1 Chapter Overview
The chapter presents the results of experiments when R134a was flowing through micro-orifices with diameters of 31.0 and 52.0 micrometers, and length-to-diameter ratio of 2.5 and 4.2, respectively. For liquid-upstream/liquid-downstream flow, the discharge coefficient was found to be independent of Reynolds number, which suggests separated flow that was defined in macroscale orifices. For liquid-upstream/two-phase-downstream flow, the experimental results indicate significant departure of flow characteristics from macroscale orifices. The flow was not choked even when the downstream pressure was reduced to more than 400.0 kPa below the saturation pressure corresponding to upstream temperature, whereas in normal size orifices with length-to-diameter ratio larger than two, the flow is typically choked as downstream pressure is reduced below the saturation pressure. This phenomenon was explained by the strengthening of metastable effect in micro-tubes. For liquid-upstream/two-phase downstream flow, a semi-empirical correlation was developed by modifying the discharge coefficient of the conventional orifice equation.

8.2 Literature Review
Orifices are used widely in heat pumps and air conditioners as throttling devices, in flow measurement instruments for metering flows, in automobiles for injecting liquid fuels into combustion chambers at high velocities, etc. Conventional scale orifices can be classified according to L/D ratio, but a well-recognized classification is not available. In this paper, we will adopt the nomenclature given Table 8-1 for consistency. In addition, the abbreviations, LL, GG, LT and TT, stand for the types of flows through an orifice in the form of Liquid-upstream/Liquid-downstream, Gas-upstream/Gas-downstream, Liquid-upstream/Two-phase-downstream, and Two-phase-upstream/Two-phase-downstream, respectively.

Table 8-1 Classification of orifices

<table>
<thead>
<tr>
<th>Thin-walled orifice</th>
<th>Thick-walled orifice</th>
<th>Short tube orifice or orifice tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>L/D</td>
<td>0.125 &lt; L/D &lt; 2</td>
<td>L/D &gt; 2</td>
</tr>
</tbody>
</table>

Orifices used as throttling devices found in conventional applications are normally larger than or close to one millimeter in diameter. However, Shannon et al. (1999) are developing a small cooling system which uses an orifice with diameter of 30.0 ~ 52.0 µm as expansion device. The whole system is about 100 mm square, 2.5 mm thick with cooling capacity of 3 ~ 20 W while operating between 20 °C (evaporation temperature) and 50 °C (condensation temperature). The design of such a system needs to characterize refrigerant flow through micro-orifices with/without flashing.

Studies on flow through macro and micro scale orifices are summarized in Table 8-2. Most of the earlier studies focused on orifices close to or larger than 1 mm, and micro-orifice studies, such as Wang et al. (1999) and Hasegawa (1997) are dealing with single-phase flow only.
Table 8-2 Summary of orifice studies in macroscale and microscale

<table>
<thead>
<tr>
<th>Reference</th>
<th>L/D</th>
<th>Diameter (µm)</th>
<th>Testing condition</th>
<th>Testing Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benjamin and Miller (1941)</td>
<td>&lt; 1</td>
<td>6000 ~ 23000</td>
<td>LT&lt;sup&gt;a&lt;/sup&gt;</td>
<td>Water</td>
</tr>
<tr>
<td>Davies and Daniels (1973)</td>
<td>&lt; 1</td>
<td>762 ~ 1422</td>
<td>LL&lt;sup&gt;b&lt;/sup&gt;, LT, TT&lt;sup&gt;c&lt;/sup&gt;</td>
<td>R12</td>
</tr>
<tr>
<td>Mei (1982)</td>
<td>7 ~ 12</td>
<td>1000 ~ 1700</td>
<td>LT</td>
<td>R22</td>
</tr>
<tr>
<td>Krakow and Lin (1988)</td>
<td>2 ~ 7</td>
<td>889</td>
<td>LT</td>
<td>R12</td>
</tr>
<tr>
<td>Aaron and Domanski (1990)</td>
<td>5 ~ 20</td>
<td>1100 ~ 1720</td>
<td>LT</td>
<td>R12</td>
</tr>
<tr>
<td>Kim and O'Neal (1994a, b)</td>
<td>5 ~ 20</td>
<td>1000 ~ 1720</td>
<td>LT, TT</td>
<td>R134a, R22</td>
</tr>
<tr>
<td>Singh &lt;i&gt;et al.&lt;/i&gt; (2001)</td>
<td>22 ~ 31</td>
<td>1220 ~ 1700</td>
<td>LT, TT</td>
<td>R134a</td>
</tr>
<tr>
<td>Ramamurthi and Nandakumar (1999)</td>
<td>1 ~ 50</td>
<td>300 ~ 2000</td>
<td>LL</td>
<td>Water</td>
</tr>
<tr>
<td>Wang &lt;i&gt;et al.&lt;/i&gt; (1999)</td>
<td>Not reported</td>
<td>150 ~ 370</td>
<td>LL, GG&lt;sup&gt;d&lt;/sup&gt;</td>
<td>Water, nitrogen</td>
</tr>
<tr>
<td>Hasegawa &lt;i&gt;et al.&lt;/i&gt; (1997)</td>
<td>0.05 ~ 1.14</td>
<td>10 ~ 1000</td>
<td>LL</td>
<td>Water, silicon oils, glycerin water</td>
</tr>
<tr>
<td>This paper</td>
<td>2.5 ~ 5</td>
<td>31 ~ 52</td>
<td>LL, LT</td>
<td>R134a</td>
</tr>
</tbody>
</table>

<sup>a</sup>LT: Liquid upstream Two-phase downstream  
<sup>b</sup>LL: Liquid upstream Liquid downstream  
<sup>c</sup>TT: Two-phase upstream Two-phase downstream  
<sup>d</sup>GG: Gas upstream Gas downstream

8.2.1 LL and GG Flow

In a flow meter, normally the thickness of the orifice plate used should not exceed 1/8 of the orifice bore (Cusick, 1961), and the flow is typically single phase. The single-phase flow through a thin-walled orifice (L/D < 0.125) can be calculated by the orifice equation:

\[
m = C_d A \sqrt{\frac{2p\Delta p}{\rho}} (1 - \beta^4)
\]

(8-1)

where \(\beta = \frac{D}{D'}\). Here, D and D' are the orifice diameter, and the conduit diameter, respectively. The discharge coefficient, \(C_d\), is a function of \(\beta\) and the orifice flow Reynolds number based on the orifice diameter (D).

\[
Re = \frac{\rho v D}{\mu}
\]

(8-2)

where \(v\) is the fluid velocity and \(\mu\) is the fluid viscosity.

Equation (8-1) can also be used for thick-walled orifices and orifice tubes, but \(C_d\) for these two types of orifices may also be a function of L/D ratio and orifice diameter, in addition to \(Re\) and \(\beta\).

Ramamurthi and Nandakumar (1999) evaluated the discharge coefficients for demineralized water flow through sharp-edged orifices of D = 0.3 ~ 2 mm, and L/D = 1 ~ 50. The single-phase flow inside the tube was categorized as separated flow or reattached flow. In the separated flow region, \(C_d\) is not a function of Reynolds number, and the pressure drop across the orifice is due mainly to the pressure loss at the orifice entry, which is proportional to the dynamic head \(\frac{\rho v^2}{2}\). For reattached flow, e.g., the flow reattached on the wall after vena contracta, and the pressure drop should include the frictional losses. Since the frictional losses are related to \(Re\), \(C_d\) for reattached flow is a function of \(Re\). The test results for 0.3 mm orifice with L/D = 5 demonstrated that \(C_d\) initially increased with an increase of Reynolds number when \(Re < 7000\) (reattached region), then abruptly dropped
to values corresponding to separated flows and did not change with Re any more. For 2 mm orifice with L/D = 5, the
transition to separated flow occurred at Re of 35,000. In separated flow region, the authors demonstrated that the
discharge coefficient was larger for smaller orifices, and attributed this behavior to the effect of surface tension.

Wang et al. (1999) used 150 ~ 370µm orifices as flow restriction device in micro check valves. They tested
water and nitrogen flow through these orifices, and found that the macro-scale model correctly predicted the
qualitative characteristics of the valve, but gave about 20-30% lower flow rate.

Hasegawa et al. (1997) tested the flow of water, silicon oils and solution of Glycerin through orifices
ranging from 10 µm to 1 mm. For all the data, the Reynolds numbers were below 1,000. The experimental results
were compared with numerical analysis of a Newtonian fluid by the finite element method. The analysis showed that
the predicted pressure drop underestimated the measured value when the size of the orifice was smaller than 35 µm.
They concluded that the flow through very small orifices is different from that through ordinary size ones that can be
solved with a Navier-Stokes equation and a different mechanism may be dominant in the flow through very small
orifices.

8.2.2 LT Flow

When orifices are used as the expansion devices in refrigeration industry, they typically work under liquid
upstream two-phase downstream (LT) condition. The flow may be choked when liquid flashes while passing
through the expansion device. Choked flow is defined as the phenomenon that occurs when the mass flow rate
remains constant even when there is a further reduction in downstream pressure. A constant-flow area expansion
device that is sensitive to the downstream pressure (not choked) would be detrimental to system performance and
reliability (Aaron and Domanski, 1990).

8.2.2.1 Unchoked Flow

Typically, an orifice with L/D < 1 does not choke the flow at normal air-conditioner/heat pump operating
conditions. Benjamin and Miller (1941) performed tests to determine the flow characteristics of saturated water
through sharp-edged thin plate orifices with L/D < 1 and D > 6 mm. They found that no flashing occurred until after
the water was through the orifice, and the flow was not choked.

The experimental investigation of Davies and Daniels (1973) demonstrated that when refrigerant flashed on
passage through an orifice with L/D < 1 and 0.762 ≤ D ≤ 1.422 mm, the flow rate was reduced. However, the
reduction in flow was independent of the upstream and downstream pressures and the pressure difference across the
orifice, but was a function of the downstream quality. They proposed a model based on correction of the orifice
equation using downstream quality.

8.2.2.2 Choked Flow

Krakow and Lin (1988) tested R12 flow through orifices with L/D of 2 and 7, and diameter of 0.889 mm.
The authors found that the flow was primarily dependent on the upstream conditions and not on the downstream
pressures, thus a choking phenomenon was indicated. Aaron and Domanski (1990), and Kim and O’Neal (1994a, b)
investigated flow through orifice with L/D between 5 ~ 20. Their results demonstrated that critical flow was
established when the downstream pressure was below the saturation pressure corresponding to the upstream
temperature. Therefore, choked flow is typically established for conventional size orifices with L/D > 2.
The speed of sound in two-phase flow is much lower than that in the corresponding single-phase flows. Thus, when flashing occurs inside the capillary tube or orifice tube, the flow may be choked. For orifices with \( L/D < 1 \), flashing occurs downstream of the orifice, as a result the flow is not choked at normal air-conditioner/heat pump operating conditions (Benjamin, 1941). However, the point of flashing in conventional scale capillary tubes have been found to be not at the saturation pressure, \( P_s \), but at a pressure \( (P_v) \) lower than \( P_s \) because of the metastable effect. \( P_v - P_s \) is designated as the underpressure of vaporization. Koizumi and Yokoyama (1980), Chen et al. (1990) and Li et al. (1990) reported that the underpressure of vaporization increases with a decrease of the inside diameter of the tube. For example, Chen et al. (1990) presented the semi-empirical correlation based on the experimental results of R12 flowing through capillary tubes of \( 660.0 \, \mu m < D < 1170.0 \, \mu m \),

\[
P_s - P_v = 0.679 \left( \frac{v_r}{v_v - v_f} \right) Re^{0.914} \left( \frac{\Delta T_{sub}}{T_{crit}} \right)^{-0.208} \left( \frac{D}{D_{ref}} \right)^{-3.18} \left( \frac{\sigma}{kT_s} \right)^{0.5} \tag{8-3}
\]

where \( T_c \) is the critical temperature, \( T_s \) is the liquid temperature, \( k \) is the Boltzmann’s constant, \( \sigma \) is the liquid refrigerant surface tension, \( D \) is the tube diameter, and \( D_{ref} \) is the reference length. Keeping mass flux constant, rearrangement of the above equation gives the following relationship between underpressure and tube diameter,

\[
(P_s - P_v) \sim \frac{1}{D^{2.27}} \tag{8-4}
\]

Equation (8-4) indicates that the underpressure increases very fast when the tube diameter decreases, and the metastable effect is strengthened in smaller tubes. The underpressure for R134a flow was calculated with different tube diameters under a typical condition of upstream temperature at 38.0 °C, upstream subcooling 10.0 °C, and mass flux of 5,000 kg/m²-s. For a tube with \( D > 1.0 \, \text{mm} \), the underpressure is less than 27.0 kPa, which is small and can be neglected. However, the underpressure for a 600μm tube is about 90.0 kPa, which must be considered. If the trend represented in Equation (8-4) continues, the underpressure for a 30.0 ~ 50.0 μm diameter tube might be so high that flashing could not be initiated inside the tube. In that case, the fluid will flow through the orifice tube as metastable superheat liquid, and the vaporization will occur downstream of the orifice tube. Therefore, it is possible that a micro-orifice tube behaves similar to a conventional scale thin-walled-orifice (\( L/D < 0.125 \)), which does not choke the flow.

The main objective of this work is to investigate liquid flow with flashing (LT) through micro-orifices. Before proceeding to LT flow, it is necessary to understand the liquid only (LL) flow in micro-orifices first.

### 8.3 Results and Discussions

#### 8.3.1 Liquid Upstream, Liquid Downstream (LL)

The discharge coefficient \( C_d \) for liquid upstream liquid downstream flow was calculated using Equation (8-1), based on the measured pressure drop, mass flow rate, and orifice cross-section area. The results along with the error bars are shown in Figure 8-1. The error bars were determined using the uncertainty propagation function based on Taylor and Kuyatt (1994).

For the orifice with a diameter of 31.0 μm, \( C_d \) is about the same for the entire range of \( 1,500 < Re < 4,500 \), which indicates a separated flow. Ramamurthi and Nandakumar (1999) showed that for an orifice with \( D = 300 \, \mu m \)
and $L/D = 5$, the discharge coefficient initially increased with increase of Reynolds number when $Re < 7000$ (reattached region), then abruptly dropped to values corresponding to separated flows and did not change with Reynolds number any more. On the other hand, the transition to separated flow for an orifice with $D = 2,000 \, \mu m$ and $L/D = 5$, occurred at $Re$ of 35,000. The $L/D$ ratio for $31 \, \mu m$ orifice in the current work is close to 5 ($L/D = 4.2$), and separated flow was reached for $Re > 1500$. Hence, it seems that the transition from reattached flow to separated flow occurs at lower Reynolds number in micro-orifices than that in larger ones. This trend is the same as what reported by Ramamurthi and Nandakumar (1999).

![Graph showing micro-orifice experimental results for liquid upstream liquid downstream (LL) flows](image)

**Figure 8-1** Micro-orifice experimental results for liquid upstream liquid downstream (LL) flows

In the separated flow region, Ramamurthi and Nandakumar (1999) demonstrated that the discharge coefficient was larger for smaller orifices, and they attributed this phenomenon to the effect of surface tension. The contribution of pressure by surface tension force is $2\sigma/r$, where $r$ is the radius of the orifice and $\sigma$ is surface tension. The surface tension of R134a at the current test condition is about 0.0083N/m, which gives driving pressure for $31.0 \, \mu m$ orifice about 1.0 kPa. This value is about the same as water flowing through $300 \, \mu m$ orifice. However, the lowest pressure drop in their experiment is about 20kPa, which makes the surface tension effect significant. In the current work, the lowest pressure drop is 136kPa, thus the surface tension effect can be neglected. This explains why $C_d$ for a $31.0 \, \mu m$ ($\sim 0.67$) is smaller than that of a $300 \, \mu m$ orifice ($\sim 0.8$) in separated flow region, but it is almost the same as a $2,000 \, \mu m$ orifice ($\sim 0.67$) for which surface tension effect is also negligible.

The discharge coefficients for a $52.0 \, \mu m$ orifice are a little larger than that for a $31 \, \mu m$ orifice, but they still can be characterized by a constant value. Discharge coefficients of 0.70 for $52 \, \mu m$, and 0.68 for $31 \, \mu m$ orifices were shown to be capable of predicting all the experimental data within $\pm 3.0\%$. In conclusion, the results suggest that a LL flow through a micro-orifice used in this study is consistent with macroscale results.
8.3.2 Liquid Upstream, Two-phase Downstream (LT)

The experimental range of liquid flow with phase change (LT) was chosen to be similar to that of Kim and O’Neal (1994a). The upstream subcooling changed from 4.0 °C to 26.0 °C, upstream pressure ranged from 900.0 to 1491.0 kPa, and downstream quality varied between 4.0% and 32.0%.

Figure 8-2 shows the relationship between the mass flow rate and pressure drop for all the experimental data. The mass flow rate is a strong function of pressure drop, no matter what the values of upstream subcooling, upstream pressure and downstream pressure are. When the mass flow rate data are compared with the pure liquid flow data under the same pressure drop, the values for LT flows are lower, but the differences are very small, as shown in Figure 8-2 (a) and (b). Figure 8-3 shows the differences between the experimental data, \( m_{\text{meas}} \), and calculated results from the liquid flow model, \( m_{\text{calc}} \). The liquid flow model underpredicts almost all of the data, but most of the errors are within \(-15\%\). In addition, it seems that the higher the downstream quality, the larger the difference.
Figure 8-3 Differences of liquid model predictions and experimental data

Figure 8-2 and Figure 8-3 indicate that the liquid flow with flashing in micro-orifices can be approximated as pure liquid flow. In a macroscale orifice tube (Krakow and Lin 1988, Aaron and Domanski 1990, Kim and O’Neal 1994a, b, Singh et al 2001), the flow is normally choked when flashing occurs, and the flow characteristics for LT flow are quite different from that of LL flow. Typically, the mass flow rate is not directly related to pressure drop, but proportional to the upstream pressure and the upstream subcooling. In addition, the flow rate is not a function (or is a very weak function) of downstream pressure. The differences between the current work and macroscale results suggest that the choking may not exist in the present experiment.

Figure 8-4 (a) presents the effect of downstream pressure on mass flow rate for a 52.0µm orifice. The upstream pressure was kept constant at 1310.0 ± 1 kPa. Three different upstream subcooling values, 4.4 ± 0.7 °C, 9.6 ± 0.5°C and 13.7 ± 0.4 °C are presented. The saturation pressure (Psat) corresponding to the upstream temperature (or subcooling) is also listed. The downstream pressure was reduced from at least 200.0 kPa below the corresponding saturation pressure. The mass flow rate increased monotonically as downstream pressure decreased, and it could be predicted by the pure liquid flow model (Equation (8-1)) within 15% error. Hence, there is no indication of choking for the present experimental range for a 52.0µm orifice. In addition, the flow rate is only a very weak function of upstream subcooling, with the lower the subcooling the lower the mass flow rate.

The downstream pressure effect on the mass flow rate for a 31.0 µm orifice under a upstream pressure of 1310.0 kPa and three upstream subcooling levels (4.8 ± 0.5 °C, 13.9 ± 0.2 °C and 21.5 ± 0.2 °C) is shown in Figure 8-4 (b). For an upstream subcooling of 21.5 °C, the flow rate increases monotonically with the decrease of downstream pressure, which suggests that choking did not occur. For an upstream subcooling of 4.8 and 13.9 °C, the flow rate increases proportionally for most of the region when the downstream pressure is reduced. Nevertheless, when the downstream pressure is reduced below 450.0 kPa, the mass flow rate remains almost constant. This might be because of experimental error or an indication of choked flow.
inlet subcooling = 4.4 C, Psat=1173 kPa
inlet subcooling = 9.6 C, Psat=1022 kPa
inlet subcooling=13.7 C, Psat = 915 kPa
Equation (8-1) prediction, Cd = 0.70

inlet subcooling = 4.8 C, Psat = 1158 kPa
inlet subcooling = 13.9 C, Psat = 909 kPa
inlet subcooling = 21.5 C, Psat = 733 kPa
Equation (8-1) predictions, Cd = 0.68

Figure 8-4 Flow dependency on downstream pressure as a function of upstream subcooling

The L/D ratio of the 31 µm orifice (L/D = 4.2) is very close to the lowest range of Kim and O’Neal (1994a), where L/D = 5 ~ 20. In addition, both investigations used the same refrigerant (R134a) and similar experimental conditions. Kim and O’Neal (1994a) demonstrated that the choked flow conditions were typically established when the downstream pressures were reduced below the saturation pressure corresponding to the upstream temperature. However, no indication of choking was observed from Figure 8-4 (b) for downstream pressure as low as 400 kPa below the saturation pressure, P_{sat}. In addition, Krakow and Lin (1988) reported choked phenomenon for R12 flowing through orifice of L/D = 2 and D = 889 µm, but choking does not exist in the current study for the flow through a 52.0 µm orifice (L/D = 2.5) for a wide range of experimental conditions.

The above analysis suggests that choked flow is much more difficult to establish in microscale orifice-tubes than in conventional scale ones, which can be attributed to the strengthening of metastable effect in smaller tubes. This means that the underpressure at the flashing point is extremely high, thus the flashing can not be initiated inside the microscale orifice-tubes if the downstream pressure is not low enough. As a result, the choked flow that normally occurs in conventional scale orifice/capillary tubes does not occur in microscale orifice-tubes under normal downstream pressure conditions. Although the current work focuses on very short orifice tubes with L/D < 5, it is suggested that even if the same experiments were run in longer orifice tubes or even capillary tubes with the same diameters, the results would be similar.

For tubes at the order of one millimeter in diameter, Chen et al. (1990) attributed the strengthening of metastable effect in a smaller tube to weaker pressure fluctuations when the tube size is reduced, but other mechanisms may be dominant in micro-tubes. For example, micro-tubes typically have a smoother surface than large ones, thus are lacking nucleate sites.
When the flashing process is suppressed in micro-tubes, the fluid ejecting from the orifice is in a metastable superheated liquid state. When the superheated liquid enters a macroscale downstream pipe, bubbles will grow instantly at a very high rate. The explosive formation of vapor may be accompanied by some pressure losses, or it may reduce the pressure recovery after the orifice-tube. For both cases, the mass flow rate will be smaller than Equation (8-1) predicts assuming pure liquid flow. The pressure drop associated with the flashing may also be proportional to the extent of metastability, and the higher the metastable effect (or the higher the downstream quality), the larger the pressure drop might be. This speculation explains the trend that is shown in Figure 8-3 – the larger differences between measured data and Equation (8-1) predictions occur at larger down stream qualities.

The above observations were used to develop a semi-empirical model for LT flow through microscale orifice-tubes, based on the correction of the discharge coefficient of the orifice equation.

\[ m = (1 - C_x x_{out}) C_d A \sqrt{2 \rho \Delta P} \]  \hspace{1cm} (8-5)

where \( x_{out} \) is the downstream vapor quality and \( C_x \) is an empirically determined constant. The discharge coefficient, \( C_d \), is empirically determined value from LL test, which is 0.70 for a 52 \( \mu \)m orifice-tube and 0.68 for a 31.0 \( \mu \)m orifice-tube. A constant \( C_x \) value of 0.416 was determined through data regression. This method predicts 90% of the experimental data within a deviation of \( \pm 5\% \), as shown in Figure 8-5 (a).

The present data points are also compared with the macroscale orifice tube models proposed by Kim and O’Neal (1994a), Kim and O’Neal (1994b), and Aaron and Domanski (1990). As shown in Figure 8-5, most of the data points are not within \( \pm 5\% \) of the model predictions. Since these models are based on choked flow, it is not surprising that they do not predict the present data well.
8.4 Summary and Conclusions

The current work focuses on micro-orifices used as flow restrictors and expansion devices. Orifice tubes with inner diameter of 31.0 and 52.0 µm, L/D ratio of 2.5 and 4.2, respectively, were tested under Liquid upstream, Liquid downstream (LL) and Liquid upstream, Two-phase downstream (LT) conditions. The diameters were measured, and the entrance surface conditions were characterized with a 1000-times magnification microscope.

For liquid flow without flashing downstream (LL), the discharge coefficients were determined to be constant for Reynolds number ranging from 1500 to 6700.

When flashing occurs at downstream of micro-orifice (LT), the flow is different from normal sized orifice tubes. The flow rate is still a strong function of pressure drop as in LL flow, but weakly dependent on upstream subcooling because of liquid density change. Experiments have been performed by keeping the upstream pressure and subcooling constant, while reducing the downstream pressure. For a 52.0 µm orifice-tube, mass flow rate increased with the decrease of downstream pressure for all three subcooling values (4.8°C, 13.9°C, and 21.5°C), even when the downstream pressure has been reduced to as low as 600.0 kPa below the saturation pressure. Similar behaviors were observed for a 31.0 µm orifice-tube when the downstream pressure as low as 400 kPa below the saturation pressure. Hence, no indication of choking was observed for a wide range of experimental conditions in the microscale orifice-tubes. However, for orifice-tubes with inner diameter around 1 mm and L/D > 2, the flow is choked when downstream pressure is lower than saturation pressure (Krakow and Lin, 1988, Aaron and Domanski, 1990 and Kim and O’Neal 1994a, b).

The distinct non-choking phenomenon observed in microscale orifice-tubes was attributed to the strengthening of metastable effect in small tubes. Semi-empirical correlation has been proposed based on correction of the discharge coefficient in the orifice equation using the downstream quality. The correlation predicts 90% of the data within ± 5%, using the experimentally determined LL flow discharge coefficients.

References


Taylor, B.N. and C.E. Kuyat, 1994, “Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results,” *National Institute of Standards and Technology Technical Note 1297*

Appendix – Instrumentation Calibration and Crosschecking

Figure 1 Micromotion® flow meter readings vs. flow rates calculated from digital balance

y = 0.0171x^5 - 0.2483x^4 + 1.2557x^3 - 2.3237x^2 + 2.3798x - 0.4516
R^2 = 0.9994

Figure 2 Rheotherm® flow rate meter (model TU1/16) calibration
Figure 3 Pressure transducer cross-checking: 0 ~ 6.8 kPa vs. 0 ~ 172 kPa

Figure 4 Pressure transducer cross-checking: 0 ~ 6.8 kPa vs. 0 ~ 68 kPa
Figure 5 Thermocouple Calibration - Deviation from NIST