Experimental Study on Heat Transfer Characteristics of Microchannel Systems Using Liquid Crystal Thermography

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Abstract

Experimental Study on Heat Transfer Characteristics of Microchannel Systems Using Liquid Crystal Thermography

Roland Ssonko Muwanga, Ph.D.
Concordia University, 2007

Microchannel heat transfer governs the performance of the microchannel heat sink, which is a recent technology aimed at managing the stringent thermal requirements of today’s high-end electronics. The present thesis investigates the single-phase and flow boiling heat transfer characteristics in microchannel systems. A state-of-the-art test facility has been constructed for the experimental investigations. It is composed of a closed loop flow setup, a microscopic imaging system and an automated data acquisition system. A technique for using un-encapsulated thermochromic liquid crystals (TLC’s) to measure the local heat transfer coefficient in microchannel geometries is developed. A unique localized calibration of the TLC material is employed to minimize the effects of lighting non-uniformity and variable coating thickness. Measurements are carried out in three different microtubes, and in two innovative parallel and radial microchannel heat sinks with 3.5 cm² footprint areas. The working fluids are distilled water, FC-72 and air.

Local single-phase heat transfer and frictional pressure drop measurements are presented for the laminar, transitional and turbulent flow regimes, in microtubes down
to 0.25 mm. Local flow boiling heat transfer coefficient data are presented for a quality up to 0.3, and for the conditions investigated, suggest a nucleation dominated region. Flow boiling oscillation characteristics in two parallel silicon microchannel heat sink configurations aimed at micro/nano-spacecraft thermal management are investigated. One is a standard heat sink with 45 parallel channels, whereas the second is similar, except with cross-linked paths at three locations. The oscillation amplitudes are relatively large and identical in frequency for the fluid pressure and temperature. Oscillation properties for the standard heat sink are correlated for different heat fluxes, while a first glimpse of the cross-linked heat sink performance under flow boiling instability conditions is presented. Optimization of a radial inflowing microchannel heat exchanger has been investigated numerically, after preliminary fabrication and experimentation trials. Unique to this optimization was consideration of channels with axially varying cross-sections. Three-dimensional conjugate analysis shows that when constrained by a fixed channel outlet area, increasing the channel inlet area will improve the thermal performance.

Overall, a method utilizing un-encapsulated TLC thermography for local heat transfer measurements in microgeometries has been developed, while investigating the flow and heat transfer characteristics in microchannel systems. The present work along with the advances in MEMS based manufacturing is expected to lead to the creation and development of a number of miniaturized technologies, from DNA analysis to power-plants-on-chips.
Acknowledgments

Achieving the work presented in this thesis would not have been possible without the help and support of a number of people. I would like to thank my supervisor and mentor Dr. Ibrahim Hassan. He continually challenged me to achieve above and beyond what I thought was possible, and believed in me at each step of the way. I would like to thank the research colleagues I have had the pleasure of working with. Many have graduated now including Korhan Pehlivan, Mohamed Abdelgawad, Ariane Immarigeon, Patricia Phutthavong and Min Min Le. As well, the present research group members, Dino Bowden, Tariq Ahmad, Minh Dang, Wael Saleh, Mohamed Ghorab, Ayman Megahed, Mohamed Rahman, Chad Zhang and Dr. Sung In Kim. The Washington Technology Center is gratefully acknowledged for their assistance and capabilities in fabricating the silicon heat sinks. I would like to extend my gratitude to Pratt and Whitney Canada for financial assistance and in particular to Sri Sreekanth and Terry Lucas.

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To my parents, Mary Victoria and Roland Sr., for all your love.
Contents

List of Figures xiv

List of Tables xxii

Nomenclature xxiii

1 Introduction 1

1.1 Motivation ............................................. 1

1.2 Objectives and Organization ........................ 2

2 Literature Review 4

2.1 Single Phase Flow and Heat Transfer ............... 4

2.1.1 Single Channel Studies .......................... 4

2.1.2 Multi-Channel Microsystems ....................... 5

2.2 Boiling Heat Transfer ................................ 8

2.2.1 Small Diameter Tubes ............................. 9

2.2.2 Flow Boiling of Dielectric Fluids in Mini and Microchannels ............................ 10
2.2.3 Flow Boiling Incipience ........................................ 11
2.2.4 Boiling Instabilities ........................................... 13
2.2.5 Novel Multi-Channel Concepts ................................. 18
2.3 Heat Transfer Measurement in Microsystems ................. 19
2.3.1 Thermography .................................................. 21
2.3.2 Summary ....................................................... 24

3 Test Facility and Test Methods .................................. 25
3.1 Flow Loop ......................................................... 25
3.1.1 Degassing Procedure .......................................... 29
3.2 Instrumentation and Data Acquisition .......................... 31
3.3 Image Acquisition ................................................ 33
3.4 Uncertainty ....................................................... 36

4 Un-encapsulated Liquid Crystal Thermography for Mini and Micro
   Geometries .................................................................. 38
4.1 Liquid Crystals for Temperature Measurement .................. 41
4.2 Color Temperature Interpretation .................................. 43
4.2.1 Intensity Based Image Processing ............................... 44
4.2.2 True Colour Image Processing (Hue Method) ................. 46
4.3 TLC Material .......................................................... 46
4.4 Coating Application ............................................... 47
4.5 Calibration ........................................................... 49
4.5.1 Stage I Calibration ................................................. 52

ix
5 Transport Phenomena in Microtubes

5.1 Single Phase

5.1.1 Data Reduction

5.1.2 Friction Factor

5.1.3 Heat Transfer Coefficient

5.2 Boiling

5.2.1 Data Reduction

5.2.2 Measurement Scope

5.2.3 Boiling Onset at Low Mass Flux

5.2.4 Wall Temperature Visualization and Measurement During Flow Boiling

5.2.5 Two-Phase Flow Heat Transfer Coefficient
## 5.3 Summary

<table>
<thead>
<tr>
<th>6 Transport Phenomena in a Cross-Linked Heat Sink</th>
<th>135</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.1 Design</td>
<td>136</td>
</tr>
<tr>
<td>6.2 Heat Sink Packaging</td>
<td>139</td>
</tr>
<tr>
<td>6.2.1 Packaging Method I</td>
<td>139</td>
</tr>
<tr>
<td>6.2.2 Packaging Method II</td>
<td>142</td>
</tr>
<tr>
<td>6.3 Single Phase Flow</td>
<td>144</td>
</tr>
<tr>
<td>6.3.1 Methods and Data Reduction</td>
<td>144</td>
</tr>
<tr>
<td>6.3.2 Pressure Drop Measurements</td>
<td>152</td>
</tr>
<tr>
<td>6.3.3 Local Heat Transfer Measurements</td>
<td>155</td>
</tr>
<tr>
<td>6.4 Flow Boiling</td>
<td>164</td>
</tr>
<tr>
<td>6.4.1 Procedure and Data Reduction</td>
<td>164</td>
</tr>
<tr>
<td>6.4.2 Flow Boiling Onset and Limits</td>
<td>165</td>
</tr>
<tr>
<td>6.4.3 Flow Boiling Oscillations</td>
<td>173</td>
</tr>
<tr>
<td>6.4.4 Discussion</td>
<td>184</td>
</tr>
<tr>
<td>6.5 Summary</td>
<td>186</td>
</tr>
</tbody>
</table>

## 7 Investigating a Radial Flow Microchannel Heat Exchanger

| 7.1 Design and Fabrication Trials                  | 189 |
| 7.2 Thermal Resistance Analysis                    | 193 |
| 7.2.1 Mathematical Model                           | 193 |
| 7.2.2 Problem Analysis                             | 198 |
| 7.2.3 Results                                      | 199 |
7.3 Three-dimensional Conjugate Analysis ......................... 203
  7.3.1 Formulation and Solution Methodology .................... 203
  7.3.2 Benchmark Study and Grid Independence Evaluation ...... 204
  7.3.3 Results ................................................. 208

7.4 Thermal Resistance Model Compared to

  Three-Dimensional Conjugate Analysis ......................... 224

7.5 Summary ..................................................... 230

8 Closing ......................................................... 231

  8.1 Conclusions and Contributions ............................... 231
  8.2 Recommendations .......................................... 234
    8.2.1 TLC Methods ........................................... 234
    8.2.2 Microtubes ............................................. 235
    8.2.3 Microchannel Heat Sinks ............................... 236

References ......................................................... 238

Publications From Thesis Work .................................... 253

A Experimental Data ............................................... 256

  A.1 Single Phase Flow and Heat Transfer in
       Microtubes ............................................. 256
  A.2 Flow Boiling in Microtubes ................................ 260
  A.3 Microchannel Heat Sink Heat Transfer and Boiling Oscillations .... 265
B Example Code

C Simplified Analysis

C.1 Derivations .................................................. 309
   C.1.1 Parallel Channel Derivations ....................... 309
   C.1.2 Radial Channel Derivations ....................... 314
   C.1.3 Parallel Channel Code Validation .................. 316

C.2 Results for Parallel Channel System .................. 319
   C.2.1 Parallel Channels: Influence of Boundary Condition .... 319
   C.2.2 Parallel Channels: Influence of Wall Contour
       Polynomial ................................................... 322

D Part Drawings ................................................. 325

D.1 Microtube Drawings ....................................... 326
D.2 Cross-linked Heat Sink Drawings ....................... 327
D.3 Radial Heat Exchanger Drawings ....................... 333
D.4 Parallel Channel Packing System I ..................... 339
D.5 Parallel Channel Packaging System II .................. 341
D.6 Radial Channel Packaging System ....................... 346
List of Figures

3.1 Schematic of the experimental test facility. ...................... 26
3.2 Schematic of the data acquisition system. ...................... 32
3.3 Schematic of the image acquisition system. .................... 34
4.1 States of liquid crystal substances, from Anderson (1999). .......... 40
4.2 Thermochromic liquid crystal characteristics. .................. 42
4.3 Color descriptor response versus temperature a)intensity of monochrome
system, b)intensity of color primaries and c)hue angle. ................ 45
4.4 Constant temperature region hue angle histogram. .............. 51
4.5 Depiction of calibration data processing. ...................... 53
4.6 Stage I corrected calibration curves. ......................... 55
4.7 Typical calibration curve fits for various ROI positions at a given tra-
verse location. ................................................. 58
4.8 Stage II corrected calibration curves. ......................... 59
4.9 Calibration curves at different time periods. .................... 60
4.10 Configuration for Test Module 1. ........................... 63
4.11 Configuration for Test Module 2. ........................... 66
4.12 Configuration for Test Module 3. ........................................ 69
4.13 Sample measured wall temperature and calculated fluid bulk temperature. ........................................ 76
4.14 Streamwise averaged Nusselt numbers for laminar flow cases. ............ 78
4.15 Streamwise averaged Nusselt numbers for various flowrates. ............ 79
4.16 Variation of the averaged Nusselt number with Reynolds number, with comparison to correlations. ................................. 81
5.1 Microtube friction factors versus Reynolds number. ......................... 90
5.2 Typical raw to processed results. ......................................... 92
5.3 Streamwise Nusselt numbers - 1.0 mm, H$_2$O. .......................... 94
5.4 Streamwise Nusselt numbers - 0.5 mm, H$_2$O. .......................... 95
5.5 Streamwise Nusselt numbers - 1.0 mm, FC-72. .......................... 96
5.6 Streamwise Nusselt numbers - 0.5 mm, FC-72. .......................... 97
5.7 Streamwise Nusselt numbers - 0.25 mm, FC-72. .......................... 98
5.8 Laminar streamwise Nusselt number - 1.0 mm, H$_2$O. .................... 99
5.9 Laminar streamwise Nusselt number - 0.5 mm, H$_2$O. .................... 100
5.10 Laminar streamwise Nusselt number - 1.0 mm, FC-72. .................. 101
5.11 Laminar streamwise Nusselt number - 0.25 mm, FC-72. .................. 102
5.12 Average Nusselt numbers. .............................................. 103
5.13 (a,c) Heat flux versus wall temperature, measured by embedded thermocouple. (b,d) Pressure drop versus heat flux. ................. 108
5.14 TLC wall temperature visualization over time during boiling onset, 79
to 88 z/D, \( q'' = 30.4 \text{ kW/m}^2, G = 770 \text{ kg/(m}^2 \cdot \text{s)} \). Red is lower
temperature region, and flow is from left to right. .................. 112

5.15 TLC wall temperature time trace - 3 separate locations imaged at
separate time instants, \( q'' = 50.7 \text{ kW/m}^2, G = 770 \text{ kg/(m}^2 \cdot \text{s)} \). ....... 114

5.16 Typical power spectrum of oscillating wall temperature as measured by
TLC, for corresponding waveforms in Figure 5.15, \( q'' = 50.7 \text{ kW/m}^2, \\
G = 770 \text{ kg/(m}^2 \cdot \text{s)} \). .................................................. 116

5.17 Wall temperature time trace at a far downstream (83 z/D) and an
upstream (18 z/D) location, \( q'' = 54.1 \text{ kW/m}^2 \). ......................... 117

5.18 Wall temperature time trace at a fixed location with increasing heat
flux. ................................................................. 118

5.19 Wall temperature oscillations and corresponding power spectrum with
increasing heat flux, taken at a distance of approximately 32 z/D. ... 120

5.20 Wall temperature fluctuations highlighting oscillatory behaviour and
high wall temperature gradient front. Red is cool and blue is hot,
black is high temperature past the clearing point of the TLC. Covering
a time span of 0.3 s, at 78 z/D to 88 z/D and \( q'' = 79 \text{ kW/m}^2, G = 770 \\
\text{kg/(m}^2 \cdot \text{s)} \), and flow is from left to right. ......................... 121
5.21 Wall temperature fluctuations highlighting oscillatory behaviour and high wall temperature gradient front. Red is cool and blue is hot, black is high temperature past the clearing point of the TLC. Covering a time span of 0.3 s, at 70 z/D to 78 z/D and $q'' = 83$ kW/m$^2$, $G = 770$ kg/(m$^2$·s), and flow is from left to right.

5.22 Measured circumferentially averaged wall temperature and calculated fluid bulk temperature along tube for varying heat fluxes, $G = 770$ kg/(m$^2$·s).

5.23 Local heat transfer coefficient as a function of quality, effect of heat flux and mass flux.

5.24 Local heat transfer coefficient, comparison with correlations at various heat fluxes, $G = 770$ kg/(m$^2$·s).

5.25 Local heat transfer coefficient, comparison with correlations at various heat fluxes, $G = 1040$ kg/(m$^2$·s).

6.1 Cross-link concept analysis results.

6.2 Cross-linked heat sink.

6.3 Heat sink support I.

6.4 Heat sink support II, millimeters [inches].

6.5 Channel inspection (a) Tapered edges, (b) Trapezoidal cross-section at entry/exit, (c) Cross-section in core.

6.6 Preprocessed heat sink TLC thermal maps, consist of 3 images joined and binary filtered.
6.7 Ratio of experimental to theoretical friction factor versus Reynolds number, before and after rapid heating at high flowrate, STR01.  153
6.8 Friction factor versus Reynolds number, STR02.  154
6.9 Ratio of experimental to theoretical friction factor versus Reynolds number, comparison with other researchers flowing water in parallel silicon microchannels.  156
6.10 Pressure drop versus flowrate for standard versus cross-linked microchannel heat sinks flowing water and air.  157
6.11 Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 142, INT02 flowing FC-72.  159
6.12 Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 210, INT02 flowing FC-72.  160
6.13 Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 307, INT02 flowing FC-72.  161
6.14 Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 559, INT02 flowing FC-72.  162
6.15 Local Nusselt numbers versus non-dimensional streamwise distance at various Reynolds numbers, INT02 flowing FC-72.  163
6.16 Inlet pressure versus applied power, $T_{in} = 70 \degree C$, $Q_{vol} = 30 \text{ ml/min.}$  167
6.17 Limiting Boiling number of STR02 heat sink for different inlet temperatures and compared to correlations.  169
6.18 Sample transient behaviour of various signals, $q'' = 7.9 \text{ W/cm}^2$, $T_{in} = 70 \degree C$, $Q_{vol} = 30 \text{ ml/min.}$  174
6.19 Inlet temperature oscillation frequency versus power, $T_{\text{in}} = 70 \, ^{\circ}\text{C}$ for
different heat sinks at $Q_{\text{vol}} = 30 \, \text{ml/min}$. 

6.20 Inlet temperature oscillation frequency versus power for STR02 at dif-
ferent flowrates and inlet temperature. 

6.21 Oscillation amplitude versus power of various signals, $T_{\text{in}} = 70 \, ^{\circ}\text{C}$ for
different heat sinks at $Q_{\text{vol}} = 30 \, \text{ml/min}$. 

6.22 Inlet pressure oscillation amplitude versus power for STR02 at different
flowrates and inlet temperature. 

6.23 Outlet temperature oscillation amplitude versus power for STR02 at
different flowrates and inlet temperature. 

7.1 Radial concept analysis results. 

7.2 Radial heat sink. 

7.3 Modeling domain depiction. 

7.4 Minimizing peak temperature: Linear profile. Solid lines are uniform
channels and dashed lines are varying cross-sectional area channels. 

7.5 Minimizing peak temperature: Influence of fin width for thermally
developing, 4-wall heated channels. Solid lines are uniform channels
and dashed lines are varying cross-sectional area channels. 

7.6 Benchmark grid comparison with experimental data. 

7.7 Maximum surface temperature versus aspect ratio. 

7.8 Evaluation of optimal aspect ratio. 

7.9 Fluid bulk temperature. 

7.10 3-wall heated area average temperature.
7.11 3-wall heated area averaged Nusselt numbers for various heat flux and
fixed channel height. .................................................. 217
7.12 3-wall heated area average Nusselt numbers for various configurations. 218
7.13 Non-dimensional temperature contours, uniform channel ($\alpha = 0.5$). . 220
7.14 Non-dimensional temperature contours, converging channel with inlet
area of $240\mu m \times 480\mu m$ ($\alpha_{in} = 0.5$). ......................... 221
7.15 Influence of heat flux on center-plane velocity (m/s), uniform channel
($\alpha = 0.5$). .......................................................... 222
7.16 Center-plane velocity (m/s) for various geometrical configurations. . 223
7.17 Parallel uniform channel: One-dimensional thermal resistance model
comparison with three-dimensional conjugate analysis. .................. 225
7.18 Radial channel: One-dimensional thermal resistance model comparison
with three-dimensional conjugate analysis. .......................... 227
7.19 Resulting inner wall heat flux distribution from three-dimensional con-
jugate analysis compared to uniform one-dimensional heat flux distri-
bution. ................................................................. 229
C.1 Comparison of developed code results with Bau (1998). .................. 318
C.2 Minimizing peak temperature: Linear profile. .......................... 320
C.3 Minimizing temperature gradient: Linear profile. ....................... 321
C.4 Minimizing peak temperature: 4-wall heated, thermally developing. . 323
C.5 Minimizing temperature gradient: 4-wall heated and thermally devel-
oping. ....................................................................... 324
List of Tables

3.1 FC-72 properties at 25 °C and 1 atm (3M, 2000). ........................................ 28
3.2 Sensor uncertainty. ......................................................................................... 37
4.1 Typical parameter uncertainty for results. .................................................... 75
5.1 Typical uncertainties for diameter dependent parameters. ....................... 89
5.2 Tube boiling measurement conditions. ......................................................... 107
6.1 Comparison of design and measured parameters. ...................................... 147
6.2 Uncertainty estimates for derived parameters. ............................................ 166
6.3 Critical heat flux correlations. ...................................................................... 170
6.4 Measurement conditions and stability limits. ............................................. 172
7.1 Benchmark grid study. .................................................................................. 206
7.2 3-D CFD simulation test matrix. Below dimensions in millimeters.

\[ W_{out} = 0.15 \text{ mm} \& H_{out} = 0.30 \text{ mm throughout. } q''_1 = 25 \text{ W/cm}^2, q''_2 = 43.75 \text{ W/cm}^2 \text{ and } q''_3 = 62.5 \text{ W/cm}^2, \Delta P_{fixed} = 6.9 \text{ kPa and } \dot{m}_{fixed} = 2.85 \times 10^{-5} \text{ kg/s.} \] .................................................. 209

A.1 1 mm, Water. ............................................................................................. 256
A.2 0.50 mm, Water. ......................................................................................... 257
A.3  1 mm, FC-72. ........................................... 258
A.4  0.5 mm, FC-72. ........................................... 258
A.5  0.25 mm, FC-72. ........................................... 259
A.6  Local heat transfer coefficient, 770 kg/(m²·s). ......................... 260
A.7  Local heat transfer coefficient, 1040 kg/(m²·s). ......................... 262
A.8  Single phase heat transfer with FC-72, INT02, Re = 142. ................. 265
A.9  Single phase heat transfer with FC-72, INT02, Re = 210. ................. 266
A.10 Single phase heat transfer with FC-72, INT02, Re = 307. ................. 267
A.11 Single phase heat transfer with FC-72, INT02, Re = 559. ................. 268
A.12 Flow boiling oscillations, STR02, $T_i = 70^\circ C$. .................. 269
A.13 Flow boiling oscillations, STR02, $T_i = 80^\circ C$. .................. 270
A.14 Flow boiling oscillation, INT01. ........................................... 271
### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area, ( m^2 )</td>
<td></td>
</tr>
<tr>
<td>Bo</td>
<td>boiling number, ( Bo = \frac{q'}{\dot{m}h_{fg}} )</td>
<td></td>
</tr>
<tr>
<td>Cp</td>
<td>fluid specific heat capacity, ( \text{kJ/(kg} \cdot \text{°C)} )</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>diameter, ( m )</td>
<td></td>
</tr>
<tr>
<td>( D_h )</td>
<td>hydraulic diameter ( \frac{4A_{ax}}{2(W_{chn} + H_{chn})} ), ( m )</td>
<td></td>
</tr>
<tr>
<td>( d_{heated} )</td>
<td>heated area hydraulic diameter ( \frac{4A_{ax}}{2H_{chn} + W_{chn}} ), ( m )</td>
<td></td>
</tr>
<tr>
<td>f</td>
<td>friction factor</td>
<td></td>
</tr>
<tr>
<td>( f_w )</td>
<td>frequency, ( \text{Hz} )</td>
<td></td>
</tr>
<tr>
<td>( f^* )</td>
<td>non-dimensional frequency, ( f^* = \frac{f_w u_{chn}}{u_{chn, in}} )</td>
<td></td>
</tr>
<tr>
<td>g</td>
<td>gravity, ( \text{m/s}^2 )</td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>mass flux, ( \text{kg/(m}^2 \cdot \text{s)} )</td>
<td></td>
</tr>
<tr>
<td>H</td>
<td>height, ( m )</td>
<td></td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient, ( W/(m^2 \cdot \text{°C)} )</td>
<td></td>
</tr>
<tr>
<td>( h_{fg} )</td>
<td>latent heat of vaporization, ( \text{kJ/(kg} \cdot \text{K)} )</td>
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</table>
\( I \) current, A

\( k \) thermal conductivity, \( \text{W/(m\cdot{}^\circ\text{C})} \)

\( K_{\text{loss}} \) entrance and exit loss factors

\( L \) length, m

\( \dot{m} \) mass flow rate, kg/s

\( N_{\text{sub}} \) subcooling number, \( (N_{\text{sub}} = \frac{C_p(T_{\text{out}}-T_{\text{in}})}{h_{fg}} \frac{\rho_f-\rho_g}{\rho_g}) \)

\( \text{Nu} \) Nusselt number

\( \bar{P} \) power generated, W

\( P \) pressure, Pa

\( P_r \) reduced pressure

\( \text{Pr} \) Prandtl number

\( \dot{q} \) heat generated per unit volume, \( \text{W/(m}^3\)\)

\( q'' \) heat flux, \( \text{W/m}^2 \)

\( Q_{\text{vol}} \) volumetric flowrate, \( \text{m}^3/\text{s} \), unless otherwise specified

\( r \) radial coordinate

\( \text{Re} \) Reynolds number

\( r^+ \) normalized radial coordinate, \( (r^+ = \frac{r_{\text{out}}-r}{r_{\text{out}}-r_{\text{in}}}) \)

\( T \) temperature, \( ^\circ\text{C} \), unless otherwise specified

\( u \) velocity, m/s

\( V \) voltage, V

\( \bar{V} \) 3-component velocity vector, m/s

\( W \) width, m
$We_D$  Weber number, $\left( We_D = \frac{G^2 D_h}{\sigma \rho_f} \right)$

$We_{Lchm}$  Weber number, $\left( We_{Lchm} = \frac{G^2 L_{chm}}{\sigma \rho_f} \right)$

$x$  quality

$z^+$  normalized streamwise coordinate, $\left( z^+ = \frac{z}{L_{chm}} \right)$

$z^*$  non-dimensional streamwise coordinate, $\left( z^* = z/RePrD_h \right)$

**Abbreviations**

BNC  bayonet Neill-Concelman

CCD  charged-coupled device

CFD  computational fluid dynamics

CHF  critical heat flux

CTF  continuous two-phase flow

DNA  deoxyribonucleic acid

DRIE  deep reactive ion etching

EBO  explosive boiling oscillations

FD  fully developed

IR  infrared

LALPO  large amplitude long period oscillations

LCD  liquid crystal display

LTAFO  liquid to two-phase alternating flow

LTVAF  liquid to two-phase to vapor alternating flow
MEMS micro-electro-mechanical systems
NTSC National Television System(s) Committee
PCR polymerase chain reaction
PIV particle image velocimetry
RGB red-green-blue
ROI region of interest
SASPO small amplitude small period oscillations
SEE standard error estimate
SIMPLE semi-implicit method for pressure-linked equations
TD thermally developing
TIFF tagged image file format
TLC thermochromic liquid crystal
UV ultraviolet

Greek Letters

\( \alpha \) channel aspect ratio (width/height)
\( \mu \) viscosity, N·s/m²
\( \rho \) density, kg/m³
\( \sigma \) surface tension, N/m
\( \theta \) Non-dimensional temperature, \( \left( \theta = \frac{T - T_{\text{ref}}}{T_{\text{ref}} / \chi_{\text{m}} / \rho_{\text{m}} / \mu_{\text{m}}} \right) \)
Subscripts

amp amplitude
avg averaged parameter
b fluid bulk property
chn channel parameter
crit critical heat flux
f liquid property
fluid fluid condition
heated heated surface parameter
i inner wall parameter
in channel inlet condition
∞ free stream condition
o outer wall parameter
out channel outlet condition
sat saturation property
sub subcooled conditions with respect to entry
tp two-phase conditions
tube tube parameter
v vapour property
w condition at the wall
\begin{align*}
y & \text{ spanwise coordinate} \\
z & \text{ streamwise coordinate} \\
0 & \text{ zero quality location}
\end{align*}
Chapter 1

Introduction

1.1 Motivation

Advances in Micro-Electro-Mechanical System (MEMS) based manufacturing has led to the creation and development of a number of miniaturized technologies, from DNA analysis (Li et al., 2006) to power generation (Epstein, 2003). For the electronics industry this has enabled the fabrication of electronic circuits with an ever decreasing size. As a result, there has been an increase in the heat produced per unit area of microelectronic chips, which in turn has generated the requirement for effective miniature cooling technologies. One such device is the pumped cooling loop using microchannel heat sinks and pioneered in the early 1980's by Tuckerman and Pease (1981). The microchannel heat sink is characterized by a set of microchannels conventionally machined or micromachined into a conducting block. It has been proposed as a viable thermal management solution for high heat flux electronics that include land based (Jiang et al., 2001), avionic (Mudawar, 2001) and extraterrestrial applications
Fundamental to the microchannel heat sink and a number of MEMS based systems under development, is the microchannel and its corresponding thermo-fluidic characteristics. The study of convective heat transfer in small diameter channels or microchannels, typically considered less than 0.5 mm for single phase flows and 3 mm for boiling flows, is still in its infancy and much fundamental work remains in this area.

1.2 Objectives and Organization

The objectives of this work are:

- To develop a test facility for local thermal measurements in microchannel systems.

- To investigate the local heat transfer characteristics in microchannel systems.

- To address flow boiling instabilities in two-phase microchannel systems.

- To develop novel microchannel heat sinks aimed at micro/nano- spacecraft thermal management applications.

- To numerically investigate heat transfer in microchannel systems with axially varying cross-sections.

Chapter 2 presents a review of the heat transfer in single and multi-channel microsystems with single phase and boiling flows. Additionally, the literature review
discusses the problem of flow boiling instabilities and oscillations, which is one of the constraints in applying two-phase microchannel heat sinks to demanding applications such as spacecraft systems. Finally, the literature review provides a summary of the current state of thermal measurement techniques for microscale applications. Chapter 3 provides a description of the test facility, which has been developed for convective heat transfer investigations in microgeometries. This includes design and procurement of the mechanical rig, the instrumentation and automated data acquisition system, and the optical measurement equipment. The methodology developed for utilizing un-encapsulated thermochromic liquid crystals (TLC's) for measurement in microgeometries is presented in Chapter 4. This includes validation through single phase heat transfer measurements in a 1 mm tube. An investigation of the transport phenomena in microtubes under single phase and flow boiling conditions is presented in Chapter 5. The work to this point provides a foundation on the methods that will be utilized in studying the heat transfer characteristics in various applications of microchannel systems. The development of a cross-linked silicon microchannel heat sink is presented in Chapter 6. Experimental measurements are presented under single phase and flow boiling conditions. Finally, Chapter 7 explores a radially flowing, varying cross-sectional area microchannel heat sink. Numerical investigations using a one-dimensional thermal resistance model and a three dimensional conjugate heat transfer model, are carried out to evaluate its performance. In the closing (Chapter 8), a summary of the contributions is presented and recommendations for future studies are provided.
Chapter 2

Literature Review

2.1 Single Phase Flow and Heat Transfer

2.1.1 Single Channel Studies

Some recent reviews on single phase flow heat transfer in mini and microchannels may be found in Hetsroni et al. (2005b), Morini (2004), Sobhan and Garimella (2000) and Bailey et al. (1995). Some of the prevalent characteristics and remarks related to single phase continuum flows include discrepancies reported between microchannel results and the accepted correlations and trends for their macro counterparts. For example, the critical Reynolds number for transition to turbulent flow in microchannels has been reported to be as low as 200 (Sobhan and Garimella, 2000), compared to the conventionally accepted range of 2100 to 2300 confirmed by other researchers in microchannels (Sharp and Adrian, 2004). In addition, significant variations between the results of different investigators have been reported for the heat transfer coefficients
and friction factor with no clear relationship for the differences in some instances (Mehendale et al., 2000). Hetroni et al. (2005b) reviewed a number of single phase flow studies, with the aim of revealing the potential sources of disparity. In general they noted that discrepancies may be due to i) differences between the experimental conditions and assumptions in the model not considered, ii) error in the measurements, and iii) increasing influence of surface forces and conditions usually neglected for conventional size systems. The authors' proposed a range of applicability of conventional size channel theory for microchannels ranging in hydraulic diameter from 15 μm to 4010 μm.

The channel size at which deviations will occur from conventionally sized channels remains undetermined and in present work microchannels are considered channels with nominal diameter of 1 mm and less. Overall, a review of the literature indicates the need for additional and improved studies of heat transfer in mini and microchannels due to a lack of consensus in a number of areas on the flow and heat transfer characteristics (Morini, 2004; Sobhan and Garimella, 2000).

2.1.2 Multi-Channel Microsystems

Recently, there have been a number of investigations considering radical modifications to the standard straight microchannel heat sink for pumped cooling loops. Amongst these, single phase flow studies have considered pin fins (Kosar and Peles, 2006a; Peles et al., 2005), fractal networks (Chen and Cheng, 2005; Alharbi et al., 2004; Pence, 2002), constructal networks with loops (Wang et al., 2006) and cross-linked
channels (Xu et al., 2005; Cho et al., 2003; Jiang et al., 2002). Radially flowing configurations, as opposed to the standard parallel channel configurations, have been proposed through fractal network and constructal network schemes (Wang et al., 2006; Alharbi et al., 2004; Pence, 2002). The aims of the fractal and constructal schemes have been for improved surface temperature uniformity and decreased pressure drop compared to equivalently sized standard parallel channel schemes. Xu et al. (2005) utilized a concept of cross-links along 10 triangular channels of 155 µm hydraulic diameter. The objective here, was to increase the heat transfer by introduction of a constantly re-developing thermal boundary layer. They compared this configuration with a standard straight channel configuration. Temperature profiles for the straight channel had a constant rise, whereas for the cross-linked section would rise, then drop giving a lower mean temperature. Additionally, the cross-linked design showed a lower pressure drop than the straight channel. Cho et al. (2003) also studied a cross-linked configuration, however with rectangular channels. Heat transfer studies considered localized heating to better simulate real chip conditions. Under single phase flow, the cross-linked configuration showed better performance, but contrary to the previous study, showed an increase in pressure drop. Colgan et al. (2005) studied a configuration with fins rather than channels combined with multiple entries and exits along the flow path. The fins were arranged in an in-line or staggered configuration and had sharp or blunt ends. The sharp versus blunt end configurations did not show any significant benefit, however the staggered fins had better thermal performance, though a higher pressure drop.
Bau (1998) proposed the use of channels with slowly varying cross sections for improving the thermal performance of micro heat exchangers. The author showed that through optimizing the wall contour with streamwise distance, the maximum wall temperature or the maximum wall gradient could be reduced in comparison to straight parallel channels. There have been a number of investigations considering flow without heat transfer in channels of slowly varying cross-sections, with one of the first studies dating back over 35 years (Mantion, 1971). Studies considering flow with heat transfer in slowly varying ducts, however, are limited, partially due to their limited applicability to conventional heat exchangers. Brod (2001) considered flow in a circular tube and a two-dimensional plane channel of slowly varying cross-section, which was directed at the process industries using melts. Assuming a similarity solution whereby the velocity at each cross-section was fully developed, the author obtained expressions for the fluid bulk temperature and heat transfer distribution. The study showed the non-dimensional fluid bulk temperature and the local Nusselt number to be invariant on the tube's contour function. MEMS based fabrication can readily produce rectangular channels with variable widths. In the work of Li et al. (2006), a polymerase chain reaction (PCR) microchip with serpentine microchannels of varying width was designed and fabricated for control of the velocity at different areas of the chip. Control of the velocity allowed for optimization of the sample exposure times at the different temperature phases, while minimizing the transition time during temperature phase transition.
2.2 Boiling Heat Transfer

Recent reviews on boiling heat transfer in mini and microchannels have been presented by Thome (2004) and Kandlikar (2002). Kandlikar (2002) suggested a channel size criterion whereby 3 mm and greater are conventional (macro) channels, 0.2 mm to 3 mm are mini channels and 0.01 mm to 0.2 mm are microchannels. With the exception of the Confinement number \(N_{\text{conf}}\), however, no physics based classification of channel sizes has been presented (Thome, 2004). The Confinement number was introduced by Kew and Cornwell (1997) and was suggested as a parameter to differentiate between microscale and macroscale boiling phenomena. It is given by,

\[
N_{\text{conf}} = \left[ \frac{\sigma}{g (\rho_f - \rho_v)} \right]^{1/2}.
\]  

(2.1)

It was found that for a Confinement number greater than 0.5, the macroscale correlations did not predict microchannel results well (Kew and Cornwell, 1997).

It has been generally accepted that both nucleate boiling and convective boiling mechanisms exist in mini and microchannels, although the dominant mechanism remains inconclusive. Some investigators have shown that there is a decreasing trend in the heat transfer coefficient with mass flux, and so have concluded that forced convection is the dominant mechanism (Lee and Lee, 2001; Qu and Mudawar, 2003). While for macro size channels, the dominant mechanism is convective boiling, the majority of the mini and microchannel studies suggest that nucleate boiling is the dominant mechanism (Thome, 2004). This has been determined by observing the
stronger dependence of the heat transfer coefficient with heat flux, and not with mass flux.

2.2.1 Small Diameter Tubes

There have been a number of recent investigations of flow boiling heat transfer in small diameter tubes. Lin et al. (2001) investigated the two-phase heat transfer characteristics of R141B in circular tubes of diameter 1.1 mm, 1.8 mm, 2.8 mm, and 3.6 mm. They deduced that there were three boiling regions. A nucleate boiling region whereby the heat transfer coefficient was dominated by the heat flux, a convective boiling region whereby the heat transfer coefficient was independent of heat flux but increased with quality, and a partial dryout region. In the partial dryout region, the heat transfer coefficient was reduced dramatically, and dryout was observed to occur earlier with increasing Confinement number. Additionally, the heat transfer coefficient was found to have a more marked decrease with quality, with decreasing tube diameter. The mean heat transfer coefficient was independent of mass flux and diameter and dependent only on heat flux in the region before dryout. Recently Saitoh et al. (2005) investigated flow boiling of R-134a in circular tubes of 0.51 mm, 1.12 mm and 3.1 mm diameter. The two-phase heat transfer coefficient was observed to be governed by convective and nucleation effects, though the convective effects decreased with decreasing diameter. Dryout was observed to occur at a lower quality with decreasing diameter. Through tracking the wall temperature, flow oscillations were observed in the 3.1 mm and 1.12 mm tube at low quality entry, but not in the
0.51 mm tube. The authors concluded that these were density wave type oscillations due to their period. Grohmann (2005) studied the two-phase and single-phase heat transfer characteristics of Argon in circular tubes, with nominal diameters of 0.50 mm and 0.25 mm. In the nucleate boiling regime, there was no influence of mass flux or quality on the heat transfer coefficient. Flow condition oscillations were sometimes observed with subcooled entry conditions, however these could be controlled with the entry quality greater than zero.

2.2.2 Flow Boiling of Dielectric Fluids in Mini and Microchannels

The fluid type is an important parameter in the heat transfer performance. Many studies on flow boiling heat transfer in mini and microchannels have used water or refrigerants, few studies however have utilized dielectric fluids. These fluids will typically have lower heat transfer performance than water, however are advantageous in situations whereby the fluid may be in contact with an electrically charged component. Warrier et al. (2002) investigated two-phase heat transfer in an array of five 0.75 mm hydraulic diameter rectangular channels using FC-84 as the working fluid. They compared their heat transfer results with correlations valid for channels ranging in size from 2.4 mm to 32 mm and found none to be in good agreement. The heat transfer coefficient was observed to have a decreasing trend with quality. They developed a correlation relating the saturated flow boiling heat transfer coefficient with quality and Boiling number. Yen et al. (2003) studied boiling heat transfer in microtubes
of 0.19 mm, 0.3 mm and 0.51 mm inner diameter, using FC-72 and HCFC123. The study was carried out at low mass flux ranging from 145 kg/(m²·s) to 295 kg/(m²·s). Local heat transfer coefficient results were only presented for HCFC123. The results showed a decreasing heat transfer coefficient with quality, but independent of mass flux, suggesting nucleate boiling as the dominant mechanism. Recently, Chen and Garimella (2005) investigated the effects of dissolved air on the subcooled flow boiling characteristics of FC-77 in an array of 10 parallel channels, 0.50 mm wide and 2.5 mm deep. The study showed boiling incipience to occur at lower wall temperatures for the un-degassed fluid compared to the degassed fluid. Flow instability fluctuations were larger for the un-degassed fluid and larger pressure drops were measured. Although the earlier boiling inception caused higher initial heat transfer coefficients for the un-degassed fluid, the authors recommended working with a degassed fluid for better predictability and control.

2.2.3 Flow Boiling Incipience

Important to the safe and reliable design of two-phase heat exchangers with subcooled entry is the prediction of the wall superheat required to initiate boiling, or inversely, for a given subcooling, prediction of the incipient heat flux. Classical approaches for boiling incipience consider a bubble to grow if the minimum temperature surrounding the bubble is equal to or greater than the saturation temperature given by the pressure inside the bubble (Collier and Thome, 1999). The superheat in this case is related explicitly to the heat flux and the pressure. This method tends to under
predict the incipient superheat, particularly for well wetting fluids such as refrigerants and perfluorocarbons, as it does not consider the fluid-solid contact angle, and it assumes all nucleation sites to be active. Recently, Basu et al. (2002) demonstrated a correlation based on adaptations of boiling incipience classical methods, which includes effects of the static contact angle of the fluid-solid system. The correlation showed good agreement with a variety of fluid-solid systems.

Qu and Mudawar (2002) investigated the incipient heat flux required for boiling in an array of rectangular microchannels. It was shown that this heat flux is dependent on the mass velocity. Additionally, boiling incipience was observed to be different from large channels, such that bubble departure occurred, rather than growth and collapse at incipience. A mechanistic model based on bubble departure criterion was developed to predict the boiling incipience from the flat surface or a corner and showed good agreement with the experimental results. Regarding single tubes, Kennedy et al. (2000) studied boiling incipience in 1.448 mm and 1.168 mm circular copper tubes with water as the working fluid. Their results for the larger tube were in good agreement with macro tube correlations, whereas the smaller tube was underpredicted. An estimation of the Confinement number for their case, taken at 50 °C (from experimental conditions) is calculated to be 0.46 for the larger tube and 0.57 for the smaller tube. Hapke et al. (2000) investigated boiling incipience in a 1.5 mm nickel alloy tube with water as the working fluid and a mass flux range from 100 kg/(m²·s) to 500 kg/(m²·s). Their results showed wall superheat dependency at incipience on mass flux.
Ghiaasiaan and Chedester (2002) developed a model for predicting boiling incipience in microchannels based on the assumption that the thermocapillary force suppressing the bubble plays a crucial role. Their model showed the incipience heat flux to be dependent on channel diameter (results for 0.75 mm to 1.45 mm diameter), however this was not experimentally verified and the results were for turbulent flows. Yen et al. (2003) experimentally investigated boiling heat transfer in tubes 0.19 mm to 0.51 mm with FC-72 and HCFC123 and at low mass fluxes of 50 kg/(m²·s) to 300 kg/(m²·s). The maximum wall superheat was found to be dependent on the tube diameter and nucleation site distribution.

2.2.4 Boiling Instabilities

Boiling instabilities in macro-systems have been extensively studied in the past and may be classified as static or dynamic (Bergles, 1981; Kakac and Veziroglu, 1983). Static instabilities are those whose cause may be related through steady state laws. During such instability the fluid may change to another steady state point or oscillate between two steady state points. An example is the Ledinegg instability which, by definition, occurs when the pressure gradient with respect to mass flow of the internal flow system is less than that of the pressure supply system. Inertia or other feedback effects cause dynamic instabilities and the steady state laws are not enough to predict them. These instabilities may be initiated by a static instability. Density waves have a low frequency that is on the order of a particle traveling through the system. Their mechanism is related to delay and feedback between the flow, pressure drop and

13
density. Other types of dynamic instabilities may include thermal oscillations, which are observed during dryout, and parallel channel instabilities, which are related to interactions between a small number of channels. In this case, various modes of flow redistribution may be observed. A third type of dynamic instability, which occurs as a secondary phenomenon, is the pressure drop oscillation. This type of instability is found to occur when there is a compressible volume upstream of the heated section. Similar to the Ledinegg instability, it occurs just after the minimum in the pressure drop versus mass flow curve. The frequency of oscillation is much lower for the pressure drop type instabilities compared to density wave instabilities and is on the order of 0.1 Hz.

A comparison of flow stability regions for single, parallel, and cross-connected parallel channels has been summarized by Kakac and Veziroglu (1983). The data were compiled from a number of experiments from a single test facility investigating boiling instabilities in vertical pipe systems using Freon-11. The pipe inner diameters were 7.5 mm (0.295 inches). The study found that for density-wave type instabilities, single channel, parallel channel and two parallel cross-connected channels all presented similar performance. For pressure-drop type instabilities the most stable system was provided by the 4 parallel cross-connected channel system. The parallel channel systems with unequal heat inputs showed the smallest stability regions with respect to pressure-drop type instabilities.

Related to microchannel flows, few studies have documented the characteristics of the flow boiling instabilities. Some studies have observed different flow pattern characteristics in the boiling instabilities depending on the imposed conditions, and
have classified them as such. Wu and Cheng (2004) investigated boiling instabilities in a parallel channel system with eight trapezoidal channels. The channels were fabricated in a silicon wafer 525 \( \mu \text{m} \) thick and had a hydraulic diameter of 186 \( \mu \text{m} \). They used a microscope and a high-speed camera for bubble visualization and placed thermocouples at the base of the wafer. The authors observed three instability modes consisting of liquid to two-phase alternating flow (LTAF), continuous two-phase flow (CTF), and liquid to two-phase to vapor alternating flow (LTVAF). For the LTAF case, bubbly flow dominated with wall temperature fluctuations of 20 °C to 46 °C. The oscillation frequency was 0.065 Hz and was characterized as large-amplitude/long period. Pressure and mass flux oscillations were almost out of phase. With an increase in heat flux and decrease in mass flux, the CTF case was observed. Temperature fluctuations decreased to 7 °C to 20 °C and the pressure and mass flux were almost in phase. Finally with additional increase in heat flux and decrease in mass flux, the LTVAF case was observed. Wall temperature fluctuations were highest in this case at 66 °C to 230 °C and the pressure and mass flux were almost out of phase. These oscillations were also classified as having a large-amplitude over a long period.

Xu et al. (2005) carried out a study of flow boiling instabilities in a copper test section of 26 channels, 300 \( \mu \text{m} \) wide and 800 \( \mu \text{m} \) deep. By gradually decreasing the mass flux, they were able to initiate flow instability. Two periodic instabilities were observed, large amplitude long period instabilities oscillations (LALPO) and small amplitude small period oscillations (SASPO). These were accompanied by thermal oscillations with fluctuations of up to 20 °C (peak to peak) in the first case and up to 4 °C (peak to peak) in the second. The SASPO instability was characterized by bubble
growth at channel exit, followed by the liquid-vapor interface moving upstream. When the interface reached the inlet plenum for a number of channels, the vapor fronts moved forward into the channels. The growing vapor front increased the temperature while decreasing the heat transfer coefficient. The working fluids were water and methanol. Similar behaviors were observed for both fluids, however, due to methanol's lower latent heat the frequency was lower.

Hetsroni et al. (2005a) investigated water boiling in 21 isosceles triangle channels with a base of 250 \( \mu \text{m} \) and an angle of 55\(^\circ\), fabricated in a silicon substrate. At boiling onset, bubble growth rapidly filled the entire channel cross-section, then grew lengthwise both upstream and downstream. The other channels accounted for the liquid that was pushed upstream. A time instant later, the bubble was vented from the channel and a brief period of vapor with liquid droplets commenced. This boiling process was described as “explosive” due to its short time period. The pressure drop amplitude was found to increase with quality and the oscillation frequency in a single channel increased with increasing heat flux. In a recent study by the same authors (Hetsroni et al., 2006), further characterization of the boiling instabilities, termed explosive boiling oscillations (EBO) was provided. They found an increase in the amplitude of bulk system parameters such as the pressure drop and the inlet and outlet fluid temperature with increasing heat flux. Additionally, they found a decrease in the frequency of the bulk system properties with heat flux. This is notably different from the characteristic of individual channels within the same system. Steinke and Kandlikar (2004a), while studying flow boiling heat transfer in six parallel channels
(214 \mu m \times 200 \mu m) in a copper substrate test section, observed flow reversals in individual channels. The fluid was water and the channels were slightly trapezoidal. The flow reversals were found to occur during the onset of the annular-slug flow regime. Flow reversals were similarly observed by Balasubramanian and Kandlikar (2005) who investigated flow boiling in six parallel copper channels, 990 \mu m wide and 207 \mu m deep. The authors noted that while bubble nucleation was the dominant flow pattern (at low surface temperatures), the pressure drop dominant frequency increased with increasing surface temperature. However, with further increase in temperature, the slug flow pattern dominated and the pressure drop dominant frequency decreased. Qu and Mudawar (2004) studied flow boiling in 21, 215 \mu m \times 821 \mu m (width \times depth) copper channels, with water as the working fluid. Two oscillation type instabilities were observed. In the first case, the liquid-vapor interface changed position drastically from close to the exit, to close to the inlet. This was accompanied with high pressure drop and temperature fluctuations. The temperature fluctuations reached 15 °C and caused premature critical heat flux (CHF). Installing a throttle upstream of the test section remedied this and it was classified as a pressure drop oscillation. The second case was a result of installing the throttle valve and was characterized by the liquid-vapor front alternating positions between channels. This was characterized as parallel channel instability and was accompanied by mild pressure drop and temperature (<1 °C) fluctuations.
2.2.5 Novel Multi-Channel Concepts

Two-phase flow novel configurations have aimed at increasing heat transfer and minimizing instabilities. Kosar and Peles (2006b), introduced restrictors at the channel inlets to suppress flow boiling oscillations. The restrictors had a diameter six times smaller than the channels. They studied the influence of the restrictor length, and observed that with increased length, the instabilities were less severe. Pressure drop fluctuations were on the order of 2 to 7 kPa with no restrictor, whereas they were less than 1 kPa with the restrictors. For heat transfer enhancement under two-phase conditions, Kosar and Peles (2005) utilized re-entrant cavities. When used in macrosized systems, such structures are known to act as artificial nucleation sites, which can enhance flow boiling. However due to the large discrepancies in two-phase heat transfer results in microchannels, the authors were unable to ascertain the benefit of this configuration. Kandlikar et al. (2006) utilized a combination of inlet flow restriction elements and fabricated nucleation sites, to stabilize the flow boiling process. The study considered restrictors with 51 percent and 4 percent of the flow area, with and without artificial nucleation sites. With the 4 percent flow area restrictor, no flow reversals were observed and the addition of the fabricated nucleation sites produced the best thermal performance. However, the penalty of this was an increase in pressure drop of almost double.
2.3 Heat Transfer Measurement in Microsystems

Most work regarding experimental heat transfer measurements in microchannels use thermocouples (point sensors) for wall temperature measurements, due to their accessibility and relatively low cost. Current fine gage wire thermocouples on the market have diameters as small as 0.013 mm for K and E-Type and 0.025 mm for J and T-Type (OMEGA, 2006). Thermocouple diameters for specialized hypodermic probes are as small as 0.200 mm, while standard small size probes have diameters as low as 1.5875 mm (OMEGA, 2006). One problem with such sensors is their large size compared to the channel dimensions and hence their potential influence on the flow characteristics. This may be alleviated by placing the sensors at a distance from the fluid-wall interface and assuming one-dimensional heat transfer, or by using a fin analysis method (Qu and Mudawar, 2003). These approximations, however, introduce inaccuracies in the internal wall surface temperature measurement. Also, the relative size of these sensors, if discreetly placed at the wall, gives only an area averaged temperature measurement. Another inconvenience of such point sensors is the need for an array of sensors to obtain full surface temperature measurements.

An overview of the various possibilities of micromachined thermal sensors may be found in Kovacs (1998). Many macro size thermal sensing concepts have been fabricated as micro transducers. These include thermo-mechanical sensors, which rely on thermal bimorph of two dissimilar metals, thermo-resistive transducers where temperature is related to change in resistance, and thermocouple type sensors, which rely on the Seebeck and Peltier effect. Self-heating, however, becomes a significant
problem as the size reduces for resistive sensors due to the excitation required. If the thermal time constants are known for the system, pulsed measurements may be incorporated to minimize this self-heating and increase accuracy. A recent example of the fabrication of an integrated resistive sensor was presented by Liu et al. (2004). For thermocouple type sensors it is attractive to manufacture them from semiconductor materials, as the Seebeck coefficient is generally larger ($\sim 10^{-3}$ V/K as opposed to $10^{-8}$ V/K), providing a more sensitive sensor from readily available material. Also common for micromachined thermal sensors are junction-based sensors utilizing diodes or transistors.

Three novel thermometry techniques have been described by Cahill et al. (2002) for temperature measurement at the micro and nano-level. The first is scanning thermal microscopy. In this case, a sharp tip is placed on the surface, and the surface temperature may be derived from the localized heat transfer between the surface to the tip. Most common setups utilize a cantilevered tip on an atomic force microscope. Spatial resolutions in the range of 30 to 50 nm have been obtained. Applicable areas in need of improvement are acquiring fundamental understanding of the tip-sample heat transfer, as well as designing and fabricating integrated multifunctional probes. Additional information on the scanning thermal microscope may also be found in Gu et al. (2002). The second method is pico-second reflectance thermometry, which is similar to reflectance thermometry but extended to pico-second time scales. This method involves generating laser pulses and directing them at the surface of interest. These pulses diffuse heat into the surface and the decay of the reflected light is used to obtain measurements. Nanometer scale resolution is possible with this method.
The third technique is scanning optical thermometry, where through the use of solid immersion lens technology, spatial resolution of optical measurements can be increased past the diffraction limit.

2.3.1 Thermography

Non-intrusive techniques such as infrared (IR) thermography, laser induced fluorescence (LIF) and liquid crystal thermography have traditionally been employed (Goldstein et al., 1998) for macro size experiments. There are a variety of challenges associated with IR thermography, which include accurately characterizing its performance and calibration, determining the measured body’s surface emissivity, identifying the measurement point, and accounting for radiation losses and additional optical equipment (Hetsroni et al., 2003). In addition, for the measurement of micro objects, accurately accounting for the background radiation is necessary. Hetsroni et al. (2003) while measuring the surface temperature of a capillary tube, utilized a technique whereby the background temperature was controlled to within that of the tube wall temperature. The aim of this was to minimize the uncertainty in measurement due to background radiation during IR measurement. Patil and Narayanan (2006) developed a method on the use of IR thermography for the near wall temperature measurement in microchannels. The system incorporated a microscope with a mid-wavelength IR camera (3.5-4.1 μm, 4.5-5.1 μm), and details on the radiation accounting procedure were provided.

Thermochromic liquid crystal (TLC) thermometry is a non-intrusive technique
based on the changing pitch size of the helical molecular structure of the material during temperature change. When illuminated with white light, TLC's will reflect light of wavelength proportional to the temperature they are experiencing. This reflected wavelength is proportional to the instantaneous pitch between molecular layers. Additional information about the characteristics of liquid crystal and their use in thermometry and heat transfer may be found in Ireland and Jones (2000) and Hallcrest (2000). The TLC technique has been used by many researchers since the late 1960's when it was first applied to non-destructive testing and flow visualization (Fergason, 1968).

The successful use of TLC's on a micro-geometry has been demonstrated by Höhmann and Stephan (2002), while investigating evaporation from a liquid meniscus. In their work, they utilized the un-encapsulated form of TLC's applied to the backside of a 20 μm stainless steel foil. They were able to obtain spatial resolutions of 0.83 μm, where the uncertainty in temperature was 0.51 °C or 7.5%. The use and limitations of TLC's in boiling heat transfer has been investigated and has been discussed by Kenning et al. (2001) and Klausner et al. (1995). These studies determined that in many instances, due to the thermal lag between the liquid crystal and the inner wall surface, the response time of the liquid crystal is not sufficient to capture all the unsteady phenomena occurring during nucleation boiling. A recent example on the use of the un-encapsulated TLC form for its high spatial and temporal resolution, during pool boiling experiments may be found in Wagner et al. (2006).

A first glimpse at the use of TLC's in mini-channels with boiling has been recently presented by Chin (1997) and Chin et al. (2002) during an investigation of boiling
incipience and convective heat transfer in narrow vertical channels. These authors used the TLC’s in their common encapsulated (as opposed to un-encapsulated) form, which due to capsule diameters of 10 to 25 μm, restricts the spatial resolution capability of the TLC material. The use of un-encapsulated TLC’s for quantitative, high resolution heat transfer measurements in small channels has, to a limited extent, been investigated (Aligoodarz et al., 1998). This is likely due to the difficulty in handling un-encapsulated TLC’s, since they are easily contaminated by solvents or dust, and prolonged exposure to ultraviolet (UV) light can disturb their calibration and even destroy their response (Hallcrest, 2000).

In the work of Aligoodarz et al. (1998), rectangular channel geometries 2×1 mm and 2×2 mm, with three walls made of 75 μm thick stainless steel, were studied for boiling flows. The fourth wall was made of glass, which was used for visualization while the opposing stainless steel wall was coated with un-encapsulated TLC. The authors document difficulties in obtaining a continuous coating of the TLC on the surface, as well as difficulties in accounting for its stringent requirements for a contaminant free area. Their choice of the un-encapsulated form, however, was motivated by the ability to apply it in thinner layers than the encapsulated form, hence obtaining an improved response time, which was approximately 4 ms. Another benefit of the un-encapsulated form is that the spatial resolution is now restricted to the optical configuration (Azar et al., 1991), thus sub-micron resolution is possible. Overall, TLC’s offer the potential for improved temperature measurements in micro-channels due to their non-intrusive character and possibility for high spatial resolution measurements.
2.3.2 Summary

Heat transfer in mini and microchannels is a recent area of study and although the number of researchers in this area has increased significantly over the last decade, a significant amount of research is still needed. Novel microchannel heat sinks using single phase flows have been proposed and investigated with the aim of increasing heat transfer while minimizing pressure drop. In particular, channels with slowly varying cross-sections have had limited heat transfer investigations.

Considering flow boiling heat transfer in small diameter channels (<2 mm), due to the complexity of the process, the number of parameters involved and the lack of correspondence with existing larger channel correlations, more data are still required considering a variety of conditions and fluids. Further, for realizing practical flow boiling microsystems, the characteristics of boiling incipience, including input parameters of influence, and the characteristics of the wall superheat and heat transfer coefficients need to be investigated. For conventional size channel systems, flow boiling instability characteristics can be significantly influenced by modification of the configuration, from single channels to parallel channels and to cross-connected parallel channels. This influence has to the author’s knowledge, yet to be investigated for microchannel configurations relevant to micro heat sink applications.

Finally, very few studies have been carried out using thermographic techniques for microsystem thermal investigations. These techniques, however, such as liquid crystal thermography, offer the opportunity for non-intrusive, high resolution measurements.
Chapter 3

Test Facility and Test Methods

The initial stage of this work required the design, procurement and construction of the test facility presented in this chapter. The aim in the design was for a state-of-the-art facility to perform single phase and flow boiling heat transfer experiments using liquid crystal thermography in microgeometries. The facility is composed of three main sections, which are, a flow loop, a data acquisition system and an image acquisition system. Over the course of the investigations, a number of additions have been made to the original layout and the final facility configuration is discussed here.

3.1 Flow Loop

A schematic of the main components used in the closed loop test facility are shown in Figure 3.1. The facility has been designed to handle a number of working fluids, which include FC-72, distilled water and air. FC-72 is a perfluorocarbon fluid manufactured by 3M and some relevant properties of this fluid at room temperature are
Figure 3.1: Schematic of the experimental test facility.
listed in Table 3.1. It was selected due to its lower saturation temperature compared to water. This is advantageous for the TLC measurements as the material is commercially available with a maximum temperature of about 120 °C. Additionally, FC-72 is considered an electronics cooling fluid and is therefore relevant to the applications of the present work. With reference to Figure 3.1, FC-72 or water (liquid) enters the loop from the main tank and is continuously circulated by a magnetically coupled gear pump. The pump is equipped with a variable speed drive allowing for a flow range up to 300 ml/min, with a maximum pressure of 517 kPa (75 psi). The tank is a one gallon stainless steel pressure vessel, with a number of ports for fluid transfer and process monitoring. The pressure and temperature in the tank are monitored with an absolute pressure gage and a thermocouple. A 1000 W screw plug immersion heater is installed for fluid temperature control via a variac.

A nutating digital output flowmeter provided by DEA Engineering is used to monitor the flowrate. This meter outputs a 5 V square wave signal at a frequency proportional to the time for the nutator to complete one cycle. The flowmeter has been calibrated using the weighing method, and has a reliable range from 10 ml/min to 250 ml/min. Upstream of the flowmeter is a filter which is used to remove any accumulating particles. A 25 μm disposable filter was used for the microtube experiments and this was later changed to a 15 μm tee-type filter (Swagelok, SS-4TF-15) for the heat sink experiments. In addition, a preheater is located at the inlet of the test section for additional flow temperature control. For the microtube experiments, this preheater consisted of a 2.743 m (9 ft) length of 12.7 mm (0.5 inches) OD serpentine copper piping. Three 0.914 m (3 ft) rope heaters rated at 125 W, were wrapped
Table 3.1: FC-72 properties at 25 °C and 1 atm (3M, 2000).

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho$</td>
<td>1680 kg/m$^3$</td>
</tr>
<tr>
<td>$\mu$</td>
<td>0.00064 Ns/m$^2$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>0.01 N/m</td>
</tr>
<tr>
<td>$C_p$</td>
<td>1100 J/kg°C</td>
</tr>
<tr>
<td>$Pr$</td>
<td>12.35</td>
</tr>
<tr>
<td>$T_{sat}$</td>
<td>56 °C</td>
</tr>
</tbody>
</table>
around the piping. This was later modified to a tube-in-tube heat exchanger for the heat sink experiments as it can provide better protection from boiling in the preheating section. This heat exchanger unit consists of a secondary tank similar to the main tank. The secondary tank is equipped with distilled water and has a 1500 W screw plug immersion heater controlled via a variac. A secondary gear pump circulates the water to the tube-in-tube preheater.

At the exit of the test section is a water-cooled heat exchanger unit, which is used to restore the temperature of the fluid. For the air line, the air enters from the building's source with a pressure of approximately 552 kPa (80 psi) after being filtered and dried. A regulator reduces the pressure to the range applicable for the experiments (34 to 69 kPa). The flow rate is adjusted by a needle valve and a mass flowmeter (Omega model FMA-1609) tracks the air flowrate. The air exits to the atmosphere just after exiting the test module. Teflon tubing (1/4”) is used throughout for fluid transfer, combined with compression fittings and standard pipe fittings. Additionally, pipe fittings in contact with FC-72 utilize a refrigerant sealant paste (Leak Lock by Highside Chemicals and Pro-Tite by Wagner Manufacturing) rather than Teflon tape, for compatibility. The flow loop components are mounted on a MayTec aluminum extrusion benchtop.

3.1.1 Degassing Procedure

Prior to flow boiling measurements with FC-72, the fluid is degassed to purge the system of non-condensables, particularly oxygen. Due to the limited supply of FC-72
fluid available, it is important to utilize a procedure which minimizes the loss of the fluid vapor to the environment. A procedure similar to that by Ammerman (1997) has been incorporated. The fluid is circulated through the system and the degassing tank at a high flowrate ($\sim 150 \text{ ml/min}$ for microtube experiments). The degassing tank is heated until the fluid in the tank boils. With reference to Figure 3.1, valves 11 and 12 are open, allowing for the vapour to travel through the condenser and condense. The air escapes as bubbles through the second degassing bottle, which is filled with water. This acts as a visual technique to track the process. The fluid is circulated for one hour after the temperature in the degassing tank has exceeded $50 \degree C$. The preheater is turned on to maintain the fluid at this high temperature throughout the system. This duration, coupled with the flowrate, allows for the entire fluid in the system to pass through the degassing tank. After this time period, bubbling in the degassing bottles is practically non-existent. At this point valve 11 is closed and the system is shut off from the environment. The flowrate is lowered to the desired setpoint and the tank heater is adjusted to maintain the tank temperature between $50 \degree C$ to $56 \degree C$. The fluid in the tank therefore remains in a two-phase state throughout the experiment and the system pressure may be changed by adjusting the tank’s temperature. To begin measurements, the flowrate is set by adjusting the motor speed and the preheater is adjusted to obtain the desired sub-cooling at test section entry.

For flow boiling measurements with distilled water, a modified procedure is utilized. The fluid in the secondary tank is heated above $90 \degree C$ while the excess vapor is expelled to the environment. The fluid is then circulated at a high flowrate for one hour, allowing the entire fluid to pass through the secondary tank twice.
3.2 Instrumentation and Data Acquisition

The facility is equipped with Type-T thermocouples, as denoted in Figure 3.1, to monitor temperature. Two 1.5 mm diameter Type-T (Omega special error limits material) thermocouples are placed in the various microchannel test section plenum chambers to measure the bulk fluid temperature at entry and exit. Pressure taps are installed in the microchannel test section plenum chambers to track the inlet and outlet gage pressures. They include two 517 kPa transducers (75 psig, Omega model PX01C1) used in Section 5.1 for inlet and outlet measurements, and inlet measurements in Section 5.2. A 345 kPa pressure transducer (50 psig, Omega model PX821) is used in Section 5.2 for outlet pressure measurements and in Chapter 6 for inlet pressure measurements. Finally, a 34.5 kPa differential pressure transducer (5 psi, Omega model PX821) is used for differential pressure measurements in Chapter 6. A Vaisala PTB220 barometer is used to obtain the absolute pressure of the surroundings.

The output from the various sensors is monitored through an automated data acquisition system using the LabVIEW™ software. Code has been developed for various measurement scenarios including microtube experiments and microchannel heat sink experiments under steady state and transient conditions. The data acquisition hardware consists of National Instrument’s SCXI 1000 signal conditioning unit with the appropriate modules and the NI 6052E 16-bit 333 kHz data acquisition card. A schematic of the data acquisition system is shown in Figure 3.2. Signals from the transducers were transferred through the signal conditioning unit, then to the computer, whereas the signals from the metering devices were directly transferred
Figure 3.2: Schematic of the data acquisition system.
through the data acquisition card to the computer.

For heat transfer investigations in the microtubes, heater power was provided by a BK Precision switch mode DC power supply (Model 1692) and a Good Will Instruments DC power supply (Model GPC-1850). Their ratings were a voltage range of 2.7 V to 15 V, with a maximum current rating of 40 A, and a voltage range of 0 V to 5 V with a maximum current rating of 20 A, respectively. For the heat sink experiments, heater power is provided by two BK Precision DC power supplies (Model 1623A) operating in parallel mode, each with a voltage range of 0 V to 60 V and a maximum current rating of 1.5 A.

### 3.3 Image Acquisition

Image acquisition was performed in LabVIEW™ and the images were captured with a size of $640 \times 480$ pixels in Red-Green-Blue (RGB) format. The Hue planes were simultaneously extracted and saved in tagged image file format (TIFF). Post-processing of the images and acquired data was performed in MATLAB. Sample code is presented in the appendix. A detailed view of the image acquisition apparatus is shown in Figure 3.3. Light originating from the illuminator box is directed through a single fibre optic cable. This has the advantage of keeping any heat generated by the light source away from the TLC coated surface. An EKE configured lamp was used, with a colour temperature rating of 3250 K. Light passes through a polarizer just before entering the zoom lens casing. Within the casing, it is deflected to the test surface by a beam
Figure 3.3: Schematic of the image acquisition system.
splitter. Depending on the magnification required, the light may pass through an infinity corrected objective for high resolution measurements or else through the default auxiliary lens. Upon reflection from the surface, the light is circularly polarized. It passes into the zoom lens through the analyzer and is directed to the CCD camera. The crossed polarizing lens pair is available to obtain improved image quality. The light reflected by the TLC material is circularly polarized and thus travels through the crossed polarizing pair essentially unaffected. The acquired image is then transferred directly to the computer as an NTSC signal through a single BNC cable. An LCD television was used for real time monitoring and for positioning. The video signal used in the television loop comes from a separate output line in the camera. The benefit of this is to minimize the accumulated noise that can occur by transferring the desired video signal through many components. Image acquisition is obtained using a Sony 3-CCD analog camera. The camera was connected to a variable zoom microscopic video lens. The camera lens combination was mounted onto a 3-axis traverse through variable length stages each with approximately a 1 \( \mu \text{m} \) resolution. The lateral and vertical axis stages allow for fine tuning of the position and focusing. Although the facility has the capability of a crossed polarizing illumination setup, this was not utilized for the microtube experiments due to the reduced intensity obtained when voltage was applied to the tubes. The fibre optic illumination source was instead fixed onto the zoom lens at a fixed angle and directly illuminated the surface.
3.4 Uncertainty

Uncertainty estimates for the various signals are presented in Table 3.2. For the pressure transducers they are based on manufacturer specifications, whereas for the thermocouples they are based on manufacturer specifications and the combined measurement system. Uncertainty in derived parameters are presented throughout this document where applicable and have been calculated using similar methods to those described by Kline and McClintock (1953).
Table 3.2: Sensor uncertainty.

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple temperatures</td>
<td>0.7 °C</td>
</tr>
<tr>
<td>Pressure transducer (PX01)</td>
<td>0.45 kPa</td>
</tr>
<tr>
<td>Pressure transducer (PX02)</td>
<td>0.57 kPa</td>
</tr>
<tr>
<td>Pressure transducer (PX821)</td>
<td>0.03 kPa</td>
</tr>
<tr>
<td>Barometric pressure</td>
<td>0.015 kPa</td>
</tr>
<tr>
<td>Liquid flowrate</td>
<td>2.25 ml/min</td>
</tr>
<tr>
<td>Air flowrate</td>
<td>0.5 SLPM</td>
</tr>
</tbody>
</table>
Chapter 4

Un-encapsulated Liquid Crystal

Thermography for Mini and Micro Geometries

The general classification of matter is of solid, liquid and gas. For solids a further classification is crystalline and amorphous solid. Crystalline solids are characterized by their ordered molecular structure, which is maintained over a long range. Upon heating a crystalline solid past its melting point, one arrives at an isotropic liquid. The liquid phase, unlike its solid counterpart, does not display any long range ordered structure at its molecular level (Chandrasekhar, 1977).

The term liquid crystal is assigned to a group of intermediate phases, or mesophases, encountered while melting some organic crystalline solids to their isotropic liquid phase. This transitioning process, due to thermal means, is termed thermotropic
mesomorphism, and liquid crystal mesophases obtained in this manner are likewise
termed, thermotropic liquid crystals. Another means of transitioning through vari-
ous mesophases would be due to solvents and the resulting intermediaries are called
lyotropic liquid crystals. In general, the main characteristics of these mesophases are
their strongly anisotropic tendency combined with a high degree of fluidity. They were
first observed by Reinitzer and Lehmann in the late 1800's, during their investigation
of some esters in cholesterol (Chandrasekhar, 1977). Since then, an enormous amount
of research has followed with experimental measurement applications utilizing liquid
crystals emerging over the last 20 years (Ireland and Jones, 2000).

Thermotropic liquid crystals (LC's) may be loosely classified as nematic, cholesteric
and smectic. These liquid crystals are composed of molecules shaped in a rod-like
manner. Figure 4.1 illustrates the various mesophases as described in the following.
Nematic LC's exhibit a preferred molecular order in their orientation direction, but
no order in their position. This orientation direction is called the director. Another
type of liquid crystal may be formed by the addition of a chiral molecule to the ne-
matic LC. This form, called the chiral nematic liquid crystal, is different from the
nematic in that its orientation order is periodic normal to the director. Furthermore,
the intermediate layers transform in a helical fashion, which gives the crystals their
optical activeness. This same structure is observed in cholesterol esters and hence the
coining of the term, cholesteric liquid crystals. Chiral nematic and cholesteric liquid
crystals are synonymous in structure. Smectic liquid crystals come in a variety of
forms (for example Smectic A and Smectic C), and their main characteristic is their
high degree of long range order in both position and direction.
Figure 4.1: States of liquid crystal substances, from Anderson (1999).
A typical phase transformation of liquid crystals can be described as follows. At the melting point one arrives at one form of the smectic mesophase. With further heating one then arrives at the cholesteric mesophase. This mesophase as described earlier and as depicted in Figure 4.2 (a), exhibits optical activity. If viewed during this heating process one would observe the liquid crystal to gradually change color from red to blue. Finally, past the cholesteric mesophase is the clearing point and then the liquid phase. Cooling the liquid crystals produces the reverse effects of heating them, however with hysteresis effects if encapsulated (Anderson, 1999).

4.1 Liquid Crystals for Temperature Measurement

The term *Thermochromic* liquid crystal (TLC) is assigned to the chiral nematic / cholesteric liquid crystals, since they display color change during a thermal process. This optical activity is brought about by their helical molecular configuration. A TLC substance can be modeled as several layers or sheets, each containing uni-directional molecules. The distance between identical directors is termed the pitch and represents a 360-degree helical rotation of the director between layers (Figure 4.2 (b)). The pitch is the parameter that determines the specific wavelength of light that is reflected. As the liquid crystal is heated the separation between the sheets increases due to increased molecular activity, causing the pitch to increase. Simultaneously and with an opposing effect, the relative angular displacement between sheets increases, which decreases the pitch. The dominant effect for most liquid crystals is the increase in
(a) Typical diagram of wavelength of light reflected with increased temperature.

(b) Pitch diagram from Chandrasekhar (1977).

Figure 4.2: Thermochromic liquid crystal characteristics.
angular displacement, and therefore as the temperature is increased, a decreasing pitch occurs. This translates into a decreasing wavelength of light being reflected.

4.2 Color Temperature Interpretation

As has been shown, the TLC's color temperature dependency makes them ideal for qualitative surface temperature mapping. However, in order to obtain quantitative information, a data acquisition and interpretation method is required. According to Moffat (1990) there are three broad classes of image interpretation methods available:

1. Human observers,

2. Intensity-based image processing systems, and

3. True-color processing systems.

Early researchers such as Clifford et al. (1983) and Jones and Hippensteele (1987) acquired images using a video camera and a video tape recorder. Temperature isotherms denoted by constant color lines, were then isolated by a human observer based on replays of the recording. Although the results obtained in this manner were of reasonable accuracy, one can imagine that this was a very tedious and time consuming process.

In order to discuss the intensity and true-color methods a brief summary on computer vision follows. A color may be represented as a combination of three color primaries. This is called the tristimulus decomposition of light. The most familiar method developed by the Committee de l'Ecllirge (C.I.E) in 1931 is the RGB
(red, green, blue) decomposition. The RGB model gives us the relative brightness or luminance of the source. In fact, the luminance of a light source is defined as a proportional summation of the three primaries. Two other descriptors of a light source are its hue and saturation value. Hue is the property that distinguishes color, for example, red light from a green light. Its value depends directly on the wavelength of a color. Saturation is the attribute that describes the relative 'whiteness' of a light. In most electronic image acquisition devices the intensity is captured and converted to an electronic signal. Through filters of various transmissivities the intensity can be converted to its R, G and B values. Finally, if required, computer algorithms can convert the RGB values to hue values.

4.2.1 Intensity Based Image Processing

In the intensity-based image processing methods, two different techniques may be employed, depending on whether a monochrome or color recorder is available. A typical intensity response to temperature change of TLC’s is shown for a transient heating test in Figure 4.3(a). The peak of the intensity plot provides the correlation with temperature as it is a fixed value depending only on the liquid crystal type. On acquisition of images, one could then track each pixel of an image acquisition series for its intensity. Pixels within a specified band of the peak intensity would then have an associated temperature for the corresponding time instant. This single temperature will be unique for the TLC’s and is determined through a calibration. Figure 4.3(b) shows the color primaries intensity response to temperature change of
Figure 4.3: Color descriptor response versus temperature; a) intensity of monochrome system, b) intensity of color primaries and c) hue angle.
a TLC. This is similar to the monochrome intensity method except that now up to three temperatures may be associated.

4.2.2 True Colour Image Processing (Hue Method)

The most prevalent method used in true color processing of TLC images is the hue method. A typical hue response of TLC's is shown in Figure 4.3(c). Curve fitting may be applied to the data points acquired and used as the calibration template. Unlike the intensity method whereby only a single point gives the color temperature relationship, with the hue method the entire curve may be used. This alleviates the need for the entire surface to undergo full color change. For example, if using the green peak in the intensity method, all points on the surface would need to clear the green peak temperature in order to convey quantitative surface temperature information. This is difficult to design for, particularly when testing complex geometries. Based on the aforementioned characteristics, the hue method has been selected as the color descriptor for the present work.

4.3 TLC Material

TLC's come with active temperature ranges of approximately -30 °C to 115 °C and color change bandwidths ranging from about 1 °C to 20 °C. Material with a bandwidth less than 5 °C is typically termed a narrow band TLC, and greater than or equal to 5 °C, a wide band TLC. Both types have their advantages and disadvantages depending on the image processing method used. The un-encapsulated liquid crystal material
used in the present investigations is a chiral nematic blend provided by LCR-Hallcrest. Acquisition sources for the un-encapsulated form of TLC material are limited and the user is subsequently limited in the formulations and volumes available. Three different formulations were purchased during the course of the investigations. Their nominal properties included, a red start of 40 °C with a bandwidth of 10 °C (R40C10W), a red start of 70 °C with a bandwidth of 20 °C (R70C20W) and a red start of 75 °C with a bandwidth of 20 °C (R75C20W). The cost per unit volume of the un-encapsulated material is substantial and more so when compared with the encapsulated form. The cost and formulation restrictions require that experiments be uniquely tailored around the available TLC material. Additionally, surface temperature predictions should be performed to identify the appropriate formulation required.

4.4 Coating Application

The aim of the TLC application procedure is to prepare a uniform coating, which will produce vibrant colors. A thicker coating will improve the vibrancy, while too thick of a coating may start to produce non-negligible temperature gradients through the TLC layer. Additionally, a variably thick layer will introduce uncertainty into the observed temperature field. A uniform coating has traditionally been applied through the use of an airbrush for the encapsulated TLC’s (Ireland and Jones, 2000). Application of the un-encapsulated TLC’s as documented by Höhmann and Stephan (2002); Aligoodarz et al. (1998); Kenning et al. (2001) has been through use of a paint brush. However, Aligoodarz et al. (1998) mentioned significant difficulties in
obtaining a continuous layer through this application method. According to Hallcrest (2000), the LC’s may be applied in their un-encapsulated form through dilution with an appropriate solvent, such as acetone. They then may be sprayed through an airbrush, resulting in an improved thin uniform layer. This approach, however, is usually not recommended due to the hazardous nature of most solvents. Similar methodologies of applying un-encapsulated LC’s with an airbrush have been used for obtaining shear stress measurements. In these cases, the researchers diluted the LC’s with petroleum ether (Fujisawa et al., 2003) or with Freon (Reda et al., 1994) before applying them with an airbrush. No literature, however, to the author’s knowledge demonstrates the application of TLC’s in their un-encapsulated form via an airbrush.

Through trial runs of applying the TLC’s with a paint brush, the difficulties observed by Aligoodarz et al. (1998) were confirmed and therefore this methodology was considered inadequate for producing reliable results. The use of an airbrush was then selected and a variety of trials both of mixture concentration and of application were then performed to produce a sound application methodology. A Badger Model 100 independent action airbrush was used for application as it allowed for variation of the air to coating material concentrations independently during spraying. The solvent used was acetone and concentrations by weight of 20:1 (solvent to TLC) based on the suggestions of Hallcrest (2000) were incorporated. This approach produced highly improved results in terms of coating uniformity and control. Subsequent trials have shown that concentrations as high as 5:1 (acetone to TLC) may also be used. It was necessary to apply the coating within a vented environment due to the toxic nature of the solvent used. Additional solvents tried included isopropyl alcohol and distilled
water. However, both required much larger amounts of solvent proportions to dilute the TLC material and both had significantly longer evaporation times. To improve the color vibrancy of the TLC response, it is common to apply a coat of black paint before the TLC layer. This was incorporated in the current coating application using a water-based black paint and also applying through the airbrush.

4.5 Calibration

To obtain quantitative thermal data from the TLC response, a calibration of the material is required to quantitatively relate the observed color to the material temperature. The definition of hue in the current work is the same as that recommended by Hay and Hollingsworth (1996), which is

$$
Hue = \arctan \left( \frac{\sqrt{3} (G - B)}{2R - G - B} \right).
$$

(4.1)

The perceived color of a TLC will depend upon the primary and background light spectral characteristics, the camera viewing angle and distance, the primary lighting angle and distance, the lights optical path, and the instance of the TLC application. For a calibration to be fully valid the above must be maintained between the calibration and the experiment. This is usually done through an in-situ calibration (Anderson, 1999). Choices for calibration include a linear temperature gradient (Chan et al., 2001), a liquid bath or circulated liquid (Hay and Hollingsworth, 1996), or an isothermal block (Anderson, 1999). A linear temperature gradient applied to a metal
block is advantageous in providing a quick calibration, however, the amount of cal-
ibration points is dependent on the number of sensors employed along the gradient.
An isothermal bath or circulated bath provides the best means of obtaining a truly
isothermal surface, however it does require a liquid supply that is either stirred or
circulated. A constant temperature block is usually difficult to work with in maintain-
ing the full viewing surface isothermal and hence not recommended. For the present
study, a calibration based on circulating the fluid through the channel was selected
due to its ease of incorporation into the setup. For the remainder of this section
the calibration procedure described is based on evaluations with the microtube test
section described in Chapter 5. Similar methods are then utilized for the heat sink
calibration (Ch. 6). The temperatures at the tube inlet and outlet were measured
and remained within 0.5 °C of each other. The fluid was slowly heated via the pre-
heater and images captured at incremental changes in temperature. One of the main
observations from this work is that the response of the un-encapsulated TLC under
increasing magnification will contain varying degrees noise and scatter. Figure 4.4
shows the histogram of a color response from a constant temperature region 30×600
pixels.

To address this in an automated fashion, an intelligent calibration process is intro-
duced. To start, a region of interest (ROI) needs to be selected to base the calibration
curve on. Lighting, viewing angle, and coating uniformity may all influence the cal-
ibration between two different regions on the tube. To eliminate such influences a
calibration curve was produced for a number of regions in the image. The minimum
Figure 4.4: Constant temperature region hue angle histogram.
size for a given ROI is a single pixel, however to account for noisy pixels using statistics, an ROI size greater than one pixel was utilized. Since the camera was traversed this meant that the ROI would be a particular tube location and that high accuracy in traversing between set locations was pertinent. The ROI should not be too large, since local variations will not be captured and for the present work, an ROI of $4 \times 3$ pixels was typically used. A typical image size then of the cropped microtube region of $60 \times 640$ pixels, produces 3200 calibration curves. It was intended to use a 5th order polynomial to fit the calibration data, which is typical in TLC measurements. To manually verify the goodness fit of each curve would be too time consuming and hence two automated approaches that consider extraneous cases were incorporated with the second being an improvement from the first. Figure 4.5 provides a depiction of the calibration data processing and details on the automated curve fitting procedure are presented in the next two sections.

4.5.1 Stage I Calibration

In the first, the fluid was incrementally heated and at each increment images were captured at each of the locations. This method produced only a few calibration points for each location. The number of calibration points was few due to the fact that traversing occurred while the temperature was slowly rising. By the time the camera was returned to a given location, the temperature had risen significantly. If an assumption is made that within the valid calibration range the hue response will increase monotonically with increasing temperature, the main point to verify is that

52
i. Image surface at a given traverse location.

ii. Crop tube section, convert from RGB to hue and define ROI size to produce N calibration curves.

iii. Generate N calibration curves using polynomial fits.

Figure 4.5: Depiction of calibration data processing.
there are no maxima or minima occurring. If a minima or maxima is present, modification of the calibration curve needs to be addressed. It was found that using a third order polynomial will produce a reasonably accurate calibration curve without these minima or maxima locations. An automated calibration curve fitting was utilized with an initial polynomial order of five, but if any local minima or maxima were present within a predefined region, its order was reduced to three. Figure 4.6 (a) shows a calibration fit with a fifth order polynomial. Figure 4.6 (b) shows a calibration curve with data points originally set to a 5th order fit then corrected to a 3rd order fit.

4.5.2 Stage II Calibration

The second approach consisted of heating incrementally at a fixed location and capturing the full bandwidth before moving to the next location. This method is implemented for all the microtube measurements using FC-72 and the heat sink measurements. The main issue is to minimize the calibration time and capture as many points as possible. In the first approach calibration time was on the order of three hours and few data points were captured. In the second approach the fluid’s inevitably increasing temperature was taken advantage of, and therefore since the camera was at a fixed location, it would continuously capture images as the temperature slowly rose. The rate of heating was approximately 0.5 °C per minute. The aim of the curve fit is to produce a monotonically increasing relationship. Depending on the quality of the data points for each ROI, this was not always possible. Extraneous cases would
Figure 4.6: Stage I corrected calibration curves.

(a) Fifth order fit.

(b) Fifth to third order fit.
have maxima or minima locations in the curves. To automatically determine a calibration data curve-fit that did not contain extraneous regions, the following logic was implemented:

1. Sort the data set with respect to increasing hue.

2. Starting from the minimum hue value, only keep data points that have a positive change in temperature with respect to increasing hue.

3. Fit a fifth order polynomial curve to the modified data set.

4. If any maxima or minima exist in the hue range of 25 to 175 discard the fifth order polynomial curve and fit a third order polynomial curve.

5. If any maxima or minima exist in the hue range of 25 to 225 modify the data sets zero location to a predefined temperature. If any maxima or minima exist in the hue range of 100 to 175, abandon curve fitting and use the neighboring ROI instead.

The ranges were selected based on repeated observations of curve fits, however depending on the experiment type and conditions, they may be adjusted. The change of hue range between steps 4 and 5 was due to the change in characteristics of the two polynomial types applied to the same data set. The change of hue range between steps 5 and 6 was because once a third order fit was applied and the zero location fixed, if an extremity in the curve occurred in the hue range of 25 to 100, it was a very slight minima that could be disregarded. That is the temperature dip at the minima was less than 0.1 °C.
4.5.3 Additional Corrections

The hue angle is defined on a polar system and hence due to scatter, hue values close to 0° may be present in the 300° to 360° range and vice versa for hue values close to 360°. Compensation similar to that of Hay and Hollingsworth (1996) is incorporated whereby the negative angle was used when such scatter occurs. This can be identified, for example, when at high temperatures a low hue value is obtained, or at low temperatures, a high hue value is obtained. The ROI will have a particular scatter in the hue values and for the present study, the nominal hue value was based on the median of the hue values. With the above three conditions incorporated, Figure 4.7 shows typical calibration curves for a given traverse location. Each calibration curve is for is for a ROI. The symbols in the figure only differentiate the curves and are not calibration data points. Figure 4.8 (a) shows a good fifth order curve fit. The circles are the section of the data set that is kept after sorting. Figure 4.8 (b) shows the modification of a fifth order fit to a third order fit due to a maximum in the curve-fit. Once again the circles are the section of the data set that are kept after sorting. Such good fits were not always possible due to scatter and noise as described previously.

The TLC calibration may change slightly over time. It was important then to verify that over a measurement run, which could be as long as three hours, the calibration curve would not shift significantly. To verify this, calibration data at a fixed location were obtained at two different times 3.5 hours apart. Figure 4.9 shows the data obtained from these two calibration runs, determined from a ROI size of

57
Figure 4.7: Typical calibration curve fits for various ROI positions at a given traverse location.
Figure 4.8: Stage II corrected calibration curves.
Figure 4.9: Calibration curves at different time periods.
4×3 pixels. As can be seen in Figure 4.9, the change in the calibration curve is insignificant, with a maximum temperature change of approximately 0.5 °C. Hence, the calibration taken prior to the measurements runs can be used over the entire measurement period.

4.6 Methodology Validation

4.6.1 Test Module Selection

Heat transfer in mini and microchannels is the main motivation for the experimental methods developed in the present study. Such channels will come in many shapes, which are predominantly dictated by the manufacturing processes available. An additional constraint will be the measurement technique utilized in the experiments. To design a heated channel test module utilizing TLC’s for measurement, a primary criterion is optical access to the measured surface. In the most basic case, a flat surface is utilized with the camera at its normal. Three test module designs were investigated during validation of the test facility, with each subsequent design being a consequence of shortcomings in the previous configuration. They are described hereafter and results are presented for the final design.

Heat transfer from a single-wall heated rectangular mini channel with TLC’s has been investigated by Chin et al. (2002) and Piasecka and Poniewski (2004) for boiling flows. The channels had widths of 20 mm and 40 mm and heights from 0.5 mm to 2 mm. The test modules were manufactured from aluminum with the channels
milled in a slot, while the fourth wall was produced from a metallic foil. Air gaps were machined on the side and bottom walls to produce approximately adiabatic conditions. The backside of the foil was coated with encapsulated liquid crystal and bolted closed with a transparent cover. The initial validation test concept, utilized similar characteristics of the above design. The base of the channel was manufactured from G-10 fiber-glass with three of the four channels produced from a milled out slot. The materials low conductivity allowed for approximately adiabatic conditions. The fourth wall was provided via a 25 μm thick and 1 mm wide foil supported on a polycarbonate cover. A schematic of this setup is shown in Figure 4.10.

The TLC material was applied on the backside of this foil before it was placed on the cover. Several difficulties were encountered with this approach. First, the use of un-encapsulated TLC’s directly in the channel caused the coating to erode over time. Unlike the encapsulated form typically used by previous researchers, the un-encapsulated TLC material does not solidify, but rather remains in an oily state. Therefore, any air pockets present on the backside of the foil where the TLC coating is located, will draw fluid and/or will allow for the TLC material to escape over the course of an experiment. Attempts to squeeze out all the air bubbles are difficult since too much pressure will force out the TLC material and create a discontinuous measurement surface. A second drawback of this design was the use of a thin foil, which was very difficult to handle and to produce accurately. In addition, alignment of the foil with the channel was challenging and each failed alignment trial required the coating to be re-applied, as it was destroyed due to contact with the cover. The use of the thin foil highlighted how any minor local fault in the foil would produce
Figure 4.10: Configuration for Test Module 1.
localized hot or cold spots, characterizing a non-uniformly heated surface. A third drawback of this design was the use of an o-ring for sealing. This seal introduced additional uncertainty into the height of the channel as well as produced secondary paths for the fluid flow. It was noted that these difficulties were likely not encountered in the works of Chin et al. (2002) and Piasecka and Poniewski (2004) to this extent, due to the width of their channels (20 mm and 40 mm as opposed to 1 mm) and their use of encapsulated TLC’s (as opposed to the un-encapsulated form). The primary conclusion from the above design was that if the TLC coating could be isolated from the flow path, it could be an effective measurement tool.

A second test module design was investigated to address the above drawbacks. In this concept, a wide thin foil was used rather than a narrow thin foil. The wider foil was easier to handle and minimized local faults produced during its slitting to size. Its size was based on the power available, since too wide a foil would draw too much current. The foil was permanently secured to the base using an epoxy and thus eliminated the need for an o-ring seal. It was important to prevent the epoxy, prior to drying, from seeping into the channel while under compression. To prevent this, a layer of double-sided tape was placed at the channel edge as a barrier. This had the added advantage of keeping the channel height fixed rather than the height being dictated by the dried epoxy, which would make it uneven. The height of the channel was increased however, so selection of a thin tape was necessary. A drawback of this semi-permanent sealing method, was that it was semi-destructive if resealing was necessary. In addition, lead-time for each new seal including drying time was about two days. Finally, each new sealing added uncertainty into the pressure the channel
could withstand due to unforeseen random errors, which occurred during assembly.

The foil required cooling in the section not exposed to the channel. A cooling sink of copper blocks with water running in a copper pipe over-top was incorporated. An added advantage of the copper blocks was that they could be bolted to the test base providing additional pressure for sealing. This was necessary since the epoxy tack to the foil or to the base was not even in all regions and had limited strength. Also, the region where the foil tacked onto the double-sided tape was prone to becoming loose with increased channel pressure, which would subsequently change the dimensions of the channel. However, even with the copper blocks the test piece was sensitive to increased channel pressure depending on how well it was sealed. The use of a metallic heat sink required electrical isolation from the foil, while minimizing thermal resistance. An alumina based epoxy that fit this requirement was utilized. It was necessary though to control the flow of the epoxy before it dried, as it was under pressure and so it may flow onto the measurement region. Once on the measurement region, it is difficult to remove without affecting the surface texture and sometimes destroying the foil. A schematic of the channel is shown in Figure 4.11.

Identification of the channel underneath the foil was challenging and it was necessary to utilize a microscope. Nusselt number measurements utilizing this test set up were found to be high in comparison to a number of correlations. After consideration of the test set up, it was concluded that these high Nusselt values were due to an inaccurate measure of the heat transfer from the top wall to the fluid. In calculating the heat transfer coefficient, the enthalpy change of the fluid was used rather than the power applied to the foil. When these two values are compared the enthalpy
change is much higher for the fluid than the energy from the foil over the channel area, which suggests an alternative source of energy. It was observed that the foil surfaces adjacent to the channel had extremely high temperatures in comparison to the surface above the channel. Therefore, although the base material had a low conductivity, heat transfer through the sidewalls produced the additional energy to the fluid. Furthermore, this meant that cooling of the foil was not as effective as it was designed for.

The Joule heating produced in the foil was not used to calculate the heat transfer since the foil surface and dimensions were not uniform, and hence the foil resistance was not uniform everywhere. This occurred since it was necessary to roughen the foil in regions where epoxy was applied for adhesion. Another source of excess energy transfer to the fluid occurred at the connection of the electrodes to the foil surface. This occurred, as the region where the two were connected was very small and difficult to reach, and hence inadequate connections produced high resistance causing this unwanted energy dissipation. Overall, assembly of this second design was difficult at all stages particularly due to the small size of the regions being worked on. This second design, however, demonstrated the assumption from the previous design that with the TLC isolated from the flow, effective measurements could be produced. However, it also demonstrated that application of accurate boundary conditions was difficult as the size was reduced.
4.6.2 Circular Tube Test Module

The main difficulties with the previous two concepts were the sealing and the application of a controlled heat load. Many setbacks were observed due to the extensive in-house assembly required, which produced many failed trials. In conceiving the third design, this point was of primary concern. Readily available on the market are relatively thin-walled stainless steel circular tubes, usually produced as hypodermic needles. Tube inner diameters are available as low as 0.254 mm and with wall thicknesses as low as 0.051 mm (Small Parts). Rectangular tubing is also available and can be custom made with inner cross sections as low as 0.510 mm (Microgroup). The use of custom-made tubing is, however, expensive and requires minimum orders not necessary for small-scale experiments. For this reason, a circular tube was selected as the channel configuration for the third design. The recent use of circular steel tubes in heat transfer studies of mini and microchannels has been performed by Lelea et al. (2004) and Owhaib and Palm (2004) using thermocouples, as well as Hetsroni et al. (2003) and Hapke et al. (2000) using infrared measurement.

A schematic of the circular tube test module used is shown in Figure 4.12. A polycarbonate sheet was machined to produce the measurement chambers for pressure and temperature and a detailed CAD drawing is presented in the Appendix. The tube had a 1.0668 mm (0.42") inner diameter and an outer diameter of 1.27 mm (0.5"). It was connected to these chambers using standard 1/16" stainless steel compression fittings with specialty ferrules to accommodate the small diameter tubing. The ferrule material is a composite of graphite and polyimide, which has a high
electrical resistance. The advantages of the compression fittings are that they are resealable and can withstand high pressures (100 psi rating). Copper stranded wire (14AWG), was used for the electrical lead connection. It should be noted that the present set-up did not utilize any cover directly in contact with the TLC surface due to the small size of the tube. Attempts were made to utilize optically transparent epoxies to encapsulate the tube, however, none of the ones tried were compatible with the TLC material. Trials were also performed with transparent heat shrink tubing. This, however, produced localized air pockets and the heated TLC would easily be expelled. As well, for smaller tube sizes, as will be presented in Chapter 5, appropriate heat shrink tube sizes were not available. Rather then, a non-contacting cover was placed over the tube to protect the TLC coated surface from dust and the room lights. The cover was an opaque plastic half tube which is open at both ends. The main drawback of this was the reduced color vibrancy. Future studies with geometries such as microtubes may consider the use of glass capillary tubes for in-contact covers.

4.6.3 Procedure

Prior to each experiment, the TLC material was calibrated as described in Section 4.5.1. To begin measurements, the flow rate was set and the system was then left to run for about ten minutes. Measurements were carried out over a flow range of 20 to 160 ml/min, with inlet temperatures ranging from 28.4 °C to 31.9 °C and outlet temperatures ranging from 32.9 °C to 38.3 °C. The room lights were dimmed and the tube was kept covered. Traversing began at the rear of the tube where the wall
temperature would be the highest. The configured magnification provided a field of view of 10.233 mm \( \times \) 7.671 mm, which translated into 15.53 \( \mu \text{m} \)/pixel for the CCD sensor size. Although the facilities magnification capabilities are much larger, a higher magnification would have required more traversing in order to capture the entire tube. This would require more time and provide additional information unnecessary for the purpose of the present work. After running flow through the system, measurements were ready to be taken and the cover was removed. The illumination system was turned on and the voltage was slowly adjusted until the TLC color response was in the predominantly green range (\( \sim \)41 \(^\circ\text{C} \) to 46.0 \(^\circ\text{C} \)). The system was then allowed to come to equilibrium, which spanned 5 to 10 minutes and afterwards a measurement was taken.

Three images were captured at a speed of 30 frames per second. These colour images were converted to hue angle, scaled on an integer range from 0 to 255, and then averaged for each pixel location. Simultaneously, the fluid temperature at inlet and outlet, as well as the flow rate measurements were captured. For each traversed location, two to four measurements were captured at a given flowrate, though the heater power may be adjusted between image capture. During data reduction, the best images by visual inspection were selected for final presentation. Whenever the voltage was adjusted, the illuminating system was turned off in order to minimize any degradation to the TLC and external heat addition to the tube as the system approached equilibrium.
4.6.4 Data Reduction

The local Nusselt number was calculated as

\[ Nu_z = \frac{h_z D}{k_{loc}}. \]  \hspace{1cm} (4.2)

The local heat transfer coefficient \((h_z)\) was obtained through the convective heat transfer relation with the following parameter definitions:

\[ h_z = \frac{q''}{(T_{w,zy} - T_{b,z})}. \]  \hspace{1cm} (4.3)

The heat flux was calculated based on the fluid enthalpy change given by

\[ q'' = \frac{\dot{m}C_p(T_{out} - T_{in})}{\pi DL_{heated}}. \]  \hspace{1cm} (4.4)

The local fluid bulk temperature \((T_{b,z})\) was determined from an energy balance and at each streamwise location was

\[ T_{b,z} = T_{in} + \frac{\pi D q''}{\dot{m}C_p} \cdot z. \]  \hspace{1cm} (4.5)

Properties for the above calculations, except the local Nusselt number, were based on the average fluid temperature between inlet and outlet. The properties were obtained from Kays and Crawford (1993) and were interpolated between temperatures. \(T_{w,zy}\) was based directly on the TLC measurement. Estimation of the internal wall temperature assuming 1-D heat transfer gave a wall temperature gradient less than
0.3 °C. This was well within the wall temperature uncertainty and not incorporated at this stage.

4.6.5 Uncertainty

The uncertainty in the TLC temperature was calculated similar to the methods discussed in Hay and Hollingsworth (1996) and in Chin (1997). The wall temperature uncertainty was taken to be:

\[ \delta T_w = \left( \frac{dT}{dHue} \delta Hue \right)^2 + \delta T_{\text{fluid}}^2 + (2 \times SEE)^2, \]  

(4.6)

where \( dT/dHue \) is the sensitivity in the temperature hue curve, which was either a fifth or third order polynomial as discussed earlier. Also, \( \delta Hue \) represents the standard deviation in hue for the ROI, and since the ROI’s were small (around 12 pixels), a constant value of 1° was selected instead. An additional 2° bias was added for the RGB to composite signal conversion uncertainty. In addition, \( \delta T_{\text{fluid}} \) was the uncertainty of the thermocouple probes used to measure the fluid temperature during calibration. Finally, the standard error estimate \( (SEE) \) takes into account the error in the polynomial curve fit and is typically defined as

\[ SEE = \left[ \sum_{k=1}^{n} \left( T_k(Hue) - T_{\text{fit},k}(Hue) \right)^2 / (n - j - 1) \right]^{1/2}, \]  

(4.7)

where \( n \) is the number of data points, and \( j \) is the polynomial fit.
The calibration yielded only a few points for each tube section and sometimes produced points with large scatter that may significantly alter the SEE unreasonably. A constant value was instead selected for the SEE based on repeated observations of the calibration curves of 0.5 °C. A typical wall temperature uncertainty for the TLC's of 1.02 °C was estimated and is similar to that of other researchers (Chin et al., 2002). Typical uncertainties for other parameters considered in the results are listed in Table 4.1. The uncertainty in the local Nusselt number is high, but this is due to the use of a heat flux derived from the fluid enthalpy change which is a function $T_{\text{fluid}}$. This was initially selected to alleviate the need for estimating the energy loss to the environment. If instead the measured power was used, an estimate of typical uncertainty in Nusselt number showed that it decreased from 23.2% to 12.7%. This value is typical of what is quoted in most works.

4.6.6 Results

Figure 4.13 shows a sample of the surface temperature variation observed as well as the corresponding fluid bulk temperature deduced from Equation 4.5. This dataset is from a condition whereby the heater flux was kept constant at each traverse. The data is from six separate images, for a range of 40 to 100 $z/D$ and is for a Reynolds number of 610. The wall temperature rises with an approximately constant slope from 41 °C to 44 °C, as would be expected for a constant heat flux boundary condition. Such a distinct wall temperature slope was not always observed, particularly for the higher flow rates whereby the wall temperature rise was much smaller. The wall temperature
Table 4.1: Typical parameter uncertainty for results.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Local Nusselt number, $N_u$</td>
<td>23.2 %</td>
</tr>
<tr>
<td>Reynolds number, $Re$</td>
<td>7.78 %</td>
</tr>
<tr>
<td>Flowrate</td>
<td>7.60 %</td>
</tr>
<tr>
<td>$z/RePrD_<em>, z^</em>$</td>
<td>9.60 %</td>
</tr>
<tr>
<td>Normalized distance, $z/D$</td>
<td>16.20 %</td>
</tr>
</tbody>
</table>
Figure 4.13: Sample measured wall temperature and calculated fluid bulk temperature.
data is a streamwise average over the visible surface and has been down-sampled for clarity. Figure 4.14 shows the streamwise averaged variation of the Nusselt number for the laminar flow cases, which was generated by considering approximately 90 diameters of tube length and which is a combination of nine adjacent tube sections imaged separately. For each tube section, there is a calibration curve for a region size of $4 \times 3$ pixels and each averaged data point consisted of 60 spanwise pixels. The data is compared with the analytical solution for laminar thermally developing flow in a circular tube, with a constant heat flux boundary condition (Shah and London, 1978). This solution has been demonstrated to have good agreement with other experimental data for tube flows down to 125 $\mu$m (Lele et al., 2004). The results presented in Figure 4.14 are in good agreement with this solution, showing a decreasing trend towards the fully developed value of 4.364. There is some scatter between the cases, which is partially due to the noisy response of the liquid crystal. Other discrepancies will arise from the ability to repeatedly traverse to a given section both during calibration and experimentation.

In Figure 4.15, the streamwise averaged Nusselt number is plotted for the laminar and turbulent flow rate cases. Each streamwise plot is a combination of nine tube sections imaged separately. For the laminar flow cases, the Nusselt number trend is relatively smooth, showing a slight decrease in the initial length, followed by a constant trend for the remaining length. For the turbulent cases, however, the Nusselt number is slightly oscillatory, particularly farther downstream. These oscillations are not physical phenomena, rather they are due to noise in the images and local non-uniformities in the coating. It was observed that the application of the voltage had
Figure 4.14: Streamwise averaged Nusselt numbers for laminar flow cases.
Figure 4.15: Streamwise averaged Nusselt numbers for various flowrates.
a minor influence on the TLC's and produced some coalescing of the coating, hence some of the local non-uniformities. Local non-uniformities also arose due to dust particles settling on the coating over time. This was a further reason for the localized calibration methodology developed. Variation between tube sections will be due to minor fluctuations during measurement in parameters such as the flow rate as well the inability to reproduce the original positions during traversing. In addition, between traversing for a fixed flow rate, the heat flux was sometimes adjusted as mentioned earlier to have the wall temperature within the TLC's active and reliable range.

The averaged Nusselt number for the laminar and turbulent flow cases is shown in Figure 4.16. Two sets of data are presented from experimental runs on different days. The turbulent results are compared with two heat transfer correlations for fully developed flow in smooth circular tubes. The laminar results are compared with an analytical solution for thermally developing flow with a constant heat flux boundary condition in a circular tube. For the turbulent results, one correlation is given by Gnielinski (1976) for macro size tubes as

\[
Nu = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)},
\]

where

\[
f = [1.82 \cdot \log(Re) - 1.64]^{-2}.
\]
Figure 4.16: Variation of the averaged Nusselt number with Reynolds number, with comparison to correlations.
The second is the Dittus-Boelter (Rosenhow et al., 1998) correlation also for macro size tubes and given by

\[ Nu = 0.024 Re^{0.8} Pr^{0.4}. \tag{4.10} \]

For the laminar results, the following solution which gives the averaged Nusselt number at a streamwise location for laminar flow was used (Shah and London, 1978),

\[ Nu = 4.364 + \frac{0.046}{(z^*)^{1/3}}. \tag{4.11} \]

The turbulent results are in very good agreement with the correlation given by Gnielinski (Equation 4.8), whereas the Dittus-Boelter (Equation 4.10) correlation appears to over predict the heat transfer. Equation 4.10 is in the functional form for which most experimental correlations of turbulent heat transfer are based (Holman, 2002). Whereas, Equation 4.8 has a similar functional form as the analytically derived relation for turbulent flow heat transfer inside tubes (Kays and Crawford, 1993). Both relations however have constants derived from empirical methods. The data suggests that the combination of the functional form and the empirically derived constants of Equation 4.8, provide a better prediction for turbulent heat transfer in smooth tubes, under the present conditions. Additionally, it is noted that Equation 4.8 is the recommended choice for turbulent heat transfer predictions in smooth tubes for the present flow ranges (Kays and Crawford, 1993; Rosenhow et al., 1998). The figure also shows data from two separate days demonstrating the good repeatability. The laminar results are also in very good agreement with the analytical solution and
this was expected since it is derived from the same solution used in the comparison for Figure 4.14.

4.7 Summary

The development of a method utilizing un-encapsulated thermochromic liquid crystals for local heat transfer measurements in microgeometries has been described. To the authors' best knowledge, this is the first use of this form of liquid crystal for heat transfer measurements in a channel this small. The un-encapsulated TLC form is difficult to handle and significant scatter and noise may be present in measurements with increased magnification. A localized calibration is utilized to minimize the influence of lighting, viewing and local variability's. A large number of calibration curves are obtained from this approach and so automation, as well as logical filtering are necessary. The calibration data generated are demonstrated to be insignificantly un-altered over the course of an experimental run. The use of an airbrush is found to produce a significant improvement in coating uniformity and thickness control as compared to application with a paintbrush.

The motivation for the methods developed was the need for increased and improved measurement in mini and microchannels. Three channel configurations were discussed and due to difficulties in the first two arrangements, attributed to the combination of the TLC's and small size, results are presented for only the third configuration, which is a circular tube. The results demonstrate that the conventional size analytical solution for laminar flow is adequate for predicting the local heat transfer in
thermally developing flows inside a 1 mm nominally sized circular tube. In addition, the Gnielinski correlation for conventional size channels is adequate for predicting the average heat transfer in a 1 mm nominal size circular tube.

The circular tube design was easily configured from readily available parts and is an effective approach for channel heat transfer studies of diameters down to 250 μm. Some of the scatter and noise in measurements was introduced due to the need to traverse the test surface. It is noted that during application of the voltage on the circular tube geometry, the TLC material coalesced in some regions altering the surface uniformity and a slight dimming of the response intensity occurred. This was not observed in the other two geometries and was not documented in previous heat transfer experiments utilizing the un-encapsulated TLC’s. This observation highlights the need for additional work related to the use of the un-encapsulated form of TLC’s for engineering thermometry applications, as they are a promising potential for micro thermography applications. Overall, the methods developed and utilized have been shown to provide the local heat transfer coefficient in a micro geometry of 1 mm nominal size, and they are a viable approach for geometries of smaller sizes.
Chapter 5

Transport Phenomena in Microtubes

Limited data is available on the thermo-fluid transport characteristics in sub-millimeter channels particularly under flow boiling conditions. This chapter presents pressure drop and heat transfer measurements in microtubes ranging from 0.25 mm to 1.00 mm nominal inner diameters. Single phase and flow boiling conditions are both considered separately. Single phase flow covers both the laminar and turbulent regimes. The test section configuration is as depicted in Figure 4.12 and described in Section 4.6.2. The un-encapsulated TLC methods developed and validated in Section 4.6 are utilized for measurement.
5.1 Single Phase

5.1.1 Data Reduction

The friction factor is obtained from the pressure drop relation,

\[ \Delta P = \left( f \frac{L}{D} + K_{\text{loss}} \right) \frac{\rho u^2}{2}. \]  \hspace{1cm} (5.1)

The losses at the entrance and exit \( (K_{\text{loss}}) \) are estimated from Streeter (1961), considering the area changes at each location between the pressure port and the tube entrance and exit. The local Nusselt number and local heat transfer coefficient are obtained as previously defined in Equations 4.2 and 4.3. The heat flux to the fluid is calculated based on the power applied to the tube and corrected for losses to the environment. This is given by,

\[ q'' = \frac{\tilde{P}_{\text{tube}} - h_\infty A_{\text{o,heated}} (T_{\text{w,TLC}} - T_\infty)}{A_{\text{i,heated}}}. \]  \hspace{1cm} (5.2)

The external heat transfer coefficient \( (h_\infty) \) is obtained from tests with no fluid circulating and a constant power applied to the tube. The power to the tube is the measured voltage and current, corrected for power loss across the lead connections. Comparisons were made between the fluid enthalpy rise based on measured inlet and outlet temperature differences versus the electrical power applied to the tube. These differences were found to be within 5% of each other for the majority of the cases. However, in a few cases where the fluid bulk temperature rise was low, differences were
on the order of 20 to 40 %, this can be attributed to fluid temperature measurement uncertainty. The internal wall temperature was calculated assuming one-dimensional radial heat conduction through the wall, with the outer wall temperatures measured by the TLC’s as boundary conditions. That is given,

\[
\frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{\dot{q}}{k} = 0
\]  
(5.3a)

\[
T (r_o) = T_{w, TLC}
\]  
(5.3b)

\[
-k \frac{\partial T (r_o)}{\partial r} = h_\infty (T_{w, TLC} - T_\infty)
\]  
(5.3c)

then the inner wall temperature \(T_{w,i}\) is given by,

\[
T_{w,i} - T_\infty = -\frac{r_o^2 \dot{q}}{4k} + C1 \cdot \ln (r) + C2
\]  
(5.4a)

\[
C1 = -\frac{r_o h_\infty}{k} (T_{w, TLC} - T_\infty) + \frac{r_o^2 \dot{q}}{2k}
\]  
(5.4b)

\[
C2 = (T_{w, TLC} - T_\infty) + \frac{r_o^2 \dot{q}}{4k} - C1 \ln (r_o).
\]  
(5.4c)

The difference between inner and outer wall temperatures were minor, however, and typically less than 0.3 °C. The local fluid bulk temperature \((T_{b,z})\) was determined from an energy balance and at each streamwise location was as defined in Equation 4.5. Properties for the above calculations, except the local Nusselt number, were based on the average fluid temperature between inlet and outlet. The properties
were obtained from Kays and Crawford (1993) for water and 3M (2000) for FC-72.

Uncertainty estimates for the TLC measured wall temperature are similar to those presented in Section 4.6.5. Uncertainty in measured signals is presented in Section 3.4. Diameter dependent uncertainties included the tube inner and outer diameter, the flowrate and the Nusselt number, and they are listed in Table 5.1. Note, the diameter uncertainties are based on manufacturing tolerance.

5.1.2 Friction Factor

The friction factor results are shown in Figure 5.1. They are compared with the friction factor for fully developed laminar flow in a pipe given by (Shah and London, 1978),

\[ f = \frac{64}{Re}, \]  \hspace{1cm} (5.5)

and Equation 4.9 for fully developed turbulent flow. Equation 4.9 was selected due to its use in the average Nusselt number relation. Very good agreement is observed for all diameters in the laminar range. In addition, the predicted curves show very good agreement for the 1 mm diameter tubes in the turbulent regime. For the 0.5 mm case the results are under predicted by Equation 4.9. This equation though is valid for smooth tubes with a \( Re > 10^4 \). This may suggest an increasing tube roughness influence with decreasing diameter, however, as seen later for the average Nusselt number results, no benefit in heat transfer is gained from the higher friction factor. Rather, it suggests that the estimated entrance and exit loss factors are the cause of the differences observed. The transition for both the 1 mm and the 0.5 mm
Table 5.1: Typical uncertainties for diameter dependent parameters.

<table>
<thead>
<tr>
<th>Nominal Tube Diameter</th>
<th>Inner Diameter</th>
<th>Outer Diameter</th>
<th>Flowrate</th>
<th>Nusselt Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.254 mm</td>
<td>±0.0254 mm</td>
<td>±0.0127 mm</td>
<td>±0.5 ml/min</td>
<td>±14-17%</td>
</tr>
<tr>
<td>0.508 mm</td>
<td>+0.0254 mm</td>
<td>+0.0127 mm</td>
<td>±1.5 ml/min</td>
<td>±8-18%</td>
</tr>
<tr>
<td>0.0127 mm</td>
<td>-0.0000 mm</td>
<td>-0.0127 mm</td>
<td>±1.5 ml/min</td>
<td>±8-18%</td>
</tr>
<tr>
<td>1.067 mm</td>
<td>±0.0254 mm</td>
<td>±0.0127 mm</td>
<td>±1.5 ml/min</td>
<td>±9-18%</td>
</tr>
</tbody>
</table>
Figure 5.1: Microtube friction factors versus Reynolds number.
tubes is observed between a Reynolds number of 2000 and 4000. This is observed for both fluids and shows that the conventional value of 2300 is reasonable for these two diameters. For the 0.25 mm case, however, a delayed transition is observed and the curve appears to be changing direction at a Reynolds value of about 5000.

5.1.3 Heat Transfer Coefficient

Figure 5.2 shows a pair of data sets from raw image to processed data. Figure 5.2 (a) is considered a good data set and is typical of the data presented in this section. In the figures the temperature is rising from left to right and covers a distance of approximately 10 mm. In Figure 5.2 (a), there is some minor scatter in the red range due to the large light intensity, however, in the green/blue range the response is much better. Most of the data presented is derived from images that fell in this green/blue range. Figure 5.2 (b) only has a small useful region, therefore such a data set is not considered for final presentation. Figures 5.3 to 5.7 show the circumferentially averaged streamwise Nusselt number. The data are compiled from a number of traversed locations and the heat flux used was not always constant. For the majority of the plots in the laminar regime (taken as Re < 2300) a decreasing Nusselt trend is observed with increasing distance, approaching an asymptotic value. Although for some laminar cases the downward slope is minimal, their slope tends to be more significant than for the turbulent and transitional cases. For the remaining cases the Nusselt number is constant. The decreasing versus constant trend of laminar versus turbulent flows are as expected based on physical considerations and observations in
Figure 5.2: Typical raw to processed results.
larger tubes (Holman, 2002). For data in the turbulent regime, a short developmental length is observed, which can be generalized as being less than 10 hydraulic diameters.

Shown in Figures 5.8 to 5.11 are the local heat transfer results for the laminar regime data. The results show that the developmental length for the laminar results are well represented by the analytical solution given in Shah and London (1978). In the fully developed region, the Nusselt number tends to the analytical result of 4.364, which is for constant heat flux in a circular tube. The results show no unexpected trend related to decreasing tube diameter.

The average Nusselt number for all the tube diameters and the two fluids is shown in Figure 5.12. These data are obtained from the average of the farthest downstream image captured to allow for thermal development. The Nusselt number is divided by $\Pr^{0.4}$ to account for the two differing fluids, with the Prandtl number taken at room temperature. The data are compared with two correlations, one the Gnielinski (1976) correlation (Equation 4.8) representing large tubes. The second, a correlation given by Adams et al. (1998) and representing small diameter channels. It is based on Equation 4.9, with a slight modification as follows,

$$\text{Nu}_{\text{Adams}} = \text{Nu}_{\text{Gniel}} (1 + F)$$

where,

$$F = 7.6 \times 10^{-5} \text{Re} \left[1 - \left(\frac{D}{1.64}\right)^2 \right].$$

The data are well correlated between each other particularly between the 0.5 mm
Figure 5.5: Streamwise Nusselt numbers - 1.0 mm, FC-72.
Figure 5.8: Laminar streamwise Nusselt number - 1.0 mm, H$_2$O.
Figure 5.9: Laminar streamwise Nusselt number - 0.5 mm, H$_2$O.
Figure 5.10: Laminar streamwise Nusselt number - 1.0 mm, FC-72.
Figure 5.11: Laminar streamwise Nusselt number - 0.25 mm, FC-72.
Figure 5.12: Average Nusselt numbers.
and 1.0 mm tubes at higher Reynolds. The influence of the delayed transition for the 0.25 mm FC-72 case is reflected also in the average Nusselt number (Figure 5.1). Comparing the data to the correlations, at first glance the Adams et al. (1998) correlation appears to be the better choice. However when considering experimental uncertainty of the experiment, correlation, or both (taken as 15%), the Gnielinski (1976) correlation also demonstrates an adequate comparison. Tube roughness is not considered a major influence as both tubes showed differing friction factors yet similar Nusselt numbers. The conclusion therefore is that in the operational range considered of \( Re < 1.2 \times 10^4 \) and Prandtl number ranging from 5.4 to 11.5, and with experimental uncertainties kept in mind, both the Gnielinski (1976) correlation for large tubes and the Adams et al. (1998) correlation suggested for small tubes are adequate. Further as a conservative design guideline, when determining the heat load applied, the Adams et al. (1998) correlation should be used, whereas when estimating the cooling benefit gained, the Gnielinski (1976) correlation should be used. These guidelines are deemed general, as there will be few applications of tubes in the sub-millimetre range with Reynolds numbers exceeding \( 10^4 \), as the pressure drop becomes unrealistically large.
5.2 Boiling

5.2.1 Data Reduction

The local heat transfer coefficient \((h_z)\) is obtained as defined in Equation 4.3. The heat flux to the fluid is calculated based on the power applied to the tube and corrected for losses to the environment as defined in Equation 5.2. The internal wall temperature is calculated from Equation 5.4. The difference between inner and outer wall temperatures was minor and typically less than 0.3 °C.

In order to determine the local heat transfer coefficient, the local bulk temperature is required. In the single phase region it is based on an energy balance from the inlet, whereas in the saturated flow boiling region, it is determined from the local saturation pressure. The local saturation pressure is assumed to be linearly decreasing within the two-phase region. This method is similar to that adopted by many recent works (Qu and Mudawar, 2003; Yen et al., 2003; Lee and Lee, 2001; Tran et al., 1996). The local quality is then calculated as,

\[
x(z) = \frac{q''\pi D(z - z_0)}{\dot{m}h_{fg}}.
\]  

Uncertainty estimates for the various sensors is presented in Section 3.4. Typical uncertainty in the two-phase heat transfer coefficient was estimated to be 9.9%.
5.2.2 Measurement Scope

Measurements were carried out at three flowrates, two low flowrates in the laminar regime and a high flowrate in the turbulent regime. For the low flowrates, the mass flux was nominally 770 kg/(m$^2$·s) and 1040 kg/(m$^2$·s), with inlet subcooling ranging from 7 °C to 18 °C. For the high flowrate, the mass flux was nominally 2050 kg/(m$^2$·s) and the inlet subcooling ranged from 5 °C to 18 °C. Details of the conditions investigated are listed in Table 5.2.

At the high mass flux, onset of nucleate boiling characteristics such as increased pressure drop and large wall temperature drop, were not observed for the heat flux range covered. Additionally, with increased heat flux the exit fluid temperature could sustain temperatures higher than the saturation temperature for small time periods before dropping to the saturation temperature. With a drop in the exit fluid temperature to its saturation temperature, vapour was observed in the tubing downstream of the test section. However, the wall temperature and pressure drop characteristics were virtually unchanged. Figures 5.13 (a-d) compares the wall temperature and pressure drop characteristics at different levels of inlet subcooling for both the high and low mass fluxes. These figures show the differences in initiating boiling at a mass flux of 770 kg/(m$^2$·s) and 2050 kg/(m$^2$·s). It is important to note that the temperature for these plots is measured by the thermocouple bonded to the tube wall close to the exit. This temperature is not representative of the actual wall temperature, due to the thermal resistance of the electrically resistive epoxy used. However, it is presented since it effectively highlights the wall temperature trend. The TLC’s are not
Table 5.2: Tube boiling measurement conditions.

<table>
<thead>
<tr>
<th>Volumetric Flow Rate (ml/min)</th>
<th>Descriptor</th>
<th>Nominal mass flux (kg/(m² · s))</th>
<th>$T_{in}$(°C)</th>
<th>$\Delta T_{sub}$(°C)</th>
<th>$P_{out}$ (bar)</th>
<th>$N_{conf}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>68</td>
<td>$\Delta T_{sub,low}$</td>
<td>2050</td>
<td>59</td>
<td>5.2</td>
<td>1.25</td>
<td>0.66</td>
</tr>
<tr>
<td></td>
<td>$\Delta T_{sub,med}$</td>
<td></td>
<td>56</td>
<td>9.9</td>
<td>1.33</td>
<td>0.55</td>
</tr>
<tr>
<td></td>
<td>$\Delta T_{sub,high}$</td>
<td></td>
<td>46</td>
<td>18.4</td>
<td>1.26</td>
<td>0.70</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>48</td>
<td>16.3</td>
<td>1.26</td>
<td>0.69</td>
</tr>
<tr>
<td>26</td>
<td>$\Delta T_{sub,low}$</td>
<td>770</td>
<td>56</td>
<td>6.7</td>
<td>1.20</td>
<td>0.55</td>
</tr>
<tr>
<td></td>
<td>$\Delta T_{sub,high}$</td>
<td></td>
<td>48</td>
<td>17.8</td>
<td>1.32</td>
<td>0.69</td>
</tr>
<tr>
<td>35</td>
<td>$\Delta T_{sub,low}$</td>
<td>1040</td>
<td>55.5</td>
<td>7.8</td>
<td>1.21</td>
<td>0.64</td>
</tr>
</tbody>
</table>
Figure 5.13: (a,c) Heat flux versus wall temperature, measured by embedded thermocouple. (b,d) Pressure drop versus heat flux.
presented at this point since the wall temperature was not always within their active range. In Figure 5.13 (a), for the low mass flux, the wall temperature increases in the single-phase regime until about 30 kW/m² when boiling is initiated. At this point, there is a large wall temperature drop. With increased heat flux the wall temperature rises slightly with increased heat flux. At a high heat flux of about 42 kW/m² for the high subcooling case and 80 kW/m² for the low subcooling case, further increase in heat flux causes a drastic wall temperature rise. The wall temperature through a large part of the boiling region fluctuates and is likely the reason for some of the scatter in the data at lower heat flux. Although the cause of the large temperature rise appears to be due to a critical heat flux condition, it needs to be investigated further as the trend is counter to the trends observed in many other works. That is, if it is a critical heat flux condition, the lower critical heat flux should be observed in the lower subcooled case (Bowers, 1994). Figure 5.13 (b) shows the pressure drop characteristics at the low mass flux case. The pressure drop slightly decreases with increasing heat flux in the single phase regime, as would be expected. At onset of boiling, the pressure drop increases, though this increase is more significant for the low inlet subcooling case.

Figure 5.13 (c) shows the wall temperature characteristics for the highest mass flux case. Unlike the low mass flux case, increased heat flux continually increases the wall temperature at an almost constant rate. For the case of high inlet subcooling, a threshold heat flux was observed, at which point further increase in heat flux caused a wall temperature jump. As the setup was not configured for critical heat flux experiments, power to the system was immediately shut down to prevent damage to
the tube. Figure 5.13 (d) shows the wall pressure drop characteristics for the high mass flux case. Similar to what is observed for single phase flows, the pressure drop slightly decreases with increasing heat flux for the range covered. For the high inlet subcooling case at a threshold heat flux of about 138 kW/m², a significant rise in pressure drop is observed, signifying two-phase or superheated flow.

For the wall thermocouple temperature measurements presented above, a wall temperature of 87 °C signifies a tube outer wall temperature greater than 100 °C. This high temperature caused vaporization of the TLC coating and long periods at this temperature would completely eliminate the TLC coating. This highlights one of the two drawbacks in utilizing un-encapsulated TLC's for boiling measurements. That is, when using the material in conditions that require high wall superheats to initiate boiling, there is a potential that the wall temperature can be too high for the TLC material and thus erode or completely eliminate the coating. The second drawback is related to effectively utilizing the TLC active range. The measurements were restricted to a system pressure that would provide wall temperature conditions in the active TLC range during boiling. This range was determined through a number of trials and is not an effective approach for a full parametric study.

From the above, boiling was only initiated at the low mass flux conditions. Detailed experimental measurement and analysis are therefore carried out at these conditions and more specifically at the low inlet subcooling condition. The remainder of this section provides results from these test conditions of low mass flux and low inlet subcooling.
5.2.3 Boiling Onset at Low Mass Flux

Flow boiling was initiated by incrementing the applied heat flux. While in single phase, the wall temperature constantly rose and this was observed with the TLC color changing from red to blue at a given location. With further increase of heat flux downstream locations of the tube past the clearing point of the TLC and wall temperature monitoring was restricted to the thermocouple embedded close to the tube end. The demarcation to the clearing point was a long smooth color change implying a low temperature gradient. With further increase, the TLC clearing point moved further upstream, signifying an increased wall temperature. No bubbles or vapour were observed in the tubing downstream of the test section, or in the outlet instrumentation chamber. At a threshold heat flux, a fast moving front with a large temperature gradient was observed to travel towards the exit of the tube. Additionally, just prior, the TLC coating was observed to partially vaporize suggesting an instantaneous wall temperature rise. A sample of this fast moving front is shown in Figure 5.14. The red region is the cooler temperature region, whereas the black is the higher temperature region, since it is past the TLC clearing point. The velocity of the front was estimated to be approximately 1.4 m/s to 1.7 m/s, which is higher than the all liquid fluid velocity of 0.58 m/s and much lower than the all vapour velocity of 47.95 m/s. Immediately following the front, the wall temperature is significantly lower than under single-phase conditions, and remains relatively constant over the two-phase length. The location of this front initiation was not identified exactly on the tube, but was approximately 1/5th to 1/4 downstream of the inlet. The large
Figure 5.14: TLC wall temperature visualization over time during boiling onset, 79 to 88 z/D, $q'' = 30.4$ kW/m², $G = 770$ kg/(m²·s). Red is lower temperature region, and flow is from left to right.
temperature spike at onset suggests a transition from all liquid to significant vapour flow. The moving wall temperature gradient further suggests that this condition is unstable, causing an immediate purging of the high vapour region by a two-phase region with better heat transfer characteristics. This onset phenomenon was found to be repeatable over the same experimental run by turning on and off the power to the tube, and was also observed on different experimental days. Its occurrence is likely function of the tubes inner surface, the tube diameter and the working fluid characteristics. Further studies are recommended to clearly identify this phenomenon.

5.2.4 Wall Temperature Visualization and Measurement

During Flow Boiling

With boiling initiated at a low heat flux, the wall temperature remained stable. The TLC color was at the red-start of the bandwidth and hence it was difficult to obtain measurements at this condition. With an increase in heat flux the wall temperature slightly increased and the TLC color response was clearly in the red range. With further increase in heat flux wall temperature oscillations were observed. Far downstream the oscillations were in the red range signifying a low wall temperature and far upstream they were in the yellow to green range signifying a higher wall temperature. With further increase in heat flux, a large increasing wall temperature gradient was observed downstream, past the clearing point of the TLC in the form of a front. This front would periodically oscillate and its color would fluctuate.

Figure 5.15 presents sample data of these oscillations taken at a low heat flux
Figure 5.15: TLC wall temperature time trace - 3 separate locations imaged at separate time instants, $q'' = 50.7$ kW/m$^2$, $G = 770$ kg/(m$^2$ · s).
and at a far downstream location of 60 z/D to 79 z/D. The time traces are produced from considering the median value of a region of 20 pixels in the streamwise direction and 72 pixels in the circumferential direction. Prior to this the images are filtered to exclude pixels that appear white or black based on consideration of their intensity, hue and saturation. The wall temperature fluctuations in Figure 5.15 are observed to be semi-periodic with amplitudes ranging from 1 °C to 5 °C. Based on this figure and other data, it was not clear the influence of variable distance on the amplitude at a given heat flux. Figure 5.16 shows a frequency spectrum of the signal typical of a variety of far downstream locations along the tube. The fundamental frequency is approximately 10 Hz. With the temporal resolution available, a constant frequency was observed across the length of the tube for a fixed heat flux. Figure 5.17 shows a second sample of the wall temperature fluctuations at low heat flux for a location close to the inlet (18 z/D) and close to the outlet (83 z/D). The mean wall temperature decreases by about 2 °C, but the frequency and amplitude of the oscillations remain similar. The influence of the heat flux on the wall temperature amplitude was not clear. Figure 5.18 shows a sample of data taken at 8 z/D with increasing heat flux. The amplitude clearly increases from 64.6 kW/m² to 73.5 kW/m², however, with further increase, the amplitude slightly decreases then remains at the same magnitude.

As the heat flux was increased the wall temperature fluctuations appeared to increase in frequency based on real-time visual inspection of the TLC color change. However, frequency spectrums of signals captured during these conditions suggest a decrease in frequency to about 6 Hz to 8 Hz. This decreasing trend however, may be caused by aliasing due to the limitation in the frame capture rate of approximately
Figure 5.16: Typical power spectrum of oscillating wall temperature as measured by TLC, for corresponding waveforms in Figure 5.15, $q'' = 50.7$ kW/m², $G = 770$ kg/(m²·s).
Figure 5.17: Wall temperature time trace at a far downstream (83 z/D) and an upstream (18 z/D) location, $q'' = 54.1$ kW/m$^2$. 
Figure 5.18: Wall temperature time trace at a fixed location with increasing heat flux.
30 Hz. Image capture at higher frequencies is therefore required to ascertain the correct trend. Figure 5.19 shows a sample of data at two different heat fluxes at the same location of 32 z/D. The frequency spectrum show a decrease with increased heat flux. At this high heat flux, the response still remains periodic, however, at the same condition far downstream, there is a significant rise in wall temperature and a large fluctuating wall temperature gradient front is observed. Figure 5.20 shows a sample of this front over a time period of 0.3 s. From 0 to 0.167 s wall temperature fluctuations are observed within the range of the TLC. The wall temperature gradient at this point is mild. From 0.2 s to 0.3 s a rapid rise in temperature is observed as the clearing point of the TLC is exceeded. This signifies a large temperature gradient. These conditions are observed at the point where the wall temperature begins to increase after the constant wall temperature region in Figure 5.13. With a further increase in heat flux the large temperature gradient front continues to move further upstream. Figure 5.21 shows a sample of the temperature gradient front at a higher heat flux and thus moved further upstream. For the duration of the time period it remains at the same location, however, a few milliseconds later its position may shift and return to the location. Upstream of the front location, however, change in the color of the TLC is observed showing that the wall temperature is fluctuating at a lower level. This large temperature gradient front suggests conditions of partial dryout inside the tube in the vicinity of the front. An important point to note during these oscillations is that there were no oscillations in mass flux, and the fluid bulk temperature at inlet and outlet remained relatively constant. The pressure, however, fluctuated at relatively small amplitude. The low pressure drop fluctuations, as well the lack of mass flow
Figure 5.19: Wall temperature oscillations and corresponding power spectrum with increasing heat flux, taken at a distance of approximately 32 z/D.
Figure 5.20: Wall temperature fluctuations highlighting oscillatory behaviour and high wall temperature gradient front. Red is cool and blue is hot, black is high temperature past the clearing point of the TLC. Covering a time span of 0.3 s, at 78 z/D to 88 z/D and $q'' = 79$ kW/m², $G = 770$ kg/(m² · s), and flow is from left to right.
Figure 5.21: Wall temperature fluctuations highlighting oscillatory behaviour and high wall temperature gradient front. Red is cool and blue is hot, black is high temperature past the clearing point of the TLC. Covering a time span of 0.3 s, at 70 z/D to 78 z/D and $q'' = 83$ kW/m$^2$, $G = 770$ kg/(m$^2$·s), and flow is from left to right.
fluctuations, suggest that these are neither pressure drop nor density instabilities. Ding et al. (1995) presented the various dynamic instabilities that may be present in a horizontal tube. Their discussion on thermal oscillations suggests that high wall temperature fluctuations will be present. Additionally, the characteristics of the thermal oscillations are initiated at the transition from a thin liquid film to an all vapour region that may oscillate due to frequent impulses of purged vapour slugs from upstream. Although at upstream locations the wall temperature oscillations are on the order of 10 °C at maximum, at the oscillating front the wall temperature gradient is likely greater than 20 °C, which is high and therefore more representative of thermal oscillations. This suggests that the mechanism responsible for the oscillations is similar to that for thermal oscillations, whereby there exists a thin film of liquid that oscillates due to bubbles being purged at the interface. This is corroborated by the fact that just upstream of the interface the wall temperature fluctuates suggesting a passing vapour slug. These observations, however, need further verification by means such as flow visualization. Nevertheless, the advantage of a thermographic technique has been highlighted in the present measurements. The spatial correspondence of discrete fluctuating signals (pixels) over time could be observed and their relationship identified as a likely vapour liquid interface. Therefore the TLC's have inadvertently provided a means of flow visualization.
5.2.5 Two-Phase Flow Heat Transfer Coefficient

To determine the two-phase heat transfer coefficient, accurate wall and fluid bulk temperature measurements are required. As mentioned earlier, determination of the fluid bulk temperature in the two-phase region is based on a linear pressure drop assumption. Additionally, accurate determination of the zero quality location requires an effective energy balance. Figure 5.22 (a-e) presents samples of the wall temperature, as well as the calculated fluid bulk temperature at incremental levels of heat flux and at a mass flux of 770 kg/(m²·s). The wall temperature is a spanwise average of 72 pixels and has been down sampled for presentation. It is compiled from snapshots at a number of locations along the tube with each location spanning 640 pixels or approximately ten diameters. Identified in the figure is the zero quality location calculated based on an energy balance. Typical in all figures is an almost constant, but slightly decreasing wall temperature in the two-phase region. In the single phase region of Figures 5.22 (d) and (e), the wall temperature rises then falls, typical of what is seen during onset of subcooled boiling. There is scatter within the data and this is partially due to the time variance of the data. The data during high heat flux when significant wall temperature variations were present and where the liquid crystal clearing point was exceeded are not included.

Figure 5.23 presents the heat transfer coefficient at two different heat fluxes and mass fluxes as a function of quality. Similar to the wall temperature measurements, they are determined by a spanwise average of 72 pixels and down sampled for presentation. In the region leading up to the zero quality location, the heat transfer
Figure 5.22: Measured circumferentially averaged wall temperature and calculated fluid bulk temperature along tube for varying heat fluxes, $G = 770$ kg/(m$^2$ · s).
Figure 5.23: Local heat transfer coefficient as a function of quality, effect of heat flux and mass flux.
coefficient rises rapidly. With increasing heat flux, the heat transfer coefficient increases when the quality is greater than zero. The increase for a mass flux of 1043 kg/(m\(^2\) \cdot s) is not as significant for as for the lower case of 770 kg/(m\(^2\) \cdot s). With the exception of the low mass flux, high heat flux case \((q'' = 83 \text{ kW/m}^2, G = 770 \text{ kg/(m}^2 \cdot \text{s})\)), the heat transfer coefficient has a mild decrease with increasing quality. Finally, at a given heat flux the influence of mass flux is mild. For the low heat flux case, the heat transfer coefficient increases slightly with increased mass flux, whereas for the high heat flux case there is a slight decrease in the heat transfer coefficient with increased mass flux. However, at this high heat flux case, there is significant scatter in the dataset, and looking back at the wall temperature trend in Figure 5.22 (e), it is expected that the heat transfer coefficient should be displaying the same trend as the other higher mass flux dataset.

In observing the trends of the lower mass flux case \((G = 770 \text{ kg/(m}^2 \cdot \text{s})\)), these data suggest a nucleation dominated region at low quality, with the heat transfer coefficient being a strong function of the heat flux and a weak function of the mass quality. These trends were typical for the other heat flux values at this mass flux. For the higher mass flux case \((G = 1040 \text{ kg/(m}^2 \cdot \text{s})\)), the heat flux influence on the heat transfer coefficient is less significant, however, and this was typical for other heat flux cases at this mass flux. It is noted that the data obtained during the large temperature gradient fronts are not presented as they are out of range of the TLC, as well they would be accompanied with significant time variance. However, if wall temperature measurements were available they would show a significant decrease in the heat transfer coefficient in the range of quality greater than 0.35.
Figures 5.24 and 5.25 compare the measured heat transfer coefficient with three different correlations. The first correlation is given by Kandlikar (1990) and Kandlikar and Balasubramanian (2004). It is based on a variety of working fluids and working conditions, and for channel diameters ranging from 4 mm to 32 mm (Kandlikar, 1990). It has been modified from its original form to account for flow in minichannels (0.20 mm ≤ Dh ≤ 3.0 mm) in the laminar and transitional range (Kandlikar and Balasubramanian, 2004). Flow is considered in the transitional range if 1600 ≤ Re_f ≤ 3000, which are the conditions of the presented data. The correlation is,

\[ h_{tp} = \text{larger of } (h_{tp,NBD} \text{ and } h_{tp,CBD}), \quad (5.8) \]

where, NBD denotes nucleation boiling dominated and CBD denotes convection boiling dominated. Each term is,

\[ h_{tp,NBD} = 0.6683Co^{-0.2}(1 - x)^{0.8}h_f + 1058.0Bo^{0.7}(1 - x)^{0.8}F_{fl}h_f \quad (5.9a) \]

\[ h_{tp,CBD} = 1.136Co^{-0.9}(1 - x)^{0.8}h_f + 667.2Bo^{0.7}(1 - x)^{0.8}F_{fl}h_f \quad (5.9b) \]

\[ Co = \left( \frac{1 - x}{x} \right)^{0.8} \left( \frac{\rho_v}{\rho_f} \right)^{0.5} \quad (5.9c) \]

where, \( F_{fl} \) is a fluid to surface relating term and for stainless steel tubes with any fluid is 1.0 (Kandlikar and Balasubramanian, 2004). For the transitional regime, it is recommended to use a single phase heat transfer coefficient based on a linear interpolation between laminar and turbulent flow. In the present work, \( h_f \) has been
Figure 5.24: Local heat transfer coefficient, comparison with correlations at various heat fluxes, $G = 770 \text{ kg/(m}^2 \cdot \text{s)}$. 
Figure 5.25: Local heat transfer coefficient, comparison with correlations at various heat fluxes, $G = 1040 \text{ kg/(m}^2\cdot\text{s)}$. 
taken as that based on the analytical solution for laminar thermally developing flow.

The second correlation by Warrier et al. (2002) was selected as it is based on a working fluid with similar chemical structure, FC-84. However, the geometry used in the study was composed of five parallel rectangular channels of 0.75 mm hydraulic diameter. The correlation is given by,

\[
\frac{h_{tp}}{h_f} = 1 + 6.0 Bo^{1/16} \left[ -5.3 (1 - 855 Bo) \right] x^{0.65}. \tag{5.10}
\]

The single-phase liquid heat transfer coefficient \((h_f)\), is for fully developed flow, however, it was taken as the thermally developing value for the present analysis. The third correlation by Gungor and Winterton (1986) is based on a database covering a variety of fluids for channel diameters ranging from 2.95 mm to 32.00 mm, and under a variety of channel orientations. It is,

\[
h_{tp} = Eh_f + Sh_{ncb} \tag{5.11a}
\]

\[
h_{ncb} = 55P_r^{0.12} \left( -0.4343 \ln (P_r) \right)^{-0.55} M^{-0.5} \left( Q/A \right)^{0.67} \tag{5.11b}
\]

\[
E = 1 + 24,000 Bo^{1.16} + 1.37 \left( \frac{1}{X_{tt}} \right)^{0.86} \tag{5.11c}
\]

\[
S = \left( 1 + 1.15 \times 10^{-6} E^2 Re_f^{1.17} \right)^{-1} \tag{5.11d}
\]

\[
X_{tt} = \left( \frac{1 - x}{x} \right)^{0.9} \left( \frac{\rho_v}{\rho_f} \right)^{0.5} \left( \frac{\mu_f}{\mu_v} \right)^{0.1}. \tag{5.11e}
\]
The parameter $E$ is an enhancement factor of the single phase heat transfer and $S$ is a suppression factor for the pool boiling term. The single phase heat transfer coefficient ($h_f$) is defined based on the Dittus-Boelter correlation (Gungor and Winterton, 1986) for turbulent flow. However, in the present as with the previous cases, it has been modified with the thermally developing laminar flow analytical solution. Due to lack of data on vapour phase viscosity of FC-72, this term was taken as 1.0. As shown in Figures 5.24 and 5.25, both the correlation of Equations 5.8-5.9 and Equation 5.10 under predict the heat transfer coefficient for all cases of heat fluxes, though Equations 5.8-5.9 are closer in magnitude. The trends presented by both correlations however are very similar to that observed in the data of Figure 5.24 ($G = 770 \text{ kg}/(\text{m}^2 \cdot \text{s})$), showing a slight decrease in heat transfer coefficient with increasing quality. For this lower mass flux case also, the correlation by Equation 5.11 provides a good measure of the magnitude of the heat transfer coefficient for the middle range of heat fluxes, however, it over predicts and slightly under predicts the data for the low and high heat flux cases respectively. For the higher mass flux case shown in Figure 5.25 however, the correlation from Equation 5.11 consistently slightly under predicts the heat transfer coefficient. The correlation of Equations 5.8-5.9 contains a nucleate boiling dominated form ($h_{tp,NBD}$) and a convective boiling dominated form ($h_{tp,CBD}$). For the conditions in the present work it was found that the nucleate boiling dominant form of the equation prevailed for the majority of the instances, particularly at the lower mass flux case ($770 \text{ kg}/(\text{m}^2 \cdot \text{s})$). On the other hand, the correlation in Equation 5.11 contains a nucleate boiling component ($Sh_{nob}$) and a convective boiling component ($E_{hf}$) and in general had the best agreement with the results in terms of magnitude.
It was found that the total heat transfer coefficient from this correlation was dominated by the convective boiling component which accounted for 60 to 70% of its value. The lack of correspondence of the correlations confirms that additional studies are still required as has been mentioned by many other researchers. Additionally with a variety of fluids to expand the correlation versatility, as has been demonstrated by the lack of fit with present data.

5.3 Summary

Through use of un-encapsulated liquid crystals, local single phase heat transfer measurements have been performed in small diameter tubes of nominal size 0.254 mm, 0.512 mm and 1.067 mm, covering the laminar, transitional and turbulent regimes. The results demonstrate that the conventional size channel analytical solution for laminar flow is adequate for predicting the local heat transfer in thermally developing flows inside sub-millimeter size circular tubes over the range studied.

Local Nusselt numbers in the turbulent regime are shown to have a developmental length less than approximately 10 z/D, which is similar to conventionally sized tubes in this Prandtl number range. Considering uncertainties in Nusselt number correlations and the present data, both the conventional correlation by Gnielinski (1976) and the microchannel correlation by Adams et al. (1998) are adequate for predicting averaged Nusselt numbers in circular tubes with diameters ranging from 0.25 mm to 1 mm, for $2500 \leq \text{Re} \leq 10^4$ and Prandtl numbers ranging from 5.4 to 11.5. Finally, the choice of the correlation should be based on the amount of conservatism required
in the design.

The investigation of flow boiling heat transfer of FC-72 in a 1.067 mm tube through use of liquid crystal thermography has been presented. Local heat transfer data are presented for a low mass flux (770 kg/(m² · s) and 1040 kg/(m² · s)) and a low inlet subcooling, partially because boiling initiation was not observed at other conditions. At these low mass flux cases, a unique phenomenon at boiling onset was observed, whereby a large temperature gradient front moved towards the exit of the tube with a cooler wall temperature behind it. This observation was enabled due to the use of a thermographic technique for wall temperature measurement.

Under two-phase conditions the wall temperature was observed to oscillate for a wide range of heat fluxes. The amplitude ranged from 1 °C to 7 °C for a low heat flux with the frequency around 10 Hz. At high heat flux, large oscillatory wall temperature gradients were observed in the form of wall temperature gradient fronts. The nature of these large gradient oscillations suggests an oscillating partial dryout regime. Although flow visualization was not directly performed, the use of a thermographic technique has inadvertently provided a means for partial flow visualization.

Local heat transfer coefficient data were presented for a quality up to approximately 0.3. In the range of heat flux covered, the heat transfer coefficient is dependent on heat flux, but only mildly influenced by quality at a lower mass flux of 770 kg/(m² · s). At a higher mass flux of 1040 kg/(m² · s), the heat transfer coefficient is less dependent on heat flux. When compared with some relevant correlations, none can accurately predict the observed magnitude and trend over the entire range of heat fluxes.
Chapter 6

Transport Phenomena in a Cross-Linked Heat Sink

This chapter presents the flow and heat transfer characteristics in a cross-linked microchannel heat sink. The aim of the cross-links is to address the issue of flow boiling instabilities and flow maldistribution. The cross-linked geometry is similar to a standard microchannel heat sink with parallel channels, except that cross-linked paths are introduced at three locations along the channels. The purpose of the gaps is to allow fluid communication between channels at additional points other than entry or exit of the channels.
6.1 Design

For the detailed design and sizing of the cross-linked heat sink concept, a one-dimensional thermal resistance model was utilized based on a design target. Additional constraints are introduced based on manufacturing limitations. The design target is motivated by thermal control for micro/nano- spacecrafts. From Paris et al. (2002), who developed a pumped cooling loop for this application, future demands will be required to manage power densities of 25 W/cm², over a 3.5 cm² footprint area (87.5 W over the footprint), while maintaining a chip temperatures below 85 °C. The footprint area is considered in the present work as the area directly above the microchannels. The channels are to be manufactured via deep reactive ion etching (DRIE) as it allows for a wider variability of shapes. The manufacturing constraints are a silicon wafer thickness of 550 μm and a maximum DRIE etch depth of 300 μm, with an aspect ratio of width to depth not exceeding 1:50. Additional constraints are that the working fluid is single-phase water and a maximum pressure of 34.5 kPa (5 psi) is available for the entire loop.

Within the model, the heat transfer coefficient is calculated considering steady, laminar, hydrodynamically and thermally fully developed flow, in a rectangular duct with constant wall temperature at three walls and the fourth wall adiabatic. The Nusselt number, which is based on the wetted perimeter hydraulic diameter, is obtained from Shah and London (1978) in tabular form, and intermediate values are based on interpolation. The pressure drop is modeled considering only the channels, that being the heat sink channel core, the channel inlet and the channel outlet. Similar to
the heat transfer results, the flow is considered fully developed, laminar and steady. In addition, entry and exit loss factors are included.

The parameters to be determined include the channel width, and the length and width of each cross-linked section. The number of channels is determined from the channel width. From the literature, the influence of the cross-links on pressure drop and heat transfer compared to a straight channel is not clear and so a conservative approach is taken. In doing so, each length of channel section is considered a new channel and therefore has an inlet and outlet pressure loss attributed. For the thermal component, the area comprising the gap space is considered to have a heat transfer coefficient equivalent to 50% of the main passages. Results from the analysis are shown in Figure 6.1. Comparing the pressure drop (Fig. 6.1 (a)) and maximum surface temperature (Fig. 6.1 (b)) for different cross-link widths and number of cross-links, only very slight differences are observed. In both cases, however, minimizing the number of cross-links results in better performance. As an initial iteration for the research program, the number of cross-links has been set to three, with the width of the cross-links equal to the determined channel width.

Figure 6.1 (c) and 6.1 (d) show the resulting pressure and temperature isotherms as a function of the channel width and flowrate, at heat fluxes of 25 W/cm² and 50 W/cm². At the lower heat flux, a wide range of channel widths meet the design target of 85 °C with a pressure drop less than 5 psi. At the higher heat flux, channel widths less than 300 μm with flowrates greater than 100 ml/min meet the target. Although the pressure penalty with increasing flowrate is not significant, the benefit gained for the target isotherm of 85 °C is limited for widths greater than 300 μm. A channel

137
(a) Pressure drop versus cross-link, width, 30 ml/min.

(b) Maximum surface temperature versus cross-link width, 30 ml/min.

(c) Channel width versus flow rate, $q'' = 25$ W/cm².

(d) Channel width versus flow rate, $q'' = 50$ W/cm².

Figure 6.1: Cross-link concept analysis results.
width of 250 μm is selected as it can comfortably meet the design target and future requirements.

The finalized design is presented in Figure 6.2. The heat sinks were fabricated at the Washington Technology Center’s Microfabrication Lab. They are fabricated from silicon wafers 500 μm thick. A silicon dioxide (SiO₂) layer is deposited on the backside for electrical isolation. A serpentine platinum heating element, 1 mm wide and 186 mm long, is deposited on the backside of the channel base to simulate the heated chip. The heating element has been designed for a nominal resistance of 90 Ω, with a potential applied serially through the entire length of the serpentine. However, the actual resistance was found to be approximately 16 times greater. To account for this, the potential is applied in parallel through the 11 strips of the serpentine. A Pyrex wafer (Corning 7740) of thickness 350 μm seals the chamber and is bonded to the silicon base piece containing the channels. The cover has eight 1 mm diameter holes etched for flow inlet and outlet.

6.2 Heat Sink Packaging

6.2.1 Packaging Method I

Figure 6.3 shows the first heat sink support with the inlet and outlet chambers, and the pressure and temperature taps. The support is fabricated from Teflon due to its high temperature limits, as well as its low thermal conductivity and high electrical resistance. The chips were originally intended to be directly mounted to the Teflon
Figure 6.2: Cross-linked heat sink.
Figure 6.3: Heat sink support I.
base piece with gaskets for sealing. However, during sealing attempts the chips easily broke, if misaligned when applying the required compressive loading for the gaskets. Rather then, a rigid midplate with two slots for fluid transfer is introduced for the chips to be permanently bonded onto, before transferring to the Teflon base. This midplate, which is easily handled, is sealed to the Teflon base using a silicone based adhesive with a high temperature limit, or gaskets. The midplate is transparent to verify that the sealant between the chips and the midplate does not leak into the heat sink port holes. Clear polycarbonate is selected due to its higher temperature limits. The vertical distance from the sensor ports to the heat sinks is 21.2 mm.

6.2.2 Packaging Method II

The second support, shown in Figure 6.4, was designed to allow for flow visualization and micro-PIV experiments. All parts are manufactured from transparent polycarbonate to provide sufficient temperature limits with optical access. A top plate, 3.18 mm (1/8" sheet) thick, supports the heat sink. A second base plate, also 3.18 mm thick, has a 1.59 mm slot machined to allow fluid transport from the pressure / temperature measurement ports to and from the heat sink. Thin plates are utilized to accommodate the working distance of the microscope objectives available. The plates and the instrument chambers are bonded together using a polycarbonate cement (SC-325 Cadillac Plastic and Chemical Company) and a transparent UV curing adhesive recommended for elevated temperatures with moisture content. The heat sinks are bonded to the top plate using the same UV curing adhesive.
Figure 6.4: Heat sink support II, millimeters [inches].
6.3 Single Phase Flow

6.3.1 Methods and Data Reduction

Pressure Drop

The pressure taps are located at some distance from the channel inlets and exits, and so contain additional pressure components due to the inlet and outlet chamber circuitry. These are accounted for through loss factors \( K_i \) for area changes and bends and are obtained from Munson et al. (2002). Additional losses will be incurred due to the channel entry and exit and flow acceleration. These are estimated using the methodology outlined by Kays and London (1964) for flows in compact heat exchangers. Finally, as the flow at entry is not fully developed, this is also considered. In summary the pressure drop in the channels \( \Delta P_{chn} \) is obtained as follows,

\[
\Delta P_{\text{Measured}} = \Delta P_{chn} + \left[ \xi_{in} + 2 \left( \frac{\rho_{in}}{\rho_{out}} - 1 \right) - \frac{\rho_{in}}{\rho_{out}} \xi_{out} + K_{\infty} - \frac{(\rho u^2)_{chn, in}}{2} + \text{Minor Losses} \right]
\]

(6.1)

where,

\[
\xi_{in} = K_e + 1 - \sigma_{por}^2
\]

(6.2a)

\[
\xi_{out} = 1 - \sigma_{por}^2 - K_e
\]

(6.2b)
and

$$\sigma_{por} = \frac{N_{chn} W_{chn} H_{chn}}{W_{total} H_{chn}} \quad (6.3)$$

and where,

$$Minor\ Losses = \sum_j K_j \frac{(\rho u^2)_j}{2} \quad (6.4)$$

The loss factors utilized in the above are from experiments in macro size configurations. These may not be fully appropriate for all the present conditions, particularly for the channel inlet and exit losses. Nevertheless, they are used as there are currently no correlations for micro configurations readily available. For comparison with theoretical predictions, the pressure drop across the channels may be obtained from

$$\Delta P_{chn} = 4 f \frac{L}{D_h \rho_{mean}} \frac{(\rho u^2)_{chn,in}}{2} \quad (6.5)$$

The friction factor is obtained from Shah and London (1978) for a rectangular duct of fixed aspect ratio as,

$$f Re = 24 \left[ 1 - 1.3553\alpha + 1.94673\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5 \right]. \quad (6.6)$$

The aspect ratio is denoted by $\alpha$ and $f Re$ was calculated to be approximately 14.25 for an aspect ratio of 0.95.

For accurate prediction of the pressure drop, accurate determination of the channel dimensions is required, particularly the cross-section. This may be done through non-destructive and destructive optical measurement, and both are utilized in the present
investigations. In the first case, the width is obtained from a top view image and
the depth is obtained from focusing on the top and bottom surfaces. This approach
can be expected to provide good measurement of the width, however, the depth
measurement will be slightly subjective. The alternative approach is obtaining the
dimensions from cross-sections of the heat sink. However, this approach is destructive
and therefore not favorable. Both approaches were explored and were found to be
in good agreement with each other. Further, significant discrepancies were observed
between the design and fabricated heat sink dimensions. A comparison of the design
and measured dimensions is shown in Table 6.1. Inspection of the channel cross-
sections showed that they are not purely rectangular as assumed in the prediction
calculations. At the inlets and exits, the channels have a slight flare as shown in the
heat sink top view of Figure 6.5 (a). Due to this flare, the cross sections at the inlet
and exits are trapezoidal with taper angles as low as 85 degrees. This characteristic is
depicted in Figure 6.5 (b). In the channel core however, the channels are found to not
have a strongly tapered angle, but do have curved edges. Figure 6.5 (c) shows a cross
section of the core and shows the locations used for approximating the dimensions.
These deviations in fabrication have been noted by some other researchers such as
Steinke and Kandlikar (2005b), and the present inspection shows the importance of
determining the 'as fabricated' channel dimensions. In addition, it is preferable to
perform such measurements prior to sealing the heat sink so that destructive testing
does not have to be performed, unless the sealing process has an influence on the final
dimensions.

146
Table 6.1: Comparison of design and measured parameters.

<table>
<thead>
<tr>
<th></th>
<th>Design</th>
<th>Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel width</td>
<td>242 ( \mu \text{m} )</td>
<td>269 ( \mu \text{m} )</td>
</tr>
<tr>
<td>Channel depth</td>
<td>300 ( \mu \text{m} )</td>
<td>283 ( \mu \text{m} )</td>
</tr>
<tr>
<td>Cross-link width</td>
<td>200 ( \mu \text{m} )</td>
<td>269 ( \mu \text{m} )</td>
</tr>
</tbody>
</table>
Figure 6.5: Channel inspection. (a) Tapered edges, (b) Trapezoidal cross-section at entry/exit, (c) Cross-section in core.
Heat Transfer

The TLC material used has a nominal red start of 40 °C with a bandwidth of 10 °C. Image capture covers a region of 9.94 mm × 7.45 mm yielding a resolution of 10 μm/pixel. To map the entire heat sink, three locations are traversed in the spanwise direction. To enable automated image pre-processing and presentation, regions of the image not active had their pixel values set to zero. This is done by applying a binary filter with the regions marked up manually. On completion, the images from a single case at all four locations are joined in an automated procedure. Where there is overlap, the average pixel value is used. Figure 6.6 shows sample pre-processed TLC mappings of the heated surface. The locations where the image is joined can be observed at ~28 and ~52 y/Dh for some cases and is due to minor error in traverse. The TLC map shows some non-uniformity in the wall temperature. There is a wavy profile in the spanwise direction, and this is due to the heating elements underneath. The color response in Figure 6.6 is highly vibrant when compared to measurements on the microtubes (e.g. Fig. 5.14). This is primarily due to the use of a cover in direct contact with the TLC (glass slide) and also because no overheating of the coating occurred to vaporize it.

Evaluation of the local heat transfer coefficient is through assuming the heat sink to behave as a one-dimensional fin with insulated ends. Such an approach has been presented by Qu and Mudawar (2003), and a similar approach is utilized as follows:

\[
q''W_{cell}L_{plan} = h (T_{w,i} - T_b) (W_{chn} + 2\eta H_{chn}) (4L_{chn,sec})
\]  

(6.7)
(a) $Re = 210$, $q'' = 3.11\text{W/cm}^2$.

(b) $Re = 563$, $q'' = 2.38\text{W/cm}^2$.

Figure 6.6: Preprocessed heat sink TLC thermal maps, consist of 3 images joined and binary filtered.
\[ \eta = \frac{\tanh(mH_{chn})}{mH_{chn}} \quad (6.8) \]

\[ m = \sqrt{\frac{h}{k_{St}W_{w/2}}} \quad (6.9) \]

\[ W_{cell} = W_{chn} + 2\left(W_{w/2}\right) \quad (6.10) \]

A unit cell \((W_{cell}/2)\) consists of a single channel with half the wall thickness on either side \((W_{w/2})\). The above neglects the heat transfer contribution within the cross-link paths and is utilized out of convenience. The heat transfer coefficient is evaluated iteratively to an accuracy of \(1\times10^{-3}\). The inner wall temperature is obtained from a one-dimensional heat transfer approximation as follows,

\[ T_{w,i} = T_{w,o} - \frac{q''(\delta_{St} - H_{chn})}{k_{St}} \quad (6.11) \]

Energy losses are found to be at maximum 5%, but in most cases less than 1% and hence deemed to be negligible. The power to the fluid is obtained from the fluid enthalpy rise given by,

\[ q'' = \frac{Q_{vol}(\rho_{out}C_{pout}T_{out} - \rho_{in}C_{pin}T_{in})}{A_{plan}} \quad (6.12) \]

TLC temperature measurement uncertainties are estimated to range from \(\pm1.1^\circ C\) to \(\pm1.5^\circ C\). Typical uncertainty in the Nusselt number are estimated at 9%.
6.3.2 Pressure Drop Measurements

Pressure drop measurements are carried out with both air and water in a flow range of 1 to 5 standard LPM and 10 to 200 ml/min, respectively. These measurements were carried out with the support system in Figure 6.3. Initial liquid pressure drop results showed a significant deviation from theoretical predictions at low Reynolds numbers. Comparison with the air data suggested that the deviations are not likely due to channel dimensions or sensors. A physical difference between the water and air, is the existence of surface tension in the case of water. Some recent works have shown that the congestion of a bubble in microchannels can pose a significant increase on the pressure drop (Kim et al., 2002). If this is indeed the case, methods to alleviate it are through applying a high liquid pressure, vibration of the channels, or a deaerating technique such as heating the flow. The chosen approach was to run at high flowrate to increase the pressure, with the fluid at an elevated temperature. Figure 6.7 compares the results before and after deaerating the flow. The large difference in pressure drop results before and after, confirm that in large, the deviation from theoretical is due to bubble congestion in the channels. Visualization would have also verified this, however in cases as in the present where this was not possible, the above procedure proved to be a valuable approach.

Figure 6.8 compares the water and air friction factors with the theoretical predictions. Good agreement is observed in both cases with a deviating trend in the lower Reynolds number ranges. In case of the air measurements, the deviation is likely due to the sensor accuracy at the low flowrates. In case of the water measurements, the
Figure 6.7: Ratio of experimental to theoretical friction factor versus Reynolds number, before and after rapid heating at high flowrate, STR01.
deviation is likely a combination of the flowrate sensor and the pressure drop sensor uncertainties, while in their low ranges. Although the air and water show differences between each other, this is still close to their uncertainty limits. Figure 6.9 compares the difference between measured and predicted friction factors from a number of microchannel studies (Steinke and Kandlikar, 2005a,b; Harms et al., 1999; Tuckerman, 1984). All studies used parallel silicon channels flowing water and accounted for entrance and exit losses in a manner similar to what has been presented. The present measurements are found to be within the range of those obtained by other researchers. At higher Reynolds numbers there is a consistent trend of an increasing deviation from predictions. This is likely due the loss factors not being fully appropriate for parallel microchannel configurations. Future works should look into obtaining loss factors appropriate for microgeometries. Figures 6.10 (a) and (b) compare the pressure drop characteristics between the standard and cross-linked heat sinks. The difference in pressure drop is observed to be negligible over the flow range investigated and no discernible benefit or detriment of the cross-linked scheme is observed.

6.3.3 Local Heat Transfer Measurements

Heat transfer measurements were carried out using FC-72, at Reynolds numbers ranging from 142 to 559, using the support system of Figure 6.4. Successful measurements were obtained only with the cross-linked heat sink, due to leakage and fracture of the standard heat sinks. Local wall temperatures and Nusselt values are presented in Figures 6.11 to 6.14. Also presented in the figures are the calculated local fluid bulk
Figure 6.9: Ratio of experimental to theoretical friction factor versus Reynolds number, comparison with other researchers flowing water in parallel silicon microchannels.
Figure 6.10: Pressure drop versus flowrate for standard versus cross-linked microchannel heat sinks flowing water and air.
temperatures and the approximate location of the cross-links. The results are spanwise averages centered at 17.6 y/D_h, 35.3 y/D_h and 52.9 y/D_h. They are averaged over 250 pixels which corresponds to 1/5th of the height. Negligible differences are observed in the wall temperatures and Nusselt values between the different spanwise locations considered. For the Reynolds range covered, the wall temperature rises non-linearly and peaks just after the second cross-link. In all cases, a slight drop in wall temperature is observed at the third cross-link. Finally, a slight change in slope is observed at the first cross-link. Overall, the cross-link influence on heat transfer for the present configuration and Reynolds number range is observed, however it is minimal. Xu et al. (2005) observed a dramatic change in their averaged streamwise wall temperature and Nusselt values from their experiments with cross-linked triangular channels. This was attributed to a re-developing thermal boundary layer. Their cross-link widths, however, were approximately three times wider than the channel width and their configuration included five cross-link paths. Thermal boundary layer re-development is similarly the likely cause of the observed inflections in temperature and Nusselt number profiles. However, given the narrower cross-link widths in the present work (cross-link width ≈ channel width), minimal influences are observed.

Figure 6.15 compares the averaged streamwise Nusselt number based on the central region (35.3 y/D_h), with the theoretical solution for a 4-wall heated channel with constant heat flux and fully developed flow (Shah and London, 1978). This solution assumes hydrodynamically fully developed flow. From this figure, it is observed that the results correlate similarly with the non-dimensional parameters of laminar thermally developing flows. The Nusselt number trend, however, has inflections caused
Figure 6.11: Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 142, INT02 flowing FC-72.
Figure 6.12: Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 210, INT02 flowing FC-72.
Figure 6.13: Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 307, INT02 flowing FC-72.
Figure 6.14: Local wall temperatures and Nusselt numbers at various spanwise locations, Re = 559, INT02 flowing FC-72.
Figure 6.15: Local Nusselt numbers versus non-dimensional streamwise distance at various Reynolds numbers, INT02 flowing
by the cross-links.

6.4 Flow Boiling

The prevalence of boiling instabilities in parallel microchannel systems was discussed in Section 2.2.4. Compared to conventional size channels, understanding of their mechanisms and characteristics is still in its infancy. Additionally, boiling oscillations in literature have only been presented for straight, standard, parallel microchannel systems. The following presents characteristics of boiling instabilities in a cross-linked microchannel heat sink and a standard microchannel heat sink, to better understand their mechanisms. The aim of the cross-links is to interrupt any rapidly expanding bubble and prevent it from engulfing the entire channel length up to the inlet.

6.4.1 Procedure and Data Reduction

Flow boiling measurements were performed using distilled water with the first support system (Fig 6.3). Before commencing experiments, the fluid is degassed as outlined is Section 3.1.1. After this, the desired flowrate is set and the preheater is adjusted to obtain the inlet temperature setpoint. Data is obtained at incremental levels of heat flux. Flow boiling measurements for the standard heat sink (STR02) have been obtained at flowrates of 20 to 50 ml/min (91 to 228 kg/(m²·s)) and inlet temperatures of 70 °C and 80 °C. For the cross-linked design (INT01), however, only one flowrate and inlet temperature is investigated due to difficulties with leakage in the setup.
Power to the fluid under single phase conditions is obtained from an energy balance as shown in Equation 6.12. During flow boiling, the heat flux was obtained from the heater power generated and corrected for losses to the environment using a power ratio factor, $\gamma$. This results in,

$$q'' = \frac{(VI) \cdot \gamma}{A_{heated}}.$$  \hspace{1cm} (6.13)

Ideally, the power ratio factor, $\gamma$, may be estimated from an energy balance through estimation of convective losses to the environment and axial conduction losses, as was shown by Hetsroni et al. (2005a). However, this requires measurement of the outer wall temperature which was not available for the boiling measurements. To alleviate this, the present work estimated this factor by combining Equations 6.12 and 6.13 under single phase conditions. For each flowrate, as the heat flux was increased, the power ratio factor reached a limiting value prior to boiling. This limiting value, which ranged from 0.85 to 0.97 for the various flowrates, was taken as the power ratio factor under boiling conditions.

Uncertainty estimates for the various sensors are presented in Section 3.4. Uncertainty estimates for derived parameters are presented in Table 6.2.

### 6.4.2 Flow Boiling Onset and Limits

The typical inlet pressure trend leading up to two-phase flow is shown in Figure 6.16 for an inlet temperature of 70 °C at a flowrate of 30 ml/min. After boiling occurred, the inlet pressure would rise slightly in some cases, due to increased vapor generation.
Table 6.2: Uncertainty estimates for derived parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_h$</td>
<td>5.5 $\mu$m</td>
</tr>
<tr>
<td>$G$</td>
<td>4.8 - 12.3%</td>
</tr>
<tr>
<td>$q''$</td>
<td>3.4 - 6.2%</td>
</tr>
<tr>
<td>$Re$</td>
<td>7.4 - 27.5%</td>
</tr>
<tr>
<td>$Bo$</td>
<td>7.1 - 13.4%</td>
</tr>
<tr>
<td>$We_D$</td>
<td>12.4 - 24.0%</td>
</tr>
<tr>
<td>$We_L$</td>
<td>12.2 - 23.9%</td>
</tr>
<tr>
<td>$N_{sub}$</td>
<td>2.4 - 4.2%</td>
</tr>
<tr>
<td>$f^*$</td>
<td>0.4 - 2.1%</td>
</tr>
<tr>
<td>$P^*$</td>
<td>1.7 - 4.7%</td>
</tr>
<tr>
<td>$T^*$</td>
<td>5.3 - 21.6%</td>
</tr>
</tbody>
</table>
Figure 6.16: Inlet pressure versus applied power, $T_{in} = 70$ °C, $Q_{vol} = 30$ ml/min.
until instabilities occurred. Figure 6.16 shows that for degassed fluids, the heat flux at which instabilities occur is similar for both cases, at approximately 7.9 W/cm². However, for a non-degassed fluid as shown for the cross-linked design, the onset of flow instabilities occurs earlier, and at approximately 66% of the heat flux for the degassed fluid. Early onset due to a non-degassed fluid has been observed by prior investigations, and this highlights the importance of utilizing a well prepared fluid for predictability Steinke and Kandlikar (2004b). Additionally, this latter point has recently been reiterated by Chen and Garimella (2005) who investigated flow boiling of a degassed and non-degassed dielectric fluid in a microchannel heat sink.

The demarcation between relatively stable flow boiling and large boiling oscillations is sometimes considered the onset of premature critical heat flux, due to the uncertain flow characteristics under such conditions. With such an assumption, the approximate heat flux demarcating stable to unstable flow boiling is plotted in Figure 6.17 in non-dimensional form, with the Boiling number (Bo) dependent on the Weber number (We). Identification of the onset of instability was based on monitoring the fluctuations of flow parameters as well as observation of the flow in the tubing downstream. The data are from the STR02 heat sink at inlet temperatures of 70 °C and 80 °C and are compared with proposed critical heat flux correlations from both microchannel (Qu and Mudawar, 2004) and conventional channel (Katto, 1981) flows. These correlations are listed in Table 6.3.

Fluid properties are taken as the saturation temperature at the exit of the system, which is taken as 100 °C. The correlation by Katto (1981) is based on conventional size
Figure 6.17: Limiting Boiling number of STR02 heat sink for different inlet temperatures and compared to correlations.
Table 6.3: Critical heat flux correlations.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Conditions</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conventional</td>
<td>Size - rectangular ducts</td>
<td>( q''<em>{crit} = q''</em>{crit0} \left(1 + K \frac{\Delta h_{sub,m}}{h_f} \right) )</td>
</tr>
<tr>
<td>Channels</td>
<td>- water and R113</td>
<td>( q''<em>{crit01} = 0.25 (Gr</em>{fg})(\frac{1}{L_{ch}d_{heated}}) )</td>
</tr>
<tr>
<td>Katto (1981)</td>
<td>- ( D_h \approx 2.1 \text{ mm} - 20.5 \text{ mm single wall heated or }</td>
<td>( q''<em>{crit02} = C (Gr</em>{fg}) W_{eL}^{0.043} \left(\frac{1}{L_{ch}d_{heated}}\right) )</td>
</tr>
<tr>
<td></td>
<td>double opposite wall heated</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>( q''<em>{crit03} = 0.15 (Gr</em>{fg}) \left(\frac{\rho_f}{\rho_f}\right)^{0.133} \left(\frac{1}{1+0.0077L_{ch}d_{heated}}\right) W_{eL}^{1/3} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>( q''<em>{crit04} = 0.26 (Gr</em>{fg}) \left(\frac{\rho_f}{\rho_f}\right)^{0.133} \left(\frac{1}{(L_{ch}d_{heated})^{1.71}}\right) W_{eL}^{0.043} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>( C = 0.25 ) for ( \frac{L_{ch}}{d_{heated}} &lt; 50 ), ( C = 0.34 ) for ( \frac{L_{ch}}{d_{heated}} &gt; 50 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>( K_1 = 1; K_2 = 0.261 \left(\frac{L_{ch}}{d_{heated}}\right); K_3 = \frac{0.556(0.0308+L_{ch}/d_{heated})}{\left(\frac{\rho_f}{\rho_f}\right)^{0.111}} W_{eL}^{1/3} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Similar to Qu and Mudawar (2004) the coefficients are determined as follows,</td>
</tr>
<tr>
<td></td>
<td></td>
<td>When ( q''<em>{crit01} &lt; q''</em>{crit02} ), ( q''<em>{crit0} = q''</em>{crit01} ), ( K = K_1 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>When ( q''<em>{crit01} &gt; q''</em>{crit02} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>if ( q''<em>{crit02} &lt; q''</em>{crit03} ), ( q''<em>{crit0} = q''</em>{crit02} ), ( K = K_2 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>if ( q''<em>{crit02} &gt; q''</em>{crit03} ) and ( q''<em>{crit03} &lt; q''</em>{crit04} ), ( q''<em>{crit0} = q''</em>{crit03} ), ( K = K_3 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>if ( q''<em>{crit03} &gt; q''</em>{crit04} ), ( q''<em>{crit0} = q''</em>{crit04} )</td>
</tr>
<tr>
<td>Microchannels</td>
<td>- rectangular and circular parallel channels</td>
<td>( q''<em>{crit} = 33.43 \left(\frac{\rho_f}{\rho_f}\right)^{1.71} W</em>{eL}^{0.21} \left(\frac{L_{ch}}{d_{heated}}\right)^{-0.36} )</td>
</tr>
<tr>
<td>Qu and Mudawar</td>
<td>- water and R113</td>
<td></td>
</tr>
<tr>
<td>(2004)</td>
<td>- ( D_h \approx 318 \mu \text{m} - 2.54 \text{ mm}</td>
<td></td>
</tr>
</tbody>
</table>
rectangular channels with one or two walls heated, from a variety of studies that pre-
dominantly used water. The hydraulic diameters in the developed correlation ranged
from 2.1 mm to 20.5 mm. The correlation by Qu and Mudawar (2004) was developed
for an array of 21, 215 μm wide × 821 μm deep parallel microchannels flowing water
and single mini and microchannels (2.54 mm and 510 μm hydraulic diameters respec-
tively) flowing R-113. Assuming instabilities to signal premature critical heat flux,
the present data show a much reduced CHF limit on the order of 5 times, compared
to conventionally sized channels denoted by the Katto (1981) correlation. However,
in comparison with the correlation by Qu and Mudawar (2004) for microchannels
the difference is much less. The data from this correlation showed no dependence
of the critical heat flux on inlet subcooling. The present data, however, shows a
dependency on the inlet subcooling. This dependency is similar to that for conven-
tionally sized channels, in that the critical heat flux is decreased with an increase of
inlet subcooling. However, the degree of influence of subcooling is more significant
for the microchannels (45% and 75% increase from 80 °C to 70 °C) compared to the
conventional channels (1.8% increase from 70 °C to 80 °C). The remainder of this sec-
tion details the characteristics of flow under unstable conditions to better understand
their source and mechanisms. Also, this section looks at the influence of cross-links
to evaluate how they compare to standard straight parallel channel heat sinks under
unstable boiling conditions. A summary of the flow conditions leading to unstable
flow and the boiling oscillation measurement conditions are listed in Table 6.4.

171
Table 6.4: Measurement conditions and stability limits.

<table>
<thead>
<tr>
<th>Heat sink configuration</th>
<th>$T_{in}$ ($^\circ$C)</th>
<th>$Q_{vol}$ (ml/min)</th>
<th>$G$ (kg/(m² · s))</th>
<th>Conditions at onset of instability $q^*$ (W/cm²)</th>
<th>Bo</th>
</tr>
</thead>
<tbody>
<tr>
<td>STR02</td>
<td>70</td>
<td>20</td>
<td>91</td>
<td>4.1</td>
<td>1.958×10⁻⁴</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>140</td>
<td></td>
<td>7.1</td>
<td>2.247×10⁻⁴</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>186</td>
<td></td>
<td>9.7</td>
<td>2.316×10⁻⁴</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>30</td>
<td>140</td>
<td>4.9</td>
<td>1.540×10⁻⁴</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>185</td>
<td></td>
<td>5.5</td>
<td>1.312×10⁻⁴</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>228</td>
<td></td>
<td>7.9</td>
<td>1.538×10⁻⁴</td>
</tr>
<tr>
<td>INT01</td>
<td>70</td>
<td>30</td>
<td>140</td>
<td>7.0</td>
<td>2.215×10⁻⁴</td>
</tr>
</tbody>
</table>
6.4.3 Flow Boiling Oscillations

Once the stable threshold was crossed, sustained oscillations in flow parameters were observed. These were relatively periodic fluctuations in various sensors including the inlet and outlet temperature and the inlet and differential pressure sensors. Samples of these signal fluctuations, for a given heat flux and flowrate, are presented in Figure 6.18. They are presented both for the STR02 and INT01 heat sinks. The oscillations are captured at a rate of 5 to 10 Hz over a 20 s to 50 s span. The time traces show relatively large amplitude oscillations. This is likely due to periodic refilling of the channels, typical of what has been observed by other researchers working with boiling in parallel channel microsystems (Hetsroni et al., 2005a, 2006; Qu and Mudawar, 2004). A characteristic trend in the flow oscillations captured, is that the dominant frequency is the same for all the sensors, and they are in phase. This is a relatively low frequency on the order of $10^{-1}$ Hz. No discernible trends are observed between a pressure time trace versus inlet or outlet temperature time trace. The inlet temperature, however, is always observed to oscillate with a smaller amplitude than the outlet temperature. Its amplitude though, grows more significantly with increased heat flux and details of these amplitude characteristics are presented later.

Through performing a Fast-Fourier Transform on the time traces and parsing for the largest peak, the dominant frequency of the signals is obtained. Figure 6.19 shows the variation in the inlet temperature oscillation frequencies, comparing both heat sink configurations. For STR02, the frequency is higher than INT01 and decreases with increasing heat flux. For INT01, the frequency also decreases with increasing
Figure 6.18: Sample transient behaviour of various signals, $q'' = 7.9 \text{ W/cm}^2$, $T_m = 70 ^\circ\text{C}$, $Q_{vol} = 30 \text{ ml/min}$. 

(a) Inlet temperature sample transient behaviour.

(b) Outlet temperature sample transient behaviour.

(c) Inlet pressure sample transient behaviour.
Figure 6.19: Inlet temperature oscillation frequency versus power, $T_{in} = 70 \, ^\circ C$ for different heat sinks at $Q_{vol} = 30 \, \text{ml/min}$. 
heat flux.

Figure 6.20 (a) shows the variation in frequency for the STR02 heat sink as a function of inlet temperature, flowrate and heat flux. For a higher temperature, the frequency is lower for the same flowrate and with a lower flowrate, the frequency is also lower. It is observed that the frequency of oscillations is dependent not only on the heat flux, but the flowrate and the inlet subcooling. Also included in Figure 6.20 (a) is the frequency data obtained from Hetsroni et al. (2006). The heat sinks in this work contained 13 triangular cross-section channels, fabricated in a silicon substrate and with a hydraulic diameter of 220 μm. The mass flux range of the data was 63.3 kg/(m² · s) to 127 kg/(m² · s) and the heat flux ranged from 17 W/cm² to 32 W/cm². Similar trends to the present data are observed, in that the frequency decreases with increasing heat flux and lower mass fluxes have lower frequencies. A correlation has been developed to compile these observations. The frequency is non-dimensionalized with the heated length and the channel inlet velocity, assuming all liquid flow at entry,

\[ f^* = \frac{f u_{chn}}{L_{chn}} \quad (6.14) \]

A number of length scales exist to non-dimensionalize the frequency, and they include the boiling length, the full channel length and the hydraulic diameter. Based on the performance of a number of correlation fit trials, the channel length was chosen as opposed to the boiling length. The hydraulic diameter is accounted for through inclusion of the Weber number based on hydraulic diameter. Since the frequency is a function of the heat flux and subcooling, we may utilize the Boiling number
Figure 6.20: Inlet temperature oscillation frequency versus power for STR02 at different flowrates and inlet temperature.
as the non-dimensional heat flux and the Subcooling number \( N_{sub} \) to account for inlet subcooling. Finally, the number of channels is also included as an independent parameter. Based on these parameters, the following relationship was developed to correlate the data in Figure 6.20 (a),

\[
f^* = C_1 Bo^{C_2} N_{sub}^{C_3} We^{C_4} N_{chn}^{C_5},
\]

(6.15)

where, \( C_1 = 3.383 \times 10^{-15}, C_2 = -1.408, C_3 = 4.328, C_4 = 0.144 \) and \( C_5 = -0.137 \), or in dimensional form as,

\[
f_\omega = 3.383 \left( 10^{-15} \right) \frac{u_{chn, in}}{L_{chn}} \left( \frac{q''}{Gh_{fg}} \right)^{-1.408} \times \left[ \frac{Cp(T_{sat} - T_{in})}{h_{fg}} \cdot \frac{\rho_f - \rho_g}{\rho_g} \right]^{4.328} \left( \frac{C^2 D_h}{\sigma \rho_f} \right)^{0.144} (N_{chn})^{-0.137}.
\]

(6.16)

Fluid properties in the above are calculated from saturation conditions at the tube exit. Equation 6.15 fits the data with a standard deviation of 25.4%. Figure 6.20 (b) compares the correlation fit with the data and includes the 95% confidence intervals based on the standard deviation. The Boiling number exponent shows that \( f^* \) is inversely proportional to the heat flux and hence a decreasing frequency with increasing heat flux will be obtained. As well, the magnitude of the Subcooling number exponent is over double that of the Boiling number, suggesting a stronger dependency of the oscillation frequency on the subcooling compared to heat flux.

Figure 6.21 shows the variation in the peak to peak amplitude of various sensor signals with increasing heat flux. In all cases \( (P_{in}, T_{in}, T_{out}) \) the amplitude increases
Figure 6.21: Oscillation amplitude versus power of various signals, $T_{in} = 70 \degree C$ for different heat sinks at $Q_{col} = 30 \text{ ml/min}$. 

(a) Inlet temperature oscillation amplitude versus power

(b) Outlet temperature oscillation amplitude versus power

(c) Inlet pressure oscillation amplitude versus power
with increasing heat flux. Fluctuations in the inlet temperature are likely due to flow reversals traveling as far upstream as the inlet plenum chamber. Such observations have been described in some recent studies (Hetsroni et al., 2005a, 2006; Qu and Mudawar, 2004). Comparing inlet pressure amplitude for heat sinks STR02 and INT01 in Figure 6.21, we observe a consistently lower amplitude for INT01. However, looking at the inlet and outlet temperatures, the behavior is opposite, whereby, STR02 has a lower amplitude than INT01. Further, the difference in the amplitude is only 1 °C to 2 °C at the inlet whereas it is as high as 10 °C at the outlet. Such a large variation, if translated to the wall temperature, could be extremely detrimental and it is recommended for future works to investigate it further.

Figure 6.22 (a) shows the variation of the inlet pressure amplitude for the STR02 heat sink at different flowrates and inlet temperatures. With increased flowrate, the amplitude of the pressure fluctuations increases, but this is also related to boiling onset and instabilities occurring at a higher heat flux. With decreased inlet temperature, the pressure amplitudes are higher. Overall, the pressure amplitude correlates relatively well with the heat flux, showing an increasing trend. To provide a more universal representation in non-dimensional form, the pressure may be non-dimensionalized with the liquid density and the inlet velocity, assuming all liquid flow at entry. This may then be set, similar to the frequency case, as a function of the Boiling number, Subcooling number and Weber number. The following correlation:
Figure 6.22: Inlet pressure oscillation amplitude versus power for STR02 at different flowrates and inlet temperature.
with the fluid properties calculated at saturation conditions at exit is obtained,

$$P^* = \frac{P_{\text{amp}}}{G^2/\rho_f} = C_6 B_o C_7 N_{\text{sub}} W e D, \quad (6.17)$$

with the coefficients determined as, $C_6 = 17.600$, $C_7 = 1.016$, $C_8 = 0.714$ and $C_9 = -0.588$, or in dimensional form as,

$$P_{\text{amp}} = 17.600 \frac{C^2}{\rho_f} \left( \frac{q''}{G h_{fg}} \right)^{1.016} \times \left( \frac{C_p (T_{\text{sat}} - T_{\text{in}})}{h_{fg}} \right)^{0.714} \left( \frac{C^2 D_h}{\sigma \rho_f} \right)^{-0.588}. \quad (6.18)$$

The correlation fits the data well with a standard deviation of 8.4%. Figure 6.22 (b) compares the fit with the experimental points and shows the 95% confidence intervals based on the standard deviation.

The outlet temperature amplitude for the STR02 heat sink for various flowrates and inlet temperatures is presented in Figure 6.23 (a) as a function of the heat flux. In general, the temperature amplitude increases with increasing power as mentioned previously. The data has been correlated as a non-dimensional temperature utilizing the saturation temperature (in Kelvin) and set once again as a function of the Boiling number and Subcooling number. Fluid properties, as previously, have been calculated at the exit saturation temperature taken as 100 °C. From this, the following is obtained,

$$T^* = \frac{T_{\text{amp}}}{T_{\text{sat}}} = C_{10} B_o C_{11} N_{\text{sub}} C_{12}, \quad (6.19)$$
Figure 6.23: Outlet temperature oscillation amplitude versus power for STR02 at different flowrates and inlet temperature.
with the coefficients determined as, $C_{10} = 5.493 \times 10^7$, $C_{11} = 1.421$ and $C_{12} = -2.169$, or in dimensional form as,

$$T_{amp} = 5.493 \times 10^7 \times T_{sat} \left( \frac{q''}{G h_{fg}} \right)^{1.421} \left[ \frac{C_p (T_{sat} - T_{in}) \rho_f - \rho_g}{h_{fg} \rho_f} \right]^{-2.169}$$

(6.20)

in degrees Kelvin. Although there is a large degree of scatter, the correlation (Eq. 6.19) fits the data with a standard deviation of 25.5%. Figure 6.23 (b) compares the correlation fit with the experimental data and shows the 95% confidence intervals based on the standard deviation. The exponents show that increasing the Boiling number and hence the heat flux, will increase the inlet temperature amplitude, whereas an increased subcooling will have decreased amplitude.

### 6.4.4 Discussion

Although flow visualization was not performed, some conclusions on the type of oscillations may be obtained when compared with other recent findings. The frequency and amplitude data by Hetsroni et al. (2006) shows similar trends as the present. Additionally, these authors were able to track the period of oscillation of flow patterns in individual channels. The period of oscillation of the system, versus an individual channel from their work, has been calculated to range from 5 to 15 times. For the present data, the period of oscillation for the system ranged from 1.5 s to 10 s, compared to the liquid particle travel time, which was estimated to range from 0.06 s to 0.16 s, for a flowrate of 20 to 50 ml/min. This ratio of the system’s oscillation period to the particle travel time, is on the order of 10 times and is similar to the range
of the data by Hetsroni et al. (2006). These qualitative and quantitative similarities lead to the conclusion that the present oscillation phenomena are explosive boiling oscillations, as described by Hetsroni et al. (2005a, 2006) and also observed by some other researchers (Steinke and Kandlikar, 2004a).

A note on the correlations presented follows. According to Yadigaroglu and Bergles (1972), density wave oscillation characteristics are governed by the combination of a variety of auxiliary phenomena. These include the heat capacity of the walls, the saturation enthalpy’s dependency on the system pressure, the gas phase compressibility, the variation in heat transfer and the relative velocity between the phases. Although these auxiliary phenomena have been described for density wave oscillations, they are expected to also have some significance during explosive boiling oscillations. The correlations presented, are then considered applicable to parallel silicon wafer substrate microchannel systems flowing water, with a system pressure of 1 atm and for low mass flux ranges.

The flow boiling results presented did not include local heat transfer measurements for two reasons. First, the measurements with the first support used water. This was necessary, as the bond for the chip to the mid-plate was silicone based, a compound not compatible with FC-72 (as opposed to the element Silicon used for the heat sink fabrication). Boiling heat transfer measurements with water would require TLC’s with temperatures greater than 100 °C which were not available. Secondly, flow boiling heat transfer measurements were indeed attempted with the second support using FC-72. However, during a degassing run, whereby significant vapour may be generated, a leak was produced in the bond between the heat sink and the top plate.
This leak was likely due to a combination of the moisture limits and rigidity of the UV adhesive utilized. Additionally, the large surface area which required bonding due to the multiple hole arrangement, also likely contributed to a weaker bond. For future investigations, a compliant adhesive is recommended. As well, using a single hole entry rather than multiple hole entry will eliminate the requirement for a transparent adhesive.

6.5 Summary

The flow and heat transfer characteristics of a cross-linked silicon microchannel heat sink have been presented. Relatively good agreement was obtained between air and water adiabatic pressure measurements compared with theoretical predictions, after accounting for inlet and outlet losses. No significant flow differences were observed between a standard and a cross-linked heat sink scheme for the cross-linked spacing investigated (cross-link width ≈ channel width).

Heat transfer measurements under single phase conditions were successfully obtained using the liquid crystal thermography methods developed in Chapter 4. Variations in the streamwise thermal profiles were observed due to the cross-links. These variations, however, were minor due to the narrow width of the cross-links used.

Flow boiling oscillations pose a restriction on the realization of MEMS based two-phase flow heat sinks for thermal management in demanding applications, for both terrestrial and extra terrestrial applications. The detailed flow characteristics of flow boiling instabilities in two different silicon microchannel heat sinks were presented.
One is a straight standard microchannel configuration, whereas the second contains cross-links along the channels. There were similarities in the characteristics of the flow boiling oscillations for both configurations and these may be summarized as follows,

1. Both configurations show a decreasing frequency of oscillation with increasing heat flux.

2. Flow oscillations for inlet temperature, outlet temperature and inlet pressure are at the same frequency.

3. In both configurations, the peak to peak amplitude for the oscillations increased with increasing heat flux.

The qualitative and quantitative characteristics of these oscillations are similar to the explosive boiling oscillations described in recent literature. Instability data were presented at two different inlet temperatures and a range of flowrates for the straight standard heat sink. The data showed that with increasing flowrate, the frequency of oscillations increased, whereas with decreasing inlet temperature, the oscillation frequency increased. The frequency and amplitude characteristics of these oscillations, for the standard heat sink at two different inlet temperatures, have been correlated with the Boiling number, the Subcooling number and Weber number. These correlations provide an intuitive means of interpreting the frequency and amplitude characteristics as a function of flowrate, inlet subcooling and heat flux, combined with a quantitative measure of their relation.
Chapter 7

Investigating a Radial Flow Microchannel Heat Exchanger

This chapter investigates the heat transfer characteristics in a radial microchannel heat exchanger scheme. Radial flowing heat exchangers are of interest in scenarios of thermal management for other radially flowing microdevices. An example is the radial micro fuel cell presented by Pattekar and Kothare (2005). In this design, radial outward flow was utilized due to the use of varying channel cross-sections, for control of the chemical reaction. For thermal management of the fuel cell a radial heat exchanger may be utilized. Another prevalent radially flowing microdevice is the microturbine (Epstein, 2003). To improve thermal efficiency, reheat or intercooling stages can be introduced, and radial flow would be advantageous to accommodate the inherent radial flow of these devices.
7.1 Design and Fabrication Trials

A radially flowing heat exchanger was designed using similar criterion as the cross-linked heat sink. This criterion was that given a 3.5 cm foot print area with a heat flux of 25 W/cm², maintain a maximum temperature of 85 °C with a maximum 5 psi supply of single phase water. A similar thermal resistance analysis as described for the cross-linked heat sink was performed to size the radial design. The channel height was fixed at 300 μm, with a central plenum of 1 mm and a minimum wall thickness of 50 μm. Results from this analysis are presented in Figure 7.1. The figure compares the resulting maximum temperature and pressure demands for a variety of channel widths and flowrates. From the figure it is observed that to meet a maximum temperature of 85 °C, a channel width just over 100 μm is required, with a pressure supply just under 3 psi.

The designed heat sink is shown in Figure 7.2. It is a three-wafer stack consisting of a 550 μm thick silicon wafer base with a SiO₂ layer on its backside for electrical isolation. This base piece contains the microchannels fabricated via DRIE. A middle wafer, also fabricated from silicon and 330 μm thick, acts to isolate the incoming and outgoing flow streams. It is adhered to the base silicon wafer using a gold eutectic bonding technique. A Pyrex glass cover 500 μm thick is anodically bonded to the silicon stack. Eight circumferential holes are located on the glass cover for flow transfer. The holes are fabricated via drilling with a high speed diamond tipped drill. Flow may then enter radially inward or outward. Similar to the parallel channel design, a platinum heating element 1 mm wide and 186 mm long is deposited on
Figure 7.1: Radial concept analysis results.
Figure 7.2: Radial heat sink.
the backside of the channel base to simulate the heated chip. Thermochromic liquid
crystal material may then be deposited on top of the heating element, followed by a
glass slide.

Fabrication trials were performed for this design by the Washington Technology
Center's Microfabrication Lab. Unique in this design is the three-wafer stack com-
posed of silicon to silicon to Pyrex. This posed unique challenges for its fabrication,
particularly with respect to bonding the different layers. Additionally, drilling of the
holes was difficult due to their close spacing and fractures between two holes would
easily occur. One chip managed to pass through all the fabrication phases, and was
provided for testing. During handling of this chip, however, the Pyrex to silicon bond
delaminated. An attempt was made to seal it and continue with the testing. The
seal, however, did not provide adequate support and during testing leaks between the
Pyrex and silicon mid-plate were present. Further experimental work with the radial
design was discontinued at that point until a reliable chip could be fabricated.

A potential benefit of a radially flowing heat exchanger is the geometries ability
to easily incorporate varying cross-sectional area channels. For example, linearly
converging channels for radially inward flow or linearly diverging channels for radially
outward flow. As highlighted in the literature review, few works are available on the
convective heat transfer characteristics of channels with slowly varying cross-sections.
In the first part of this chapter, a thermal resistance model is adapted for use with
radially inward flow channels with streamwise varying cross-section. The analysis
evaluates the benefits and characteristics of flow in channels of varying cross-sections.
The second part of the chapter performs a similar investigation, but with a three-dimensional conjugate heat transfer model. For both analyses, the general heat sink dimensions are based on the fabricated design presented in this section.

7.2 Thermal Resistance Analysis

7.2.1 Mathematical Model

Microfabrication through techniques such as DRIE enables a variety of channel shapes to be fabricated. This section evaluates the possibility of improved performance with a linearly varying channel cross-section. Also considered is the influence of the wall heating condition. The model is an adaptation of the model by Bau (1998), which only considered a 3-wall heated boundary condition under fully developed flow conditions. This model was a one-dimensional thermal resistance network model which assumed steady state, laminar and incompressible flow within a heated parallel channel system. The present adaptations are 4-wall heating, 4-wall heating with thermally developing flow and radially configured channels. Thermally developing flow is more realistic for microsystems, since due to the large pressure drop, short channels are usually encountered. The fluid properties are taken as constant, viscous dissipation and axial conduction are neglected as are heat transfer due to natural convection and radiation.

The modeled region consists of a unit cell which is depicted in Figure 7.3. This cell consists of a single channel with a fixed height \(H_{chn}\) and length \(L_{chn}\) which spans the outer \(r_{out}\) to inner \(r_{in}\) radius. The wall thickness resistance is neglected
Figure 7.3: Modeling domain depiction.
assuming the material to have a high thermal conductivity. The channel width is defined as some continuous function of the axial location \((W_{chn}(r) = r\beta_2)\), whereas the channel height is constant. Correspondingly the channel aspect ratio, defined as the width to height ratio, is a function of the radial location \((\alpha = W_{chn}(r)/H_{chn})\).

The total thermal resistance is a combination of the convective thermal resistance \((R_{conv})\) and the calorimetric thermal resistance \((R_{cal})\). Considering convective transport within a single channel and due to a heat flux applied on the top outer wall, we have,

\[
\left[\frac{q''r(r^2_{out} - r^2_{in})}{N_{chn}}\right]/A\frac{3}{4} = h(r)(T_w(r) - T_b(r)).
\]  

(7.1)

The convective thermal resistance may then be derived as,

\[
R_{conv} = \theta_w(r) - \theta_b(r) = \frac{2\alpha\pi (r^2_{out} - r^2_{in})}{Nu(r)(1 + \alpha)\left(L_{chn}H_{chn}\left(\frac{\alpha_2 + \alpha_1}{2}\right) + 2\right)N_{chn}},
\]  

(7.2)

for a 3-wall heated channel and

\[
R_{conv} = \theta_w(r) - \theta_b(r) = \frac{2\alpha\pi (r^2_{out} - r^2_{in})}{Nu(r)(1 + \alpha)\left(2L_{chn}H_{chn}\left(\frac{\alpha_2 + \alpha_1}{2}\right) + 1\right)N_{chn}},
\]  

(7.3)

for a 4-wall heated channel. \(\theta\) is the non-dimensional temperature defined as,

\[
\theta = \frac{T - T_{b,in}}{q''H_{chn}/k}.
\]  

(7.4)

Numerically derived solutions of the Nusselt number under fully developed flow
for cases of 3 and 4 wall heated conditions, may be obtained from Shah and London (1978) as a function of aspect ratio. Similar to Bau (1998), the solutions for periphery constant wall temperature are taken. For thermally developing flow, Lee and Garimella (2006) recently provided a correlation based on numerical solutions for 4-wall heated conditions with constant heat flux. This was selected as data is not readily available for a constant periphery wall temperature, for a variety of aspect ratio channels and under thermally or simultaneously developing flow conditions.

For the calorimetric thermal resistance, an energy balance within the channel gives,

$$\frac{q'' \pi (r_{out}^2 - r^2)}{N_{chn}} = \dot{m}C_p (T_b(r) - T_b(r_{out})).$$  \hspace{1cm} (7.5)

Following the considerations by Bau (1998), the calorimetric thermal resistance is defined as,

$$R_{cal} = \theta_b(r) = \frac{\pi \left\{ r_{out}^2 - \left( r_{out} - r^+ (r_{out} - r_{in}) \right)^2 \right\}}{N_{chn}} \left( \frac{k_f}{C_p} \right) \left( \frac{1}{\dot{m}^* \cdot \dot{m}_o H_{chn}} \right)$$  \hspace{1cm} (7.6)

where

$$r^+ = \frac{r_{out} - r}{r_{out} - r_{in}}$$  \hspace{1cm} (7.7)

and

$$\dot{m}^* = \frac{1}{\left( \frac{1}{8} \int_0^1 P_\alpha(\alpha) \frac{(1+\alpha)^2}{\alpha^3} dr^+ \right)}.$$  \hspace{1cm} (7.8)
The parameter \( \dot{m}_0 \) is the characteristic mass flowrate parameter given by,

\[
\dot{m}_0 = \frac{H_{chn}^4 \Delta P}{\nu L_{chn}}.
\]  

(7.9)

To obtain the above definition of calorimetric thermal resistance, Bau (1998) assumed the flow to be locally fully developed within the channel and the pressure drop due to acceleration to be negligible. The Poiseuille number \( (Po(\alpha)) \) has been obtained from Shah and London (1978) and is given by,

\[
Po(\alpha) = f \text{Re} \\
= 96[1 - 1.3553\alpha + 1.94673\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5].
\]  

(7.10)

This is a correlation for numerical solutions of fully developed flow within a channel of aspect ratio \( \alpha \). The total thermal resistance is then obtained from summation of the calorimetric and convective resistances. This is equivalently the wall temperature,

\[
R_{tot} = \theta_w(r) = R_{conv} + R_{cal}.
\]  

(7.11)

Detailed presentation on the development of these equations may be found in the Appendix (Section C.1).
7.2.2 Problem Analysis

The performance criterion evaluated was the channel configuration which produces the smallest peak wall temperature and defined as,

\[ \min [\max \{\theta_w(r)\}] . \]  \hspace{1cm} (7.12)

The aspect ratio may be defined as an \( N^{th} \) order polynomial,

\[ \alpha(r) = \sum_{k=0}^{N} \alpha_k r^k . \]  \hspace{1cm} (7.13)

For the present study, linear, quadratic and cubic polynomials were considered. The minimization of Equation 7.12 was performed using a Golden Section search and parabolic interpolation minimization routine available using the MATLAB \texttt{fminbnd} and \texttt{fminsearch} functions. The \texttt{fminbnd} function is a one parameter minimization routine and was used to optimize the uniform channel configurations. The \texttt{fminsearch} function is a multi-parameter minimization routine and was used for the varying cross-sectional channel searches. This function may produce local minima and hence was found to occasionally be start point dependent. For this reason, the channel configurations using this multi-parameter function are not identified as optimum, but improved.

General inputs for the analysis include a channel depth of 300 \( \mu \)m and an outer and inner radius of 11.05 mm and 0.5 mm respectively. The working fluid is water with properties set at 25 \(^\circ\)C. Restrictions were placed on the channel width and the
fin width at exit to be at minimum 50 μm.

7.2.3 Results

The resulting channel configurations determined using the developed 1-D analysis for radially inward flowing channels, for a variety of boundary conditions, are presented in Figure 7.4. For the present conditions, at a heat flux of 25 W/cm², a difference of \( \theta_w = 0.01 \) translates into temperature difference of approximately 1.2 °C. In all cases, a lower peak temperature is obtained by using a converging channel. Looking at the uniform cross-sectional area channels, a consistent trend is observed. This trend shows a decreasing channel width with increasing pressure availability. The increased pressure allows for smaller channel diameters, which in turn improves the thermal performance. This is due to both higher heat transfer coefficients and increased convective surface area, due to more channels. For the channels with varying cross-sectional areas, however, such a trend is not readily observed. Comparing the channels with varying cross-sectional areas to uniform channels, in all cases the varying cross-sectional area channels have a wider inlet and a narrower exit for a given pressure drop. This is possible as the wider inlet provides a lower frictional resistance, and this added benefit may then be utilized at the narrower exit. The resulting wall profiles for the fully developed cases approach each other, both for uniform and varying cross-section configurations as the available pressure increases. The resulting non-dimensional temperatures are also similar, however, the magnitudes may vary substantially when considering the dimensional values.
Figure 7.4: Minimizing peak temperature: Linear profile. Solid lines are uniform channels and dashed lines are varying cross-sectional area channels.
Observing the wall temperature profiles in all cases, a non-linear rise in wall temperature is obtained. For 4-wall heated fully developed (Fig. 7.4 (c) & (d)) versus thermally developing flow (Fig. 7.4 (e) & (f)), a more realistic temperature profile is obtained whereby the initial wall to fluid temperature difference approaches zero and rises rapidly due to the developing thermal boundary layer. Comparing Figures 7.4 (c) and (e), for the uniform channels, the influence of a fully developed versus a thermally developing profile influences the entrance characteristics. The exit temperatures, however, are similar and the peak occurs at the exit. Therefore, in terms of minimizing the peak temperature, if considering uniform channels, fully developed and thermally developing flows will provide a similar conclusion. However, this is substantially different when considering a channel with linearly varying cross-section in a radial configuration. Although the exit temperatures are similar in magnitude, the peak does not necessarily occur at the exit. Subsequently, the improved configuration can be significantly different depending on whether fully developed or thermally developing flow is used.

The previous case considered a fixed fin width at the channel exit. The exit fin width will depend on fabrication constraints and structural requirements. Figure 7.5 shows the resulting wall temperatures and channel configuration for exit fin widths of 0 \( \mu \text{m} \), 50 \( \mu \text{m} \) and 100 \( \mu \text{m} \). No general trend in channel configurations is observed with increasing exit fin width and pressure supply. The wall temperatures, however, increase with increasing exit fin width. This is due to a decreasing number of channels with increasing fin width and is further due to the decreased area available. It would be advantageous to have a zero thickness exit fin width, however based on design
Figure 7.5: Minimizing peak temperature: Influence of fin width for thermally developing, 4-wall heated channels. Solid lines are uniform channels and dashed lines are varying cross-sectional area channels.
and fabrication constraints, the present results show that the characteristics of the improved channel configuration will be design specific.

7.3 Three-dimensional Conjugate Analysis

The results presented in the previous section are limited by the assumptions incorporated. Particular assumptions that may have significance for actual microchannel systems are the flow development (thermally and hydrodynamically), negligible axial conduction and constant fluid properties. An investigation was carried out on a radially inward flowing micro-heat exchanger, using a three-dimensional conjugate heat transfer model.

7.3.1 Formulation and Solution Methodology

The computational domain is the shaded region depicted in Figure 7.3 (a), spanning the full radius. It is composed of a single half-section of a channel and fin portion, with the adjacent surfaces set to symmetry boundary conditions. The top outer wall is heated by a constant heat flux and the base wall is adiabatic. All other outer walls are also set to adiabatic. The inlet is set to pressure inlet or uniform velocity depending on the case being evaluated, while the outlet is set to zero gage pressure. The solid material is silicon \((k = 157 \text{ W/m-K})\), while the fluid is water with properties obtained from Collier and Thome (1999). The modeling assumptions were as follows:

- Steady state flow and heat transfer,

- Laminar flow,

203
• Constant solid properties and temperature dependent fluid properties,

• Negligible radiation heat transfer.

The commercial package Gambit 2.0 is used for grid generation. A structured mesh with hexahedral elements is used in all the test cases.

The analysis is carried out using the commercial package FLUENT 6.1 to solve the continuity, momentum and energy equations within the fluid and given by,

$$\nabla \cdot \left( \rho \vec{V} \right) = 0, \quad (7.14)$$

$$\vec{V} \cdot \nabla \left( \rho \vec{V} \right) = -\nabla P + \nabla \cdot \left( \mu \nabla \vec{V} \right) \quad (7.15)$$

$$\vec{V} \cdot \nabla (\rho C_p T) = \nabla \cdot (k \nabla T), \quad (7.16)$$

and the conduction equation within the solid given by,

$$\nabla \cdot (k \nabla T) = 0. \quad (7.17)$$

Discretization of pressure, velocity and energy terms is through a second order upwind scheme. Coupling of pressure and velocity is performed using the SIMPLE algorithm. Finally, the solution of the equations is through a Segregated solver.

7.3.2 Benchmark Study and Grid Independence Evaluation

A benchmark investigation was performed to evaluate the efficacy of the modeling method and assumptions. The selected case was from the study by Lee and Garimella.
(2005). This study involved heat transfer investigations in a copper substrate with 10 channels. The selected case consisted of 194 \( \mu m \) wide by 884 \( \mu m \) deep channels (\( D_h = 318 \mu m \)), 25.4 mm long. The simulated domain and boundary conditions are similar to those intended for the radial heat sink. The domain consisted of one half the channel width and wall thickness and the full channel length. Symmetry boundary conditions were applied on the side surfaces, a uniform heat flux on the top surface and adiabatic conditions on the bottom surface. The flow was taken as laminar, with pressure boundary conditions and temperature dependent fluid properties. The double precision form of FLUENT was necessary to produce reasonably decreasing residuals. A number of grid densities were evaluated and by tracking the outlet temperature a grid density of \( 50 \times 220 \times 120 \) was deemed satisfactory. The results from the mesh evaluations are presented in Table 7.1. A comparison of the average Nusselt number from the selected mesh and experimental data are shown in Figure 7.6. The figure shows that satisfactory agreement is obtained when compared to the experimental data. Lee and Garimella (2005) also performed numerical simulation with a half channel section using FLUENT. Satisfactory agreement was obtained with their experimental data using a mesh of \( 50 \times 160 \times 100 \) cells. Their study however neglected the temperature dependence of the fluid properties. Discrepancies between the present numerical results and the experimental data, are likely due to a combination of experimental error and data reduction of the experimentally defined average Nusselt number. Experimental errors were quoted in the range of 6 to 17\% and estimates with the present data show differences of about 12\% at \( Re = 800 \) and 4\% at \( Re = 400 \). The average Nusselt number was obtained from four thermocouples located
Table 7.1: Benchmark grid study.

<table>
<thead>
<tr>
<th>Case</th>
<th>Mesh ((W_{chn} \times H_{chn} \times L_{chn}))</th>
<th>(T_{out})</th>
<th>% change</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>(30 \times 40 \times 50)</td>
<td>308.9783</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>(40 \times 100 \times 100)</td>
<td>309.1931</td>
<td>0.070</td>
</tr>
<tr>
<td>3</td>
<td>(50 \times 160 \times 100)</td>
<td>309.225</td>
<td>0.010</td>
</tr>
<tr>
<td>4</td>
<td>(50 \times 220 \times 120)</td>
<td>309.2292</td>
<td>0.001</td>
</tr>
<tr>
<td>5</td>
<td>(60 \times 280 \times 150)</td>
<td>309.235</td>
<td>0.002</td>
</tr>
</tbody>
</table>
Figure 7.6: Benchmark grid comparison with experimental data.
along the channel and placed a distance from the fluid-wall interface. Additionally the wall temperature was calculated based on a one-dimensional extrapolation of the thermocouple temperature.

Considering now the radial scheme, two grid sizes were evaluated with mesh densities of $50 \times 160 \times 100$ and $50 \times 220 \times 120$. The percent change in outlet temperature was 0.001% and hence the $50 \times 220 \times 120$ grid size was selected for the study.

### 7.3.3 Results

A number of configurations of varying cross-sectional area channels were analyzed and compared to a uniform channel with an inlet and outlet aspect ratio of 0.5. A summary of the test cases are listed in Table 7.2. For all the test cases the outlet aspect ratio is set to 0.5 with a channel height of 300 $\mu$m. The aspect ratio of 0.5 was selected based on the results from the one-dimensional thermal resistance analysis, at a pressure of 6895 Pa (1 psi). The channel height was selected based on the fabrication considerations. The cases considered were the influence of inlet width for aspect ratios ranging from 0.4 to 1.0, for a fixed pressure drop and a heat flux ranging from 25 W/cm² to 62.5 W/cm². Similarly the influence of inlet width for inlet aspect ratios ranging from 0.4 to 1.0, but for a fixed mass flowrate and a heat flux of 25 W/cm². The influence of inlet height for aspect ratios of 0.4, 0.5 and 1.0 for a fixed pressure drop and a heat flux of 25 W/cm². In the last case both the inlet height and width were variable and the inlet aspect ratio fixed at 0.5, for a fixed pressure drop and a heat flux of 25 W/cm².
Table 7.2: 3-D CFD simulation test matrix. Below dimensions in millimeters. $W_{out} = 0.15$ mm & $H_{out} = 0.30$ mm throughout. $q_1'' = 25$ W/cm$^2$, $q_2'' = 43.75$ W/cm$^2$ and $q_3'' = 62.5$ W/cm$^2$, $\Delta P_{fixed}$ = 6.9 kPa and $\dot{m}_{fixed}$ = $2.85 \times 10^{-5}$ kg/s.

<table>
<thead>
<tr>
<th>$\alpha_{in}$</th>
<th>$H_{in} = 0.300$</th>
<th>$W_{in} = 0.150$</th>
<th>$\alpha_{in} = \alpha_{out} = 0.5$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\Delta P_{fixed}$</td>
<td>$\dot{m}_{fixed}$</td>
<td>$\Delta P_{fixed}$</td>
</tr>
<tr>
<td></td>
<td>$q_1''$</td>
<td>$q_2''$</td>
<td>$q_3''$</td>
</tr>
<tr>
<td>0.4</td>
<td>$W_{in} = 0.120$</td>
<td>$W_{in} = 0.120$</td>
<td>$W_{in} = 0.120$</td>
</tr>
<tr>
<td>0.45</td>
<td>$W_{in} = 0.135$</td>
<td>$W_{in} = 0.135$</td>
<td>$W_{in} = 0.135$</td>
</tr>
<tr>
<td>0.5</td>
<td>$W_{in} = 0.150$</td>
<td>$W_{in} = 0.150$</td>
<td>$W_{in} = 0.150$</td>
</tr>
<tr>
<td>0.55</td>
<td>$W_{in} = 0.165$</td>
<td>$W_{in} = 0.165$</td>
<td>$W_{in} = 0.165$</td>
</tr>
<tr>
<td>0.6</td>
<td>$W_{in} = 0.180$</td>
<td>$W_{in} = 0.180$</td>
<td>$W_{in} = 0.180$</td>
</tr>
<tr>
<td>0.8</td>
<td>$W_{in} = 0.240$</td>
<td>$W_{in} = 0.240$</td>
<td>$W_{in} = 0.240$</td>
</tr>
<tr>
<td>1</td>
<td>$W_{in} = 0.300$</td>
<td>$W_{in} = 0.300$</td>
<td>$W_{in} = 0.300$</td>
</tr>
</tbody>
</table>
Figure 7.7 presents the maximum surface temperature determined for the various test cases. With increasing inlet width and thus increasing aspect ratio, a decreased maximum temperature is obtained. Equivalently with an increasing inlet height and thus decreasing aspect ratio, decreased maximum temperature is obtained. In the case of a fixed aspect ratio, increasing the inlet area similarly decreases the maximum temperature. The average surface temperature plots showed similar trends to the maximum surface temperature.

The decreasing peak temperature with increasing inlet aspect ratio is primarily due to the increasing flowrate. Figure 7.8 (a) shows the mass flowrate for various heat fluxes as a function of inlet aspect ratio and the thermal resistance also as a function of inlet aspect ratio (Figure 7.8 (b)), for fixed height channels. Note that the total thermal resistance in Figure 7.8 (b) is calculated from the summation of the convective and calorimetric resistances, whereby the convective resistance is based on the channel averaged heat transfer coefficient. Increasing the inlet aspect ratio increases the mass flowrate (Fig. 7.8 (a)) which in turn will decrease the calorimetric thermal resistance (Fig. 7.8 (b)). The diverging channels ($\alpha_{in} < 0.5$) will produce higher heat transfer coefficients and hence lower convective resistance. However, the benefit in heat transfer coefficient due to decreased channel inlet size is not significant compared to the mass flow benefit for increased channel inlet size. This arises due to the restriction on the number of channels, which is subsequently due to the restriction of the exit radius and fin width at the exit.

The mass flow alone, however, is not the only criterion for improved heat transfer. This is observed in the fixed pressure drop case of $\alpha_{in} = 0.6$ (0.3mm×0.18mm) and
Figure 7.7: Maximum surface temperature versus aspect ratio.
Figure 7.8: Evaluation of optimal aspect ratio.
$\alpha_{in} = 0.4$ (0.15mm×0.375mm), which are fixed height and fixed width cases. Their inlet cross-sectional areas are within 4.2% of each other, with $\alpha_{in} = 0.6$ having the lower cross-sectional area. The mass flowrates, however, are reversed with $\alpha_{in} = 0.4$ having a mass flowrate approximately 3.8% lower than $\alpha_{in} = 0.6$. Looking at Figure 7.7, however, an aspect ratio of 0.4 produces the lower maximum temperature ($\theta_{\text{max}} = 0.44$), compared to an aspect ratio of 0.6 ($\theta_{\text{max}} = 0.48$). This is primarily due to the differences in heat transfer coefficients, with the aspect ratio of 0.6 having an average Nusselt number of 8.6, whereas the aspect ratio of 0.4 produces an average Nusselt number of 14.0. Some benefit is also likely gained from a reduced inner to outer wall thickness for the 0.4 aspect ratio case, due to a larger height.

In Figures 7.9 and 7.10, the fluid bulk and inner wall temperature characteristics of the radial heat exchanger are presented. The fluid bulk temperature is calculated based on an energy weighted average, whereas the wall temperature is calculated based on an area weighted average. In all cases, the fluid bulk temperature shows a non-linear rise with increasing distance. A lower fluid bulk temperature rise is observed for cases with lower maximum wall temperatures. This is because the lower maximum temperature was obtained from an increase in mass flow. Due to a balance of energy then, for the same heat flux, a lower fluid bulk temperature rise will be obtained when the mass flowrate is increased. This is also a consequence of the fluid property changes having minimal influence. The inner wall temperature shows a different inner wall temperature trend compared to parallel channel systems. The inner wall temperature rises until approximately 50% of the length then drops. The rising wall temperature is expected due to the increasing fluid temperature. The drop,
(a) Constant channel height, various inlet aspect ratios.

(b) Constant mass flowrate, various inlet aspect ratios.

(c) Constant channel width, various inlet aspect ratios.

(d) $\alpha_m = 0.5$, various inlet areas.

Figure 7.9: Fluid bulk temperature.
however, arises from the thinning of the wall fins inherent in the radial configuration. In the case of a fixed mass flowrate (Fig 7.9 and 7.10 (b)), the fluid temperature rise is the same for all the cases. However, a lower inner wall temperature is obtained with inlet aspect ratio of 0.4. This results from a higher Nusselt number for the 0.4 aspect ratio channel, compared to the 0.8 aspect ratio channel under the same mass flow conditions.

Local Nusselt numbers for the various cases are presented in Figures 7.11 and 7.12 and are based on the local hydraulic diameters. In all cases, the local Nusselt values range from approximately 21 to 5. Additionally, they are still decreasing at the exit signifying that the flow is still thermally developing. For a 3-wall heated channel under fully developed conditions and for an aspect ratio of 0.5 with constant periphery wall temperature, the Nusselt number is 3.842 (Shah and London, 1978).

Considering varying channel inlet widths (fixed height, Fig. 7.11) at fixed heat fluxes, the Nusselt number changes only slightly between the cases. Additionally, the local Nusselt values rise slightly with increased channel width. Similarly in the case of varying channel inlet heights (fixed width, Fig. 7.12 (b)), the Nusselt number slightly rises with increased channel height. In general, an increase in the inlet area will produce a slightly higher Nusselt value for a fixed pressure drop.

Contours of the temperature resulting in a uniform channel and a converging channel of the same aspect ratio ($\alpha = 0.5$) are presented in Figures 7.13 and 7.14 respectively. Looking at the uniform channel case, Figure 7.13 (a) shows the surface temperature, with the channel located along the centre. The surface temperature
(a) Constant channel height, various inlet aspect ratios.

(b) Constant mass flowrate, various inlet aspect ratios.

(c) Constant channel width, various inlet aspect ratios.

(d) $\alpha_{in} = 0.5$, various inlet areas.

Figure 7.10: 3-wall heated area average temperature.
Figure 7.11: 3-wall heated area averaged Nusselt numbers for various heat flux and fixed channel height.
(a) Constant mass flowrate, various inlet aspect ratios.

(b) Constant channel width, various inlet aspect ratios.

(c) $\alpha_{in} = 0.5$, various inlet areas.

Figure 7.12: 3-wall heated area average Nusselt numbers for various configurations.
shows the elevated temperature to lie in the mid-streamwise section, with the inlet and outlet temperatures both cooler. This is similarly related to the inner wall temperature rise and drop that was presented earlier (Fig. 7.10). Cross-sections of the silicon wafer temperature gradients normal to both inner wall surfaces, spread over a large distance at the inlet plane (Fig. 7.13 (b)). For subsequent planes, the reduction in area causes both a lower temperature gradient and spread. The inner wall temperature contours (Fig. 7.13 and 7.14 (c)) show that the temperature is somewhat constant around the periphery in the inlet and outlet regions. However, as the center of the channel is approached, the periphery temperature is increasingly more non-uniform. Comparing the uniform and converging channels, similar trends in the profiles are observed. The magnitudes, however, are significantly different as expected from the averaged plots presented earlier.

The velocity contours for the symmetry plane of a uniform channel ($\alpha = 0.5$) under varying heat flux, are presented in Figure 7.15. With an increase in heat flux, higher velocities are attained. This is expected as the fluid’s viscosity decreases with increased temperature, providing a lower flow resistance. The velocity contours are also observed to be unsymmetrical with respect to the height, with higher velocity gradients obtained on the upper half of the channel. This is related once again to viscosity changes in the fluid, as the upper wall is heated and thus the fluid bulk temperature is hotter on the upper half.

Similarly, Figure 7.16 shows velocity contours for the symmetry plane for different geometrical configurations. Figure 7.16 (a) is for a channel with decreasing width, (b) is for a channel with decreasing height and in (c) both the width and the height are
Figure 7.13: Non-dimensional temperature contours, uniform channel ($\alpha = 0.5$).

(a) Heated surface.

(b) Silicon cross-section.

(c) Inner wall.
Figure 7.14: Non-dimensional temperature contours, converging channel with inlet area of \(240\mu m \times 480\mu m\) \((\alpha_{in} = 0.5)\).
Figure 7.15: Influence of heat flux on center-plane velocity (m/s), uniform channel ($\alpha = 0.5$).
(a) Varying channel width (fixed height), $\alpha_m = 0.6$.

(b) Varying channel height (fixed width), $\alpha_m = 0.4$.

(c) Varying channel width and height (fixed aspect ratio), $\alpha_m = 0.5$.

Figure 7.16: Center-plane velocity (m/s) for various geometrical configurations.
decreasing. Increasing velocities are now obtained as a combination of a decreasing viscosity and a decreasing flow area. The case of simultaneously varying channel width and height has the highest velocity, and correspondingly the highest mass flowrate, and lowest maximum temperature.

7.4 Thermal Resistance Model Compared to

Three-Dimensional Conjugate Analysis

The model developed by Bau (1998) and the present adaptations for the thermal resistance model, incorporating thermally developing flow and radially inward flow, have not been previously verified. These simplified models are advantageous for rapid geometry optimization, however, they contain a number of assumptions which may pose a limited applicability on their use. In this section, a comparison is made between these simplified models and a conjugate three-dimensional analysis, to explore the range of their applicability. The results for the conjugate three-dimensional analysis are derived from the previous section. In comparison to the simplified analysis, the three-dimensional model incorporates the fluid's temperature dependent properties, solid and fluid axial conduction, and flow development.

Figure 7.17 compares the one-dimensional thermal resistance model for parallel channels considering fully developed and thermally developing flows. The CFD analysis is from the benchmark case, which consisted of uniform channels with hydraulic diameters of approximately 318 μm and aspect ratios of approximately 0.22. Good
(a) 3-wall and 4-wall heated fully developed (FD) flow models.

(b) 4-wall heated thermally developing (TD) flow models. Constant heat flux (H1) and constant temperature (T) boundary conditions.

Figure 7.17: Parallel uniform channel: One-dimensional thermal resistance model comparison with three-dimensional conjugate analysis.
agreement is obtained between the CFD analysis and the thermally developing flow cases with respect to the general trend and magnitude. The fully developed cases, however, largely over predict the wall temperature. This over prediction is in part due to the higher heat transfer coefficients available in the flow development region, which subsequently lowers the wall temperature. The thermally developing cases, however, do not predict the entrance or exit region wall temperatures well. For the entrance this is due to the thermal resistance model not accounting for axial conduction, as is discussed later. For the exit it is likely due to neglecting the fluids viscosity decrease with distance. The decreasing viscosity will produce a higher local Reynolds number, which increases the heat transfer coefficient while in the thermally developing regime.

The thermally developing cases are for the 4-wall heated cases, whereas the CFD analysis is a 3-wall heated case. This difference, however, produces negligible differences as is seen when comparing the 3-wall heated and 4-wall heated fully developed cases. Comparing the constant heat flux and constant wall temperature thermally developing cases, negligible differences are observed. The constant heat flux case, which has been utilized in the previous sections, however, is advantageous as it is available in a wider range of non-dimensional streamwise coordinates \( z^* = z/(RePrD_h) \), to better reproduce the inlet characteristics and for a wide range of aspect ratios. These results have demonstrated that for parallel channel systems, when optimizing for a channel with uniform cross-section, consideration of the thermal entrance region is imperative for accurate comparisons. This conclusion is also expected for parallel channels with varying cross-sectional areas.

Figure 7.18 compares the fully developed and thermally developing one-dimensional
Figure 7.18: Radial channel: One-dimensional thermal resistance model comparison with three-dimensional conjugate analysis.
thermal resistance models with the three-dimensional conjugate analysis for radially inward flow configuration. Two cases are considered, a uniform channel ($\alpha = 0.5$) and a converging channel with unity inlet aspect ratio. Similar to the parallel channel cases, the trend and magnitudes are not well predicted by the fully developed flow cases when compared to the three-dimensional conjugate analysis. In addition, however, the thermally developing cases do not provide a good match with this analysis. To investigate this further, plots of the resulting heat flux in the conjugate analysis compared to a uniform heat flux distribution are presented in Figure 7.19. It is observed in this figure that in both the parallel and radial channel cases, the heat flux is non-uniform. However, in the case of the radial channel it is significantly greater. The mean deviation, defined as

$$MD = \frac{1}{L_{chn}} \int_0^{L_{chn}} \left( q''_{CFD} - q''_{1D,Unifom} \right) dz,$$

(7.18)

is calculated to be 25% for the radial channel compared to 2.9% for the parallel channel. This non-uniformity is due primarily to axial heat conduction within the solid travelling towards the channel entry. This can be expected particularly in the case of the radial channels, as a large part of the volume is solid and this region gets larger towards the inlet. In summary, the radial model which is an adaptation of the parallel channel model does not provide good predictability of the surface temperature, due to the assumption of negligible wall thickness. This negligible wall thickness translates into an assumption of a uniform heat flux distribution at the inner wall surface, which is a stark difference from the physical reality.
Figure 7.19: Resulting inner wall heat flux distribution from three-dimensional conjugate analysis compared to uniform one-dimensional heat flux distribution.
7.5 **Summary**

The design of a radial microchannel heat sink has been presented. Fabrication trials were performed but due to delamination of the various layers, experimental investigations were not pursued. The optimization of a radial inflowing microchannel heat exchanger has been investigated numerically. Unique to this optimization was consideration of channels with axially varying cross-sections. Three-dimensional conjugate analysis shows that when constrained by a fixed channel outlet area, increasing the channel inlet area will improve the thermal performance. This is due to increased mass flow providing for a decreased calorimetric resistance.

A one-dimensional thermal resistance model was explored for rapid optimization of channels with varying cross-section, for radially inflowing microchannel heat exchangers. This model is an adaptation of a previous model for parallel channel systems. In general, the analysis shows that linearly converging channels are superior to uniform channels. However, the model predictions do not fair well when compared to the three-dimensional conjugate analysis. This is mainly due to the assumption of a negligible wall thickness thermal resistance, which neglects the significant axial conduction possible in a radially inflowing configuration.
Chapter 8

Closing

8.1 Conclusions and Contributions

A state-of-the-art test facility has been developed for local heat transfer measurements in microchannel systems. The facility can accommodate a number of fluids including air, water and engineered fluids. Single phase and flow boiling measurements may be investigated with the facility.

The facility utilizes a unique Thermochromic Liquid Crystal technique for surface temperature measurement. High spatial resolution measurements, necessary for microgeometries, are enabled through use of the liquid crystal material in its un-encapsulated form. Test methods for using the un-encapsulated form of liquid crystal have been advanced through the development of a localized calibration procedure to account for inconsistencies in coating thickness and lighting, as well, through measurements on microtubes, down to 250 μm and microchannel heat sinks with 3.5 cm² footprint areas.
Single phase flow laminar heat transfer in microtubes down to 250 μm, is found to be well represented by conventional analytical treatment. The average turbulent heat transfer coefficient in microtubes down to 500 μm, is well represented using the correlation by Gnielinski (1976) and Adams et al. (1998).

The use of un-encapsulated TLC's has aided in partial flow visualization of flow boiling phenomena in a microtube. This includes a unique flow boiling onset of FC-72 in a 1 mm microtube while in a flow range of 770 kg/(m²·s) to 1040 kg/(m²·s). Wall temperatures during flow boiling of FC-72 in that range, were found to oscillate for a wide range of heat fluxes. The amplitude was on the order of 1 °C to 7 °C for low heat flux and the frequency around 10 Hz. The local heat transfer coefficient under flow boiling conditions for FC-72 in a 1 mm tube, has been presented for a quality up to 0.3. For the range of heat flux covered, the heat transfer coefficient is dependent on heat flux, but only mildly influenced by quality at a mass flux of 770 kg/(m²·s). At a mass flux of 1040 kg/(m²·s), the heat transfer coefficient is less dependent on the heat flux.

A cross-linked silicon microchannel heat sink was designed with 45, 250 μm wide channels and 3 cross-link paths with widths approximately equal to the channel width. Single phase flow in such a heat sink showed negligible differences to a standard heat sink of similar geometry, whereas, single phase heat transfer showed inflections in the wall temperature in the vicinity of the cross-links. Flow boiling in standard and cross-linked silicon parallel microchannel heat sinks has been identified as explosive, similar to other recent observations. This flow boiling phenomena is characterized by periodic oscillation in the fluid bulk parameters.
Flow boiling oscillations in a standard and a cross-linked silicon microchannel heat sink of related geometric configuration are similar. Both configurations show a decreasing oscillation frequency with increasing heat flux. Flow boiling oscillations for inlet temperature, outlet temperature and inlet pressure occur at the same frequency. For both configurations, the peak to peak oscillation amplitude increased with increasing heat flux. Flow boiling oscillation data for a standard silicon microchannel heat sink has been presented and correlated. The frequency of oscillation has been successfully correlated with the Boiling number, the Subcooling number, the Weber number and the number of channels. The inlet pressure amplitude has been successfully correlated with the Boiling number, the Subcooling number and the Weber number. The outlet temperature amplitude was successfully correlated with the Boiling number and the Subcooling number.

A radially inflowing microchannel heat exchanger was investigated through fabrication trials and a numerical study. A three-dimensional conjugate analysis of a radially inflowing microchannel heat exchanger was performed. Unique to this design was the incorporation of channels with linearly varying cross-sections. This analysis showed that when constrained by a fixed channel outlet area, increasing the channel inlet area will improve the thermal performance. The use of a one-dimensional thermal resistance model for optimization of radially inflowing microchannels with varying cross-section was explored. This model showed poor correspondence with the three-dimensional conjugate analysis and this was primarily attributed to its exclusion of axial conduction effects. These are found to be imperative for accurate modeling of radially inward flowing microchannel heat sinks.
8.2 Recommendations

8.2.1 TLC Methods

The biggest challenge in utilizing the un-encapsulated TLC material for thermal measurement is packaging it onto the surface of interest, while protecting it from environmental effects. Highly vibrant colour response and hence high signal to noise ratio, is possible when a transparent cover is placed in direct contact with the material. The use of transparent encapsulants compatible with TLC material should be investigated, as they offer the possibility for both an in-contact cover and protection from the environment. Packaging should be considered at the initial experimental design stages, when intending to use un-encapsulated TLC’s. The glass slides used to seal the heat sinks, in Section 6.3.1, were effective for providing a highly vibrant color response. However, over time the coating would decay partially due to expansion on heating, which would cause it to seep out of the edges. Use of a tape around the edges provided sufficient compressive seal to prevent this. The sealed coating in this form was found to be highly vibrant and responsive for extensive periods, on the order of months. Based on this, not only is a cover in direct contact necessary, but also the coating should be prevented from escaping. Covering and sealing are increasingly difficult as the geometry reduces in size and offer unique challenges to the experimenter. For the microtubes, future investigators may consider using glass capillary tubes as covers. The capillary tubes diameter could be strategically selected to provide a directly contacting cover for the coated microtubes.
In applying the TLC coating, repeated trials were sometimes necessary, particularly with the microtubes, to obtain the necessary color response. This, however, is time consuming and can also be costly. Rather, a standard coating machine could be developed to provide consistently reproducible coatings. For planar surfaces an airbrush on a traversing mechanism could be developed.

8.2.2 Microtubes

The single phase measurements presented, investigated circular microtubes only. The methods used could similarly be applied to non-circular microtubes. The investigator would, however, need to address the two-dimensional conduction effects at the corners.

The flow boiling microtube experiments were advantageous in developing understanding of heat transfer mechanisms in microchannels, due to the use of a single channel. An expansion of the flow boiling investigations in sub-millimeter microtubes should be performed over a wider range of conditions. In addition to the above, alternative cross-sections could also be investigated. This will aid in understanding the unique flow boiling onset phenomenon of FC-72 in the microtube. Additionally, there was a lack of correspondence of the flow boiling heat transfer coefficient correlations, when compared with the 1 mm tube data flowing FC-72. Expanding the investigation with a variety of fluids would aid in developing versatile correlations.
8.2.3 Microchannel Heat Sinks

A limited experimental program of the cross-linked heat sinks was presented due to challenges with chip support packaging. A compliant adhesive is recommended for permanent chip bonding, however non-permanent bonding methods would be the most ideal.

Future works should investigate the single and two-phase heat transfer of the cross-linked scheme under a wider range of conditions and with varying gap widths. Visualization of flow boiling phenomena and two-phase flow patterns should also be investigated with the second packaging method. Through the use of two cameras, or appropriate optics, simultaneous flow visualization and thermal measurements could be performed. The added perspective of such an approach on the flow boiling physics would be of great value.

Challenges in fabrication produced a low yield of the radial heat sink. A modification to the design that may ease fabrication challenges, is to have the entry or exit at the outer radius, rather than re-diverting the flow towards the top of the heat sink. This would then only require a two-layer, rather than a three-layer stack. An experimental program of the radial heat sink under single phase and flow boiling conditions should be pursued for further characterization. Experiments need to consider both radially inward and outward flows.

An experimental program with MEMS based heat sinks can be an expensive endeavor and so the geometries of interest need to be selected strategically. One-dimensional thermal resistance models offer the possibility of rapid optimization to
identify geometries of interest. The one-dimensional thermal resistance model explored for the radial heat sink was limited. A simplified model accounting for axial conduction needs to be developed to aid in the rapid optimization of channels with varying cross-sections. Additionally, numerical correlations need to be developed for various wall heating conditions, for channels of various aspect ratios and possibly shapes, under thermally and simultaneously flow developing conditions. Other microsystems may be exposed to different wall heating conditions and readily available correlations would aid in their geometrical optimization.

Only linearly varying channels were considered in the present numerical program. Experimental measurements need to be performed in microchannels of varying cross-sections both for parallel and radially flowing systems. Finally, micro-PIV investigations should also be pursued for the cross-linked and radial heat sinks to provide validation data for numerical investigations.
References


Publications from Thesis Work

Journal


Conference


Appendix A

Experimental Data

A.1 Single Phase Flow and Heat Transfer in Microtubes

Table A.1: 1 mm, Water.

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<th>$N_u_{avg}$</th>
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Ambient Temperature, 24.6 °C

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Table A.2: 0.50 mm, Water.

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Table A.4: 0.5 mm, FC-72.

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\[
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Ambient Temperature, 25.5 °C

259
## A.2 Flow Boiling in Microtubes

Table A.6: Local heat transfer coefficient, 770 kg/(m²·s).

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### A.3 Microchannel Heat Sink Heat Transfer and Boiling Oscillations

Table A.8: Single phase heat transfer with FC-72, INT02, $Re = 142$.

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Table A.9: Single phase heat transfer with FC-72, INT02, \( Re = 210 \).

\[
\begin{align*}
Q_{vol} &= 60 \text{ ml/min} \\
Q &= 13.3 \text{ W} \\
T_{in} &= 26.4^\circ C
\end{align*}
\]

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Table A.10: Single phase heat transfer with FC-72, INT02, $Re = 307$.

\[
\begin{align*}
Q_{vol} &= 85 \text{ ml/min} \\
Q &= 13.7 \text{ W} \\
T_{in} &= 28.0^\circ C
\end{align*}
\]

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267
Table A.11: Single phase heat transfer with FC-72, INT02, Re = 559.

\[ Q_{vol} = 150 \text{ ml/min} \]
\[ Q = 15.5 \text{ W} \]
\[ T_{in} = 30.2^\circ C \]

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Table A.12: Flow boiling oscillations, STR02, $T_{in} = 70^\circ C$.  

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Table A.13: Flow boiling oscillations, STR02, $T_{in} = 80^\circ C$.

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<td>12.15</td>
<td>0.032551</td>
</tr>
<tr>
<td>$G = 185$ kg/(m²·s)</td>
<td>5.7016</td>
<td>0.000136</td>
<td>0.365829</td>
<td>0.0301</td>
<td>4.83406</td>
<td>0.133196</td>
<td>9.56</td>
<td>0.025616</td>
</tr>
<tr>
<td>$N_{sub} = 59.890$</td>
<td>5.9725</td>
<td>0.000142</td>
<td>0.225126</td>
<td>0.0185</td>
<td>4.36540</td>
<td>0.120282</td>
<td>11.37</td>
<td>0.030469</td>
</tr>
<tr>
<td>$We = 0.1703$</td>
<td>6.4778</td>
<td>0.000154</td>
<td>0.205025</td>
<td>0.0169</td>
<td>4.70768</td>
<td>0.129713</td>
<td>11.09</td>
<td>0.029708</td>
</tr>
<tr>
<td></td>
<td>6.922</td>
<td>0.000165</td>
<td>0.180905</td>
<td>0.0149</td>
<td>5.27640</td>
<td>0.145383</td>
<td>11.77</td>
<td>0.031546</td>
</tr>
<tr>
<td>$G = 228$ kg/(m²·s)</td>
<td>8.1815</td>
<td>0.0001587</td>
<td>0.393970</td>
<td>0.0264</td>
<td>4.79720</td>
<td>0.088083</td>
<td>12.30</td>
<td>0.032953</td>
</tr>
<tr>
<td>$N_{sub} = 59.890$</td>
<td>8.8191</td>
<td>0.000171</td>
<td>0.237186</td>
<td>0.0159</td>
<td>5.42910</td>
<td>0.099686</td>
<td>12.32</td>
<td>0.033005</td>
</tr>
<tr>
<td>$We = 0.2556$</td>
<td>9.7413</td>
<td>0.0001889</td>
<td>0.168844</td>
<td>0.0113</td>
<td>5.83984</td>
<td>0.107228</td>
<td>13.65</td>
<td>0.036580</td>
</tr>
<tr>
<td></td>
<td>9.7384</td>
<td>0.0001889</td>
<td>0.172864</td>
<td>0.0116</td>
<td>5.83984</td>
<td>0.107228</td>
<td>14.70</td>
<td>0.039406</td>
</tr>
</tbody>
</table>
Table A.14: Flow boiling oscillation, INT01.

\[
T_{in} = 70^\circ C
\]
\[
G = 140 \text{ kg/(m}^2 \cdot \text{s)}
\]

<table>
<thead>
<tr>
<th>( q'' ) (W/cm(^2))</th>
<th>( f_\omega ) (Hz)</th>
<th>( T_{amp,\text{in}} ) (K)</th>
<th>( T_{amp,\text{out}} ) (K)</th>
<th>( P_{amp,\text{in}} ) (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.6334</td>
<td>0.3308</td>
<td>5.9</td>
<td>16.7</td>
<td>5.40730</td>
</tr>
<tr>
<td>7.8571</td>
<td>0.3008</td>
<td>8.5</td>
<td>17.7</td>
<td>6.16020</td>
</tr>
<tr>
<td>8.1260</td>
<td>0.2406</td>
<td>9.2</td>
<td>19.1</td>
<td>6.62880</td>
</tr>
<tr>
<td>8.4066</td>
<td>0.2206</td>
<td>9.7</td>
<td>19.3</td>
<td>6.83410</td>
</tr>
<tr>
<td>8.6896</td>
<td>0.1905</td>
<td>11.3</td>
<td>19.5</td>
<td>7.16580</td>
</tr>
<tr>
<td>8.9529</td>
<td>0.1805</td>
<td>11.9</td>
<td>20.3</td>
<td>7.33430</td>
</tr>
<tr>
<td>9.1153</td>
<td>0.1805</td>
<td>12.9</td>
<td>20.4</td>
<td>7.01840</td>
</tr>
<tr>
<td>9.5861</td>
<td>0.1604</td>
<td>14.9</td>
<td>20.4</td>
<td>7.40270</td>
</tr>
</tbody>
</table>
Appendix B

Example Code

% TLC Rig
% Author: Roland S Muwanga
% Date: Jan 23/07
% Based on Jun 30/06 and Sept 14/05
%
% Program: calibproc***
%
% Description: File to post-process TLC calibrator data
%
% Requires:
% 1. Calibration images and file location
% 2. image location rotation
%
% Promises:
% 1. calculate ROI calibration curves and save them for multiple image location
% 2. files saved with number equal to location number N x N polynomial equations
% 3. filepolycoeff - Output file of polynomial coefficients
%
% Additional Notes:
% i. to be run from calibration data directory
% ii. to run type filename without '.m' at Matlab prompt

close all;
warning off MATLAB:polyfit:RepeatedPointsOrRescale
clear all;
% INPUTS
Nimg = 21; % number of images
Nloc = 1; % number of image locations for measurement
NlocCalib = 1; % number of calibrated locations
krepresent = 0; % location which will be used for calibration
% representation of uncalibrated locations
widthIMG = 640; heightIMG = 72; % TLC ROI of interest in pixels
dystart = 2; % number of pixels relative to y-coordinate
dx = 4; % interrogation ROI dimensions for calibration in pixels
% (needs to be divisible by respective widthIMG and heightIMG)
dy = 3;
Nx = widthIMG/dx; Ny =heightIMG/dy;

% Filename inputs
datafilename = 'calib.lvm'; % calibration data filename (*.lvm)
% from LabVIEW - user input from LabVIEW
imagefilename = 'calib_0'; % Beginning of Hue image name (*.tif) to
% before increments - user input from LabVIEW
imagefilenameRGB = 'calib_0_rgb'; % Beginning of RGB image name (*.TIFF)
% to before increments
% - user input from LabVIEW
calibhuefile = 'calibhue_4x3_0'; % ROI 4x3 lots of small region | one
% file per location - program created
filepolycoeff = 'polycoeff_'; % calibration polynomial coefficients |
% one file per location -
% program created (4x3)
coordinatesfile = 'coordcalib.txt'; % coordinates file - user creatd

% Process inputs
templstyle = 'inlet_temps';
% choice between (1)average - 'average_temps'; (2)inlet temperature -
% inlet_temps; (3)exit temperature - exit_temps;
% (4)interpolated - interp_temps
storedhuedata = 'yes';
savepolycoeffs = 'yes'; % save generated polynomial coefficients
SmoothZeros = 'yes'; % smooth zero polynomial coefficients
fitpolys = 'yes'; % generate polynomial coefficients
readpolys = 'no'; % read in poly coefficients for smoothing
xstart = 0;
rdist = 320;
Nsmooths = 10; % number of smoothings to do to get rid of
% zero polynomial coefficients
fitorder = 5; % initial polynomial order of curve fit

% Additional corrections
% To obtain improved curve fit
ZeroTemp = 38; % TLC restart
nzeros = fitorder - 3;
imgDisp = [1 2 3 4 5 6 7]; % location number RGB cropped image to be
% viewed
addforcedhue = 'true'; % Impose peak asymptotic temp for better fit
deltaforcehue = 25; % hue addition at which peak asymptotic hue will
% be added
Tempforce = 53; % Imposed peak asymptotic temp for better curve fit
FindZeroLow = 0; % lower limit to search for peaks and troughs in
% fitted curves
FindZeroHigh = 175; % upper limit to search for peaks and troughs in
% fitted curves
Minus360HueUpper = 300; % upper limit for hue value at which negative
% angle is used
dTtempcalibration = 1.5; % minimum increment in calibration images;
% prevents extraneous points from being
% included

% Optional
imgview = 'yes'; % 'yes' or 'no'
imgview_kval = 1;

% REMINDERS
% i) change imglocArr appropriately
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Calibration data file constants %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
nrows = 4;
headerlines = 23;
headercolumns = 0;
numblinesdatfile1 = 250; % because joining two files
imgMid = 10000; % number of images in 1st round / put to high number if
% only one round
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Initialize coordinates %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
[xindArr1 yindArr1] = textread(coordinatesfile...
', '%f %f', 'delimiter', '	');
xleftArr = linspace(0, 0, Nloc);
xrightArr = zeros(size(xleftArr));
yleftArr = zeros(size(xleftArr));
yrightArr = zeros(size(xleftArr));
rotArr = zeros(size(xleftArr));
rotArr1 = zeros(size(xleftArr));
% Read coordinates file into variables
j=1; % Nloc / 1, depends on which is first, for last location 1st use
% 'Nloc'
for i=1:2:2*Nloc
xleftArr(j) = xindArr1(i);
xrightArr(j) = xindArr1(i+1);
yleftArr(j) = yindArr1(i);
yrightArr(j) = yindArr1(i+1);
rotArr(j) = atan((yrightArr(j)-ylleftArr(j))/(xrightArr(j)-...)
xleftArr(j)))*180/pi;
theta = rotArr(j)*pi/180;
dyrot = rdist*sin(theta);
yposArr(j) = yrightArr(j) - floor(dyrot)+dystart;
j = j-1;
end

% Array initializations
tempArr = zeros(1,Nloc);
polycoeff = zeros(Nx*Ny,fitorder+1,Nloc);

% Read calibration file
% skip first and last point as different size columns than the rest
Tamb = linspace(0,0,Nimg); leftknob = linspace(0,0,Nimg); rightknob =...
linspace(0,0,Nimg);
imgLocArr = linspace(0,0,Nimg);
counter2Img = linspace(1,1,Nloc); % re-initialize counter for image
% locations
[Xval TempIn TempOut Pout Pin Pdiff Voltage Current TubeTemp...
preheater1 preheater2 TankTemp PumpOutTemp PumpInTemp Tamb Tank2Temp...
PumpOutTemp flowalnminTestInSigma TestOutSigma PressOutSigma...
PressInSigma PressDiffSigma VoltsSigma CurrentSigma Pbarometer...
leftknob rightknob casenumb Comment] = txtread(datafilename,'%f %f...
%f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f
%f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f
imgLocArr = leftknob-rightknob+1; %unique function to denote the number
% of locations, starting from 1 -> to create integer indexing
% Counter initializations
counterImg = linspace(0,0,Nloc);
for k=1:Nimg
counterImg(imgLocArr(k)) = counterImg(imgLocArr(k))+1;
end

275
Tamb = linspace(0,0,Nimg); leftknob = linspace(0,0,Nimg); rightknob = ...
        linspace(0,0,Nimg);
for k=1:Nimg
    indexCount = imgLocArr(k);
    switch tempstyle
        case 'average_temps' % Recommended if temperature difference is
            % ~0.5C or less
calibdata = (TempIn(k)+ TempOut(k))/2;
        case 'inlet_temps' % Recommended if concerned with inlet region
            % and larger temperature difference
calibdata = TempIn(k);
        case 'exit_temps' % Recommended if concerned with exit region
            % and larger temperature difference
calibdata = TempOut(k);
        case 'interp_temps' % treat as linear drop in temperature -
            % Recommended if concerned region and larger
            % temperature difference
calibdata = (TempIn(k) - TempOut(k))/(100)*(leftknob(k)+...  
                   rightknob(k))+TempIn(k);
end
    indexCount = imgLocArr(k);
    tempArr(counter2Img(indexCount),indexCount) = calibdata;
    counter2Img(indexCount) = counter2Img(indexCount)+1;
end
NimgForLoc = max(counterImg);
disp('finished reading in temperature data');
counter2Img = linspace(1,1,Nloc); % re-initialize counter for image
% locations
counter = 1; % counter for ROI locations
hueImgArr = zeros(NimgForLoc,Nx*Ny,Nloc); % hue array [(No of images for
% locs)x(NxNy)x
% (No of image locs)]
xind = xstart;

% Parsing out and storing hue dat for each ROI
kimg = 1;
if strcmp(storedhuedata,'yes')
    for i=1:Nimg
        kimg = i;
        indexCount = imgLocArr(i);
yind = yposArr(imgLocArr(i));
Filename = [imagefilename int2str(kimg) '.tif'];
FilenameRGB = [imagefilenameRGB int2str(kimg) '.TIFF'];
imgA = imread(Filename,'tif');
imgA = imrotate(imgA,rotArr(indexCount),'crop');
for i = imgDisp(indexCount);
    imgRGB = imread(FilenameRGB,'tif');
    imgRGB = imrotate(imgRGB,rotArr(indexCount),'crop');
    imgRGB = imcrop(imgRGB,[xind yind widthIMG heightIMG]);
    imshow(imgRGB);
end
for j=1:Nx
    for k=1:Ny
        imgCrop = imcrop(imgA,[xind yind dx dy]);
        imgDbl = double(imgCrop);
        sizeArr = size(imgDbl); rowsize = sizeArr(1); colsize = sizeArr(2);
        hueDat = reshape(imgDbl,rowsize*colsize,1);
        roiMedian = median(hueDat)/255*360;
        hueImgArr(counter2Img(indexCount),counter,indexCount) = roiMedian;
        yind = yind+dy;
        counter = counter+1;
    end
    xind = xind+dx;
    yind = yposArr(indexCount);
end
counter2Img(indexCount) = counter2Img(indexCount)+1;
counter = 1;
xind = xstart;

% Writing
for k = 1:NlocCalib
    Filename = [calibhuefile int2str(k) '.txt'];
    dlmwrite(Filename,hueImgArr(:, :, k),'	');
end
if NlocCalib < Nloc % if there are more measured than calibrated
    % locations represent uncalibrated with location krepresent right
    % now assumes representing location is last location
    for k = 1:krepresent-1
        Filename = [calibhuefile int2str(k) '.txt'];
        hueImgArr(:, :, k) = hueImgArr(:, :, krepresent);
        dlmwrite(Filename,hueImgArr(:, :, k),'	');
        counterImg(k) = counterImg(krepresent);
        tempArr(:, k) = tempArr(:, krepresent);
    end
end
disp('finished writing hue data...beginning poly curve fitting');
else
    % Reading
    for k = 1:Nloc
        Filename = [calibhufile int2str(k) '.txt'];
        hueImgArr(:,:,k) = dlmread(Filename,'\t');
    end
    disp('finished reading in saved hue data....');
end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Polynomial fitting %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
donething = 0;
if strcmp(fitpolys,'yes')
    disp('beginning polynomial curve fitting.....');
    counterzero = 0; counter3 = 0; counter5 = 0; counterback = 0;
    hueX = linspace(0,260,30);
    for k = 1:Nloc
        for i = 1:Nx*Ny
            %%% Initially assume 5th order fit %%%
            hueDat_orig = hueImgArr(1:counterImg(k),i,k);
            tempDat_orig = tempArr(1:counterImg(k),k);
            hueDat_old = hueImgArr(1:counterImg(k),i,k);
            tempDat_old = tempArr(1:counterImg(k),k);
            sizearr = size(tempDat_old); sizearr = sizearr(1);
            sizearrCount = sizearr;
            for j = 1:sizearr
                if hueDat_orig(j) > Minus360HueUpper;
                    hueDat_old(j) = 360 - hueDat_old(j);
                end
            end
            hue_temp_arr = [hueDat_old tempDat_old];
            hue_temp_sorted = sortrows(hue_temp_arr,1);
            % sorts data with respect to increasing hue values
            hueDat = hue_temp_sorted(1,1); tempDat =...
            hue_temp_sorted(1,2);
            jprime = 2;
            for j = 2:sizearr
                if hue_temp_sorted(j,2) > tempDat(jprime-1) \&\&...
                    (hue_temp_sorted(j,2)-...
                        tempDat(jprime-1))<dTtempcalibration
                        % if the temperature increment is positive and less
                        % than 1.5C the increment is a valid point
                        hueDat(jprime) = hue_temp_sorted(j,1);
                        tempDat(jprime) = hue_temp_sorted(j,2);
                        jprime = jprime+1;
        end
    end

278
end
end
sizenew = size(huedat);
if strcmp(addforcedhue,'true');
    huedat = [huedat huedat(sizenew(2))+deltaforcehue];
    tempdat = [tempdat Tempforce];
end
polycoefftest = polyfit(huedat,tempdat,fitorder);
flag = '5th';

%%% Check to verify real numbers else set to zero %%%
isnanSum = sum(isnan(polycoefftest));
polycoefftest2 = 0;
if isnanSum > 0
    polycoeffnew = 0;
    counterzero = counterzero+1;
else
    %%% If roots to derivative assume 3rd order fit %%%
polyderiv = polyder(polycoefftest);
isnanSum = sum(isnan(polyderiv));
if isnanSum == 0
    tempY = polyval(polycoefftest,hueX);
    dTempY = polyval(polyderiv,hueX);
    rootvals = roots(polyderiv);
    n=1;
    while n < fitorder
        if isreal(rootvals(n)) && rootvals(n) >...
            FindZeroLow &....
            rootvals(n) < FindZeroHigh
            % if 5th has root stick
            % with 3rd, should be 25-175
            flag = '3rd';
            polycoefftest2 = [linspace(0,0,nzeros)...
                polyfit(huedat,tempdat,3)];
            polyderiv = polyder(polycoefftest2);
            isnanSum = sum(isnan(polyderiv));
        end
    end
    if isnanSum == 0
        % if can fit 3rd order see if has roots
        n = n+1;
        tempY = polyval(polycoefftest2,hueX);
        dTempY = polyval(polyderiv,hueX);
        rootvals2 = roots(polyderiv);
        y = 1;
        while y <= 2
            if isreal(rootvals2(y)) &...
rootvals2(y) >...
FindZeroLow & &...
rootvals2(y) < FindZeroHigh
y = y+1;
% forcing the 0 hue to ZeroTemp
tempdatchek =...
[ZeroTemp tempdat];
huedatchek = [0 huedat];
polycoeftest3 =...
[linspace(0,0,nzeros) ... polyfit(huedatchek,tempdatchek,3)];
polyderiv =...
polyder(polynomialtest);
isnanSum = sum(isnan(polyderiv));
if isnanSum == 0
tempY =...
polyval(polycoeftest3...
 ,hueX);
dTempY =...
yval(polyderiv,hueX);
rootvals3 = roots(polyderiv);
z = 1;
while z <= 2
% see if forcing the 0
% hue to ZeroTemp solves
% problem + new range
if...
isreal(rootvals3(z)) ...
& & rootvals3(z) >...
FindZeroLow & & rootvals3(z) <...
FindZeroHigh
z = 3;
y = 3;
% third fails,
% kick out of all
n = fitorder;
% third fails, kick
% out of all
flag = 'old';
else
flag = '3rdMOD';
z = z+1;
end
end
else
y = 3;
n = fitorder;
flag = 'old';
    end
  else
    y = y+1;
  end
end
else
  flag = 'old';
  n = fitorder;
  % 3rd no good forcing kick out
end
else
  n = n+1;
end
else
  polycoeffnew = 0;
  flag = 'old';
end
switch flag
  % -Sets the polynomial type to:
  % Modified 3rd order (3rdMOD), 3rd order,
  % or 5th order
  % -If dataset is junk (e.g. max hue < prescribed)
  % treats as a zero
  case '3rdMOD'
    if max(huedat) < 50;
      polycoeffnew = 0;
      %disp('zero');
      counterzero = counterzero+1;
    else
      polycoeffnew = polycoefftest3;
      %disp('3rd');
      counter3 = counter3+1;
    end
  case '3rd'
    if max(huedat) < 50;
      polycoeffnew = 0;
      %disp('zero');
      counterzero = counterzero+1;
    else
      polycoeffnew = polycoefftest2;
      %disp('3rd');
      counter3 = counter3+1;
    end
  case 'old'
    if max(huedat) < 50;
polycoefficient = 0;
"disp('zero');
counterzero = counterzero+1;
else
    polycoefficient = 0;
    "disp('old');
counterback = counterback+1;
end

    case '5th'
        if max(huedat) < 100;
            polycoefficient = 0;
            "disp('zero');
counterzero = counterzero+1;
        else
            polycoefficient = polycoefficient;
            "disp('5th');
counter5 = counter5+1;
end

    end

    if strcmp(imgview,'yes') && k > imgview_kval
        tempfinal = polyval(polycoefficient,hueX);
temp3 = polyval(polycoefficient,hueX);
temp5 = polyval(polycoefficient,hueX);
disp(tempfinalnumb = ['ROI no. ' num2str(i)...
        ' location ' num2str(k)];
disp(tempfinalnumb);
plot(huedat,tempdat,'bo',hueX,temp5, 'r-',...,
hueX,temp3,'y-',...
hueX,tempfinal,'g-',huedat_orig,tempdat_orig,...
'm*');
legend('corrected data','5th order','3rd order',...
'final curve','original data');
axis([0 360 35 55]);
pause;
end
end % 1st inan if statement
polycoefficient = polycoefficient;
end

    if strcmp(savepolycoeffs,'yes')
        disp('finished poly coefficient determination....
writing out coefficients');
        for k = 1:Nloc
            Filename = [filepolycoefficient int2str(k) '.txt'];
dlmwrite(Filename,polycoefficient(:,k),'	');
end

disp('finished gen & saving poly coeff...plotting & ending');
else
disp('finished generating...plotting and ending');
end

disp(['counter zero = ' num2str(counterzero)]);
disp(['counter backwards = ' num2str(counterback)]);
disp(['counter 3rd order = ' num2str(counter3)]);
disp(['counter 5th order = ' num2str(counter5)]);
countertotal = counterzero+counterback+counter3+counter5;
disp(['total number of locations is ' num2str(countertotal)]);
end

if strcmp(SmoothZeros,'yes')
disp('reading in polynomial coefficients for smoothing');
if strcmp(readpolys,'yes')
    for k = 1:Nloc
        Filename = [filepolycoefficient int2str(k) '.txt'];
        polycoefficient(:,:,k) = dlmread(Filename,'\t');
    end
end

counterzero2 = 0;
disp('finished reading poly coefficients for... smoothing...beg smoothing');
for k=1:Nloc
    for i = 1:Nx*Ny
        if polycoefficient(i,:,k) == 0
            counterzero2 = counterzero2+1;
        end
    end
end

disp(['number of zeros initially = ' num2str(counterzero2)]);
counterzero2 = 0;
for k=1:Nloc
    for h = 1:Nsmooths
        for i = 1:Nx*Ny
            if polycoefficient(i,:,k) == 0
                rowi = ceil(i/Nx);
                coli = Nx + (i-Nx*rowi);
                if i == 1
                    kappa = 2;
                elseif coli == 1
                    kappa = i-Nx;
                else
                    kappa = i+1;
                end
            end
        end
    end
end
polycoeff(i,:,k) = polycoeff(kappa,:,k);
end
end
for i = 1:Nx*Ny
    if polycoeff(i,:,k) == 0
        rowi = ceil(i/Nx);
coli = Nx + (i-Nx*rowi);
    if i == 1
        kappa = 2;
    elseif coli == Nx
        kappa = i-1;
    else
        kappa = i+1;
    end
    polycoeff(i,:,k) = polycoeff(kappa,:,k);
end
end
counterzero2 = 0;
for k=1:Nloc
    for i = 1:Nx*Ny
        if polycoeff(i,:,k) == 0
            counterzero2 = counterzero2+1;
        end
    end
    disp(['counter zero second time = ' num2str(counterzero2)]);
    disp('...writing out coefficients');
    for k = 1:Nloc
        Filename = [filepolycoeff int2str(k) '.txt'];
dlmwrite(Filename,polycoeff(:,k),'	');
    end
    disp('finished generating smoothing and saving polynomial coefficients...
plotting and ending');
end
disp('Good bye');
% END
% TLC Rig
% Author: Roland S Muwanga
% Date: July 05/06
% Based on Oct 05, 2005 and Sept 28/05
% Program - datprocboil***
% Description: Main program for post processing heat transfer data from
% a microtube flow boiling experiment
%
% Requires:
% 1. Calibration output file of polynomial coefficients from calibproc***
% 2. Outputed data file from LabVIEW - DAQMain_Boiling02.vi or similar
% 3. NING number of RGB and hue TLC images
% 4. Coordinates file identifying area of interest within images
%    - coordinates file contains 2 coordinates for each traversed location
% 5. nucalboil_example.m or similar program for calculating the local
%    heat transfer coefficients is available
%
% Promises:
% 1. To calculate the local heat transfer coefficient from a flow boiling
%    microtube experiment
% 2. Output - 'NussStrm' which is the averaged streamwise Nusselt number
%
% Additional Notes:
% i. to be run from measuremetn data directory
% ii. to run type filename without '.m' at Matlab prompt

close all;
regeneratedata = 'true'; % true/false - if data already loaded can set
% to false and continue with postcalculations
postcalculations = 'false'; % may include calculations for correlation
% comparison and plotting

if strcmp(regeneratedata,'true')
    close all;
clear all;

    inputfile = 'meas_mod.lvm'; % output data file from LabVIEW
    imagenumber = 1; % image no. to start at
    coordfile = '..\Calibration\coordinatet.txt'; % coordinates file
    calibfile = '..\..\July0506_1.00BoilCalibRetrial\Calibration\polycoeff_mod_';
% output from calibration post processing

FiguresFilename = 'Figures\all_img'; % output filename for
    % post-processed figure
fileOutput = 'SavedData\100FC72boilout.txt'; % miscellaneous
    % output data
    % filename
imgname = 'meas_0'; % beginning of image filename
writefigures = 'true'; % true/false to write post-processed figures
postcalculations = 'true'; %true/false - see above

Nimg = 104; % number of images
Nloc = 7; % number of traversed locations
deltay = 72; % height of imaged location of interest (pixels)

Dh = 1.0668e-3; % hydraulic diameter (m)
Lhydraulic = 152.4/1000; % tube length (m)
heaterL = 0.106; % heated tube length (m)
powerratio = 0.95; % should be less than 1;
    % resistanceTube/resistanceTotal
umperpix = 15.528; % micrometers per pixel - resolution
pixperum = 1/umperpix; % pixles per micrometer

xinpix = 380; % based on (0,0) location at inlet
deltaxin = ((640-xinpix)*umperpix/1000)*(-1);
xlength = 640; % image length (pixels)
rdist = xlength/2; % distance from center of image rotation (pixels)
xorigin = 0;
xstart = 0; % number of pixels in length direction into the ROI to
    % to parse out ROI
dystart = 1; % number of pixels in height direction into the ROI to
    % to parse out ROI
streamX = xlength-xstart; % length of imaged region to present
    % (pixels)
streamY = deltax; % height of imaged region to present
    % (pixels)
dxCalib = 4; %calibration streamwise pixel difference (ROI)
dyCalib = 3; %calibration spanwise pixel difference (ROI)

% Read in LabVIEW data file
numheaderlines = 23; % header lines in LabVIEW data file
[Xval TempIn TempOut Pout Pin Pdiff Voltage Current TubeTemp...
    preheater1 preheater2 TankTemp PumpOutTemp PumpInTemp...
    AmbientTemp Tank2Temp PumpOutTemp2 flowalmin TestInSigma...
    TestOutSigma PressOutSigma PressInSigma PressDiffSigma...
    VoltSigma CurrentSigma Pbarometer leftknob rightknob casenumb...
Comment] = textread(inputfile,'%f %f %f %f %f %f %f %f %f %f %f %f ... %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f %f
'headerlines',numheaderlines,'delimiter','\t');

% clear unused parameters
clear Xval Preheat1 Preheat2 Temp1 Temp2 TankTemp PumpInTemp... PumpOutTemp Comment;

% Initialize coordinates
imgLocArr = round((leftknob+rightknob+5)/10)-3;
xin = (leftknob + rightknob + deltaxin)/1000

% Read in coordinates data file
[xIndArr yIndArr] = textread(coordfile,'%f %f','delimiter','\t');
xleftArr = linspace(0,0,NLoc);
xrightArr = zeros(size(xleftArr));
yleftArr = zeros(size(xleftArr));
yrightArr = zeros(size(xleftArr));
rotArr = zeros(size(xleftArr));

j=NLoc; % NLoc/1, depends on which is first, for last location 1st
% use 'NLoc'
for i=1:2:2*NLoc
xleftArr(j) = xIndArr(i);
xrightArr(j) = xIndArr(i+1);
yleftArr(j) = yIndArr(i);
yrightArr(j) = yIndArr(i+1);
rotArr(j) = atan((yrightArr(j)-yleftArr(j))/(xrightArr(j)-...
xleftArr(j)))*180/pi;
theta = rotArr(j)*pi/180;
dyrot = rdist*sin(theta);
yposArr(j) = yrightArr(j) - floor(dyrot);
j = j-1; % +/- depends on which is first - for last location
% 1st use '+'
end
xInd = xstart; %yInd = yposArr(imgLocArr(1))+dystart;
Qf1Arr = .flowmlmin*(1e-6)/60; % convert flowrate to m3/s

% Initialize arrays of various parameter
xstar = zeros(Nimg,xlength); % x/RePrDh
xd = zeros(Nimg,xlength); % x/D for a given image
PrLocalArr = zeros(Nimg,xlength); % Prandtl number
xqualityArr = zeros(Nimg,xlength); % thermodynamic quality
rhoFLocalArr = zeros(Nimg,xlength); % liquid density
CpLocalArr = zeros(Nimg,xlength); % fluid specific heat
kcondLocalArr = zeros(Nimg,xlength); % liquid thermal conductivity
sigmaLocalArr = zeros(Nimg,xlength); % fluid surface tension
muFLocalArr = zeros(Nimg,xlength); % liquid viscosity
rhoVLocalArr = zeros(Nimg,xlength); % vapor density
hfgLocalArr = zeros(Nimg,xlength); % fluid latent heat of vaporization

AvgNuThresh = linspace(0,0,Nimg); % Average Nu from AvghtcThresh
AvghtcThresh = linspace(0,0,Nimg); % Average htc from AvgTwThresh
AvgTwThresh = linspace(0,0,Nimg); % Average wall temperature
% thresholded between min and max
% temperature limits

ReynNum = linspace(0,0,Nimg); % Average Reynolds number
NuGnielArr = linspace(0,0,Nimg); % Nu from Gnielinski correlation
ImgHueAvg = linspace(0,0,Nimg); % Image average Hue value
TwStrm = zeros(size(xstar)); % Averaged streamwise wall temperature
TsatArr = zeros(size(xstar)); % Saturation temperature
TbArrAll = zeros(size(xstar)); % Fluid bulk temperature
TbArrtheo = zeros(size(xstar)); % Fluid bulk assuming no boiling
maxdelNuArr = zeros(size(ReynNum)); % Maximum Nu uncertainty
maxdelTwArr = zeros(size(ReynNum)); % Maximum Tw uncertainty
rhoIn = zeros(size(ReynNum)); % Inlet liquid density

frictfactArr = linspace(0,0,Nimg); %initialize friction factor array
Pdiff = (Pdiff+Pcorrection)*6894.75729317; %psi to Pascals
Pout = Pout*6894.75729317; %psi to Pascals
Pin = (Pin+Pcorrection)*6894.75729317; %psi to Pascals

Pbarometer = Pbarometer*6894.75729317; %psi to Pascals
PinCorr = Pout./Pout - 1; % zero array size of Pout
PinDelta = Pout./Pout - 1; % zero array size of Pout
PoutDelta = Pout./Pout - 1; % zero array size of Pout

% begin iterating through images
for i=imgstart:Nimg
  % Read in hue image
  indexCount = imgLocArr(i);
  dispimagenumb = ['image no.' num2str(i)];
  disp(dispimagenumb);
  Filename = [imgname int2str(i) '.tif'];
  imgA = imread(Filename,'tif');

  % Rotate and crop hue image
  imgA = imrotate(imgA,rotArr(indexCount),'crop');
  imgCrop = imcrop(imgA,[xorigin yposArr(indexCount) streamX...
                      streamY]);
  imgInt = fliprl(imgCrop);
  imgDb1 = double(imgInt)*360/255; % convert to degrees scale
% Read in RGB image
RGBname = [imagename '_rgb' int2str(i) '.TIFF'];
RGBimg = imread(RGBname,'tif');

% Rotate and crop RGB image
RGBrot = imrotate(RGBimg ,rotArr(indexCount),'crop');
RGBcrop = imcrop(RGBrot,[xorigin yposArr(indexCount) streamX...
streamY]);
for k = 1:3
RGBflip(:,:,k) = fliplr(RGBcrop(:,:,k));
end

imgSize = size(imgDb);
xlengthI = imgSize(2);
xstar = xstar(:,1:xlengthI);
xd = xd(:,1:xlengthI);

% Read in polynomial coefficients
polycoeffFile = [calibfile int2str(indexCount) '.txt'];
polycoeff = dlmread(polycoeffFile,'	'); % calibration

% Calculate friction factor and correct for entry/exit losses
[tempIn(i), tempOut(i), QflArr(i), Pdiff(i),Dh,...
Lhydraulic,'fc72'];
% adjusting Pin/Pout for klosses at entrance/exit
PinCorr(i) = Pin(i)-PinDelta(i); PoutCorr(i) =...
Pout(i)+PoutDelta(i);

% Calculate Nusselt number
[xstar(:,:) xd(:,:) NussStrm(:,:) AvgNuThresh(i) ReynNum(i)...
TwaArray PercIn(i) outofboundsI NuArray TwaStrm(i,:)]

NuGnSelArr(i) ImgHueAvg(i) maxdelNuArr(i)...
maxdelTwaArr(i) UTwcArr TbaExit(i) delValArr(i)...
BoilIndicate(i) QtoFluid(i) TsatOut(i) imgHue...
xtoBoil(i) TbaArrAll(i,:) PrLocalArr(i,:)...
xqualityInd(i,:) htcStrm(i,:) AvghtcThresh(i)...
AvgTWhResh(i) rhoLocalArr(i,:) CpLocalArr(i,:)...
KcondLocalArr(i,:) sigmaLocalArr(i,:)
myFLocalArr(i,:) rhoVLocalArr(i,:) hfgLocalArr(i,:)
ToutSing(i) TsatArr(i,:) TbaRrtheo(i,:) rhoIn(i);]

volAge(i),Current(i),QflArr(i),xin(i),polycoeff,...
fileOutput, dxCalib, dyCalib, umperpix, TestInSigma(i),
TestOutSigma(i), AmbientTemp(i), powerratio, Pdiff(i),...
PoutCorr(i), PinCorr(i), Pbarometer(i),i);
xdtoboilArr(i,:) = [xdtoboil(i), xdtoboil(i), 0, 150];

QperA = QtoFluid/(pi*Dh*heaterL); % heat flux/unit area
Gflux = (rho.In.)*QflArr/(pi*Dh^2/4); % mass flux

% Present processed figures if 'writefigures' is true
if strcmp(writefigures,'true')
    % Hue image
    titleSTR = ['Location ' num2str(i) ', Image No.'...
        num2str(i),', G' ' num2str(Gflux(i),'%10.0f'),...
        'kg/(m^2\cdots), Q/A^\circ', num2str(QperA(i)/1000,...
        '%10.0f') 'kW/m^2'];
    SUBPLOT(5,1,1), imagesc(imgHue);
    colormap(jet);
    colorbar; axis equal;
    caxis([0 360]);
    ylabel('hue value');
    title(titleSTR);

    % Average Tw and Tb along streamwise direction
    SUBPLOT(5,1,2), plot(xd(i,:),TwStrm(i,:),'bo',xd(i,:),... 
    TbArrAll(i,:),'r');
    AXIS([fLOOR(xd(i,1)) CEIL(xd(i,xlength)) 40 100]);
    xlabel('x/D'); ylabel('T (\circ C)');

    % Average Nu along streamwise direction
    plothigh = 1.5*AvgNuThresh(i);
    if plothigh <= 0
        plothigh = 10;
    end
    SUBPLOT(5,1,3), plot(xd(i,:),NussStrm(i,:),'bo');
    AXIS([FLOOR(xd(i,1)) ... 
    CEIL(xd(i,xlength)) 0 plothigh]);
    xlabel('x/D'); ylabel('Nu');

    % Average htc along streamwise direction
    plothigh = 1.5*AvghtcThresh(i);
    if plothigh <= 0
        plothigh = 1000;
    end
    SUBPLOT(5,1,4), plot(xd(i,:),htcStrm(i,:),'bo');
    AXIS([FLOOR(xd(i,1)) ... 
    CEIL(xd(i,xlength)) 0 plothigh]);
    xlabel('x/D'); ylabel('h (W/(m^-2\circ C)');

    % Average RGB intensity

    % End for loop over i

290
SUBPLOT(5,1,5), imshow(RGBflip);
ylabel('raw image');

% Write out plot to file
X=getframe(gcf);
plotfiletitle = ['FiguresFilename num2str(i) '.jpg'];
imwrite(X.cdata, plotfiletitle,'jpg');
end
end

if strcmp(postcalculations,'true')
    %%%%%%%%%%%%%%%%%%%%% Misc Post calculations %%%%%%%%%%%%%%%%%%%%%
    % Miscellaneous plotting and file saving commands may be included as
    % necessary
    end
% END %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% TLC Rig
% Author: Roland S Muwanga
% Date: July 05, 2006
% Modified: Oct 21, 2005
% - added calculation for Tw mean and htc mean
% - modified htc to 28 from 25
% - outputting htc average and stream
% Based on Sept 28/05 and Oct0105

% function [xstar, xd, AvgNuStrm, AvgNuThresh, Reyn, TwFilter, PercIn,]
% outofbounds, NuArrFilter, AvgTwStrm, NuGniel, ImgHueAvg,...
% maxdelNu, maxdelTw, UTwc, TbArrExit, delVals, BoilIndicate,...
%_QtoFluid, TsatOut, imgHue, xdtoboil,TbArr, PrLocal, xquality,
% AvghtcStrm, AvghtcThresh, AvgTwThresh, rhoLocal, CpLocal,...
% kcondLocal, sigmaLocal, myuLocal, rhoVLocal, hfgLocal,...
% ToutSing, Tsat, TbArrtheo, rhoIn] = NuCalcTubeJun3006(Tin,...
% Tout, imgHue, Vvolt, Iamp, Qfl, xin, polycoeff,ofileOutput,...
% dxCalib,dyCalib,unperpix, TinStdev, ToutStdev,xlength,deltay,
% % Tamb,powerratio,Pdiff,Pout,Pin,Pbaro,imgNumber)

% Requires:
% 1. Hue image 'imgHue' of only of area to be converted to Tw based on
% polynomial coefficients in file 'polycoeff'
% 2. 'polycoeff' formatted in output based on calibproc*** or similar
% 3. Vvolt, and Iamp are power applied to tube
% 4. Qfl is flowrate in m3/s
% 5. Pin, Pout, Pbar, Pdiff are inlet, outlet, barometric and
% differential pressure in Pascals
% 6. Tin, Tout are inlet and outlet temperatures of tube
% 7. dxCalib and dyCalib the x and y ROI sizes from calibration

% Promises: Nusselt number calculations for a microtube
% boiling experiment with FC-72

% % %

% % %

% function [xstar, xd, AvgNuStrm, AvgNuThresh, Reyn, TwFilter, PercIn,...
% outofbounds, NuArrFilter, AvgTwStrm, NuGniel, ImgHueAvg,...
% maxdelNu, maxdelTw, UTwc, TbArrExit, delVals, BoilIndicate,...
% _QtoFluid, TsatOut, imgHue, xdtoboil,TbArr, PrLocal, xquality,...
% AvghtcStrm, AvghtcThresh, AvgTwThresh, rhoLocal, CpLocal,...
% kcondLocal, sigmaLocal, myuLocal, rhoVLocal, hfgLocal,...
% ToutSing, Tsat, TbArrtheo, rhoIn] = NuCalcTubeJun3006(Tin,...
% Tout, imgHue, Vvolt, Iamp, Qfl, xin, polycoeff,ofileOutput,...
dxCalib, dyCalib, umperepix, TinStdev, ToutStdev,
Tamb, powerratio, Pdiff, Pout, Pin, Pbaro, imgNumber)

% SOLUTION & OUTPUTS
maxdelNu = 0; maxdelTw = 0; UTwc = 0;
heaterL = 0.106; % heated length
ChannelLength = heaterL; % flow length
Dh = 1.0668e-3; % hydraulic diameter
Dhouter = 1.27e-3; % tube outer diameter
houter = 28; %(W/m2C) - identified from pretesting
kcondSS = 16.2; %(W/mC) Type304 ANSI SS - Matweb
Acs = pi*Dh^2/4; % inner cross-sectional area

% Area of interest size
imgSize = size(imgHue);
xlength = imgSize(2);
ylength = imgSize(1);

% FC-72 Properties
[rhoIn, CpIn, kcondIn, sigmaIn, myuIn, PrIn, rhoVin, hfgIn] = ... fc72properties(Tin); % Inlet fluid properties, ref 1.
hfgIn = hfgIn*1000;
[rhoOut, CpOut, kcondOut, sigmaOut, myuOut, PrOut, rhoVOut, hfgOut] = ... fc72properties(Tout); % Outlet fluid properties, ref 1.
hfgOut = hfgOut*1000;

% Average fluid Properties
TbAvg = (Tin+Tout)/2;
[rho, Cp, kcond, sigma, myu, Pr, rhoV, hfg] = fc72properties(TbAvg);

% ref 1.
hfg = hfg*1000;

% Fluid Properties interpolation
% Local fluid properties along tube assuming all liquid flow
kcondLocal = linspace(kcondIn,kcondOut,xlength);
PrLocal = linspace(PrIn,PrOut,xlength);
myuLocal = linspace(myuIn,myuOut,xlength);
rhoLocal = linspace(rhoIn,rhoOut,xlength);
CpLocal = linspace(CpIn,CpOut,xlength);

Reyn = rho*(Qf1/Acs)*Dh/myu; % Average Reynolds number
mdot = Qf1*rhoIn; % mass flowrate
ReynLocal = rhoLocal*(Qf1/Acs)*Dh/myu; % Local Reynolds number

imgMax = 210; imgMin = 0; % max and min values for filtering identified
% from calibration
Imghueavg = mean(mean(img Hue));

% Initialization of parameter for conversion of hue to temperature
x0 = 1; x1 = dxCalib; y0 = 1; y1 = dyCalib;
xind = 0; yind = 0;
Nx = xlength/dxCalib; Ny = ylength/dyCalib;
counterY = 1;

% Calculate Tw Outer
imghue = flipr(imghue);
Utwc = zeros(size(imghue)); % initialize temperature-hue uncertainty array

for k = 1:Nx*Ny
    Dpolycoeff = polyder(polycoefficient(k,:)); % derivatives of coefficients
    TwArr(y0:y1,x0:x1) = polyval(polycoefficient(k,:),imghue(y0:y1,x0:x1));
    % TwArr - wall temperature
    Utwc(y0:y1,x0:x1) = polyval(Dpolycoeff,imghue(y0:y1,x0:x1));
    % UTwc - wall temperature gradient for uncertainty calc
    counterY = counterY+1;
    if counterY < Ny+1
        y0 = y1;
        y1 = y1+dyCalib;
    else
        counterY = 1;
        y0 = 1;
        y1 = dyCalib;
        x0 = x1;
        x1 = x1+dxCalib;
    end
end
TwArr = flipr(TwArr);
TwMedian = reshape(TwArr,xlength*ylength,1);
TwAvg = median(TwMedian);

% Q to fluid
DelHfl_fluid = mdot*(CpOut*Tout-CpIn*Tin); % single phase energy balance
Qgen = Vvolt*Iamp*powerratio; % generated power times correction ratio
% accounting for losses to environment
QtoFluid = Qgen - houter*(pi*Dhouter*heaterL)*(TwAvg - Tamb);
PercIn = delHfl_fluid/(mean(QtoFluid))*100;
QperA = QtoFluid/(pi*Dh*heaterL); mean(QperA); % power per unit area

% Account for radial heat conduction
ThetaTLC = TwArr - Tamb; mean(mean(ThetaTLC));
QgenperA = (Qgen)/(pi*(Dhouter^-2-Dh^-2)*heaterL);
Ro = Dhouter/2; Ri = Dh/2;
C1 = (-2*Dhouter*ThetaTLC + Ro*QgenperA)*Ro/(2*kcondSS);
C2 = ThetaTLC + (Ro^-2/4)*(QgenperA/kcondSS) - C1*log(Ro);
ThetaHTC = -(Ri^-2/4)*(QgenperA/kcondSS) + C1*log(Ri) + C2;
TwInner = ThetaHTC + Tamb; mean(mean(TwInner));

% Calculate HTC

imgHue = fliplr(imgHue);
TbArr = zeros(1,xlength);
TbArrtheo = zeros(1,xlength);
Lch = ChannelLength;
x = zeros(1,xlength);

% Tbulk / Tsat Calcs

% Assumes boiling occurs when Tbulk = Tsat
% Flow is initialized to Tbulk based on Tin and Tout linear interp
% Local Tsat is calculated based Pin/Pout interpolation
% @ location where Tbulk=Tsat, all location after are set to local Tsat
TsatIn = 1562/(9.729-LOG10(Pin+Pbaro))-273.15; % inlet Tsat, ref 1.
TsatOut = 1562/(9.729-LOG10(Pout+Pbaro))-273.15; % outlet Tsat, ref 1.
[rhoSat, CpSat, kcondSat, sigmaSat, myuSat, PrSat, rhoVSat, hfgSat]...
= fc72properties(TsatOut); % calculate FC-72 properties at outlet
hfgSat = hfgSat*1000;
TimgSatIn = TsatIn - ((TsatIn-TsatOut)/heaterL)*xin; % Sat temp at inlet
% of image
TimgSatOut = TsatIn - ((TsatIn-TsatOut)/heaterL)*...
(xin+(xlength*umperpix/1e6)); % Sat temp
% at outlet of image
Tsat = linspace(TimgSatIn,TimgSatOut,xlength); % local Sat temp
% accross image

xd(j) = (xin+((j-1)*umperpix/1000000))/(Dh);
Tsatflag = 0; % when zero boiling location not found, when 1 boiling
% location found
for i=1:xlength
x(i) = xin + (i-1)*umperpix/1e6;
TbArr(i) = Tin + pi*Dh*x(i)*QperA/(mdot*CpLocal(i));
TbArrtheo(i) = TbArr(i);
if Tsat(i) < TbArr(i) % simple correction for saturation
% temperature assuming linear drop
TbArr(i) = Tsat(i);
end
[rhoLocal(i), CpLocal(i), kcondLocal(i), sigmaLocal(i),...
  myuLocal(i), PrLocal(i), rhoVLocal(i), hfgLocal(i)] =...
fC72properties(TbArr(i)); % update properties w/ Tbulk improved
end
hfgLocal = hfgLocal*1000;
TbArrExit = TbArr(xlength);
ToutSing = Tin + pi*Dh*heaterL*QperA/(mdot*Cp);
Psat = Pin + Pbaro + (Pdiff)/Lch*x;

%%% Length to boil

% Calculated from equating:
% (Tb,out-Tb,in)/heaterL = (Tsatt,boiling-Tb,in)/lengthtoboil
% & (Tsat,out-Tsat,in)/heaterL = (Tsatt,boiling-Tsat,in)/lengthtoboil
BoilIndicate = mdot*(CpOut*Tout-CpIn*Tin);%- QtoFluid;
TsatSlope = (TsatOut - TsatIn) / heaterL;
TbulkOutTheo = Tin + pi*Dh*heaterL*QperA/(mdot*Cp); % calculated with
% average prop's
TbulkSlope = (TbulkOutTheo - Tin) / heaterL; %
lengthtoboil = (TsattIn - Tin) / (TbulkSlope - TsatSlope);
xDtoBoil = lengthtoboil/Dh;
xquality = (QperA*pi*Dh*(x-lengthtoboil))./(mdot*hfg);

% Heat Transfer Coefficient Calculation
hArr = zeros(size(TwInner));
for i=1:xlength
    hArr(:,i) = QperA*((TwInner(:,i)-TbArr(i)).^(-1));
end

% Filter Results
% Filters based on max and min Tw expected
% - usually based from calibration
NuArr = zeros(size(TwInner));
NuArr = hArr*Dh/kcond;
inu = 1; jnu = 1;

NuArrFilter = NuArr;
hArrFilter = hArr;
TwInnerFilter = TwInner;
TwFilter = TwArr;
imgHueFilter = imgHue;
maxNu = max(max(NuArrFilter));
outofbounds = 1;
Tmax = 97; Tmin = 68;
for j=1:xlength
    for i=1:ylength
        if ((TwInner(i,j) > Tmax) || (TwInner(i,j) < Tmin))

296
outofbounds = outofbounds + 1;
end
if ((imgHue(i,j) > imgMax) || (imgHue(i,j) < imgMin))
    NuArrFilter(i,j) = 0;
    hArrFilter(i,j) = 0;
    TwInnerFilter(i,j) = 0;
    imgHueFilter(i,j) = 0;
    TwFilter(i,j) = 0;
end

% Uncertainty Calc
delNu = ones(size(NuArr));
delh = ones(size(NuArr));
delTw = ones(size(NuArr));
[delNu delh delf delTw delRe delxd delxstar delQf1 delTb] = ...
    uncertainty2phasetube(Dh, heaterL, Cp, myu,Pr, kcond, Qf1, rho,...
        x, xlength, umperpix, Reyn, Lch, Pdiff, Iamp, Vvol, UTwc,...
        hArrFilter, Lch,Dhouter, houter, TwFilter, Tamb, TbArr, kcondSS,...
        QtoFluid, TinStdev, ToutStdev, '1.00mm', 'powerinput', Tsat,...
        TbArrtheo, Pin, Pout, Phar0, Psat);

% Calculate averages and output to file
AvgTwall = mean(mean(TwInner));
AvgNu = mean(mean(NuArr));
counter = 0;
counterStrm = 0;
AvgNuSum = 0;
AvghtcSum = 0;
AvgTwSum = 0;
AvgNuStrm = zeros(1,xlength);
maxdelNu = 0;
maxdelTw = 0;
AvgdelTw = 0;
AvgdelNu = 0;
Avgdelh = 0;
for j=1:xlength %Column-wise
    AvgNuColSum = 0; % Reset the column sum to zero
    AvghtcColSum = 0; % Reset the column sum to zero
    AvgTwColSum = 0;
    xstar(j) = (xin+((j-1)*umperpix/1000000))/(ReynLocal(j)*...
PrLocal(j)*Dh);

    xD(j) = (xin+(j-1)*umperpix/1000000)/(Dh);
    for i=1:length yRow-wise
        if ((imgHue(i,j) < imgMax) && (imgHue(i,j) > imgMin))
            % Check if original image values within calibration range
            if ((TwInner(i,j) < Tmax) && (TwInner(i,j) > Tmin))
                AvgNuSum = AvgNuSum + NuArr(i,j);
                AvgNuColSum = AvgNuColSum + NuArr(i,j);
                AvghtcSum = AvghtcSum + hArr(i,j);
                AvghtcColSum = AvghtcColSum + hArr(i,j);
                AvgTwSum = AvgTwSum + TwInner(i,j);
                AvgTwColSum = AvgTwColSum + TwInner(i,j);
                counter = counter+1;
                counterStrm = counterStrm+1;
                AvgdelNu = AvgdelNu + delNu(i,j);
                Avgdelh = Avgdelh + delh(i,j);
                AvgdelTw = AvgdelTw + delTw(i,j);
                if (delNu(i,j) > maxdelNu)
                    maxdelNu = delNu(i,j);
                    inu = i; jnu = j;
                end
                if (delTw(i,j) > maxdelTw)
                    maxdelTw = delTw(i,j);
                    iTw = i; jTw = j;
                end
            end
        end
    end
    if counterStrm > 0
        AvgNuStrm(j) = AvgNuColSum/counterStrm;
        AvghtcStrm(j) = AvghtcColSum/counterStrm;
        AvgTwStrm(j) = AvgTwColSum/counterStrm;
    else
        AvgNuStrm(j) = 0;
        AvghtcStrm(j) = 0;
        AvgTwStrm(j) = 0;
    end
    counterStrm = 0;
end
    AvgNuThresh = AvgNuSum/counter;
    AvghtcThresh = AvghtcSum/counter;
    AvgTwThresh = AvgTwSum/counter;
    AvgdelNu = AvgdelNu/counter;
    AvgdelTw = AvgdelTw/counter;
    AvgdelNuPerc = AvgdelNu/AvgNuThresh*100;
    MaxdelNuPerc = delNu(inu,jnu)/NuArr(inu,jnu)*100;
MaxdelhPerc = delh(inu,jnu)/hArr(inu,jnu)*100;
sizeImg = size(delNu); sizeNew = sizeImg(1)*sizeImg(2);
histArr1 = reshape(delNu,sizeNew,1);
histArr2 = reshape(NuArr,sizeNew,1);
typicaldelNu = median(histArr1);
typicaldelNuPerC = typicaldelNu/median(histArr2)*100;
histArr1 = reshape(delh,sizeNew,1);
histArr2 = reshape(hArr,sizeNew,1);
typicalDelh = median(histArr1);
typicaldelhPerc = typicalDelh/median(histArr2)*100;
histArr1 = reshape(delTw,sizeNew,1);
histArr2 = reshape(TwFilter,sizeNew,1);
typicaldelTwTLC = median(histArr1);
typicaldelTwTLCPerC = typicaldelTwTLC/median(histArr2)*100;
histArr1 = reshape(delTb,xlength,1);
histArr2 = reshape(delTb,xlength,1);
typicalDelTb = median(histArr1);
typicaldelTbPerC = typicaldelTb/median(histArr2)*100;

AvgHue = mean(mean(imgHue));
Avgh = mean(mean(hArr));
AvgNuStrmSize = size(AvgNuStrm);

% Screen outputs

disp('Image no.

num2str(imgNumber));
disp(['max Nusselt uncertainty = ' num2str(maxdel1Nu)]);
disp(dispexit1); maxdelTw = max(max(delTw));
disp(dispexit2);
disp(dispexit3) = ...
['Supposed percentage energy to fluid = ' num2str(PercIn) '%'];
disp(dispexit3);
disp(dispexit4) = ['average filtered Nusselt no. = ' num2str(AvgNuThresh)];
disp(dispexit4);
disp(dispexit5) = ...
['Average wall temperature uncertainty = ' num2str(AvgdelTw)];
disp(dispexit5);
disp(dispexit6) = ['Average Nusselt number uncertainty = ' num2str(AvgDelNu)];
disp(dispexit6);
disp(dispexit7) = ['Average Nusselt unceratinty (percentage)= '

num2str(Avgdel1NuPerC) '%'];
disp(dispexit7);
disp(['Nusselt: (' num2str(inu) ', ' num2str(jnu) ') Temp: ('...

num2str(iTw) ', ' num2str(jTw) ')']);
disp(dispexit8) = ['Max nusselt unceratinty = ' num2str(Maxdel1NuPerC) '%'];
disp(disptext8);
378 disp(['Nusselt: (' num2str(inu) ')',' num2str(jnu) ') Temp: ('...
379 num2str(iTw) '',' num2str(jTw) ');']);
380 disptext9 = ['Max htc unceratinty = ' num2str(MaxehlPerc) ''];
381 disp(disptext9);
382 disptext10 = ... 
383 ['Typical Nusselt uncertainty = ' num2str(typicalNNuPerc) ''];
384 disp(disptext10);
385 disptext11 = ...
386 ['Typical htc uncertainty = ' num2str(typicalelhPerc) ''];
387 disp(disptext11);
388 disptext12 = ...
389 ['Reynoldes no. uncertainty = ' num2str(delRe/Reyn*100) ''];
390 disp(disptext12);
391 disptext13 = ...
392 ['Typical x/d uncertainty = ' num2str(delxd(100)/xd(100)*100) ''];
393 disp(disptext13);
394 disptext14 = ['Typical xstar uncertainty = '...
395 num2str(delxstar(100)/xstar(100)*100) ''];
396 disp(disptext14);
397 disptext15 = ['Flow rate uncertainty = ' num2str(delQf1/Qf1*100) ''];
398 disp(disptext15);
399 disptext16 = ['average filtered Tw = ' num2str(AvgTwThresh)];
400 disp(disptext16);
401 disptext17 = ['average filtered htc = ' num2str(AvghtcThresh)];
402 disp(disptext17);
403 disptext18 = ...
404 ['Typical Tw uncertainty = ' num2str(typicalelTwTLC) '\circC'];
405 disp(disptext18);
406 disptext19 = ...
407 ['Typical Tbkle/sat uncertainty = ' num2str(typicalelTb) '\circC'];
408 disp(disptext19);
409
410 % File outputs
411 disp('XXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXX
412 disp('XXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXX
413 fid = fopen(fileOutput,'a');
414 fprintf(fid,'\%s\n',disptext0);
415 fprintf(fid,'\%s\n',disptext1);
416 fprintf(fid,'\%s\n',disptext2);
417 fprintf(fid,'\%s\n',disptext3);
418 fprintf(fid,'\%s\n',disptext4);
419 fprintf(fid,'\%s\n',disptext8);
420 fprintf(fid,'\%s\n',disptext9);
421 fprintf(fid,'\%s\n',disptext10);
422 fprintf(fid,'\%s\n',disptext11);
423 fprintf(fid,'\%s\n',disptext12);
```c
424  fprintf(fid,'\n',disptext7);
425  fprintf(fid,'\n',disptext8);
426  fprintf(fid,'\n',disptext9);
427  fprintf(fid,'\n',disptext10);
428  fprintf(fid,'\n',disptext11);
429  fprintf(fid,'\n',disptext18);
430  fprintf(fid,'\n',disptext19);
431  fprintf(fid,'\n',disp00);
432  fprintf(fid,'\n');
433  fclose(fid);
434
435  delReynPerc = (delRe/Reyn*100);
436  delxdperc = delxd(100)/xd(100)*100;
437  delxstarperc = delxstar(100)/xstar(100)*100;
438  delQflperc = delQfl/Qfl*100;
439
440  delVals = [typicaldelNuPerc typicaldelhPerc delf typicaldelTwTLC... 
    delReynPerc delxdperc delxstarperc delQflperc];
441  XXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXX
442
443  % END XXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXX
444
445  % References:
446  % 1. 3M Fluorinert Electronic Liquid FC-72, Product Information, 2000.
```
% TLC Rig
% Author: Roland S Muwanga
% Date: Apr 03/06
%
% Program: function manuallfilter(imgdir,imgdirFilt,rgbbeg,imgi)
%
% % Promises: Manually filter an image and return a binary mask
% % identifying the filtered region
% %

function manuallfilter(imgdir,imgdirFilt,rgbbeg,imgi)

sizeimgi = size(imgi); sizeimgi = sizeimgi(2);

for i = 1:sizeimgi
    j = imgi(i);
    dispimagenumb = ['image no.' num2str(j)]; disp(dispimagenumb);
    FilenameIn = [imgdir rgbbeg int2str(j) '.TIFF'];
    RGBimg = imread(FilenameIn,'tif');
    HSVimg = rgb2hsv(RGBimg);
    Vimg = HSVimg(:,:,3);
    VimgBW = roipoly(Vimg);
    HSVimg(:,:,1) = HSVimg(:,:,1).*VimgBW;
    HSVimg(:,:,2) = HSVimg(:,:,2).*VimgBW;
    HSVimg(:,:,3) = HSVimg(:,:,3).*VimgBW;
    RGBimg = hsv2rgb(HSVimg);
    imshow(RGBimg); pause;
    FilenameOut = [imgdirFilt rgbbeg '_filter' int2str(j) '.TIFF'];
    imwrite(VimgBW,FilenameOut,'tif');
end

% END %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% TLC Rig
% Author: Roland S Muwanga
% Date: Apr 03/06

% Program: function [HimgNew] = hueBrightDarkfilter(Filename)
%
% Promises: Automated filtering to remove black paint regions
% and bright spots from hue image
%

function [HimgNew] = hueBrightDarkfilter(RGBimg)

threshVal = 300/360;
VAL1 = 0.5; VAL2 = 0.9;
HUE1 = 0.1; HUE2 = 0.9;
SAT1 = 0.1;

HSVimg = rgb2hsv(RGBimg);
Himg = HSVimg(:,:,1);
Vimg = HSVimg(:,:,3);
Simg = HSVimg(:,:,2);

% modify red color showing high hue to appropriately show low hue
Mult1 = ceil(threshVal - Himg); % less than threshval to 1
Mult2 = ceil(Himg - threshVal); % greater than threshval to 1

Himg = (Himg.*Mult1 + (1-Himg).*Mult2);

% black has low value and may have high hue -> MODIFIED APPROACH
ValSplit = ceil(Vimg - VAL1); % lose dark val -> val < VAL1 to 1 % rest to 1
HueSplit = ceil(HUE2 - Himg); % lose high hue
HueBlack1 = HueSplit.*ValSplit; %
HueBlack1 = im2bw(HueBlack1,0.5); % keep image with with val > VAL1 and % hue < HUE1
HueBlack2 = 1;

% white has dark saturation and bright value
ValSplit = ceil(VL2 - Vimg); % lose bright val -> bright val to 0 % rest to 1
SatSplit = ceil(Simg - SAT1); % lose dark sat -> dark sat to 0 rest to % 1
HueWhite = ValSplit + SatSplit;
HueWhite = im2bw(HueWhite,0.5); % keep image with sat > SAT1 combined
    % with val < VAL2
HueMult = 1; % HueBlack1 .* HueBlack2 .* HueWhite;
% bright and dark filter % temporarily ignore filtering
HimgNew = Himg.*HueMult;
close all
% END %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
function [iend, ImgJoin] = joinImage(SubCase, subcaseIter, i, ...
    totLengthPix, xPixSize, yPixSize, imgname, imgdirFilt, rgbbeg, ...
    indexMM, ypixpos, Nimg, plane, filtering);

switch plane
    case 'single'
        ImgOnes = ones(yPixSize, xPixSize);
        ImgJoin = zeros(totLengthPix, xPixSize);
        ImgDev = zeros(totLengthPix, xPixSize); % counter for number of active elements
while SubCase(i) < subcaseIter
    dispImageNumb = ['image no., ' num2str(i)];
    disp(dispImageNumb);
    FilenameIn = [imname int2str(i) '.tif'];
    imgInt = imread(FilenameIn, 'tif');
    imgDbI = double(imgInt) + 1; % imagesc(imgDbI);
    ImgJoin((ypixpos(indexMM(i),1));
    (ypixpos(indexMM(i),2)), :) = ...
    ImgJoin((ypixpos(indexMM(i),1));...
    (ypixpos(indexMM(i),2)), :) + imgDbI;
    ImgDev((ypixpos(indexMM(i),1));...
    (ypixpos(indexMM(i),2)), :) =...
    ImgDev((ypixpos(indexMM(i),1));...
    (ypixpos(indexMM(i),2)), :) + ImgOnes;
    i = i+1;
end
if i > Nimg, break, end
end
ImgDev = ceil(abs(ImgDev - 0.1));
ImgJoin = (ImgJoin./ImgDev);
Iend = i;
case 'triple' % triple plane image (RGB/HSV)
    ImgOnes = ones(xPixSize, xPixSize);
    ImgJoinHue = zeros(totLengthPix, xPixSize);
    ImgDevHue = zeros(totLengthPix, xPixSize); % counter for number
    % of active elements
    ImgJoinSat = zeros(totLengthPix, xPixSize);
    ImgDevSat = zeros(totLengthPix, xPixSize);
    ImgJoinVal = zeros(totLengthPix, xPixSize);
    ImgDevVal = zeros(totLengthPix, xPixSize);
    HSVJoin = zeros(totLengthPix, xPixSize, 3);
while SubCase(i) < subcaseIter
    dispImageNumb = ['image no., ' num2str(i)];
    disp(dispImageNumb);
    FilenameIn = [imname int2str(i) '.TIFF'];
    RGBimg = imread(FilenameIn, 'tif');
    HSVimg = rgb2hsv(RGBimg);
    Himg = HSVimg(:, :, 1);
    Simg = HSVimg(:, :, 2);
Vimg = HSVimg(:, :, 3);

if strcmp(filterimg,'true')
    FilenameFilter = [imgdirFilt rgbbeg '_filter'
        int2str(i) '.TIFF'];
    Filterimg = imread(FilenameFilter,'tif');
    Himg = Himg.*Filterimg;
    Imgadj = im2bw(Himg,0.8333); % to take negative hue of
    % angles >300
    Imgmult = ~(Imgadj);
    Himg = Imgmult.*Himg + (ImgOnes-Himg).*Imgadj;
    Simg = Simg.*Filterimg;
    Vimg = Vimg.*Filterimg;
end

ImgJoinHue((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) = ...
ImgJoinHue((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) + Himg;
ImgDevHue((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) = ...
ImgDevHue((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) + ImgOnes;

ImgJoinSat((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) = ...
ImgJoinSat((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) + Simg;
ImgDevSat((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) = ...
ImgDevSat((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) + ImgOnes;

ImgJoinVal((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) = ...
ImgJoinVal((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) + Vimg;
ImgDevVal((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) = ...
ImgDevVal((ypixpos(indexMM(i),1)));
(ypixpos(indexMM(i),2)),:) + ImgOnes;
i = i+1;

if i > Himg, break, end
end

ImgDevHue = ceil(abs(ImgDevHue - 0.1)); % sets zero terms to 1
142 \texttt{ImgJoinHue} = (\texttt{ImgJoinHue ./ ImgDevHue}); \quad \% \text{so don't divide by zero}
143 \texttt{ImgDevSat} = \text{ceil}(\text{abs}(\texttt{ImgDevSat} \cdot 0.1)); \quad \% \text{sets zero terms to 1}
144 \texttt{ImgJoinSat} = (\texttt{ImgJoinSat ./ ImgDevSat}); \quad \% \text{so don't divide by zero}
145 \texttt{ImgDevVal} = \text{ceil}(\text{abs}(\texttt{ImgDevVal} \cdot 0.1)); \quad \% \text{sets zero terms to 1}
146 \texttt{ImgJoinVal} = (\texttt{ImgJoinVal ./ ImgDevVal}); \quad \% \text{so don't divide by zero}
147
150 \texttt{HSVJoin(:, ;, 1)} = \texttt{ImgJoinHue};
151 \texttt{HSVJoin(:, ;, 2)} = \texttt{ImgJoinSat};
152 \texttt{HSVJoin(:, ;, 3)} = \texttt{ImgJoinVal};
153 \texttt{ImgJoin} = \text{hsv2rgb}(\texttt{HSVJoin});
154
156 \texttt{iend} = \texttt{i};
157
159 \% END
Appendix C

Simplified Analysis

C.1 Derivations

C.1.1 Parallel Channel Derivations

General Definitions:

\[ Nu = \frac{hD_h}{k}, \]

\[ \alpha(z) = \frac{W_{chn}(z)}{H_{chn}}, \]

\[ A_{cs} = W_{chn}H_{chn} = \alpha H_{chn}^2, \]

\[ A_3 = (W_{chn} + 2H_{chn}) L_{chn} = (\alpha + 2) H_{chn} \cdot L_{chn}; \]

\[ A_4 = 2(W_{chn} + H_{chn}) L_{chn} = (\alpha + 2) H_{chn} \cdot L_{chn}; \]

\[ A_{plan,cell} = W_{cell}L; \]

\[ D_h = \frac{4A_{sk}}{Per_{cs}} = \frac{2\alpha H_{chn}}{(1+\alpha)}; \]

\[ Per_{cs} = \frac{4A_{sk}}{D_h} = 2(W_{chn} + H_{chn}) = 2H_{chn}(1 + \alpha); \]

\[ \bar{m}_0 = \frac{H_{chn}^4 \Delta P}{\nu L_{chn}}; \]
Average Shear Stress:

\[ \Delta P = f \frac{L_{chn} \rho V_{chn}^2}{D_h} \frac{2}{2} \]

The average shear stress in a constant cross-sectional area channel may be obtained as,

\[ \tau_w = \Delta P \cdot \frac{A_{cs}}{L_{chn}} \frac{P_{er}}{2} = f \frac{L_{chn} \rho V_{chn}^2}{D_h} \cdot \frac{A_{cs}}{L_{chn}} \frac{P_{er}}{2} \]

Given,

\[ Re = \frac{\rho V D_h}{\mu} \text{ or } \rho V = \frac{Re \mu}{D_h} \]

then,

\[ \tau_w = f \frac{L_{chn} \rho V D_h}{D_h} \frac{Re \mu}{D_h} \frac{V_{chn}}{2} \cdot \frac{D_h A_{cs}}{L_{chn} 4 A_{cs}} = f \frac{Re \mu V_{chn}}{D_h} \frac{8}{8} \]

and

\[ P_o (\alpha) = f Re = 96 \left[ 1 - 1.3553\alpha + 1.94673\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5 \right] \]

we have,

\[ \tau_w = \frac{P_o \mu V_{chn}}{D_h} \frac{8}{8}. \]

For a channel with varying cross-section then,

\[ \tau_w (\alpha) = \frac{P_o (\alpha) \mu V_{chn}}{D_h} \frac{8}{8}. \]
Mass Flow Rate:

\[ \Delta P \cdot A_{cs} = \int_{0}^{L_{chn}} \tau_w \cdot Per \cdot dx \]

\[ \Delta P \cdot A_{cs} = \int_{0}^{L_{chn}} \frac{1}{8} \frac{\mu V_{chn}}{D_h} \cdot Po \cdot Per \cdot dz \]
and given \( \dot{m} = \rho V_{chn} A_{cs} \) and \( Per = \frac{4A_{cs}}{D_h} \), we have

\[ \Delta P \cdot A_{cs} = \int_{0}^{L_{chn}} \frac{1}{8} \rho D_h A_{cs} \cdot Po \cdot \frac{4A_{cs}}{D_h} \cdot dz \]

or

\[ \Delta P \cdot A_{cs} = \int_{0}^{L_{chn}} \frac{1}{2} \frac{\dot{m}}{D_h} \cdot Po \cdot \frac{\nu}{D_h} \cdot dz. \]

Finally,

\[ \dot{m} = \frac{\Delta P \cdot A_{cs}}{\int_{0}^{L_{chn}} \frac{1}{2} \frac{\nu}{D_h} \cdot Po(\alpha) \cdot dz} \]

or

\[ \dot{m} = \frac{\Delta P \cdot W_{chn} \times H_{chn}}{\int_{0}^{L_{chn}} \frac{1}{2} \frac{\nu}{4\alpha^2 H_{chn}^2/(1+\alpha)^2} \cdot Po(\alpha) \cdot dz} \]

\[ \dot{m} = \frac{\Delta P \cdot \alpha H_{chn}^2}{\int_{0}^{L_{chn}} \frac{1}{2} \frac{\nu}{4\alpha^2 H_{chn}^2/(1+\alpha)^2} \cdot Po(\alpha) \cdot dz} \]

Non-dimensionalizing with \( \dot{m} \),

\[ \frac{\dot{m}_0}{\dot{m}} = \int_{0}^{L_{chn}} \frac{1}{8} Po(\alpha) \frac{(1+\alpha)}{\alpha^3} \left( \frac{dz}{L_{chn}} \right) = \int_{0}^{L_{chn}} \frac{1}{8} Po(\alpha) \frac{(1+\alpha)}{\alpha^3} dz = \frac{1}{\dot{m}^+} \]

Convective Resistance:

\[ q'' A_{plan,cell} = h A_3 (T_w(z) - T_b(z)) \]

\[ q'' W_{cell} L_{chn} = h ((W_{chn} + 2H_{chn}) L_{chn}) (T_w(z) - T_b(z)) \]

311
we define,

\[ \theta_s(z) = \frac{T_w(z) - T_b(0)}{\frac{q''H_{chn}}{k}} \text{ and } \theta_b(z) = \frac{T_b(z) - T_b(0)}{\frac{q''H_{chn}}{k}} \]

substituting the above plus general definitions we have,

\[ q''W_{cell}L_{chn} = \frac{Nu k (1+\alpha)}{2\alpha H_{chn}} (H_{chn} (\alpha + 2) L_{chn}) \]
\[ \times \left[ \frac{\theta_w(z)q''H_{chn}}{k} + T_b(0) - \frac{\theta_b(z)q''H_{chn}}{k} - T_b(0) \right] \]

simplifying,

\[ W_{cell} = \frac{Nu (1 + \alpha)}{2\alpha} (\alpha + 2) H_{chn} [\theta_w(z) - \theta_b(z)] \]

and rearranging,

\[ \theta_s(z) - \theta_b(z) = \frac{2\alpha}{Nu (1 + \alpha) (2 + \alpha) H_{chn}} W_{cell} = R_{conv} \]

Defining,

\[ \alpha_{max} = \max \left( \frac{W_{chn}(z)}{H_{chn}} \right) = \max (\alpha(z)); \quad W_{w,min}^+ = \min \left( \frac{W_w(z)}{H_{chn}} \right) \]

we have,

\[ R_{conv,3} = \frac{2\alpha}{Nu (1 + \alpha) (2 + \alpha)} (\alpha_{max} + W_{w,min}^+) \]

Similarly following the above except using the 4-wall heated area \( A_4 \) we have,

\[ R_{conv,4} = \frac{\alpha}{Nu (1 + \alpha)^2} (\alpha_{max} + W_{w,min}^+) \]

312
Calorimetric Resistance:

$$q'\prime W_{cell}z = \dot{m}C_p\left(T_b(z) - T_b(0)\right)$$

Substituting previous definitions of $\theta_b$ and $\dot{m}$ we have,

$$q''W_{cell}z = \dot{m}_0\dot{m}^+C_p\theta_b(z)q''H_{chn}/k_f$$

therefore,

$$\theta_b(z) = \frac{k_fW(z^+L_{chn})}{C_pH_{chn}(\dot{m}_0\dot{m}^+)} = R_{cal}$$

and finally substituting $\chi$,

$$R_{cal} = \chi \frac{(\alpha_{\text{max}} + W_{w^+}^+)}{\dot{m}^+}z^+.$$

**Total Resistance:** For 3-wall heated,

$$R_{Total,3} = R_{\text{conv}} + R_{cal} = \left(\alpha_{\text{max}} + W_{w,\text{min}}^+\right) \left\{ \frac{2\alpha}{Nu(1+\alpha)(2+\alpha)} + \frac{\chi}{\dot{m}^+}z^+ \right\}$$

and for 4-wall heated,

$$R_{Total,4} = R_{\text{conv}} + R_{cal} = \left(\alpha_{\text{max}} + W_{w,\text{min}}^+\right) \left\{ \frac{\alpha}{Nu(1+\alpha)^2} + \frac{\chi}{\dot{m}^+}z^+ \right\}.$$
C.1.2 Radial Channel Derivations

General Definitions: Aspect ratio as a function of radius:

\[
\alpha (r) = \frac{W_{chn} (r)}{H_{chn}} = \frac{r^2 \beta_2}{H_{chn}};
\]

\[
\alpha_i = \frac{r_{i,in}^2 \beta_2}{H_{chn}} \quad \text{and} \quad \alpha_o = \frac{r_{o,out}^2 \beta_2}{H_{chn}}
\]

Wall surface areas:

\[
A_{hor} = \int_{r_{i,in}}^{r_{o,out}} (r \beta_2) \, dr = \frac{r_{o,out}^2 \beta_2}{2} \frac{r_{o,out}^2 - r_{i,in}^2}{2} \beta_2 = \frac{(r_{o,out} - r_{i,in})(r_{o,out} + r_{i,in}) \beta_2}{2}
\]

\[
= \frac{(r_{o,out} - r_{i,in})(\alpha_o + \alpha_i) H_{chn}}{2} = L_{chn}(\alpha_o + \alpha_i) H_{chn}
\]

\[
A_{ver} = (r_{o,out} - r_{i,in}) H_{chn} = L_{chn} H_{chn}
\]

\[
A_3 = A_{hor} + 2A_{ver} = L_{chn}(\alpha_o + \alpha_i) H_{chn} + 2L_{chn} H_{chn}
\]

\[
A_3 = L_{chn} H_{chn} \left( \frac{(\alpha_o + \alpha_i)}{2} + 2 \right)
\]

\[
A_4 = 2A_{hor} + 2A_{ver} = L_{chn} (\alpha_o + \alpha_i) H_{chn} + 2L_{chn} H_{chn}
\]

\[
A_4 = L_{chn} H_{chn} \left[ (\alpha_o + \alpha_i) + 2 \right]
\]

Cross-sectional area, perimeter and hydraulic diameter definitions remain the same as for the parallel channel definitions.

Convective Resistance:

\[
\frac{q'' \pi (r_{o,out}^2 - r_{i,in}^2)}{N_{chn}} = hA_3/4 (T_w (r) - T_b (r))
\]
Substituting the previously defined non-dimensional temperatures ($\theta_{W/B}$), the Nusselt number ($Nu$) and the appropriate areas ($A_{3/4}$) we have,

$$\theta_w (r) - \theta_b (r) = \frac{2\alpha \pi (r_{out}^2 - r_{in}^2)}{N_u (1 + \alpha)} \left( L_{chn} H_{chn} \left( \frac{(\alpha + \alpha_1)}{2} + 1 \right) \right) N_{chn} = R_{cs}$$

for 3-wall heating case and

$$\theta_w (r) - \theta_b (r) = \frac{2\alpha \pi (r_{out}^2 - r_{in}^2)}{N_u (1 + \alpha)} \left( 2L_{chn} H_{chn} \left( \frac{(\alpha + \alpha_1)}{2} + 1 \right) \right) N_{chn} = R_{cs}$$

for 4-wall heating case.

**Calorimetric Resistance:**

$$\Delta P \cdot A_{cs} = \int_{r_{out}}^{r_{out}} \tau_w \cdot P_{cs} \cdot dr$$

Following similar steps as for the parallel channel case we arrive at,

$$\dot{m} = -\frac{\Delta P \cdot A_{cs}}{\int_{r_{out}}^{r_{in}} \frac{1}{2} \frac{\nu}{D_h} P_0 (\alpha) \cdot dr}$$

Note, the negative is due to a negative velocity due to radially inward flow. Substituting for $D_h$ and $A_{cs}$,

$$\dot{m} = -\frac{\Delta P \cdot \alpha H_{chn}^2}{\int_{r_{out}}^{r_{in}} \frac{1}{2} \left( 2 \alpha H_{chn} (1 + \alpha) \right) P_0 (\alpha) \cdot dr} = -\frac{\Delta P \cdot \alpha^3 H_{chn}^4}{\int_{r_{out}}^{r_{in}} \frac{1}{8} \frac{\nu}{D_h} (1 + \alpha) P_0 (\alpha) \cdot dr}$$

Dividing by the characteristic mass flow to get the non-dimensional mass flow $\dot{m}_0 = \frac{H_{chn}^4 \Delta P}{\nu L_{chn}}$,

$$\dot{m}^+ = \frac{\dot{m}}{m_0} = -\frac{\Delta P \cdot \alpha^3 H_{chn}^4}{\int_{r_{out}}^{r_{in}} \frac{1}{8} \frac{\nu}{D_h} (1 + \alpha) P_0 (\alpha) \cdot dr} \times \frac{\nu L_{chn}}{H_{chn}^4 \Delta P} = -\frac{\alpha^3 L_{chn}}{\int_{r_{out}}^{r_{in}} \frac{1}{8} (1 + \alpha) P_0 (\alpha) \cdot dr}$$

315
Defining \( r^+ = \frac{r_{out} - r}{r_{out} - r_{in}} \) which will then span 0 to 1 from outer to inner radius. Then,

\[
\frac{dr^+}{r_{out} - r_{in}} = \frac{-dr}{r_{out} - r_{in}}
\]

Substituting it into the non-dimensional mass flow,

\[
\dot{m}^+ = \frac{\alpha^3 L_{chn}}{f_0^{1/8} (1 + \alpha) Pr (r_{out} - r_{in})} \frac{\alpha^3}{f_0^{1/8} (1 + \alpha) Pr (r_{out} - r_{in})} dr^+ = \frac{\alpha^3}{f_0^{1/8} (1 + \alpha) Pr (r_{out} - r_{in})} dr^+
\]

Which is the same form as for parallel channels except integrated with respect to \( r \).

For the thermal resistance term:

\[
\frac{q''^D (r_{out}^2 - r^2)}{N_{chn}} = \dot{m}C_p (T_b (r) - T_b (r_{out}))
\]

Substituting the previously defined non-dimensional temperatures (B), and mass flowrate parameters,

\[
\theta_b = \pi \left\{ \frac{r_{out}^2 - (r_{out} - r^+ (r_{out} - r_{in}))^2}{N_{chn}} \right\} \left( \frac{k_f}{C_p} \right) \left( \frac{1}{\dot{m}^+ \cdot \dot{m}_o H_{chn}} \right)
\]

Where,

\[
r^+ = \frac{r_{out} - r}{r_{out} - r_{in}}
\]

### C.1.3 Parallel Channel Code Validation

The equations presented have been coded into MATLAB to determine optimal channel configurations. To verify the code, comparisons were made with the test case
presented by Bau (1998). For this case the working fluid was water, flowing in a 1500 μm deep channel, 1 cm long and with a 20 kPa pressure supply. Fluid properties were noted to be calculated at room temperature for which the present has assumed to be at 20 °C. Bau (1998) considered the objective function given by,

\[ J(K_1, K_2) = \min \left\{ K_1 \max (\theta_w(z)) + K_2 \int_0^1 \left( \frac{d\theta(z)}{dz} \right)^2 dz \right\} \]

The case \((K_1 = 1, K_2 = 0)\) will determine the minimum peak temperature and \((K_1 = 0, K_2 = 1)\) the minimum gradient. Depending on the problems objective, \((K_1 = 1 \text{ and } K_2 = 0)\) may be adjust appropriately. In the analysis for minimizing the maximum temperature gradient the coefficients were set to \((K_2/K_1 = 5 \times 10^3)\). This has also been incorporated in the following for comparison. The polynomial form for the wall contours has been set to second order.

Figure C.1 shows the results from the analysis compared with the results from Bau (1998). The figure presents the wall temperature profiles and the corresponding wall contours. Reasonable agreement is observed both for minimizing the peak temperature and the temperature gradient. Differences are attributed to differences in the numerical minimization routines utilized. These results validate the developed code for evaluation of optimum channel configurations.
(a) Non-dimensional wall temperature, 3-wall fully developed.

(b) Aspect ratio, 3-wall fully developed.

Figure C.1: Comparison of developed code results with Bau (1998).
C.2 Results for Parallel Channel System

C.2.1 Parallel Channels: Influence of Boundary Condition

Figure C.2 presents the optimized wall temperature and aspect ratios for channels with linearly varying cross-sections and for different thermal boundary conditions. These results are for the case of minimizing the peak wall temperature and the results are compared with an optimized uniform channel. The analysis was performed for $\chi$ values of $2.60e^{-3}$, $2.60e^{-4}$, $2.60e^{-5}$ and $5.21e^{-6}$, corresponding to pressure drops of 0.1 psi, 1 psi, 10 psi and 50 psi. In the case of fully developed flow (C.2 a & c), the wall temperature linearly rises, whereas a non-linear rise is observed for thermally developing flow. Comparing the uniform channel to varying cross-sectional area channels, negligible differences are observed in the thermal profile and consequently the aspect ratio. Observed in the results in general is a trend towards optimized channels with smaller cross-sections for increased pressure drop availability. This is not surprising as with increasing pressure drop availability, narrower channels can be accommodated and are advantageous due to the increased heat transfer coefficients.

Figure C.3 presents the results for minimizing the wall temperature gradient, comparing uniform channels with linearly varying cross-sectional area channels. For the 3-wall heated case at higher pressures, a slightly converging channel produces both a lower peak temperature and lower temperature gradient. With higher pressures in all the cases however, the temperature rise is small and hence the motivation for minimizing the gradient is low. For the 4-wall heated cases however, negligible benefit is obtained from using a channel with varying cross-section. Additionally the optimal
Figure C.2: Minimizing peak temperature: Linear profile.
Figure C.3: Minimizing temperature gradient: Linear profile.
aspect ratio for 4-wall heated channels is 1. Comparing Figures C.2 and C.3, a lower
temperature gradient may indeed be obtained, however this in all cases occurs at the
cost of an increased maximum wall temperature.

C.2.2 Parallel Channels: Influence of Wall Contour

Polynomial

The previous section considered the case of channels with linearly varying cross-
sections only. This section presents results for channels with quadratic and cubic
contoured channels. In particular the 4-wall heated case with thermally developing
flow is presented as it is a more realistic scenario due to the flows thermal devel-
opment. Figure C.4 presents the wall temperature and contour profiles for the case
of minimizing the peak wall temperature. With exception of the low pressure case,
the contoured channels all tend toward a significantly converging channel at the exit.
The wall temperature profiles show negligible differences however when compared to
the uniform channel profiles. In the case of minimizing the wall temperature gradient
uniform channels with aspect ratio of 1 are found to be optimal over a quartic or
cubic wall profile (Figure C.5).
Figure C.4: Minimizing peak temperature: 4-wall heated, thermally developing.
Figure C.5: Minimizing temperature gradient: 4-wall heated and thermally developing.
Appendix D

Part Drawings
D.1 Microtube Drawings
D.2 Cross-linked Heat Sink Drawings
All dimensions in millimeters (mm)

Cross-linked Design

Silicon Base - Detail

VIEW A
D.3 Radial Heat Exchanger Drawings
Silicon Base - Detail
Radial Design

All Dimensions in millimeters (mm)
Heater pattern location
Radial Design
All dimensions in millimeters

5.55
3.055
5.055

31.110
31.111
D.5 Parallel Channel Packaging System II
Heat Sink Packaging II

Tolerance +/-.010"  
Permament bond with appropriate  
Transparant adhesive [not specified]  
Primary dimensions in millimeters  
Material: Polycarbonate  

% N PT X 10 DEEP  
ø6.15 THROUGH
D.6 Radial Channel Packaging System