Experimental and numerical evaluation of single phase adiabatic flows in plain and enhanced microchannels

Akhilesh V. Bapat
Experimental and Numerical Evaluation of Single Phase Adiabatic Flows in Plain and Enhanced Microchannels

by

AKHILESH V. BAPAT

A Thesis Submitted in Partial Fulfillment of the Requirements for the MASTER OF SCIENCE IN MECHANICAL ENGINEERING

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ABSTRACT

Thermal Dissipation is a critical issue in the performance of semiconductor devices. The current practice is to use forced convection by air over a heat sink which is bonded to the microelectronic device. With increased packing density of the circuits inside a chip, large amounts of heat is generated and air cooling is no longer sufficient. Forced liquid convection using microchannels is considered to be a viable option for cooling of these microprocessor chips. This work deals with the evaluation of the single phase pressure drop in microchannels.

There are two types of microchannels under consideration. Plain microchannels which have basically long uninterrupted flow channels while the enhanced channels which have the interrupted flow lengths. Enhanced microchannels, because of their offset strip fin geometry, significantly increase both the heat transfer as well as the pressure drop. This work deals with the evaluation of single phase flow pressure drop in both plain and enhanced microchannels.

For plain microchannels there have been a few investigations in the literature which suggest that the microchannel performance can generally be predicted using the classical fluid flow equations. However there are some experiments that still show departure from the classical theory that cannot be explained. It is proposed in this work that the reason for this discrepancy can be traced to the effects due to flow maldistribution in plain microchannels. A systematic experimental investigation is performed to study the effects of slight variations in channel dimensions and their influence on the flow maldistribution in an attempt to validate the applicability of classical theory to microchannel flows. Enhanced microchannels however have not been investigated thoroughly. There is very few data available in the literature. FLUENT is a CFD software which can be used as a tool to design and optimize these enhanced channels. However it has to be first validated with experiments. Thus pressure drop experiments are carried out on an offset strip fin silicon microchannel and the data is predicted using FLUENT, which is CFD software. Also existing predictive models for friction factor in offset strip fin minichannels are tested to check their validity for microchannel flows.

For plain microchannels, it seen that with uniform flow assumption, the friction factor is either underpredicted or overpredicted using the theory depending upon the reference channel dimension. However by accounting flow maldistribution in plain microchannels, friction factor can be accurately determined using theoretical equations. For enhanced microchannels it is observed that FLUENT can predict the pressure drop within 10%.

In this work only adiabatic flows are considered. It is recommended that this work should be extended to flows with heat transfer.
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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$a$</td>
<td>channel width</td>
<td>m</td>
</tr>
<tr>
<td>$A$</td>
<td>area</td>
<td>m$^2$</td>
</tr>
<tr>
<td>$b$</td>
<td>channel height</td>
<td>m</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat</td>
<td>J kg$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$D$</td>
<td>diameter</td>
<td>m</td>
</tr>
<tr>
<td>$D_h$</td>
<td>hydraulic diameter</td>
<td>m</td>
</tr>
<tr>
<td>$f$</td>
<td>Fanning friction factor</td>
<td>$= \frac{\rho \Delta p D_h}{2 L G^2}$</td>
</tr>
<tr>
<td>$G$</td>
<td>mass flux</td>
<td>kg m$^{-2}$ s$^{-1}$</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity</td>
<td>W m$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$K$</td>
<td>loss coefficient</td>
<td></td>
</tr>
<tr>
<td>$L$</td>
<td>channel length</td>
<td>m</td>
</tr>
<tr>
<td>$L_h$</td>
<td>hydrodynamic entrance length</td>
<td>m</td>
</tr>
<tr>
<td>$L^+$</td>
<td>non-dimensional entrance length</td>
<td></td>
</tr>
<tr>
<td>$n$</td>
<td>number of channels</td>
<td></td>
</tr>
<tr>
<td>$P_w$</td>
<td>wetted perimeter</td>
<td>m</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
<td>$= \frac{G D_h}{\mu}$</td>
</tr>
<tr>
<td>$s$</td>
<td>fin thickness</td>
<td>m</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature,</td>
<td>°C</td>
</tr>
</tbody>
</table>
\( x^+ \) non-dimensional flow distance
\( x^* \) non-dimensional thermal flow distance

**Greek**

\( \alpha_c \) microchannel aspect ratio \( = \frac{a}{b} \)
\( \alpha_c \) fin aspect ratio \( = \frac{s}{b} \)
\( \Delta \) difference

\( \mu \) viscosity \( \text{N} \text{ s} \text{ m}^{-2} \)
\( \rho \) density \( \text{kg} \text{ m}^{-3} \)

**Subscripts**

app apparent
avg average
b bulk or mean
c cross section
FD fully developed
i inlet
LMTD log mean temperature difference
m mean
o outlet
s surface
tot total
w wall
CHAPTER ONE
INTRODUCTION

Thermal management of VLSI chips is very important. The reliability of these chips decreases dramatically as the chip temperature increases beyond 110°C. Considering various thermal resistance paths, it is customary to limit the temperature of the back side of an IC chip to around 80 or 85 °C. The current industry practice employs a heat sink which is bonded to the back side of the chip. A fan is placed on top of the heat sink which forces air through the extended surface. Figure 1.1 shows a typical heat sink setup for cooling computer chips as depicted by Steinke (2005). As shown in the fig. 1.1, the heat sink is typically made of copper or aluminum. It is bonded using a Thermal Interface Material to the silicon device, to accommodate for unequal thermal expansion. Heat Spreader is used to uniformly increase the heated surface area for better uniform cooling. With the advances in microelectronic industry, the cooling requirements are increasing and forced air cooling is no longer a viable option. Table 1.1 lists the estimates of heat transfer coefficients for different mechanisms.

Table 1.1 Heat Transfer Coefficients for Forced Convection, Holman (1976)

<table>
<thead>
<tr>
<th>HEAT TRANSFER MECHANISM</th>
<th>h (W/m²°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Forced Convection, Air</td>
<td>~100</td>
</tr>
<tr>
<td>Forced Convection, Water</td>
<td>~15000</td>
</tr>
<tr>
<td>Flow Boiling, Water</td>
<td>~100000</td>
</tr>
</tbody>
</table>
For cooling requirements in excess of 100 W/cm², liquid cooling seems to be the right solution. Heat transfer coefficients in the order of 100000 W/m²°C can be achieved with flow boiling of water. However employing flow boiling also requires additional components like a condenser. The overall cooling system gets very complicated and bulky. However it is possible to achieve heat transfer coefficients more than 10000 W/m²°C with forced convection of water by using microchannels as shown by Colgan et al. (2005). Jet impingement and evaporative
spray cooling are also attractive options which are considered for electronics chip cooling. Microchannel performance is evaluated in this thesis.

Microchannels have very small hydraulic diameters in the order of 200μm. Figure 1.2 shows a schematic of plain microchannels.

![Figure 1.2: Schematic of Flow in Plain Microchannels](image)

The small flow cross sectional area associated with microchannels presents a large surface area to the volume ratio which is responsible for increased heat transfer. This inherent characteristic of microchannels enables them to be used for high heat flux cooling. The microchannel configuration shown in Fig. 1.2 is called plain microchannels where the fluid has uninterrupted flow through the length of the channel. The hydraulic diameter of these
microchannels falls in the range of 10-200µm. Further increase in heat transfer can be achieved by modifying the path of the flow of the fluid in the microchannels. The fin walls, which run continuous along the flow length for plain channels, are periodically spaced and also offset. Figure 1.3 shows the schematic of an enhanced microchannel which has an offset strip fin configuration.

![Fig. 1.3: Schematic of an “Enhanced” Microchannel](image)

Each new fin interrupts the flow and in the process there is a formation of boundary layer at the start of each fin which continues along the fin length. Heat transfer in the boundary regions
is very high. This enables the offset strip fin microchannels to push the heat transfer coefficient even higher than plain microchannels. However, the enhanced heat transfer also comes with additional pressure drop. Pressure drop is increased because of boundary layer formation and frictional form drag losses. This makes the hydraulic prediction of these channels very important. Because of the complexity of the flow in these channels, it is very difficult for an analytical solution. This work deals the evaluation of flows through both these configurations of microchannels – plain and enhanced.

1.1 Brief History of Microchannel Heat Sink

The first concept for using microchannels as a heat exchanger device for VLSI circuits was conceived by Tuckerman and Pease (1981). It was found that the microchannels are very effective in heat dissipation. The reason for very high heat removal was associated with the fact that microchannels have very small hydraulic diameters ranging to up to a few hundred microns. This was in conjunction with the significant area enhancement due to the finned surface. Because of smaller free flow areas, the power needed to pump fluid through smaller passages also increased a lot. Thus increased pressure drop was a concern. Very high pressures required to maintain the flow through these channels assured that microchannels predominantly operated in the laminar regime. Although microchannels promised to be very effective, initially microchannel research did not receive an impetus. Intel developed the 80286 microprocessor which had significantly reduced the power consumption of the microelectronic device which implied less waste heat.

By mid 90’s increased complexity and density of microelectronic circuits gave rise to increase in the thermal dissipation requirements. Microchannel cooling was widely researched by the academia and industry. Microchannels were a viable option for the practical implementation (Tuckerman and Pease (1981), Phillips (1987)). The next question was the applicability of the
conventional equations to predict the thermo-hydraulic performance of these microchannels. There have been many papers in the literature with different views. Steinke and Kandlikar (2006) carried out an exhaustive review of investigations dealing with microchannel cooling.

Other viable configurations which would utilize the potential of microchannels investigated. It was found that the enhanced channels were the most promising for high heat flux removal. These enhanced microchannels exhibit an interrupted fin design with the fins offset. This causes periodic destruction of the boundary layer which accounts for very high values of heat transfer coefficients. This type of configuration is widely used in the heat exchanger applications like automotive cooling. Researchers have gathered data spanning over few decades and come up with correlations to predict the pressure drop and heat transfer in offset strip fin heat exchangers. However for offset strip finned microchannel no predictive models have been developed.

The present work deals with the evaluation of single phase fluid flow in microchannels. Both plain and enhanced microchannels have been investigated. The single phase predictions would be evaluated for plain and enhanced channels.

1.2 Thesis Outline

This work deals with the experimental and numerical evaluation of single phase adiabatic flows for both plain and enhanced microchannels. An outline is laid out in this section to understand the development and presentation of the work reported in this thesis.
1. Review literature for studies on applicability of classical fluid flow equations to plain microchannel flows

2. Investigate the effect of geometric variation in the channels on flow distribution and the fluid flow prediction using conventional theory

3. Systematic experimental setup to facilitate flow maldistribution and to evaluate experimentally the effects of flow maldistribution on the theoretical prediction

4. Development of equations to analyze flow maldistribution effects on friction factor

5. Estimate individual channel flow rates based on the channel cross section and measured pressure drop

6. Calculate for each channel the frictional pressure losses

7. Comparison of theoretical prediction with the experimental data based on maldistribution analysis

Fig. 1.4 (a): Flow Chart for Experimental Evaluation of Plain Microchannel Flows
ENHANCED MICROCHANNELS

1. Review literature for studies on offset strip finned microchannels for any predictive models for friction factor
2. Use CFD to simulate microchannel flow
3. Identify experimental data and correlations for hydraulic performance of enhanced microchannels
4. Generate pressure drop data in the laminar range on enhanced silicon microchannels
5. Creating mathematical model of unit cell to represent the offset strip fin microchannel and replicate the experimental flow conditions numerically
6. Check the applicability of existing minichannel correlations for predicting fluid flow performance in enhanced microchannels
7. Validation of CFD analysis of fluid flow through offset strip fin microchannels with the experimental pressure drop data
8. Comparison of enhanced minichannel correlations for predicting friction factor with enhanced microchannel data

Fig. 1.4 (b): Flow Chart for Numerical Evaluation of Enhanced Microchannel Flows
Figure 1.4 (a) shows the outline for the experimental evaluation of flows in plain channels. Plain channels have been extensively investigated in the literature. A brief review is conducted in Chapter Two. However there exist a lot of contradictory views about the validity of classical fluid flow equations to the microchannel flow. It is seen that the conventional theory either over predicts or underpredicts the microchannel pressure drops. These predictions are based on equal flow distribution. Flow maldistribution introduced due to slight geometric variation is thought to be the reason for discrepancy in the literature. In this work, maldistribution analysis is developed to test the validity of the classical equations for microchannel data and explain the discrepancy in the literature. The analysis involves accurate estimation of individual channel flow rates based on the channel dimensions. This flow rate is then used to calculate the experimental and theoretical channel friction factor. Friction factor predictions using uniform flow distribution are also made. By comparing the predictions based on uniform flow assumption and maldistributed flow assumption, the effect of geometric variation in the channel dimension on the fluid flow prediction using classical equations are analyzed.

For offset strip fin microchannels however, there is not a large pool of data sets available in the literature. Figure 1.4 (b) is the flow chart with the outline for numerical evaluation of adiabatic flow in enhanced microchannels. Because of the complexity of flow involved in enhanced channels, no analytical model has been developed. Many researchers have developed correlations based on experimental data for enhanced minichannels. However for enhanced microchannels there are no predictive models developed yet. In this work, CFD is tested as a tool to predict the flow in enhanced microchannels. The CFD analysis is carried out in FLUENT and the analysis is validated by running experiments on enhanced microchannels. The existing
correlations for minichannel offset strip fins are also compared with the pressure drop data on the enhanced microchannels.
CHAPTER TWO
LITERATURE REVIEW

Plain microchannels have been investigated extensively. Many researchers have compared the classical theory to the microchannel data. There are conflicting views on whether the conventional theory can predict the friction factor in microchannels. A brief review of few articles dealing with predictive models on microchannel friction factor is presented. The enhanced microchannels, with offset strip fin configuration however have not been tested experimentally. There is hardly any experimental data in the literature. For enhanced minichannels however there are many data points which have been generated for over 40 years and many correlations have also been developed to predict minichannel performance for enhanced channels. No predictive model has been developed for enhanced microchannels yet. Following the literature review on plain microchannels, review of work on offset strip fin configuration channels is presented.

2.1 Plain Microchannels

Many researchers over the years have investigated the microchannel performance. The highlight of this work is to evaluate the microchannel performance in regards to the conventional fluid flow theory. In the literature review for plain microchannels, few relevant investigations dealing with the pressure drop predictions of microchannels have been highlighted. A brief summary of previous investigations on single-phase flow in microchannels and their performance prediction using macroscale correlations are described in this section. Table 2.1
presents a summary of some of the important investigations in this area. Many investigations have reported aberrant behavior of flow and heat transfer in microchannels.

Table 2.1: Summary of investigation on single phase flows in microchannels

<table>
<thead>
<tr>
<th>Author</th>
<th>Channel Details</th>
<th>Hydraulic Diameters (µm), Measurement Verification</th>
<th>Flow rate Measurement Apparatus</th>
<th>Classical Theory Predictions in Laminar Regime</th>
<th>Experimental Uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peng et al.</td>
<td>Rectangular channels machined on steel substrate, single channel</td>
<td>133-367, NA</td>
<td>Rotameter</td>
<td>Significant deviation from theory for Nu and friction factor</td>
<td>NA</td>
</tr>
<tr>
<td>(1994)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yu et al.</td>
<td>Microtubes, single channel</td>
<td>19-102, SEM</td>
<td>NA</td>
<td>Friction factor overpredicted, no heat transfer prediction in laminar</td>
<td>11.9% in fRe</td>
</tr>
<tr>
<td>(1995)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mala and Li,</td>
<td>Fused silica, Stainless Steel microtubes,</td>
<td>50-254, NA</td>
<td>Flow sensor, confirmed by actual measurement</td>
<td>Friction factor underpredicted for smaller diameter tubes,</td>
<td>NA</td>
</tr>
<tr>
<td>(1999)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Authors</td>
<td>Methodology</td>
<td>Channels</td>
<td>Measurement</td>
<td>Instrumentation</td>
<td>Results</td>
</tr>
<tr>
<td>-------------------------------</td>
<td>--------------------------------------</td>
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<td>-------------</td>
<td>-----------------</td>
<td>-------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Hegab et al. (2001)</td>
<td>Milled Aluminium plates, multiple</td>
<td>112-210,</td>
<td>Flowmeter</td>
<td>Agreement for</td>
<td>3-23% for friction factor, no heat transfer study</td>
</tr>
<tr>
<td></td>
<td>channels</td>
<td>digital dial calipers</td>
<td></td>
<td>friction factor</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>no heat transfer study</td>
<td></td>
</tr>
<tr>
<td>Bucci et al. (2003)</td>
<td>Microtube, Single channel</td>
<td>172-520, NA</td>
<td></td>
<td>Friction factor agreement below Re 800-1000, heat transfer coefficient</td>
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<td></td>
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<td></td>
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<td>higher than</td>
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<td>thermally</td>
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<td></td>
<td></td>
<td>developing flow</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td>theory</td>
<td></td>
</tr>
<tr>
<td>Solomon and Sobhan (2005)</td>
<td>Milling copper plates, multiple</td>
<td>280-3670, NA</td>
<td>Rotameter</td>
<td>fRe underpredicted, Nu over predicted</td>
<td></td>
</tr>
<tr>
<td></td>
<td>channels</td>
<td></td>
<td></td>
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<td></td>
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<tr>
<td>Steinke and Kandlikar</td>
<td>Etched Silicon</td>
<td>222, SEM</td>
<td>Rotameter</td>
<td>Agreement for</td>
<td>6.5% in f</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td>fRe and Nu</td>
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</tbody>
</table>
Peng et al. (1994) began investigating into the microchannel performance in comparison to the conventional theory. They found a discrepancy in the two. They experimentally investigated heat transfer in rectangular microchannels with hydraulic diameters of 133-367µm. Their experiments indicated early transition to turbulent regime and fully developed flow was observed for Re 400-1500. Nusselt number in the laminar regime was found to be dependent on Re^{0.62}. Fluid properties were calculated at the fluid inlet temperature. The relationship between friction factor and Nusselt number was observed to be significantly different for the laminar flow. The authors stated that for microchannels the friction factor in the laminar regime was highly dependent on the hydraulic diameter and the channel depth to height ratio. They found

| (2005) | channels, multiple channels |  |  |  |  |
| Rands et al., (2006) | Fused silicon microtubes, single channel | 16-30, SEM | Graduated flask and stop watch | Agreement with theory for fRe, no heat transfer study | 16-29 % for fRe |
| Hrnjak and Tu, (2007) | Machined in PVC substrate, single channel | 69-304, Stylus surface profilometer | Rheotherm mass flow meter | Agreement for friction factor, no heat transfer study | ±3.5% for f |
that the experimental friction factor was higher than the classical prediction. The laminar to
turbulent transition was found to be a strong function of the hydraulic diameter.

Mala and Li (1999) investigated water flow through tubes of diameters ranging from 50
to 254μm. The flow characteristics for smaller tubes deviated significantly from the conventional
theory. Material dependence on the friction factor was observed and the values obtained
experimentally were higher than the predictions. At lower flow rate the results were in rough
agreement with the theory however, significant deviation was noted at higher Re flows. Two
possible explanations were discussed. One was that there was early rise to transition flow and
hence laminar flow equations cannot be used. The second reason which also explained the early
transition was the surface roughness which might be thought to play an important role. It was
however apparent that the early transition seemed to be the prominent reason for the discrepancy.

Harms et al., (1999) applied the developing flow theory for both single channel and
multiple channel systems to characterize flow and heat transfer in minichannels. The channels
were 1000μm deep and 251μm wide. They experimentally observed that the local Nusselt
number agreed well with the classical developing flow theory. However, for multiple channel
design, agreement was reasonably well at higher flow rates but deviated significantly from
theory at low flow rates. Authors reported this deviation to the flow bypass in the manifold. But
it was concluded that the classical theory does apply to plain microchannels as well

Hegab et al. (2001) found that the friction factors values were consistently lower than
values predicted by macroscale correlations in the transition and the turbulent regime. R-134a
was used as the test fluid for 112-210 μm hydraulic diameter rectangular channels. For the heat
transfer calculations at lower Reynolds numbers, the uncertainties reported were as high as 67%.
Hence the heat transfer predictions in the laminar regime were not reported.
Hrnjak and Tu (2007) investigated fully developed liquid and vapor flow through rectangular microchannels with hydraulic diameters of 69 to 304 μm. For low surface roughness the $f_{laminar}$ approached the conventional values for all the channels tested. No heat transfer studies were performed.

Steinke and Kandlikar (2005) conducted an exhaustive survey of literature and experimentally investigated friction factor and Nusselt number in silicon microchannels. They used the simultaneously developing flow condition since the channel lengths were small, and compared the data with the developing flow theory. It was reported that the developing flow theory was in very good agreement with the data. It may be noted that the individual channel dimensions were measured and were very close to each other.

It is observed from literature that there are contradictory findings, especially as related to the applicability of the laminar flow theory for microchannels flows. However, there seems to be a consensus building up in the recent years which agree that fluid flow in microchannels in the laminar regime is not different from macroscale phenomenon in some of the carefully conducted experiments with same size channels. Although the continuum assumption is widely accepted for microchannels, the effect of individual channel size variations is believed to be a factor responsible for these deviations in parallel microchannels. This work attempts to analyze the effects of flow maldistribution on the single phase flow predictions in microchannels.

2.2  Enhanced Microchannels

Enhanced microchannels mentioned here on refer to the offset strip fin configuration. This type of minichannel configuration has been widely researched for many decades for its very high heat transfer surface area. For the design of such a heat exchanger it is very essential to be able to predict the performance. Although these prove to be promising technology for electronics
cooling, not many papers have dealt with rectangular offset strip fins microchannels. Recently Kosar and Peles (2006), Prasher et al (2007) have investigated offset pin fin geometry. Steinke and Kandlikar (2006) and Colgan et al (2005) are the two publications with data on enhanced microchannels. However, for mini channel offset strip fin geometry, various researchers have collected data and many correlations have been developed. The scope of current work is extended only towards adiabatic flow in microchannels.

Kays (1993) developed a very simple analytical model for predicting hydraulic performance of enhanced channels. This earliest model was based on the forced convection on a plate with an additional term which included the drag coefficient. Weiting (1977) obtained correlations from curve fits to data from over 20 core geometries covering laminar and turbulent ranges. Sparrow and Liu (1979) investigated numerically the heat transfer and pressure-drop results for laminar airflow through arrays of inline or staggered plate segments. Joshi and Webb (1987) presented analytical models to predict the heat transfer coefficient and the friction factor of the offset strip fin heat exchanger surface geometry in the laminar and turbulent flow regimes. They also studied the transport of energy and momentum in the boundary layers of the fins because of the oscillating velocities developed from the wakes. Thus the wake distribution was also studied by them to take into considerations the effect of fin length, fin thickness and the fin spacing on the wake flow pattern. This was done to accurately determine the transition flow. The models developed are very complicated. Manglik and Bergles (1995) gave an exhaustive summary of the existing correlations for minichannels. They also developed a correlation with a consistent definition of hydraulic diameter. The equations for friction factor were a continuous form covering the laminar, transition and turbulent flow regime.
All this work has been carried out at the minichannel level, with the flow ranges occurring in the practical application of around 200-4000 Reynolds number. Also almost all the predictive models developed are based on multiple regression analysis. The application of these correlations for microchannel data needs to be ascertained.

Colgan et al (2005) obtained data on enhanced microchannel for laminar flow regime with the Reynolds number in the range of 20-300. They obtained cooling of over 300 W/cm² with very little pressure drop. This was achieved by designing an intricate manifold structure interconnected to each other rather than just an elongated opening which serves as the inlet. Figure 2.1 shows the photograph of the manifold facing the chip. The result of the multiple inlet and outlet inlet vias leads to a complex and short flow lengths which significantly enhance the performance. As a result this data cannot be used as a comparison to test the minichannel correlations due to heavy cross flows caused by the multiple inlets and outlets.

Bapat and Kandlikar (2006) investigated the minichannel correlations used to predict offset strip fin heat exchanger performance. The correlations were compared with experimental microchannel data on offset strip fins obtained by Colgan et al (2005). Two reasons were cited why these correlations should not be used to compare microchannel data. Almost all of the
correlations are based on series of experimental data collected on various geometries. The flow ranges employed in gathering that data were in the practical limits of 200-4000 Reynolds number range. For laminar flow which is normally used in microchannels, the range is around 20-200 Reynolds number. Thus the correlations should be tested with large data sets on various aspect ratios. The second reason cited was that the experimental data used for comparison is not a good reference. The microchannel coolers are very advanced with very short flow lengths and multiple inlets and outlets which are responsible for a complex flow path. Thus the data may not be representative of simple offset strip fin microchannels.

Enhanced microchannels with offset strip fin configuration provide an attractive alternative for electronics cooling application. However no predictive model has been developed to predict the fluid flow in enhanced microchannels. Fluid flow in enhanced microchannels is predicted using CFD in this work.

2.3 Objective

This work addresses two points evident in the literature review. For plain channels it is seen that although there are few investigations which show that the classical equations can predict the microchannel flow, there are some experiments which show that there is deviation from the theoretical prediction. Flow maldistribution in parallel plain microchannels is believed to be one of the reasons for this discrepancy. This premise is experimentally investigated by analyzing frictional pressure losses in parallel microchannels. Also, for enhanced channels, it is seen that there is no analytical model developed yet to predict the fluid flow performance. Also the existing correlations based on minichannels are not applicable to the microchannel data. The second major objective of this work is to show that the flow in enhanced microchannels can be predicted using CFD. FLUENT, commercially available CFD software is used to predict the
performance. Experiments are also carried out to validate the computer simulations. Following are the objectives of the thesis

- To estimate flow maldistribution in parallel microchannels induced by channel size variation
- Experimentally evaluate the effect of flow maldistribution on the friction factor prediction using classical fluid flow equations
- Predict the fluid flow performance in enhanced microchannels using CFD analysis
- To experimentally validate the CFD results on adiabatic flows in enhanced microchannels
- Test the validity of existing enhanced minichannel correlations to predict the enhanced microchannel data
CHAPTER THREE

EXPERIMENTAL SETUP FOR PLAIN MICROCHANNELS

It is identified from the literature that there is discrepancy in the validity of the classical equations for plain microchannels. Maldistribution is thought to be one of the reasons for this deviation from theory. The typical test setup consists of a common inlet and outlet header feeding parallel channels. Slight variation in the dimensions of the channel could lead to flow maldistribution. This work investigates the effects of flow maldistribution on the friction factor prediction for microchannels. This is experimentally investigated by machining six parallel microchannels with different cross sectional areas to introduce flow maldistribution. This chapter describes the design of the experimental setup for investigating the effects of flow maldistribution on friction factor predictions in microchannels. The setup consists of a storage tank, water pump, digital flow meters, a set of parallel microchannels and a receiver tank. The following sections describe in the detail the flow circuit, instrumentation and data acquisition circuit and the experimental procedure for the investigation for flow maldistribution on the pressure drop and heat transfer in microchannels.
3.1 Flow Loop

Fig. 3.1: Flow loop for investigation single phase flows in plain microchannels

A schematic of the experimental loop is shown in Fig. 3.1. Degassed water is stored in the tank. The pump directs the water from the tank to the test section through a set of digital flow meters. Two flow meters one with 0-100ml/min and other with 100-1000ml/min capacity are used in parallel. This is an open loop and the heated water is led into a receiver where the flow rate can be further confirmed manually with a stop watch.
3.2 Test Section Design

![Photograph of Copper Block Test Section with Parallel Microchannels](image)

Fig. 3.2: Photograph of Copper Block Test Section with Parallel Microchannels

Figure 3.2 shows the actual photograph of the test section. It is a copper block with 6 parallel microchannels milled on the top surface. It has a hole below the microchannels which houses a cartridge heater which is used to heat the surface of the channels. It has two layers of thermocouples inserted on either sides of the copper block. The two layers are underneath the copper surface at two different heights so that heat flux normal to the surface of the channel based on the temperature differential in the two layers.
Figure 3.3 shows the header made of polycarbonate with the inlet and outlet manifolds. The water flow is marked on the diagram. It enters through the inlet manifold into the microchannels and then exits through the outlet header. The passage which connects the two manifolds with the inlet and outlet connections also has a provision for inserting thermocouples. E-type thermocouples are used. This ensures accurate inlet and outlet manifold temperatures. Also taps are provided on the inlet and outlet passage for measuring the pressure drop across the channel.
Fig. 3.4: Assembly of the cover plate, test section and the insulating base

Fig. 3.4 shows the three layer assembly of the test section. The bottom of copper block is attached to a phenolic base. This acts as an insulating block. This assembly is mounted on a stand and is covered with cotton which further acts as an insulation and then wrapped with cloth.
3.3 Channel Dimensions

Table 3.1 Individual channel dimensional variability

<table>
<thead>
<tr>
<th>CHANNEL</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean Height μm</td>
<td>209.3</td>
<td>207.3</td>
<td>194.0</td>
<td>190.1</td>
<td>208.4</td>
<td>210.1</td>
</tr>
<tr>
<td>Mean Width μm</td>
<td>268.5</td>
<td>264.6</td>
<td>265.8</td>
<td>262.3</td>
<td>236.9</td>
<td>266.4</td>
</tr>
<tr>
<td>Cross Section Area A_c x10^2 μm^2</td>
<td>560</td>
<td>546</td>
<td>514</td>
<td>498</td>
<td>492</td>
<td>559</td>
</tr>
</tbody>
</table>

The current test section has six parallel long channels. The individual channel dimensions are measured using a surface profilometer at 11 different locations along the flow length. Table 3.1 gives the height and width of each channel along with the cross sectional area of each channel. It is seen from the cross sectional area that channel one and five have the maximum and minimum cross sectional area respectively. Thus actual flow in these channels is going to vary the most for same pressure drop across these channels. Later when comparison will be made to check the effects of flow maldistribution, these two channels are considered for comparison purpose.

3.4 Instrumentation and Data Acquisition

Pressure is measured using a differential pressure transducer. The pressure transducer is a PX-26 series from Omega. The transducer is a silicon diaphragm that uses a wheat-stone bridge resistor network to measure the deflection. The excitation voltage is 10 VDC and the
output is from 0 to 100 mV. The pressure transducer range is selected to give the highest level of accuracy, with a range of 0 to 6.8 atm. The pressure transducers are calibrated using known pressures and the measured response of the transducer. A pressure calibrator from Omega is used to apply a known value of pressure. The range of the pressure calibrator is -100 to 200 kPa. The high side pressure port on the differential pressure transducer is exposed to the known values of pressure. Over twenty points are taken within the range of the specific pressure transducer. A linear curve fit is assumed and used to generate the calibration equation. If the range of the pressure transducer is 690 kPa, the transducer is calibrated up to 200 kPa and linear behavior is assumed thorough the remainder of the range.

A gear pump manufactured by Micropump, Inc., model: GA-V23.J9FS.G, is used to pump the working fluid. The maximum allowable pressure setting is 172 kPa, and it can deliver a flow rate ranging from 42 mL/min to 350 mL/min. The pump has a pulsation of 1.5%. Water is pumped through this pump. LabVIEW is also used to control the pump output by sending 0-5 DC voltage.

Two digital flowmeters (FLR 1007 and FLR 1010 from Omega) are used to accurately measure the flow rate. The FLR 1007 model measures flow rate from 10-100 ml/min while the FLR 1010 is for 100-1000 ml/min. The excitation voltage is 12.8 volts DC. The flow rate is measured by a miniature turbine wheel housed inside the flow meter. The wheel rotates with a speed proportional to volumetric flow rate of the fluid. The output signal of the flowmeters is compared with the set point for the flow rate and the pump is controlled according to the difference by LabVIEW.
3.5 Experimental Uncertainties:

The uncertainty is determined by the method of evaluating the bias and precision errors. The following equation is used to calculate the uncertainty in the experimental results which is based on Kline and McClintock (1953).

\[
U = \sqrt{\left(\frac{\partial R}{\partial v_1} U_1\right)^2 + \left(\frac{\partial R}{\partial v_2} U_2\right)^2 + \ldots + \left(\frac{\partial R}{\partial v_n} U_n\right)^2}
\]  

(3.1)

where \( U, R, \) and \( v \) are uncertainty interval, result, and variable, respectively.

The pressure transducer has an uncertainty of ±0.69 kPa. The temperature reading has an uncertainty of ±0.1°C. The power supply used to provide input power within ±0.05 V and current within ±0.005 amps uncertainty. The flow meter uncertainty in the volumetric flow measurement is ±0.0588 cc/min. The power measurement has an accuracy of ±0.5 Watts. The temperature difference measurements have an uncertainty of ±0.2°C. The resulting uncertainties are calculated for the pressure drop is 7.19%, and friction factor is 4.80%, at a median flow case. The major source of error is the temperature reading. This uncertainty is based on Steinke and Kandlikar [14] as the same instrumentation setup is used.

3.6 Experimental Procedure

Water is first degassed to ensure homogeneity of the medium. This is done by boiling the water in the storage tank and increasing the pressure to around 15 psi and then venting the pressure. This is repeated twice. Water at atmospheric condition and over 90 °C releases all the dissolved gasses. This degassed water is then allowed to cool down to room temperature. The current study focuses on the classical equations based on the laminar flow regime. Thus the Reynolds number for the experiments is kept under 2000.
A pump is used to direct the flow from the storage tank to the test section through flow meters. The flow rate is recorded in LabVIEW and flow rate is compared with the set point. This feedback is used to control the pump in order to push the exact quantity of water through the circuit. The pressure drop across the inlet and outlet header is measured by the differential pressure gage. Inlet temperature of water is also measured for determining the viscosity of fluid. Readings are recorded only after the flow has been stabilized.

Pressure drop readings and the corresponding flow rates are recorded and are used to calculate the apparent friction factor in each channel. Apparent friction factor includes the pressure drops due to both developing and fully developed flow. The experimental value is compared with the theoretical value to compare the effects of maldistribution on flow prediction using classical equations.
There has been very few data on enhanced microchannels. Enhanced Microchannels are capable of dissipating very large quantities of heat because of periodic interruption of boundary layer. However very high pressure drops are also encountered in this flow. Because of the complexity of the flow caused by the flow over the fins, there has been no analytical solution to predict the performance. In such a situation, predictive models are basically based on numerical models or correlations which need lot of experimental data. For minichannels, of similar configuration, there has been extensive research and many correlations have been developed based on the existing data. However, there has been no predictive model yet developed for enhanced microchannels. In this study, the pressure drop data is numerically modeled using CFD. Also, experiments are also carried out to validate the numerical results and existing minichannel correlations are tested on this data. Simple pressure drop experiments are carried out on one offset strip fin silicon microchannel for the above purpose.

Experiments on an enhanced silicon microchannel are carried out for flow range of 20-200 Reynolds number in order to validate the numerical scheme. The following sections will describe the flow loop and experimental procedure.

4.1 Flow Loop

A simple flow circuit is used which includes a pump, different pressure sensor and the offset strip fin microchannel. Figure 4.1 details the schematic diagram of the experimental setup.
Flow rates up to 80ml/min for water can be circulated through the enhanced microchannels using the peristaltic pump. This roughly translates to around 250 Reynolds number. Pressure drop is measured using a pressure differential transducer. Temperature of the water is also recorded before it enters the channel. This value of temperature is used to calculate the correct representative viscosity while numerically modeling the flow.
4.2 Test Section Details

The actual silicon microchannel is shown in fig. 4.2. The overall dimensions of the chip are 10X10 mm. There are 80 channels with channel width of 50 µm and height of 200 µm across the flow length while along the flow length; there are 40 rows of fins with fin length of 250 µm. Fins have smooth entry and exit passages. The silicon chip is housed inside a cavity which is then covered with an acrylic plate. Figure 4.3 shows the cover plate with the chip underneath it.

Fig. 4.2: Enhanced Silicon Microchannel, fin length 250µm, channel depth 200µm, channel width 50µm
The pump outlet connection is made with the inlet connection for the test fixture. The path takes a 90° bend to reach the header on the chip. Similar path is taken by the fluid while exiting the channels. Pressure drop is measured across the inlet and outlet feed lines. This pressure drop does include entry and exit losses associated with the area changes. These losses are deducted from the measured pressure drop.

For the enhanced channels only adiabatic flow is investigated just like plain microchannels. Experiments are carried out for flow in the laminar flow regime and the pressure drop across the channels is calculated. This data is then compared with the minichannel correlations. Since there are no predictive models developed so far in the literature for enhanced
microchannels, numerical modeling is carried out in order to predict the microchannel pressure drop data. The next chapter deals with the setup for the simulation of microchannel flows in offset strip fin configurations.

4.3 Test Procedure:

Very high pressures are observed in enhanced channels. Experiments are carried out in the laminar flow regime and at very flow Reynolds number range of 20-200. This range is selected as because of the large frictional losses encountered in microchannels, laminar flow regime is employed in practice. The pressure drop across the channels is measured with a differential pressure gage while the volumetric flow rate is given by the peristaltic pump. Steady state readings are recorded. Chapter Seven gives the details of the experimental runs.
Enhanced microchannels with offset strip fin configuration have not been extensively researched in the literature. There have been no predictive models developed yet for the same. Since performance of these microchannels is not evaluated, a numerical modeling is carried out to verify whether the enhanced microchannel flow can be modeled using the classical flow equations. The pressure drop is this numerically predicted using numerical modeling based on finite volume method. Commercially available Computational Fluid Dynamics Solver FLUENT is used for the computer simulations. It involves first recreating the flow geometry with the region of interest, assigning correct boundary conditions, meshing of the geometry, and then solving the Navier-Stokes Equation along with the continuity equation for prediction the pressure drop in the flow. This chapter discusses in detail the problem setup for the numerical modeling for pressure drop in enhanced microchannel. Heat transfer is not modeled in this work.

5.1 Flow Geometry:

The geometry is created and meshed using GAMBIT. The enhanced chip as described in the above section contains 80 of fins in the transverse direction with fin lengths of 250 μm. The total flow length is 10000μm which implies around 40 fins along the flow length. This type of repeated geometry makes the flow also repeatable after initial few rows of fins. Since the number of fins along the flow length is very large, it is a safe assumption to model the flow after a steady flow has been established. This is called as the periodic flow condition which is different than the conventional use of the term for straight channels. The first part in making the geometry is to
create identical flow path. Figure 5.1 shows a highly magnified image of the fin structure for the chip under consideration.

**Fig 5.1: 1000X magnified image of offset strip fins from KEYENCE microscope**

The numbers marked in the snapshot have the following values; 1-50µm, 2-50µm, 4-100µm. It is clear from the figure that the fins do not have flat front faces. There is a smooth entry and exit for the fluid into the channel. This geometry is very difficult to create in GAMBIT. An approximation is used to create this geometry. Marked on the above figure you see the dimensions which are used to make the geometry. Figure 5.2 shows the schematic of the flow path and the computational domain which is considered for pressure drop prediction.
Fig. 5.2 Schematic of Unit Cell and Computational Domain
Figure 5.3 shows the actual 3D model which is created in GAMBIT which is the graphical representation of the schematic of the computational domain. The symmetry and periodic nature of the flow which is inherent in the flow is the reason for modeling on part of geometry in FLUENT. The boundary conditions which are applied to this geometry are discussed in detail in the following section.
5.2 Zone Specifications

The physical and the operating characteristics of the computational model at its boundaries are given by the zone type specifications. There are basically two types; Boundary Type and the Continuum Type.

The Continuum Type is set to fluid for the current setup. This enables to characterize the unit cell within the domain as a fluid. The momentum and continuity equations are applied to the nodes or the cells that exist within the volume. Since the unit cell is a 3D representation of the enhanced channels, the boundary conditions are specified for the faces of the model. Each face or boundary of the model needs to given a boundary condition. Following are the conditions specified.

Wall Boundary Condition: The no slip boundary condition is given by assigning the faces as wall and giving the zero velocity. The top, bottom and fins are given the no slip condition.

Symmetry: The unit cell is the representation of the complete enhanced channel. The unit cell can be replicated along the symmetry lines. The symmetry faces are marked on the fig. 5.3. By specifying symmetry boundary condition, the flow and pressure gradients are identically zero along these faces. Because of this the physical conditions in the regions immediately adjacent to either side of the edge are identical to each other.

Periodic Boundary Condition: The inlet and the exit faces of the domain are marked as periodic. The mesh on these faces also need to be linked. When these faces are hard linked, GAMBIT associates the faces with each other. And any operation applied to one face is linked with the other face. In other words, the flow characteristics are at the exit face are replicated at the inlet wall. This leads to a “fully developed” flow condition. The term fully developed does not assume the conventional meaning. For the present case, the term fully developed refers to the repeatable flow pattern. The actual flow might be developing within the unit cell. However the
same flow pattern is obtained in the preceding and following unit cells. This is one of the assumptions which is used while modeling the flow in enhanced channels. The boundary conditions for the unit cell are clearly marked on the faces as shown in fig. 13.

5.3 Mesh Generation

The enhanced channel is approximated using all rectangular faces. Since the geometry is computational very straightforward, Quad Elements are used for meshing. This specifies that the mesh includes all quadrilateral mesh elements. For each mesh element have a range of meshing schemes that can be used. A Quad-Map Meshing scheme is used which creates a regular, structured grid of mesh elements. Quad-Map meshing scheme is applicable primarily to faces that are bounded by four or more edges and also the face should be logically rectangle [FLUENT docs ref]. As the geometry satisfies both these conditions a Quad Map Meshing Scheme is used. 163200 mesh volumes are created for the geometry. Figure 5.4 shows a top view of the meshed geometry.
5.4 Mesh Quality

It is important before the simulations are run that the mesh quality be checked. If the choice of meshing scheme is not correct, the results of the numerical modeling may not make any sense. EquiAngle Skew and MidAngle Skew are used to test the mesh quality. EquiAngle Skew is a normalized measure of skewness. \( Q_{EAS} = 0 \) suggest a equilateral element while \( Q_{EAS} = 1 \) represents a completely degenerate or poorly shaped element. The MidAngle Skew or \( Q_{MAS} \) applies only to quadrilateral and hexahedral elements. By definition of \( Q_{MAS} \) its value also lies in the range of 0 to 1. A relation of \( Q_{EAS} \) and mesh quality is defined in the FLUENT
documentation. It is assumed that in general high quality meshes contain elements that possess average $Q_{EAS}$ values of 0.1 for 2D element and 0.4 for 3D element.

For the current Quad-Map Meshing Scheme, 100% of total mesh elements have $Q_{EAS}$ value between 0 and 0.3 with worst element having a value of 0.295 while 90% of the total mesh elements have $Q_{MAS}$ value fall in the range of 0 and 0.3 with the worst element having value of 0.41. Thus overall for the type of geometry under consideration, the mesh quality is very good.

5.5 **Mesh Size Dependence**

Trial runs are performed on the meshed geometry to check the mesh size sensitivity. Mass flow rate of $2.4732e^{-6}$ kg/s is given to the unit cell for 3 different mesh sizes. The pressure gradient computed by FLUENT to maintain the specified flow rate is compared. Table 5.1 gives the mesh size, the value of pressure gradient for above given flow rate and the percent change in the pressure gradient.

**Table 5.1: Mesh Size Sensitivity**

<table>
<thead>
<tr>
<th>Mesh Size $\mu$m</th>
<th>Pressure Gradient Pa/m</th>
<th>% Change in Pressure Gradient</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>2206444</td>
<td>0.3</td>
</tr>
<tr>
<td>2.5</td>
<td>2213288</td>
<td>0.1</td>
</tr>
<tr>
<td>3</td>
<td>2217718</td>
<td>-</td>
</tr>
</tbody>
</table>
It is seen from the Table 5.1, that by reducing the mesh size from 3 µm to 2.5 µm, there is only 0.1 % change in the value of the pressure gradient, while changing the mesh size further down to 2 µm, there is a change of 0.3 % in the result. Thus mesh size of 2.5 µm is selected as the final mesh size. For 2 µm mesh size, the computational time increased a lot and since there is negligible change in the result, mesh size of 2.5 µm was selected to save computational time. Any further reduction in mesh size was not possible because of the processor limitations. For mesh size of 2.5 µm a total of 163200 mesh volumes are generated.

This meshed mathematical model is simulated in FLUENT. The objective of this simulation is to confirm if flow in enhanced microchannels can be predicted. Experiments in the Reynolds number range of 20-200 are performed as described in chapter four. Hence for numerical modeling, the same flow conditions are simulated which were used for the experiments.

In Chapters Three, Four and Five, the setup for experimental and numerical analysis for both plain and enhanced microchannels has been explained. The development of maldistribution analysis for plain microchannels is explained in the following chapter.
In the literature it is seen that there are contradictions on the applicability of conventional fluid flow equations on the microchannel data. It is thought that flow maldistribution in multiple channels could lead to this discrepancy in the literature. In this chapter, the maldistribution analysis is explained in detail. This includes estimating the flow in each channel and then estimating the friction characteristics and comparing it with theory. The following section describes the logic behind the estimation of the maldistribution and the data comparison with the theory.

For comparing the predictions based on maldistribution analysis with the uniform flow assumption, the uniform flow calculations are similar except for non uniform analysis, individual channel dimensions are considered while for uniform flow calculations each channel is assumed to have the same dimension, which ensures that there is no flow maldistribution. The calculations described below are for one with maldistribution. Same procedure is followed for uniform flow analysis.

6.1 Estimating Flow Maldistribution

Figure 6.1 illustrates the test section designed to investigate the effects of flow maldistribution on the prediction of friction factor using the classical equations. As described in the Chapter Four, the copper block has 6 parallel channels machined on the surface and water is fed through the inlet and outlet manifold. The pressure drop is measured across the header as
shown in the schematic. Since for all parallel channels have the same pressure drop across its end, the flow is uniformly distributed only if all the channels have the same dimension. However as shown in the fig. 6.1 six channels of varying dimensions are shown. This results in flow maldistribution in the channels.

Fig 6.1: Schematic of multiple microchannels with the inlet and outlet manifold configuration
Mass flow rate can be expressed in terms of pressure drop $\Delta p$, apparent friction factor $f_{app}$ and Reynolds number $Re$ by the following eq. (6.1).

\[
\dot{m} = \frac{\Delta p \cdot \rho}{2 \left( f_{app} \cdot Re \right) \mu L} A_c D_h^2
\]  

(6.1)

where $A_c$ and $D_h$ are the channel cross sectional area and hydraulic diameter respectively, $f_{app}$ represents the apparent friction factor which includes the pressure drops due to developing flow as well as fully developed flow.

For a channel with fixed cross section, the variables involved which would determine the mass flow rate are $f_{app}$, $Re$ and $\Delta p$. The channel length is around 55mm and for the flow rates under purview of this study, the developing flow length is not significant and for fully developed flows, the product $f_{app} \cdot Re$ is constant. The flow rate is hence dependent on the pressure drop and the channel dimensions. From the above equation, mass flow rate in each channel for the measured pressure drop can be found to be proportional to the product of cross sectional area and the square of hydraulic diameter as shown in eqn. (6.2)

\[
m_i \propto A_{ci} D_{hi}^2
\]  

(6.2)

Where $D_{hi}$ and $A_{ci}$ are the hydraulic diameter and cross sectional area of the $i^{th}$ channel and $m_i$ is the mass flow rate in that channel

The total flow rate is measured accurately using two digital flow meters. If $m_i$ is the flow through channel $i$, then

\[
\sum m_i = \dot{m}_{total}
\]  

(6.3)

Using eq. (6.2) and (6.3) the flow rate in each channel is found using
This flow rate is considered henceforth while predicting the friction factor for the individual channels.

### 6.2 Friction Factor Prediction

Once the mass flow in each channel is estimated, friction factor is found for each channel and it is compared with the theory. Using the individual channel flow rate and the measured pressure drop is used to calculate the experimental friction factor is each channel using the following eqn. (6.5)

\[
f_{\text{app}} \text{Re} = \frac{\Delta \rho \text{ext} \cdot \rho \cdot A_{ci} \cdot D_{hi}^2}{2 \mu \bar{m} \cdot L}
\]  

(6.5)

For theoretical value \(f_{\text{app}} \text{Re}\), table developed by Phillips (1987) for flow in rectangular microchannels is used. It takes into account the pressure drop due to developing flow as well as fully developed flow. However, in the present case, because the flow is nearly fully developed, the value of \(f_{\text{app}} \text{Re}\) tends to be same as fully developed flow for rectangular channel of that aspect ratio. The expression for \(f_{\text{app}} \text{Re}\) is given by Eq. (6.6), and the values of the constants \(a-f\) are obtained from Kandlikar et al. (2005).

\[
f_{\text{app}} \text{Re} = \frac{a + cx^{0.5} + ex^+}{1 + bx^{0.5} + dx^+ + fx^{+1.5}}
\]  

(6.6)
where the non-dimensional hydrodynamic entry length is represented by $x^+$ given by the following equation:

$$x^+ = \frac{L}{Re \cdot D_h} \quad (6.7)$$

The above calculated value represents the theoretical friction factor in the channel for the measured pressure drop and for the estimated flow rate in that channel. This theoretical friction factor, eqn. (6.6) is compared with experimental friction factor given by eqn. (6.5) for all 6 channels. There are thus 6 different plots. However for the uniform flow analysis only one plot is employed since all the six channels are of same dimension, same flow rate is used in each channel and hence same theoretical and experimental flow rate for each channel. The plots are presented in Chapter Eight.

Throughout the above mentioned calculations, the pressure drop was measured across the inlet and outlet manifold. It includes the losses in the 90° bends and the expansion and contraction losses. Equation (6.8) is used to calculate the pressure losses; $K_{90}$, $K_c$ and $K_e$ represent the coefficients for losses in the 90° bends, contraction and expansion losses. The values for these coefficients are based on Phillips (1987).

$$\Delta p_t = \left[ \left( \frac{A_c}{A_p} \right)^2 \cdot 2K_{90} + K_c + K_e \right] \frac{m^2}{2A_c \rho} \quad (6.8)$$
The pressure drop $\Delta p$ is referenced in this paper henceforth after deducting the pressure losses from the actual measured pressure drop and represents the net pressure drop in the microchannels. The apparent friction factor $f_{app}$ is the non dimensional form representing the same pressure drop.

The analysis presented here is applicable only to plain channels. For enhanced channels, there is no analytical solution to predict the performance. Existing minichannel correlations are tested with the new experimental data and the numerical modeling is validated by the data. The next chapter details the minichannel correlations which are used to predict the enhanced microchannel fluid flow performance.
CHAPTER SEVEN

ENHANCED MICROCHANNELS – FRICTION FACTOR ANALYSIS

Minichannel offset strip fin heat exchangers have been used for a long time. As the name suggests, these are called enhanced channels because of the enhancement in the heat transfer achieved by periodically interrupting the fluid flow length. However, as a result of fins interrupting the flow, wakes are formed and fluid mixing occurs. This makes it very difficult to analytically predict the performance. All the predictive models developed so far for minichannels with offset strip configuration have been based on either numerical modeling or correlations which are based on experimental data collected on different aspect ratio channels. For microchannels however neither numerical study has been carried out nor have any correlations been developed. This work aims to evaluate the flow in enhanced microchannels by numerically modeling the flow as well by generating experimental data and testing the validity of minichannel correlations for this data. There are few correlations which have been widely used for predicting performance in the minichannels. This chapter presents those correlations which would be used to analyze the microchannel data.

7.1 Enhanced Microchannel Data Reduction

7.1.1 Experimental Friction Factor

Experiments were carried out an enhanced silicon microchannel for low Reynolds number flow as described in Chapter Four. Table 7.1 gives the values of flow rates and the corresponding pressure drops recorded for the experiments on enhanced microchannels.
Table 7.1: Experimental Data for Enhanced Microchannels

<table>
<thead>
<tr>
<th>$\dot{m} \times 10^{-4}$ (kg/s)</th>
<th>1.98</th>
<th>3.90</th>
<th>5.83</th>
<th>7.76</th>
<th>9.68</th>
<th>11.63</th>
<th>13.54</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta P$ (Pa)</td>
<td>9426</td>
<td>20635</td>
<td>32875</td>
<td>47057</td>
<td>61853</td>
<td>77478</td>
<td>94879</td>
</tr>
</tbody>
</table>

The measured pressure drop is converted into non dimensional parameter $fRe$ given by eqn. (7.1)

$$fRe = \frac{\Delta p \cdot \rho \cdot A_c \cdot D_h^2}{2 \cdot \mu \cdot m \cdot L}$$  \hspace{1cm} (7.1)

where $\Delta p$ is the experimentally measured pressure drop from Table 7.1, $L$ is the total length of the test chip and $A_c$ and $D_h$ are the total free flow area and hydraulic diameter as defined in eqn. (7.2) and (7.3).

$$A_c = n \cdot a \cdot b$$  \hspace{1cm} (7.2)

where $n$ represents the number of channels (80- for current chip), $a$ is the channel width and $b$ is channel height.

Hydraulic Diameter is calculated as defined by Steinke (2005)

$$D_h = \frac{2 \cdot a \cdot b \cdot L_f}{s \cdot b + \frac{L_f}{2} \left( a + 2 \cdot b \right)}$$  \hspace{1cm} (7.3)

where $L_f$ is the fin length and $s$ is fin thickness.

$fRe$ which is also called as Poiseuille number is plotted against the Reynolds number given by eqn. (7.4)

$$Re = \frac{G \cdot D_h}{\mu}$$  \hspace{1cm} (7.4)

where $G$ represents the mass flux based on the measured flow rate and the total cross sectional area $A_c$. 

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### 7.1.2 Numerical Friction Factor

The above section describes the calculation of experimental friction factor for enhanced microchannels. The objective of the experiment is to validate the CFD results given by FLUENT on enhanced microchannel flows. Chapter Five described the numerical setup of the problem. FLUENT calculates the value of pressure gradient required to maintain the mass flow rate for a periodic flow. This value of pressure gradient is converted into \( f_Re \) using the same eqn. (7.1). The value of \( \Delta p/L \) is replaced by the pressure gradient term given by FLUENT while all the other parameters have the same value. The experimental and numerical value of \( f_Re \) is compared to validate the CFD results for enhanced microchannels.

### 7.2 Enhanced Minichannel Correlation Data Reduction

The widely used models for mini channel offset strip fins are developed by Manglik and Bergles (1995), Joshi and Webb (1987), and Weiting (1977). These correlations mainly cover the practical range of Reynolds number used in industry for minichannels, from 200-4000. Following are the correlations used to predict the friction factor. However, the current experiments were carried out in flow ranges much lower than the above Reynolds values. So when minichannel correlations are tested against the microchannel data, extrapolation of the correlations is needed. Hence this comparison of minichannel correlations for microchannel data is just to check the validity of these correlations in low Reynolds number flow. However, these correlations have different definitions of hydraulic diameter which results in different value of Reynolds number for same flow rate and for the same geometry. Thus friction factor plots cannot be compared with Reynolds number. For this reason, the experimental flow rate and the test geometry is used to calculated the respective Reynolds number and corresponding friction factor.
is calculated using eqn. (7.5)-(7.10). This value of friction factor as predicted by the enhanced minichannel correlation is then converted into the pressure drop by using eqn. (7.1). It is then directly compared with the experimental pressure drop for enhanced microchannel as measured in table 7.1.

Following are expressions for the minichannel correlations.

Manglik and Bergles(1995):

\[ f = 9.624\left(\frac{S}{H}\right)^{-0.186}\left(\frac{L}{L}\right)^{0.305}\left(\frac{L}{S}\right)^{-0.266}\frac{Re}{Dh}^{-0.742} \]  \hspace{1cm} (7.5)

\[ Dh = \frac{4sHL}{2(sL + HL + tH) + ts} \]  \hspace{1cm} (7.6)

Where \( f \), \( t \), \( s \), \( H \), \( L \) stand for friction factor, fin thickness, fin spacing, channel depth and fin length respectively.

Joshi and Webb(1987):

\[ f = 8.12\left(\frac{L}{Dh}\right)^{-0.41}\left(\frac{S}{H}\right)^{-0.02}\frac{Re}{Dh}^{-0.74} \]  \hspace{1cm} (7.7)

\[ Dh = \frac{2(s - t)HL}{sL + HL + tH} \]  \hspace{1cm} (7.8)

Weiting(1977):

\[ f = 7.661\left(\frac{L}{Dh}\right)^{-0.384}\left(\frac{S}{H}\right)^{-0.092}\frac{Re}{Dh}^{-0.712} \]  \hspace{1cm} (7.9)

\[ Dh = \frac{2sH}{s + H} \]  \hspace{1cm} (7.10)
The existing correlations are compared with the new experimental pressure drop data which was collected on the offset strip fin microchannel. This comparison is a valid estimate to check the applicability of minichannel correlations for low Reynolds number microchannel flow.
CHAPTER EIGHT

RESULTS

Experiments were performed on both plain and enhanced microchannels in order to check the validity of classical flow equations on the microchannel data. For plain microchannels the effects of flow maldistribution on the predictive models is investigated while for enhanced microchannels the existing minichannel correlations were tested with the data. Enhanced channels were also numerically modeled. In this chapter the results for the maldistribution analysis and the results of numerical modeling for enhanced channels are presented.

8.1 Plain Channels

In Chapter Two, past studies on plain microchannels were reviewed with focus on work dealing with the flow prediction using the classical equations by considering equal flow distribution. It was pointed out that there is discrepancy in predicting the friction factor in parallel microchannels using the classical equations. Flow maldistribution was assumed to be one of the reasons for this discrepancy. In Chapter Six, flow maldistribution analysis was developed where in the flow in each channel was estimated based on the hydraulic diameter and the cross sectional area of the channels as per eqn. (6.4). Table 8.1 gives the results of estimated channel flow rates for all the six channels. \( \Delta P \) represents the pressure drop across the test section which generates the total flow rate \( \dot{m}_{\text{total}} \). \( \dot{m}_{\text{uniform}} \) represents the mass flow rate in each channel without considering flow maldistribution. So \( \dot{m}_{\text{uniform}} \) is just one sixth of the measured total flow rate \( \dot{m}_{\text{total}} \). It is seen that since channel one and five have the extreme cross sectional areas, the mass flow rates in these channels also form the limits amongst all the six parallel channels.
Table 8.1: Estimated Channel Flow Rates and Measured Pressure Drop for Plain Microchannels

<table>
<thead>
<tr>
<th>( \dot{m}_{\text{total}} \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_{\text{uniform}} \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_1 \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_2 \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_3 \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_4 \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_5 \times 10^{-5} ) (kg/s)</th>
<th>( \dot{m}_6 \times 10^{-5} ) (kg/s)</th>
<th>( \Delta P ) (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>18.0</td>
<td>2.99</td>
<td>3.36</td>
<td>3.20</td>
<td>2.80</td>
<td>2.63</td>
<td>2.62</td>
<td>3.34</td>
<td>100.9</td>
</tr>
<tr>
<td>23.8</td>
<td>3.96</td>
<td>4.45</td>
<td>4.24</td>
<td>3.71</td>
<td>3.48</td>
<td>3.47</td>
<td>4.43</td>
<td>149.4</td>
</tr>
<tr>
<td>29.5</td>
<td>4.92</td>
<td>5.53</td>
<td>5.26</td>
<td>4.62</td>
<td>4.33</td>
<td>4.31</td>
<td>5.51</td>
<td>189.6</td>
</tr>
<tr>
<td>35.3</td>
<td>5.89</td>
<td>6.61</td>
<td>6.30</td>
<td>5.52</td>
<td>5.17</td>
<td>5.16</td>
<td>6.58</td>
<td>223.6</td>
</tr>
<tr>
<td>46.9</td>
<td>7.82</td>
<td>8.78</td>
<td>8.36</td>
<td>7.33</td>
<td>6.87</td>
<td>6.84</td>
<td>8.74</td>
<td>302.1</td>
</tr>
<tr>
<td>64.3</td>
<td>10.7</td>
<td>12.0</td>
<td>11.5</td>
<td>10.0</td>
<td>9.41</td>
<td>9.38</td>
<td>12.0</td>
<td>423.2</td>
</tr>
<tr>
<td>70.0</td>
<td>11.7</td>
<td>13.1</td>
<td>12.5</td>
<td>10.9</td>
<td>10.3</td>
<td>10.2</td>
<td>13.0</td>
<td>451.5</td>
</tr>
<tr>
<td>81.7</td>
<td>13.6</td>
<td>15.3</td>
<td>14.6</td>
<td>12.8</td>
<td>12.0</td>
<td>11.9</td>
<td>15.2</td>
<td>557.1</td>
</tr>
<tr>
<td>93.3</td>
<td>15.6</td>
<td>17.5</td>
<td>16.6</td>
<td>14.6</td>
<td>13.7</td>
<td>13.6</td>
<td>17.4</td>
<td>667.0</td>
</tr>
<tr>
<td>99.1</td>
<td>16.5</td>
<td>18.5</td>
<td>17.7</td>
<td>15.5</td>
<td>14.5</td>
<td>14.5</td>
<td>18.5</td>
<td>727.1</td>
</tr>
<tr>
<td>105.0</td>
<td>17.5</td>
<td>19.6</td>
<td>18.7</td>
<td>16.4</td>
<td>15.4</td>
<td>15.3</td>
<td>19.6</td>
<td>793.3</td>
</tr>
<tr>
<td>110.7</td>
<td>18.4</td>
<td>20.7</td>
<td>19.7</td>
<td>17.3</td>
<td>16.2</td>
<td>16.1</td>
<td>20.6</td>
<td>853.0</td>
</tr>
</tbody>
</table>

Using this channel flow rate and the measured pressure drop as given in Table 8.1, both experimental and theoretical friction factor are calculated using eqns. (6.5) and (6.6) respectively. The comparison is made for each channel. As described in Table 3.1, each channel has a slightly varying dimension. This introduces flow maldistribution and thus each channel have different frictional losses. Thus each channel is individually analyzed and in the next 6 plots, fig. 8.1 a-f, the results for the theoretical and experimental friction factor comparison using the maldistribution analysis are shown.
(a) Comparison of Theoretical Prediction and Processed Experimental Data on Friction Factor for Channel 1
(b) Comparison of Theoretical Prediction and Processed Experimental Data on Friction Factor for Channel 2
(c) Comparison of Theoretical Prediction and Processed Experimental Data on Friction Factor for Channel 3
(d) Comparison of Theoretical Prediction and Processed Experimental Data on Friction Factor for Channel 4
(e) Comparison of Theoretical Prediction and Processed Experimental Data on Friction Factor for Channel 5
Theoretical Prediction

(f) Comparison of Theoretical Prediction and Processed Experimental Data on Friction Factor for Channel 6

Fig. 8.1 Plot of dimensionless pressure drop using maldistribution for channels 1(a) to 6(f)

The y axis is the factor fRe which represents the non dimensional pressure drop and it is plotted against the Reynolds number. It is evident from the plots that using maldistribution analysis, friction factor can be very accurately predicted using the classical fluid flow equations. These results are compared with analysis which does not include maldistribution.

For the uniform flow analysis, all the 6 channels are considered to be of same dimension and thus eqn. (6.4) ensures that the mass flow rate in all the 6 channels would be same. Thus for uniform flow analysis, all channels will have the same flow rate, and same pressure drop across it, and hence resulting in same friction factor. Hence there is no need for separate plots for each channel and fig. 8.2 (a) and (b) represent the comparison between theoretical and experimental
friction factor using equal flow in all six channels. The difference in fig. 8.2 (a) and (b) is of importance.

![Graph showing comparison of theoretical prediction with experimental data for friction factor (fRe) vs. Reynolds number (Re)](image)

*(a) Uniform Flow Results Based On Channel 1 Reference Dimension*

Plot 8.1 (a) shows the comparison of theoretical prediction based on classical equations with the experimental friction factor. As mentioned before, for uniform flow analysis, all the channels are considered to be of same dimension. For the above plot, channel 1 dimensions are applied to all channels. It is seen from the plot that if all the 6 microchannels had dimensions of channel one as given in table 3.1, then the theory under predicts the microchannel data by over 20%. It would be good to note at this time, that the variation in the channel dimensions is just in the order of ten microns. Thus if flow maldistribution is not accounted and all the channel dimensions are considered to be of channel one, then it can be concluded from fig. 8.2 (a) that the conventional theory cannot predict the microchannel data.
Channel one and five represent the channels with maximum and minimum flow areas. From fig. 8.2 (a) it is seen that with channel one as the reference dimension for all the channels, the theory underpredicts the microchannel friction factor. Figure 8.2 (b) however represents the comparison of theoretical prediction of friction factor with the experimental data with all the channels assuming the value of channel five as per table 3.1. Although the theoretical prediction is off only by around 10%, it is interesting to point out that all the experimental data lies below the theoretical prediction, meaning the theory overpredicts the microchannel data. Thus it is clear from these plots that if maldistribution is present and not accounted for then the predictions using conventional theory cannot predict the data accurately. Since the variation in channel
dimensions is very less, it can go unaccounted and it might lead to erroneous conclusion that the conventional theory is not applicable to microchannel flows. Next results of CFD analysis for adiabatic flows in enhanced microchannels is presented.

8.2 Enhanced Microchannels

8.2.1 CFD Results of Enhanced Microchannels

FLUENT is used to predict the adiabatic flow in enhanced Silicon Microchannels. Table 8.2 gives results of the CFD analysis in comparison with the experimental values.

<table>
<thead>
<tr>
<th>Mass Flux kg/m²s</th>
<th>Re</th>
<th>Experimental Pressure Gradient (kPa/m)</th>
<th>Numerical Pressure Gradient (kPa/m)</th>
<th>Expt/Numerical Pressure Gradient Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>250.76</td>
<td>23.82</td>
<td>942.6</td>
<td>1035.0</td>
<td>0.91</td>
</tr>
<tr>
<td>494.65</td>
<td>46.99</td>
<td>2063.5</td>
<td>2213.2</td>
<td>0.93</td>
</tr>
<tr>
<td>738.53</td>
<td>70.16</td>
<td>3287.5</td>
<td>3587.8</td>
<td>0.92</td>
</tr>
<tr>
<td>982.42</td>
<td>93.33</td>
<td>4705.7</td>
<td>5151.5</td>
<td>0.91</td>
</tr>
<tr>
<td>1226.30</td>
<td>116.50</td>
<td>6185.3</td>
<td>6890.1</td>
<td>0.90</td>
</tr>
<tr>
<td>1470.19</td>
<td>139.67</td>
<td>7747.8</td>
<td>8787.6</td>
<td>0.88</td>
</tr>
<tr>
<td>1714.07</td>
<td>162.84</td>
<td>9487.9</td>
<td>10829.6</td>
<td>0.88</td>
</tr>
</tbody>
</table>

In Chapter Five it is mentioned that the flow problem for CFD analysis has been set up as a periodic flow. For FLUENT, periodic flow field is computed either by inputing the mass flow rate or the pressure gradient across the boundary. Since we are simulating the experimental flow conditions, we input the mass flow rate for the unit cell. The initial value of pressure gradient to maintain this flow is not known. FLUENT iterates the solution to calculate the pressure gradient until the mass flow rate specified in the periodic flow is achieved. This value of pressure gradient is given in table 8.2.
Figure 8.3 shows the variation of the static pressure across a plane which is midway between the top and bottom walls. It is seen that the sharp peaks in the pressure corresponds to the tip of fin where there is a pressure build up. The static pressure is represented against the x coordinate. Since the length of the unit cell is 500 µm, the absolute value of pressure drop across this unit cell is very small.
Figure 8.4 shows the velocity plots for the inlet and outlet face of the unit cell. This plot confirms the periodic flow assumption where the velocities at the outlet are repeated at the inlet boundary, thus establishing a “fully developed flow”. 0 m position on the x coordinate has the symmetry boundary condition, just the tangent at that point would be horizontal meaning that there is no gradient of velocity at the symmetry point. The extreme point at 25 µm represents the first fin as shown in fig. 5.3. Since this is a wall boundary condition, there is no slip boundary condition and thus zero velocity.
Figure 8.5 represents the velocity plot over the fin. The fin is highlighted in the plot for illustration. This velocity plot is for the maximum flow rate tested experimentally on the enhanced microchannels. It is seen that even at the high flow, there is no flow separation or a formation of recirculation zone.
Fig. 8.6: Comparison of Experimental and Numerical Poiseuille Number

The results of CFD analysis of enhanced channels are shown in fig. 8.6. The experimental pressure drop and numerical pressure gradient is converted into Poiseuille number as described in section 7.1. It is seen that the numerical predictions are within 10% of the experimental data. This difference can be associated with the approximation of the fin. The fin is modeled to be with sharp edges while the actual fin is one with smooth edges. Also the pressure gradient calculated by FLUENT is for a “fully developed” flow condition. However, for first few rows of fins along the flow length, the flow pattern will not be fully developed which also will adds to the difference of the results.
It was seen in Fig. 8.6 that the enhanced microchannel friction factor can be predicted using CFD. Figure 8.7 shows the comparison of the pressure drop as predicted by the enhanced minichannel correlations with the experimental friction factor for enhanced microchannels. Section 7.2 describes the calculation of pressure drop using minichannel correlations. As mentioned earlier, the minichannel correlations are extrapolated to match the flow rates employed in experiments for enhanced microchannels. Since these correlations are based on experiments carried out on compact heat exchanger, the validity of these correlations is not known. It is seen from Fig. 8.7 that the Weiting (1977) correlation predicts the enhance
microchannel data accurately. However Manglik and Bergles (1995) and Joshi and Webb (1987) correlation does not accurately predict the microchannel performance. It would be unwise to conclude anything at this juncture since the correlations are tested only against one set of enhanced microchannel geometry. However it is apparent that some enhanced minichannel correlations can also predict the pressure drop for low Reynolds number flow in enhanced microchannels.
CHAPTER NINE

CONCLUSIONS AND FUTURE WORK

9.1 Conclusions

The objective of the present was to evaluate experimentally and numerically single phase adiabatic flows in plain and enhanced channels. For plain channels, a discrepancy is noted in the literature for the theoretical predictions of microchannel data using equal flow distribution in a set of parallel microchannels. As outlined in the flowchart in fig. 1.4 (a), flow maldistribution induced by slight variation in the channel dimension is thought to be one of the reasons for this discrepancy. A careful set of experiments is designed to study the effect of non-uniform flow distribution in each channel on the friction factor prediction of microchannels. The pressure drop measurements are made across the inlet and outlet manifolds. The developing flow theory equations are used to compare with the experimental data. Since each channel is of different dimensions, mass flow rate is calculated based on the flow cross sectional area and the measured pressure drop. From the maldistribution analysis following conclusions can be drawn.

1. Slight variation in channel dimensions leads to uneven flow distribution in parallel microchannels
2. The pressure drop predictions based on the non-uniform flow distribution resulting from the channel area variation are in good agreement with the experimental data when the developing flow theory is used.
3. Channel dimensions need to be measured accurately. Variations in the individual channel dimensions are responsible for deviations from the theoretical friction factor predictions during laminar flow in smooth microchannels.
Enhanced Microchannels were also experimentally and numerically evaluated. Literature review indicates that although offset strip fin minichannels have been extensively in the past, very few investigations have been done on enhanced microchannels. Many correlations have been developed to predict the friction factor of enhanced minichannels however there is no predictive model for enhanced microchannels. FLUENT, a CFD software, is used to predict adiabatic flows in enhanced microchannels. To validate the CFD analysis, experiments were carried out on enhanced microchannels. Validity of existing minichannel correlations in predicting pressure drop of enhanced microchannels is also carried out as outlined in fig. 1.4 (b). Following points can be concluded from the numerical evaluation of adiabatic flows in enhanced microchannels.

1. CFD can predict the friction factor for adiabatic flows in enhanced microchannels within 10%.
2. Select minichannel correlations can be applied to the present microchannel data, pressure drop is predicted within 20%.

9.2 Future Work

Adiabatic flows in plain and enhanced microchannels are investigated experimentally and numerically. For plain channels the discrepancy in the friction factor prediction using classical fluid flow equations was investigated and flow maldistribution due to channel size variation was attributed as one of the possible reasons. This work could be extended for heat transfer and maldistribution effects on heat transfer predictions should be carried out. For enhanced microchannels, since no predictive models have been developed for enhanced microchannels, CFD analysis was carried out to predict the friction factor. It is seen that both the minichannel correlations and the CFD analysis can predict the pressure drops. However in order to use the
existing minichannel correlations to predict the friction factor in enhanced microchannels, the
correlation should be tested with more microchannel data on channels with different aspect ratio.
Since there is no data available in the literature, more experiments needs to be carried out to
characterize the flow in enhanced microchannels. Also heat transfer predictions should also be
investigated using CFD analysis and the applicability of existing minichannel correlations for
heat transfer should be validated for enhanced microchannel data.
REFERENCES


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