Experimental Study of Heat Transfer and Pressure Drop Over an Array of Short Micro Pin Fins

Aditya Bhat
agb6235@rit.edu

Follow this and additional works at: https://scholarworks.rit.edu/theses

Recommended Citation

This Thesis is brought to you for free and open access by RIT Scholar Works. It has been accepted for inclusion in Theses by an authorized administrator of RIT Scholar Works. For more information, please contact ritscholarworks@rit.edu.
EXPERIMENTAL STUDY OF HEAT TRANSFER AND PRESSURE DROP OVER AN ARRAY OF SHORT MICRO PIN FINS

Aditya Bhat

A Thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

Department of Mechanical Engineering
Kate Gleason College of Engineering

Rochester Institute of Technology

Rochester, New York

December 1st, 2015
Approved by:

___________________________________________________________
Dr. S.G. Kandlikar, Professor
Thesis Advisor, Department of Mechanical Engineering

___________________________________________________________
Dr. Robert Stevens, Professor
Committee Member, Department of Mechanical Engineering

___________________________________________________________
Dr. Jason Kolodziej, Associate Professor
Committee Member, Department of Mechanical Engineering

___________________________________________________________
Dr. A. Crassidis, Professor
Department Representative, Department of Mechanical Engineering
Contents

Nomenclature ........................................................................................................................................ 6
Abstract .................................................................................................................................................... 7
1. Introduction ........................................................................................................................................... 8
   1.1 Convective heat transfer .................................................................................................................. 8
   1.2 Heat sinks .......................................................................................................................................... 9
       1.2.1 Design considerations for heat sinks ...................................................................................... 9
   1.3 Electronics cooling .......................................................................................................................... 12
2. Literature Review ............................................................................................................................... 14
   2.1 Microchannels ................................................................................................................................ 14
   2.2 Micro pin fins ................................................................................................................................... 15
   2.3 Short micro pin fins and microchannels ......................................................................................... 18
3. Objectives of the present study ......................................................................................................... 21
4. Experimental details ........................................................................................................................... 22
   4.1 Test Surface (copper chip) .............................................................................................................. 22
   4.2 Gasket ............................................................................................................................................... 23
   4.3 Ceramic Base .................................................................................................................................. 24
   4.4 Gasket compression experiment .................................................................................................... 24
       4.4.1 Gasket compression after assembly ................................................................................... 24
       4.4.2 The gasket experiment ........................................................................................................... 25
   4.5 Flow channel and top cover ........................................................................................................... 26
   4.6 Housing .......................................................................................................................................... 27
   4.7 Flow loop ......................................................................................................................................... 28
   4.8 Measurements ............................................................................................................................... 28
5. Data reduction ....................................................................................................................................... 29
   5.1 Experimental values of Heat Transfer Coefficient and Nusselt number ........................................ 29
   5.2 Experimental values of Pressure drop and Friction factor ............................................................ 30
       Pressure loss in straight pipes ........................................................................................................ 30
       The pressure drop at the bends ....................................................................................................... 31
       Experimental pressure drop ........................................................................................................... 31
   5.3 Comparison with correlations ....................................................................................................... 31
       Friction factor .................................................................................................................................. 31
       Nusselt Number .............................................................................................................................. 32
5.4 Uncertainties.................................................................................................................. 32
5.5 Experimental procedure ................................................................................................ 32
6. Set-up validation .............................................................................................................. 33
   Nusselt number for a theoretical plain surface ............................................................... 33
   Friction factor for a theoretical plain surface ................................................................. 34
   Discrepancy in heat transfer coefficient ......................................................................... 35
7. Results ............................................................................................................................. 36
   7.1 Heat transfer coefficient ......................................................................................... 37
   7.1.1 Effect of fin height on heat transfer coefficient .................................................... 37
   7.1.2 Effect of free flow area on the heat transfer coefficient from a surface .............. 37
   7.2 Pressure drop ........................................................................................................... 39
   7.2.1 Effect of free flow area on pressure drop ............................................................. 41
   7.3 Comparison with correlation ................................................................................. 41
      7.3.1 Comparison of pressure drop values ................................................................. 42
      7.3.2 Nusselt Number correlation ............................................................................. 44
   7.4 Repeatability ............................................................................................................ 46
      7.4.1 Pressure drop .................................................................................................. 46
      7.4.2 Heat transfer coefficient .................................................................................. 46
      Repeatability issues with heat transfer coefficient .................................................... 47
    7.5 Analysis of the correlation ..................................................................................... 47
8. Conclusion ....................................................................................................................... 49
9. References ...................................................................................................................... 50
Figure 1 Example of thermal resistances generally encountered in commercial chip cooling applications.

Figure 2 Types of fins (shape, arrangement and orientation) used for commercial applications.

Figure 3 Moore’s Law [20].

Figure 4 Circular and Oblong fin geometries used by Metzger.

Figure 5 Different micro pin fin shapes studied by Tullius et al.

Figure 6 Nusselt number and pressure drop values observed for square staggered micro pin fins for fin height to channel height ratios from 0 to 0.75.

Figure 7 Test surface (copper chip).

Figure 8 Ceramic Base.

Figure 9 Schematic of ceramic base with chip and gasket and a top cover.

Figure 10 Heater Block.

Figure 11 Flow loop.

Figure 12 Test setup.

Figure 13 Bend loss coefficient proposed by Babcock and Wilcox Co. (1978).

Figure 14 Set-up validation plots showing heat transfer and pressure drop over a plain surface.

Figure 15 Experimental heat transfer coefficient for chips (a) c1, (b) c2 & (c) c3 with varying gasket thicknesses (g1, g2 & g3).

Figure 16 Experimental heat transfer coefficient for chips c1, c2 & c3 for different gasket thicknesses g1 (a), g2 (b) & g3 (c).

Figure 17 Experimental values of pressure drop vs Reynolds number at varying gasket thicknesses.

Figure 18 Experimental pressure drop values for chips (a) c1, (b) c2 and (c) c3 at gasket thickness g1, g2 and g3.

Figure 19 Comparison of experimental pressure drop values with the correlation proposed by Tullius et al.

Figure 20 Comparison of experimental Nusselt number values with the correlation proposed by Tullius et al.

Figure 21 Comparison of experimental Nusselt number values with the correlation proposed by Tullius et al.

Figure 22 Pressure drop repeatability plots for c1-g2 and c2-g2.

Figure 23 Heat transfer coefficient repeatability plot for c1-g2 (a) and c2-g2 (b).

Table 1 Fin dimensions and heat transfer area enhancement.

Table 2 Gasket dimensions.

Table 3 Test Matrix.

Table 4 List of friction factor correlations studied.

Table 5 List of Nusselt Number correlations studied.

Table 6 Experimental Uncertainties.

Table 7 Constants used in equation 12.

Table 8 mean error values for correlations developed by Tullius et al [17].
### Nomenclature

<table>
<thead>
<tr>
<th>Name</th>
<th>Units</th>
<th>Name</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_{as}$</td>
<td>$mm^2$</td>
<td>$L_f$</td>
<td>mm</td>
</tr>
<tr>
<td>$A_f$</td>
<td>$mm^2$</td>
<td>$N_f$</td>
<td>--</td>
</tr>
<tr>
<td>$c$</td>
<td>--</td>
<td>$Nu$</td>
<td>--</td>
</tr>
<tr>
<td>$D$</td>
<td>$mm$</td>
<td>$\Delta p$</td>
<td>kPa</td>
</tr>
<tr>
<td>$D_c$</td>
<td>$mm$</td>
<td>$Pr$</td>
<td>--</td>
</tr>
<tr>
<td>$D_f$</td>
<td>$mm$</td>
<td>$q''$</td>
<td>W/m$^2$</td>
</tr>
<tr>
<td>$D_p$</td>
<td>$mm$</td>
<td>$R_p$</td>
<td>mm</td>
</tr>
<tr>
<td>$f$</td>
<td>--</td>
<td>$Re$</td>
<td>--</td>
</tr>
<tr>
<td>$g$</td>
<td>$mm$</td>
<td>$S$</td>
<td>mm</td>
</tr>
<tr>
<td>$h$</td>
<td>$W/m^2K$</td>
<td>$T$</td>
<td>°C</td>
</tr>
<tr>
<td>$H_c$</td>
<td>$mm$</td>
<td>$u$</td>
<td>m/s</td>
</tr>
<tr>
<td>$H_f$</td>
<td>$mm$</td>
<td>$W_c$</td>
<td>mm</td>
</tr>
<tr>
<td>$k$</td>
<td>$W/m^2$</td>
<td>$W_f$</td>
<td>mm</td>
</tr>
<tr>
<td>$k_b$</td>
<td>--</td>
<td>$x^*$</td>
<td>--</td>
</tr>
<tr>
<td>$L_c$</td>
<td>$mm$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

#### Symbols

| $\alpha$                      | Aspect ratio |
| $\mu$                         | Dynamic viscosity |
| $\eta$                        | Enhancement factor |
| $\eta_f$                      | Fin efficiency |

#### Subscripts

| $app$                         | Apparent |
| $avg$                         | Average |
| $c$                           | Channel |
| $cu$                          | Copper |
| $f$                           | Fin |
| $fl$                          | Fluid |
| $l$                           | Longitudinal (pitch) |
| $LMTD$                        | Log Mean Temperature Difference |
| $m$                           | Mean |
| $p$                           | Perimeter |
| $pi$                          | Pipe |
| $plain$                       | Plain surface (unfinned) |
| $s$                           | Surface |
| $t$                           | Transverse (pitch) |
| $th$                          | Theoretical |
Abstract

Studies on thermal enhancement for electronic chips has been gaining prominence as increased transistor density in the chips calls for larger heat dissipation. Various enhancement techniques have been proposed ever since 1981, to enhance the heat dissipation from the chip surface. Micro pin fins have been gaining recognition as a highly favorable surface enhancement due to the design versatility it provides in the form of myriad geometric shapes and fin arrangements as opposed to conventional microchannels. The micro pin fins however, present a larger pressure drop over the surface as compared to other conventional methods which reduces the thermal efficiency of the chip surface. To reduce the pressure drop associated with micro pin fins, short micro pin fins were proposed. A short micro pin fin arrangement is similar to micro pin fin arrays, with one change, in that short micro pin fins have a clearance between the fins and the top of the channel. The current study focusses on heat transfer and pressure drop over short micro pin fin arrays.

Experimental studies were conducted over 10 mm × 10 mm with fin heights varying from 200 to 500 µm and clearance over the fins varying from 265 to 900 µm. Distilled water was used as the cooling medium. The heat transfer coefficient and pressure drop characteristics were evaluated at varying fin heights and varying clearance of the surfaces with an aim to identify optimum fin height and clearance parameters. The heat transfer coefficient and pressure drop data obtained from experiments were also evaluated with the correlation proposed by Tullius et al. [17].

Data showed that the highest heat transfer coefficient was observed for fins with the largest fin height. When fin clearance was evaluated for its effect on heat transfer coefficient, a hint of mixing phenomenon leading to enhancement in heat transfer coefficient was observed at higher clearance values. A higher pressure drop was observed at longer fins owing to the increased friction factor at the fin walls. The highest pressure drop of over 100 kPa was observed for a chip gasket combination which consisted of the longest fins with the least amount of clearance. It was also observed that the Nusselt number and Pressure drop correlations proposed by Tullius et al was not able to accurately predict the experimental data. However, the correlation did show the same trend as the experimental data, hence, the present correlation could be modified or used as a basis for new correlations of Nusselt number and friction factor.
1. Introduction

Electronic devices now-a-days show a tendency for faster, smaller and denser chips to cope with the high computational demands expected of them. As the chips get more powerful and smaller, thermal management of these chips becomes imperative. As the transistor density increases, the heat flux of the system increases. Without an efficient way of dissipating this generated heat into the environment, the temperature of the chips would increase and will cause the chip to stop working. To address the problem of thermal management in electronic chips, various types of heat sinks have been designed by researchers.

Heat transfer through a system takes place in the form of conduction, convection and radiation. Conduction heat transfer takes place when hot, rapidly vibrating atoms and molecules interact with adjacent molecules, thereby transferring some of their energy to them. Conduction is the primary mode of heat transfer in solids and the interface between any two media. To take advantage of conduction heat transfer, heat sinks are made of highly conductive materials, generally metals.

Convective heat transfer is the transfer of heat through motion of the fluid. Convective heat transfer can be further divided into two categories based on how the motion of the fluid takes place. As imagined, convective heat transfer is the primary source of heat transfer for most fluids.

1. Free or natural convection: The motion of the fluid is dictated by the density difference brought about by changes in temperature. The difference in density gives rise to buoyant forces that cause motion of the fluid. Since the heat of the system is used to drive the fluid in this process, this process gets the name natural convection.

2. Forced convection: when external forces, e.g. using fans, mixing the fluid, etc., bring about the motion of fluid over the heat sink.

Heat transfer through radiation occurs due to escape of heat to the surroundings in the form of electromagnetic waves. Since the propagation of heat is through electromagnetic waves, radiation heat transfer takes place even in the absence of a transfer medium. Heat transfer through radiation becomes significant when the difference in temperatures between the system and the surrounding are large. At low temperatures, like the working conditions in this study, heat transfer due to radiation can be ignored.

1.1 Convective heat transfer

Convective heat transfer, often the dominant form of heat transfer in gases and liquids, is the transfer of heat the motion of fluids. Although, often referred to as a distinct form of heat transfer, convection involves the processes of conduction and advection (heat transfer by bulk fluid flow).
Convective cooling, often called ‘Newton’s Law of Cooling’, states that the rate of heat loss from a body is proportional to the difference in temperatures between the body and its surroundings. Convective cooling is defined by the following equation.

\[
\frac{dQ}{dt} = hA\Delta T(t)
\]

Where, \(Q\) is the thermal energy in joules, \(h\) is the heat transfer coefficient (W/m\(^2\)K), \(A\) is the heat transfer area in m\(^2\), and \(\Delta T\) is the temperature difference between the body and its surroundings.

1.2 Heat sinks

All systems that produce work generate heat. This heat is dissipated into the surroundings though a combination of heat transfer methods discussed earlier. This dissipation of heat to the environment keeps the systems at working temperatures. Some systems tend to generate huge quantities of heat very quickly. To keep these systems under working conditions, engineers have to design efficient ways of removing the heat from the systems. A heat sink is a device that enables efficient transfer of heat from the system to the surrounding. Heat sinks are attached to the systems from which heat needs to be dissipated. They are made of materials with high thermal conductivity. Heat sinks also increase the surface area presented to the surrounding fluid, thereby aiding in the heat transfer. They achieve this by introduction of fins, enabling multiple fluid channels and by increasing the number of transfer units. The performance of a system can be limited by its ability to dissipate its generated heat, which is why thermal management of systems becomes an imperative part of design considerations for a system.

1.2.1 Design considerations for heat sinks

**Material**

Copper has excellent heat transfer characteristics which makes it one of the best materials to be used for heat sinks. Copper is also resistant to corrosion and bio fouling. Copper finds its application in industrial thermal facilities, solar power systems, power plants, HVAC systems, etc. However, most of the heat sinks used commercially for electronic chip cooling are made of aluminum alloys.

Aluminum alloys have heat transfer coefficients of around 200 W/m-K which is about half that of copper. Aluminum alloys are preferred over copper for commercial applications as they are cheaper than copper.
**Thermal Resistance**

Thermal is a measure of the temperature difference by which a material resists heat flow. Thermal conduction is a material property. Thermal resistance is the temperature difference across a system when heat flows through it. Thermal resistance finds its analogy with electrical resistance in an electrical circuit. Thermal resistance gains its importance in systems and not individual components. Resistances occur at the interface between two connected parts. When individual parts are connected, micro gaps are formed at the interface due to grooves and crests that are inherently present on the surface. The gaps thus formed get filled with air, which has a low thermal conductivity. This means that the energy has to get transferred through a medium of low conductivity. This reduces the overall heat transfer coefficient of the system. Although thermal resistance between surfaces cannot be totally overcome, it can be mitigated. There are various ways of managing these resistances.

1. Use of thermal tape. Twin sided tapes made of thermally conductive materials are one of the widely used, cheapest form of attachment that is used. The tape, being compressive, reduces the number of air pockets when pressure is applied.

2. Epoxy. Epoxy paste can be applied between the two contacting surfaces to reduce the thermal resistance. It is not as common as thermal tape as it is more expensive than the tape.

3. Pressure. Contacting surfaces can be held tightly in place by use of pressure. Pressure is exerted on the surfaces by use of clips, push pins with compression springs, etc.

In practical applications a combination of tape/paste along with pressure is used to achieve optimum performance of thermal systems.

*Figure 1 Example of thermal resistances generally encountered in commercial chip cooling applications.*
**Fin arrangement**

To increase the heat transfer area of a heat sink, fins are placed over the base of a heat sink. Fins can be square, rectangular, elliptical, cylindrical, etc. The idea here is to pack as many fins over the base as possible as more fins provide more heat transfer area. However, higher fin density causes a hindrance to the flow of fluid over the fins. If the flow of fluid is natural (without use of fans, pumps, etc.), higher fin density may lead to a reduced heat transfer. For forced fluid flows, higher fin densities results in high power consumption, thereby reducing the efficiency of the system.

Engineers hence have to find an appropriate fin density and fin arrangement that works optimally for a given system. This gives rise to a myriad of possible fin arrangements. Figure (2) gives a few examples of the types of fins and the fin arrangement observed in commercial applications.
1.3 Electronics cooling

‘Moore’s Law’ is the observation that the transistors in electronic chips double approximately every two years. This law has been held true partly because it is used by the semiconductor industry to guide its R&D.

As the number of transistors in the chip increase, so does the heat flux generated by the chip. The physics of semiconductor devices are strongly affected by their temperatures. An increase in junction temperature increases the reverse saturation current. This reduces electrical isolation provided by reverse biased current in integrated circuits. Corrosion and interfacial diffusion phenomena are also increased at higher temperatures. Higher density devices also reduce inter and intra chip interconnect delays [1]. It was also shown that the time delay in electronic chips is inversely proportional to the maximum thermal energy that can be removed from a unit area in unit time. These are only a few problems that occur due to high temperatures. Researchers have been studying various methods of effective heat dissipation from electronic chips.

Thermal management of most of the commercially available chips is achieved by use of a simple finned heat sink with airflow forced by a fan acting as the cooling medium. This simple form of heat exchanger has worked perfectly up until the past few years. However, high performance chips made now-a-days need to be cooled by a liquid medium. Water-cooling has become a norm for almost all high performance gaming systems. If the trend for smaller, denser, high performance chips continues, we may see air cooled systems becoming obsolete in the coming years.
An additional hurdle in designing liquid cooled systems for electronic chips comes in the form of packaging the heat sink with the chip. Unlike air cooled systems, liquid cooled systems need to be sealed, thereby necessitating the use of flow channels. This not only makes the system complicated to manufacture, but also increases the pumping power required to pump the fluid, thereby reducing the efficiency of the system. Currently, electronic chips must sustain low surface temperatures for them to perform efficiently and to avoid damage due to overheating. The advancement of these chips is limited to the available means of removal of heat that has been generated. The need for smaller, highly efficient miniature heat sinks has never been greater.
2. Literature Review

Over the past decades, extensive research has been conducted towards optimization of heat sinks. Single phase flow through mini and micro channels has been studied, in order to gain insights into the different phenomenon governing heat transfer and pressure drop, with a final goal of designing efficient mini and micro heat sinks for applications such as electronic chip cooling, cooling of turbine blades, and many more. Numerous enhancement techniques for have been proposed in literature to enhance the performance of miniature heat sinks. An effective method of enhancing the performance of heat sinks is to engineer heat transfer surfaces that promote maximum heat transfer with the least pressure drop. Following is a concise list of enhancement techniques available in literature.

2.1 Microchannels

Microchannels are fluid flow channels with hydraulic diameters less than 1mm. Microchannels are engineered on to the required surfaces either by precision machining or by using micro fabrication techniques.

In 1981, Tuckerman and Pease [3] showed that microchannel arrays could be engraved on the surface of a silicon chip to increase the heat transfer from the chips. These chips were cooled using water as the cooling medium. The microchannel arrays had a basic configuration of 50 µm wide and 350 µm long. Each microchannel in the array was separated by a distance of 100 µm. The array spanned over an area of 1 × 1 cm².

In 1984, Mahalingam [1] discussed the importance of thermal management in packaging of semiconductor devices. They used a heat sink made of silicon substrate. Microchannels measuring 200µm wide and 1700 µm deep were engraved on the substrate. Experiments conducted using water yielded thermal resistances of about 0.03°C/W and 0.02°C/W for flow rates of 12 and 63 cm³/s respectively.

Although this work showed that surface geometries at the micron scale would make for excellent heat sink enhancement, investigations on them were abated due to the limitations in fabricating them. However, with the recent advancements in microfabrication processes, microchannels and alternate surface geometries can be fabricated with relative ease.

In 2006, Colgan et al [2] conducted a study on 19 silicon microchannel coolers to determine the average heat transfer coefficient. The 19 coolers provided a wide range of microchannel designs. The microchannel designs were varied in their pitch (50-100 µm), channel width (20–60 µm) and fin arrangement (continuous and staggered). At optimum performance, the authors observed a unit thermal
resistance of 16.2°C.mm²/W between the chip surface and cooling water when the heat exchanger was
attached to the chip using Ag epoxy. The authors believe that coolers of this type, if bonded with chips
thereby forming a single module, should be able to cool chips with average power densities of 500 W/cm².
Using the optimum design parameters from [2], Colgan et al [3] demonstrated that chips with power
densities of greater than 500 W/cm² could be cooled by using water. To achieve this degree of cooling the
authors reduced the pitch of the fins, used thinner chips and used bonding materials with better
conductivity than the ones used in [2]. They also demonstrated that by using fluorinated fluid with an inlet
temperature of -30°C, a chip with a power density of 270 W/cm² could be maintained at 35°C. However,
the increased surface area increases the pressure drop in the channel.
This increases the pumping power required to pump the fluid. It was also shown that correlations
developed for macro scale transport did not scale well when applied to micro scale geometries.

2.2 Micro pin fins
Flow over micro pin fin arrays is gaining recognition as one of the alternate surface geometry for
microchannel flow.
Siu-ho et al. [4] investigated heat transfer and pressure drop characteristics of a micro pin fin heat
exchanger at two heat fluxes of 50 and 100 W/cm². The pin fins measured 200 × 200 µm² in cross section
and 670 µm long. Using deionized water as the coolant. The inlet Reynolds number of the water ranged
from 93 to 634 for q" = 50 W/cm² and 127 to 634 for q" = 100 W/cm².
Kosar et al. [5] showed experimentally that the macro scale correlations did not hold true when predicting
pressure drops obtained at micro scales. In the test they used circular and diamond shaped pin fin bundles,
100 µm long with hydraulic diameters of 50 and 100µm placed in inline and staggered arrangement. The
authors noted that pressure drop in micro fin or microchannel channels are mainly caused due to two
major factors. The friction at the fins or channel walls and the endwall (top and bottom surfaces) effects.
The authors observed that the endwall effects are diminished at higher Reynolds numbers (>100). Hence
macro scale correlations, which mainly take into consideration the effects at the fin walls, could predict
the results observed in the experiment. However at lower Reynolds numbers, the endwall effects on
friction factor are significant. Hence the conventional correlations under-predicted the experimental data.
Based on their results, they proposed a modified friction factor correlation. The correlation includes the
effects caused at the end walls and effects of fin density given by the parameters ‘π₁’ and ‘π₂’.
While studying single phase flow of R-123 across a bank of micro pin fins, Kosar and Peles [6] discovered
that the Nusselt number correlation proposed by Zukauskas [7] did not accurately predict the
experimental results for lower mass fluxes. This was true not just for R-123 data, but also for water data.
Using the least squares method, the authors developed a new correlation for Nusselt number to describe the experimental data. The correlation is described in section 4.2 of this literature.

Using a commercial computational tool, John et al. [8] simulated fluid flow over circular and square pin fins. The authors, in the study, varied various parameters of the pin fins such as the axial and transverse pitch distances, aspect ratio and hydraulic diameters of the pins. To determine the overall performance of the heat sink, pressure drop and heat transfer characteristics of the heat sink were evaluated over a range of Reynolds numbers between 50 and 500. The study showed that at lower Reynolds numbers (below 300), circular fins performed better than square fins and vice versa for higher Reynolds numbers. For both the types of pin fins studied, it was observed that increasing the transverse pitch increased the performance of the chip, mainly due to a decrease in pressure drop. A computational study conducted by Koz et al [5] shows that the effect of end walls on the friction factor decreases with an increase in Reynolds number. At higher Reynolds numbers, the ratio between the viscous forces to the total forces decreases with an increase in Reynolds number.

Metzger [9], in 1982 investigated square staggered micro pin fins with the pin length to diameter ratio of 1 and a transverse and longitudinal pitch to diameter ratios of 1.5 and 2.5 respectively. Apart from overall heat transfer and flow friction behavior, their study also focused on stream wise heat transfer variation. For all the Reynolds numbers tested, the authors observed a general trend in the stream wise heat transfer variation. The Nusselt number showed an increasing trend for the first 3-5 rows of the fins, after which a gradual decline was observed. When the stream wise pin spacing was larger, the system would achieve its peak heat transfer earlier (third row) than when the pin spacing was smaller, however the overall heat transfer was higher for the system with a closer pin spacing. A significant take away from this study is that the pin fin arrays studied here (H/D = 1) performed significantly better than long fins (larger H/D ratio), especially at lower Reynolds numbers.
In a separate study, Metzger et al. [10] compared the performance of two different shapes of pins. The two types of fins studied were circular fins, similar to the ones used in their previous study [9], and oblong fins. The fins were arranged in a staggered arrangement. However, within the staggered arrangement, the fin orientation to the flow was changed as shown in figure 4. Three orientation angles corresponding to 30, 45 and 90 degrees were studied. The oblong fins showed an increase in heat transfer, of close to 20%, when compared to circular fins in a similar arrangement. But, this enhancement was offset by an increase in pressure loss of 100%. The orientation of the staggered fins with respect to flow direction does cause significant changes in the thermal performance of the system for both the fin shapes studied. An arrangement close to the inline arrangement increased the heat transfer by 9% and reduced the pressure drop by 18%. The authors also believe, with some reservation, that the heat transfer coefficient observed at the pin fins is almost twice that observed at the end walls. However, since the observation was indirect, the authors suggested restraint while using this conclusion. Since the Reynolds numbers observed in micro heat sinks are generally small, this study shows that the use of pin fin with a small H/D ratio is favorable for enhancing heat transfer.

It was also shown, by Selvarasu et al. [11] and Shafeie et al. [12], when measuring heat capacity and pressure drop the fins showed best performance in the laminar regime. However, flow through micro pin fin arrays severely increases the pressure drop of the flow.

Figure 4: Circular and Oblong fin geometries used by Metzger.
2.3 Short micro pin fins and microchannels

Fin arrays with a clearance as a clearance over the fins is frequently encountered during chip packaging design. A clearance over the fins also provides for complex flow regimes as opposed to a no clearance case. These complex flow patterns, in theory should enhance the heat transfer of the fins.

Micro pin fins arrays can become excellent heat sinks for electronic cooling applications if the pressure drop in the flow can be mitigated. Short micro pin fins show the potential to provide an answer for this problem. Pin fins are said to be short if the height of the fins is smaller than the channel height, thereby leaving a free flow area on top of the fins for fluid flow. It was shown that, shortening the fins reduced the thermal performance of the fins while also decreasing the pressure drop.

Sparrow and Kadle [13] investigated the heat transfer characteristics of longitudinal fin arrays (microchannels) with a clearance over the fins. The study focused on the effect of fin size and the clearance over the fins on the heat transfer from the fins. The clearance over the fins varied from 0 to 0.336 times the height of the film and two fin thicknesses which were 3.75 and 7 times the fin height. Air was used as the cooling fluid. Presence of a clearance over the fins reduced the heat transfer from the fins. For example, clearance heights equal to 10, 20 and 30 percent of fin height resulted in heat transfer coefficients were 85, 74 and 64 percent of the no clearance case.

Numerical modelling of a microchannel heat exchanger by Harpole and Eninger [14] observed that optimum performance was reached when the fin height was about half the total channel height. The heat

Figure 5 Different micro pin fin shapes studied by Tullius et al.
transfer coefficient achieved in this study was close to 100 W/cm²K with pressure drops of close to 1 to 2 bars.

Min et al [15], in 2004, conducted a computational study to determine the thermal performance of microchannels with a clearance over the top of the channels, similar to short micro pin fins. The clearance over the channel tops were varied from 0 to 1 times the channel height. Their study showed that the optimum height of the fins should be about 60% of the channel height for optimal performance. They also observed a modest increase in the performance, of about 3%, when compared to microchannels without tip clearance.

Knight et al [16] conducted optimization studies on pin fins and developed equations to optimize finned heat sinks. The equations presented in this paper enable the determination of heat sink dimensions that yield the lowest thermal resistances. The study also found that laminar flows yielded lower thermal resistances when the pressure drops were lower and turbulent flows performed better when the pressure drops were higher.

Tullius et al [17] conducted an extensive computational study on the heat transfer and pressure drop in micro pin fins. They studied 6 different shapes of fins, with varying fin heights ranging from 0.75 times the channel height down to an unfinned surface.

Figure 6 Nusselt number and pressure drop values observed for square staggered micro pin fins for fin height to channel height ratios from 0 to 0.75

An increase in Nusselt number and pressure drop were observed with an increase in fin height. For the entire range of fin geometries studied, the performance of the all the fins were the same at low flow rates. However, at higher flow rates, the square fins showed not only the highest heat transfer, but also highest pressure drop.

The focus of this study was to see if the existing correlations could accurately predict the results from the study. However, the data obtained by the authors was not accurately predicted by the correlations for Nusselt number and friction factor present at the time. The Mean Effective Error for the Nusselt number
correlations ranged from 2.1 to 82.62% and 5.98 to 279.5% and 5.98 to 279.5 for friction factor. The authors hence proposed a new Nusselt number and friction factor correlation for a range the fin shapes based on their study.
3. Objectives of the present study

It is evident that correlations for heat transfer and pressure drop for flow over micro pin fins are available in literature and have been validated to a certain extent using computational techniques. An experimental investigation, however, is warranted to determine the usage of these correlations for practical purposes. The purpose of this study can hence be categorized into two main objectives:

1. To assess the enhancement in heat transfer and increase in friction factor brought about by the introduction of fins and the effect of tip clearance on the thermal enhancement of a finned surface over a plain surface.

2. To determine the fit of correlation for Nusselt number and friction factor proposed by Tullius et al. [17] with experimental values.

The current study focuses on rectangular staggered short micro pin fins. It studies the effect of fin height and fin clearance on heat transfer and pressure drop characteristics of the heat sink. Hence, apart from the fin height and the fin clearance, all other parameters of the fins were kept the same. The pressure drops and heat transfer studies were conducted over a range of flow rates, thereby enabling the study at a range of Reynolds numbers.

The experimental values obtained from this study were compared with theoretical values obtained by using the Nusselt number and Pressure drop correlations developed by Tullius et al. [17] to study how well the correlations fit to the experimental data.
4. Experimental details

The study focusses on heat transfer and pressure drop characteristics of finned surfaces with a clearance over the fins. To facilitate this study, we need a test setup which will enable us to conduct the study. The finned surface was created using a copper chip. Fins were machined on the copper chip. To complete the flow chamber, a top cover was fabricated. The space between the fins and the top cover makes up for the flow chamber. A gasket was placed between the chip and the top cover to provide sealing for the chamber. The thickness of the gasket also provides for the free flow area or the clearance over the fin tops. The components that went into fabricating the test section are detailed in the following sections.

4.1 Test Surface (copper chip)

Staggered micro pin fin arrays with a plan area of 10 mm x 10 mm were machined at the center of 20 X 20 mm square copper chips using a CNC machine. The test section constitutes the 10 mm x 10 mm finned area. The heat to the fins is supplied at the base of the copper chip. Grooves at the base, surrounding the heated surface ensure one-dimensional heat conduction into the base of the fins and the channel walls.

The copper chip is seated in a slot created on a ceramic base. Three types of chips were fabricated as shown in table 1.

The base area in this study is a 10mm x 10mm test section area, which makes the area available for heat transfer from a plain surface to be equal to 0.0001m². With the introduction of fins however, the heat transfer area increases leading to an enhancement in area. The area enhancement brought about by the different fin heights corresponding to chips c1, c2 and c3 are presented in the following table.
Table 1 Fin dimensions and heat transfer area enhancement

<table>
<thead>
<tr>
<th>Chip</th>
<th>Fin height (Hf)</th>
<th>Area enhancement</th>
<th>Fin constants</th>
</tr>
</thead>
<tbody>
<tr>
<td>c1</td>
<td>500 µm</td>
<td>1.78</td>
<td>l_f = 1000 µm</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>S_t = 800 µm</td>
</tr>
<tr>
<td>c2</td>
<td>300 µm</td>
<td>1.47</td>
<td>w_f = 300 µm</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>S_t = 1000 µm</td>
</tr>
<tr>
<td>c3</td>
<td>200 µm</td>
<td>1.3</td>
<td>N = 60</td>
</tr>
</tbody>
</table>

4.2 Gasket

Three gaskets were cut to be used between the chip and the top cover. Each gasket had two functions. The gasket served as a sealant for the test chamber. The gasket also provided for the free flow area above the fin tops for the fluid. The height of the free flow section was the same as the gasket thickness. Table (2) gives the dimensions of the gaskets used in the study.

Table 2 Gasket dimensions

<table>
<thead>
<tr>
<th>Gasket</th>
<th>Thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>g1</td>
<td>900 µm</td>
</tr>
<tr>
<td>g2</td>
<td>420 µm</td>
</tr>
<tr>
<td>g3</td>
<td>265 µm</td>
</tr>
</tbody>
</table>

This chip-gasket combination resulted in varying channel heights (h_c) and varying channel to fin height ratios. The chip-gasket combinations used in this study are given in table (3).

Table 3 Test Matrix

<table>
<thead>
<tr>
<th>Chip – Gasket</th>
<th>Total Channel Height</th>
<th>Chip – Gasket</th>
<th>Total Channel Height</th>
<th>Chip – Gasket</th>
<th>Total Channel Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>c1 – g1</td>
<td>1400 µm</td>
<td>c2 – g1</td>
<td>1200 µm</td>
<td>c3 – g1</td>
<td>1100 µm</td>
</tr>
<tr>
<td>c1 – g2</td>
<td>920 µm</td>
<td>c2 – g2</td>
<td>720 µm</td>
<td>c3 – g2</td>
<td>620 µm</td>
</tr>
<tr>
<td>c1 – g3</td>
<td>765 µm</td>
<td>c2 – g3</td>
<td>565 µm</td>
<td>c3 – g3</td>
<td>465 µm</td>
</tr>
</tbody>
</table>
4.3 Ceramic Base

The copper chip is seated on a ceramic chip. A 20mm × 20mm slot was machined on an 80mm × 80mm ceramic piece. The depth of the slot was equal to the thickness of the chip, which was 3mm. A heater block was used to provide the required heat flux into the chip. A 10mm × 10mm hole was machined at the center of the base to facilitate introduction of the heater block onto the base of the chip.

4.4 Gasket compression experiment

Three different gaskets, with pre-compression thicknesses of 275, 500 and 975 µm were cut. The primary role of the gasket is to provide sealing to the flow chamber. The presence of this gasket provides for a free flow area for fluid flow equal to the thickness of the gasket seen in figure (9). However, once the set-up is assembled, the gaskets compress, thereby reducing the effective gasket thickness. A separate experiment was conducted to measure the actual post-compression thickness value of each of the gaskets considered. Details of the separate gasket experiment are given in the following sections. The gasket measures 30mm × 30mm. A 10mm × 10mm square hole was cut inside the gasket. This hole coincides with the 10mm × 10mm finned surface on the chip. Finally, a top cover sits on the gasket, making the whole assembly leak proof.

4.4.1 Gasket compression after assembly

The ceramic base, chip, gasket and the top cover assembly are held together with the help of bolts. This pressure exerted by the bolts causes the gasket to compress. As mentioned before, the thickness of the gasket equals the free flow area available for fluid flow. However, the compression of the gasket could not be measured directly. This warranted the need for a separate experiment to determine the compression and the final thickness of each gasket after assembly.
4.4.2 The gasket experiment

To get the value of gasket compression after assembly, a separate experiment was conducted. The aim was to place an object in the chamber, which would compress with the assembly. Another constraint present in the experiment that the object should not stick to the top of the chamber (top cover), when the test setup was disassembled. If the object stuck to the top of the chamber, it would not give a proper measurement of the total channel thickness. Hence, floral foam was chosen as the preferred object for this experiment.

A small piece of floral foam was placed in the flow chamber, with an original height greater than the total channel height. The setup was then assembled and kept aside for about 20-30 mins. The setup was then carefully disassembled making sure that the floral foam inside was not disturbed. In floral foam was disturbed, the experiment was repeated. With the undisturbed, compressed floral foam on the chip, the thickness of the compressed foam was measured using a Laser confocal microscope.

The measured floral foam thickness values were used as final gasket thicknesses given in table (2).
4.5 Flow channel and top cover

The flow channel was formed between the copper chip and a top cover fabricated out of polysulfone. A gasket was placed between the top cover and the copper chip. The gasket served two purposes. First, it provided sealing between the copper chip and the top cover. It also facilitated for a gap between the fin tops and the channel roof providing for a free flow area above the fin tops, crucial for this study. The free flow height above the fins was dictated by the gasket thickness. Thus, the total height of the channel is the addition of the fin height and the gasket thickness. 4.5.6 Heater block

The heater block consists of a solid copper block as shown in figure (10). The base of the block has 4 holes drilled into it to accommodate cartridge heaters. The top part of the heater has three 0.003” holes spaced 5 mm apart to accommodate thermocouples. The K-type Omega thermocouples themselves have a diameter of 0.0027”. Four 200 W cartridge heaters were inserted into the base of the heater to heat it. The cartridge heaters are held in place using a thermal paste.
4.6 Housing

A square housing on top of the top cover incorporated the inlet and outlet channels. Two slits at the base of the housing, spaced 10 mm apart, made for the actual inlet and outlet of the flow chamber. The entire assembly is held in place with the help of a holding plate, and bolted together using bolts. The holding plates make sure that the heater and the setup are in perfect contact with each other. The holding plates ensure this by fixing the entire setup horizontally to the base plate. The test section after assembly of the chamber components, holding plate and the heater block looks like figure (12).

The base plate for the chamber assembly and the plate for heater seating are mounted on a stand that sits on a vibration isolation table. It is necessary to ensure that the entire assembly is held perfectly upright (without any tilts) for efficient conduction of heat into the system.

Figure 12 Test setup

Figure 11 Flow loop
4.7 Flow loop

The flow loop starts with a vessel or a reservoir for the distilled water. The water is then pumped into the system using a Micropump® pump. A flow meter before the test section is used to read the flow rate of the water. The water then flows through the test section before finally going back into the reservoir. A thermocouple, at the inlet and outlet are used to measure the inlet and the outlet temperatures. A pressure sensor, connected before and after the test section, measures the pressure difference in the test section.

4.8 Measurements

A K type thermocouple inserted inside the chip is used to measure the temperature of the chip. The thermocouple is located at the center of the chip and is at a distance of 5mm from the base or the top. The actual temperature of the chip surface is calculated by knowing the heat input into the base of the chip and the temperature of the chip from the thermocouple. Three K-type thermocouples are inserted into the thermocouple slots on the heater at equal intervals. These temperature measurements are used to calculate the heat flux at the top of the heater surface using Fourier’s 1D law of heat conduction. The details of the calculations are given in the following section. One thermocouple each is used at the inlet and the outlet of the test assembly to measure the inlet and the outlet temperature of the water. Two different pressure transducers were used in the study. The two pressure transducers varied only in their sensitivity. The sensitivity of PS 1 (pressure transducer 1) was 0-1 PSID, whereas PS 2 (pressure transducer 2) could be used over the range of 0-1 atm. The position of both the transducers was the same. They were placed between the inlet and the outlet of the test assembly.

Finally, an Omega© flow meter was used to measure the flow rate of the water in the test loop.
5. Data reduction

This section provides information of the calculations that were involved in calculating the parameters like the heat flux into the chip \(q''\), the average heat transfer coefficient \(h_{avg}\), the average Nusselt number \(Nu_{avg}\), the pressure drop \(\Delta p\) and the experimental friction factor \(f\).

The parameters measured directly were,

1. \(T_1, T_2\) and \(T_3\) which give the temperatures of the heater at an interval of 5mm as shown in section 3.6.
2. The inlet and the exit temperature of water \((T_{in}, T_{out})\)
3. The temperature of the base at the center of the channel \((T_s)\).
4. The pressure drop between the inlet and exit of the channel.

The data was obtained in this study with the help of LabView© VI. Data for temperatures and pressure drop was collected at steady state. The data was obtained over a period of 20 seconds with a sampling rate of 5 samples/second, resulting in 100 data points for each steady state. These values are then averaged to get the temperature and pressure value at each steady state.

5.1 Experimental values of Heat Transfer Coefficient and Nusselt number

As mentioned in section 3.6, the heat into the chip is provided by a copper heater, which in turn is heated using cartridge heaters.

The heat flux \(q''\) at the surface of the heater is calculated using Fourier’s law for one-dimensional heat conduction,

\[
q'' = -k_{Cu} \frac{dT}{dx} \tag{1}
\]

Three thermocouples \(T_1, T_2, T_3\) were inserted into the heater at a fixed distance \(\Delta H = 5\text{mm}\).

Using a three point backward difference Taylor series approximation, the temperature gradient \(\frac{dT}{dx}\) was calculated.

Therefore, \(\frac{dT}{dx} = \frac{3T_1 - 4T_2 + T_3}{2\Delta h}\) \(\tag{2}\)

The chip surface temperature was obtained from the measured chip temperature \(T_4\), the heat flux, and the distance 1.5mm between the chip thermocouple and the chip surface.

To evaluate the thermal characteristics of the heat sink, the average heat transfer coefficient \(h_{avg}\) needs to be calculated.

\[
q = h_{avg}(\eta_f N_f A_{ts} + WL - N_f A_f) \left[T_s - \frac{(T_{in}-T_{out})}{2}\right] \tag{3}
\]
Where, \(\eta_f = \frac{\tanh(m_f H)}{m_f H}\), and \(m_f = 2 \sqrt{\frac{h_{avg}}{D_f R_e}}\)

Also, \(N_f\) is the number of fins.

\(A_t\) is the total surface area of the fin. That is the area exposed to the fluid and \(A_f\) is the area of the fin top.

In the current study, \(A_t\) is given by

\[A_t = 2(Wf + WfH_f + LfH_f)\]  \hspace{1cm} (4)

\(A_f\) is given by \(A_f = Wf*Lf\)  \hspace{1cm} (5)

Here, \(W_f, H_f,\) and \(L_f\) are the width, height and the length of the fin respectively.

\(W\) and \(L\) are the width and the length of the channel.

\(D_f\) is the hydraulic diameter at the fin given by,

\[D_f = \frac{2WfH_f}{W_f+H_f}\]

The Nusselt number is then calculated using the \(h_{avg}\) calculated earlier.

\[Nu = \frac{h_{avg}D_c}{k_f}\]  \hspace{1cm} (6)

\(D_c\) is the hydraulic diameter of the channel given by

\[D_c = \frac{2(W+H)}{W+H}\]  \hspace{1cm} (7)

\(k_f\) in the above equation is the thermal conductivity of the fluid being used; water in this study.

5.2 Experimental values of Pressure drop and Friction factor

The inlet and the exit pressure ports are situated 5 mm away from the inlet and the exit. Hence the pressure drop recorded by the transducer, records the pressure drop at the pipes before and after the inlet and exit of the chamber. Also from section 3.5 and figure (11), we can see that the inlet channel consists of 45-degree bends and slanting inlet and exit sections. Hence, to calculate the actual pressure at the chamber, the following calculations of miscellaneous pressure drop were made.

Pressure loss in straight pipes

The pressure drop in straight pipes (\(\Delta p_p\)) and the slanting inlet and exit sections in the top cover is given by the following equation.

\[\Delta p_p = \frac{2f_p \rho u_l^2}{D_p}\]  \hspace{1cm} (8)

Here the friction factor at the pipes (\(f_p\)) is given by the following equation

\[f_p = \frac{16}{R_e}\]

Where \(R_e\) is the Reynolds number of the fluid in the pipe.
The pressure drop at the bends

\[ \Delta p_b = 0.5 f_p \rho u^2 \frac{\pi R_b}{D} \frac{\theta}{180} + 0.5 k_b u^2 \rho \]  \hspace{1cm} (9)

Here, \( R_b \) is the radius of the bend
\( \theta \) is the angle of the bend in degrees
\( k_b \) is the bend loss coefficient obtained from figure (13) proposed by Babcock and Wilcox Co. (1978)

Experimental pressure drop

The experimental pressure drop (\( \Delta P \)), after accounting for pressure loss in the pipes and bends, is used to calculate the friction factor at the test chamber.

\[ f = \frac{\Delta P D_c}{2 \rho u^2 m L} \]  \hspace{1cm} (10)

5.3 Comparison with correlations

Friction factor

<table>
<thead>
<tr>
<th>Author</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tullius et al. [17]</td>
<td>( f = 5.28 \left( \frac{S_t}{D_f} \right)^{0.2} \left( \frac{S_x}{D_f} \right)^{0.2} \left( \frac{H_f}{D_f} \right)^{0.18} \left( 1 + \frac{g}{D_f} \right)^{0.2} \left( Re_f \right)^{-0.435} )</td>
</tr>
</tbody>
</table>
**Nusselt Number**

*Table 5 List of Nusselt Number correlations studied*

<table>
<thead>
<tr>
<th>Author</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tullius et al. [17]</td>
<td>[ Nu_f = 0.0937 \left( \frac{S_t}{D_f} \right)^{0.2} \left( \frac{S_t}{H_f} \right)^{0.2} \left( 1 + \frac{g}{D_f} \right)^{0.4} \left( Re_f \right)^{-0.6} P_r^{0.36} \left( \frac{P_r}{P_r_s} \right)^{0.25} ]</td>
</tr>
</tbody>
</table>

**5.4 Uncertainties**

To assess the accuracy of the measurements, an uncertainty analysis was performed. Here, the uncertainties in the calculated parameters like \( Nu, q^\prime\prime, f \), etc. are generally denoted by \( \delta y \), if \( y \) is a function of the variables \( x_1, x_2, x_3, \ldots x_n \).

\[
y = f \left( x_1, x_2, x_3, \ldots x_n \right),
\]

\[
\delta y = \left[ \left( \frac{\partial y}{\partial x_1} \delta x_1 \right)^2 + \left( \frac{\partial y}{\partial x_2} \delta x_2 \right)^2 + \ldots + \left( \frac{\partial y}{\partial x_n} \delta x_n \right)^2 \right]^{1/2}
\]

(11)

Where, \( \delta x_1, \delta x_2, \ldots, \delta x_n \) are the uncertainties in the independent variables.

The uncertainties in the experimental parameters are listed in the following table.

*Table 6 Experimental Uncertainties*

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Nu</th>
<th>( T_s )</th>
<th>Pr. drop</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uncertainties</td>
<td>5-7%</td>
<td>0.05°C</td>
<td>0.5 – 1%</td>
</tr>
</tbody>
</table>

**5.5 Experimental procedure**

Distilled water is used as the working fluid in this study. Water is circulated through the test loop at the desired flow rate using the Micropump. The flow meter placed before the inlet indicates the flow rate. The temperature of the water to the inlet of the test section is not controlled and is kept at room temperature. The power to the heater is then turned on to obtain a low initial heat flux of about 6-8 W/cm². Measurements are recorded using the Lab View VI at steady state conditions. Once the required data is saved, the heat input is increased by changing the voltage across the heaters by about 5V. This procedure is repeated until the surface temperature of the copper chip reaches more than 80°C. At this point, the heaters are turned off and the data is further analyzed.
6. Set-up validation
Before beginning with the testing various finned surfaces on the test set-up, the test set-up needed to be validated. To validate the results obtained from the experiments, tests were conducted over a plain surface with a constant gasket thickness $g_3$ (265 µm). A handbook of heat transfer and pressure drop in mini and microchannels gives theoretical equations for obtaining pressure drop values in microchannels. The theoretical values obtained from the equations presented in the book were then compared with the experimental pressure drop data obtained in the study. The experimental data and theoretical values obtained from [21] are presented in the following figure. The Mean error in the pressure drop of experimental and predicted values was calculated to be 15%.

The theoretical values of pressure drop and Nusselt number were calculated using the equations presented in the following section.

From figure (14 a) we see that the predicted theoretical values of heat transfer coefficient do not agree with the experimental values within the uncertainty limits. The reasoning for this discrepancy is discussed in the section after the next.

**Nusselt number for a theoretical plain surface**
For rectangular channels with one-sided heating configuration is given by the following equation, presented in the book Heat Transfer and Fluid Flow in Minichannels and Microchannels.

$$Nu = a + bx^+ + c(\ln(x^+))^2 + d \ln(x^+) + e(x^+)^{-1.5}$$  \hspace{1cm} (12)

Where $\alpha_c$ is the channel aspect ratio given by $\alpha_c = A/B$
Where A is the unheated side and B is the heated side.

In the current study, \( \alpha_c = h_c/w_c \).

The constants used in equation 12 are given in the following table

<p>| | | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>B</td>
<td>C</td>
<td>d</td>
<td>e</td>
</tr>
<tr>
<td>9.132</td>
<td>-3.7531</td>
<td>0.4822</td>
<td>2.5622</td>
<td>5.16*10^-6</td>
</tr>
</tbody>
</table>

Equation (12) and its constants have been taken from Appendix A at the end of chapter 3 of the book mentioned above.

**Friction factor for a theoretical plain surface**

As flow enters a channel, the velocity profile develops along its length, ultimately reaching the fully developed Hagen-Poiseuille velocity profile. The length of the hydrodynamic developing region \( l_h \) is given by the equation

\[
\frac{L_h}{D_h} = 0.05Re
\]

Shah and London [22] provided the following equation for a rectangular channel with a channel aspect ratio \( \alpha_c \) with the short side A and long side B.

\[
fRe = 24(1 - 1.3553\alpha_c + 1.9467\alpha_c^2 - 1.7012\alpha_c^3 + 0.9564\alpha_c^4 - 0.2537\alpha_c^5) \quad (13)
\]

Where \( \alpha_c \) is the channel aspect ratio given by \( \alpha_c = A/B \)

Where A is the short side and B is the long side.

In the current study, \( \alpha_c = H/W \).

In the current study, the flow regime is developing through the entire length of the channel. To account for the developing flow, an apparent friction factor is presented \( (f_{app}) \). The apparent friction factor accounts for the pressure drop in the developing region.

The difference between the apparent friction factor over a length \( x \) and the fully developed friction factor \( f \) is expressed in terms of an incremental pressure defect \( K(x) \).

\[
K(x) = (f_{app} - f) \frac{4x}{D_e} \quad (14)
\]
For $x > L_h$ the incremental pressure defect $K(x)$ attains a constant value $K(\infty)$, known as Hagenbach’s factor.

Steinke and Kandlikar [21] obtained the following curve-fit equation for Hagenbach’s factor in rectangular channels

$$K(\infty) = 0.6796 + 1.2197\alpha_c + 3.3089\alpha_c^2 - 9.5921\alpha_c^3 + 8.9089\alpha_c^4 - 2.9959\alpha_c^5$$ (15)

Since the flow in the current study is in the developing regime, the friction factor obtained from the experimental measurements is the apparent friction factor. Hence, for use in equation (11), $f_{\text{plain}}$ is the same as $f_{\text{app}}$ (equation (14)) for the theoretical plain surface.

**Discrepancy in heat transfer coefficient**

The correlation for Nusselt number provided, used for theoretical calculation of Nusselt number for a plain channel holds true for a narrow range of parameters. The authors of the book [21] suggest the use of a set of equations/correlations to determine the theoretical values of Nusselt numbers for flow through a plain channel configuration. The Nusselt number correlations are bound tightly to certain heating conditions (constant heat flux vs. constant surface temperature) and the number of heated boundaries (1-wall heating, 2-sided heating, 4-sided heating, etc.) The conditions that closely match the test parameters in the study are that the heat flux into the surface is constant and the surface is heated just at the base making for a one sided heating condition. The constants used in the correlations also vary with the aspect ratio of the channel. The aspect ratios studied in the handbook range from 0.1 to 10.

These very tight restrictions for the use of the correlations could not be achieved in the experiment performed in the study. Although the equation can be used for one sided heating condition, the side walls/gasket also heats up in the experiment, thereby contributing heat input to the fluid, and the one sided heating condition could not be fully upheld. Similarly, the aspect ratio studied herein measures 0.0265, with the heated side being considered as the longer side. This would make the constants used in the correlation not correct. These factors mitigated the validation testing of the setup on the basis of heat transfer coefficient. However, the set-up shows great agreement with the experimental and theoretical values of pressure drop. Hence, it can be concluded that the test set-up could be validated on the basis of the pressure drop values and the set-up works as expected.
7. Results

This study focusses on the heat transfer and pressure drop characteristics of short micro pin fins. Nine chip – gasket combinations were studied for five flow rates of 50, 100, 200, 400 and 500 ml/min. Pressure drop across the test section, the inlet and outlet temperatures, and the temperature of the chip were measured at steady state conditions. The flow domain in this study is in the laminar regime for all the flow rates, gasket configurations and fin heights considered. It is important to note that the flow domain in the experiment is hydro dynamically as well as thermally developing.

Figure 15 Experimental heat transfer coefficient for chips (a) c1, (b) c2 & (c) c3 with varying gasket thicknesses (g1, g2 & g3)
7.1 Heat transfer coefficient

7.1.1 Effect of fin height on heat transfer coefficient

Figure (15) illustrates the averaged heat transfer coefficients (considering total area, i.e. area enhancement by the fins as well as considering fin efficiency) obtained at flow rates ranging from 50 to 500 ml/min, for the entire range of the test matrix discussed in section 3.2, table 3.
To study the effect of fin height on the heat transfer coefficient, the chips/finned surfaces are tested at a constant gasket thickness/free flow area above the fins. For all the fin heights considered, the heat transfer coefficient increases with an increase in Reynolds number.
Increasing fin height increases the surface area available for heat transfer. It is expected that as the fin height increases, so does the heat transfer coefficient. Such a trend was observed for the experimental data. At all the gasket thicknesses, considered, the maximum heat transfer coefficient was observed for chip c1 with the longest fins, measuring 500 μm, whereas the shortest fin c3, measuring 200 μm, consistently showed lower values of heat transfer values. However, a maximum value of heat transfer coefficient was observed for the chip gasket combination c1-g1, with a total channel height of 1400 μm. It is expected that the channel with a lower total channel height (hₐ), corresponding to a lower hydraulic diameter, should in theory present a higher heat transfer coefficient compared to a channel with a larger hydraulic diameter. The reason for this disparity in the experimental values is still unknown and will need to be studied in detail further.

7.1.2 Effect of free flow area on the heat transfer coefficient from a surface

To study the effect of free flow area on the heat transfer coefficient, the chips/finned surfaces are tested at varying gasket thickness/free flow area above the fins.
Figure (16) illustrates the projected heat transfer coefficients (ignoring fin efficiency and considering the projected base area as the heat transfer area) obtained at flow rates ranging from 50 to 500 ml/min, for the entire range of the test matrix discussed in section 3.2, table 3.

Analysis of all the sub figures in figure (16) shows that a clear relationship between the fin top area and the heat transfer coefficient cannot be made. It is expected that as the free flow area over the fins decreases, the heat transfer coefficient should increase owing to a higher $h_f/h_c$ ratio and reduction in hydraulic diameter. Such a relationship is observed only for chip c1 with a fin height of 500 µm. For other fins c2 and c3 the same is not true. For fins with a lower fin height, the heat transfer coefficient increases as the gasket thickness increases from 420 µm to 900 µm. This increase in heat transfer coefficient may possibly be explained by a mixing phenomenon taking place over the fins.

The presence of a free flow area above the fins can, in theory, increases the heat transfer coefficient of the system due to flow mixing, not observed in confined microchannel flows. Such a mixing phenomenon was not directly observed in the present study. However, there is some merit in saying that the theory may be in effect in the present study from the discussion in the section above. A larger free flow area, coupled with disturbance of the boundary layer bought about by the fins, may lead to significant amounts of flow mixing. It is possible to conduct a flow visualization study on similar channel configurations to decisively assert this claim. It is possible that by adopting different flow patterns or structures on the fin and microchannel walls or a combination of both can result in flow mixing in short pin fins. This mixing
phenomenon can be used to further enhance the heat transfer coefficient of short micro pin fins and microchannels alike.

### 7.2 Pressure drop

Reduction in pressure drop is one of the main reasons why micro pin fins are studied over micro channels and fins that extend the full height. The pressure drop in the chamber is caused due to end wall effects and the friction at the fins. Micro pin fins have smaller fins, which reduces the pressure drop at the fin walls. Also compared to microchannels, micro pin fins generally tend to have larger inlet cross sections, thereby resulting in lower Reynolds numbers in the flow, and lower Reynolds numbers lead to a lesser pressure drop. Figures 17 (a)-17 (c) illustrate the pressure drop obtained for the entire range of chip – gasket combinations tested in the study.

*Figure 17 Experimental values of pressure drop vs Reynolds number at varying gasket thicknesses*
The thinnest gasket ‘g3’, indicated by the ‘+’ inside the individual symbols, shows the largest pressure drop for all the flow rates tested. Similar to heat transfer coefficient, the pressure drop increases as the free flow area is reduced and more liquid contacts the fins. The highest pressure drop of about 100 kPa is observed for the chip gasket combination c1-g3 representing the combination of the tallest fins with the least free flow area over the fins.

This combination not only results in a very low inlet cross section (due to a thin gasket) but also causes maximum hindrance to the flow inside the channel owing to the tall fins. This is why the lowest pressure drop of 0.23 kPa was observed for the chip-gasket combination c3-g1 with the largest gasket thickness (free flow area over the fin tops) and the shortest fins.

Figure 18 Experimental pressure drop values for chips (a) c1, (b) c2 and (c) c3 at gasket thickness g1, g2 and g3.
7.2.1 Effect of free flow area on pressure drop

An increase in free flow area over the fin tops increases the channel hydraulic diameter. A larger free flow area also reduces the fin height to channel height ratio. Which means that the pressure drop caused at the fin walls gets reduced considerably. Hence it is expected that pressured drop values should decrease as the free flow area or the gasket thickness (in this study) increases. When the fin height is kept constant, as seen in figures (18 (a) – 18 (c)) we can see that maximum pressure drop values are consistently observed for gasket g3 as expected. Gasket g3 is the thinnest which makes for the lowest free flow area and consistently the smallest total channel heights, hence the high pressure drop.

However gasket g1 does not behave as expected. Gasket g1 is the thickest gasket, providing the largest free flow area. Hence in theory, the pressure drop expected for gasket g1 should be lower than the pressure drop for gasket g2 which has a thickness of 420 µm compared to 900 µm for g1. However, analysis of all the figures in figure 18 shows exactly the opposite trend. In all the figures we can see that pressure drop observed for gasket g1 is consistently higher than the values observed for gasket g2. A systematic and controlled study will need to be conducted to assert this claim.

7.3 Comparison with correlation

The experimental results obtained in the present study was compared with the values predicted by the Nusselt number and Pressure drop correlations predicted by Tullius et al. [17]. The data obtained agrees (with some reservation) to the correlation proposed by Tullius et al. [17] in a specific range of total channel heights close to 1000 µm; the channel height studied by Tullius et al. [17]. The test section studied by Tullius et al. [17] has the following dimensions. The total channel height \( h_c \) considered by the authors was 1000 µm or 1 mm. The fins introduced over the surface were squares measuring 1mm \( \times \) 1mm. The fin heights studied in the study varied from zero (plain surface) to 75% of the channel height i.e. 750 µm. The channel heights considered in the present study vary from 465 to 1400 µm. Consider figure (18 d) that shows the experimental pressure drop data plotted against Reynolds numbers for the chip gasket combination c1-g2. The total channel height for this set-up equals 920 µm with a fin to channel height ratio of 0.54. This configuration comes quite close to the fin and channel configurations studied by Tullius et al. [17] in their study.
7.3.1 Comparison of pressure drop values

Figure (19) shows the experimental Pressure drop data compared against the correlation developed by Tullius et al. [17] at various Reynolds numbers. The figures (figure (19 a – 19 i)) are arranged in descending order of their total channel height.
The values of pressure drop obtained from the experiments are not predicted accurately by the correlation proposed by Tullius et al. [17]. At total channel heights close to 1000 µm, the predicted values show one of the least mean error of 30% and -17% for c1-g2 with a total channel height of 920 µm and (figure (19 d)) for c3-g1 (figure (19 c)) with a total channel height of 1100 µm respectively. As the channel height increases or decreases beyond 1000 µm, the discrepancy between the experimental and the predicted pressure drop values increases. Consider chip c3-g2 (figure (19 g)) with a total channel height of 620 µm and chip c2-g2 (figure (19 f)) with a total channel height of 720 µm. The experimental values obtained are not predicted well with the correlation. The mean error of the correlation is –72% and -70% respectively for the two conditions.
Similarly, for total channel heights greater than 1000 µm namely c1-g1 (figure (19 a)) with a total channel height of 1400 µm and c2-g1 (figure (19 b)) with a total channel height of 1100 µm, the observed mean error, of 115% and -56% respectively, is large.

It is important to note that, although the correlation developed by Tullius et al. [17] does not accurately predict all the experimental data, the data trend for all the chip-gasket combinations is the same as the trend predicted by Tullius et al. [17]. This means that the correlation can be used a basis to develop a refined correlation that predicts data for a larger range of channel and fin configuration values.

### 7.3.2 Nusselt Number correlation

Figure (20) shows the experimental Nusselt number data compared against the correlation developed by Tullius et al. [17] at various Reynolds numbers. The figures (20a to 20i) are arranged in descending order of their total channel height ($h_c$).

Nusselt number data follows the same trend like the pressure drop data when compared with the correlation proposed by Tullius et al. [17]. The experimental Nusselt number data is not accurately
predicted by the correlation proposed by Tullius et al. [17]. However, the data does show the same trend as the correlation predicts.
The computational study conducted by Tullius et al. [17] was for square short fins with a clearance over the fin tops. The fin heights ranged from 250 to 750 µm and the transverse and the longitudinal pitch were 2mm. The total channel height considered in the study was a constant 1000 µm. The fins had a width of 1mm. This correlates closely with the fin dimensions considered in the study.

7.4 Repeatability
7.4.1 Pressure drop
To assess the repeatability of the tests, the chip gasket combinations c1-g2 and c2-g2 were tested twice.

![Pressure drop repeatability plots for c1-g2 and c2-g2](image)

The heat transfer coefficient and pressure drop values are compared in the following plots.

Figure (22) shows the repeatability plots for c1-g2 (a) and c2-g2 (b). Pressure drop is plotted on the y-axis whereas flow rate is plotted on the x-axis. From the plots, it can be seen that there is good repeatability in the pressure drop data. The differences in the two values of the tests were 32% and 31% for c1-g2 and c2-g2 respectively. The two plots presented herein show that the pressure drop values show good repeatability in the experiments conducted.

7.4.2 Heat transfer coefficient
Figure (23) shows the repeatability plot for plots c1-g2 (a) and c2-g2 (b) for heat transfer coefficient. Heat transfer coefficient is plotted on the y-axis and flow rate on the x-axis. Two tests were conducted on the two chip-gasket combination c1-g2 and c2-g2. The repeatability of heat transfer coefficient is not as good as seen for pressure drop. Some of the values do show repeatability, for example, the heat transfer coefficient for a flow rate of 50 and 200 ml/min for c1-g2 and the heat transfer coefficient at the same flow rate values for c2-g2. However, the differences in the two tests is large at the other two flow rates.
tested for both the chip-gasket combinations. Hence, it is somewhat difficult to determine the repeatability of heat transfer coefficient in the present study.

![Figure 23 Heat transfer coefficient repeatability plot for c1-g2 (a) and c2-g2 (b) ](image)

**Repeatability issues with heat transfer coefficient**

Flow thorough structured surfaces is not uniform. There is a constant disruption of boundary layer, unlike flow through plain channels and surfaces. Flow through grooved and structured microchannels undergoes resonance and oscillatory motion as observed by Ghaddar et al. [22] in their numerical study. The authors also observed vortex shedding behind the grooves. The location of the resonance and vortex does not remain constant, causing dynamic areas in the flow stream to undergo vortex shedding, resonance and oscillatory flow. This leads to considerable uncertainties in the flow itself, which in turn causes the prediction of Nusselt number a difficult job. Although over longer spans, the average number of vortices tends remain the same, hence the Nusselt number values will remain more or less the same at longer flow lengths. However, at shorter flow lengths, as studied in the current study, stray vortices and resonance effects will lead to a certain uncertainties in heat transfer coefficient that cannot be account for without some visualization studies at least.

**7.5 Analysis of the correlation**

The pressure drop values obtained for using this correlation yielded error values between -17 to 244%. Although almost all of the pressure drop values lie outside the experimental error range, the values follow the same trend as the experimental ones. Hence it would be useful to use the correlation developed by Tullius et al. [17] as a basis for improving pressure drop correlation for short micro pin fins. Here it is also important to note that the correlation comes close to predicting experimental data for chip gasket.
combinations which yield a total channel height close to 1000 µm which was the total channel height studied by Tullius et al. [17]

The Nusselt number correlation developed by Tullius et al. [17] show the same trend as the experimental values. The Mean Errors values range from 21 to about 150%. Considering that the trend does remain the same as the experimental values, this correlation is can be used as a basis for a new correlation or a modified for Nusselt numbers.

The following table gives the mean error for all the chip-gasket combinations studied. Traversing left to right in the table, decreases the gasket thickness (free flow area over the fins) while keeping the fin height constant and traversing top to bottom decreases the fin height while keeping the free flow area above the fins constant (gasket thickness).

*Table 8 mean error values for correlations developed by Tullius et al [17]*

<table>
<thead>
<tr>
<th></th>
<th>Pressure drop (%)</th>
<th>Nusselt number (%)</th>
<th></th>
<th>Pressure drop (%)</th>
<th>Nusselt number (%)</th>
<th></th>
<th>Pressure drop (%)</th>
<th>Nusselt number (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>c1-g1</td>
<td>89.93</td>
<td>97.50</td>
<td>c1-g2</td>
<td>108.78</td>
<td>149.97</td>
<td>c1-g3</td>
<td>244.00</td>
<td>139.42</td>
</tr>
<tr>
<td>c2-g1</td>
<td>-35.64</td>
<td>137.92</td>
<td>c2-g2</td>
<td>-17.11</td>
<td>47.69</td>
<td>c2-g3</td>
<td>120.11</td>
<td>84.45</td>
</tr>
<tr>
<td>c3-g1</td>
<td>-51.49</td>
<td>20.95</td>
<td>c3-g2</td>
<td>-69.73</td>
<td>55.04</td>
<td>c3-g3</td>
<td>-83.20</td>
<td>51.82</td>
</tr>
</tbody>
</table>
8. Conclusion

0.3 x 1 mm. rectangular staggered pin finned surfaces with varying fin heights were tested in this study. The fin heights studied were 200, 300 and 500 µm. A free flow area was introduced over the fin tops with the incorporation of gaskets. Three gaskets were chosen with gasket thickness of 265, 420 and 900 µm, providing a range of flow channel heights, from 465 to 1400 µm. The pin fins were machined on to a copper chip over a surface area of 10mm x 10mm. Pressure drop and heat transfer coefficients were analyzed over this test surface, using water as the cooling medium.

An increase in channel Reynolds number causes an increase in heat transfer coefficient as well as pressure drop in the channel. Heat transfer and pressure drop values also increase with an increase in fin height. However, it was also observed that heat transfer coefficient also increased with an increase in the free flow area over the fins (for some cases). Although such a behavior is counter intuitive, it can possibly be explained by flow mixing brought about by the presence of the free flow area and the disruption of the boundary layer brought about by the presence of fins. However, no direct observation of the mixing phenomenon was observed in the study. Further studies, possibly visualization, will need to be conducted to validate this claim of flow mixing phenomenon guiding heat transfer in channels with short micro pin fins.

The Nusselt number and pressure drop values obtained from the study were compared with the correlations proposed by Tullius et al [17]. Comparison of the experimental and predicted values showed that neither the Nusselt number, nor the pressure drop values obtained from the experiments could be accurately predicted by the correlations proposed. It is possible that the correlations proposed hold true for a small configuration of channel and fin parameters. Even in the present study, the mean error of -17% was observed for a total channel height of 920 µm. This coincides with the total channel height of 1000 µm studied by Tullius et al. [17]. The experimental data does however show the same trend as the predicted values. This means that the existing correlation proposed by Tullius et al. [17] can either be modified or used as a base for a new correlation for predicting heat transfer and pressure drop values in flows over short micro pin fins.
9. References


