Thermal–Hydraulic Characterisation of Obstacles in Microchannel Flow: The Influence of Confinement

John Patrick O’Connor

Stokes Laboratories
School of Engineering
Science & Engineering
University of Limerick

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Declaration

The substance of this thesis is the original work of the author and due reference
and acknowledgement have been made, where necessary, to the work of others. No
part of this thesis has been submitted in candidature for any degree.

John Patrick O’Connor (Candidate)

Dr. Jeff Punch (Supervisor)

This thesis was defended on July 6th 2017

Examination Committee

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<tr>
<th>Role</th>
<th>Name</th>
<th>Institution</th>
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<tr>
<td>Chair</td>
<td>Dr. Mark Davies</td>
<td>University of Limerick</td>
</tr>
<tr>
<td>External Examine</td>
<td>Dr. Tadhg O’Donovan</td>
<td>Heriot-Watt University</td>
</tr>
<tr>
<td>Internal Examiner</td>
<td>Dr. David Newport</td>
<td>University of Limerick</td>
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Abstract

Internet usage has increased exponentially in recent times, challenging the capacity of optical communication infrastructure. To address this issue, denser Photonic Integrated Circuit (PIC) devices being developed with smaller footprint area, have larger heat fluxes (∼1kW/cm²). Tight operating temperature tolerance (±0.1K), and proximity of neighbouring laser devices creates thermal crosstalk. In the context of a microchannel cooling solution for integration within PIC packages, the objective was to identify the role of confinement on heat transfer enhancement and pressure drop penalty associated with a cylindrical pillar within a microchannel. Functionally, the pillar could serve as electrical interconnection.

Two experiments were conducted to complete this objective. Firstly, head loss was measured on a 0.225mm x 1.5mm channel containing a centrally placed pillar of confinement ratio of 10-70% for ReDh 100-1,200. Secondly, heat transfer measurements were recorded using infrared thermography on a 6.25mm x 10.25mm heated area located 1mm and 8mm downstream of cylinder for channel confinements of 10-70% in a 1mm x 6.25mm channel, for a ReDh of 100-800. Triangular, rhombus and square pillar shapes were also studied.

Increases in both pressure drop and heat transfer were observed beyond a confinement of 60% and Re√A ≥ 400; postulated to be due to vortex shedding within the channel. The pillars’ contributions to the total head loss were correlated with pillar diameter, and were found to predict the pillar head loss to within ±20% up to a Re_d of ≃425. The closer placement of pillar to heat source (1mm) yielded higher heat transfer. Larger confinements (β ≥ 0.4) yielded lower heat transfer at low ReDh, due to a large wake generated. At the onset of vortex shedding, the heat transfer, NuDh, was greater than conventional channel flow. Smaller confinements (β ≤ 0.2) induced higher heat transfer than conventional channel flow. The pressure drop penalty was greater than the heat transfer enhancement for all configurations.

A smaller confinement (β < 0.3) is preferred in channels where the downstream flow is below the critical Re_d for vortex shedding, however larger confinement (β ≥ 0.5) is superior beyond this point. The findings of this thesis are important in identifying effective pillar placements and in defining appropriate pumping systems for a microfluidic cooling solution for integration within PIC packages.
Acknowledgements

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I wish to extend my sincerest gratitude to Dr. Jason Stafford and Nicholas Jeffers for their technical insights over the course of this thesis.
To everyone at Stokes Laboratories and Nokia Bell Labs: it has been a pleasure and privilege to work with you. In particular to Paddy and Ollie for tips in good craftsmanship, Fionnuala for doing all the difficult and sometimes frustrating administrative work effortlessly. I would like to thank Science Foundation Ireland for funding this doctoral work under grant no. 10/CE/I1853.
To my friends Jonathan, Alistair, Philip, Lester, Declan, Andrew, Ciaran, Valeria, Betta, and Bram, thank you for all the entertaining discussions inside and outside of the office and lab.
Finally, I would like to thank my family for all the support while undertaking this endeavour.
“The struggle itself toward the heights is enough to fill a man’s heart. One must imagine Sisyphus happy.”

-Albert Camus
To all the loved ones who are here and not here now
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# Nomenclature

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<th>Symbol</th>
<th>Description</th>
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<td>$A$</td>
<td>area</td>
<td>m$^2$</td>
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<tr>
<td>$C_D$</td>
<td>drag coefficient</td>
<td>–</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat capacity at constant pressure</td>
<td>Jkg$^{-1}$K$^{-1}$</td>
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<td>diameter</td>
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\[ m \quad \text{mass-flow rate} \quad \text{kg s}^{-1} \]

\[ P \quad \text{perimeter} \quad \text{m} \]

\[ Q \quad \text{heat} \quad \text{W} \]

\[ q \quad \text{heat flux} \quad \text{W m}^{-2} \]

\[ R \quad \text{thermal resistance} \quad \text{KW}^{-1} \]

\[ T \quad \text{temperature} \quad \text{K} \]

\[ t \quad \text{thickness} \quad \text{m} \]

\[ U \quad \text{mean velocity} \quad \text{m s}^{-1} \]

\[ V \quad \text{voltage} \quad \text{V} \]

\[ w \quad \text{width} \quad \text{m} \]

\[ X, Y \quad \text{pixel width} \quad \text{m} \]

\[ x, y \quad \text{Cartesian coordinates} \quad - \]

\[ x^* \quad \text{thermal development length} \quad - \]

\[ X_T \quad \text{thermal entrance length} \quad \text{m} \]

\[ Z^+ \quad \text{hydrodynamic development length} \quad - \]

Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>[ \beta ] = [ d/w ]</td>
<td>confinement ratio</td>
<td>–</td>
</tr>
<tr>
<td>[ \Delta ]</td>
<td>difference/drop</td>
<td>–</td>
</tr>
</tbody>
</table>
Nomenclature

\[ \Delta P \] pressure drop \quad \text{Pa}

\[ \epsilon \] channel aspect ratio \quad -

\[ \epsilon_s \] surface emissivity \quad -

\[ \kappa \] pillar head loss \quad -

\[ \mu \] viscosity \quad \text{Pa.s}

\[ \Phi \] normalised head loss \quad -

\[ \rho \] density \quad \text{kg m}^{-3}

\[ \sigma \] Stefan-Boltzmann constant, \( 5.669 \times 10^{-8} \) \quad \text{Wm}^{-2}\text{K}^{-4}

\[ \varepsilon \] surface roughness \quad \text{m}

\[ \zeta \] hydrodynamic loss \quad -

Dimensionless Numbers

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Gr )</td>
<td>Graetz number</td>
<td>( D_h Pr Re D_h / L )</td>
</tr>
<tr>
<td>( Nu )</td>
<td>Nusselt number</td>
<td>( h D_h / k )</td>
</tr>
<tr>
<td>( Pr )</td>
<td>Prandtl number</td>
<td>( \rho c_p / k )</td>
</tr>
<tr>
<td>( Re )</td>
<td>Reynolds Number</td>
<td>( \rho U D_h / \mu )</td>
</tr>
<tr>
<td>( Sr )</td>
<td>Strouhal number</td>
<td>( f d / U )</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>---------------------</td>
<td>-------</td>
</tr>
<tr>
<td>∞</td>
<td>ambient</td>
<td>-</td>
</tr>
<tr>
<td>A</td>
<td>foil area</td>
<td>-</td>
</tr>
<tr>
<td>app</td>
<td>apparent</td>
<td>-</td>
</tr>
<tr>
<td>bm</td>
<td>bulk mean</td>
<td>-</td>
</tr>
<tr>
<td>c</td>
<td>conduction</td>
<td>-</td>
</tr>
<tr>
<td>Channel</td>
<td>channel</td>
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</tr>
<tr>
<td>crit</td>
<td>critical</td>
<td>-</td>
</tr>
<tr>
<td>f</td>
<td>foil</td>
<td>-</td>
</tr>
<tr>
<td>f.conv</td>
<td>forced convection</td>
<td>-</td>
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<tr>
<td>fc</td>
<td>forced convection</td>
<td>-</td>
</tr>
<tr>
<td>gen</td>
<td>generated</td>
<td>-</td>
</tr>
<tr>
<td>h</td>
<td>hydraulic</td>
<td>-</td>
</tr>
<tr>
<td>in</td>
<td>inlet</td>
<td>-</td>
</tr>
<tr>
<td>k</td>
<td>Kapton</td>
<td>-</td>
</tr>
<tr>
<td>lat.</td>
<td>lateral</td>
<td>-</td>
</tr>
<tr>
<td>losses</td>
<td>empirical head loss</td>
<td></td>
</tr>
<tr>
<td>m</td>
<td>mean</td>
<td>-</td>
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### Nomenclature

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<td>manifold</td>
<td>-</td>
</tr>
<tr>
<td>n.conv</td>
<td>natural convection</td>
<td>-</td>
</tr>
<tr>
<td>out</td>
<td>outlet</td>
<td>-</td>
</tr>
<tr>
<td>p</td>
<td>paint</td>
<td>-</td>
</tr>
<tr>
<td>r</td>
<td>radiation</td>
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</tr>
<tr>
<td>rad</td>
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<td>-</td>
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</table>

### Acronyms

<table>
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<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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</thead>
<tbody>
<tr>
<td>IR</td>
<td>infrared</td>
<td></td>
</tr>
<tr>
<td>MWIR</td>
<td>mid wavelength infrared</td>
<td></td>
</tr>
<tr>
<td>PIC</td>
<td>photonic integrated circuit</td>
<td></td>
</tr>
<tr>
<td>PIV</td>
<td>particle image velocimetry</td>
<td></td>
</tr>
<tr>
<td>TEC</td>
<td>thermo-electric cooler</td>
<td></td>
</tr>
<tr>
<td>TIPS</td>
<td>thermally integrated photonic systems</td>
<td></td>
</tr>
<tr>
<td>TSV</td>
<td>through silicon via</td>
<td></td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Motivation

The emergence of smart technologies, enabled by the evolution of new mobile phone technology, portable tablet computers, and other personal telecommunications paraphernalia has led to a dramatic increase in internet traffic. This increased network traffic exerts an ever increasing load on the current network infrastructure, and this trend is predicted to grow exponentially in the near future (Brodkin, 2014; CISCO, 2016; Holland et al., 2013). Figure 1.1 illustrates this growing trend on a network which is not limited by the current optical infrastructure, that is to say an unlimited network.

As a result, major telecommunication equipment providers, such as Nokia, Ericsson and Huawei, are investing in new infrastructure which can handle these future demands (Luckerson, 2014; Soref, 2010). The industry is now focusing more on increasing optical network capacity due to its superior data transfer rates, when compared to traditional copper wire networks (Bell-Labs, 2013; Reed, 2008).
To accommodate the near exponential increase in bandwidth demand, each component within the telecommunication network requires innovation to achieve the necessary data rate while maintaining or increasing efficiency. One technology where improvements can be made to optical networks is the transmitter and receiver devices, which feature Photonic Integrated Circuits (PICs) (Soref et al., 2006). Standard optical transceivers are composed of many components such as modulators, photo-detectors, multiplexers and laser diodes.

At present, PICs are expensive, not scalable and thermally inefficient: future PIC architectures require more functionality, higher data throughput and small footprint area, while maintaining or reducing costs. Integrating both electronic and photonic components at the same level is necessary to ensure feasibility. This entails significant mechanical design challenges, in particular effective component level cooling. A reduction of the device size combined with increased data throughput means increased power density, yielding higher and more localised heat fluxes.
1.1 Motivation

Conventional air cooling methods are both large in volume and inefficient in power usage, pointing towards a limit in their cooling capacity. A paradigm shift in the current PIC package design is required to evolve the capacity of these devices further. A solution which satisfies the optical, electronic and heat removal requirements, is needed.

Figure 1.2 illustrates the current state-of-the-art, which consists of four levels of packaging. The first level features the PIC, consisting of a transceiver device composed of a III-V material, such as Indium Phosphide (InP). The optical properties of this material are thermally sensitive and, as such, the thermal management of this device requires control rather than cooling below a threshold (in the manner of conventional ICs). Specifically, temperature fluctuations greater than $\pm 0.1^\circ C$ create a measurable shift in wavelength, incurring a signal breakdown as the receiver cannot demultiplex incoming data. Each laser device requires $\sim 100$ mW to operate. The geometry of the laser bars in such a device is long ($\sim 0.5–1$ mm) with comparably small cross-section areas ($\sim 2 \times 2 \mu m^2$) – yielding heat fluxes of approximately 1 kW/cm$^2$ per laser.

At the second package level, the photonic devices are placed on a thermo-electric cooler (TEC) inside a Kovar package. Additional optical devices, such as lenses, multiplexers and optic fibre connections, are placed in the Kovar package. At present, the approach is to use the TEC to sub-cool all photonic devices, while deposited resistive heaters are employed nearby each laser to raise their temperature to a suitable threshold. This method is inefficient as the TEC expends more energy than needed to cool each laser to the desired temperature. At the walls of the Kovar package are electronic drivers, used to electrically operate each laser; they are also thermally connected to the Kovar package, where wires are bonded across to the photonic devices within.

The third level package contains both the previously described Kovar package,
optical transmitter and receiver, along with supporting IC modules. This is referred to as a CFP package, which is pluggable so that each device can be replaced or upgraded, if needed. The fourth level of packaging connects the PIC package to the circuit packs found in typical data centre racks. At this level, forced convection air cooling is used to remove the thermal energy from these circuit packs, thereby placing an upper limit on the overall performance of the PIC components.

**Figure 1.2:** Representative model of the four different levels of packaging for a typical telecommunication data centre rack, transmitting and receiving information: PIC devices reside at the first level and are encased in their protective second level package, a Kovar package. A modular CFP package is used to contain the second level package, while the CFP package itself is located in a server blade (Jeffers et al., 2014)
1.1 Motivation

To ensure the next generation of PICs to be smaller, yielding higher throughput, and increasing efficiency while maintaining or reducing cost, the IC industry provides adequate inspiration and expertise. Complementary Metal Oxide Semiconductor (CMOS) processing techniques can further integrate electrical and optical components on a smaller footprint area, yielding a superior PIC architecture. Conventional CMOS processes is a preferred method in creating PICs, as it has been used by the semiconductor industry for over 40 years (Ginsberg, 1992; Soref et al., 2006). This technology shows the path to complete electrical and optical component integration on the same wafer. One limitation of CMOS is that silicon does not make an effective semiconductor laser, as it is an indirect band-gap material (Liang and Bowers, 2010). A hybrid approach is being considered: it involves depositing the active III-V materials first as the active laser device, while other optical components, such as passive sections, wave guides, and multiplexer, are fabricated on the silicon wafer (Duan et al., 2014).

The Thermally Integrated Photonics Systems (TIPS) architecture\(^1\) proposed by Jefters et al. (2014) and Enright et al. (2014) is illustrated in figure 1.3. The TIPS concept was proposed to increase functionality, device density and cost, over the older architecture by incorporating higher component densities and integration of photonic/electronic devices. TIPS will integrate optical, electronic and thermal layers into one package. Laser bars and electronic components are connected using Through Silicon Vias (TSVs). As shown in figure 1.3, the TSVs pass though the thermal plane. Between the optical and electronic plane, lies the thermal feature of the package, the microchannel array for microfluidic cooling of the TECs. This package combines both temperature control and the conventional thermal management practice of removing heat to reduce the global temperature below a threshold value. Firstly, a heat-spreading material is deposited by the laser

\(^1\)H2020 Grant Agreement 644453, http://www.tips2020.eu/project-goals
1. INTRODUCTION

Figure 1.3: Detailed illustration of the proposed next-generation TIPS, with the current PIC architecture for comparison (Jeffers et al., 2014).

bar, reducing the heat fluxes generated from more than 1 kW/cm$^2$ to less than 0.1 kW/cm$^2$ (Enright et al., 2014). The heat spreading from the laser bar is then directed into the thermal layer using micro-TECs. This heat is then convected away using a microchannel array. At the same time, heat generated from the lower electronic plane is convected into the microchannel array. This system is more efficient as it does not dissipate excess energy via heaters. TIPS will incorporate a wide range of optical, thermal and electronic design methods, into a compact and scalable solution. The final aspect of this system is the removal of heat to the ambient temperature server room, via conventional forced air cooling.

Figure 1.4 summarises the proposed TIPS thermal management system (Enright et al., 2014). A microchannel array is contained in the silicon layer between the optical and electronic planes, convecting heat into the fluid contained in the array. The working fluid is driven by a micro-scale pump from the primary heat exchanger,
Figure 1.4: Proposed TIPS for the next generation PIC device (reprinted from Enright et al. (2014)): Micro-scale pumps are used to drive fluid into the thermal plane between the electrical and optical planes – removing heat using conduction through the silicon wafer and convection into the fluids contained in the microchannel array. The working fluid is cooled using a secondary heat exchanger.

the microchannel array, into the secondary heat exchanger, removing heat from the system. Jeffers et al. (2014) investigated the limiting conditions for optimised heat transfer and head loss performance for the primary heat exchanger system. One design choice not investigated was the role of TSVs placed within the channel. Integrating a pillar containing TSVs into a channel will disturb the downstream flow, creating both a wake region and shedding flow region (Wang et al., 2013). Figure 1.5 illustrates the flow concept. This arrangement poses the question of how this flow feature can be understood and utilised effectively in comparison with a conventional channel flow. A literature survey presented in chapter 2 highlights gaps in the understanding of the role of the pillar width relative to the channel width (confinement ratio, $\beta$) – specifically the thermal and hydrodynamic performance of such configurations compared to a microchannel without pillars.
1. INTRODUCTION

Figure 1.5: Concept of the microchannel array integrated into a laser array. Cross-sectional and plan views of channel containing a TSV integrated into a pillar embedded in the channel. Plan view illustrates the typical flow behaviour downstream of the pillar in a laminar flow regime.

1.2 Research Objectives

The objective of this thesis is to investigate the thermal and hydrodynamic performance of a single pillar placed in a microchannel upstream of a heat source. Specifically, to identify the influence of pillar confinement and location upstream of the heat source. It is postulated that the heat transfer would increase due to the complex downstream flow generated, however resulting shedding vortices are not well understood from a thermal hydraulic perspective. To address this, two experiments will be carried out in order to understand the thermal and hydrodynamic characteristics of such a configuration. The key objectives and experiments of this thesis are:

- to measure the head loss contributions of a single channel containing a pillar across a range of laminar flow Reynolds No. for a range of pillar widths.
1.3 Thesis Compendium

- to characterise the downstream wall heat transfer contribution of the same range of pillar widths across a range of laminar flow Reynolds No. and different upstream locations.

Completing and understanding these experiments will yield design guidelines for the current and future iterations of the TIPS architecture for future optical communications transceivers.

1.3 Thesis Compendium

The research undertaken in this thesis has been divided into six chapters as follows:

- *Chapter 2* presents a comprehensive review of the fluid mechanics associated with pillars within channels. Heat transfer and hydrodynamic characteristics of this type of flow configuration are discussed.

- *Chapter 3* details the two experiments performed to measure the head loss and heat transfer characteristics of the pillar channel configurations.

- *Chapters 4 & 5* respectively describe the head loss and heat transfer behaviour of the pillar channel configurations.

- *Chapter 6* presents the conclusions to this thesis and offers recommendations based on those conclusions for future research on this topic.

Appendices including the author’s published work and extended experimentation details are available at the end of this thesis.

The following chapter will present a literature survey addressing the typical flow regimes associated with channel configurations considered in the TIPS system. Literature gaps will show where the experiments undertaken in this thesis are relevant to the current understanding of confined pillars in a channel.
1. INTRODUCTION
Chapter 2

Literature Review

This chapter surveys the literature addressing the fluid flow regimes and heat transfer performance of confined pillars within channels. The chapter is divided into several parts, describing the classical fluid mechanics of flow past a circular cylinder, effects of confinement, behaviour of head loss with respect to changing confinement and, finally, what effect this confinement has on heat transfer – specifically on a downstream wall. The following section presents the classical flows associated with a circular pillar in cross flow.

2.1 Flow Past a Circular Cylinder

Fluid flow past bluff bodies, such as aerofoils, spheres and cylinders, has been a topic of extensive research since the late 19th century (Karman, 2013; Prandtl and Tienjens, 1929; Strouhal, 1878). It involves phenomena such as flow separation, flow stability, boundary layer theory, and turbulent fluid flow behaviour. This knowledge can be applied to practical scientific studies, such as thermal management, aerodynamics, passive flow control, and structural mechanics in the design of airplanes, buildings and heat exchangers (Amini et al., 2013; Holman,
2. LITERATURE REVIEW

When a fluid flows past a bluff body such as a cylinder or sphere, various flow structures can appear downstream depending on the Reynolds number based on the diameter of said body. Changes in the fluid velocity or geometry at a fixed temperature can be non-dimensionally scaled using the Reynolds number to characterise the form of the flow past the cylinder (Holman, 2010; Schlichting, 2016; Zdravkovich, 1997a,b). Considering a cylinder in free flow, the Reynolds number is described in equation 2.1

\[ Re = \frac{\rho Ud}{\mu} \]  

(2.1)

This is the product of the fluid density, \( \rho \), average fluid velocity in the direction of the cylinder, \( U \), the cylinder diameter, \( d \), and the fluid viscosity, \( \mu \). Table 2.1 presents a tabular description of the flow forms associated with increasing Reynolds number. At creeping flow rates or approximately zero flow rate, a steady laminar flow exists with no area of separated, recirculating flow occurring around the cylinder body. An area of separation begins to occur as Reynolds number increases, with two symmetric regions of recirculating flow. This is known as a wake. This wake formation behind the cylinder increases in length as a function of increasing Reynolds number. Further increases in flow rate past the cylinder lead to quasi-unsteady events occurring further downstream of the cylinder. The pressure field surrounding the cylinder cannot maintain its symmetry and begins to oscillate the wake, leading to vortices shedding from the cylinder. This type of flow form is known as a Von Kármán vortex street. As the Reynolds number grows, the wake regions begin to reduce in size and the vortices become larger and more frequent. Ultimately, the wake region and downstream flow become fully turbulent in nature.

When fluid flows past a bluff body, an equal and opposite drag force acts upon the fluid. When equating the forces across a bluff body, the drag force resisting the fluid motion can be described as a function of the pressure and fluid speed
Table 2.1: Tabular description of the downstream flow regimes of a circular cylinder for a Reynolds No. range of \( \sim 0 - 3.5\times10^5 \). Strouhal No. ranges are highlighted for each flow regime. This information has been re-tabulated from Schlichting (2016). Fluid flow is from left to right with a Reynolds No. range of 30-2000 being directly relevant to this thesis.

<table>
<thead>
<tr>
<th>Reynolds No. regime</th>
<th>Flow regime</th>
<th>Flow form</th>
<th>Flow characteristic</th>
<th>Strouhal No., Sr</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{Re} \rightarrow 0 )</td>
<td>Creeping flow</td>
<td></td>
<td>Steady, no wake</td>
<td>–</td>
</tr>
<tr>
<td>( 3 \leq \text{Re} &lt; 30 )</td>
<td>Vortex pairs in wake</td>
<td></td>
<td>Steady, symmetric separation</td>
<td>–</td>
</tr>
<tr>
<td>( 30 \leq \text{Re} &lt; 90 )</td>
<td>Onset of Karman vortex street</td>
<td></td>
<td>Laminar, unstable wake</td>
<td>–</td>
</tr>
<tr>
<td>( 90 \leq \text{Re} &lt; 300 )</td>
<td>Pure Karman vortex street</td>
<td></td>
<td>Karman vortex street</td>
<td>( 0.14 &lt; \text{Sr} &lt; 0.21 )</td>
</tr>
<tr>
<td>( 150 \leq \text{Re} &lt; 1.3 \times 10^5 )</td>
<td>Subcritical regime</td>
<td></td>
<td>Laminar, with vortex street instabilities</td>
<td>( \text{Sr} = 0.21 )</td>
</tr>
<tr>
<td>( 1.3 \times 10^5 \leq \text{Re} &lt; 3.5 \times 10^6 )</td>
<td>Critical regime</td>
<td></td>
<td>Laminar separation</td>
<td>No preferred frequency</td>
</tr>
<tr>
<td>( 3.5 \times 10^6 \leq \text{Re} )</td>
<td>Supercritical regime (transcritical)</td>
<td></td>
<td>Turbulent separation</td>
<td>( 0.25 &lt; \text{Sr} &lt; 0.30 )</td>
</tr>
</tbody>
</table>
past this body. It is represented by the equation 2.2.

\[ F_{\text{drag}} = 0.5C_D \rho U^2 A_{\text{frontal}} \]  

(2.2)

The drag force \( F_{\text{drag}} \) is a function of the fluid density, \( \rho \), drag coefficient, \( C_D \), fluid velocity, \( U \), and the surface area facing upstream of the flow, \( A_{\text{frontal}} \). The drag force behaviour can alter greatly when considering flow past a cylinder, as shown in figure 2.1.

![Figure 2.1: Drag coefficient of a cylinder in free flow as a function of Reynolds No. Flow lines around cylinder indicate the angle of separation, \( \theta \), at different Reynolds No. Flow past cylinder is from left to right. Graph redrawn from experimental and analytical data (Bejan, 2013; Schlichting, 2016).](image)

At low Reynolds No., the measured drag force coefficient is high, due to viscous forces dominating the fluid flow. As the Reynolds No. is a ratio of inertial to viscous
2.1 Flow Past a Circular Cylinder

forces inside a volume of moving fluid, it is a natural conclusion (Holman, 2010). At a Reynolds No. of \( \sim 10^3 \), the ratio of drag to viscous force is approximately equal. From \( \text{Re}_d = 10^3 - 3 \times 10^5 \), the drag force remains approximately constant; beyond this point, a sharp reduction in the drag coefficient is observed. This is known as the critical point, where the flow surrounding the cylinder becomes fully turbulent. The boundary layer surrounding the cylinder has a wider angle of separation, causing the drag coefficient to reduce significantly as the frontal area increases. The vortex shedding street downstream of the cylinder has a characteristic frequency band, meaning that the vortices are periodic and ordered in nature. Such behaviour can be represented by a non–dimensional No., the \textit{Strouhal No.} This dimensionless number relates the shedding frequency, \( f \), to the geometric scale, in this case the cylinder diameter, \( d \), and the fluid velocity, \( U \), in equation 2.3:

\[
\text{Sr} = \frac{f d}{U}
\]  

Equation 2.3

Table 2.1 indicates typical Strouhal No. values of 0.14–0.21, across a range of Reynolds No. up to \( \text{Re}<10^3 \), where the viscous forces are dominant over the inertial. From \( 10^3 > \text{Re} > 3 \times 10^5 \), the Strouhal No. remains approximately constant at a value of 0.21. Beyond the critical Reynolds No., the Strouhal No. increases to a range of 0.25-0.3. At this point, the fluid flow is turbulent in nature. The inherently unstable nature of the flow creates additional vortices of their own higher frequencies, increasing Strouhal No. and its range.

This section presented a baseline understanding of the fluid mechanics for a cylinder in cross flow. A comprehensive review of the literature addressing 2D flow past a circular cylinder can be found in the work of Zdravkovich (1997a). The majority of experiments involving this flow were conducted in a wind/water tunnel. In the context of this thesis, this literature gives a good understanding of
the flow mechanisms expected downstream of a circular pillar. However, it does not comment on the implications of how these flow mechanisms affect global head loss and heat transfer characteristics in a microchannel, typical of the application studied in this thesis. Furthermore, confinement of the duct width and height relative to the cylinder diameter can influence the flow regime by comparison to the idealised case. The next section will discuss how the idealised case varies when confinement is a factor. Figure 2.2 presents Strouhal No. as a function of Reynolds No. across a cylinder. Variations are evident when the drag and viscous forces around the cylinder are not equivalent.

![Figure 2.2: Variation of Strouhal No. as a function of Reynolds No. for a cylinder in free flow. Redrawn from White (2011). Dashed lines (−−) indicate uncertainty of the measurement at various Reynolds No.](image)

Figure 2.2: Variation of Strouhal No. as a function of Reynolds No. for a cylinder in free flow. Redrawn from White (2011). Dashed lines (−−) indicate uncertainty of the measurement at various Reynolds No.
2.2 Confined Flow Past Cylinders

Research provided by both experimentalists and theoreticians up to recently, commonly retreats to the “comfortable dream–world” of two dimensionality, when studying the flow behaviour of a cylinder in free flow (Morkorvin, 1964). Fluid flow past a bluff body, and all aerodynamic bodies, tend to be empirically studied in a wind/water tunnel of some scale. To simplify the analysis, the vertical and horizontal confinements of the cylinder are often not considered. The following section discusses the influence of confinement on the two–dimensional case as it is applicable to understanding the head loss and wall heat transfer characteristics studied in chapters 4 and 5.

2.2.1 Aspect Ratio

In most experiments found in the literature, which study flow past a cylinder, the cylinder length to the diameter ratio (aspect ratio = L/D) was sufficiently large to assume two dimensional conditions. In reality, the full flow field is more complicated, with the classical flow classifications appearing along with horseshoe vortices at the ends of the cylinder. This flow structure is represented in figure 2.3. Having a suitably large aspect ratio is necessary to simplify any numerical and analytical experiment, and lesser values of aspect ratio have been shown to diverge the typical flow characteristics from the ideal. Parameters such as boundary layer thickness and incoming flow velocity have been demonstrated to increase the number of horseshoe vortices appearing upstream of the cylinder for a range of Reynolds No. Zdravkovich (1997b) presents a detailed graphical map from the literature which predicts the number of vortices generated as a function of the
2. LITERATURE REVIEW

Figure 2.3: Horseshoe vortex and flow behaviour generated by the interaction of the wall and cylinder with incoming flow. Taken from Melville and Coleman (2000).

Reynolds No. \( (Re_D) \) and the ratio of the cylinder diameter to boundary layer height \( (D/\delta_B) \). At a high Reynolds No. \( (>2700) \) that is outside the range considered in this thesis, numerous unsteady swirling vortices are generated, which are unstable and oscillatory in nature. Taniguchi et al. (1981) conducted surface pressure distribution measurements downstream of the cylinder with an incoming turbulent boundary layer for three aspect ratios \( (H/D = 1,3,5) \) for a Reynolds No. range of 460–33,000. Time-averaged isobar contours showed a “kidney” shaped feature which elongated with increasing aspect ratio. Belik (1973) measured the pressure distribution at the mid-plane, upstream of the cylinder for \( H/D = 3.3 \), \( D/\delta_B = 2 \) and \( Re = 36–220k \). The pressure coefficient, \( C_p \), was shown to be lower as the flow approached the cylinder by comparison to potential flow, due to horseshoe vortices. The pressure behaviour upstream was predicted using an analytical solution based on \( \theta \), \( r \), and \( D \), where \( \theta \) is a spanning angle, \( r \), the radial distance upstream and \( D \), the cylinder diameter. For the experimental range considered, the separation length, \( x_s \), upstream was found to vary by 0.45\( D \)–0.6\( D \). The upstream separation at the wall is influenced by the confinement of the duct height and width as the boundary layer is the variable influencing the behaviour. Baker (1991) showed that
the wake formation is independent of the boundary layer for \( H/D > 1 \). Below this threshold, the layers begin to reduce in size.

The behaviour of the laminar wake, onset of vortex shedding and subsequent high Reynolds No. flows have been examined for a long time. The flow structures of the laminar wake were first studied by Nishi (1925). This showed the three dimensional nature of the wake flow at \( Re = 15 \) and \( H/D = 8.5 \). Flow visualisation at the mid-plane of the duct exhibited a typical wake formation, however the span-wise direction revealed the flow to be directed from the mid-plane to the top/bottom walls, creating a saddle-point upstream of the swirling vortices inside of the wake. As the flow speed increases, a vortex street is produced. The Reynolds No. at which this occurs is influenced by \( H/D \) below a certain threshold (Nishioka and Sato, 1974). Figure 2.4 demonstrates this relationship, and illustrates that the onset of vortex shedding is increased significantly. This finding is important regarding this thesis, as the aspect ratio is kept low indicating that the onset of vortex shedding will be delayed. This is shown in chapters 4 and 5, specifically in sections 4.1, 5.1, and 5.2 where changes in both head loss and heat transfer are evident at higher Reynolds No.

Ribeiro et al. (2012) investigated the wake length and surrounding x,y,z velocity profiles for a range of aspect ratios (0.5–128), small confinement ratio (\( W/D = 2 \)) and for \( Re = 2–200 \). Particle-Image Velocimetry (PIV) and numerical simulations for various aspect ratios revealed two to three regions of accelerated and de-accelerated flow along the cylinder axis. Low aspect ratios (\( H/D = 0.5 \)) showed the flow profile to be more uniform and comparable to the channel flow. These axial flow profiles continue downstream. Once vortex shedding occurs, a stratified flow regimes forms, with a number of cells (central, side and end) and thin discontinuous regions between cells. As discussed in subsection 2.1, this simplified 2D flow regime is periodic in nature and so are all cells in the confined case. However, the frequency
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![Figure 2.4: Reynolds No. at onset of vortex shedding as a function of the ratio of the height of the cylinder to its diameter. Redrawn from Nishioka and Sato (1974).](image)

at which each cell sheds is different and, as flow speed increases, these cells can disappear and reappear from the wall (Konig et al., 1990). Shedding occurs at a higher frequency in the centre of the flow than near the channel walls. This literature is of direct relevance to this thesis: it implies that the desired change from laminar to unsteady flow occurs at higher Reynolds No. for the confined scenario than for the unconfined cases. This conclusion will help to explain the head loss and heat transfer behaviour measurement in chapters 4 and 5 receptively.

2.2.2 Confinement Ratio

Zdravkovich (1997a) presents a concise and informative overview of the literature surrounding the effect of wall confinement on the cylinder diameter. At flow regimes ranging from fully formed vortex shedding to turbulent flow, Zdravkovich (1997b) states that the flow can be compared to the two-dimensional flow under the following conditions:
(i) for confinement ratios of less than 10%, the effect of the confinement is small and flow physics are assumed to follow a cylinder in free flow.

(ii) for confinement ratios between 10–60%, the flow is influenced but may be corrected for by the use of measured data.

(iii) for confinement ratios greater than 60%, the flow is heavily influenced and altered around the cylinder region, making data nearly impossible to correct. White (1964) examined the influence of confinement ratio at very low flow rates, by measuring the flow resistance of wires falling in a glycerol solution. As the Reynolds No. approached zero for a range of W/D: 10-500, each confinement ratio plateaued. Resistance was found to divert from Lamb’s equation \( C_R = \frac{5.6}{\log(7.4/Re)} \) for Re<1. Coutanceau and Bouard (1977) studied the role of confinement on the point of separation beyond the creeping flow regime, and it was found to have a minimal effect. When the wake begins to form, increasing confinement ratio retards the wake length. Consequently, the wake became more stable as the onset of vortex shedding was delayed to a higher Reynolds No. Figure 2.5 shows the role of confinement on the delay of flow instability in the wake. Finally, the vortex shedding was shown to originate further downstream than the unconfined case.

Another feature of the flow affected by confinement is spacing between vortices. Rosenhead and Schwabe (1930) measured these spacings for confinements of up to 60% of the duct width, shown in figure 2.6. Confinements of 33% or greater showed the spacings to be independent of Reynolds No. Flow visualisations of similar confinement ratios are presented in literature, illustrating that vortex shedding from the cylinder is delayed until it is suppressed at a confinement of 67% (Zdravkovich, 1997b). Additional features of the flow are that the vortices generated downstream become as wide as the duct width, and are shown to cling
2. LITERATURE REVIEW

![Graph showing Re vs D/W](image)

**Figure 2.5:** Onset of vortex shedding observed at different rates of confinement and Reynolds No. Data reprinted from Zdravkovich (1997b).

...to the surface. This was simulated for confinements up to 90% with regions of downstream wall-separation and multiple stability criteria (Sahin and Owens, 2004). Instability of the wake at this point is shown in figure 2.7. The rate of change of separation angle with Reynolds No. \((d\theta/dRe)\) was shown to be sensitive to confinement. Higher confinement delayed the onset of wake formation, and the slope between \(\theta\) and Re \((d^2\theta/dRe^2)\) increased non-monotonically to a value of 150° (Mitry, 1977). At the higher Reynolds No. range prior to turbulent flow, visualisation experiments by Okamoto and Takeuchi (1975) showed that strength in vorticity increased with confinement (D/W = 0.093, 0.181, 0.25, 0.34). Accelerated flow past the sides of the cylinder are thought to be responsible for generating the stronger eddies.

The drag coefficient was shown to increase exponentially with confinement in experiments by Hiwada and Mabuchi (1981); a range of D/W = 0.22–0.8 increased \(C_D= 1–20\) respectively for Re = 45x10³. Huang and Feng (1995) found a similar relationship with confinement. The influence of confinement on Strouhal No. has been investigated for laminar and unsteady downstream flow. Strouhal No. was found to increase sharply from 0.22–0.5 with confinement of 0.2–0.6 for Re =
2.2 Confined Flow Past Cylinders

![Diagram](image)

Figure 2.6: Vortices spacing after onset of Von Kármán vortex street for a range of confinement ratios. Data reprinted from Rosenhead and Schwabe (1930).

Figure 2.7: Onset of vortex shedding observed at different ranges of large confinement and Reynolds No. of 200. Data reprinted from Sahin and Owens (2004).
45x10³ (Hiwada and Mabuchi, 1981) with the shedding frequency spectra ceasing at D/W ≥ 0.6. It was also shown by Ramamurthy and Ng (1973) that Strouhal No. was independent of Reynolds No. for D/W = 0.071-423 at Re = 13x10³–230x10³. Confinement has been shown to reduce the critical Reynolds No. for an increase in D/W ratio (Richter and Naudascher, 1976). Finally, confinement was also found to decrease the wake width significantly by Okamoto and Takeuchi (1975), as shown in figure 2.8.

Figure 2.8: Relationship between wake width, b/D, as a function of wake length, x/D, for Re = 32x10³. Data adapted from Okamoto and Takeuchi (1975).

The literature presented here describes how the wake and downstream flow change with flow rate and confinement. This literature details what the expected flow features are at different Reynolds No., downstream of the pillar. In particular, how the expected turbulent kinetic energy (TKE) increases both the head loss and heat transfer measurements in chapters 4 and 5.
2.3 Multiple Cylinders and Cross-Section Shape

Using an array of pillars for heat transfer enhancement has been widely studied, with the cross-section of the shape receiving particular attention. As lower order cylinders and several cross-section shapes are studied later in chapter 5, the subsequent literature survey will focus on the typical flow interactions of such configurations.

The flow around arrays of cylinders has many applications, however research surrounding the subject initially was *ad hoc* with the *a priori* assumption of the flow behaving as a single cylinder without interference effects between them (Zdravkovich, 1997b). Cylinder arrays in line, perpendicular or staggered to the flow direction are sensitive to interference based on spacing ratio between them. Initial studies on twin cylinders with ratio of downstream spacing, $S$, and lateral spacing, $T$, to cylinder diameter, $D$, could be categorised into the flow regimes described in section 2.2. Increasing the number of cylinders in forms of clusters adds complexity both to the upstream and downstream flow behaviour. For the purposes of this thesis, heat transfer from arrays of pillars is not directly relevant, with the exception of the twin pillar cases presented in chapter 5. A detailed review is written by Zdravkovich (1997b). Detailed literature surrounding the topic in terms of fluid behaviour, head loss and heat transfer can be found in several textbooks: Bejan (2013); Holman (2010); Idelchik (1960); Kays and London (1998); Tullius et al. (2012); Zdravkovich (1997b).

Flow past bluff bodies is a cornerstone and fundamental area of study in aerodynamics, giving rise to flight. When comparing bodies of different cross-sectional shape, such as square, triangular, diamond, etc., the drag coefficient is the appropriate metric to compare each shape (White, 2011). The drag behaviour as a function of Reynolds No. follows a similar trend to the classical two-dimensional
2. LITERATURE REVIEW

Table 2.2: Description of three twin cylinder arrangements and the flow regimes associated with changes in their respective spacing (Zdravkovich, 1997b). Diagram shows pillar orientations with respect to incoming fluid flow, U.

<table>
<thead>
<tr>
<th>In-line</th>
<th>Regime</th>
</tr>
</thead>
<tbody>
<tr>
<td>S/D = 1-1.2</td>
<td>The separation of the wake continues behind the downstream cylinder leading to a single vortex street.</td>
</tr>
<tr>
<td>S/D = 1.2-3.65</td>
<td>Reattachment on the downstream cylinder was observed to reattach in three ways; alternating, permanent or intermittent as a function of increasing spacing ratio. Single vortex streets are generated downstream.</td>
</tr>
<tr>
<td>S/D = 3-4.0</td>
<td>Bistable flow regime with intermittent changes producing two vortex streets.</td>
</tr>
<tr>
<td>S/D = 3.9-6</td>
<td>Coupled vortex shedding occurs which is synchronised in frequency and phase.</td>
</tr>
<tr>
<td>S/D &gt; 6</td>
<td>De-coupled twin vortex streets with different frequencies are generated.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Perpendicular</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>T/D = 1-1.2</td>
<td>Single vortex street is produced, as cylinders act together as a bluff body.</td>
</tr>
<tr>
<td>T/D = 1.2-2.2</td>
<td>Wide and narrow vortices appear behind each cylinder. Flow between cylinders forms a jet. Wakes reduce intermittently and alternate in length with jet biasing towards narrower wake.</td>
</tr>
<tr>
<td>T/D = 2.2-5</td>
<td>Wakes are coupled and equal in size. Vortices generated are synchronised in both frequency and phase. Two vortex streets are present in the downstream flow.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Staggered</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>S/D = 1.1-3.5, T/D &gt; 0.2S</td>
<td>Strong gap flow applies a large lift force on downstream cylinder, affecting its wake. Flow between cylinders stops intermittently cancelling out the lifting force on downstream cylinder.</td>
</tr>
<tr>
<td>S/D &gt; 2.8, T/D &gt;0.4</td>
<td>Downstream flow influences upstream wake formation. Lift force is applied to the upstream cylinder.</td>
</tr>
</tbody>
</table>

![Diagram showing pillar orientations with respect to incoming fluid flow, U.](image)
2.3 Multiple Cylinders and Cross-Section Shape

In the laminar separation and vortex shedding regimes, the drag coefficient remains relatively constant due to the balance of viscous to frictional forces in the fluid. This constant changes relatively with the shape and a list of common shapes and their drag coefficient is presented in Table 2.3.

Table 2.3: Drag coefficients of some simple shapes and world record holder of the 100m sprint, Usain Bolt (Hernandez et al., 2013; White, 2011).

<table>
<thead>
<tr>
<th>Shape</th>
<th>$C_D$</th>
<th>Shape</th>
<th>$C_D$</th>
</tr>
</thead>
<tbody>
<tr>
<td>○</td>
<td>1.2</td>
<td>□</td>
<td>2.1</td>
</tr>
<tr>
<td>◊</td>
<td>1.6</td>
<td>▵</td>
<td>1.6</td>
</tr>
<tr>
<td>○</td>
<td>1.0</td>
<td>▾</td>
<td>1.2</td>
</tr>
<tr>
<td>△</td>
<td>1.7</td>
<td>▶</td>
<td>2.0</td>
</tr>
<tr>
<td>Usain Bolt</td>
<td>1.2</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The drag coefficient is an effective way of quantifying hydrodynamic loss, however this thesis addresses the application of a cylinder within channels. Identifying parameters such as head loss and heat transfer coefficient are of a practical consequence. The previous section detailed the flow types and associated fluid mechanics in what is considered to be a free or partially constrained case. There are not many sources of experimental data, measuring head loss in a channel containing a pillar. Idelchik (1960) provides a model to describe head loss in a channel that merges the role of confinement with the mechanics of a bluff body. Equation 2.4 describes the head loss contributions of a singular bluff body in a channel as:

$$\kappa = 1.15C_d \frac{\frac{A_{\text{surface}}}{D_h}}{\left(1 - \frac{A_{\text{surface}}}{D_h}\right)^3 \left(1 - \frac{2y}{D_h}\right)^{\frac{1}{4}}}$$

(2.4)

This equation was stated to be valid for a wide range of Reynolds No. ($10^{-1} - 10^6$) and simplified to three parameters for square and circular channel cross-sections: ratio of cylinder frontal surface area to hydraulic diameter, $A/D_h$, drag coefficient, $C_d$, and ratio of cylinder height to hydraulic diameter, $y/D_h$. The constant $1.15$ is modified to 0.15 to get an equation similar to the one of Idelchik (1960) who used $\kappa = 1.5C_d$. The approach of adding the head loss contribution of a cylinder to the total head loss in a channel was also studied by Idelchik (1960) who postulated that the coefficient of head loss is $\kappa = 1.5C_d$ where $C_d$ is the drag coefficient of a singular bluff body in a channel.

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$C_d$, and proximity of cylinder to wall, $y/D_h$. This head loss can be added to the frictional loss of the duct to describe the total head loss behaviour in equation 3.8 (Idelchik, 1960). The validity of such an equation is not commented on here, and should however be treated with caution. Furthermore, evidence in the literature implies that drag coefficient changes as a function of confinement, and points to the conclusion that channels with high confinement may not conform to equation 2.4 (Hiwada and Mabuchi, 1981). Several numerical studies which investigate variation of drag coefficient as a function of Reynolds No. and confinement ratio strengthen this claim (Bharti and Chhabra, 2007; Rao et al., 2011). This gap in the literature is addressed by the head loss measurements presented in chapter 4.

2.4 Heat Transfer Involving Pillars

Heat transfer behaviour of pillars has been studied in depth for the two-dimensional case for both free and forced convection (Zdravkovich, 1997a). Typically, two regions are of interest, the pillar itself and the downstream walls. Studies addressing the heat transfer around the pillar have carried out by several authors (Bharti et al., 2006; Chatterjee and Bittagopal, 2012; Nitin and Chhabra, 2005; Park, 2013; Seyyedi et al., 2013; Sharma and Eswaran, 2005). This section will detail the role of confinement and contemporary studies surrounding the subject.

Utilising a pillar to augment heat transfer has been demonstrated to be an effective method, with a modest head loss penalty (Kandlikar, 2014). The presence of an upstream pillar upsets the growing boundary layer of the fluid, promoting enhanced fluidic mixing and, by extension, heat transfer. This is due to the reduced areas of separation surrounding the cylinder in the triangular case.

Srikanth et al. (2010) compared heat transfer between triangular and square pillars of confinement ratio, $\beta = 25\%$. The triangular shape performed better than
2.4 Heat Transfer Involving Pillars

the square cylinder. Alfieri et al. (2013) numerically investigated a variety of pin-fin arrangements for cooling 3D chip stacks using vortex shedding regimes (Re = 60-450), and examined the effect of three different confinement ratios: lateral, vertical and longitudinal confinement. The variation of these parameters was appraised by measuring the Strouhal No., heat transfer enhancement and chip temperature. Drag coefficient of the cylinder was found to increase non-monotonically with confinement, and decrease with Reynolds No. The following was observed in a single pin case: Strouhal frequency increased by 100% with a Nusselt No. increase of 30%. Temperature and vorticity plots with a confinement of 50% showed the vortices to cling to the wall, revealing segmented regions of vorticity and temperature downstream of a heated cylinder. In the 25 & 50 pin row cases, the Nusselt No. increased significantly from the open channel case (300%). Finally by increasing the confinement ratio to 0.5, the magnitude of vortex shedding experienced in the channel was diminished greatly.

Meis et al. (2010) numerically investigated a 2D channel domain with an obstacle placed in the channel to quantify the heat transfer enhancement from the channel sidewall (which had an isothermal boundary condition). Three pillar types were investigated for a range of shapes (circular/elliptical, rectangular and yawed rectangular), aspect ratios, and confinement ratio and degree of yaw angles (±40°). It was found that the optimal parametric permutation was a rectangular pillar with confinement ratio, $\beta = 1/2$, rotated clockwise at 30°, however the heat transfer gains (150%) did not outweigh the measured pressure drop penalty (250%).

Wang et al. (2013) experimentally investigated the downstream heat transfer performance past a single pillar within a microchannel ($D_h = 391 \mu m$ and channel aspect ratio of 6) for Re = 100-5,600 with air, and Re = 100-2,400 with water for several shape configurations: triangular, diamond, twin circular and two circular pillars with a varying diameter. Confinement ratio ranged from 10-20% with
heat-transfer performance measured downstream of pillar configurations over a 1x1mm² heater in air and PIV carried out in water. Larger confinement ratio, W/D, with circular shapes showed an increase in heat transfer, with twin pillar configurations yielding higher heat transfer. Shapes with lower drag coefficient – such as the circular shape – yielded higher heat transfer. High turbulent kinetic energy values were observed over the heated area and these values were higher at the mid-plane in comparison to the wall. TKE was correlated to an increase in heat transfer. Jung et al. (2012) investigated the flow field of a single pillar geometry for comparable channel geometry and Reynolds No. range of 100–700. It was found that the stagnation region began to decay after a channel Reynolds No. of 400, with the TKE growing significantly after this point. TKE values of up to 10% were observed just three diameters downstream. Armellini et al. (2010) compared the downstream flow profiles for circular, triangular, square and diamond shaped pillars. Flow profiles showed that the diamond and triangular shapes induced low recirculation zones and higher levels of fluidic mixing downstream. This is in line with the heat transfer measurements of Wang et al. (2013). Wang et al. (2012) measured local wall heat transfer at Reynolds No. 50–100,000. The channel flow was thermally fully developed with a channel aspect ratio of 4, and the channel was fitted with a rectangular cylinder of 80% confinement. The local wall heat transfer was shown to correlate to \( Re^{0.51} \).

The heat transfer studies presented in the section address only single or small range of confinement ratios; nor does the literature surveyed present a study on the role of pillar location or shape on downstream wall heat transfer. This thesis investigates in a single experimental series a wide range of confinement ratios, sensitivity of pillar location downstream versus wall heat transfer and a comparison of pillar shape. The details of this experiment are presented in section 3.2 and the measurements are discussed in chapter 5.
2.5 Closure

This chapter presented the literature addressing pillars in cross flow for both classical and confined cases, describing the mechanics of the flow. Heat transfer and hydrodynamic performance has been explained by the readily available literature describing the fluid mechanics. However, literature investigating the heat transfer and hydrodynamic losses as a function of confinement exclusively have not been addressed. A study such as this would help to answer the practical questions for the TIPS system previously described in section 1.2. The next chapter will address the experimental methods used to study the role of confinement from a heat transfer and head loss perspective.
Chapter 3

Experimentation

This chapter concerns itself with the experimental methods and test facilities used to characterise the downstream fluid dynamics and heat transfer associated with obstacles placed in a channel for thermal management applications. Head-loss coefficient and area-averaged heat transfer measurements using Infrared (IR) thermography were used in order to capture both hydrodynamic and heat transfer behaviour of this flow configuration.

The structure of this chapter is as follows: a detailed description of the methods used, construction of the channel geometries, the commissioned facilities for each experiment, and data reduction for each data set. Finally, the chapter concludes with an uncertainty analysis of the primary and derived variables used for each experiment.
3. EXPERIMENTATION

3.1 Head-Loss Coefficient

This section describes the details of the head loss coefficient experiment. This consists of the geometric characterisation of the fluidic channels fabricated, a description of the test facility as outlined in the procedural steps, and benchmarking of the facility against the theory using a conventional fluidic channel.

3.1.1 Channel Fabrication

Channel samples of nominal height, \( h = 225\mu m \), width, \( w = 1.5\text{mm} \), and length, \( L = 30\text{mm} \), were precision machined in a Datron M7HP CNC from a 2.0mm \((\pm 10\mu m)\) thick Acrylic sheet. Figure 3.1 shows the channel layout with the measured parameters. The cylindrical pillars of diameter, \( d \), were positioned in the centre of the channel for various confinement ratios, \( \beta = d/w = 10\% - 70\% \). The channels were cleaned in an ultrasonic bath with water containing detergent, and then wiped down with commercial Kimitek wipes to remove any debris or swarf which resided in the channels after machining.

The pillar diameter, \( d \), and channel length, \( L \), were measured using a Keyence microscope, with each channel cross-sectioned to measure the height and width. Each channel was sectioned and then potted in an epoxy resin. A sequence of polishing steps perpendicular to the channel direction was taken to expose a clean cross-section of the channel in a Lab-Struers polishing facility. The height and width were also measured for each channel. Table 3.1 presents the dimensional characteristics for all the channel samples. Given the size of the fluidic channels and method of fabrication chosen for this experiment, the magnitude of the roughness was considered to have an effect on the measured hydrodynamic characteristics. Surface roughness in microchannels has been shown by to induce earlier flow transition, (Kandlikar, 2012) which can shift with the magnitude of the relative
3.1 Head-Loss Coefficient

Figure 3.1: Sample cross-sectional (a), and overhead views (b,c), of the channel with parameters: height, h, width, w, pillar diameter, d, and channel length, L. An isometric sketch of the channel sample (d), is shown with a zoomed in view of the pillar (e).

Table 3.1: Channel dimensional characteristics with hydraulic diameter \( D_h = \frac{4hw}{2(h+w)} \).

<table>
<thead>
<tr>
<th>( \beta ) (-)</th>
<th>( L ) (mm)</th>
<th>( h ) (( \mu )m)</th>
<th>( w ) (mm)</th>
<th>( d ) (( \mu )m)</th>
<th>( D_h ) (( \mu )m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>29.97</td>
<td>227</td>
<td>1.958</td>
<td>0</td>
<td>409</td>
</tr>
<tr>
<td>0.1</td>
<td>30.00</td>
<td>231</td>
<td>1.476</td>
<td>150</td>
<td>399</td>
</tr>
<tr>
<td>0.2</td>
<td>29.94</td>
<td>228</td>
<td>1.478</td>
<td>395</td>
<td>395</td>
</tr>
<tr>
<td>0.3</td>
<td>29.97</td>
<td>224</td>
<td>1.478</td>
<td>429</td>
<td>389</td>
</tr>
<tr>
<td>0.4</td>
<td>29.97</td>
<td>233</td>
<td>1.478</td>
<td>585</td>
<td>402</td>
</tr>
<tr>
<td>0.5</td>
<td>29.97</td>
<td>199</td>
<td>1.479</td>
<td>740</td>
<td>351</td>
</tr>
<tr>
<td>0.6</td>
<td>29.99</td>
<td>231</td>
<td>1.460</td>
<td>881</td>
<td>398</td>
</tr>
<tr>
<td>0.7</td>
<td>30.03</td>
<td>232</td>
<td>1.473</td>
<td>1031</td>
<td>401</td>
</tr>
</tbody>
</table>
roughness, $\varepsilon/D_h$. For conventional silicon microchannels, surface roughness is negligible (root mean squared roughness values of $\sim30\text{nm}$ for a typical silicon fabrication process (Lee et al., 2007)), however in this experiment, it was necessary to quantify the surface roughness for each channel sample as it was an unknown quantity. The roughness was measured along each channel surface, including over the tool paths, using a profilometer; the height in the measured surface roughness, $\varepsilon$, was approximately $0.5\mu\text{m}$ for all samples, yielding an average relative roughness value of $\varepsilon/D_h = 0.00128$. This is greater than the $\varepsilon/D_h < 0.00025$ threshold for smooth pipes and implies an earlier transition to turbulent flow (Kandlikar, 2012).

The laminar–to–turbulent model, proposed by Schmitt and Kandlikar (Kandlikar, 2014), was used to verify the critical Reynolds No. and is shown in equation 3.1:

$$Re_{D_h} = 2300 - 18.75 \times 10^3 \left( \frac{\varepsilon}{D_h} \right)$$  \hspace{1cm} (3.1)

The transition criterion was found to occur at a value of $Re_{D_h} \approx 2275$, indicating that the machined channels effectively conform to standard smooth microchannels.

### 3.1.2 Experimental Facility

The test facility, shown in figure 3.2, used a 36L reservoir (1) containing distilled and degassed water which was fed to the modular test section using a Tuthill 2VDC positive displacement gear pump (2) with a DDS.19 pump head. The system was filtered using a $7\mu\text{m}$ in-line filter (3) fitted downstream of the pump to prevent contamination of the channel sample (4). The test section was fitted with a Validyne DP15 pressure transducer (5) and equipped with a cross-over valve (6) to accommodate system priming prior to testing. K-type thermocouples (T) were placed at the inlet and exit of the test section and within the reservoir to record fluid temperature. A Bronkhorst Liqui-Flow 30 mass-flow meter was placed
3.1 Head-Loss Coefficient

downstream of the test section to record mass-flow rate (7). Pressure drop and
mass-flow rate data were recorded using a NIDAQ 6211 data acquisition card, and
temperature was recorded using a NIDAQ 9162 device.

Figure 3.2: Layout of experimental test facility with reservoir (1), gear pump (2),
filter (3), test section (4), pressure transducer (5), cross-over valve (6), and mass
flow meter (7). K-type thermocouples (T) are placed at the inlet and exit of the test
section and in the reservoir.

Figure 3.3 illustrates the test section used to measure the head loss coefficient
across the microchannel samples, with dimensions outlined. The test section was
clamped with a Polycarbonate cover plate (1) which had a viewing window and a
0.6mm thick gasket sheet placed between the cover plate and test section to prevent
leakage. Each sample (2) resided in a slot in the modular manifold test section
(3) to ensure consistent test conditions for each experiment. Finally, the pressure
ports (4) were located at 9mm either side of the test piece to record pressure drop.

A 3D numerical analysis of the exit manifold was conducted using Star-
CCM+v10.04. This confirmed the suitability of pressure port locations for accurate
measurement of the static pressure. A Reynolds No. of 1200 was chosen for the case,
and this was based on the channel hydraulic diameter. This value was chosen as it
was just outside the experimental envelope and therefore a suitable limit. As shown
in figure 3.4, the simulated velocity vector lines are parallel to the manifold wall at
3. EXPERIMENTATION

the pressure port location. This indicates that the static pressure measurement at this location is suitable for the Reynolds No. range tested in this experiment.

**Figure 3.3:** Schematic of the modular manifold test section with the associated design dimensions: (a) a cross-section of the modular manifold with cover plate (1), and test piece (2); (b) plan view of the exposed manifold block (3) with visible pressure port (4) locations 9mm from the entrance and exit of the test piece.

**Figure 3.4:** Centreline slice of the exit manifold geometry, showing a scalar image of the simulated local velocity fields with vectors highlighting flow direction. Vector arrows are parallel to the manifold wall at the pressure port location 9mm downstream of the channel exit.
3.1.3 Test Procedure

The test procedure for measuring head loss across the facility was as follows:

1. Each test sample was placed in the manifold block with the channel side face down.

2. The gasket was clamped in place with the cover plate and the system was primed for approximately 30 minutes prior to testing. At this time, all fluid lines, the pressure transducer and the mass-flow meter were bled to remove any air bubbles.

3. 5–minute intervals were allowed between measurements to ensure that the system reached steady state. Pressure drop and mass-flow rate were then recorded for 2 minutes at a sample rate of 100Hz.

4. This process was repeated until the maximum calibrated pressure drop (32.5kPa) was reached. Temperature was recorded at a rate of 2Hz over the same period.

3.1.4 Calibration Procedures

All instrumentation used for this experiment was calibrated using the following procedures. The differential pressure transducer was calibrated using a Martell T-140 pressure calibrator. Figure 3.5 shows the calibration chart for the transducer with an illustration of the calibration setup. The positive side of the transducer (1), calibrator readout (2), and the hand pump (3), were all connected via gas-tight plumbing connections. The pressure was increased in both the calibrator and the transducer to create a known pressure difference across each sensor. This created an equivalent pressure differential on each device as the negative pressure port of each instrument was open to atmosphere. The pressure difference was
3. EXPERIMENTATION

incrementally increased up to a value of 32.5kPa. A linear trend was evident between the differential pressure and output voltage. The uncertainties of the Martell calibrator and Validyne differential pressure transducer were ±10Pa and ±87.5Pa respectively. Although the uncertainty in the measurement is large at low differential pressures, a large sample size was recorded to reduce variance in the measurement. This yielded an adequate reading as shown by the high chi-square value, $R^2 = 1$.

![Calibration graph of the differential pressure transducer](image)

**Figure 3.5:** Calibration graph of the differential pressure transducer for a given voltage output with calibration correction equation (left) and a diagram of the calibration setup with transducer (1), Martell pressure calibrator (2) and hand-pump which generates pressure in the positive chamber of the transducer (right).

Figure 3.6 shows the thermocouple calibration chart and an illustration of the setup used. TJ-36 K-type thermocouples (1), purchased from Omega Engineering, were calibrated in a temperature-controlled water bath (2) which could maintain

---

1It is noted that calibration equation intersects y-axis in figure 3.5. This is due to manual adjustment of the instrument output voltage prior to calibration. This offset was fixed for the duration of the experimental series and accounts for the zero pressure drop at zero flow rate.
3.1 Head-Loss Coefficient

the water temperature to within $\pm 0.1^\circ C$, while measuring the temperature with a FLUKE 1502a thermometer (3). The thermometer probe was accurate to within $\pm 0.01^\circ C$ and was kept in close proximity to the thermocouples for an accurate estimate of water temperature. See appendix B.4 for calibration certificate. The thermocouples were calibrated over a temperature range of 15-30$^\circ C$. A linear behaviour was noted for the thermocouples on the graph illustrated in figure 3.6 with their calibration correction equation and chi-square values.

Figure 3.6: Thermocouple calibration chart (left) with calibration correction equations and chi-square fits. The thermocouples calibration setup (right) comprising a temperature controlled water bath (1), thermocouples (TC) (2), Fluke thermometer (3), and NIDAQ 9162 temperature recording device (4).

The Bronkhorst Liqui-Flow 30 mass-flow meter was calibrated over a range of 0-10kg/hr to an accuracy of $\pm 1\%$ F.S. using an ISO17025 accredited calibration centre. The calibration certificate is contained in appendix B.4.
3. EXPERIMENTATION

3.1.5 Data Reduction

The pressure drop across the arrangement shown in figure 3.3 with an open channel test section can be expressed as follows:

\[
\Delta P = \frac{1}{2}\rho U^2 \left( \frac{4f_{\text{app}}L}{D_h} + K_{\text{Losses}} + 2 \left( \frac{A}{A_{\text{man}}} \right)^2 \right)
\]  

(3.2)

The static pressure drop, \(\Delta P\), is equal to the product of the dynamic pressure drop and the frictional, reversible and irreversible losses due to the change in area at inlet and exit. The channel head loss term is a function of the apparent friction factor, \(f_{\text{app}}\), channel length, \(L\), and hydraulic diameter, \(D_h\). The irreversible inlet and exit losses of the channel, \(K_{\text{Losses}}\), are approximated to be 1.8 from empirical data (Kandlikar, 2014). The reversible losses due to the ratio of channel area, \(A\), to manifold area, \(A_{\text{man}}\), are assumed to be negligible in comparison. Therefore, they are omitted from this analysis.

For an open channel without a pillar, the channel pressure drop can be represented in the conventional form:

\[
\Delta P_{\text{Channel}} = \frac{2f_{\text{app}}ReU\mu L}{D_h^2}
\]  

(3.3)

There are a number of correlations available which describe the apparent frictional losses across a channel and, for this analysis, the correlation of Muzychka and Yovanovich (2009) is invoked due to its robustness and accuracy (±10%) for a range of non-circular channels. This correlation does not use the conventional length scale hydraulic diameter, \(D_h\), but the square root of the channel cross-sectional area, \(\sqrt{A}\). As the length scale is different, equation 3.3 is reformulated as follows:

\[
\Delta P_{\text{Channel}} = \frac{f_{\text{app}}Re\sqrt{\text{min}}P\mu L}{2\rho A^{5/2}}
\]  

(3.4)
3.1 Head-Loss Coefficient

The Reynolds No. is defined using the new length scale, $\sqrt{A}$, as:

$$Re\sqrt{A} = \sqrt{A} \frac{\dot{m}}{\mu A} \tag{3.5}$$

The correlation for defining the apparent friction factor (Muzychka and Yovanovich, 2009) $f_{app}Re\sqrt{A}$ takes the form:

$$f_{app}Re\sqrt{A} = \left[ \left( \frac{3.44}{\sqrt{Z^+}} \right)^2 + \left( \frac{12}{\sqrt{\epsilon(1+\epsilon)}} \left[ 1 - \frac{192\epsilon}{\pi^3} \tanh\left( \frac{\pi}{2} \epsilon \right) \right] \right)^2 \right]^{\frac{1}{2}} \tag{3.6}$$

Where the dimensionless length scale $Z^+$ is defined as:

$$Z^+ = \frac{L/\sqrt{A}}{Re\sqrt{A}} = \frac{\mu L}{\dot{m}} \tag{3.7}$$

When comparing the channels containing the pillar of variable width to a blank channel case, the total head loss can be represented as a superposition of the frictional and drag losses of an open channel and cylinder respectively (Idelchik, 1960). This approach is similar to a valve loss coefficient study by Waddell et al. (2015), which quantified the effective loss of a unique valve design within a channel. The pressure drop across the microchannel containing a pillar can now be represented as:

$$\Delta P_{\text{Channel}} = \frac{1}{2} \frac{\dot{m}^2}{\rho A^2} \left( \frac{f_{app}PL}{A} + \kappa \right) \tag{3.8}$$

Where the pillar-loss coefficient is represented by $\kappa$. This term quantifies the additional hydrodynamic drag losses incurred by the pillar within the channel configuration studied in this thesis and is presented in equation 2.4.
3. EXPERIMENTATION

3.1.6 Validation

The test facility was validated using an open channel sample of hydraulic diameter, $D_h = 409\mu m$. The apparent frictional losses of the channel, $f_{app}$, were calculated from equation 3.4, and compared to the correlation of Muzychka and Yovanovich (2009) in figure 3.7. The data was within the prediction bounds of the correlation used, with a maximum discrepancy of 9%, indicating that the facility was adequate for measuring the head loss coefficient.

Figure 3.7: Measured apparent friction factor, $f_{app}$, as a function of Reynolds No., $Re\sqrt{\frac{\alpha}{\mu}}$, compared with correlated values.
3.1.7 Hydrodynamic Uncertainty

The experimental uncertainty from each device used in the measurement of head loss has been outlined in subsection 3.1.2. Tables 3.2 lists variables measured in the experiment with associated accuracy and percentage uncertainty for the experimental range. The uncertainty values of pressure drop and mass-flow rate were the highest at the lowest mass-flow rate values, which reduce sharply for a corresponding increase in mass-flow rate. Uncertainty values in the primary variable were deemed to be acceptable for this experimental series.

Table 3.2: Uncertainties of each measured parameter for the head loss experiment.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Accuracy</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ΔP</td>
<td>Pressure drop, Pa</td>
<td>87.5 Pa</td>
<td>± 0.3–8.8</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass-flow rate, kg/hr</td>
<td>0.1 kg/hr</td>
<td>± 0.2–5.5</td>
</tr>
<tr>
<td>T</td>
<td>Temperature, K</td>
<td>0.1K</td>
<td>~± 0.0035</td>
</tr>
<tr>
<td>L</td>
<td>Length, m</td>
<td>10 μm</td>
<td>~± 0.03</td>
</tr>
<tr>
<td>h</td>
<td>Height, m</td>
<td>2 μm</td>
<td>± 1.4–1.6</td>
</tr>
<tr>
<td>w</td>
<td>Width, m</td>
<td>2 μm</td>
<td>~± 0.2</td>
</tr>
<tr>
<td>d</td>
<td>Diameter, m</td>
<td>2 μm</td>
<td>± 0.3–2.3</td>
</tr>
</tbody>
</table>

Table 3.3 lists the measured and derived parameters with their uncertainties. For this experiment, the magnitude of the uncertainties in pillar-loss coefficient, $\kappa$, Reynolds No., $\text{Re}\sqrt{\tau}$, and confinement ratio, $\beta$, were deemed acceptable, with larger variances observed at the lowest mass-flow rates.
3. EXPERIMENTATION

Table 3.3: Derived parameter uncertainties for the ranges considered in the head loss experiment.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Area, m$^2$</td>
<td>1.4–1.6</td>
</tr>
<tr>
<td>$P$</td>
<td>Perimeter, m</td>
<td>1.4–1.6</td>
</tr>
<tr>
<td>$D_h$</td>
<td>Hydraulic diameter, m</td>
<td>2.0 – 2.3</td>
</tr>
<tr>
<td>$f_{app}$</td>
<td>Apparent friction factor, (-)</td>
<td>1.8–8.0</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>Normalised head loss, (-)</td>
<td>3.4–14.6</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>Pillar-loss coefficient, (-)</td>
<td>2.0–14.1</td>
</tr>
<tr>
<td>$Re_{\sqrt{A}}$</td>
<td>Reynolds No., (-)</td>
<td>1.1–5.6</td>
</tr>
<tr>
<td>$Re_d$</td>
<td>Pillar Reynolds No., (-)</td>
<td>1.5–6.2</td>
</tr>
<tr>
<td>$Z^+$</td>
<td>Dimensionless length scale, (-)</td>
<td>0.2–5.5</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>Aspect ratio, (-)</td>
<td>1.4–1.6</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Confinement ratio, (-)</td>
<td>0.4–2.3</td>
</tr>
</tbody>
</table>
3.2 Heat Transfer Measurements

The following section details the manner in which local quasi-steady heat transfer measurements were conducted using the Joule heated foil technique. A description of this technique, the experimental rig, test procedure, and validation of benchmark case are described in the following subsections.

3.2.1 Joule Heated Foil Technique

The Joule heated foil technique was used to determine the downstream forced convective heat transfer contributions of the pillar configurations investigated in this thesis. This was accomplished by placing a thin metallic foil (thickness $\approx 12.5\mu m$) at the channel wall and supplying electrical power to produce heat – Joule heating. Local surface temperature of the foil was monitored with an IR camera. Measuring both the energy input and corresponding temperatures, the local surface heat transfer can be quantified. The thermal energy of the foil is dissipated through several heat transfer mechanisms, specifically conduction, radiation, free convection, and forced convection. The purpose of this technique is to quantify the forced convection heat transfer from the foil to the fluid. As such, an energy balance is required to quantify the magnitude of each heat transfer mode. Stafford et al. (2009) created a detailed approach to quantify the additional heat transfer modes in order to deduce the forced convection characteristics of surfaces adjacent to axial flow fans. Figure 3.8 illustrates the Joule heated foil configuration under investigation herein with an energy balance of an element that is representative of a single pixel on the IR camera sensor. Equation 3.9 presents the heat flux balance of the unit cell $A$ shown in figure 3.8, isolating the forced convective component, $q_{fc}$. See appendix B.1 for a full description of the derivation for equation 3.9 from the unit cell $A$ in figure 3.8.
3. EXPERIMENTATION

Figure 3.8: Illustration of the heated thin foil configuration used with a unit cell A displaying the different heat transfer modes through the unit. Modified from Stafford et al. (2009).

\[ q_{fc} = q_{gen} - q_r - q_c + (k_f t_f + k_p t_p + k_k t_k) \left( \frac{\partial^2 T_m}{\partial x^2} + \frac{\partial^2 T_m}{\partial y^2} \right) \]  

(3.9)

The forced convective heat flux, \( q_{fc} \), is the difference between the heat flux generated through Joule heating, \( q_{gen} \), and the radiative, \( q_r \), conductive, \( q_c \), and lateral conductive heat fluxes emanating from the heated foil. Heat flux produced by the Joule effect is expressed in equation 3.10 and is defined as a function of the voltage drop across the foil and current passing through it, divided by the area of unit cell A:

\[ q_{gen} = \frac{VI}{A_A} \]  

(3.10)
3.2 Heat Transfer Measurements

Heat flux losses via conduction and radiation are defined by equations 3.11 and 3.12:

\[
q_c = k \frac{\partial T}{t_{\text{glass}}} \tag{3.11}
\]

\[
q_r = \varepsilon_s \sigma (T_f^4 - T_\infty^4) \tag{3.12}
\]

Equation 3.11 defines the heat flux via conduction as a product of the thermal conductivity of the material, \(k\), and the temperature gradient normal to the heat flux area, \(\partial T / t_{\text{glass}}\). The heat flux via radiation through the IR crystal is expressed as a product of the surface emissivity, \(\varepsilon_s\), Stefan-Boltzmann constant, \((\sigma = 5.669 \times 10^{-8} \text{ W/m}^2\text{K}^4)\), and difference between the foil and ambient temperatures.

The effects of lateral heat conduction across the foil have been shown by Stafford et al. (2009) to be the most pronounced heat transfer mode, second only to the forced convection. It was shown by increasing the thickness from 14.3 to 41.7 \(\mu\)m, a normalisation of convective heat transfer coefficient of up to 32% was observed. As such, quantifying the heat flux vector between each pixel element in an IR image is critical to resolving a normalisation of the local temperature field due to the foil thickness, resulting in loss of information. The latter part of equation 3.9 calculates the heat flux across the foil; it is defined as the summation of the product of layer thickness, \(t\), and thermal conductivity, \(k\), of each layer times the Laplacian of the temperature field as a function of pixel width. This analysis will be used in subsection 3.2.8 to reduce the data obtained from the experiment and to validate the heat transfer information obtained against the theory.

### 3.2.2 Channel Geometries

Three cases were considered for the heat transfer characterisation experiment:

- Case I, influence of pillar width 1mm upstream of heated foil
3. EXPERIMENTATION

- Case II, influence of pillar width 8mm upstream of the heated foil
- Case III, influence of geometry and dual pillar configuration which were placed 1mm upstream of the heated foil

Figure 3.9 illustrates a sketch of the insert design with key dimensions outlined: channel height, H, channel width, W, pillar width, d, and distance of pillar from left hand side of the insert, L. An isometric view with its physical dimension is also presented. To change the channel geometries consistently, modular aluminium inserts were machined to fit into a polycarbonate holder as described in subsection 3.2.4 in figure 3.11.

The additional cases investigated were square, diamond and triangular pillars along with two twin pillar cases. Area averaged heat transfer and near surface PIV
measurements of Wang and Peles (2013) were conducted, however the cases studied in this thesis will demonstrate the heat transfer characteristics of the surface. This will add to the existing literature and assist practitioners in choosing better forms of flow augmentation to achieve higher heat transfer. This will add to the existing literature and in choosing a better form of flow augmentation for higher heat transfer. In order to compare the different shapes, the chord length of each shape was constrained to the circular case as shown in figure 3.9. In the two twin pillar cases, a spacing of approximately one third of channel width was taken, comparable to the study of Wang and Peles (2013). The measurements of all key dimensions studied are presented in table 3.4. A vernier calipers was used to measure all parameters in table 3.4 to an accuracy of ±0.01mm.

The critical Reynolds No., $Re_{crit}$, for maintaining a developed hydrodynamic flow profile was calculated. This is relevant when comparing the behaviour of the pillar configuration to the blank channel case; this is a Graetz flow problem and requires a fully developed hydrodynamic flow profile prior to the heated section of the channel.

### 3.2.3 Experimental Facility

An illustration of the experimental facility is shown in figure 3.10. The test section (1) was connected to a Tuthill gear pump (2) equipped with a DDS.19 gear pump head which fed degassed and filtered water to the test section. The mass flow rate was measured using a Bronkhorst Liqui-Flow 30 mass-flow meter (3). The mass-flow meter was fitted downstream of the test section as it heats the fluid to discern its mass-flow rate. The working fluid temperature was regulated to 20°C by a Lauda E100-R104 water bath (4) providing a controlled inlet temperature. K-Type thermocouples (T) were placed at the inlet and exit of the channel section to monitor fluid temperature. The power supplied to the heated foil, by a Sorensen
Figure 3.10: Heat transfer test facility: the test section (1) fitted with thermocouples (T) is connected to a gear pump (2) and temperature controlled water bath (4). The mass flow is measured by a mass-flow meter (3). The heated foil was powered by the power supply (5), where the voltage was measured using sense wires across the foil of known distance and the current was measured by a multimeter. Thermal images of the foil were acquired by the Merlin IR camera (6).
### Table 3.4: CASE I: Dimensions of circular pillar designs which were positioned 1mm upstream of the heated foil. CASE II: circular pillar designs positioned 8mm upstream of foil. CASE III: additional designs and blank channel used for benchmark case.

<table>
<thead>
<tr>
<th>CASE I</th>
<th>H (mm)</th>
<th>W (mm)</th>
<th>d (mm)</th>
<th>d2 (mm)</th>
<th>L (mm)</th>
<th>Re$_{crit}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>B1</td>
<td>0.98</td>
<td>6.20</td>
<td>0.61</td>
<td>-</td>
<td>55.22</td>
<td>642</td>
</tr>
<tr>
<td>B2</td>
<td>0.99</td>
<td>6.19</td>
<td>1.23</td>
<td>-</td>
<td>54.06</td>
<td>629</td>
</tr>
<tr>
<td>B3</td>
<td>0.98</td>
<td>6.15</td>
<td>1.86</td>
<td>-</td>
<td>53.64</td>
<td>624</td>
</tr>
<tr>
<td>B4</td>
<td>1.00</td>
<td>6.20</td>
<td>2.49</td>
<td>-</td>
<td>53.00</td>
<td>616</td>
</tr>
<tr>
<td>B5</td>
<td>1.00</td>
<td>6.20</td>
<td>3.10</td>
<td>-</td>
<td>52.87</td>
<td>615</td>
</tr>
<tr>
<td>B6</td>
<td>1.00</td>
<td>6.18</td>
<td>3.74</td>
<td>-</td>
<td>51.74</td>
<td>602</td>
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<tr>
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<td>6.13</td>
<td>4.41</td>
<td>-</td>
<td>51.13</td>
<td>595</td>
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<table>
<thead>
<tr>
<th>CASE II</th>
<th>H (mm)</th>
<th>W (mm)</th>
<th>d (mm)</th>
<th>d2 (mm)</th>
<th>L (mm)</th>
<th>Re$_{crit}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>B1</td>
<td>1.01</td>
<td>6.22</td>
<td>0.61</td>
<td>-</td>
<td>48.35</td>
<td>562</td>
</tr>
<tr>
<td>B2</td>
<td>1.02</td>
<td>6.25</td>
<td>1.23</td>
<td>-</td>
<td>47.78</td>
<td>556</td>
</tr>
<tr>
<td>B3</td>
<td>1.02</td>
<td>6.24</td>
<td>1.85</td>
<td>-</td>
<td>47.12</td>
<td>548</td>
</tr>
<tr>
<td>B4</td>
<td>1.00</td>
<td>6.22</td>
<td>2.48</td>
<td>-</td>
<td>46.51</td>
<td>541</td>
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<tr>
<td>B5</td>
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<td>-</td>
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<tr>
<td>B6</td>
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<td>-</td>
<td>45.15</td>
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<tr>
<td>B7</td>
<td>1.00</td>
<td>6.22</td>
<td>4.40</td>
<td>-</td>
<td>45.09</td>
<td>524</td>
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</table>

<table>
<thead>
<tr>
<th>CASE III</th>
<th>H (mm)</th>
<th>W (mm)</th>
<th>d (mm)</th>
<th>d2 (mm)</th>
<th>L (mm)</th>
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</tr>
</thead>
<tbody>
<tr>
<td>Circular</td>
<td>0.98</td>
<td>6.20</td>
<td>0.61</td>
<td>-</td>
<td>55.22</td>
<td>642</td>
</tr>
<tr>
<td>Triangular</td>
<td>1.02</td>
<td>6.20</td>
<td>0.62</td>
<td>-</td>
<td>55.43</td>
<td>644</td>
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<tr>
<td>Square</td>
<td>1.00</td>
<td>6.17</td>
<td>0.61</td>
<td>-</td>
<td>55.19</td>
<td>642</td>
</tr>
<tr>
<td>Diamond</td>
<td>1.03</td>
<td>6.20</td>
<td>0.61</td>
<td>-</td>
<td>55.00</td>
<td>640</td>
</tr>
<tr>
<td>Twin Circ.</td>
<td>1.03</td>
<td>6.20</td>
<td>0.62</td>
<td>1.87</td>
<td>55.76</td>
<td>648</td>
</tr>
<tr>
<td>Twin Tri.</td>
<td>1.01</td>
<td>6.21</td>
<td>0.63</td>
<td>1.85</td>
<td>55.36</td>
<td>644</td>
</tr>
<tr>
<td>Blank</td>
<td>1.0</td>
<td>6.25</td>
<td>-</td>
<td>-</td>
<td>65.00</td>
<td>-</td>
</tr>
</tbody>
</table>

DCS 80-37 (4), was quantified by measuring the voltage across and current flowing through the foil. Voltage was measured using 50µm diameter sense wires placed ∼7.0mm apart, across the centre of the foil. A Fluke 45 dual display multimeter, placed in line within the electrical circuit, was used to measure the current directly, see appendix B.4 for its calibration certificate. The voltage, ambient, inlet and exit temperature measurements were respectively acquired by NIDAI 6211 and 9171 cards. The thermal images were recorded using a ThermCam Merlin camera (6)
3. EXPERIMENTATION

with an InSb detector operating in the 3-5\(\mu\)m MWIR spectral range. ThermCAM Researcher Pro v2.8 software was interfaced with the camera using an I-Port frame grabber to acquire thermal images for each test.

3.2.4 Experimental Test-Section

Figure 3.11 presents exploded and cross-sectional views of the test section to evaluate the downstream heat transfer characteristics of the pillar configurations studied. The test section comprised a matt-black spray-painted Aluminium plate (1). Residing within the plate was a Nylon holder containing a rectangular IR glass. The IR glass used for this experiment was fabricated from a FLUKE CV400 95mm IR glass. It was polished with a grinding wheel with 1200 grade SiC sand paper to a 12x15mm rectangular window. The heated foil (2), of Inconel 600, was adhered to the IR glass using double-sided Kapton tape with a Silicone pressure-sensitive adhesive. The tape has IR transmissive properties, making emissivity calibration possible, while the foil remained flat against the glass. The calibration details are discussed in subsection 3.2.6. A black Acetal sandwich plate (3), which contained O-ring groves either side to seal the fluidic channel with \(\varnothing 1\)mm high temperature O-rings, was placed upon the Nylon holder and aligned with Hex bolts. The sandwich plate (4) contained machined slots for the fluidic channel and \(\varnothing 27\)mm 5VDC fans (3) which provided external cooling to the non-liquid cooled foil area to prevent melting or heat-induced mechanical deformation during the experimentation. The heated foil was electrically connected to an external power supply via Copper busbars (6), which were mounted to Acetal struts (5). The struts served as structural support and held the rig together when different channel configurations were switched out. A Polycarbonate modular insert (7) was used as the final part to form the fluidic channel. A machined slot in the modular
3.2 Heat Transfer Measurements

Figure 3.11: Exploded and cross-sectional views of heated foil test section with aluminium base plate (1), IR glass and heated foil (2). Acetal sandwich plate (4) was fitted with two high temperature O-rings and two 27mm fans (3). Acetal struts (5) held both sandwich and Aluminium plates together while Copper busbars (6) were mounted for electrical connection to the foil from the power supply. The modular insert (7) with channel inserts formed the fluidic channel (8), as shown in the cross sectional view, when assembled.

Insert allowed for quick change of channel inserts between experiments. Once the experimental test section was assembled, it formed the 6.25mm wide and 1mm deep fluidic channel (8).

3.2.5 Heated Foil Preparation

The heated foil material used for this experiment was Inconel® 600 heat resisting alloy purchased from Goodfellow. The foil was light tested prior to purchase to ensure no holes or defects were present in the batch. The nominal thickness of the
foil was 12.5µm. A single strip of width 10.25mm was cut from 150mm x 150mm sheet with the area of interest spray-painted matt black to ensure a high emissivity, low reflectivity surface for capturing the thermal images \( \varepsilon \approx 0.96 \) for matt black paints (Holman, 2010). For the purpose of this experiment, it is assumed that there is no air gap present between the foil and the paint layer; this assumption was validated experimentally by Nogueira et al. (2003) and Raghu and Philip (2006).

To ensure that neither liquid nor air would be trapped between the heated foil and IR glass during the image acquisition, a single strip of 12.25mm wide double-sided Kapton tape was applied perpendicularly to the foil length over the painted area. Kapton tape thickness was 100µm with a \( \sim 50 \)µm of Kapton and \( \sim 25 \)µm of pressure sensitive Silicone adhesive. To ensure that no air gap was present between the glass, Kapton and painted foil layers, the following procedure was used:

- IR Glass & Nylon holder were cleaned with Isopropyl alcohol (IPA) and Kimitek wipes. They were then allowed to air dry.

- The double-sided Kapton tape was applied face down along the channel length, with the other adhesive side being covered with protective backing. The tape was massaged repeatedly until no void was visible along the entire length.

- The protective backing was removed from the Kapton tape and the painted foil was positioned over the glass area.

- Protective backing was reapplied over the foil and tape. The foil area was massaged and inspected from the glass side with light to detect air gap defects. This was repeated until no defect was visible.

The Kapton tape used in this experiment was IR transmissive, which is accounted for by the \textit{in-situ} emissivity calibration described in section 3.2.6. Copper
sense wires of $\varnothing 50 \mu m$ were placed on the heated foil 7mm apart at either side of the channel area. Prior to assembly, each wire was held in place with Kapton tape. The foil length was measured as 10.25mm with a vernier calipers to an accuracy of $\pm 0.01 \text{mm}$. Figure 3.12 shows a prepared heated foil with the Copper sense wires in place prior to assembly, with a machinist ruler for scale. On completion of the experimental series, the foil was removed from the glass and cross-sectioned. Prior to cross-sectioning, the foil and tape were potted in epoxy resin and polished using a LabStruers polishing station with increasingly finer SiC sandpaper grades (800, 1200, 2400, 4000). A Hitachi 2000 Scanning Electron Microscope (SEM) was used to take images of the foil cross-section. Figure 3.13 shows the cross-sectional image. A custom Matlab code was used to measure the layer thickness from its profile. Table 3.5 presents the layer measurements for the paint and foil layers.

![Figure 3.12: An image of a prepared heated foil with sense wires. The length of the heated foil and sense wire distance are scaled using the machinist’s ruler as a reference in the picture frame.](image)

<table>
<thead>
<tr>
<th>Layer</th>
<th>Nominal, $\mu m$</th>
<th>Measured, $\mu m$</th>
<th>Std.Dev.</th>
<th>% Dev.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foil</td>
<td>12.5</td>
<td>14.0</td>
<td>1.65</td>
<td>+12 %</td>
</tr>
<tr>
<td>Paint</td>
<td>-</td>
<td>18.5</td>
<td>2.05</td>
<td>-</td>
</tr>
</tbody>
</table>
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Figure 3.13: SEM image of heated foil cross-section. Dashed lines indicate the paint layer thickness measured.

3.2.6 Calibration

The live IR image produced by the ThermCam software coupled with the lens configuration produced a cold spot in the centre of the camera’s field of view; this effect will be referred to as a “thermal vignette” and means that the displayed temperature field in not representative. To eliminate this effect, an in-situ emissivity calibration was performed by pumping fluid of known temperature through the test section. To create a known temperature condition, water was fed through the test section at the pump’s maximum flow rate to ensure that the temperature difference between inlet and exit was within their calibrated uncertainties (±0.1°C). The following procedure was used to calibrate the foil for a known temperature range:

1. The test facility was assembled with a blank channel test piece and primed with the working fluid.

2. The water bath maintained the fluid temperature through the test section at 20°C, with the bath’s fluid temperature being monitored by a Fluke 1502a thermometer.

3. The temperature drop was monitored across the test section while the fluid was being pumped until a temperature difference of less than 0.1°C was maintained.
4. A series of 1000 images of the heated foil at quiescence was recorded at a frequency of 50Hz, while the inlet and exit temperatures of the test were measured using a NIDAQ 9171.

5. Steps 2–4 were repeated for a series of fluid temperatures up to a maximum of 42°C, the maximum which could be achieved using the less than 0.1°C temperature difference condition across the test section.

The measured and observed temperatures of each pixel of the heated foil were correlated to each other using a 4th order polynomial scheme to achieve an accurate fit. This two-dimensional field of polynomial curve fits was used to correct the temperature field of the image for a known range of 20-42°C. Figure 3.14 illustrates the calibration concept with a graphical relationship between the measured and observed foil temperature of a single pixel on the IR camera sensor. A 4th order polynomial was also fitted through the calibration data set, and was chosen so as to interpret the local temperature with statistical confidence. The thermocouples used for this experiment were subjected to the same calibration procedure outlined in subsection 3.1.4 using the Fluke 1502a thermometer. Other instruments used in this experiment, such as the Fluke multimeter and Bronkhorst mass-flow meter, were calibrated by a test house; their certificates are shown in appendix B.4.

3.2.7 Experimental Procedure

This subsection describes the setup prior to testing and the procedural steps for the heat transfer experiment. The experimental setup and calibration steps are as follows:

a) The test facility was plumbed and tested for leaks by using the pump to prime the channel fluid lines at the highest flow rate the pump could produce.
3. EXPERIMENTATION

![Figure 3.14](image.png)

**Figure 3.14:** Calibration concept for emissivity correction. Left: Foil at quiescence being heated by incoming fluid of known temperature either side. Thermal vignette is present. Right: Graph showing the relationship between the observed pixel temperature and average fluid temperature. A 4th order polynomial is fitted through each data set.

b) The temperature-controlled water bath was set to an initial temperature of 20°C and the facility was allowed 60 minutes to cool to a set temperature.

c) The camera was set up to view the entire foil and IR glass area. The live image was focused until a sharp thermal image was visible.

Once the experiment was set up, the calibration and experimental procedural steps were completed as follows:

1. The emissivity calibration of the foil was carried out to correct for the non-uniformity of the observed temperature field.

2. The flow rate was adjusted to achieve a channel Reynolds No. of 100.

3. The foil was then powered on.

4. The facility was monitored and left idle until steady readings were observed of voltage drop, inlet/exit thermocouples and the live IR camera image.
5. The foil voltage drop, current, inlet/exit temperatures were simultaneously recorded while the IR camera captured 1000 images of the heated foil at a frequency of 50Hz.

6. Steps 4-5 were repeated by increasing the Reynolds No. in steps of 50 to a maximum of 800.

7. The test vehicle was disassembled and reassembled with a different channel insert. Steps 2–7 were repeated for all channel inserts.

### 3.2.8 Data Reduction

The experimental facility was benchmarked using a blank channel and compared to conventional channel flow theory. In this case, a Graetz flow configuration is assumed to exist within the channel. Graetz flow is defined as a fully developed hydrodynamic flow profile in an unheated region entering a heated region generating a developing thermal boundary layer (Holman, 2010). To validate the experimental results, the area-averaged Nusselt No. ($\overline{\text{Nu}}(x^*)$), plotted as a function of inverse Graetz No. ($x^*$), is compared to the theoretical prediction of Shah for a Graetz flow problem (Bejan, 2013). These dimensionless numbers are defined in equation 3.13 and 3.14 (Bejan, 2013):

$$\overline{\text{Nu}}(x^*) = \frac{\dot{q}_{fc}D_h}{(T_{foil} - T_{bm})k}$$

(3.13)

$$x^* = \frac{L}{D_hRe_DhPr}$$

(3.14)

The variables $\dot{q}_{fc}$, $D_h$, $T_{foil}$, $T_{bm}$, $k$, $L$, $Re_Dh$ and $Pr$, represent respectively forced convective heat flux, hydraulic diameter, area-averaged foil temperature, bulk mean temperature, thermal conductivity, foil length, Reynolds No. based on hydraulic diameter and Prandtl No. The bulk mean temperature is used to adjust for the
increasing bulk fluid temperature as the foil is cooled by the fluid. It is described
by equation 3.15, and the variables $T_{in}$, $Q_{fc}$, $\dot{m}$, and $C_p$, are respectively inlet fluid
temperature, power supplied to the foil, mass-flow rate, and specific heat capacity
at constant pressure:

$$T_{bm} = T_{in} + \frac{Q_{fc}}{\dot{m}C_p}$$  \hspace{1cm} (3.15)

$T_{bm}$ is fixed for the experimental series as the power input does not change. Therefore
increases in heat transfer are proportional to the reduction in foil temperature.

For the benchmark case, it is important to consider the thermal entrance length
for this configuration. Equation 3.16 describes the foil length required to reach a
thermally developed flow:

$$X_T \sim 0.1D_h Re Pr$$  \hspace{1cm} (3.16)

The shortest thermal entrance length is for the $Re = 100$ case, and is calculated
to be $\sim 127$mm for $D_h = 1.72$mm and Prandtl No., $Pr = 6.78$. This states that
the heated foil region of length 10.25mm only experiences the first 8.27\% of the
thermal boundary layer growth. For this experiment, the flow is wholly thermally
developing. As such, the Nusselt No. approximation proposed by Shah (Bejan,
2013) for thermally developing Hagen-Poiseuille flow can be compared to the
experimental with equation 3.17:

$$Nu(x^*) = 1.302x^{*^{-1/3}}$$  \hspace{1cm} (3.17)

The theory discussed in this subsection described the convection heat transfer
mechanics and how it can be used to benchmark the experiment. As heat is also lost
through conduction and radiation, it is therefore prudent to identify the relative
magnitude of each heat transfer mode in the experiment. To this end, the following
subsection presents a thermal resistance network of each heat transfer path in the
experiment.
3.2.9 Heat Flow Path Assessment

To quantify all thermal losses from the heated foil, a thermal resistance network analysis was carried out on single cell of the heated-thin foil, presented in figure 3.8. It is important to identify the magnitude of each loss, with one effective measure being the thermal resistance \( R_{th} \) of each loss mode. This analysis is applied to the heated foil experiment herein to understand the dominant heat transfer modes present. This differs from the previous analysis of Stafford et al. (2009) in subsection 3.2.1 as each mode of heat transfer is decoupled from each other and only requires the material and geometric information to quantify the magnitude of each resistance path. Figure 3.15 illustrates a single cell of the heated thin foil, paint layer, Kapton tape, and IR Glass, with a resistance network of the cell considering all modes of thermal transport from the foil layer.

![Thermal Resistance Network](image)

**Figure 3.15:** Single cell of the heated-thin-foil, paint layer, Kapton tape and IR glass (left) with a thermal resistance network of the cell (right).

\( R_{th} = \frac{\Delta T}{Q} \) (Holman, 2010).
3. EXPERIMENTATION

\[
R_{\text{cond.}} = \frac{t}{k\Delta X \Delta Y}, \quad R_{\text{lat. cond.}} = \frac{\Delta X}{kt\Delta Y}, \\
R_{\text{conv.}} = \frac{1}{h\Delta X \Delta Y}, \quad R_{\text{rad}} = \frac{1}{h_{\text{rad}}\Delta X \Delta Y}
\] (3.18)

Equations 3.18 describe the thermal resistances for each loss mode in figure 3.15, which are respectively one-dimensional conduction, lateral conduction, forced convection, natural convection, and radiation. Variables \( t, k, \Delta X, \Delta Y, h, \) and \( h_{\text{rad}} \), are defined respectively as the thickness, thermal conductivity, pixel width, pixel breadth, heat transfer coefficient in natural convection and heat transfer coefficient in radiation. Thermal radiation in this model is only considered from the paint side to ambient air, and not the forced convection side, as the transmission of thermal radiation through the water is assumed to be negligible. The value of \( h_{\text{rad}} \) is taken from empirical data from Holman (2010) for radiation heat transfer from a black body to a large enclosure. For the temperature ranges measured for this experimental series (293-315K), the range of radiative heat transfer coefficients were 5.5 - 7 W/m\(^2\)K. To account for natural convection, a conservative heat transfer coefficient value of 8W/m\(^2\)K was chosen (Holman, 2010). Table 3.6 shows the thermal resistances for each heat transfer mode. For these calculations, the pixel dimension is constant for lateral directions: \( \Delta X = \Delta Y = 43\mu\text{m} \).

**Table 3.6:** Thermal resistance values of the network under assessment.

<table>
<thead>
<tr>
<th>Layer</th>
<th>Resistance (K/W)</th>
<th>Layer</th>
<th>Resistance (K/W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( R_{f,\text{conv}} )</td>
<td>2.7x10^5</td>
<td>( R_{\text{Kapton}} )</td>
<td>1.18x10^5</td>
</tr>
<tr>
<td>( R_{\text{foil}} )</td>
<td>605</td>
<td>( R_{\text{lat.Kapton}} )</td>
<td>2.17x10^4</td>
</tr>
<tr>
<td>( R_{\text{lat.foil}} )</td>
<td>5.95x10^4</td>
<td>( R_{\text{Glass}} )</td>
<td>1.35x10^5</td>
</tr>
<tr>
<td>( R_{\text{paint}} )</td>
<td>7.25x10^4</td>
<td>( R_{\text{lat.Glass}} )</td>
<td>15.0</td>
</tr>
<tr>
<td>( R_{\text{lat.paint}} )</td>
<td>3.9x10^4</td>
<td>( R_{\text{rad}} )</td>
<td>7.7x10^7</td>
</tr>
<tr>
<td>( R_{n,\text{conv}} )</td>
<td>6.8x10^8</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The thermal resistance associated with the main thermal paths are presented in table 3.6. The foil, paint and Kapton layers have very low conductive resistance,
and their lateral resistances are of the same order of magnitude as the forced convection component. The IR glass resistances too are low, while the external resistances due to natural convection and radiation are very large compared to the forced convection component; hence, the majority of thermal energy is assumed to transfer into fluid. The glass and additional layer resistances serve to normalise the local temperature field due to the gradient which exists along the foil. The previous analysis in section 3.2.1, which considers the temperature normalisation through the foil thickness, will be extended upon in the following subsection to account for internal lateral fluxes within the glass substrate. The next subsection describes how the measured temperature fields are used in conjunction with a numerical finite element model to calculate the internal heat conduction within the IR glass.

3.2.10 Correction Technique

The following subsection describes a correction technique used to calculate the internal normalising heat fluxes across the convection-cooled heated foil. Figure 3.16 illustrates how normalisation through conduction within the IR glass affects the measured convective heat transfer behaviour.

![Figure 3.16](image)

Figure 3.16: Graphical illustrations show the normalisation due to heat conduction with theoretical graphs of heat transfer coefficient and temperature over foil length. The physical behaviour is in line with the theory: the temperature and heat flux profiles change shape due to the normalisation.

As the flow is fully developed in nature, the convective heat transfer is assumed to
3. EXPERIMENTATION

follow a power-law decay profile along the foil length. The conductive coupling of
the foil to the glass and lateral conduction within the glass, indicates a normalisation
of the measured temperature profile along the foil. An apparently non-uniform heat-
flux field now exists due to transverse conduction through the glass. Numerically,
this behaviour – and associated normalisation – can be described by Laplace
equation for heat conduction, equation 3.19:

\[ \dot{q} = k \Delta T \]  

(3.19)

Temperature fields recorded through IR measurement, in conjunction with
material details and geometry, can be used to calculate internal heat flux fields.
In this subsection, a numerical representation of the experimental configuration –
simplified to 2D – is used to compare the heat flux along the foil for the full model
and a equivalent model which only uses the temperature field to generate the heat
flux field. Two simulations are presented in figure 3.17; the first one presents a
simulation with perfect thermal contact between the Kapton tape and IR glass
(top right), while the other applies a thin resistive layer of air to partially insulate
the IR glass from the heated foil.

Comsol Multiphysics v5.0 with a conjugate heat transfer solver was used to
simulate the 2D geometry with the boundary conditions presented in figure 3.17.
The boundary conditions for this case were chosen based on the thermal resistance
values presented previously in table 3.6:

1. A fully developed velocity profile of Reynolds No. 100 of hydraulic diameter
   1.72mm and fluid temperature 293.15K, was assigned at the entrance of the
   fluid (water).

2. The top side of the fluid, leading and trailing edges of the IR glass had
3.2 Heat Transfer Measurements

Figure 3.17: Two-dimensional simulation of the heated foil experiment. (Top left) Topology of experiment with 1mm fluid (water), 4mm IR glass, with assigned boundary conditions. (Top right) Temperature field, with internal heat flux vectors (magenta) illustrating the heat flow path within the IR glass. (Bottom centre) Temperature field with heat flux vectors (magenta) where there is a thermally insulating layer between Kapton and IR glass.

adiabatic conditions (/\), while a natural convection boundary condition was applied at the bottom of the IR glass for a nominal laboratory temperature of 298.15K.

3. A uniform heat flux of 30kW/m² was applied to the foil.

4. Material properties for water, Polycarbonate, Inconel foil, Acrylic paint, Kapton tape, Spinel IR window, and air, were taken from the software’s material database.

The temperature maps of the insulated case show that the heat is being convected into the fluid, highlighting the role of the air gap in insulating the heat spreading
3. EXPERIMENTATION

effect of the IR glass – this assumption holds true for similar heated foil experiments performed by other authors (Jeffers, 2009; Kearney, 2009; Waddell, 2015). For this simulation, an air gap of thickness 10µm was applied as the thermally resistive layer. The latter simulation shows the effect of normalisation, as the heat flux vectors (magenta) migrate towards the leading edge of the foil where the thermal boundary layer is not fully developed and, as a consequence, the heat transfer coefficient is higher. A lesser portion of the heat flux migrates towards the trailing edge, due to the difference between the high temperature of trailing foil edge and lower mean fluid temperature beyond the trailing edge. This simulation shows that the apparent heat flux field is not uniform along the foil length.

To benchmark the experiment to theory, the assumption of applying a constant heat flux field across the foil must be redefined, assuming good thermal contact between Kapton and IR glass. A constant heat flux boundary condition across the foil is not representative; it is effectively a summation of the constant heat flux supplied by Joule heating and the heat flux associated with the lateral conduction of heat along the foil. The latter heat flux field can be calculated using exclusively the temperature field. To demonstrate this, a third simulation applying the normalised temperature profile along the Kapton/fluid & paint/Kapton interface, was completed. Boundary conditions for IR glass were the same as the conjugate heat transfer model, as illustrated in figure 3.18.

The heat conduction in solids module within Comsol Multiphysics was used for this simulation. The heat flux field entering the paint and Kapton layer interface of both simulations are compared in figure 3.18. The two cases are effectively equal with deviations of less than 0.5% \(^1\), which implies that the apparent heat flux field for the experiment can be calculated using the temperature field of the foil coupled with the Joule effect. This technique can be extended to the whole

\(^1\)It is presumed that the deviation for the three dimensional case will be larger, however figure 3.19 shows this deviation for the experimental case.
3.2 Heat Transfer Measurements

3D geometry by using the full experimentally measured temperature field. This measured temperature field includes the foil area and channel areas before and after the foil. This region is used in the simulations with all emissivities calibrated for a temperature range of 20–40°C. The calibration procedure has been discussed previously in subsection 3.2.6.

![Figure 3.18: Comparison between the heat flux field leaving the paint layer into the IR glass for the full model and the temperature model. This shows that by knowing the temperature field, the apparent heat flux entering the paint layer can be calculated.](image)
3. EXPERIMENTATION

3.2.11 Validation

Figure 3.19 presents a dimensionless comparison of the corrected and uncorrected data sets in terms of local heat transfer (Nusselt No., Nu(x*)) as a function of thermal development length (x*) for a Reynolds No. of 100.

![Figure 3.19: Comparison between the corrected and uncorrected data using the numerical technique for a channel Reynolds No. of 100. Data sets are compared to theoretical predictions with uncertainty bars of ±10%. Shah’s theoretical Graetz flow solution is used (Nu(x*) = 1.302x*^{-0.333}) and taken from Bejan (2013).](image)

To benchmark the experimental technique, presented in subsection 3.2.10, an analytical solution for Graetz flow proposed by Shah was used for comparison as it closely represents the physics involved (Bejan, 2013). This theory is discussed in detail in the following subsection 3.2.8. The correction technique reduces the discrepancy in the experimental data, which is postulated to be due to the unac-
3.2 Heat Transfer Measurements

counted normalisation heat flux across the foil. The slope of the corrected data has a better agreement with theory, with largest deviations seen at the leading and trailing edges. The simulation in the top right of figure 3.17 shows the heat travelling into the glass and back into the fluid at the leading and trailing edges of the foil. The magnitude of this discrepancy was within the experimental uncertainty, which is presented later in subsection 3.2.12: \( \text{Nu}(x^*) \) and \( x^* \) uncertainties were 9.3 and 8.3 % respectively.

Figure 3.20 presents the corrected experimental area-averaged Nusselt No. as a

![Figure 3.20](image)

**Figure 3.20:** Non-dimensional log-log plot of the foil mean Nusselt No. \( (\bar{\text{Nu}}(x^*)) \) as a function of thermal development length \( (x^*) \) for a Reynolds No. range: 100-783. Data is compared to thermally developing flow equation (Bejan, 2013) with maximum data deviations of \( \pm 9\% \).
function of the inverse Graetz No. for a Reynolds No. range of 100 to 798, and also the theoretical prediction of Shah (Bejan, 2013). The experimental data shows good agreement with theory, with maximum discrepancies of approximately ±9%. The trend shows that the theory over-predicts the Nusselt No. with increasing Reynolds No. This deviation can be explained by the unaccounted heat under the O-rings and outside of the IR camera’s field of view. When the Reynolds No. increases, the foil lowers in temperature creating a greater potential for heat to conduct into the glass and into the leading and trailing foil edges. This addition of heat would lower the Nusselt No. in these locations and reduce the area averaged value and deviation from theory. It was postulated that a change in the foils resistivity could be a source of error; however, for the temperature range considered (20-190°C), the change in the resistivity of the foil (1%) is insufficient to make a difference. The presented data set gives confidence in the behaviour of the experimental facility to provide a good benchmark case when comparing the behaviour of this flow with the augmented flow configurations under investigation in this thesis.

3.2.12 Heat Transfer Uncertainty

Table 3.7 describes each variable with their respective accuracy and uncertainty values for the heat transfer experiment. The dominant source of uncertainty in this list is mass-flow rate, and it is largest at the smallest value of mass-flow rate measured in the experiment. Table 3.8 contains the uncertainties for all derived values in the same experiment. Notable sources of uncertainty are temperature difference and Reynolds No.; these uncertainties stem from the calculation of bulk mean temperature and measurement of the mass-flow rate. The magnitude of these uncertainty percentages are deemed to be acceptable for the ranges considered for the heat transfer experimentation.
Table 3.7: Uncertainties of each measured parameter for the heat transfer experiment.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Accuracy</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>V</td>
<td>Voltage drop, V</td>
<td>0.005 V</td>
<td>±1.2</td>
</tr>
<tr>
<td>I</td>
<td>Current, A</td>
<td>0.01 A</td>
<td>±0.2</td>
</tr>
<tr>
<td>L&lt;sub&gt;foil&lt;/sub&gt;</td>
<td>Foil length, m</td>
<td>10µm</td>
<td>±0.1</td>
</tr>
<tr>
<td>H</td>
<td>Channel height, m</td>
<td>10µm</td>
<td>±0.1-1.02</td>
</tr>
<tr>
<td>W</td>
<td>Channel width, m</td>
<td>10µm</td>
<td>±0.2</td>
</tr>
<tr>
<td>L&lt;sub&gt;Vol&lt;/sub&gt;</td>
<td>Sense wire distance, m</td>
<td>10µm</td>
<td>±0.1</td>
</tr>
<tr>
<td>(\dot{m})</td>
<td>Mass-flow rate, kg/hr</td>
<td>0.1 kg/hr</td>
<td>±1.2-7.9</td>
</tr>
<tr>
<td>d</td>
<td>Pillar diameter, m</td>
<td>10µm</td>
<td>±0.3-1.6</td>
</tr>
<tr>
<td>T&lt;sub&gt;foil&lt;/sub&gt;</td>
<td>Local foil temperature, K</td>
<td>0.14K</td>
<td>±0.2-0.4</td>
</tr>
<tr>
<td>T&lt;sub&gt;in&lt;/sub&gt;</td>
<td>Inlet temperature, K</td>
<td>0.1K</td>
<td>±0.5</td>
</tr>
<tr>
<td>T&lt;sub&gt;out&lt;/sub&gt;</td>
<td>Outlet temperature, K</td>
<td>0.1K</td>
<td>±0.5</td>
</tr>
</tbody>
</table>

Table 3.8: Derived parameter uncertainties for the ranges considered in the heat transfer experiment.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Uncertainty(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A&lt;sub&gt;channel&lt;/sub&gt;</td>
<td>Channel area, m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>± 1.0</td>
</tr>
<tr>
<td>A&lt;sub&gt;foil&lt;/sub&gt;</td>
<td>Foil area, m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>± 0.2</td>
</tr>
<tr>
<td>P</td>
<td>Channel perimeter, m</td>
<td>±1.0</td>
</tr>
<tr>
<td>D&lt;sub&gt;h&lt;/sub&gt;</td>
<td>Hydraulic diameter, m</td>
<td>±1.4-1.5</td>
</tr>
<tr>
<td>Re&lt;sub&gt;Dh&lt;/sub&gt;</td>
<td>Reynolds No., (-)</td>
<td>±2.1-8.1</td>
</tr>
<tr>
<td>Re&lt;sub&gt;d&lt;/sub&gt;</td>
<td>Pillar Reynolds No., (-)</td>
<td>±1.6-8.2</td>
</tr>
<tr>
<td>(\Delta T)</td>
<td>Temperature difference, K</td>
<td>±0.8-4.3</td>
</tr>
<tr>
<td>Q</td>
<td>Power, W</td>
<td>±1.17</td>
</tr>
<tr>
<td>(\dot{Q})</td>
<td>Heat flux, W/m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>±1.2</td>
</tr>
<tr>
<td>T&lt;sub&gt;bm&lt;/sub&gt;</td>
<td>Bulk mean temperature, K</td>
<td>±1.7-8.0</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient, W/m&lt;sup&gt;2&lt;/sup&gt;K</td>
<td>±2.4-9.2</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt No., (-)</td>
<td>±2.74-9.3</td>
</tr>
<tr>
<td>x*</td>
<td>Thermal development length, m</td>
<td>±2.5-8.3</td>
</tr>
</tbody>
</table>

3.3 Closure

This chapter presented the experimental methods used to characterise the
head loss and downstream wall heat-transfer of channels containing pillars for a
3. EXPERIMENTATION

confinement ratio, $\beta$, of 10-70%. Geometric measurements of the test pieces are presented with a description of the experimental facilities, in which the test pieces are characterised. Data reduction methods for the primary experimental measurements are presented, coupled with validated benchmark for each. A correction technique applied to calculate the apparent heat flux field in the heat transfer experiment was explained. Finally, a detailed uncertainty analysis of the primary and derived values used for each experimental method was presented.

The following chapter analyses the head loss measurement of channels containing centrally located circular pillars for confinement ratios, $\beta$, of 10–70%. Data reduction methods presented in this chapter 3 are used to isolate the pillar losses from the total head loss measured. A regression analysis of the experimental data is performed to describe the behaviour of the head loss coefficient of the channel configuration and the isolated pillar loss.
Chapter 4

Hydrodynamic Measurements

This chapter presents the hydrodynamic characterisation of the measured pillar configurations detailed in chapter 3. The head loss behaviours of the pillar configurations are compared to an open channel case, and the trends of each are discussed. A regression analysis of the frictional contributions was performed to collapse the behaviour across the range of Reynolds No. under consideration. The role of length scale necessary to collapse the data sets is shown with a regression analysis for each scale. The chapter concludes with a summary of the impact of these findings for microchannels containing the circular pillar configurations for downstream flow augmentation. The following section presents and discusses the hydrodynamic performance of the channels tested. The non-dimensionalised head loss characteristics for each configuration are shown and discussed.

4.1 Pressure-Drop Behaviour

The channel pressure-drop as a function of mass-flow rate curves are presented in figure 4.1 for a confinement ratio, $\beta$, range of 10–70%. It is evident from the graph that each curve follows a non-linear trend with increasing mass-flow rate.
4. HYDRODYNAMIC MEASUREMENTS

Figure 4.1: Pressure-drop–mass-flow rate curves for each channel containing a circular pillar with a confinement ratio range, $\beta$, of 10–70%. The data set from the open channel benchmark case is also presented. Additional annotations above curve indication order of each curve as they appear from left to right.

A non-proportionate increase in pressure drop for a corresponding increase in confinement ratio, $\beta$, is evident. In particular, for the $\beta = 0.3$ & 0.4 and $\beta = 0.5$ & 0.6 curves respectively, an overlap with each data set for the mass-flow rate range is observed. This is due to the difference in the physical channel sizes used in the experiment, explaining the non-proportionate increase between each curve. The geometric measurements for each channel and pillar geometry, which were presented previously in table 3.1, can be used to correct for scaling differences between channels. The respective change in head loss can be presented in terms
4.1 Pressure-Drop Behaviour

of Poiseuille No. \((f_{\text{app}}Re)\), which represents the total frictional increase of each channel (Idelchik, 1960; White, 2011). Figure 4.2 presents the normalised head loss coefficient, \(\Phi\), as a function of Reynolds No. for all confinement ratios considered.

The normalised loss term, \(\Phi\), is defined as the ratio of the measured channel head loss to that of an open channel of equivalent geometry\(^1\). The normalised head loss relationship is defined in equation 4.1.

\[
\Phi = \frac{f_{\text{app}}Re\beta}{f_{\text{ann}}Re_{\text{Channel}}}
\]  

Figure 4.2: Normalised head loss for various confinement ratios as a function of the channel Reynolds No.

\(^1\)The friction factor model of Muzychka and Yovanovich (2009) was used to calculate the theoretical Poiseuille No. of each channel and is presented previously in section 3.1.5 equation 3.6.
A power-law decay is evident for confinement ratios up to $\beta = 0.5$. This is due to the frictional losses of the channel and drag loss of the pillars (Idelchik, 1960). At lower Reynolds No., the effect of confinement is not dominant but it becomes more apparent with increasing Reynolds No., as the hydrodynamic losses due to the expansion and contraction at the pillar location become larger. A proportionate increase in head loss is evident for an increase in confinement ratio. At the higher values of confinement ratio, a distinct increase is noticeable beyond a Reynolds No. of $\sim 400\text{–}560$ for $\beta = 0.6$ and $0.7$. This change in head loss indicates a change in the flow behaviour downstream of the pillar from laminar to unsteady – or even quasi-unsteady – flow due to the presence of the pillar, which results in an increase in the turbulent kinetic energy as shown by Jung et al. (2012). A sharp increase in the turbulent kinetic energy was observed beyond a pillar Reynolds No. of 400. This change in flow regime occurs at higher $Re_d$ due to increasing confinement, as seen by other authors in figures 2.4 and 2.5. A similar change has been observed by Renfer et al. (2013), where a populated staggered pin-fin array was seen to undergo a transition from laminar to oscillatory flow due to the pin-fin arrangement, at low channel Reynolds No., $Re \geq 200$. This change in flow regime creates an increase in pressure drop, above a critical Reynolds No. A similar change in flow regime occurs in this arrangement, increasing head loss at higher confinement ratios. The circular pillars studied generate different downstream flow structures, similar to a circular pillar in cross-flow (Idelchik, 1960; Schlichting, 2016; Zdravkovich, 1997a). This flow structure is identified by the pillar Reynolds No., $Re_d$: $Re_d = \rho \bar{U}_{ch} d/\mu$.

Plotting the Reynolds No. based on the pillar diameter against the channel Reynolds No., $Re_{\sqrt{\pi}}$, can highlight regions of flow transition that correspond to the behaviour in head loss coefficient. As this configuration behaves similarly to a cylinder in cross flow, vortex shedding exists downstream of the cylinder beyond a threshold Reynolds No., $Re_d$. Figure 4.3 plots the corresponding Reynolds No., with
illustrations of where the flow transitions to different downstream flow structures at their respective Reynolds No. Work by Sahin and Owens (2004) visually shows the level of mixing with an increase in confinement in figure 2.7:

![Diagram showing flow regimes](image)

**Figure 4.3:** Illustration of downstream flow regimes for an unconfined pillar for a range of Reynolds No. Adapted from Idelchik (1960). Plot of Reynolds No. based on square root of channel cross-sectional area, $Re_{\sqrt{A}}$, versus the Reynolds No. based on the pillar diameter for each channel case.

As shown previously in figure 4.2, the head loss coefficient begins to increase
beyond $\text{Re}_\sqrt{A} \simeq 500$ for $\beta = 0.6 \& 0.7$. In figure 4.3, these corresponding curves show that the flow tends towards unsteady flow very rapidly, given the low slope. Conversely, the opposite is true for the lower confinement ratios, with a corresponding increase in mass-flow rate generating different downstream flow structures. This highlights the type of hydrodynamic loss experienced in each case: a combination of frictional loss by restarting the boundary layer downstream of the pillar (Schlichting, 2016), and the increasing inertial loss due to the downstream unsteady and oscillatory behaviour associated with Von Kármán vortex shedding. Figure 4.3 illustrates the flow regimes downstream of an unconfined pillar for a range of Reynolds No., and represents the typical flow fields experienced by the channels studied in this thesis. Micro-Scale PIV experiments on similar channel configurations show comparative flow transition behaviours (Jung et al., 2012; Wang et al., 2013). Furthermore, the shedding frequency of the vortices downstream of the pillar ($f = Sr U_{ch}/d$) is proportional to the incoming fluid velocity, and this frequency continues to rise for a corresponding increase in mass-flow rate. The inertial portion of the head loss becomes the more dominant contribution to be total head loss in the channel for high confinement ratios.

4.2 Influence of Pillar Loss

The overall hydrodynamic behaviour of the channels containing the cylindrical pillars, with postulations as to the cause of their behaviour, was presented in section 4.1. The total measured head loss coefficient can be described as a superposition of drag loss and frictional loss (Idelchik, 1960). Furthermore, with adequate knowledge of channel geometry through geometric measurement and reliable correlations of friction factor, the contribution of the pillar to the overall head loss coefficient can
be isolated and characterised. This yields important information when designing a channel with a pillar for flow augmentation.

Figure 4.4 presents the isolated pillar-loss coefficient, $\kappa$, for all confinement ratios and Reynolds No. investigated. The term, $\kappa$, previously shown in equation 3.8, is derived from the difference between the measured head loss and frictional loss of an open channel of equivalent dimension. This frictional loss term was calculated using the apparent friction factor correlation, defined in equation 3.6, for the measured geometry of each channel. The pillar-loss Reynolds No. curves in figure 4.4 follow a power-law decay similar to figure 4.2. A general trend appears, showing that the slope for each changes at different Reynolds No. and that this change, or inflection point, in each curve occurs at a lower Reynolds No. for greater
values of confinement ratio, $\beta$. As previously discussed, the pillar Reynolds No. dictates the flow regime present in the channel, and the change in slope of curves is due to the change in flow regime downstream of the pillar.

To predict the behaviour of this arrangement, the pillar loss, $\kappa$, was curve-fitted to the form of $\kappa = aRe^b + c$ for each confinement ratio, $\beta$. The equation parameters are compiled in table 4.1, with the high coefficient of determination, $R^2 \simeq 0.999$, and low root mean squared error (RMSE) revealing a good fit to the data. The parameters, $a \& b$, chosen for the curve fit vary weakly with the confinement ratio, $\beta$. The pillar loss is strongly influenced by the confinement ratio, and it exhibits a non-linear relationship, evident in the trend of the $c$-parameter in table 4.1. The $c$ parameter in this curve-fit equation captures the inflection point in each curve and represents the change in flow regime downstream of the pillar.

**Table 4.1:** Correlated values for pillar-loss coefficient, $\kappa$, for each confinement ratio, based on channel Reynolds No., $Re\sqrt{A}$.

<table>
<thead>
<tr>
<th>$\beta$</th>
<th>Parameters</th>
<th>Statistics</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$a$</td>
<td>$b$</td>
</tr>
<tr>
<td>0.1</td>
<td>1.54x10^5</td>
<td>-1.655</td>
</tr>
<tr>
<td>0.2</td>
<td>1.00x10^5</td>
<td>-1.556</td>
</tr>
<tr>
<td>0.3</td>
<td>1.04x10^5</td>
<td>-1.570</td>
</tr>
<tr>
<td>0.4</td>
<td>1.10x10^5</td>
<td>-1.577</td>
</tr>
<tr>
<td>0.5</td>
<td>1.03x10^5</td>
<td>-1.553</td>
</tr>
<tr>
<td>0.6</td>
<td>1.04x10^5</td>
<td>-1.558</td>
</tr>
<tr>
<td>0.7</td>
<td>1.07x10^5</td>
<td>-1.550</td>
</tr>
</tbody>
</table>

To further collapse this data, a surface fit analysis of the confinement ratio, $\beta$, and Reynolds No. was performed. Reynolds No. based on the square root of channel area, $\sqrt{A}$, square root of open area ratio, $\sqrt{A(1-\beta)}$, and pillar diameter, $d$, were investigated. The length scale $\sqrt{A}$ was chosen, as it was used in the correlation developed by Muzychka and Yovanovich (2009), which collapsed friction factor of a channel for a range of channel cross-section types. The open-area ratio, $A(1-\beta)$, is
4.2 Influence of Pillar Loss

an inclusive geometric term which uses both the channel cross-section information and the pillar information; as the data suggests, the head loss behaviour is the product of both the channel frictional loss and the downstream flow behaviour of the pillar. Finally, the pillar diameter length scale, $d$, was chosen as it is possible that the pillar head loss is exclusively related to the pillar diameter. This is similar to the conclusions of Waddell et al. (2014), who studied curved obstructions and collapsed the head loss contributions using the obstruction width.

A surface fit analysis of the form, $\kappa = aRe^b\beta^c$, using the length scales previously discussed, was carried out in order to correlate the three variables. The surface fit based on the channel Reynolds No., $Re\sqrt{A}$, showed poor agreement as an average deviation of 41% was observed, with standard deviation, maximum and minimum values of 71%, 348%, and -12% respectively. The square root of the open area, $\sqrt{A(1-\beta)}$, was also examined and showed poor agreement with large under predictions at high Reynolds No. The poor agreement could be due to the use of inappropriate length scales, $\sqrt{A}$ and $\sqrt{A(1-\beta)}$, hence choosing a more appropriate length scale is necessary. Using the Reynolds No. based on the diameter of the pillar would be more appropriate as the flow structures generated are from its presence in the channel. As such, the pillar-loss coefficient is plotted against the Reynolds No. based on pillar diameter, $Re_d$, in figure 4.5a. The data curves in figure 4.5a show similar trends to figure 4.4: a power-law decay is evident with inflection points that increase with the confinement ratio; this transition begins at approximately $Re_d \approx 400-500$. This inflection point corresponds to an increase in heat transfer at the wall downstream which is presented in chapter 5. Superimposing the typical flow structures, that would be expected of an unconfined cylinder for this Reynolds No. range, Von Kármán vortex streaks would be present with the vortices growing in strength with an increase in Reynolds No. (Idelchik, 1960; Zdravkovich, 1997a).
4. HYDRODYNAMIC MEASUREMENTS

Figure 4.5: (a) Pillar-loss coefficient, $\kappa$, as a function of pillar diameter Reynolds No., $Re_d$. Dashed lines (—) indicate typical range of flow transition points of an unconfined cylinder. (b) Collapsed pillar loss coefficient, $\kappa$, as a function of $Re_d$, where dashed lines (—) indicate flow transition point of an unconfined cylinder in cross-flow.
4.2 Influence of Pillar Loss

In a similar form to table 4.1, table 4.2 plots the correlation parameters based on the newly defined length scale, \( Re_d \), for each confinement ratio of the form; 
\[ \kappa = aRe^b + c. \]
High coefficient of determination, \( R^2 \) and RMSE values were observed, matching the data. A similar trend is seen for both the \( b \) and \( c \) parameters while the \( a \) parameter increases in a non-linear fashion: the change in this parameter value with Reynolds No. represents the change in pillar diameter for each curve.

### Table 4.2: Correlated values for pillar-loss coefficient, \( \kappa \), for each confinement ratio, based on the pillar diameter Reynolds No., \( Re_d \).

<table>
<thead>
<tr>
<th>( \beta )</th>
<th>Parameters</th>
<th>Statistics</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( a )</td>
<td>( b )</td>
</tr>
<tr>
<td>0.1</td>
<td>1.85E4</td>
<td>-1.71</td>
</tr>
<tr>
<td>0.2</td>
<td>3.30E4</td>
<td>-1.54</td>
</tr>
<tr>
<td>0.3</td>
<td>6.95E4</td>
<td>-1.57</td>
</tr>
<tr>
<td>0.4</td>
<td>1.05E5</td>
<td>-1.56</td>
</tr>
<tr>
<td>0.5</td>
<td>1.63E5</td>
<td>-1.53</td>
</tr>
<tr>
<td>0.6</td>
<td>1.96E5</td>
<td>-1.56</td>
</tr>
<tr>
<td>0.7</td>
<td>2.45E5</td>
<td>-1.54</td>
</tr>
</tbody>
</table>

Finally, a surface-fit was performed on the same data set of the form \( \kappa = aRe_d^b \beta^c \) and is based on the cylinder diameter Reynolds No., \( Re_d \). This is plotted in a collapsed form, \( \kappa/\beta^c = aRe_d^b \), and the equation in figure 4.5b, reveals parameter values, \( a = 4 \times 10^5 \) and \( b = c = 1.5 \). The correlation was found to collapse the data well and predict the experimental data up to a Reynolds No., \( Re_d \), of 425 within \( \pm 20\% \) accuracy. It is evident that the correlation does not quite fully collapse the effect of confinement ratio, \( \beta \), beyond this Reynolds No. A limitation of the surface-fit correlation presented in figure 4.5b is that it does not take into account the channel width term in the Reynolds No. used, \( Re_d \). As a result, the correlation presented is specific to the flow configurations studied. Further experimentation would be
4. HYDRODYNAMIC MEASUREMENTS

needed at higher channel Reynolds No., and a wider range of channel aspect ratios to modify and extend the correlation presented here: specifically, identifying the role of pillar aspect ratio (d/h), which was inadvertently varied in this study, and would have been expected to have a damping effect on the downstream flow for high aspect ratio channels. However, for the purpose of this thesis, the channel aspect ratios and range of Reynolds No., that were correlated, capture the practical operation range of a microfluidic cooling system for thermal management of PIC packages.

4.3 Closure

This chapter presented the hydrodynamic behaviour of a range of high-aspect ratio microchannels containing a centrally placed circular cylinder of confinement ratios 10–70% over a wide laminar flow Reynolds No. range. A regression analysis of the data set comparing the head loss behaviour to an unconfined cylinder in cross-flow revealed the following:

1. The head loss coefficient showed an increase of 109-317% over an open channel of equivalent geometry. The head loss followed a power-law decay relationship as a function of the channel Reynolds No., \( \text{Re}\sqrt{A} \), with each channel decaying to a near constant, which increased with confinement ratio, \( \beta \).

2. Higher confinement ratios showed an increase in head loss beyond a critical Reynolds No. This increase was due to a change in flow regime. For a fixed channel Reynolds No., the pillar Reynolds No, \( \text{Re}_d \), increases with pillar diameter, \( d \). Unsteady vortex shedding occurs downstream of the pillar at lower Reynolds No. for larger pillar diameters, increasing the inertial losses observed in the head loss coefficient measurement. The frequency of
this vortex shedding flow regime increases with the channel Reynolds No., explaining the further increase in head loss coefficient.

3. The regression analysis of the isolated pillar-loss coefficient was correlated to the form: \( \kappa = aRe^b \sqrt{\frac{A}{\beta}} + c \). Good agreement was observed and the obtained parameters provided a useful tool for predicting the head loss coefficients of channels containing circular pillars of confinement ratios 10–70%.

4. Three length scales were investigated to collapse the pillar head loss, \( \kappa \), Reynolds No., \( Re \), and confinement ratio, \( \beta \): square root of channel area, \( \sqrt{A} \), square root of open area, \( \sqrt{A(1-\beta)} \), and pillar diameter, \( d \). It was found that the pillar diameter, \( d \), was the most suitable, and it collapsed the experimental data to the form \( \kappa = 4 \times 10^5 (\beta/Re_d)^{1.5} \), up to a Reynolds No. of 425 to an accuracy of \( \pm 20\% \).

Using microchannels containing single pillars is an advantageous design choice not only to augment heat transfer, but also as electrical conduits to connect optical and electrical planes in a single PIC package. Understanding and characterising the hydrodynamic performance of the pillar–channel configurations presented in the chapter are important for the pump design of such a microfluidic cooling system. The following chapter will present the downstream wall heat-transfer characteristics for pillars of the same confinement ratio range for different downstream locations.
4. HYDRODYNAMIC MEASUREMENTS
Chapter 5

Heat Transfer Measurements

This chapter presents the downstream wall heat transfer characteristics of the measured pillar configurations detailed in chapter 3. Three cases of heat transfer augmentation are presented: the first case investigates the role of confinement from 10-70% of the channel width for a pillar placed 1mm upstream of the heated area; the second case investigates the same confinement ratio with the pillar placed 8mm upstream; and the final case investigates the role of singular and twin pillars of width 10% of channel width placed 1mm upstream of the heated area. The nominal Reynolds No., $Re_{Dh}$, range for all three cases is 100–800. The non-dimensional heat transfer enhancement, in terms of Nusselt No., of each case is presented and compared to open channel measurements. Local contour plots obtained from the heat transfer experiment described in subsection 3.2 are presented and show the thermally enhanced regions with their fluidic behaviour correlated to that of a unconfined pillar in cross flow. The following subsection presents the heat transfer enhancement of a pillar placed 1mm upstream of the heated region.
5. HEAT TRANSFER MEASUREMENTS

5.1 CASE I: 1mm

The first experimental case addressed the downstream heat transfer characteristics of pillars with confinement ratio, $\beta$, of 10–70%, placed 1mm upstream of the heated foil area. Figure 5.1 presents the area-averaged heat transfer coefficient, $\bar{h}$, as a function of the mass-flow rate, $\dot{m}$, for the series of confinement ratios investigated. The illustration in figure 5.1 represents the experimental configuration.

![Figure 5.1: Downstream area averaged heat transfer coefficient, $\bar{h}$, curves of the pillar arrangements for confinement ratios, $\beta = 10–70\%$, as a function of channel mass-flow rate, $\dot{m}$. Pillar location is 1mm upstream of the heated-foil area.](image)

As the confinement ratio increases, an initial enhancement in heat transfer
coefficient is seen, but then diminishes at $\beta = 0.3$. The decrease in heat transfer coefficient is incremental until $\beta = 0.5$, where a clear change in slope is evident. Similar trends are seen in figure 5.2 where heat transfer coefficient, $\bar{h}$, and mass-flow rate, $\dot{m}$, are respectively non-dimensionalised into Nusselt No., $Nu_{Dh}$, and Reynolds No., $Re_{Dh}$.

![Figure 5.2: Downstream area averaged Nusselt No., $Nu_{Dh}$, as a function of Reynolds No., $Re_{Dh}$, for confinement ratios, $\beta$, of 10-70%. Pillar location is 1mm upstream of the heated foil area.](image)

The heat transfer performance of this flow configuration highlights the role of the wake formation downstream of the pillar which occurs over the heated foil area. In the laminar flow regime ($Re_d < 200$), the wake behind the cylinder continues to grow until it reaches a critical flow rate for vortex shedding to occur (Zdravkovich, 91)
The wake then begins to reduce in length and the recirculation/wake region has a higher turbulent kinetic energy than the steady wake region, yielding a sharp increase in heat transfer (Jung et al., 2012). The correlation between the strength of TKE and increase in heat transfer has been shown by Wang et al. (2013), with visual examples of the fluidic mixing present in figure 2.7 by Sahin and Owens (2004). As shown in subsection 4.2, a sharp increase in head loss was observed beyond a critical pillar diameter of 0.6 and Reynolds No., $Re_{Dh}$, of 400. This implies that the increase in both head loss and heat transfer is an artefact of the shedding vortices from the pillar.

A confinement ratio of 0.7 has the widest range, and shows the lowest and highest heat transfer enhancement. At a Reynolds No., $Re_{Dh}$ of 400, the corresponding $Re_d$ is 975. At this value of $Re_d$, the wake region begins to decrease with the vortex shedding becoming more turbulent in nature (Idelchik, 1960; Wang et al., 2013). As the confinement ratio decreases from 0.7 to 0.5, this transition point moves to a higher channel Reynolds No., $350 < Re_{Dh} < 450$. As the confinement ratio increases, the inflection in the measured heat transfer coefficient occurs at a lower Reynolds No. and has a steeper slope. This is a result of the pillar Reynolds No. range increasing with pillar diameter, $d$, showing that by increasing the confinement ratio, vortex shedding occurs sooner. The confinement ratio has been shown to delay the onset of vortex shedding (Zdravkovich, 1997b), however the increase of confinement does not delay the increase in heat transfer.

Figure 5.3 presents a normalised local Nusselt No. contour map of three confinement ratios ($\beta = 0.2, 0.5, & 0.7$) at three Reynolds No. ($301 > Re_{Dh} > 804$). The local Nusselt No. values are normalised by the maximum Nusselt No., $Nu_{max}$, of each contour respectively with their value presented in the figure caption. At low confinement ratios and Reynolds No., the accelerated flow passed the pillar (1) dominates the heat transfer enhancement. As the Reynolds No. increases to
the onset of vortex shedding, the shedding eddies (2) are present and are now the dominating feature of the flow. Increasing the confinement yields a larger wake and region of recirculating flow over the heated foil (3). This contributes to the poor heat transfer performance over the heated foil area. As the confinement ratio increases to 70%, the accelerated flow becomes more apparent at low Reynolds No. (4). This accelerated flow causes the TKE to increase with shedding vortices occupying the full width of the channel (Rosenhead and Schwabe, 1930; Sahin and Owens, 2004). This increases the observed heat transfer over the heated foil area.

Figure 5.3: Downstream area averaged heat transfer coefficient contours of the pillar arrangements for confinement ratios, $\beta = 10\text{-}70\%$, as a function of channel mass-flow rate, $\dot{m}$. Contour plots are normalised by $\text{Nu}_{\text{max}} = (a) 15.1$ (b) 23.5 (c) 30.3 (d) 12.1 (e) 20.7 (f) 28.1 (g) 11.7 (h) 15.8 (i) 23.7.
Figure 5.4 compares the downstream heat transfer enhancement created from each confinement ratio. Each case is normalised by the a regression model of the open channel case. A 4th order polynomial was fitted to the open channel case data, yielding high R² and low values of 0.9997 and 0.018 respectively. Each case was then normalised at their respective Reynolds No. A confinement ratio of $\beta = 0.7$ yielded the highest and lowest heat transfer enhancement, with a maximum range of 69–157%. As shown by the contour map in figure 5.3, the growth and decay of the wake with increasing Reynolds No. is the primary reason for the enhancement measured. As the confinement ratio decreases, the breakdown of the wake is delayed due to the physical size of the pillar within the channel. Also, the minimum enhancement of each case decreases with a reduction in the confinement ratio. This shows that the dead volume of the wake region appears to be the primary contribution to the minimum enhancement measured as seen in figure 5.3 (3). To obtain an increase in heat transfer from the range of confinement ratios examined with a proximity of 1mm upstream of the heated region, a confinement ratio of $\beta > 0.5$ and a channel Reynolds No. of $Re_{Dh} > 400-625$ would be required.

The magnitude in heat transfer enhancement, shown previously, indicates that wider pillars provide greater heat transfer than those with smaller confinement. It is important, however, to compare the ratio of heat transfer enhancement to the frictional penalty incurred by the flow. This influences the pumping power required to remove heat from the device and, furthermore, indicates the effectiveness of the system. Figure 5.5 presents the quotient of the measured area averaged Nusselt No., $Nu_{Dh}$, and the head loss coefficient of a channel of similar geometries. The characteristics for hydrodynamic loss, $(\zeta = 4f_{app}L/D_h + \kappa)$, are derived from the experimental results presented in chapter 4. The frictional penalty incurred by the pillars outweighs the heat transfer gains. All curves tend towards unity as the
Figure 5.4: Normalised Nusselt No., $Nu_{Dh}^*$, as a function of Reynolds No, $Re_{Dh}$, for confinement ratios, $\beta$, of 10-70%. Pillar location is 1mm upstream of the heated foil area. Each curve is normalised by the open channel case with a confinement ratio of $\beta = 0$. 
5. HEAT TRANSFER MEASUREMENTS

Reynolds No. increases. Narrow pillars are shown to be more effective than pillars with larger confinement ratios. Furthermore, the gains in heat transfer from the larger confinement ratios begin to plateau at a confinement of 0.7 or greater.

The following subsection presents the heat transfer characteristics of the second case. This identifies the influence of placing the pillar further upstream to 8mm for the same confinement ratios and Reynolds No. range.

Figure 5.5: Ratio of Nusselt No., $ Nu_{Dh} $, enhancement to frictional penalty, $ \zeta $, as a function of Reynolds No., $ Re_{Dh} $, for confinement ratios of 10–70%. Curves are compared to conventional channel flow which equates to unity.
5.2 CASE II: 8mm

The second experimental case investigates the same configuration as section 5.1, but with a change of the pillar location from 1mm to 8mm upstream of the heated foil area. Figure 5.6 presents the area-averaged heat transfer coefficient, $\bar{h}$, as a function of mass-flow rate, $\dot{m}$ for confinement ratios, $\beta$, of 10–70%. The illustration in figure 5.6 represents the experimental configuration.

Figure 5.6: Downstream area averaged heat transfer coefficient, $\bar{h}$, curves of the pillar arrangements for confinement ratios, $\beta = 10-70\%$, as a function of channel mass-flow rate, $\dot{m}$. Pillar location is 8mm upstream of the heated-foil area.
At low mass-flow rates ($< 1.25 g/s$), the heat transfer coefficient decreases with confinement ratio, $\beta$. As the shape of the curve also changes with increasing confinement ratio, an S-shape or inflection point begins to appear with small variations in the heat transfer coefficient value for $\beta = 0.3-0.4$. This trend is enhanced at higher confinement ratios, and it is a result of the flow changing from a steady separated flow to an unsteady flow with vortex shedding (Idelchik, 1960; Schlichting, 2016; Zdravkovich, 1997b). An improvement of heat transfer coefficient compared to the open channel only begins to appear at the higher mass-flow rates and confinement ratios of 0.5-0.7. Furthermore, the inflection point for the $\beta$ range of 0.5–0.7 becomes more pronounced when the confinement increases and begins at lower mass-flow rates, as shown in each data curve. The confinement ratio range of 0.5–0.7 shows a greater growth in heat transfer coefficient over the lower confinement ratios. Similar trends are seen in figure 5.7, where the data is non-dimensionalised in terms of Nusselt No., $Nu_{Dh}$, and Reynolds No., $Re_{Dh}$.

All confinement ratios showed higher heat transfer enhancement over the open channel case beyond a Reynolds No. of 600, showing that the disturbed flow downstream of the pillar yielded superior heat transfer to the open channel case. At Reynolds No. below this value, a decrease in heat transfer enhancement is seen. This is due to the competing role of the wake and the pillar distance upstream of the heated area. The accelerated flow and shedding vortices generated passed the pillar are further upstream and have more time to decay than the previously presented 1mm case (Jung et al., 2012). Furthermore, the length of the wake past the pillar is longer for larger confinement ratios, and now has a larger portion of the heated foil area. The Von Kármán vortex street, which is the expected flow phenomenon for such a flow configuration, features less flow acceleration at the cylinder edges this far downstream, contributing to the lower heat transfer.
enhancement. The maximum heat transfer enhancement is less than the 1mm case for a higher confinement ratio, again due to the distance of the pillar upstream of the heated foil area. Although the wake breaks down and becomes more turbulent with increased Reynolds No., the flow has time to decay before it reaches the heated area, reducing the measured enhancement (Jung et al., 2012). An illustration of the fluidic mixing as shown in figure 2.7 highlights the rapid decay in downstream mixing from confinement ratios, $\beta \leq 0.7$ Sahin and Owens (2004).

Local Nusselt No. contour maps over the foil area are presented in figure 5.8 for confinement ratios 0.2, 0.5 & 0.7 and channel Reynolds No, $Re_{Dh}$, 300, 560, & 800.
5. HEAT TRANSFER MEASUREMENTS

Each contour plot is normalised by the maximum Nusselt No. of their respective plot with their value presented in the figure caption. At low confinement ratios and Reynolds No., the accelerated flow from the cylinder and gap between cylinders form a “W-shaped” heat transfer profile over the heated foil region (1). The wake region is not present this far downstream and, as the Reynolds No. increases, the vortex shedding from the pillar becomes the dominating feature in the contour map (2). As the confinement ratio increases, the accelerated flow between the pillar and the walls shows regions of higher heat transfer forming a “B-shaped” profile on the contour (3). As the Reynolds No. increases, the wake increases and becomes evident in contour map (e); it also and breaks down into a region of increased heat transfer (4) in contour map (f). At the highest confinement ratio, the recirculating wake region dominates the heated region (g) but breaks down sooner than for the lower confinement ratios (h), where the accelerated flow from the pillar and shedding vortices create a large uniform region of heat transfer (5). The placement of the cylinder is critical to maximising the heat transfer enhancement. Placing wider pillars close to the heated region will maximise the heat transfer enhancement of the wake breakdown at a flow rate which corresponds to the breakdown of the wake region generated by the pillar. This is advantageous from a device integration point of view, as closer spacing allows for more device modules to be placed on a single plane.

The effective heat transfer enhancement is presented in figure 5.9 in terms of normalised Nusselt No., $Nu^*_{Dr}$, and channel Reynolds No., $Re_{Dr}$, for each case. Each curve is normalised by a regression model of the open channel case with the same regression model used for the 1mm case presented in section 5.1. The range of heat transfer enhancement was -65–115%, with the highest confinement ratio $\beta = 0.7$ encompassing this range. In comparison with the 1mm case, placing pillars this far upstream reduces the overall heat transfer enhancement, showing that the
breakdown in the wake and turbulence generated by this change in flow behaviour decays too rapidly to increase heat transfer.

As previously discussed in subsection 5.1, comparing the heat transfer enhancement to the head loss can illustrate the effectiveness of a thermal management system. Figure 5.10 presents the measured area averaged Nusselt No., $N_u_{Dh}$, as a function of the channel head loss, $\zeta$, for an equivalent geometry. The head loss
Figure 5.9: Normalised Nusselt No., $Nu_{Dh}^*$ as a function of Reynolds No., $Re_{Dh}$, for confinement ratios, $\beta$, of 10-70%. Pillar location is 1mm upstream of the heated foil area. Each curve is normalised by the open channel case with a confinement ratio of $\beta = 0$. 
Figure 5.10: Ratio of Nusselt no., $N_u D_h$, enhancement to frictional penalty, $\zeta$, as a function of Reynolds No., $Re_D h$, for confinement ratios of 10–70%. Curves are compared to conventional channel flow which equates to unity.
outweighs the heat transfer gains, however it shows that the curve exhibits greater heat transfer enhancement to frictional loss as the Reynolds No. increases.

5.3 CASE III: Shapes 1mm

The third case investigated the role of pillar profile and twin pillar dividing the channel into three segments for a nominal Reynolds No., $Re_{Dh}$, range of 100-800 with pillar location being 1mm upstream of the heated foil area. Pillar shape profiles such as circular, triangular, rhombus, and square of cord diameter 10% of channel width, were investigated. This dimension translates to a confinement ratio, $\beta$, of 0.1. The geometry of each shape was previously presented in subsection 3.2.2. Two twin pillar configurations of triangular and circular shapes were also investigated.

Figure 5.11 presents the area-averaged heat transfer coefficient as a function of mass-flow rate for the series of pillar configurations. The family of curves is compared to the open channel case for comparison. The following observations can be made:

- At low mass-flow rates, the twin circular pillar and square pillar channels show respectively the most and least enhancement in heat transfer coefficient.

- As the mass-flow rates increase, all pillar configurations are equivalent to, or outperform, the open channel case.

- The square and rhombus pillar shapes show the best performance of the single pillar profiles, while the triangular pillar shape shows the least enhancement.

- The twin circular pillars show the largest enhancement of the two twin pillar configurations presented and the greatest enhancement overall.
5.3 CASE III: Shapes 1mm

Figure 5.11: Downstream area averaged heat transfer coefficient, $\bar{h}$, curves for various pillar profiles as a function of mass-flow rate, $\dot{m}$. Pillars are at a confinement ratio of 10% and placed 1mm upstream of heated area.

In this experiment, only two twin pillar shapes were considered, with the difference in performance between each shape consistent to that of the single pillar shape profiles. To further increase heat transfer enhancement, it is postulated that a twin rhombus shaped pillar configuration would yield a higher enhancement. This is because the twin vortex streets generated by each pillar are believed to be present over the heated region yielding a larger area to disturb the boundary layers developing on the channel walls. At the same time, they do not disturb each other due to the spacing between them (Zdravkovich, 1997b) – increasing the convective heat transfer characteristics. Similar trends are seen in the non-dimensional form.
of Nusselt No., $Nu_{Dh}$, and Reynolds No., $Re_{Dh}$, shown in figure 5.12. Comparing the best heat transfer performance in all three cases undertaken in this experiment, the twin cylinder configuration shows consistent enhancement across the Reynolds No. range considered but does not yield the highest performance. The maximum heat transfer between the $\beta = 0.7$ at 8mm upstream and twin circular case 1mm upstream are of the same magnitude ($Nu = 8.8$ and 8.9 respectively) with the $\beta = 0.7$ at 1mm upstream having a 38.2–39.7% higher Nusselt No. than the other cases. Twin cylinder arrangements produce an adequate increase in heat transfer
at lower Reynolds No., especially in the higher confinement ratio cases where vortex shedding does not occur at low flow rates. The head loss or heat transfer enhancement in each case must be considered if pumping power or heat removal are the limiting factors, respectively.

Normalised forms of each shape configuration are presented in figure 5.13, showing the greatest enhancement seen by the twin circular pillars with the least performance presented by the single cylinder. The enhancement range for these pillar configurations are 96–114%.

**Figure 5.13:** Normalised Nusselt No., $Nu_{Dh}^*$, as a function of Reynolds No., $Re_{Dh}$, for confinement ratios, $\beta$, of 10-70%. Pillar location is 1mm upstream of the heated foil area. Each curve is normalised by the open channel case with a confinement ratio of $\beta = 0$. 

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Figure 5.14 presents contour plots of the local Nusselt No. normalised by the maximum Nusselt No. in the heated region. Three Reynolds No. cases are present: 300, 550 and 800. In the single pillar cases, the shape of the wake is different in each case. For the singular pillars at low flow rates, the wake behind the pillar has little influence over the heated region. The space between the pillar and walls creates accelerated flow past the pillar which decays along the foil length (1). At higher flow rates, the shedding vortices from the pillar are evident on the foil (2). The rhombus pillar has a wider wake region as indicated by the area of low heat transfer downstream of the pillar (3). The appearance of vortex streets occurs at lower Reynolds No. (4) for the square pillar and the wake width appears to be smaller (5). The twin pillar shapes showed similar contour profiles with the circular pillars showing longer streets for similar Reynolds No. The cylinder and rhombus shapes show the largest wake regions, while the triangular and square show the smallest wake regions. This is postulated to be due to their respective aerodynamic shapes, characterised by their drag coefficients (Holman, 2010), which are displayed in table 2.3. This can be used to quantify how effective their wake is on heat transfer enhancement.

The thermal-hydrodynamic effectiveness of the pillar configurations has not been presented in this subsection as no hydrodynamic measurements were taken. The measurement of the drag coefficient is a good indicator of the head loss performance of a pillar, however, values for each shape studied in this experiment are detailed in table 2.3. The drag coefficients for the triangular, square and rhombus shapes are greater than the circular shape by respectively 33.3, 75, and 33.3 %. The thermal-hydrodynamic effectiveness of such pillar configurations would not exceed the open channel case. Similar trends are shown in figures 5.5 and 5.10. Furthermore, the difference in drag coefficient value outweighs the increase in heat transfer for the range studied.
Figure 5.14: Contour plots of each pillar configuration examined. Contour plots are normalised by $\text{Nu}_{\text{max}} = (a) 14.1 (b) 18.8 (c) 24.6 (d) 15.2 (e) 21.2 (f) 25.9 (g) 17.9 (h) 25.5 (i) 32.0 (j) 15.8 (k) 22.9 (l) 34.6 (m) 16.6 (n) 22.9 (o) 27.14 (p) 16.6 (q) 22.8 (r) 27.0.
5. HEAT TRANSFER MEASUREMENTS

5.4 Closure

This chapter presented the heat transfer characteristics of three experimental cases examining the role of confinement ratio, the distance upstream of the heated region, the role of pillar shape, and twin pillar configurations at fixed spacing. This was performed over a channel Reynolds No., $Re_{Dh}$, of 100-800. The following points were concluded from this experiment:

1. Increasing confinement ratio induced lower heat transfer enhancements at lower Reynolds No. due to the presence of an increasing wake area over the heated region. Beyond a critical Reynolds No., the heat transfer enhancement increased dramatically; this critical Reynolds No. correlates well with the pillar Reynolds No., $Re_d$, indicating that the wake region behind the cylinder is becoming increasingly turbulent.

2. Increasing the distance of the pillar upstream of the heat region from 1mm to 8mm reduced heat transfer enhancement measured over the heated region. Although an equivalent flow structure is present for each Reynolds No., $Re_{Db}$, the distance downstream is far enough for the TKE generated to decay sufficiently to decrease heat transfer enhancement. From a photonic integration and heat transfer perspective, it is an advantage to have the pillar close to the heated region to enhance heat transfer.

3. A rhombus pillar placed 1mm upstream with cord diameter of 10% of channel width, showed the highest heat transfer enhancement in comparison with the other pillar shapes. This heat transfer enhancement over the open channel case was consistent for the Reynolds No. range, and it outperformed the larger single circular pillars below their critical Reynolds No. The twin pillar configuration showed an even greater enhancement over the rhombus shape;
this was due to more areas of the perturbed and accelerated flow over the heat region. Again, this remained consistent over the range of Reynolds No., indicating that shape and number of pillars are important design parameters if greater cooling performance is required for low Reynolds No. flows.

4. In all cases, the hydrodynamic penalty outweighed the heat transfer enhancement in comparison to an open channel. Lower confinement ratios were found to perform better in terms of hydrodynamic penalty incurred in the channel.

The following chapter will present conclusions based on the experimental data and analyses presented in chapter 4 and chapter 5. This is concluded with a list of recommendations of future experiments and analyses that would further complement the body of work presented in this thesis.
Chapter 6

Conclusions and Recommendations

6.1 Conclusions

The hydrodynamic and thermal characteristics of a pillar within a channel have been studied for its use application as an electrical conduit in a TIPS system. Parameters such as width of pillar, distance upstream of the heat source, and shape were considered in the experiments. Micro/mini-scale experimental rigs were commissioned and benchmarked against open channels, with existing theoretical and analytical models, to characterise the head loss and area averaged heat transfer characteristics. The following conclusions are drawn from the hydrodynamic and heat transfer measurements:

6.1.1 Hydrodynamic Characteristics

- The increase in head-loss coefficient for the channel configuration studied was in the range, 109\%–317\%, in comparison with an open channel, and the
observed behaviour followed a power-law decay strongly dependent on channel Reynolds No. For higher confinement ratios ($\beta = 0.6–0.7$), the head-loss coefficient was found to increase, which corresponded to a change in flow regime, from steady laminar to quasi-unsteady.

- The isolated pillar-loss coefficient behaviour of the cylindrical pillar was found to follow a power-law decay. Good agreement was observed, and the obtained curve-fit parameters provide a useful tool for predicting the pillar-loss coefficient behaviour.

- The surface fit of the pillar-loss coefficient, $\kappa$, to Reynolds No., $Re_d$, and confinement ratio, $\beta$, collapsed well, and was valid up to $Re_d = 425$ to within $\pm 20\%$. The use of the pillar diameter length scale, $d$, provided better agreement than the square root of cross-sectional area, $\sqrt{A}$, or open area ratio, $\sqrt{A(1 - \beta)}$, in predicting the pillar loss behaviour.

### 6.1.2 Thermal Characteristics

- Complex heat transfer characteristics were observed when increasing the confinement ratio of the pillar within the channel. Heat transfer was found to decrease with increasing Reynolds No. until the flow regime changed from steady to unsteady flow, producing a Von Kármán vortex street. Heat transfer in the steady flow regions was observed to decrease for an increase in confinement ratio from $\beta \geq 0.5$. Local heat transfer maps revealed that the wake region beyond the pillar broke down, creating a unsteady region.

- Varying the pillar distance upstream of the heated region from 1mm to 8mm highlighted a noticeable difference in heat transfer performance. It was found
that placing the pillar 1mm upstream of the cylinder produced up to a ∼40% greater heat transfer performance than the 8mm upstream case for equivalent Reynolds No. A similar trend in heat transfer as a function of mass-flow rate was found, with the minimum and maximum performance being -64–115% and -65–157% for the 8mm and 1mm respectively at $\beta = 0.7$, when compared to the open channel case. It is postulated that the accelerated flow regions due to the pillar presence decayed significantly downstream, while the increasing boundary layer growth reduced heat transfer.

- Pillar shapes showed measurable variance in heat transfer when compared to the circular pillar profile. A rhombus shaped pillar of chord diameter 10% featured the highest performance, compared to the triangular, square and circular cases, also measured with a peak heat transfer performance of 109%, compared to an open channel. The heat transfer performance across the range of Reynolds No. showed an consistent increase in heat transfer to an open channel. The twin pillar arrangements showed the greatest increase in heat transfer, with the circular shaped profile producing equivalent heat transfer (114%) to the 8mm case at a confinement ratio of 0.7 (115%), when compared to an open channel.

- The thermal-hydrodynamic behaviour was shown to be dominated by the frictional losses present in the channel over the heat transfer enhancement created. As the Reynolds No. increased, the heat transfer enhancement showed a higher rate of increase than the frictional loss for lower confinement ratios ($\beta < 0.3$). At higher confinements, however, the increases in frictional losses were equivalent to the heat transfer increase.

The results of both the hydrodynamic and thermal studies provide practical design choices for electrical conduits or blockages placed inside a channel for the
6. CONCLUSIONS AND RECOMMENDATIONS

TIPS architecture. Firstly, placing a wide pillar (> 50% of channel width) close to the heated regions yields heat transfer greater than an open channel beyond a critical Reynolds No. Secondly, this type of arrangement is an advantage from a further integration perspective, as multiple laser arrays with associated cooling architecture can be populated within the same device. If pumping power is a limiting design factor, yielding low flow rates, it would be preferable to choose a dual pillar arrangement with a shape that provides increased heat transfer enhancement. This would be a twin pillar arrangement with a rhombic shape profile.

6.2 Recommendations

• PIV measurements of the wake breakdown region for high confinement ratios could be taken, and compared to the heat transfer contour maps measured in this thesis. TKE could be measured and correlated to confinement ratio. Furthermore, local velocity fields can be integrated and compared to the head loss measured in this thesis.

• Channel aspect ratio was fixed for both experiments as it was found to be the optimal shape in terms of pumping power and heat transfer for the TIPS architecture. Extending the study to fixed confinement ratio with varying channel height would confirm whether or not the behaviour observed is consistent regardless of channel aspect ratio.

• Change the working fluid to a non-Newtonian type, such as visco-elastic, to determine if it affects the wake breakdown for high confinement ratios at their respective critical Reynolds No.

• The experimental measurements presented in this thesis are suitable as
validation of computational models. Once validated, optimised configurations of confinement ratio and pillar location could be identified for a microchannel design.

- The goal of the TIPS project is to implement an embedded micro scale cooling system in electronics packaging. Once optimised configurations of confinement and location are identified, fabrication and testing of this microchannel architecture should be completed with the chosen micro scale pump and secondary heat exchanger to create a demonstrator.
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Appendix

Appendix A

Publications


A. PUBLICATIONS

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Appendix B

Experimental Appendix

B.1 Derivation of Heated-Foil Equation

Section 3.2.1 presented an equation for the deduction of the forced convective heat flux component in the heated-foil technique. The following section will describe in detail the derivation such an equation on the control element illustrated in figure B.1. This technique is an energy balance exercise represented by equation B.1. The energy stored within a control element is the summation of the energy input to that element and energy generated minus the energy outputted by the element:

\[ \dot{E}_{st} = \dot{E}_{in} + \dot{E}_{gen} - \dot{E}_{out} \]  

(B.1)

To simplify the analysis, the following assumptions have been taken:

- Experiment is carried out at steady state, i.e. \( \dot{E}_{st} = 0 \).

- Each element has the same temperature field across all layers, as the Biot number is low: \( Bi < 0.1 \). This is due to a low Bi and negligible contact resistance due to the adhesion between paint and foil, and tape and paint layers. Conduction through each layer is assumed negligible, simplifying the analysis to two-dimensional.
Expanding on equation B.1 by considering the foil, paint and Kapton layers illustrated in figure B.1, the equation now becomes:

\[
[q_x]_f + [q_y]_f + [q_x]_p + [q_y]_p + [q_x]_k + [q_y]_k + q_{gen}dxdy.t_f \\
- [q_x+dx]_f - [q_y+dy]_f - [q_x+dx]_p - [q_y+dy]_p - [q_x+dx]_k - [q_y+dy]_k \\
- q_{f_c}dxdy - q_c.dxdy - q_r.dxdy = 0
\]  

Figure B.1: Illustration of the heated thin foil configuration used with a unit cell A displaying the different heat transfer modes through the unit. Modified from Stafford et al. (2009) to represent the experiment carried out in this thesis.

The incoming heat perpendicular from the lateral faces in the x- and y-directions can be described using Fourier’s law (Holman, 2010):

\[
q_x = -k_t.dy \frac{\partial T_{xy}}{\partial x}
\]  

(B.3)
B.1 Derivation of Heated-Foil Equation

\[ q_y = -k.t.dx \frac{\partial T_{xy}}{\partial y} \]  \hspace{1cm} (B.4)

Heat leaving the control element across their respective differential lengths, \(dx\) and \(dy\) can be expressed as the following:

\[ q_{x+dx} = q_x + \frac{\partial q_x}{\partial x} dx \]  \hspace{1cm} (B.5)

\[ q_{y+dy} = q_y + \frac{\partial q_y}{\partial y} dy \]  \hspace{1cm} (B.6)

Substituting equations B.3 - B.6 in equation B.2 simplifies to the following:

\[ \left( k_{ft} + k_{tp} + k_{tk} \right) \left( \frac{\partial^2 T_{xy}}{\partial x^2} + \frac{\partial^2 T_{xy}}{\partial y^2} \right) dx dy + \dot{q}_{gen} dx dy. t_f - q_{fc} dx dy - q_c dx dy - q_r dx dy = 0 \]  \hspace{1cm} (B.7)

Finally, a rearrangement of equation B.7 to isolate the forced convection component yields the following equation:

\[ q_{fc} = q_{gen} - q_r - q_c + \left( k_{ft} + k_{tp} + k_{tk} \right) \left( \frac{\partial^2 T_m}{\partial x^2} + \frac{\partial^2 T_m}{\partial y^2} \right) \]  \hspace{1cm} (B.8)

Equation B.8 corresponds to equation 3.9 in this thesis.
B.2 Sample Uncertainty Calculation

This section describes how uncertainty for the experimental measurements and derived quantities for each data set were calculated using the Kline-McKlintock method (Holman, 2012; Kline and McClintock, 1953). This method can be described as follows: given that the calculated result of derived parameter, \( f \), is represented as a function of several independently measured variables such as:

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

the associated uncertainty of the result, \( \omega_f \), is calculated from the root-sum squares of the individual parameters in the same equation. This is presented by equation B.10, which is the generic form of the uncertainty formula:

\[
\omega_f = \left( \left( \frac{\delta f}{\delta x_1} \omega_1 \right)^2 + \left( \frac{\delta f}{\delta x_2} \omega_2 \right)^2 + \ldots + \left( \frac{\delta f}{\delta x_n} \omega_n \right)^2 \right)^{\frac{1}{2}}
\]

(B.10)

The uncertainty of primary variables in all experiments are calculated as a quotient of the stated uncertainty and the measured variable as shown in equation B.11:

\[
\omega_W = \frac{\delta W}{W}
\]

(B.11)

An uncertainty calculation example of heat transfer coefficient using equations B.10 and B.11 is presented in this appendix using measured data. A sample uncertainty calculation for area averaged heat transfer coefficient for the heated-foil experiment is presented as follows. Equation B.12 presents the heat transfer coefficient formula, where heat flux, \( \dot{Q} \), and temperature difference, \( \Delta T \), are
expanded to show all primary variables considered in the analysis.

\[ h = \frac{\dot{Q}}{\Delta T} \]  \hspace{1cm} (B.12)

Where:

\[ \dot{Q} = \frac{VI}{A_{foil}} = \frac{VI}{L_{foil}W_{channel}}, \quad \Delta T = \bar{T}_{foil} - T_{in} - VI/\dot{m}C_P \]

Values for voltage, \( V \), current, \( I \), foil length, channel width, average foil temperature, inlet temperature, and mass-flow rate, \( \dot{m} \), are presented with their respective uncertainties below \(^1\):

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
<th>Uncertainty (( \delta ))</th>
<th>Relative Uncertainty (( \omega ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>( V )</td>
<td>0.326V</td>
<td>0.005V</td>
<td>0.015</td>
</tr>
<tr>
<td>( I )</td>
<td>6.018A</td>
<td>0.01A</td>
<td>0.0017</td>
</tr>
<tr>
<td>( L_{foil} )</td>
<td>10.25mm</td>
<td>10 ( \mu )m</td>
<td>0.00098</td>
</tr>
<tr>
<td>( W_{channel} )</td>
<td>6.25mm</td>
<td>10 ( \mu )m</td>
<td>0.0016</td>
</tr>
<tr>
<td>( T_{foil} )</td>
<td>308.22K</td>
<td>0.141K</td>
<td>0.00045881</td>
</tr>
<tr>
<td>( T_{in} )</td>
<td>295.5K</td>
<td>0.1K</td>
<td>0.000338</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>1.27kg/hr</td>
<td>0.1kg/hr</td>
<td>0.0787</td>
</tr>
</tbody>
</table>

The uncertainties of heat flux, \( \dot{Q} \), and temperature difference, \( \Delta T \), are shown below and combined to calculate the uncertainty in heat transfer coefficient, \( \omega h \)

\[ \omega \dot{Q} = \left[ \omega V^2 + \omega I^2 + \omega L_{foil}^2 + \omega W_{channel}^2 \right]^{1/2} = 0.015 \]
\[ \omega \Delta T = \left[ \omega T_{foil}^2 + \omega T_{in}^2 + \omega V^2 + \omega I^2 + \omega \dot{m}^2 \right]^{1/2} = 0.0812 \]
\[ \omega h = \left[ \left( \frac{\partial h}{\partial \dot{Q}} \omega \dot{Q} \right)^2 + \left( \frac{\partial h}{\partial \Delta T} \omega \Delta T \right)^2 \right]^{1/2} \]
\[ \omega h = \left[ (0.015)^2 + (0.0812)^2 \right]^{1/2} = 0.0826 = 8.26\% \]

\(^1\)Values presented in this section are relevant to the heat-transfer experiment presented in chapter 5.
B. EXPERIMENTAL APPENDIX

B.3 Fluid Properties

Fluid properties of water, such as specific heat capacity at constant pressure, \( C_p \), density, \( \rho \), kinematic viscosity, \( \mu \), thermal conductivity, \( k \), and Prandtl No., \( Pr \), for a range of temperatures, are presented in table B.1. In each analysis, the measured temperature was used to calculate fluid properties. A 5\(^{th}\) order polynomial fit was applied to the table data with a chi-squared value, \( R^2 \), of no less than 0.999. These regression equations were used to infer the measured fluid properties.

Table B.1: Properties of water (saturated liquid) taken from Holman (2010).

<table>
<thead>
<tr>
<th>Temperature, °C</th>
<th>( C_p ) (kJ/kgK)</th>
<th>( \rho ) (kg/m(^3))</th>
<th>( \mu ) (x10(^{-4}))</th>
<th>( k ) (W/mK)</th>
<th>( Pr ) (-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.00</td>
<td>4.195</td>
<td>999.2</td>
<td>13.1</td>
<td>0.585</td>
<td>9.40</td>
</tr>
<tr>
<td>15.56</td>
<td>4.186</td>
<td>998.6</td>
<td>11.2</td>
<td>0.595</td>
<td>7.88</td>
</tr>
<tr>
<td>21.11</td>
<td>4.179</td>
<td>997.4</td>
<td>9.8</td>
<td>0.604</td>
<td>6.78</td>
</tr>
<tr>
<td>26.67</td>
<td>4.179</td>
<td>995.8</td>
<td>8.6</td>
<td>0.614</td>
<td>5.85</td>
</tr>
<tr>
<td>32.22</td>
<td>4.174</td>
<td>994.9</td>
<td>7.65</td>
<td>0.623</td>
<td>5.12</td>
</tr>
<tr>
<td>37.78</td>
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<td>6.82</td>
<td>0.630</td>
<td>4.53</td>
</tr>
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<td>43.33</td>
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<td>0.637</td>
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<td>48.89</td>
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<td>5.62</td>
<td>0.644</td>
<td>3.64</td>
</tr>
<tr>
<td>54.44</td>
<td>4.179</td>
<td>985.7</td>
<td>5.13</td>
<td>0.649</td>
<td>3.30</td>
</tr>
<tr>
<td>60.00</td>
<td>4.179</td>
<td>983.3</td>
<td>4.71</td>
<td>0.654</td>
<td>3.01</td>
</tr>
<tr>
<td>65.55</td>
<td>4.183</td>
<td>980.3</td>
<td>4.30</td>
<td>0.659</td>
<td>2.73</td>
</tr>
<tr>
<td>71.11</td>
<td>4.186</td>
<td>977.3</td>
<td>4.01</td>
<td>0.665</td>
<td>2.53</td>
</tr>
<tr>
<td>76.67</td>
<td>4.191</td>
<td>973.7</td>
<td>3.72</td>
<td>0.668</td>
<td>2.33</td>
</tr>
<tr>
<td>82.22</td>
<td>4.195</td>
<td>970.2</td>
<td>3.47</td>
<td>0.673</td>
<td>2.16</td>
</tr>
<tr>
<td>87.78</td>
<td>4.199</td>
<td>966.7</td>
<td>3.27</td>
<td>0.675</td>
<td>2.03</td>
</tr>
<tr>
<td>93.33</td>
<td>4.204</td>
<td>963.2</td>
<td>3.06</td>
<td>0.678</td>
<td>1.90</td>
</tr>
</tbody>
</table>
B.4 Calibration Certificates

CALIBRATION CERTIFICATE

We herewith certify that the instrument mentioned below has been calibrated in accordance with the stated values and conditions. The calibration standards used are traceable to national standards of the Dutch Metrology Institute VSL.

Identifications

Calibrated instrument

Calibration Standard

Type: Flow meter (D)  Coriolis meter
Serial number: M5204160A  EX996219101A
Model number: L30Z-ABD-33-0  RHM-015-AAD-GNT
Certificate no.: BHTL48/1516627  ODS/20121026

Conditions

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Pressure</th>
<th>Temperature</th>
<th>Flow</th>
<th>Output range</th>
</tr>
</thead>
<tbody>
<tr>
<td>H2O</td>
<td>5 psi (g)</td>
<td>20 °C</td>
<td>10 kg/h</td>
<td>0 - 100 %</td>
</tr>
<tr>
<td>H2O</td>
<td>4.0 bar (a)</td>
<td>21.1 °C</td>
<td>Room temperature</td>
<td>21.1 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Atm. pressure</td>
<td>1014 hPa</td>
</tr>
</tbody>
</table>

Results

<table>
<thead>
<tr>
<th>Nominal Flow Setting</th>
<th>Calibrated Output Signal</th>
<th>Customer Flow</th>
<th>Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0 %</td>
<td>0.0000 %</td>
<td>0.0000 kg/h</td>
<td>0.0 %FS</td>
</tr>
<tr>
<td>25.0 %</td>
<td>25.19 %</td>
<td>2.521 kg/h</td>
<td>0.0 %FS</td>
</tr>
<tr>
<td>50.0 %</td>
<td>49.96 %</td>
<td>5.001 kg/h</td>
<td>-0.1 %FS</td>
</tr>
<tr>
<td>75.0 %</td>
<td>75.04 %</td>
<td>7.510 kg/h</td>
<td>-0.1 %FS</td>
</tr>
<tr>
<td>100.0 %</td>
<td>100.0 %</td>
<td>10.01 kg/h</td>
<td>-0.1 %FS</td>
</tr>
</tbody>
</table>
An INAB Accredited Calibration Laboratory Reg No.001C

<table>
<thead>
<tr>
<th>Date of Issue:</th>
<th>13TH JULY 2015</th>
<th>Approved Signatory:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Category:</td>
<td>A - C.S.L, CO. LIMERICK</td>
<td>Timmy Davern (Head of Laboratory.)</td>
</tr>
<tr>
<td>Customer:</td>
<td>UNIVERSITY OF LIMERICK</td>
<td>Tony O'Mara(Technical Manager.)</td>
</tr>
<tr>
<td>Address:</td>
<td>CO. LIMERICK</td>
<td></td>
</tr>
</tbody>
</table>

Details Of Unit Calibrated:

<table>
<thead>
<tr>
<th>Manufacturer:</th>
<th>FLUKE</th>
<th>Date Received:</th>
<th>6TH JULY 2015</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model:</td>
<td>45</td>
<td>Date Calibrated:</td>
<td>10TH JULY 2015</td>
</tr>
<tr>
<td>Serial No:</td>
<td>7594003</td>
<td>Calibrated By:</td>
<td>MICHAEL KENNEDY</td>
</tr>
<tr>
<td>Ref No:</td>
<td>N/A</td>
<td>Temperature:</td>
<td>(23± 5) °C</td>
</tr>
<tr>
<td>Description:</td>
<td>DUAL DISPLAY DIGITAL MULTIMETER</td>
<td>Humidity:</td>
<td>(50±25) %</td>
</tr>
</tbody>
</table>

The above instrument was tested against the manufacturer's accuracy specification at the points shown and the results are tabulated in the following report. This report relates solely to the instrument described above.

This certificate is issued in accordance with the conditions of accreditation laid down by the Irish National Accreditation Board, which has assessed the measurement capability of the Laboratory. The reported results are traceable to recognised National and International standards. The copyright of this report is reserved to Calibration Specialists Limited (CSL) and it shall not be used either in whole or in part for the purposes of advertising, publicity, litigation or otherwise without the prior consent of CSL. This calibration certificate contains information belonging to Calibration Specialists Ltd., which is confidential and/or legally privileged. Information is intended only for the use of the entity named above. If you have received this certificate in error, please notify us by telephone immediately at the above number.

The reported expanded uncertainty is stated as the standard uncertainty of measurement multiplied by the coverage factor \( k=2 \), which for a normal distribution corresponds to a coverage probability of approximately 95%. The standard uncertainty of measurement has been determined in accordance with EAL Publication EA-4/02.
Method: The unit under test was first allowed to stabilise. Precisely known values were then applied to the input of the unit under test and the resultant readings compared against manufacturer’s specifications. The Manufacturer Published Performance Test Procedure is carried out in reference to the Fluke 45 Dual Display User Manual P/N 855981 Rev.2, 12/89 published in 1989.

### DC Voltage Test

<table>
<thead>
<tr>
<th>Range (V)</th>
<th>Value (V)</th>
<th>Min (V)</th>
<th>Actual (V)</th>
<th>Max (V)</th>
<th>Uncertainty (±V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(slow) 100 m</td>
<td>Short</td>
<td>-0.006 m</td>
<td>0.002 m</td>
<td>0.006 m</td>
<td>0.0005 m</td>
</tr>
<tr>
<td>90 m</td>
<td>89.971 m</td>
<td>89.981 m</td>
<td>90.029 m</td>
<td>0.004 m</td>
<td></td>
</tr>
<tr>
<td>(slow) 1000 m</td>
<td>900 m</td>
<td>899.71 m</td>
<td>899.79 m</td>
<td>900.29 m</td>
<td>0.02 m</td>
</tr>
<tr>
<td>300 m</td>
<td>Short</td>
<td>-0.02 m</td>
<td>0.000 m</td>
<td>0.02 m</td>
<td>0.005 m</td>
</tr>
<tr>
<td>300 m</td>
<td>299.90 m</td>
<td>299.95 m</td>
<td>300.10 m</td>
<td>0.01 m</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>2.9990</td>
<td>2.9994</td>
<td>3.0010</td>
<td>0.0001</td>
</tr>
<tr>
<td>-3</td>
<td>-3</td>
<td>-3.0010</td>
<td>-2.9992</td>
<td>-2.9990</td>
<td>0.0001</td>
</tr>
<tr>
<td>30</td>
<td>30</td>
<td>29.990</td>
<td>29.994</td>
<td>30.010</td>
<td>0.001</td>
</tr>
<tr>
<td>300</td>
<td>300</td>
<td>299.90</td>
<td>299.94</td>
<td>300.10</td>
<td>0.01</td>
</tr>
<tr>
<td>1000</td>
<td>1000</td>
<td>999.5</td>
<td>999.7</td>
<td>1000.5</td>
<td>0.1</td>
</tr>
</tbody>
</table>

### AC Voltage Test

<table>
<thead>
<tr>
<th>Range (V)</th>
<th>Value (V)</th>
<th>Min (V)</th>
<th>Actual (V)</th>
<th>Max (V)</th>
<th>Uncertainty (±V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>300 m</td>
<td>#Short</td>
<td>-0.75 m</td>
<td>0.19 m</td>
<td>0.75 m</td>
<td>0.005 m</td>
</tr>
<tr>
<td>15 m, 1kHz</td>
<td>14.87 m</td>
<td>15.01 m</td>
<td>15.13 m</td>
<td>0.01 m</td>
<td></td>
</tr>
<tr>
<td>15 m, 100kHz</td>
<td>13.75 m</td>
<td>14.14 m</td>
<td>16.25 m</td>
<td>0.02 m</td>
<td></td>
</tr>
<tr>
<td>300 m, 1kHz</td>
<td>299.30 m</td>
<td>300.01 m</td>
<td>300.70 m</td>
<td>0.05 m</td>
<td></td>
</tr>
<tr>
<td>300 m, 100kHz</td>
<td>294.50 m</td>
<td>293.08 m</td>
<td>315.50 m</td>
<td>0.07 m</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>3, 1kHz</td>
<td>2.9930</td>
<td>3.0002</td>
<td>3.0070</td>
<td>0.0005</td>
</tr>
<tr>
<td>30</td>
<td>30, 1kHz</td>
<td>29.930</td>
<td>30.002</td>
<td>30.070</td>
<td>0.005</td>
</tr>
<tr>
<td>300</td>
<td>300, 1kHz</td>
<td>299.30</td>
<td>300.01</td>
<td>300.70</td>
<td>0.09</td>
</tr>
<tr>
<td>750</td>
<td>750, 1kHz</td>
<td>747.5</td>
<td>750.0</td>
<td>752.5</td>
<td>0.3</td>
</tr>
</tbody>
</table>

### DC Current

<table>
<thead>
<tr>
<th>Range (A)</th>
<th>Value (A)</th>
<th>Min (A)</th>
<th>Actual (A)</th>
<th>Max (A)</th>
<th>Uncertainty (±A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30 m</td>
<td>30.000 m</td>
<td>29.982 m</td>
<td>FAULTY</td>
<td>30.018 m</td>
<td>0.007 m</td>
</tr>
<tr>
<td>100 m</td>
<td>90.00 m</td>
<td>89.93 m</td>
<td>FAULTY</td>
<td>90.07 m</td>
<td>0.007 m</td>
</tr>
<tr>
<td>10</td>
<td>1.900</td>
<td>1.891</td>
<td>1.899</td>
<td>1.909</td>
<td>0.001</td>
</tr>
</tbody>
</table>
Date of Calibration: 10TH JULY 2015

AC Current @ 1kHz

<table>
<thead>
<tr>
<th>Range (A)</th>
<th>Value (A)</th>
<th>Min (A)</th>
<th>Actual (A)</th>
<th>Max (A)</th>
<th>Uncertainty (±A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30 m</td>
<td>30.000</td>
<td>29.840</td>
<td>FAULTY</td>
<td>30.160</td>
<td>0.020 m</td>
</tr>
<tr>
<td>100 m</td>
<td>90.000</td>
<td>89.400</td>
<td>FAULTY</td>
<td>90.60</td>
<td>0.07 m</td>
</tr>
<tr>
<td>10 m</td>
<td>1.900</td>
<td>1.871</td>
<td>1.898</td>
<td>1.929</td>
<td>0.0020</td>
</tr>
</tbody>
</table>

Resistance

<table>
<thead>
<tr>
<th>Range (Ω)</th>
<th>Value (Ω)</th>
<th>Min (Ω)</th>
<th>Actual (Ω)</th>
<th>Max (Ω)</th>
<th>Uncertainty (±Ω)</th>
</tr>
</thead>
<tbody>
<tr>
<td>300 #Short</td>
<td>#Short</td>
<td>0.00</td>
<td>0.02</td>
<td>0.04</td>
<td>0.005</td>
</tr>
<tr>
<td>3 k</td>
<td>99,999</td>
<td>99.99</td>
<td>100.00</td>
<td>100.06</td>
<td>0.01</td>
</tr>
<tr>
<td>30 k</td>
<td>9.999</td>
<td>9.99</td>
<td>10.00</td>
<td>10.01</td>
<td>0.01</td>
</tr>
<tr>
<td>3 M</td>
<td>0.9999</td>
<td>0.999</td>
<td>1.000</td>
<td>1.0006</td>
<td>0.001</td>
</tr>
<tr>
<td>30 M</td>
<td>9.998</td>
<td>9.970</td>
<td>10.00</td>
<td>10.026</td>
<td>0.001</td>
</tr>
</tbody>
</table>

Frequency Test (Rate Slow/Medium @1V)

<table>
<thead>
<tr>
<th>Range (Hz)</th>
<th>Value (Hz)</th>
<th>Min (Hz)</th>
<th>Actual (Hz)</th>
<th>Max (Hz)</th>
<th>Uncertainty (±Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 k</td>
<td>100.000</td>
<td>99.949</td>
<td>100.00</td>
<td>100.060</td>
<td>0.0100</td>
</tr>
</tbody>
</table>

Diode Test

<table>
<thead>
<tr>
<th>Range (V)</th>
<th>Value (V)</th>
<th>Min (V)</th>
<th>Pass / Fail</th>
<th>Max (V)</th>
<th>Uncertainty (±V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>n/a</td>
<td>short</td>
<td>-0.0008</td>
<td>(tone)</td>
<td>0.0008</td>
<td>0.0001</td>
</tr>
<tr>
<td>n/a</td>
<td>open</td>
<td>OL</td>
<td>OL</td>
<td>OL</td>
<td>n/a</td>
</tr>
</tbody>
</table>

Standards Used:
Asset No. 192 CALIBRATOR Cal Due: MAR - 2016

Comments:
The AC/DC 30mA and 100mA Current Range were found to be FAULTY, no repair was carried out.
This unit was within the manufacturer's specifications at all other points tested.

# Functional Test, Short verification for 2 Wire Resistance and AC Volts do not form part of our Accredited Schedule.

END OF CERTIFICATE

Procedure:Fluke45.xls Control No.X0027 Rev No.5.00 Date:30/01/2013