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**CONVECTIVE HEAT TRANSFER IN
MICROCHANNELS WITH VARYING FLOW
CROSS-SECTION**

CHENG KAI XIAN

**SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING**

2019

**CONVECTIVE HEAT TRANSFER IN
MICROCHANNELS WITH VARYING FLOW
CROSS-SECTION**

CHENG KAI XIAN

School of Mechanical and Aerospace Engineering

A thesis submitted to Nanyang Technological University,
Singapore in partial fulfilment of the requirement for the
degree of
Doctor of Philosophy

2019

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

31st July 2019

Date

Cheng Kai Xian

Supervisor Declaration Statement

I have reviewed the contents and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

31 JUL 2019

Date

Professor Ooi Kim Tiow

Authorship Attribution Statement

This thesis contains material from 2 papers published in the following peer-reviewed journals in which I am listed as an author.

Chapter 4 is published as Z.H. Foo, K.X. Cheng, A.L. Goh, and K.T. Ooi, Single-phase convective heat transfer performance of wavy microchannels in macro geometry, *Applied Thermal Engineering*, 141 (2018) 675-687.

DOI: 10.1016/j.applthermaleng.2018.06.015.

The contributions of the co-authors are as follows:

- Mr. Z.H. Foo proposed the idea of wavy microchannel, drafted the manuscript and conducted the experimental measurements.
- I provided the project direction, conducted the numerical simulations and discussed with Mr. Foo on the content of the manuscript. I also assisted in the collection of measurement data.
- Dr. Goh and Dr. Ooi provided guidance on the interpretation of the data and edited the draft manuscript.

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The contributions of the co-authors are as follows:

- I proposed the idea and project direction and drafted the manuscript.
- Mr. Y.S. Chong collected the measurement data.
- Dr. Ooi edited the draft manuscript.

31st July 2019

Date

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Abstract

The microchannel heat sinks offer superior heat transfer capabilities. In the past three decades, these microchannels have been studied extensively in terms of design and implementation, fundamental understanding of heat transfer and fluid flow, as well as the enhancement techniques in heat transfer. These thermally-efficient channels have found applications in the industries where space and thermal performance are the key drivers. However, cost-effective manufacturing is a hurdle for the proliferation of microscale heat transfer in the wider commercial applications.

Kong and Ooi demonstrated the ability to realise microscale heat transfer by superimposing two macro geometries. In one case, they demonstrated that an annular gap of less than 1 mm in size can be obtained by inserting a macro-sized rod into a macro-sized hollow cylinder. The heat transfer coefficients achieved in these microscale gaps are comparable with that in the existing microchannels. Uniquely, these macro geometries were machined using readily-available conventional machining methods and thereby eliminating costly micro-fabrication processes for the realisation of microscale heat transfer. Goh and Ooi studied nature-inspired configurations for an enhanced heat transfer in these microscale annular gaps. A triangular wavy design, which resembles that of a durian-thorny skin, outperformed the other nature-inspired profiles, in terms of thermo-hydraulic performance.

The present Ph.D. study aims to investigate the thermo-hydraulic performance of various microchannel configurations with a varying flow cross-section, implemented by superimposing two macro geometries, realising gap sizes less than 1 mm. In this study, the different configurations of the microchannel are realised by placing solid cylinders of different surface profile into a hollow cylinder of macro-scale with a diameter of 20 mm and a length of 30 mm. Implementation of surface profiles on these cylinders and superimposition of these cylinders at different orientation result in flow channels with a varying cross-section along the flow direction. These parts are machined using simple turning processes.

The first configuration is a microchannel with a sinusoidal wavy profile on the non-heated surface of the channel, differentiating this study from the existing works available in open literature. As the heating surface remains flat, this channel features a periodically expanding and contracting flow channel. A parametric study is conducted on the amplitude and wavelength of the waveform. A total of six wavy microchannels are studied and the thermal and hydrodynamic performance are compared against that of a straight microchannel. Experimental measurements are conducted for Re of 1400 to 4600 for a heat flux of 53.0 W/cm^2 . The channels are also modelled numerically to visualise the flow field and heat transfer. This channel configuration is able to remove up to 64 per cent more heat as compared to the straight microchannel, evaluated at the same pumping power. The amplitude of the waveform shows a more dominant effect on the thermo-hydraulic performance as compared to the wavelength. In addition, working correlations for the average Nusselt number and friction factor are proposed for this channel configuration, with a maximum discrepancy of 15 and 12 per cent respectively, when comparing the predicted data with the measured data.

The second configuration is a microchannel with a sinusoidal wavy profile on the heated surface of the channel and the non-heated surface remains flat. This channel is compared with three other channels: the first configuration with a wavy non-heated surface, a serpentine configuration and a straight channel, in terms of thermo-hydraulic performance. The performances are predicted using a numerical model for Re range of 750 to 2200. The predicted Nu and f show an average difference of 4.8 and 18.6 per cent as compared to the measured data gathered for the first configuration. Although the channel with a wavy heated surface has similar pressure drops as that with a wavy non-heated surface, the former has a higher removal capability. This second channel configuration achieves a maximum performance index of 1.88, implying an 88 per cent higher heat removal capability at the same pumping power as the straight microchannel. This figure is 36.8 and 74.1 per cent higher than that of the first configuration and the serpentine channel.

The third configuration is a microchannel with a sinusoidal wavy profile of a varied wave amplitude along the flow direction, on both the heated and non-heated surface. For each of the configurations, three types of waveform are implemented: increased-

amplitude, decreased-amplitude and uniform-amplitude, yielding a total of six wavy microchannels. Experimental measurements are collected for the channel with a wavy non-heated surface, for the Re range of 550 to 3400 and a heat flux of 42.4 W/cm^2 . The numerical model, which is validated with the measured results, is used to predict the heat transfer and flow field of the channels with a heated wavy profile, for the Re range of 1580 to 3110. For the channels with a wavy non-heated wall, a decreased amplitude along the flow direction achieves an enhancement of 19.9 and 21.3 per cent of heat removal capability as compared to the waveforms with an increased and uniform amplitude respectively, under a low pumping power condition of 0.5 W. As the pumping power increases, the channel with a uniform-amplitude waveform achieves an improvement in performance index, outperforming those with a varied amplitude along the flow direction. Similar trends are observed with the wavy profile implemented on the heated surface, with the channel with the uniform-amplitude waveform incurring a 44.8 to 49.0 per cent lower pressure drop, despite the similar heat transfer coefficients as the decreased-amplitude waveform.

The two other configurations which are being analysed numerically are an eccentric channel and a skewed channel. Three eccentric ratios and three skewed ratios are studied for annular microchannels with radius ratios of 0.95 and 0.97. The predicted Nu and f , for the range of 550 to 2300, are validated with the measured results for the concentric configurations, and the analytical solutions for the eccentric channels. The results showed that the Nu and f of the eccentric and skewed channels deviate more from the concentric channel at a higher radius ratio. The skewed channels have Nu deviated less than 5 per cent from that of the concentric channels, but an increment more than 20 per cent in f for the highest skewed ratio. The eccentric channel with an eccentric ratio of 0.75 has a Nu which is more than 10 per cent lower, and a lower f of similar magnitude range, as compared to the concentric channel.

The last configuration that is being studied is a converging flow channel, i.e. a channel with a reducing hydraulic diameter along the flow direction. Four converging gradients are studied, and the thermo-hydraulic performance is compared against a straight channel. Experimental measurements are gathered for Re range of 1400 to 5200. All the converging flow channels possess a performance index less than unity, implying that

these channels remove less heat as compared to the straight channel, when operating at the same pumping power.

The findings of these channel configurations contribute to the development of a high performance compact microchannel heat exchanger.

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Nomenclature

Latin symbol

A	Area	[m ²]
c	Specific heat capacity	[J/kg·K]
C	Coefficient	[-]
D	Diameter	[m]
E	Increment ratio	[-]
f	Friction factor	[-]
h	Average heat transfer coefficient	[W/m ² ·K]
I	Current	[A]
k	Thermal conductivity	[W/m·K]
K	Heat conductance	[W/K]
Kn	Knudsen number	[-]
l	Length	[m]
L	Total length	[m]
$\dot{m}$	Mass flow rate	[kg/s]
Nu	Nusselt number	[-]
p	Pressure	[Pa]
P	Perimeter	[m]
Pr	Prandtl number	[-]
q	Heat flux	[W/m ²]
Q	Heat transfer rate	[W]
r	Radius	[m]
R	Resistance	[ohm]
Re	Reynolds number	[-]
St	Stanton number	[-]
T	Temperature	[K]
V	Velocity	[m/s]
$\dot{V}$	Volumetric flow rate	[m ³ /s]
x	Axial distance	[m]
x^+	Dimensionless axial distance	[-]
y	Distance from wall	[m]

Greek symbol

ρ	Density	[kg/m ³]
μ	Dynamic viscosity	[Pa·s]
τ	Shear stress	[N/m ²]
θ	Dimensionless temperature profile	[-]
ν	Kinematic viscosity	[m ² /s]
α	Thermal diffusivity	[m ² /s]
ε	Eccentricity ratio	[-]
δ	Offset distance	[m]
λ	Free mean path	[m]
η	Performance index	[-]

ε	Surface roughness	[m]
ε^*	Dimensionless surface roughness	[-]
Ω	Pumping power	[W]

Subscript	Parameter
b	Bulk
c	Cross-sectional
$conv$	Conventional
cp	Constant properties
$crit$	Critical
cu	Copper
e	Enthalpy
$--E$	Enhanced
f	Friction
fd	Fully developed
h	Hydrodynamic
i	Inner
in	Inlet
lam	Laminar
m	Mean
o	Outer
out	Outlet
p	Constant pressure
r	Radial
ref	Reference
s	Heat transfer surface
t	Thermal
tur	Turbulent
w	Wall

Chapter 1 Introduction

This chapter introduces the Ph.D. research study. Firstly, the research motivation is described. Secondly, the objectives of this study are clearly stated. Thirdly, the scope of the work is clearly defined to achieve the stated objectives. Lastly, the organisation of this thesis is described.

1.1 Research motivation

Microscale passages are commonly found in natural transport systems such as lungs and kidneys in humans and other living organisms. The scale of these passages facilitates the transport processes due to the increased area-to-volume ratio. The perks of these micro-sized channels were realised in an engineering device in 1981 when Tuckerman and Pease [1] introduced rectangular channels with a hydraulic diameter of 100 μm in a heat sink for the cooling of electronic devices. A high removal capability of 790 W/cm^2 was demonstrated with a maximum temperature increment of 71 $^{\circ}\text{C}$, as compared to the fluid inlet temperature. Generally, channels with a hydraulic diameter between 1 μm to 1 mm are categorised as microchannels [2, 3]. Microchannels result in a more compact heat exchange device, featuring material saving, smaller fluid hold-up and higher thermal efficiency as compared to the conventionally-sized channels. Convective heat transfer in microchannels has since then become a popular research area, mainly driven by the increased heat flux dissipation in microelectronic devices and the emergence of microscale devices that require cooling [4].

Within three decades since the inception of the microchannel in heat removal devices, it has been well studied in terms of design, the underlying physics and practical implementation [5]. However, the application of microchannel is still limited to niche areas where space and thermal performance are the key drivers. Cost-effective manufacturing is one of the main hurdles for the proliferation of microscale heat transfer in commercial products [6]. Kandlikar et al. [7] mentioned in 2012 that an area recommended for research is the development of high duty heat exchangers at competitive costs using microscale passages to replace those employing macro-scale passages.

The need for a cost-effective and simple method of implementing microscale heat transfer motivated Kong and Ooi [8] to implement microscale channels by superimposing macro geometries, which were machined using readily-available conventional machining methods. As there was still a need for single-phase liquid cooling in microchannels while keeping the pressure drop low [9], Goh and Ooi [10-12] successfully achieved improved thermo-hydraulic performances using nature-inspired profiles, as compared to a straight microchannel. They adopted the similar methodology of implementing a microchannel as [8].

The triangular wave profiles, named Durian profiles in the original article [12], depicted the highest thermo-hydraulic performance, as compared to the other profiles being investigated, which were fish-scale and inverted fish-scale profiles [10, 11]. This Ph.D. study attempts to improve the thermo-hydraulic performance of the design which has thorny edges, by implementing smoother edges, resulting in a sinusoidal wavy profile. While sinusoidal wavy channels are not uncommon [13-16], this study investigates the effects of implementing the wavy profile on the non-heated surface, while the heated surface remains smooth. This yields a channel with a changing hydraulic diameter along the flow direction, as well as a constant heat transfer area. This configuration is useful when modifications to the existing heated surface are not feasible. The increasing and decreasing hydraulic diameter introduces re-entrant effects to enhance the heat transfer. Heat transfer and hydrodynamic correlations are proposed for this orientation for the adoption of this channel configuration. This study is presented in Chapter 4 of this thesis.

While the channel configuration described above yields a positive thermo-hydraulic performance as compared to the straight microchannels, it is of scientific interest to investigate the different possible implementations of wavy microchannels. There are four possible implementations of a wavy microchannel, i.e., a serpentine channel and a racoon channel which are double-wavy-wall configurations, as well as two single-wavy-wall configurations with different heating boundary. The latter yields a changing hydraulic diameter along the flow direction on top of the channel curvature technique. While it has been proven that racoon channels possess lower thermo-hydraulic

performance as compared to serpentine channels [14], racoon channels have been excluded for comparison. This study is detailed in Chapter 5.

As Chapter 5 summarises that single-wavy-wall channels possess higher thermo-hydraulic performances as compared to the serpentine and straight microchannels, Chapter 6 optimises the single-wavy-wall channels by varying the wave amplitude along the flow direction. This modification has been proven to be effective in serpentine wavy microchannels [13, 17]. The difference in the thermal and hydrodynamic performance between wavy profiles with uniform and varying amplitude, at the same average amplitude, is quantified using the measured data for the non-heated wavy profiles and the predicted data for the heated wavy profiles.

Implementing the microscale gap in a concentric manner comes with a high cost and long machining hours as a tight tolerance is expected on all the parts. Otherwise, an eccentricity might result. Deformation in the service also potentially distorts the concentricity of the channel. Although the effects of eccentricity, both uniform and changing eccentricity along the flow direction, have been intensively investigated in conventionally-sized pipes [18-20], there is a need to investigate the effect of eccentricity in a microscale gap. This is mainly because, as the size of the channel scales down, the factors which are not significant in conventional-sized pipes, such as axial conduction in the substrate wall and variation in fluid properties, become significant particularly at low Reynolds numbers. Furthermore, the flow in a microscale channel mainly remains in the developing regime. The aforementioned factors affect the heat transfer and hydrodynamic performance of an eccentric annular microchannel. The findings are concluded in Chapter 7 of this thesis.

Converging flow channel is another channel configuration that has gained attention recently. This channel has a decreasing flow cross-section along the flow direction. The superiority of this channel configuration in promoting temperature uniformity has been demonstrated in [21-23]. Nevertheless, Wong and Ang [24] claimed that a straight microchannel outperforms a converging configuration in terms of thermo-hydraulic performance. A converging channel inevitably changes the convective heat transfer area and the conductive heat transfer area in the substrate. This yields an optimum tapering

ratio for a uniform temperature distribution [21]. The current experimental setup enables the evaluation of the performance of a channel with a decreasing hydraulic diameter under a constant heat transfer area. By keeping the heat transfer area constant, the effect of reducing the hydraulic diameter in a microscale gap on the heat transfer coefficient can be studied. The measured data is reported in Chapter 8.

Therefore, this Ph.D. study investigates the heat transfer and hydrodynamic performance of different microchannel configurations with a varying flow cross-section along the flow direction. These include different wavy-channel implementations, eccentric and skewed annular microchannels, as well as a channel with a decreasing hydraulic diameter. The findings are crucial for the implementation of a microchannel, based on the evaluation of an increased heat removal capability at the same pumping power as compared to that of a straight configuration.

1.2 Objectives

The main objective of this Ph.D. research is to study various micro-scale channel configurations with a varying flow cross-section along the flow direction, in terms of heat transfer and hydrodynamic performance. The following goals are to be achieved:

- To achieve microscale heat transfer effects in microchannels implemented using conventional machining methods.
- To enhance the heat transfer by increasing the convective heat transfer coefficient using passive heat enhancement techniques, particularly the channel curvature technique and re-entrant effects.
- To compare the heat transfer and hydrodynamic performance of channel configurations with different design parameters.
- To understand the fluid flow and heat transfer in different micro-sized channel configurations by solving the governing equations using numerical methods.

1.3 Scope

In order to achieve goals stated in Section 1.2, the tasks to be accomplished are as follows:

- To review heat transfer and hydrodynamic theories in conventionally-sized flow channels, as well as the existing convective heat transfer studies on microscale single-phase liquid flow. The former is crucial to the grasp of the underlying concepts on convective heat transfer phenomena. The latter presents an overview of the history, up-to-date developments and research needs in this field. The challenges, limitations and opportunities in the existing studies are identified to advance this research topic.
- To design and implement various channel configurations to address the limitations in the existing literature. This includes preliminary numerical studies to predict the performance of the channels and metrology on the fabricated parts.
- To collect steady-state measurements to investigate the heat transfer and hydrodynamic performance of various channel configurations. Calibration of measuring devices, re-alignment of test loop and validation of the measured data have been performed to ensure the reliability of the data. 13 channel configurations have been implemented to gather the measured data. The studies are conducted for Reynolds number range of 550-5200 and heat flux of 42.4 to 53.1 W/cm². 600 steady-state measurements are gathered for each data point through the data acquisition system and reduced to the parameters of interest. The dependent variables include the heat transfer coefficient, Nusselt number, pressure drop and friction factor. The uncertainty of the measured data and reduced parameters is evaluated.
- To develop conjugate heat transfer numerical models and solve the models using ANSYS CFX or FLUENT. Mesh independence tests are conducted for each channel configuration. Besides, the models are validated using existing correlations and available measured data. The flow field and temperature distribution are used to predict the performance of the channels and explain the underlying mechanisms.
- To analyse and compare various channel configurations of different design parameters. The measured or predicted performance is presented, followed by

the flow field and temperature distribution to explain the observed phenomena. For the new channel configuration in Chapter 4, Nusselt number and friction factor correlations are proposed for the industrial adoption of the design.

1.4 Organisation of thesis

This Ph.D. thesis is organised as follows:

Chapter 1 presents an introduction of the Ph.D. research study, which includes the research motivation, the desired objectives and the scope. This chapter also describes the organisation of the thesis.

Chapter 2 covers the literature survey conducted. The first two sections detail the heat transfer and hydrodynamic theories for a circular duct and an annular channel. The third section describes the existing studies on single-phase convective heat transfer in microchannels, in the timeline of development. The needs and methods to achieve enhanced heat transfer in microchannels are also highlighted.

Chapter 3 presents the investigation methodologies employed in this Ph.D. study. The first section introduces the methodology of implementing the microchannels in this study. The second section elaborates the experimental methods. These include the measurement system, test module, calibration processes, data reduction method, uncertainty analysis and validation of the measured data. The third section describes the numerical modelling. Numerical modelling entails governing equations, boundary conditions, turbulence modelling, discretisation of the test module and validation of the predicted data.

Chapter 4 summarises the works on a single-wavy-wall microchannel configuration. A parametric study on the amplitude and wavelength of the profile is conducted. This chapter begins with the discussion on the research motivations, the experimental and numerical models, as well as the validation of the models. The measured data is then presented with the predicted flow field and heat transfer at a certain flow rate to understand the underlying mechanisms. Nusselt number and friction factor correlations are also proposed for the adoption of this channel configuration.

Chapter 5 compares the heat transfer and hydrodynamic performance of three wavy microchannel configurations. A literature survey is conducted on the existing work. This is followed by the description of the numerical model and the validation of the model using available measured data. The predicted heat transfer coefficient and pressure loss in different configurations are presented, followed by the heat and flow field to understand the underlying physics. Different configurations are then evaluated under the same pumping power requirement.

Chapter 6 presents the thermal and hydrodynamic performance of uniform, increased as well as decreased wave amplitude in the single-wavy-wall microchannels. This chapter has the similar organisation as Chapter 4.

Chapter 7 presents the investigation on the effect of eccentricity, both uniform and changing eccentricity along the flow direction, on the heat transfer and hydrodynamic requirement of microscale gaps. Two straight channels of different hydraulic diameters, together with three eccentric and skewed configurations are investigated. This chapter has the same organisation as Chapter 5.

In contrast to the previous chapters, Chapter 8 discusses the measured data of four tapering channels. Each of the channels has different tapering gradient at the same average hydraulic diameter. The temperature and flow parameters are measured, and the performance is deduced and presented.

Chapter 9 concludes the works accomplished during this Ph.D. study and summarises the main findings from the investigations. Upon reflection on the key findings, key strategies and opportunities for future research are recommended.

Chapter 2 Literature Review

This chapter reviews and evaluates the existing literature which is relevant to the current study. The first two sections present the heat transfer and hydrodynamic theories for forced internal convection in a circular duct and an annular channel respectively. The third section presents the past studies in microchannels, highlighting the important developments and challenges in single-phase liquid flow in microchannels. This includes the need and methods to achieve and quantify heat transfer enhancement. The fourth section discusses the methodologies of implementing a microchannel available in the literature. The last section identifies the potential research direction based on the literature survey.

2.1 Forced internal convection in a circular duct

Convection is the heat transfer mode associated with the bulk movement of a fluid. For forced internal convection, Newton's law of cooling assumes the following form:

$$Q = hA_s(T_s - T_m) \quad (2-1)$$

The convective heat transfer coefficient, h depends strongly on the fluid properties, heat transfer surface geometries and conditions, as well as the flow condition. The multi-factor dependence of the parameter renders it difficult to be determined theoretically. In general, the magnitude of the convective heat transfer coefficient depends on the type of convection condition, as shown in Table 2-1.

Table 2-1: Range of convective heat transfer coefficient [25]

Type of convection	h (W/m ² ·K)
Free convection with air	$2 < h < 25$
Forced convection with air	$10 < h < 500$
Forced convection with water	$100 < h < 15000$
Condensation of water	$5000 < h < 100000$

As the fluid flow is often confined by solid surfaces, the interaction between the solid surfaces and the fluid flow is the crux of the convection process. This interaction results in the formation of two boundary layers: velocity boundary layer and thermal boundary layer.

2.1.1 Fluid flow considerations

The fluid assumes a zero-velocity at the wall attributed to the viscosity of the fluid. As the viscous effect penetrates the fluid in the direction normal to the fluid flow, the region in which the viscous effect is significant grows in thickness along the fluid flow direction. The velocity boundary layer is the region in which the fluid velocity varies from zero to 99 per cent of the free-stream velocity. For an internal flow, the velocity boundary layers form on each side of the wall and merge at a distance $x_{fd,h}$ downstream, as shown in Figure 2-1. $x_{fd,h}$ is the hydrodynamic entry length, and is defined in Equation (2-2) and (2-3) for laminar and turbulent flows respectively in a circular duct with a rounded converging nozzle as shown in the figure [26].

Figure 2-1: Formation of laminar velocity boundary layer [26]

$$(x_{fd,h})_{lam} = 0.05 \text{Re} D_h \quad (2-2)$$

$$(x_{fd,h})_{tur} = 10 D_h \quad (2-3)$$

The extent of the hydrodynamic entrance region depends on the flow regime of the flow, either laminar or turbulent. The flow regime of a certain pipe flow is invariably determined by Reynolds number which is defined as:

$$\text{Re} = \frac{\rho V D}{\mu} \quad (2-4)$$

There is no definite value in which the transition occurs. However, the ranges below are ubiquitous in practical cases for a circular duct:

Laminar flow	$\text{Re} < 2300$
Transitional flow	$2300 \leq \text{Re} \leq 10000$
Turbulent flow	$\text{Re} > 10000$

In the $0 < x < x_{fd,h}$ region, the flow is referred as hydrodynamically developing flow, whereas for $x > x_{fd,h}$, the flow is a hydrodynamically fully developed flow [26].

Local Fanning and Darcy friction factors are defined in Equation (2-5) and (2-6), respectively [27]. The friction factor is the highest at the entrance region and decreases gradually to a constant value in a fully developed flow. There are two factors causing a higher friction factor at the entrance region: a larger velocity gradient at the wall when the velocity profile is developing and an additional drag force which is taken into the consideration of the friction factor attributed to the accelerating velocity in the core [28].

$$C_f(x) = \frac{\tau_s(x)}{\frac{1}{2} \rho V_m^2} \quad (2-5)$$

$$f(x) = D_h \left(-\frac{\partial p}{\partial x} \right) \bigg/ \frac{1}{2} \rho V_m^2 \quad (2-6)$$

The average friction factors are given by:

$$C_f = \frac{1}{l} \int_{x=0}^l \tau(x) dx \bigg/ \frac{1}{2} \rho V_m^2 = \frac{A}{P_{wetted} l} (p_{x=0} - p_{x=l}) \bigg/ \frac{1}{2} \rho V_m^2 \quad (2-7)$$

$$f = \frac{D_h}{l} \int_{x=0}^l \left(-\frac{\partial p}{\partial x} \right) dx \bigg/ \frac{1}{2} \rho V_m^2 = \frac{D_h}{l} (p_{x=0} - p_{x=l}) \bigg/ \frac{1}{2} \rho V_m^2 \quad (2-8)$$

Therefore, Darcy friction factor is four times of Fanning friction factor:

$$f = 4C_f \quad (2-9)$$

The friction factor correlations for hydrodynamically developing flows are presented in Table 2-2. The dimensionless axial position x^+ is given by:

$$x^+ = \frac{x}{D_h \text{Re}} \quad (2-10)$$

Table 2-2: Friction factor correlations for circular ducts

Conditions	Correlation	Equation
Laminar, developing flow	$f \text{Re} = \frac{13.76}{(x^+)^{1/2}} + \frac{64 + 1.25 / (x^+) - 13.76 / (x^+)^{1/2}}{1 + 0.000021(x^+)^{-2}}$ <p>Shah [29]</p>	(2-11)
Turbulent, developing and fully developed flow	$f = A(\text{Re})^B$ $A = 0.3716 + \frac{4.06448}{L / D_h}$ $B = -0.268 - \frac{0.31930}{L / D_h}$ <p>$\text{Re} < 28000$</p> <p>Phillips [30]</p>	(2-12)

2.1.2 Thermal considerations

Akin to the velocity boundary layer, the thermal boundary layer is formed owing to the temperature difference between the solid surface and the fluid. Figure 2-2 shows the development of thermal boundary layer in a circular duct. In the fully developed region, the non-dimensional temperature profile θ , as defined in Equation (2-13), becomes independent of x :

$$\frac{\partial \theta}{\partial x} = \frac{\partial}{\partial x} \left[\frac{T(r, x) - T_s(x)}{T_m(x) - T_s(x)} \right] = 0 \quad (2-13)$$

Since θ is not a function of x :

$$\frac{\partial \theta}{\partial r} = \frac{\partial}{\partial r} \left[\frac{T - T_s}{T_m - T_s} \right] \neq f(x) \quad (2-14)$$

Local heat transfer coefficient, as given by Equation (2-15), remains constant along axial position in the thermally fully developed flow. As the thickness of the thermal boundary layer is zero at the entrance, h_x is very large. This value decays rapidly until it becomes constant in the thermally fully developed region, as shown in Figure 2-3.

$$h_x = \frac{q_s}{T_s - T_m} = \frac{-k \frac{\partial T}{\partial y} \Big|_{y=0}}{T_s - T_m} = k \frac{\partial T}{\partial r} \Big|_{r=r_o} \neq f(x) \quad (2-15)$$

Figure 2-2: Development of thermal boundary layer [26]

Figure 2-3: Variation of local heat transfer coefficient along axial direction [26]

The position at which the thermal boundary layer becomes fully developed, $x_{fd,t}$ is given as Equation (2-16) and (2-17) for laminar and turbulent flow respectively in a circular duct.

$$(x_{fd,t})_{lam} = 0.05 \text{Re Pr } D_h \quad (2-16)$$

$$(x_{fd,t})_{tur} = 10 D_h \quad (2-17)$$

Equation (2-16) is a product of Equation (2-2) and Prandtl number, Pr. Pr is defined as the ratio of the momentum diffusivity to thermal diffusivity of the fluid:

$$\text{Pr} = \frac{\nu}{\alpha} = \frac{c_p \mu}{k} \quad (2-18)$$

Figure 2-4: Development of boundary layers for different Prandtl number [27]

When $\text{Pr} \ll 1$, the development of the boundary layers resembles that in Figure 2-4(b). A larger thermal diffusivity enables the thermal effect to penetrate the flow field faster than the viscous effect, resulting in a thermal boundary layer which is always thicker than the velocity boundary layer. An opposite situation occurs when $\text{Pr} \gg 1$, as shown in Figure 2-4(a). For $x < x_{fd,f}, x_{fd,t}$, the region is referred to as the simultaneously developing flow. Water has a Prandtl number of 5.43 at a temperature of 30 °C.

Heat transfer coefficients can be determined from heat transfer correlations that have been formulated based on dimensional analysis or correlating the experimental measurement data in terms of dimensionless analysis. Nusselt number correlations, as defined in Equation (2-19), are presented in Table 2-3 for different flow configurations.

$$\text{Nu} = \frac{hD_h}{k} \quad (2-19)$$

Table 2-3: Nusselt number correlations for circular ducts

Conditions	Correlation	Equation
Laminar flow, fully developed velocity profile, thermally developing, constant heat flux	$\text{Nu} = 1.953(\text{Re Pr } D_h / L)^{\frac{1}{3}}$ <p>for $\text{Re Pr } D_h / L \geq 33.3$</p> $\text{Nu} = 4.364 + 0.0722 \text{Re Pr } D_h / L$ <p>for $\text{Re Pr } D_h / L < 33.3$</p> <p>Shah and London [31]</p>	(2-20)
Transition flow, fully developed flow, constant heat flux	$\text{Nu} = (1 - \gamma) \text{Nu}_{lam} + \gamma \text{Nu}_{turb}$ $\text{Nu}_{lam} = \left\{ 4.364^3 + 0.6^3 + \left[1.953 \sqrt{2300 \text{Pr } D_h / L} - 0.6 \right]^3 \right\}^{\frac{1}{3}}$ $\text{Nu}_{turb} = \frac{(\xi / 8) \times 10000 \times \text{Pr}}{1 + 12.7 \sqrt{\xi / 8} (\text{Pr}^{2/3} - 1)} \left[1 + (D_h / L)^{2/3} \right]$ $\xi = (1.8 \log_{10} 10000 - 1.5)^{-2}$ $\gamma = \frac{\text{Re} - 2300}{10^4 - 2300}$ <p>$2300 < \text{Re} < 10^4$</p> <p>Gnielinski [32]</p>	(2-21)

2.1.3 Relationship between heat transfer and fluid friction

Colburn [33] presented an empirical formula to express the relationship between heat transfer and fluid friction. The empirical formula is stated as:

$$\text{St}_x \text{Pr}^{2/3} = \frac{1}{2} C_{f,x} \quad (2-22)$$

When $\text{Pr} = 1$, the Colburn's formula is commonly recognised as Reynolds analogy between heat transfer and wall friction in turbulent flow. It is proven in [34] that

Reynolds analogy can be derived from the view that each eddy has the same propensity to convect heat as it has to transfer momentum in the direction normal to the wall. Similar expression as that proposed by Colburn can be obtained through the concept of density of contact spots.

2.2 Forced internal convection in concentric annulus

Concentric annular channels can be classified by r^* which is defined by:

$$r^* = \frac{r_i}{r_o} \quad (2-23)$$

An annular channel can be approximated as a parallel configuration when $r^* \geq 0.5$ [35]. When r^* approaches 1, it is a parallel-plate condition, while $r^* = 0$ defines a circular duct with an infinitesimal core at the centre. The r^* of the current study ranges from 0.95 to 0.97. For this range of r^* , the hydrodynamic and thermal entrance lengths for the laminar flow are given by [36] as:

$$(x_{fd,f}^+)_{lam} = 0.0108 \quad (2-24)$$

$$(x_{fd,t}^+)_{lam} = 0.04101 \quad (2-25)$$

The second fundamental boundary condition which defines a constant heat flux on one wall and an adiabatic condition on the other wall, is assumed for Equation (2-25). For a fully turbulent fluid flow, in Re range of 16000 and 70000, the hydrodynamic entrance length is about 20 to 25 hydraulic diameters [37].

The flow in an annular duct can also be approximated as an internal flow of circular cross section, by using an effective diameter as the characteristic length or the hydraulic diameter:

$$D_h = \frac{4A_c}{P} \quad (2-26)$$

By using this definition, the hydraulic diameter of a concentric annulus is defined by:

$$D_h = D_o - D_i \quad (2-27)$$

Goh and Ooi [11] reported that the laminar-to-turbulent region for an annular microchannel of $r^* = 0.97$ to be between 2200 to 3400, which coincides with that of parallel plate configuration reported by Beavers et al. [38]. Gnielinski [39] identified $2300 < Re < 10000$ to be in the transition region for this channel configuration.

In contrast to the circular ducts which are categorised as singly connected ducts, concentric annuli are doubly connected ducts. This configuration has more combinations of boundary conditions as there are two walls. Therefore, in contrast to the friction factor in which the correlations of the circular duct can be conveniently adopted by using the concept of hydraulic diameter, the Nusselt number correlations of the concentric annulus require special attention and are presented in Table 2-4.

Table 2-4: Nusselt number correlations for concentric annular

Conditions	Correlation	Equation
Laminar, fully developed velocity profile, thermally developing, heated isothermal outer wall and adiabatic inner wall	$Nu = \sqrt[3]{Nu_1^3 + Nu_2^3}$ $Nu_1 = 3.66 + 1.2 \left(\frac{D_i}{D_o} \right)^{\frac{1}{2}}$ $Nu_2 = 1.615 \left[1 + 0.14 \left(\frac{D_i}{D_o} \right)^{\frac{1}{3}} \right] \sqrt[3]{Re Pr D_h / L}$ <p>Gnielinski [39]</p>	(2-28)
Transition flow, fully developed velocity profile, thermally developing, heated isothermal outer wall and adiabatic inner wall	$Nu = (1 - \gamma) Nu_{lam}^* + \gamma Nu_{turb}$ <p>*Nu_{lam} is obtained from Equation (2-28) with $Re = 2300$</p>	(2-29)

	$\text{Nu}_{tur} = \frac{(\xi_{ann}/8) \times 10000 \times \text{Pr}}{a + 12.7 \sqrt{\xi_{ann}/8} (\text{Pr}^{\frac{2}{3}} - 1)} \left[1 + \left(\frac{D_h}{L} \right)^{2/3} \right] F_{ann}$ $a = 1.079 - \frac{0.63}{(1 + 10\text{Pr})}$ $\xi_{ann} = [1.8 \log_{10}(\text{Re}^*) - 1.5]^{-2}$ $\text{Re}^* = 10000 \frac{\left[1 + \left(\frac{D_o}{D_i} \right)^2 \right] \ln \left(\frac{D_o}{D_i} \right) + \left[1 - \left(\frac{D_o}{D_i} \right)^2 \right]}{\left(1 - \frac{D_o}{D_i} \right)^2 \ln \left(\frac{D_o}{D_i} \right)}$ $F_{ann} = \left[0.9 - 0.15 \left(\frac{D_i}{D_o} \right)^{0.6} \right]$ $\gamma = \frac{\text{Re} - 2300}{10^4 - 2300}$ $2300 < \text{Re} < 10^4$ <p>Gnielinski [39, 40]</p>	
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2.3 Forced internal convection in eccentric annulus

Eccentricity often results from manufacturing tolerances and imposed service conditions. Even moderate values of eccentricity greatly affect the flow rate in an annular with large r^* [34]. The eccentricity is defined by:

$$\varepsilon = \frac{\delta}{r_o - r_i} \quad (2-30)$$

where the parameters are defined in Figure 2-5.

Figure 2-5: Front view of an eccentric annulus

Fully developed laminar flow through eccentric annuli has been studied analytically and numerically by Cheng and Hwang [41] as well as Trombetta [19]. The latter obtained velocity and temperature solutions for the fundamental problems of the first, second and fourth kinds. The analytical solutions for the developing flows are presented in [42].

2.4 Corrections for fluid properties

The convective heat transfer solutions generally assume constant fluid properties. Errors arise when there is a large temperature difference between the fluid and the wall. The fluid properties vary with temperature and affect the variation of velocity and temperature within the boundary layer. However, the analytical investigation on the effect of variable fluid properties on heat transfer is a complicated task owing to the different variation with temperature from one fluid to another, and difficulty in expressing the variations in an analytical form [43]. Therefore, appropriate correlations based on the constant-property assumption have been proposed.

For liquids, the variation of viscosity is the most significant effect amongst all the property effects. Therefore, the variable-property Nusselt numbers and friction factors are correlated by:

$$\frac{\text{Nu}}{\text{Nu}_{cp}} = \left(\frac{\mu_b}{\mu_w} \right)^n \quad (2-31)$$

$$\frac{f}{f_{cp}} = \left(\frac{\mu_b}{\mu_w} \right)^m \quad (2-32)$$

Deissler [44] determined $n = 0.14$ and $m = -0.58$ (for heating only) for laminar flow, with $\text{Pr} > 0.6$, through a circular duct at constant heat flux boundary condition. Petukhov [45] identified $n = 0.11$ for $10^4 < \text{Re} < 5 \times 10^6$, $2 < \text{Pr} < 40$, and $0.08 < \mu_w / \mu_b < 40$ and $m = \frac{1}{6}(7 - \mu_b / \mu_w)$ for $0.35 < \mu_w / \mu_b < 2$, $10^4 < \text{Re} < 2.3 \times 10^5$, and $1.3 < \text{Pr} < 10$.

The calculation of Reynolds number, however, needs to be determined from the main-stream viscosity as it determines the nature of the flow [46].

2.5 Convective heat transfer in microchannels

All modes of microscale heat transfer: the fundamental modes of conduction in microstructures, convection in microchannels and microscale radiation phenomena have been pursued by the researchers. Among these, convective heat transfer in the microchannels is the most studied [47]. There are two main broad categorisations of the microchannel. Mehendale et al. [48] first introduced a set of definitions which are merely based on the channel dimension. Microscale heat exchangers are those with a channel dimension of 1 to 100 μm whereas those having a channel dimension of 100 μm to 1 mm are coined mesoscale heat exchangers. Kandlikar and Grande [49] further refined the classification based on flow considerations. The hydraulic diameter of the minichannels falls between 200 μm and 3 mm while that of the microchannels ranges between 200 to 10 μm . A broader definition of the microchannel with a characteristic dimension between 1 μm and 1 mm has been widely accepted [2, 3, 50, 51].

When it comes to engineering applications, the thermal transport phenomena such as phase change, single and two-phase flows and combined mode heat transfer are also important considerations. Although two-phase systems have inherently higher heat removal capabilities associated with the latent heat transport, there are complexities in the implementation such as saturation temperature, condensation, nucleation site activation and critical heat flux. Therefore, for immediate heat flux dissipation, single-phase flow appears to be more attractive [52].

The pioneering work in convective heat transfer in a microscale channel was introduced by Tuckerman and Pease in 1981 [1]. They introduced parallel rectangular micro-sized channels in a heat sink, spurred by an increasing demand for a high-performance heat dissipation system in the semiconductor sector. The heat sink achieved a heat removal capacity of 790 W/cm^2 with a maximum substrate temperature increment of 71°C from the inlet water temperature.

As compared to a conventional-sized flow channel, a microchannel enables material saving and smaller working fluid hold-up owing to the compactness of the device. It also has higher thermal efficiency and shorter response time [2].

2.5.1 Single-phase convective heat transfer and fluid flow in microchannel

Following the work by Tuckerman and Pease, single-phase convective heat transfer and liquid flow in a microchannel have become a popular research area, as illustrated in Figure 2-6. Kandlikar [53] summarised the research work in terms of the design and implementation, fundamental understanding of the fluid flow and heat transfer, as well as the practical implementation with enhancements, according to the timeline of development.

Figure 2-6: Publication histogram showing papers relevant to single-phase liquid heat transfer and fluid flow in microchannels [53]

The initial works focused on the design and implementation for almost one decade after the introduction of the microchannel heat sinks. Sasaki and Kishimoto [54] evaluated the optimal channel width for a microchannel to acquire the maximum allowable power density at a constant pumping power. With a constant pumping capacity, they concluded that the wider channels bode well for the heat removal capability as the total flow rate is higher, in spite of a higher finned surface area. Both researchers [55] carried on with the optimisation of a microchannel for high-power semiconductor devices by introducing a diamond-shaped structure. These boundary-layer-reinitialising structures resulted in a more uniform temperature distribution, as compared to the conventional parallel channels. Knight et al. [56] presented the governing equations for fluid dynamics and heat transfer in dimensionless form. They utilised these equations to determine the optimised dimensions for microchannels which result in the minimum thermal resistance.

Following that, a spate of research works focused on the fundamental understanding of the convective heat transfer and fluid flow at micro-scale. Some experimental results are remarkably different from that of the conventional-size pipes, raising interest among researchers to investigate the validity of continuum theory for microchannels. Furthermore, the measured results in the microchannel can be contradicting, as shown in Table 2-5. Morini [3] presented a detailed summary on the findings by the researchers, up to the year 2004. They reported that the friction factor, Nusselt number and critical Reynolds number can vary in the range following range: $0.5 < f / f_{conv} < 3.5$, $0.21 < Nu / Nu_{conv} < 16$ and $300 < Re_{crit} < 2300$ [57].

For gas flows, the continuum theory is invalid when the mean free path of the molecules is the same order as the hydraulic diameter of the channel [58]. This relationship is expressed as the Knudsen number:

$$Kn = \frac{\lambda}{D_h} \quad (2-33)$$

Slip flow occurs when at a high Kn when the interaction between the molecules close to the surface is negligible. The continuum theory is valid when Kn is less than 0.001.

By 2005, it was accepted that the liquid flow in the microchannel follows the continuum theory [7]. The discrepancy is mainly due to the size effect [57], i.e. the variation in the dominant factors and phenomena as the scale of the heat transfer device decreases. These include surface roughness, axial heat conduction in the channel wall and fluid properties variation. Furthermore, the entrance effects and measurement uncertainties might also lead to the discrepancies between the experimental results and theoretical values.

Table 2-5: Examples of work on the fundamental understanding of single-phase convective heat transfer and fluid flow in the microchannels

Literature	D_h (mm)	Channel geometry	Working fluid	Nu	f/Re
Peng et al. [59]	0.133 – 0.367	Rectangular	Water	↓	↑
Harms et al. [60]	0.404	Rectangular	Water	≈	≈
Qu et al. [61]	0.062-0.169	Trapezoidal	Water	↓	×
Qu et al. [62]	0.051-0.169	Trapezoidal	Water	×	↑
Xu et al. [63]	0.030-0.344	Rectangular	Water	×	≈
Celata et al. [64]	0.130	Circular	R114	↑	≈
Qu and Mudawar [65]	0.349	Rectangular	Water	≈	≈
Gao et al. [66]	0.199-1.923	Rectangular	Water	↓	≈
Liu and Garimella [67]	0.244-0.974	Rectangular	Water	×	≈
↑ Experimental results are higher than classical predictions ↓ Experimental results are lower than classical predictions ≈ Experimental results agree with classical predictions × Not available					

2.5.2 Challenges with convective heat transfer in microchannel using single-phase liquid flow

There are two concerns of employing single-phase liquid flow through the microchannel: the high-pressure loss across longer channels and the control of the fluid temperature increment by having a larger mass flow rate [68]. The latter inevitably adds to the pressure drop penalty. Therefore, there is still a need for enhanced microchannel designs with affordable pressure drop.

Cost-effective manufacturing is another hurdle for the proliferation of microchannel into commercial products [7, 68]. Currently, micro-sized channels are only commonly found in applications where space and thermal performance are the key drivers, such as microelectronic, space and automotive devices.

2.5.3 Single-phase heat transfer enhancement techniques

The heat transfer surface area and channel hydraulic diameter affect the heat transfer performance whereas the channel cross-sectional area and channel hydraulic diameter affect the pressure drop. Therefore, it is desirable to have a large cross-sectional area in conjunction with a high heat transfer coefficient. This can be achieved in a microchannel with larger hydraulic diameters providing a larger cross-sectional area, while the heat transfer is enhanced by using heat transfer enhancement features [52].

Single-phase enhancement techniques have been extensively investigated and can be categorised into two main classes: active and passive enhancement. The former requires an external power such as electric or acoustic fields whereas the latter employs various surface geometries or fluid additives for enhancement. Passive methods are more commonly found as active methods incur a higher operating cost and complications owing to noise and vibration [43].

Steinke and Kandlikar [69] reviewed all the conventional single-phase heat enhancement techniques and adopted these techniques for the microchannel flow. Their findings are summarised in Table 2-6.

Table 2-6: Passive heat enhancement techniques for single-phase convective heat transfer in microscale channel

Passive Enhancement Techniques	Application in Microscale Flow
Surface roughness	Use of etches or different surface treatments
Flow disruptions	Use of sidewall or in-channel disruptions
Channel curvature	Introduction of radius of curvature or large number of serpentine channels; more feasible than conventionally-sized channels
Re-entrant obstructions	Introduction of re-entrant structures such as orifices; Sudden expansion and contractions in the flow area
Secondary flow	Geometries to promote fluid mixing in channel
Out of plane mixing	Three-dimensional geometries
Fluid additives	Micro- or nanoparticles

Tao et al. [70] presented three mechanisms behind various passive enhancement techniques: decreasing thermal boundary layer thickness through the adoption of enhancement surfaces, increasing fluid interruptions and increasing the velocity gradient near a heated source.

2.5.4 Channel configurations

The channel configuration design (CCD) is a key factor determining the thermal performance of a heat transfer device. The CCDs which are commonly investigated include single-layer parallel channels, double-layer parallel channels, serpentine wavy channels and channels with a changing hydraulic diameter such as tapering channels.

The earlier works on CCD involve single-layer parallel channels. These channels of various geometrical shapes [71-73] and aspect ratio [74-76] have been studied in terms of heat transfer and hydrodynamic characteristics. These single-layer parallel channels, however, result in a huge temperature variation within the heat sink between the inlet and outlet mainly due to the small amount of coolant [21]. This high-temperature rise is undesirable in microelectronic devices as the resultant thermal stresses lead to thermal stability and reduced reliability. Although a high depth-to-width ratio addresses this issue, it incurs a high pumping power and complicated fabrication processes [23]. For a straight channel, the effects of eccentricity and skewness have been examined in this Ph.D. study. The eccentricity and skewness exist in an annular channel, which is formed through the superimposition of two cylinders.

Vafai and Zhu [77] introduced double-layer parallel channels with counter-flow configuration to address the aforementioned issue. An extensive amount of work has then been devoted to optimise the performance of this channel configuration [78-84]. The parameters which are being studied include the channel number, channel aspect ratio, channel-to-pitch width ratio and operating conditions [79]. These works reveal that a double-layer configuration outperforms the single-layer configuration in terms of cooling performance, temperature uniformity and hydrodynamic performance.

A wavy channel promotes fluid mixing through the formation of Dean vortices due to centrifugal force and chaotic advection at the troughs and crests, enhancing the heat transfer performance of this channel configuration. The augmented thermal performance with affordable pressure loss relative to the straight channel configuration spawned studies in this configuration. These studies, as covered in Chapter 4, focused mainly on the parametric studies on the amplitude, wavelength and aspect ratio of the channel. This Ph.D. study further explores another possible wavy-channel configuration in Chapter 4 and subsequently compares the performance of this channel configuration with the existing implementations in Chapter 5. In contrast to the existing serpentine wavy channels, these new single-wavy-wall channels studied realise a changing flow cross-section along the flow direction. In addition, given that the waveform amplitude is one of the factors affecting the thermal performance of the channels, the effects of varying the waveform amplitude along the flow direction on the heat transfer performance are investigated in Chapter 6.

Another emerging area of study is a converging flow channel, i.e. a flow channel with a reduction in the hydraulic diameter. A converging channel increases the heat transfer coefficient with the increment in flow velocity. The local wall temperature is reduced, promoting temperature uniformity of the flow channel and reducing the thermal resistance of the heat sink. This channel configuration has been studied in both single-layer and double-layer configurations, as covered in Chapter 8 of this Ph.D. thesis. Changing the tapering ratio affects the convective heat transfer area and the distribution of heat flux in the substrate. Therefore, this Ph.D. study also examines the effect of reducing the hydraulic diameter by keeping the heat transfer area constant.

2.6 Performance evaluation criteria

The performance of a conventional straight flow channel can be substantially enhanced by numerous techniques. Webb and Eckert [85] mentioned that there are three design objectives of an enhanced flow channel:

- a. To reduce heat transfer surface area for equal heat exchange capability and pumping power
- b. To increase heat exchange capability for equal heat transfer surface area and pumping power

- c. To reduce pumping power for equal heat transfer capability and surface area

Therefore, there is a need for performance evaluation criteria (PEC) which quantify the performance benefits of an enhanced heat exchange device, relative to a referenced design subject to various design constraints. This reference design is invariably the conventional device with smooth surfaces.

The most commonly used PEC, developed by Webb and Eckert [85], are in the form of Equation (2-34):

$$\eta = \frac{(St/St_{ref})}{(f/f_{ref})^{1/3}} \quad (2-34)$$

This index is derived from the comparison of the relative heat conductance of the enhanced and smooth channels:

$$\frac{K}{K_{ref}} \approx \frac{hA_s}{(hA)_{ref}} \quad (2-35)$$

The heat transfer rate of the enhanced and smooth channels is given as:

$$\frac{Q_E}{Q_{ref}} = \frac{hA_s(T_w - T_f)}{[hA_s(T_w - T_f)]_{ref}} \quad (2-36)$$

The relative heat conductance only quantifies the relative heat transfer rate, i.e. Equation (2-35) equals Equation (2-36) when the temperature difference between the wall and the bulk fluid, is the same for both enhanced and smooth channels. However, it is not the same for two different channels, at the same pumping power. The inadequacy of the ratio of heat transfer coefficient to fluid pumping power to represent the improved thermal performance has been highlighted in [86].

Therefore, this Ph.D. adopts the PEC, in the form of Equation (2-37), to evaluate the increment in heat transfer rate at the same pumping power. The assumptions and the derivation are described in Section 3.2.6.5.

$$\frac{Q_E}{Q_{ref}} = \frac{\left(2 + \text{St} \frac{A_s}{A_c}\right)_{ref} (hA_s)_E}{\left(2 + \text{St} \frac{A_s}{A_c}\right)_E (hA_s)_{ref}} \quad (2-37)$$

Equation (2-37) indicates that the relative heat conductance quantifies the increment in the heat removal capability when:

$$\frac{\left(2 + \text{St} \frac{A_s}{A_c}\right)_{ref}}{\left(2 + \text{St} \frac{A_s}{A_c}\right)_E} \approx 1 \quad (2-38)$$

2.7 Microchannel fabrication techniques

The four main microchannel fabrication techniques are photolithographic-based processes, X-ray micromachining, micromechanical machining and surface and surface-proximity-micromachining processing techniques [3].

Photolithographic-based processes involve the usage of UV light to produce a pattern on a substrate. A schematic illustration of this process is shown in Figure 2-7. It is most frequently used in semiconductor processing [87]. Photolithography comes in many variants depending on the light source being used. The material removal process is usually completed by etching processes. Etching is the chemical or physical removal of materials from a substrate, in a pattern defined by a protective mask layer such as a resist or an oxide [88]. Wet etching uses pools of liquid as the tools to remove materials in an anisotropic or isotropic manner. In anisotropic wet etching, potassium hydroxide solutions produce microchannels with specific orientations such as rectangular, trapezoidal or double-trapezoidal microchannels, depending on the crystallographic nature of the silicon used [89]. It is worth mentioning that a few of the prominent works [1, 90-92] in the microchannel studies utilise this microchannel fabrication method. Isotropic etching utilises hydrofluoric acid, nitric acid and acetic acid as the etchant. This technique etches in all crystallographic directions at the same rate, resulting in rounded isotropic features in single-crystalline silicon. Although wet etching techniques feature a low-cost batch production, these techniques are also not environmentally friendly.

Alternatively, dry etching techniques such as plasma-assisted dry etching and reactive-ion etching are gaining popularity mainly because they are more environmentally friendly and possess better critical dimension control as compared to the wet etching techniques [88]. Nevertheless, this technique requires expensive and specialised equipment, in addition to the need for a very clean operating line.

Figure 2-7: Schematic illustration of photolithography [93]

LIGA (Lithographie-Galvanoformung-Abformung) involves lithography, electroplating and molding, as shown in Figure 2-8. LIGA uses hard X-rays of around 0.1 nm. The high energy of the rays enables the highly precise manufacture of high aspect ratio micro-structures with very tight tolerances [88]. On top of that, this technology is insensitive to the nature of the substrate material [94], opening door to different design possibilities and materials such as ceramic and alloys [95]. The patterned resist can be used as a finished part or a molding master. However, X-ray lithography is considered to be expensive and time-consuming from design-to-realisation, especially the making of the mask [49, 96]. Furthermore, the mask is difficult to manufacture [94]. Therefore, LIGA is a secondary option if other fabrication methods are possible.

Figure 2-8: LIGA process [49]

The surface-proximity-micromachining method, such as epi-micromachining, are complementary methods to build multifunctional systems, such as system-on-chip. This method enables the implementation of sensors for active sensing, computation, fluidic components, onboard power sources and communication devices [49].

Micromechanical cutting includes micro-milling, micro-turning, micro-drilling and micro-grinding [97]. The former two are the most commonly applied methods [98]. These processes can achieve a minimum dimension of 10 to 100 μm [99]. Aptly suitable for customised individual production, these processes also feature good surface finish, high form accuracy, high machining speed and good adaptability to a huge range of materials such as steel, aluminium, brass, plastics, ceramics and polymers. Micromechanical cutting facilities feature decreased heat deformation, less material and energy consumption, smaller vibration amplitudes, smaller footprint and smaller space occupation [100], as compared to traditional machining methods. Nonetheless, as the micro-machining techniques are relatively new, the fundamentals of the process are still being understood and there is a minimal experience pool in the industry for these techniques [101]. Besides, tool cutting-edge geometry, grain size and orientation, which

are considered to have little influence at larger-scale cutting, affect the quality of finished products to a great extent [102]. Some other challenging issues include predictability, producibility and productivity [103]. More importantly, the working dimension of the micro-machining facilities is limited for macro-sized geometries.

Readily-available conventional machining methods for macro-sized geometries such as turning and milling appear to be attractive alternatives to replace the existing micro-fabrication techniques in producing a microchannel. Although ultra-precision manufacturing using these traditional computer-numerical-controlled (CNC) methods is possible, it is not energy efficient [101]. Therefore, the machining of the parts is restricted to near macro-size scale to ensure energy and resource efficiency. Thereafter, the superimposition of these macro-sized geometries yields a micro-sized channel. Kong and Ooi [8] first demonstrated this idea in 2013. Microchannels with nominal gap sizes of 1000 and 300 μm were realised by placing a solid cylinder with outer diameter 18.0 and 19.4 mm respectively, concentrically into another hollow cylinder with an inner diameter of 20.0 mm, as shown in Figure 2-9.

Figure 2-9: Implementation of an annular channel [8]

This work sheds light on an economical and simple manner of implementing microchannel. The geometries needed for this microchannel can be readily obtained from the market without the need of customised parts. This can be advantageous in terms of cost. Furthermore, the machining of these geometries requires a low technical level as only simple turning and milling processes are involved. The machining can be done in any mechanical workshop which does the mechanical cutting for macro-sized geometries. From another perspective, it raises the prospect of implementing microscale

heat transfer in macro-sized heat exchange systems given that the microscale heat transfer can be realised by placing two macro geometries at a distance in the order of 100 μm . Given the pumping power limit of existing industrial systems, this concept can be adopted to realise microscale heat transfer in these systems to improve the thermal performance without the need to replace the whole systems.

2.8 Enhanced heat transfer using macro geometries

Inspired by the success of implementing microchannel by using conventional machining methods, Goh and Ooi [10] explored the possibility of achieving enhanced heat transfer by using similar methods. They adopted biomimicry in their design of the surface profile on the inner cylinder, as shown in Figure 2-9, to enhance the heat transfer in the microchannel. In the view that fishes are naturally streamlined to minimise flow disturbance, they hypothesised that Inverted Fish Scale profile, as shown in Figure 2-10, can cause maximum flow disturbance which plays a crucial role in heat transfer enhancement. They conducted a parametric study on the e/H and P/e ratio.

Figure 2-10: Inverted Fish Scale parameters [10]

Goh and Ooi [11] subsequently examined the effect of fish scale profile, as shown in Figure 2-11, on the thermo-hydraulic performance. These profiles enhance heat transfer performance by promoting an early transition to turbulence. The effect of scale height is more prominent than the frequency of perturbation along the flow direction in promoting early turbulence. The critical Reynolds numbers for different e/H and P/e were reported.

Figure 2-11: Fish scale profile [11]

A preliminary study which includes all the nature-inspired profiles introduced by Goh and Ooi [104], suggested that a durian profile, as shown in Figure 2-12, has higher thermo-hydraulic performance as compared to the fish scale and inverted fish scale profile. A durian is a tropical fruit which has a thorn-covered husk. It resembles a wavy profile, showing potential to cause flow disturbance to enhance heat transfer. Therefore, Goh and Ooi [12] conducted a more detailed parametric study on this profile, varying e/H and P/e ratio.

Figure 2-12: Key parameters of Durian profile [12]

The maximum thermo-hydraulic performance index reported is near to 1.67, achieved by the profile with the highest e/H ratio at a constant P/e ratio of 10. On the other hand, at a constant e/H ratio of 0.5, a smaller P/e ratio favours the thermal-hydraulic performance at higher Reynolds numbers and vice versa.

2.9 Research gap

Single-phase liquid convective heat transfer in microchannels appears to be an effective heat removal technique for immediate heat flux applications. Although there have been numerous works on this topic, there is still a need for a balance between the enhancement in heat transfer and the unavoidable increment in pressure drop. This Ph.D. study

explores different enhanced channels and these channels are analysed for the thermo-hydraulic performances.

Among the heat transfer techniques, the channel curvature and re-entrant effects appear to be more relevant to micro-sized channels as compared to the others, owing to the small radius of curvature required and the relatively short length of the microchannels.

Inspired by the earlier implementations by Goh [105], this Ph.D. study first explores a single-wavy-wall configuration in Chapter 4. This configuration adopts the re-entrant effects as the hydraulic diameter varies along the flow direction, with a flat heating surface. Furthermore, this channel eases the implementation as it involves simple turning process and no modification to the existing heating surface is required.

Subsequently, the performance of this channel configuration is compared against a serpentine channel which realises the channel curvature technique, and a single-wavy-wall configuration with a heated wavy surface which combines both the re-entrant effects and the channel curvature techniques. This study is conveyed in Chapter 5.

After proven that single-wavy-wall channels have better thermo-hydraulic performances than the double-wavy-wall configurations, there is a need to improve the performances of these channels. Therefore, Chapter 6 investigates the effects of varying the waveform amplitude on the thermo-hydraulic performances. Although changing the relative waviness along the flow direction has been proven to be able to reduce the thermal resistance of the double-wavy-wall channels, the effects on the heat removal capability and the thermo-hydraulic performances of single-wavy-wall channels have yet to be investigated.

While conventional manufacturing methods such as CNC milling and turning have successfully implemented enhanced microchannels, the tolerance of the manufacturing becomes a concern. A loose tolerance and deformation during service result in an eccentric channel or a skewed channel. Although the effect of eccentricity has been well studied in conventionally-sized pipes, there is a lack of work in a short and small

gap where the flow mainly remains thermally developing. Hence, a study on eccentric and skewed micro-sized annuli is conducted and reported in Chapter 7.

Another channel configuration of interest is the converging flow channel. Converging flow channels have been proven as effective in promoting surface temperature uniformity. However, it has also been proven ineffective in terms of thermo-hydraulic performance. As the converging flow channels have been realised in a substrate, the convective heat transfer area, conductive heat transfer in the substrate and distribution of heat flux are inevitably changed. The current measurement system enables the measurement of the improvement in heat transfer coefficient eliminating the changes in the conductive and convective heat transfer area. This channel configuration is analysed through measured data in Chapter 8.

Chapter 3 Methodology

This chapter explains the research methodology employed in the studies. The first section details the implementation of the micro-sized gap. The second section describes the measuring system which includes the test loop, test module, measuring devices and data acquisition system. Pre-experimental and post-experimental procedures are also described in this section. The former includes metrology on the fabricated parts, calibration of the measuring devices, selection of experimental variables and validation of the measured data. The latter entails data reduction and uncertainty analysis. The third section presents the numerical models which are employed in the studies. The discretised test module, boundary conditions, selection of the turbulence model and discretisation scheme, mesh independence tests, data reduction and validation of the predicted data are discussed in this section.

3.1 Implementation of the microchannel

3.1.1 Straight channel

This study works on an annular microchannel, formed by superimposing a hollow copper cylinder and a solid stainless steel cylinder concentrically [106], as shown in Figure 3-1. These macro-sized geometries are machined using readily-available conventional computer numerical controlled methods such as milling and turning processes.

Microchannels of different hydraulic diameter, as defined in Equation (3-1), can be formed by varying the outer diameter of the inner solid cylinder. The mineral-insulated heating cable is housed permanently in the copper cylinder and thus the inner diameter of the hollow channel is fixed at 20.0 mm in the test module. The length of the channel is also fixed at 30.0 mm. Insert holders and O-rings are present in the test module to ensure the concentricity of the channel and the details are elaborated in Section 3.2.2.

This straight channel is used as the benchmark for the comparison of thermal and hydrodynamic performance for various studies conducted, as described in the subsequent chapters.

$$D_h = D_o - D_i \quad (3-1)$$

Figure 3-1: Formation of annular flow channel

3.1.2 Enhanced channel

Figure 3-2 illustrates the cross-sectional views of the enhanced microchannels. There are two configurations of enhanced channel which are being investigated.

For the first configuration, the surface profile is introduced on the surface of the solid cylinder which is placed concentrically into the hollow cylinder, as shown in Figure 3-2(a). As the heat is being supplied to the hollow copper cylinder, the heat transfer surface area remains constant. This configuration is not uncommon in the heat transfer applications. Twisted tapes are inserted in the existing heat exchangers to upgrade their heat duties and in new designs for size reductions [107]. These inserts promote vortex motion or induce turbulence and thereby, increasing the heat transfer coefficient.

Modifications to the heating surface are not always possible or cost-friendly in an existing implementation and the machining space becomes limited as the dimension of the new device scales down, particularly when it involves machining on an internal surface. Therefore, it is of practical value to explore and quantify the heat transfer performance of such configuration where the passive heat transfer enhancement is realised by modifying the non-heated surface.

The second configuration involves modifications to the copper surface forming the annular channel with the solid cylinder, as shown in Figure 3-2(b). This results in a configuration where the modifications are made to the heated surface.

Both configurations realise a flow channel with a varying flow gap, aiming to create flow disturbance to increase the heat transfer performance. The design parameters of the surface profile are elaborated in subsequent chapters, investigating different enhanced microchannels.

Figure 3-2: Cross-sectional view of an enhanced microchannel

3.1.3 Eccentric channel

Eccentric channels are studied numerically. The eccentricity is formed by introducing an offset between the center of the solid cylinder and the hollow cylinder. The eccentric ratio is defined in Equation (3-2) and the parameters are defined in Figure 3-3. Eccentric ratios range from 0.25 to 0.75 are studied in Chapter 7.

$$\varepsilon = \frac{\delta}{r_o - r_i} \quad (3-2)$$

Figure 3-3: Front view of an eccentric channel

3.1.4 Skewed annulus

A skewed annular is defined as an annular having changing eccentricity along the flow direction. This results in a changing gap size along the flow direction, as shown in Figure 3-4. This channel configuration is studied in Chapter 7 where the design parameters are presented.

Figure 3-4: Comparison between a skewed annulus on the left and an eccentric annulus on the right (Cross-sectional view)

3.1.5 Converging flow channel

Figure 3-5 shows a converging flow channel. The outer diameter of the inner solid cylinder increases linearly from the inlet to the outlet of the channel. This results in a

decreasing hydraulic diameter along the flow direction. Four different gradients of convergence are described and studied in Chapter 8.

Figure 3-5: Cross-sectional view of a converging flow channel

The solid cylinder, referred to as an insert, has a front and rear holding part in the test module, as shown in Figure 3-6. The protruded ends are slotted into the insert holders to hold the insert securely in place, as depicted in Figure 3-8.

Figure 3-6: An insert

3.2 Experimental method

The experimental method involves data collection and reduction from the measuring devices installed in the test loop and test module. The experimental loop employed in this study, as shown in Figure 3-7, is designed by Goh [105]. More descriptions on the design considerations and limitations of the current experimental loop and equipment can be found in [105]. These limitations are the important considerations in deciding the

measurement range of this work. Some of the specifications are included here to justify the constraints on the range of the measured data.

3.2.1 Experimental loop

Figure 3-7 illustrates the experimental test loop which consists of a test loop and a bypass loop. The test module is installed along the test loop, together with five Type-T thermocouples, five WIKA A-10 pressure transmitters and a Coriolis flow meter which are connected to the data acquisition system. An additional ten Type-J thermocouples are incorporated in the test module. These devices are detailed below.

Distilled water is used as the working fluid and is stored in the integrated reservoir in the chiller unit. The air-cooled chiller cools the water to the desired inlet temperature and a positive displacement pump channels the liquid water into the test loop where the test module is installed. The volumetric flow rate is controlled by the by-pass loop and a needle valve along the test loop. The bypass loop also serves to remove the air bubbles in the liquid water. A 40-micron inline filter is used to remove solid particulates in the water while the working fluid is regularly replaced.

Figure 3-7: Experimental test loop [10]

3.2.2 Test Module

The test module, as shown in Figure 3-8, consists of two insert holders, two insulation pipes, one copper block with embedded mineral-insulated heating cable, one insulation

cover and an insert. The insert is placed concentrically into a copper block with an inner hollow channel, forming an annular gap. The concentricity of the channel is ensured by restricting the radial tolerance to $\pm 10 \mu\text{m}$. ISO 3601 A0035 O-rings are also used in the insert holders to tightly secure the insert in place. The mineral-insulated heating cable with a maximum capacity of 3000 W is housed within the groove on the surface of the copper block. The heat input is regulated using a variac and the current is monitored using an ammeter. Ten slots are present in the copper block, five on each side of the microchannel, evenly spaced to house the Type-J thermocouples. The insulation pipes and insulation cover are employed to minimise the heat loss to the surrounding, maximising the heating efficiency of the test module. The fluid enters the microscale gap and removes heat from the copper wall, as shown in Figure 3-9.

Figure 3-8: Cross-sectional view of the test module

Figure 3-9: Heat removal from copper wall in a straight microchannel

3.2.3 Measuring Devices

The measuring devices which are employed in the test loop are Type-J thermocouples, Type-T thermocouples, pressure transmitters, flow meter and ammeter. These devices are calibrated or validated to ensure the accuracy of the results obtained.

3.2.3.1 Type-J Thermocouple

Mineral-insulated Type-J thermocouples, based on the DIN EN 60584 standard, have a measuring range from -40 to 750 °C. These thermocouples are calibrated at five temperature points in the expected working temperature range of 10 to 110 °C. The calibration process is elaborated in the subsequent section. Characterisation functions are formulated into the data acquisition system to correct the raw measured temperature values. These functions have r-squared values of 1.000, indicating a good representation of the characterisation function at the calibrated temperature points. The standard error of estimate, as shown in Table 3-1, quantifies the uncertainty of the measured temperature values obtained through the calibration functions. The uncertainty values which are derived from the calibration process are discussed in the subsequent section and have been taken account into the uncertainty analysis.

Table 3-1: Calibration results for Type-J thermocouples

Thermocouple	Characterisation function	Standard Error of Estimate (S.E.E.)
TCJ-01	$y = 1.0105x - 2.9377$	0.136
TCJ-02	$y = 1.0076x - 1.7715$	0.163
TCJ-03	$y = 1.0098x - 2.4582$	0.0953
TCJ-04	$y = 1.0079x - 1.8534$	0.123
TCJ-05	$y = 1.0096x - 2.4563$	0.0411
TCJ-06	$y = 1.0063x - 1.3123$	0.138
TCJ-07	$y = 1.0105x - 2.5668$	0.0871
TCJ-08	$y = 1.0080x - 1.7864$	0.120
TCJ-09	$y = 1.0085x - 1.9151$	0.102
TCJ-10	$y = 1.0087x - 1.9812$	0.114

3.2.3.2 Type-T Thermocouple

As compared to Type-J thermocouples, Type-T thermocouples have a smaller working temperature range of -40 to 350 °C. However, as the temperature of the liquid water is expected to increase up to a maximum of 40 °C at the lowest flow rate, these devices are calibrated at five temperature points with the range of 10 to 50 °C. The calibration results for these devices are tabulated in Table 3-2. These characterisation functions have equally good r-squared values of 1.000.

Table 3-2: Calibration results for Type-T thermocouples

Thermocouple	Characterisation functions	Standard Error of Estimate (S.E.E.)
TCT-01	$y = 1.0012x - 0.1532$	0.0143
TCT-02	$y = 1.0010x + 0.1857$	0.0135

3.2.3.3 Pressure transmitters

WIKA A-10 pressure transmitters are used to measure the local fluid pressure. This model of pressure transmitter has a working pressure range of 0 to 6 bar(g), powered by

a 24 V DC Agilent power supply unit. It outputs a current from 4 to 20 mA which is translated into pressure reading through the data acquisition system. The measuring sensors are installed on a hydraulic deadweight tester where the standard weights are placed on the piston, at an increment of 0.5 bar for the whole working pressure range, to calibrate the output readings on the data acquisition system. This calibration device is further described in the next section. The characterisation functions for both pressure transmitters and the corresponding errors are shown in Table 3-3.

Table 3-3: Calibration results for pressure transmitters

Pressure transmitter	Characterisation functions	Standard Error of Estimate (S.E.E.)
PT-01	$y = 375.3x - 1.4945$	0.00131
PT-02	$y = 375.9x - 1.4948$	0.00120

3.2.3.4 Flow meter

The flow meter that is used is Coriolis flow meter. The selected Tricor TCM 0650 Coriolis flow meter is capable to measure flow rate range of 0.1 to 10 l/min, within an uncertainty of ± 0.1 % of reading.

3.2.3.5 Ammeter

An ammeter is used to measure the magnitude of current supply to the heater. The current together with the reduced resistance of the heater are used to approximate the wattage of the heater according to $I^2 R$. However, this device is not calibrated as the data obtained is not used in the data reduction process. The exact heat supply of the heater is calculated using the energy balance equation, explained in Section 3.2.5. Nevertheless, another probe is used to countercheck the readings to ensure that it is functioning normally.

3.2.3.6 Data acquisition system

The data acquisition system comprises of two National Instruments (NI) 9213 modules, one NI 9208 module, NI LabVIEW software and Hewlett-Packard xw4600 workstation.

All the thermocouples, both Type-T and Type-J, are connected to the NI 9213 module. Internal reference junctions are used for the cold-junction compensation. Therefore, extra care has been taken to ensure that the module is far from heat sources which can affect the cold-junction compensation sensor. The error incurred by the internal cold-junction compensation has been accounted for by using the same data acquisition system during the calibration process.

NI 9208 module, which is powered by a 24 V DC power supply, acquires data from the flow meter and pressure transmitters. These modules are housed in a 4-slot chassis NI cDAQ-9174 which is connected to the workstation. The raw data is corrected by the characterisation functions and the key parameters are output directly on the LabVIEW interface for monitoring purposes.

3.2.4 Metrology

3.2.4.1 Surface roughness

The surface roughness of the components employed in this study is crucial as surface roughness is one of the ways to acquire heat transfer enhancement, particularly when the size effect is dominant in a microscale channel [57]. However, only smooth channels are being considered in this study. The dimensionless surface roughness is given by [108]:

$$\varepsilon^* = \varepsilon \frac{u_\tau}{\nu} \quad (3-3)$$

where u_τ is computed as:

$$u_\tau = \left(\frac{\tau_w}{\rho} \right)^{0.5} = \left(\frac{f}{8} \right)^{\frac{1}{2}} \left(\frac{4\dot{m}}{\pi\rho(D_o^2 - D_i^2)} \right) \quad (3-4)$$

A threshold of $\varepsilon^* = 5$ is proposed for a channel to be considered hydraulically smooth [43]. The surface roughness is examined by SurfTest SJ 301 Stylus Profilometer with a resolution of 0.01 μm . Five readings are taken at random positions for each of the specimens. The surface roughness of the components in the test module has been measured by Goh [105] and reported as below:

Table 3-4: Mean surface roughness of the components in test module

Component	Mean surface roughness, R_a (μm)
Copper block (inner channel)	1.58
PEEK insulation pipe 1	1.83
PEEK insulation pipe 2	1.56

The surface roughness of the inserts is also being considered and the maximum dimensionless surface roughness is reported in separate studies.

3.2.4.2 Dimensions

The dimensions of the copper block which constitutes the microchannel, as described in Section 3.1, have been measured by Goh [105]. The values, tabulated in Table 3-5, are taken into consideration for the actual size of the microchannels.

Table 3-5: Measured key dimensions

Parameter	Nominal value (mm)	Actual value (mm)
Inner diameter of the copper block	20.000	19.969
Microchannel length	30.00	30.00

The external diameters of the fabricated inserts are inspected using IM-EC1-071132 electronic calliper, with a resolution of 0.01 mm. The actual microchannel sizes are presented in Chapter 4, 6 and 8 where the inserts are fabricated for the realisation of the microchannels in the measurement system.

3.2.5 Calibration

3.2.5.1 Calibration Systems

The calibration systems are described briefly here as the specifications are needed for the calculation of uncertainty values which are discussed in the subsequent section. Fluke 7103 Micro-bath Thermocouple Calibrator and Fluke Calibration 5628 Platinum Resistance Thermocouple are used for the calibration of both Type-J and Type-T thermocouples. The values reported for Fluke Calibration 5628 Platinum Resistance

Thermocouple (PRT) are obtained from a report from Fluke South East Asia Pte. Ltd., based on the calibration done on the PRT in the laboratory. The specifications of these devices are presented in Table 3-6 and Table 3-7 respectively.

Table 3-6: Specifications of Fluke 7103 Micro-bath Thermocouple Calibrator

Parameters	Description
Temperature range	-35 to 125 °C
Temperature stability	± 0.03 at -25 °C ± 0.05 at 125 °C
Temperature uniformity	± 0.02 °C
Working fluid	Silicone oil

Table 3-7: Specifications of Fluke Calibration 5628 Platinum Resistance Thermocouple

Parameters	Description
Temperature range	-200 to 661 °C
Calibrated uncertainty	± 0.0085 at -195.65 °C ± 0.0064 at -39.91 °C ± 0.0026 at 0.01 °C ± 0.019 at 232.05 °C ± 0.021 at 419.72 °C ± 0.014 at 660.32 °C
Long term drift	< 0.004 °C/ 100 hours at 661 °C

The Flotech deadweight tester is used to calibrate the pressure transmitters. It has an uncertainty of 0.015 % and calibrating range of 1 to 35 bar.

The same data acquisition system for the measurement process, as described in the preceding section, is used during the calibration processes to take into consideration the uncertainties incurred during the data logging process. This includes resistance of the thermocouple wires and cold-junction temperature. The latter can incur an error up to 1.7 °C and an average of 0.8 °C for the operating range from 0 to 70 °C [109].

3.2.5.2 Calibration Process

There are two types of thermocouple calibration, namely tolerance testing and characterisation. Tolerance testing involves measuring the voltage output at various temperatures and comparing the error to that listed in the American Society for Testing and Materials (ASTM) error ratings. This method ensures that the thermocouple adheres to the standard model and thus requires no data fitting or complex calculations. Characterisation method, which is more suitable for high accuracy applications, is chosen as it quantifies the difference between measured and standard voltage, followed by a curve fitting to correct the discrepancy. As the data acquisition employs an internal cold-junction compensation, the output temperature is characterised to the standard temperature directly.

During the calibration process, enough immersion is ensured to prevent stem losses as the thermocouple junctions are at the tip of the probe. An immersion depth of 15 times the probe diameter plus the length of the sensing element is recommended for probes with thin wall construction and situations with good heat transfer [110]. The probes are placed as close as possible without getting into contact and on the same plane.

3.2.6 Data reduction

3.2.6.1 Reference state fluid properties

The temperature of the fluid varies with the position in the microchannel. It is convenient to take the mean fluid temperature for the evaluation of fluid properties as the average of the inlet and outlet temperature [34] :

$$T_{f,m} = \frac{T_{in} + T_{out}}{2} \quad (3-5)$$

Another reference state taken for the fluid properties is the average fluid pressure which is measured during the experimental run:

$$p_{f,m} = \frac{p_{in} + p_{out}}{2} \quad (3-6)$$

However, it should be noted that pressure has insignificant influence on fluid properties. Patek et al. [111] stated that the change in property due to pressure is smaller by more than one order of magnitude than the uncertainty of the property itself. Hence the selection of property quoted at different fluid pressure yields no difference in accuracy, for pressure range between liquid saturation pressure and 0.3 MPa.

The only exception is the density which is evaluated at the outlet fluid temperature. This is because the volumetric flow rate is measured by the Coriolis flow meter which is installed downstream of the test module in the test loop.

The data is acquired for a 10-minute steady-state condition, at a frequency of 1 Hz. The steady-state condition range is set as 0.3 °C, 0.04 bar and 0.05 l/min for temperature, pressure and volumetric flow rate readings respectively.

3.2.6.2 Hydraulic diameter

The data reduction adopts the hydraulic diameter of concentric annular channels as the hydraulic diameter of the microchannel:

$$D_h = D_o - D_i \quad (3-7)$$

This is consistent with that fact that concentric annular channel can be approximated as parallel plate configuration with $D_h = 2 \times \text{gap size}$, when $r^* \geq 0.5$ [35]. r^* is defined as the ratio of the inner radius to the outer radius:

$$r^* = \frac{r_i}{r_o} \quad (3-8)$$

3.2.6.3 Hydrodynamic data

The pressure drop is deduced from the pressure difference between the inlet and outlet of the test module:

$$\Delta p_{\text{module}} = p_{in} - p_{out} \quad (3-9)$$

In this study, the pressure drop across the test module is approximated as the pressure drop of the channel:

$$\Delta p_{\text{module}} \approx \Delta p_{\text{channel}} \quad (3-10)$$

Table 3-8 shows that the difference between Δp_{module} and $\Delta p_{\text{channel}}$ for the simulated channels falls below 14.0%, even at lower flow rates. This approximation eliminates the potential errors due to the numerical difference between measured and predicted data.

Table 3-8: Simulation results for Δp_{module} and $\Delta p_{\text{channel}}$

Channel	Flow rate (L/min)	$\Delta p_{\text{channel}}$ (Pa)	Δp_{module} (Pa)	$\frac{\Delta p_{\text{channel}}}{\Delta p_{\text{module}}}$
straight	2	7744.2	8822.1	0.878
straight	3	13061.4	14981.2	0.872
straight	4	19849.2	22778.2	0.871
sin_005_3	3	12659.5	14726.5	0.860
sin_010_3	3	29167.9	33375.4	0.874
sin_015_3	3	73589.0	76675.4	0.960
sin_015_6	3	40406.0	43138.2	0.937

The friction factor in the present study is defined by:

$$f = \frac{\Delta p_{\text{channel}}}{\frac{1}{2} \rho V_m^2} \frac{D_h}{L} = \frac{\Delta p_{\text{channel}}}{\text{Re}^2} \left(\frac{2D_h^3}{L} \times \frac{\rho}{\mu^2} \right) \quad (3-11)$$

The Reynolds number for the annular channel is computed by:

$$\text{Re} = \frac{\dot{m} D_h}{A_c \mu} = \frac{4 \rho \dot{V}}{\pi (D_o + D_i) \mu} \quad (3-12)$$

The pumping power required is given by:

$$\Omega = \Delta p_{\text{channel}} \times \dot{V} \quad (3-13)$$

3.2.6.4 Heat transfer data

To account for the heat loss to the surrounding, the heat transfer rate to the liquid water is calculated through the energy balance equation, as the flow work is negligible:

$$Q = \rho \dot{V} c_p (T_{out} - T_{in}) \quad (3-14)$$

This actual heat supply is ensured to be within 5 per cent of the nominal heat supply.

The wall temperature is deduced from the Type-J thermocouples, taking the thermal resistance of the copper block in the radial direction into consideration:

$$T_{w,i} = T_{r,i} - Q \frac{\ln(D_r/D_w)}{2\pi L k_{cu}} \quad (3-15)$$

Therefore, the wall temperature is the average of the ten readings:

$$T_{w,m} = \frac{1}{10} \left(\sum_{i=1}^{10} T_{w,i} \right) \quad (3-16)$$

The average heat transfer coefficient is then obtained from Newton's law of cooling:

$$h = \frac{Q}{A_s (T_{w,m} - T_{f,m})} \quad (3-17)$$

The average Nusselt number is calculated from the average heat transfer coefficient:

$$Nu = \frac{h D_h}{k_{fluid}} \quad (3-18)$$

3.2.6.5 Thermo-hydraulic performance

This section presents the relative heat transfer rate between two channels under comparison. The author wishes to credit this piece of work to Mr. Toh Zheng Yong for the incredible input.

The following conditions are assumed for the subsequent derivations:

- i. Same inlet temperature
- ii. Insignificant difference in the specific heat capacity, density and average heat transfer coefficient when the heat input varies
- iii. Same pumping power input for both channels
- iv. Negligible flow work as compared to the heat input

Although the viscosity of the water is a strong function of the temperature, the effect of the viscosity on the Nusselt number is minimised due to the opposing effects on the Reynolds number and Prandtl number. Figure 3-10 depicts the heat transfer coefficient of a straight channel under two different heat fluxes. Although the heat flux is doubled, the difference in h varies from 2.9 to 6.0%. Compounded with the uncertainties in the measuring sensors, Assumption (ii) is reasonable.

Figure 3-10: Difference in heat transfer coefficient with different heat input

The average wall temperature of a channel can be obtained from the Newton's law of cooling:

$$T_w = \frac{Q}{hA_s} + \left(\frac{T_{in} + T_{out}}{2} \right) \quad (3-19)$$

Rearranging the energy balance equation, assuming negligible flow work:

$$T_{out} = T_{in} + \frac{Q}{\dot{m}c_p} \quad (3-20)$$

Substituting Equation (3-20) into(3-19):

$$T_w - T_{in} = Q \left(\frac{1}{hA_s} + \frac{1}{2\dot{m}c_p} \right) \quad (3-21)$$

Equation (3-22) compares the heat removal from both channels with the same average wall temperature and fluid inlet temperature.

$$\frac{Q_E}{Q_{ref}} = \frac{\left(\frac{1}{hA_s} + \frac{1}{2\dot{m}c_p} \right)_{ref}}{\left(\frac{1}{hA_s} + \frac{1}{2\dot{m}c_p} \right)_E} \quad (3-22)$$

Equation (3-22) can be re-written as:

$$\frac{Q_E}{Q_{ref}} = \frac{\left(2 + \frac{h}{\rho V c_p} \frac{A_s}{A_c} \right)_{ref} (hA_s)_E}{\left(2 + \frac{h}{\rho V c_p} \frac{A_s}{A_c} \right)_E (hA_s)_{ref}} \quad (3-23)$$

This can be further simplified into:

$$\frac{Q_E}{Q_{ref}} = \frac{\left(2 + St \frac{A_s}{A_c} \right)_{ref} (hA_s)_E}{\left(2 + St \frac{A_s}{A_c} \right)_E (hA_s)_{ref}} \quad (3-24)$$

Equation (3-24) computes the increment in the heat removal capability of the channel of interest at the same pumping power as the reference channel.

The computation of this parameter requires a linear interpolation between neighbouring data points to obtain the heat transfer coefficient and Stanton number at the same pumping power.

3.2.7 Uncertainty Analysis

3.2.7.1 Standard Uncertainty

Uncertainty arises due to the lack of exact knowledge of the value of the measurand. Although efforts have been made to correct the recognised systematic error, the measurand is still an estimate of the quantity of interest due to random effects during the data logging process and imperfect correction of the system for the systematic effects.

Uncertainties are grouped into two categories, namely Type A and Type B, based on the two different evaluation methods. Both types of evaluation are based on the probability distribution and are quantified by variances or standard deviations. The former is

obtained from a probability density function derived from an observed frequency distribution whereas the latter is obtained from an assumed probability distribution based on the judgment that an event will occur. The judgment is invariably made based on a pool of comparatively reliable information such as previous measurement data, manufacturer's specifications and handbooks.

3.2.7.2 Type A evaluation of standard uncertainty

The arithmetic mean of n data points under the same measurement conditions, as shown in Equation (3-25), is often the best available expected value of a quantity. The individual data points differ in value attributed to the random variations in the influence quantities and the dispersion about their respective mean is quantified through experimental standard deviation using Equation (3-26). The experimental standard deviation of the mean, calculated using Equation (3-27), is an indicator of how well the mean estimates the expected value of the measured quantity and is a measure of the uncertainty of mean value. Therefore, it is called Type A standard uncertainty.

$$\bar{q} = \frac{1}{n} \sum_{i=1}^n q_i \quad (3-25)$$

$$s(q) = \sqrt{\frac{1}{(n-1)} \sum_{i=1}^n (q_i - \bar{q})^2} \quad (3-26)$$

$$s(\bar{q}) = \frac{s(q)}{\sqrt{n}} = \sqrt{\frac{1}{n(n-1)} \sum_{i=1}^n (q_i - \bar{q})^2} \quad (3-27)$$

3.2.7.3 Type B evaluation of standard uncertainty

In contrast to Type A evaluation method, Type B evaluation is based on scientific judgment on different types of sources mentioned above. These sources may exist in different types of the probability distribution function. It is important to recognise the general type of assumed probability distribution and the corresponding standard deviation, which is listed in Table 3-9.

Table 3-9: Common types of probability distribution [112]

Probability distribution	Standard deviation/ Uncertainty
Normal distribution (99.73 % level of confidence)	$u = \frac{a}{3}$
Rectangular distribution	$u = \frac{a}{\sqrt{3}}$
Triangular distribution	$u = \frac{a}{\sqrt{6}}$

where a is the half-width of the interval

The items considered for the uncertainty budget for Type B evaluation are as follows:

- PRT calibration uncertainty
- PRT drift with an approximation of 500 working hours
- Bath temperature uniformity
- Bath temperature stability
- Standard estimate of error of the calibration function

The values for Type B uncertainties are reported in Appendix A.

The computed uncertainty values are presented in the respective chapters where the measured data is reported.

3.2.8 Validation of measured data

An annulus can be conveniently approximated to a circular duct by using the concept of hydraulic or wetted diameter. Therefore, the measured friction factor is validated using Equation (2-11) and (2-12), as shown in Figure 3-11. The variation of viscosity in the measurement owing to the heating process has also been accounted for using Equation (2-31). The difference of friction factor falls within ± 10.0 %.

Figure 3-11: Validation of measured friction factor

However, validation of heat transfer data using correlations of the circular duct may result in over-prediction. This is mainly due to the current implementation of the channel: the outer wall is heated whereas the inner wall has a conservative heat flux condition. An approximation to a circular duct results in an annulus with both walls heated. Therefore, the heat transfer data is compared with Gnielinski's correlations for an annular channel, as shown in Equation (2-27) and (2-28), for laminar and transitional flow regime respectively. This comparison is done in Figure 3-12. These equations, however, assume an isothermal boundary condition with outer wall heated and an adiabatic inner wall, as explained in Section 2.2. The difference in Nusselt number falls within $\pm 15.0\%$.

Figure 3-12: Validation of measured Nusselt number

3.3 Numerical methods

Numerical methods are used to predict the fluid and heat flow in the test module. The governing equations are solved using commercially available solvers ANSYS CFX and ANSYS FLUENT in the studies. Although both solvers use finite volume method which discretises the spatial domain using a mesh, CFX uses the vertex-centred method while FLUENT adopts the cell-centred method, as shown in Figure 3-13. In the cell-centred approach, the cells serve as control volumes and the average values are stored in the centre of the cell. For the vertex-centred method, the control volume is formed by combining sub-volumes around the vertex and the variable values are stored in the vertex.

Figure 3-13: Vertex-centred control volume (left) versus the cell-centred control volume (right) [113]

3.3.1 Governing equations

A three-dimensional convective heat transfer is being modelled in this study. Therefore, the governing equations which are solved in this study are continuity equation, momentum equation, energy equations for both solid and fluid domains and the empirical models for transitional and turbulent flows. The steady-state continuity equation is written as:

$$\nabla \cdot (\rho \mathbf{V}) = 0 \quad (3-28)$$

The momentum equation for steady-state incompressible flow and in the absence of body forces is expressed as:

$$\nabla \cdot (\rho \mathbf{V} \otimes \mathbf{V}) = -\nabla p + \nabla \cdot \left[\mu \left(\nabla \mathbf{V} + (\nabla \mathbf{V})^T - \frac{2}{3} \delta \nabla \cdot \mathbf{V} \right) \right] \quad (3-29)$$

The thermal energy equation, applied to both solid and incompressible fluid, neglecting viscous dissipation and body forces, is written as:

$$-\nabla \cdot (\rho h_e \mathbf{V}) = \nabla \cdot (k \nabla T) \quad (3-30)$$

3.3.2 Boundary conditions

The boundary conditions are reported in the respective studies as there are slight differences owing to the difference in geometry and simplification. The solid and fluid properties which are universal to the studies are included here. The properties for the solid domains are assumed constant, as presented in Table 3-10.

Table 3-10: Properties of the solid domains

Domains	Properties		
	ρ (kg/m ³)	c_p (J/kg·K)	k (W/m·K)
Insert and Holders	7854	434	60.50
Copper block	8933	385	401.00
PEEK insulation pipes	1400	1800	0.92
MICA insulation cover	2100	500	0.31

International Association for the Properties of Water and Steam (IAPWS) Industrial Formulation 1997 [114] standards are adopted for the variation in fluid properties. This database is integral to ANSYS CFX. In ANSYS FLUENT where direct adoption of the database is not possible, the regression done by Hong et al. [115] based on the aforementioned standards for the temperature range of 283 to 373 K is applied. These equations are shown in Table 3-11.

Table 3-11: Thermophysical properties of water [115]

Properties	Unit	Function
μ	Pa·s	$0.0194 - 1.065e^{-4} \times T + 1.489e^{-7} \times T^2$
k	W/m·K	$-0.829 + 0.0079 \times T - 1.04e^{-5} \times T^2$
c_p	J/kg·K	$5348 - 7.42 \times T + 1.17e^{-2} \times T^2$
ρ	kg/m ³	998.2

Note: The unit of T is Kelvin (K).

3.3.3 Turbulence modelling

Turbulent flow is a complex phenomenon due to the fact that it is three-dimensional, unsteady and consists of many scales, both in terms of time and length [116]. These characteristics are, nevertheless, attribute to many engineering phenomena at high Reynolds numbers. In principle, the Navier-Stokes equation describes both laminar and turbulent flows without any additional information. However, the huge number of time and length scales render it impossible to acquire a full numerical resolution of these flows through Direct Numerical Simulation (DNS).

The other alternatives to solving Navier-Stokes equation involving turbulent flows are Large-Eddy Simulation (LES) and Reynolds Averaged Navier-Stokes (RANS) equation. LES solves spatial- averaged Navier-Stokes equation in which large eddies are solved directly whereas the eddies smaller than the mesh are modelled. Therefore, LES is more computationally efficient than DNS.

However, the cost of implementing LES is still relatively higher as compared to RANS. For most engineering applications, RANS is sufficient to predict turbulent flows. Therefore, RANS is selected over DNS and LES in the studies [117].

Contrary to LES, RANS equation is time-averaged Navier-Stokes equation. This form of Navier-Stokes equation was introduced by Reynolds [118] in 1895. The properties in Reynolds averaging are decomposed into two components, a mean and a fluctuating component. The modification introduces an additional component called Reynolds stresses, $-\rho \overline{V_i' V_j'}$ in RANS equation and thus incurs closure problems. Turbulence modelling is thus employed to implement various assumptions to evaluate Reynolds stresses. Eddy viscosity models are the most extensively used models for turbulent flow predictions based on RANS method [117].

3.3.3.1 Eddy Viscosity Turbulence Models

The main idea behind eddy viscosity turbulence models is Boussinesq's eddy-viscosity concept which assumes that the turbulent stresses are proportional to the mean velocity gradient. These models treat turbulent eddies as the carriers of thermal energy and momentum, in a similar way that the molecules are treated and analysed in kinetic theory [119]. The Reynolds stresses are modelled as in Equation (3-31) for eddy viscosity turbulence models.

$$-\overline{V_i' V_j'} = \nu_t \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) - \frac{2}{3} k \delta_{ij} \quad (3-31)$$

$$k = \frac{1}{2} \left(\overline{u'^2} + \overline{v'^2} + \overline{w'^2} \right) \quad (3-32)$$

where ν_t is the turbulent or eddy viscosity and k is the turbulent kinetic energy. ν_t depends strongly on the turbulence state and not a fluid property.

When Equation (3-31) is implemented in the momentum equation, the second portion of the eddy viscosity equation is incorporated into the pressure term and hence, determination of the eddy viscosity is the main objective. Turbulence models, ranging from simple zero-equation models to two-equation models seek a prescription for the eddy viscosity to model Reynolds stresses. In the present study, the two-equation model

is used as it offers a good compromise between numerical effort and computational accuracy [120]. Both the velocity and length scale are solved using separate transport equations.

3.3.3.2 Two-equation Turbulence Models

k - ε model and k - ω model are the most commonly used two-equation turbulence models. Improvement to the standard k - ε model yields RNG k - ε and realizable k - ε models. Similarly, baseline model (BSL) k - ω model and shear stress transport (SST) are more versatile versions of the k - ω model. These models are discussed below.

k - ε models

In the standard k - ε model, the turbulence viscosity is evaluated using the expression:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (3-33)$$

The transport equations for the evaluation of k and ε can be found in [120, 121]. The standard k - ε model, however, is only accurate at high Reynolds numbers and in the absence of large rates of deformation [122]. The RNG (Renormalisation Group) k - ε model extends the k - ε model to a wider class of flows with higher accuracy, including rapidly strained flows, swirling flows and flows at lower Reynolds number [121]. The transport equations are the same as that of the standard model. An additional term is introduced to account for the interaction between turbulence dissipation and mean shear.

As compared to the standard and RNG k - ε models, realizable k - ε model satisfies the mathematical constraints on the Reynolds stresses, consistent with the physics of turbulent flows. This model introduces a new eddy-viscosity formula and a modified ε -equation.

k - ε models, in general, do not predict the separation zones well. The model predicts an excessive level of turbulent stress particularly in the presence of adverse pressure gradients, leading to the suppression of separation on curved walls. Furthermore, it does not give a good prediction in the stagnation or near wall-regions owing to the excessive level of turbulence [122]. Therefore, the k - ω model outperforms k - ε model in the

boundary layer modelling, particularly in the sublayer and logarithmic zone [123]. It also eliminates the complex non-linear damping functions required for the k - ε model and therefore, is more accurate and robust [120].

k - ω models

The k - ω model assumes that the turbulence viscosity is linked to the turbulence kinetic energy and turbulent frequency, ω through Equation (3-34). The k - ω model developed by Wilcox [124, 125] is adopted in ANSYS CFX and FLUENT [120, 121]

$$\mu_t = \rho \frac{k}{\omega} \quad (3-34)$$

Nevertheless, this model is sensitive to free stream conditions where it is very dependent on the ω specified for the inlet [126]. Baseline (BSL) k - ω model is implemented to address this issue by applying the k - ω model near the surface and the k - ε model in the outer region [127]. A blending function is introduced for the transformation. As a result, the BSL model is significantly independent on the free stream conditions.

SST further improves the BSL model which fails to predict the onset of flow separation by placing a limiting function to the eddy viscosity. This modification accounts for the transport of principal turbulent shear stress which is important in predicting adverse pressure gradient flows.

3.3.4 Selection of turbulence model

As the near-wall conditions such as friction factor and heat transfer coefficient are of our concern, the more versatile version of k - ω model, i.e. the SST model is selected for the prediction of turbulent characteristics. The empirical model which includes the transport equations and the constant can be found in [120]. To benefit from this model, a minimum resolution of 10 layers is needed in the boundary layer [120].

In FLUENT, an intermittency transition option is available to model transitional flows as well as flows in the low freestream turbulence environments [121]. Therefore, this option is coupled with the SST model for the modelling of the flows at low Reynolds

numbers. This model is compared with the measured data in Chapter 6 to validate the accuracy for low Reynolds number applications.

3.3.5 Discretised test module

A multi-scale approach is adopted in the discretisation of the test module. This implies that elements of different sizes are generated, depending on the respective length scale of the domain and the physics involved, as depicted in Figure 3-14. Mesh independence tests are conducted to ensure that the size chosen for each of the domain does not affect the predicted results. The mesh independence tests for the individual studies are reported in Chapter 4 to 7 respectively. A finer mesh is required for the flow in the microscale gap as the boundary layers need to be resolved accurately.

The General Grid Interface (GGI) has been enabled in the vortex-centred solver CFX to connect the non-matching grids between the solid and fluid domain. These two domains are coupled in FLUENT to allow for conservative flux boundary conditions.

The inflation layer is created to resolve the boundary layer by creating a finer mesh perpendicular to the wall. Figure 3-15 shows the inflation mesh at the fluid-solid interface.

Figure 3-14: Multi-scale meshing across different domains

Figure 3-15: Inflation mesh

A minimum of 10 nodes in the inflation layer has been created [117] to ensure that the turbulence model can be solved properly. y^+ is the dimensionless perpendicular distance from the wall, providing the near-wall resolution as it represents the dimensionless distance of the first node from the wall. Nevertheless, when y^+ is too small, the equations are too stiff, and the solutions might not be able to converge. A guideline [117] points out that the y^+ for SST model should be less than 2, if possible. Furthermore, a fine grid with y^+ around 1 is needed for heat transfer predictions [120]. Figure 3-16 illustrates that the y^+ values in the fluid domains are below 1 across the flow rates being modelled.

Figure 3-16: Distribution of the y^+ values in the fluid domain

3.3.6 Mesh quality

CFX and FLUENT require different mesh requirement as these solvers have different codes in the computational process. The criteria, listed in Table 3-12 and Table 3-13, are set to ensure that the quality of the mesh generated is acceptable.

Table 3-12: Grid generation criteria for FLUENT solver

Metric	Range
Skewness	< 0.9
Aspect ratio	< 40, except in the inflation layers

Table 3-13: Grid generation criteria for CFX solver

Metric	Range
Mesh orthogonality	> 0.1
Aspect ratio	< 100
Expansion factor	< 20

3.3.7 Discretisation of governing equations

CFX is a pressure-velocity coupled solver and a high-resolution scheme is chosen for the spatial discretisation. In FLUENT, Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm is adopted. The standard scheme is used for the discretisation of pressure, while the momentum, energy and turbulence parameters are computed using a second order upwind scheme.

3.3.8 Convergence

CFX Solver Modelling Guide [117] provides a guideline to monitor the convergence of the solutions based on the root-mean-square (RMS) residual level, as listed in Table 3-14.

Table 3-14: Interpretation on the RMS residual level

RMS residual level	Description
Above 10^{-4}	Loose convergence. Sufficient for qualitative analysis.
Between 10^{-5} and 10^{-4}	Relatively loose convergence. Sufficient for many engineering applications.
Between 10^{-6} and 10^{-5}	Good convergence. Sufficient for most engineering applications.
Below 10^{-6}	Very tight convergence. Hard to achieve, especially when a single precision solver is used.

For FLUENT solvers, the default scaled residual which is deemed sufficient for convergence is 10^{-3} for all equations, except the energy equation, for which the criterion is 10^{-6} [128]. Other than the value of the residual itself, the behaviour of the residual is also monitored. The solution process is only halted when the value of the residuals remains low and relatively constant for at least 50 iterations. The values of the convergence of the solutions are reported in the respective studies.

3.3.9 Data reduction

The local heat transfer parameters are extracted from the predicted data. The local heat transfer coefficient of each of the cross section is computed as:

$$h_i = \frac{q_x}{T_{w,x} - T_{f,x}} \quad (3-35)$$

The bulk fluid temperature is estimated based on the mass flow average:

$$T_{f,x} = \frac{1}{\rho V_m A_c} \int_{A_c} \rho V T_f dA_c \quad (3-36)$$

The wall temperature is the perimeter averaged wall temperature:

$$T_{w,x} = \frac{1}{P} \int_P T_w dP \quad (3-37)$$

Similarly, the local heat flux is the perimeter averaged heat flux:

$$q_x = \frac{1}{P} \int_P q dP \quad (3-38)$$

The average heat transfer coefficient is therefore calculated:

$$h = \frac{1}{n} \sum_{i=1}^n h_i \quad (3-39)$$

The average Nusselt number is computed as:

$$\text{Nu} = \frac{hD_h}{k_{fluid}} \quad (3-40)$$

The volume-averaged properties are taken as the fluid properties.

3.3.10 Mesh independence tests and validation of predicted data

Mesh independence tests are conducted for each channel geometry to ensure that the predicted data is independent on the grid size. The grid size is selected such that further refinement of the grid does not yield more than 3 per cent difference in the predicted data. Furthermore, the predicted data is compared against the available measured data and existing correlations to ensure the reliability of the data. The mesh independence tests and validation results for each of the study are presented in the respective chapter.

Chapter 4 Single-wavy-wall microchannel

This chapter presents the findings of an experimental study on a single-wavy-wall microchannel, i.e. a microchannel with a varying hydraulic diameter along the flow direction. The sinusoidal wavy profiles are introduced on the surface of the inner solid cylinder. A parametric study is conducted on three amplitudes and three wavelengths respectively, by keeping the other parameters constant. The measured results are supplemented by numerical predictions of the fluid field and heat flow at a certain flow rate to explain the underlying mechanisms. Working correlations for the average Nusselt number and friction factor are subsequently proposed for the single-wavy-wall microchannel.

4.1 Introduction

Wavy sinusoidal passages started as triangular wavy passages as introduced by Goldstein and Sparrow [129]. This channel configuration was improvised to a round-edge wavy profile [130], in the interest of the hydrodynamic performance. Rush et al. [131] summarised a significant body of research on heat transfer and flow in wavy passages. This section focuses more on the existing studies on wavy microchannels, which are more relevant to the current study.

While wavy passages had been proven their superior thermo-hydraulic performance, this channel configuration was first adopted in a microchannel by Sui et al. [132]. A numerical study was performed on wavy rectangular-cross-section microchannels with constant wavelength and different amplitude. The heat transfer enhancement outweighs the increment in pressure drop penalty. Channels with changing relative waviness are proposed to mitigate the temperature rise of the chip along the flow direction and the presence of hot spots. The same group of researchers [16] studied experimentally the effect of amplitude at different Re. The enhancement in Nusselt number exceeds the increment in friction factor, with reference to the straight channel, for both wavy microchannels of different amplitude. M. Khoshvaght-Aliabadi et al. [133] evaluated the heat transfer rate to the pumping power ratio for wavy microchannels of different amplitude and wavelength at different Pr and Re. Longer wavelengths, lower amplitudes,

lower Re and Pr yield considerably higher ratio. Z.G. Mills [134] examined the performance of a wavy channel under three flow regimes, namely unidirectional, circulatory and unsteady, which are formed in sequence when the flow rate increases. The effects of the amplitude and wavelength in these flow regimes are detailed. Unsteady flow in a low-amplitude wavy channel, at a Re greater than 385, favours the thermo-hydraulic performance.

The performance of a wavy microchannel is proven to be a strong function of the channel aspect ratio, amplitude and wavelength. Xie et al. [135] optimised the thermal performance the design of a wavy microchannel in the low Re regime based on the interpolate-spacing, the wavelength ratio and the amplitude ratio. These parameters, coupled with the working Re , affect the thermal performance of the microchannels studied. Sakanova et al. [15] concluded that high amplitudes and short wavelengths reduce the thermal resistance, being more noticeable at a high Re . The improvement of a wavy channel over a traditional rectangular channel is more obvious when water is used as the working fluid, as compared to the nanofluids which possess high dynamic viscosity. Mohammed et al. [136] studied numerically wavy channels with different amplitudes and constant hydraulic diameter. They reported an optimum range of amplitude that yields better thermal performance as compared to the straight microchannel. Rostami et al. [137] similarly proved the existence of an optimum geometry for an optimal heat transfer rate, at a constant hydraulic diameter, pitch and wavelength, owing to the presence of vortices which their shapes and intensities depend on the geometrical parameters and operating conditions. M. Khoshvaght-Aliabadi et al. [133] proposed working correlations for the average Nusselt number and friction factor, as a function of Re , Pr , amplitude and wavelength of the profile. L. Lin et al. [17] explored designs with varying amplitude and wavelength along the flow direction. These designs perform better than the conventional wavy design with constant parameters in terms of thermal resistance and temperature uniformity in the substrate.

The existence of centrifugal force due to the channel curvature leads to the formation of Dean vortices and chaotic advection at the crest and the trough, enhancing heat transfer performance through fluid mixing [138]. Nevertheless, under certain circumstances, Dean vortices are absent at low Re and low curvature in a microchannel [14]. The

channel geometry shifts the maximum velocity from the centreline, leading to the thinning of the thermal and hydrodynamic boundary layers. Similar observations can be found in [15].

There are two wavy channels which are conventionally employed, namely the serpentine channel and racoon channel. Serpentine channel is one with a phase shift of 0° , i.e., the crests and troughs are facing each other alternately. Racoon is one with a phase shift of 180° , i.e., the crests and troughs are facing each other. The former has a constant hydraulic diameter whereas the latter constitutes a converging-diverging flow path, having a changing hydraulic diameter. Racoon channels possess lower thermo-hydraulic performance as compared to serpentine channels and straight channels [14, 139].

The promising performance of a wavy microchannel, coupled with the simplicity of implementation, makes the wavy microchannel a promising study direction. This chapter explores the third type of channel configuration, namely a single-wavy-wall microchannel. This implementation is useful in applications where modifications to the heated surface are not feasible or costly. Furthermore, this implementation makes it possible to enhance the performance of existing flow channels by placing an insert with wavy profiles. Furthermore, Goh and Ooi [12] studied the triangular wavy profile using similar implementation and achieved higher thermo-hydraulic performance as compared to other designs [11, 105]. Therefore, it is compelling to investigate the thermal and hydrodynamic performance of this channel configuration which has a smoother edge as compared to the triangular wavy profile. Working correlations are also proposed to enable the adoption of this design.

A straight microchannel of a gap size of 300 microns is used as the reference for the performance evaluation. At the same average gap size, three waveform amplitudes of 0.05, 0.10 and 0.15 mm and three wavelengths of 3, 6 and 10 mm, are studied at a constant wavelength of 3 mm and an amplitude of 0.15 mm respectively.

4.2 Channel design

A sinusoidal wavy profile as described by Equation (4-1) is introduced on the surface of the solid cylinder described in Section 3.1.4.

$$y = a \sin\left(\frac{2\pi x}{\lambda}\right) \quad (4-1)$$

a and λ represent the amplitude and wavelength respectively, whereas x is the position along the flow direction.

Figure 4-1 illustrates the definition of these parameters. The average gap size H , is kept constant at 0.3 mm for all the wavy microchannels. The wavy microchannels are subsequently named in the format of sin_a_b where a is the amplitude in 10^{-2} mm and b represents the wavelength in mm, as summarised in Table 4-1.

Figure 4-1: Wave profiles parameters

Table 4-1: Nomenclature for the microchannel

Microchannel	Insert design parameter		Average gap size, $H/$ mm
	Amplitude/mm	Wavelength/mm	
sin_005_3	0.05	3.0	0.30
sin_010_3	0.10	3.0	
sin_015_3	0.15	3.0	
sin_015_6	0.15	6.0	
sin_015_10	0.15	10.0	

Each parametric study comprises of three wavy microchannels as shown in Figure 4-2. The performance of the wavy microchannels is compared to a straight microchannel, as described in Section 3.1.1 with the same average gap size. Table 4-2 tabulates the actual

maximum and minimum gap sizes that result from the superimposition of the fabricated inserts and the hollow copper channel.

Figure 4-2: Single-walled sinusoidal wavy microchannel [140]

Table 4-2: Measured maximum and minimum gap size of the channels

Microchannel	Mean value of the maximum gap size (mm)	Mean value of the minimum gap size (mm)	Average gap size (mm)
straight	-	-	0.29
sin_005_3	0.33	0.24	0.29
sin_010_3	0.38	0.19	0.29
sin_015_3	0.43	0.14	0.29
sin_015_6	0.43	0.14	0.29
sin_015_10	0.43	0.14	0.29

The largest dimensionless surface roughness calculated using Equation (3-3) and (3-4) is 3.11 and thus the microchannels are considered hydraulically smooth, based on the criterion stated in Section 3.2.4.

4.3 Methodology

Steady-state measurements are acquired using the experimental system. Subsequently, the heat and flow fields in the channels at the flow rate of 3 l/min are predicted using the numerical model. Both the experimental and numerical models are described below.

4.3.1 Experimental model

The flow rate of the study ranges from 2.00 to 7.00 l/min with an interval of 0.25 l/min, with an inlet temperature of 28 °C. The heat supply to the channel is kept constant at 1000 W, equivalent to a heat flux of 53.0 W/cm². The validation of the measuring system using the straight channel is described in Chapter 3. Repeatability tests are conducted using different channels at random flow rates to ensure the consistency of the system, as shown in Table 4-3.

Table 4-3: Repeatability tests for measured data

Profile	Flow Rate (l/min)	Discrepancy for h (kW/m²·K)	Discrepancy for Δp (bar)
sin_010_3	2.00	0.19%	0.48%
sin_015_6	2.75	-1.01%	-0.24%
sin_005_3	3.00	0.01%	1.01%
straight	4.25	-0.20%	0.24%
sin_015_3	5.00	0.73%	0.26%
in_015_10	6.50	0.82%	-0.02%

The uncertainties of the derived parameters are reported with a confidence level of 95% in Table 4-4.

Table 4-4: Maximum experimental uncertainties

Parameter	Maximum uncertainty (%)
h	13.3
Nu	13.4
Re	0.28
Δp	11.0
f	11.1

4.3.2 Numerical model

The steady-state conjugate heat transfer model governed by the equations in Section 3.3.1 are solved using ANSYS CFX, coupled with SST $k-\omega$ turbulence model. The following boundary conditions are applied to the discretised test module:

- a. A uniform velocity fluid inlet with a temperature of 28 °C
- b. An atmospheric pressure fluid outlet
- c. Free convection at the outer surfaces of the test module with a heat transfer coefficient of 8 W/m²·K at an ambient temperature of 30 °C [106]
- d. Total volumetric heat generation of 1000 W for the heating cable
- e. Conservative heat flux at the solid-solid and solid-fluid domain
- f. Constant solid properties [106]
- g. Variable fluid properties based on IAPWS IF97 standards [114]

A one-eighth model is employed instead of the full-model owing to the symmetry of the model. Mesh independence tests, as summarised in Figure 4-3, are conducted to ensure that the choice of grid size does not affect the predicted solutions. In this case, the model with 2.3 million element counts is selected as an increment of element by nearly 300% yields only a marginal difference which is less than 2% in the observed parameters. Subsequently, the predicted data from the numerical model is compared against the measured data from the experimental model to ensure the accuracy of the predicted values. The results are tabulated in Table 4-5.

Figure 4-3: Mesh independence test for predicted data

A consistent under-prediction is observed for the pressure drop data. This likely arises due to the measurement uncertainties and the limitations in the experimental fluid pressure measurement. Firstly, the maximum uncertainty for the measured pressure drop is 11.0%. Secondly, there are minor differences in the pressure loss measurement between the numerical model and the experimental setup. In the numerical model, the pressure loss can be conveniently computed at the inlet and the outlet of the test module. Whereas in the experimental implementation, the pressure transmitters and the thermocouples are installed at a short distance away from the test module due to practical limitations. The installation of the thermocouples and pressure transmitters might have disturbed the flow to a small extent, resulting in slightly higher pressure losses to be registered experimentally. In fact, this is common in microchannel implementations, where numerical difference up to 11 percent has been registered [67].

Table 4-5: A comparison between measured and predicted data

Profile	h (kW/m ² ·K)		Δp (bar)	
	Measured	Predicted	Measured	Predicted
Plain	12.6	13.2	0.148	0.129
Deviation (%)	4.8		-12.9	
sin_005_3	13.8	13.4	0.165	0.147
Deviation (%)	-2.9		-10.9	
sin_010_3	24.1	25.6	0.363	0.334
Deviation (%)	6.2		-8.0	
sin_015_3	34.4	38.3	0.898	0.769
Deviation (%)	11.3		-14.4	
sin_015_6	28.2	29.6	0.503	0.431
Deviation (%)	5.0		-14.3	
sin_015_10	20.4	21.3	0.357	0.317
Deviation (%)	4.4		-11.2	

The root-mean-square residual of 10^{-4} is observed for all the converged solutions. The data reduction for both experimental system and numerical model is described in Chapter 3.

4.4 Results and discussion

4.4.1 Local heat transfer coefficient

Figure 4-4 shows the variation of the local heat transfer coefficient with respect to the wave position. The highest h_x is recorded near the peaks of the waveform. The smallest flow area is formed between the copper wall and the peak of the waveform. By the virtue of mass conservation, and since water is virtually incompressible, the velocity of the flow is the highest at this location, gradually increasing after passing through the trough of the waveform. Therefore, the variation of h_x follows the shape of the waveform, with the highest value at the peak and the lowest value at the trough, particularly obvious in sin_005_3 channel.

Figure 4-4: Local heat transfer coefficient with respect to wave position for channels with different amplitude and same wavelength

Recirculation zones are found in channel with high amplitude, i.e. sin_015_3, as observed in Figure 4-6(c). The recirculation zone reduces the effective hydraulic diameter of the channel, shifting the position with the lowest bulk fluid velocity slightly to the right of the trough. This results in a thicker thermal boundary as shown in Figure 4-5(c), and thereby a lower local heat transfer coefficient. The flow is accelerated towards the upward slope of the subsequent wave. The wave geometry drives the flow upwards, aligning the velocity vector in the direction of the temperature gradient, as observed in Figure 4-7(c). This aids in the heat removal process and therefore the maximum h_x is shifted slightly to the left of the peak. This phenomenon is obvious in channels with higher amplitude, as shown in Figure 4-4.

There is no observable recirculation zone found in sin_010_3 channel. After passing the peak, the high-amplitude downslope waveform drives the flow away from the heated wall, in the absence of recirculation zone which can be found in sin_015_3. This is depicted in Figure 4-6(b). Therefore, the thermal boundary layer becomes thicker abruptly, as shown in Figure 4-5(b), resulting in the low h_x . The slow-moving fluid in the

trough of the waveform causes the maximum velocity in the flow direction shifts away from the trough, thinning the velocity and thermal boundary layer at the heated wall.

Figure 4-6(d) shows that the recirculation zones found in the troughs of sin_015_6 are less obvious as compared to that of sin_015_3. The former has a waveform of longer wavelength. Right after passing the peak of the waveform, the flow is guided downwards by the profile of the waveform. This results in a recirculation zone and a thicker thermal boundary layer, as depicted in Figure 4-5(d) and Figure 4-6(d). These phenomena can also be found in sin_010_3 where there is no recirculation zone found in the trough.

4.4.2 Effect of wave parameter on heat transfer and fluid flow

The measured heat transfer coefficient is plotted against volumetric flow rate in Figure 4-8. The highest h attained is $51.1 \text{ kW/m}^2\cdot\text{K}$ at a flow rate of 7 l/min , double of that of the straight channel which records h of $25.3 \text{ kW/m}^2\cdot\text{K}$.

Channel with low-amplitude waveform, i.e. sin_005_3, poses minimal disturbance to the flow, causing the flow transitions to occur at a similar Re to that of the straight channel, as shown in Figure 4-8. Channels with higher amplitudes might have induced transition at lower Re , and therefore the flow is in the transitional or turbulent flow regime across the tested Re range.

Therefore, sin_015_3, which has the waveform with the highest amplitude, registers the highest h throughout the flow rates. As compared to sin_010_3, sin_015_3 registers a minimum enhancement in h of 22.5 per cent, and this difference can be as much as 61.8 per cent at the lowest flow rate. Comparing sin_005_3 and sin_015_3, the latter has a h which is 151.8 per cent higher at the lowest rate and maintains the enhancement at 45.5 per cent at the highest flow rate.

For the wavelength study, Figure 4-8 illustrates that the heat transfer coefficient is inversely related to the wavelength. This is physical sound as a decreased wavelength implies a higher frequency of the flow and boundary layer disturbance. The heat transfer enhancement mechanisms are triggered more frequently in channels with shorter-wavelength waveforms. However, the difference between the wavelengths is relatively

(a) sin_005_3

(b) sin_010_3

(c) sin_015_3

(d) sin_015_6 (scaled differently to include the whole waveform)

Figure 4-5: Fluid temperature distribution in the channel

(a) $\sin_{005_3}$

(b) $\sin_{010_3}$

(c) sin_015_3

(d) sin_015_6

Figure 4-6: Two-dimensional streamlines in the channels

(a) `sin_005_3`

(b) `sin_010_3`

(c) `sin_015_3`

Figure 4-7: y-component velocity in the channel

less significant. As compared to sin_015_6 and sin_015_10, sin_015_3 has an advantage, in terms of h at the same flow rate, of a maximum 29.5 and 96.5 per cent respectively.

The enhancement in heat transfer is nonetheless, accompanied by undesirable pressure loss as shown in Figure 4-9. A higher amplitude which implies higher degree of flow disturbance; and a shorter wavelength which suggests higher frequency of flow disturbance, resulting in a greater pressure loss across the channel. Furthermore, all the enhanced channels depict an increasing gradient in pressure loss with the increment in flow rate. This is contrary to the increment in heat transfer coefficient in the enhanced channels, where the gradient of increment decreases with the increment in flow rate. Therefore, this necessitates the investigation in the thermo-hydraulic performance of the channels, which is discussed in the subsequent section.

Figure 4-8: Heat transfer coefficients against flow rate

Figure 4-9: Pressure loss across the channels against flow rate

4.4.3 Thermo-hydraulic performance

Figure 4-10 shows the increment in the heat removal of the wavy channels as compared as the straight microchannel, evaluated at the same pumping power. The highest performance index achieved is 1.64, implying that sin_015_3 can remove 64% more heat as compared to the straight channel at the same pumping power.

The thermo-hydraulic performance is a strong function of the amplitude of the waveform. It can be observed that, the performance index for the channel with high-amplitude waveform, i.e. sin_015_3, decreases with increasing pumping power. On the other hand, the index for sin_005_3 increases with increasing pumping power. The difference between these two channels decreases from 43.6% from 9.5% as the pumping power supplied increases. This can be understood as, at the same relatively higher pumping power, sin_005_3 can operate at higher Re as compared to sin_015_3, owing to the lower pressure loss across the configuration. Although sin_010_3 has a higher heat transfer capability as compared to sin_005_3, this channel has the lowest performance index. The maximum performance index recorded is 1.08. On the other hand, sin_005_3 can achieve a maximum performance index of 1.25.

There is no obvious change in trend for the comparison among the channels with the same-amplitude and different-wavelength waveforms. Although sin_015_3 has twice

the number of waveforms within the microchannel length as compared to sin_015_6, the difference in performance between two channels is less than 6.4 per cent. Similarly, as the number of waveforms reduces from 10 in sin_015_3 to 3 in sin_015_10, within the length of the microchannel, these two channels only differ by an average of 25.7 per cent in the performance index.

Figure 4-10: Increment in the heat removal at the same pumping power

4.4.4 Working correlation

Nusselt number and friction factor correlations are proposed for this channel configuration to enable the adoption of this design. These correlations are proposed in the form of Equation (4-2) and (4-3). The terms outside the square brackets denote the general expression for a fully-developed turbulent, where A , B , C and P are constants to be derived. The second term in the square brackets accounts for the entrance effect, with D and R as coefficients to be determined. The third term models the effects of the profiles on the heat transfer and hydrodynamic parameters. E , F , G , S , T and U are constants to be derived.

$$Nu = A Re^B Pr^C \left[1 + \frac{D}{L/D_h} + E \frac{(a/H)^F}{(\lambda/L)^G} \right] \quad (4-2)$$

$$f = \frac{1}{\text{Re}^p} \left[Q + \frac{R}{L/D_h} + S \frac{(a/H)^T}{(\lambda/L)^U} \right] \quad (4-3)$$

92 steady-state measurements are employed for the regression process which is performed using a particle swarm optimisation algorithm, implemented using MATLAB. Particle swarm optimisation considers a population of individuals. Each individual has a position and velocity defined. The algorithm moves the individuals around in the defined domain to search for a potential solution considering the individual's current position, the individual's and the population's best outcome [141]. The solutions are deemed to be converged when further iterations yield no further improvement in the regression coefficients to accurately predict the measured data. Table 4-6 shows the regression coefficients. The applicable range of the correlations is shown in Table 4-7. Comparison between the predicted Nusselt number and friction factor shows an acceptable difference, yielding an average discrepancy of 5% and 4%, and a maximum discrepancy of 15% and 12% respectively.

Table 4-6: Regression coefficients

Coefficient	Value
<i>A</i>	0.034
<i>B</i>	0.7
<i>C</i>	$\frac{1}{3}$
<i>D</i>	7
<i>E</i>	0.6
<i>F</i>	1.3
<i>G</i>	0.7
<i>P</i>	0.3
<i>Q</i>	0.7
<i>R</i>	0.7
<i>S</i>	2.7
<i>T</i>	3.0
<i>U</i>	1.0

Table 4-7: Applicable range of correlation

Microchannel	Re range
sin_005_3	2800 – 4600
sin_010_3	2200 – 4600
sin_015_3	1380 – 4600
sin_015_6	1550 – 4600
sin_015_10	1710 – 4600

Chapter 5 Microchannel with single-wavy-wall and serpentine-wavy-wall

This chapter compares the thermal and hydrodynamic performance of three different configurations of wavy microchannel. These include a double-wavy-wall channel – a serpentine channel, as well as two single-wavy-wall channels, each with a different heating boundary. The first one realises a wavy heating surface while the second one has a wavy non-heated surface. The heat transfer and flow field in the channels are solved using finite volume methods, in which the predicted data is validated using available measured data. The governing equations are solved for the Re range of 750 to 2200. The predicted heat transfer coefficient and pressure loss across the channels are presented, together with explanations on the underlying mechanisms. The channels are evaluated for the heat transfer improvement given the same pumping power.

5.1 Introduction

The previous chapter demonstrates that the changing of the hydraulic diameter of the channel by a wavy non-heated profile along the flow direction enhances the thermo-hydraulic performance significantly, as compared to a straight microchannel. Recirculation zones are formed at high-amplitude waveforms, but these vortices do not aid in the heat removal directly as they are formed at the non-heated surface. Thus, this channel configuration realises the re-entrant effects while the benefits of channel curvature have been downplayed.

As these vortices might help to remove the heat from the heated wall, this chapter examines the performance of a wavy-heated surface, keeping the non-heated surface flat. It is also important to compare the performance of these single-wavy-wall channels to the serpentine wavy channel which is commonly found. The latter has a uniform hydraulic diameter along the streamwise direction.

Therefore, this chapter investigates the thermo-hydraulic performance of these three wavy channel configurations at the same average hydraulic diameter. The amplitude and wavelength of the waveforms are kept the same.

5.2 Channel design

Including the straight microchannel, there are four channel configurations which are studied. These channels are straight channel, serpentine channel, wavy-insert channel and wavy-heater channel, as shown in Figure 5-2. The wavy-insert channel has a flat heating surface whereas the wavy-heater channel constitutes a wavy heating surface. Contrary to the single-wavy-wall channels as aforementioned, the serpentine channel has two wavy walls which are in-phase, resulting in a constant hydraulic diameter along the flow direction.

The wavy-insert channel adopts sin_015_3 channel which is studied in the previous chapter. This channel, which has a high-amplitude and short-wavelength waveform, outperforms the other channels and is, therefore, used as a reference for parameter selection in this chapter. This also enables the validation of the predicted data by using the measured data from the previous chapter.

Figure 5-1 shows the key wave parameters. The length of the channels is kept constant at 30.0 mm. The waveforms have a constant amplitude of 0.15 mm and a constant wavelength of 3.0 mm. The mean gap size of the channels is the same at 300 μm .

Figure 5-1: Key wave parameters

Figure 5-2: Nomenclature of the microchannels

5.3 Numerical modelling

These channels are placed in a test module which is shown in Figure 5-3. This test module resembles that in the measurement system. This enables the validation of the predicted data by using the measured data for the straight and wavy-insert channel. A three-dimensional model is discretised, and the governing equations which are stated in Section 3.3.1 are solved using ANSYS CFX solver. SST $k-\omega$ model is employed to model the turbulence. The following boundary conditions are assumed:

- a. Single-phase, steady-state liquid water flow with a uniform velocity inlet for Re range of 750 to 2200.
- b. An inlet liquid water temperature of 28 °C.
- c. An atmospheric pressure outlet condition.
- d. A free convection outer boundary, with a heat transfer coefficient of 8 W/m²·K and an ambient temperature of 30 °C [106].
- e. A uniform heat flux of 50 W/cm² at the outer surface of the copper block.
- f. Constant thermodynamic properties for the solid domains [106].
- g. Fluid properties evaluated at the average fluid temperature which is computed based on the energy balance equation.

Figure 5-3: Numerical test module [140]

For the thermo-hydraulic performance evaluation, the pumping power is taken as constant in the simulations. An initial inlet velocity is assumed for the calculations, then the resultant pressure drop and pumping power are computed. Iterations are carried out by changing the inlet velocity so that the resultant pumping power equals to the specified pumping power of 0.14 to 0.54 W, at an interval of 0.1 W.

A one-eighth model is employed to reduce the computational efforts, given the symmetry of the model. Mesh independence tests are conducted for each configuration at the highest Re. Figure 5-4 shows the mesh independence test conducted for the straight microchannel. The model with 2.6 million elements is chosen since a further massive increment of the number of elements by 42 per cent, only yields a maximum difference of 0.9 per cent in the predicted data.

Figure 5-4: Mesh independence test for the straight channel

The solutions are considered converged once the root-mean-square residuals reduce to the order of 10^{-6} for all variables.

5.4 Results and discussion

5.4.1 Validation of predicted results

The predicted data is validated against the measured data to ensure its reliability. The predicted Nusselt number and friction factor of straight and wavy-insert channels are compared against that gathered from the experimental setup in Figure 5-5 and Figure 5-6 respectively. The predicted and measured heat transfer data are in good agreement for both straight and wavy-insert channels, yielding an average difference of 2.0 and 4.8 per cent in Nusselt number.

The numerical model constantly underpredicts the friction factor by an average of 15.5 and 18.6 per cent for straight and wavy-insert channel respectively. Similar underpredictions are obtained in Chapter 4 and the possible reasons are elaborated in the same chapter.

Figure 5-5: Validation of predicted Nu with measured Nu

Figure 5-6: Validation of predicted f with measured f

5.4.2 Heat transfer performance

Figure 5-7 illustrates the heat transfer coefficients of various channel configurations, measured at the same flow rates. The maximum h achieved is $46.2 \text{ kW/m}^2\cdot\text{K}$ at the highest flow rate in the wavy-heater channel, recording a 261.8 per cent augmentation as compared to the straight channel. At the same flow rate, the serpentine channel and the wavy-insert channel register a h which is 79.1 and 194.1 per cent higher than the straight channel.

With reference to the results presented in Chapter 4, the flow regime in the straight channel remains laminar within the flow rate of 1.0 to 3.5 l/min. Similarly, the flow in the wavy-insert, which is the sin_015_3 channel, is in the transitional or turbulent regime. Therefore, the wavy-heater channel, which has a similar periodic flow cross-section constriction, causes the flow to transit into transitional or turbulent flow regime. The serpentine channel depicts a transition in trend near 2 l/min, as shown in Figure 5-7.

Figure 5-7: Heat transfer coefficient for various channel configurations against flow rate

All the single-wavy-wall channels and serpentine wavy configurations depict better heat transfer performance as compared to the straight microchannel. The single-wavy-wall channels outperform the serpentine channel. The wavy-insert and wavy-heater channel achieve a minimum enhancement of 47.1 and 102.0 per cent respectively with respect to the serpentine channel, predicted at the same flow rate. The wavy-heater channel registers a consistently higher h over the wavy-insert channel, with a minimum enhancement of 23.0 per cent. Figure 5-8 illustrates the local heat transfer coefficient, h_x along the stream-wise direction and Figure 5-9 shows the position in the actual channel implementations.

Figure 5-8: Local heat transfer coefficient along the stream-wise direction

Figure 5-9: Position definition for the local heat transfer coefficient along the stream-wise direction

The maximum h_x in the wavy-insert channel occurs at the minimum flow area, at 0.25λ where the peak of the waveform on the insert surface is. At this location, the boundary layers are relatively thinner due to the acceleration of the fluid by the virtue of mass conservation for an incompressible fluid. A recirculation zone can be observed in the wavy-insert at the trough of the waveform, as shown in Figure 5-10(b). This recirculation zone effectively reduces the cross-sectional flow area and thus the core velocity remains

high while passing through the maximum flow area at 0.75λ . The flow is brought towards the insert surface or the non-heating surface near to λ , resulting in the minimum h_x .

In the wavy-heater configuration, the maximum h_x occurs near 0.5λ , somewhere in between the peak and the trough. Akin to the wavy-insert configuration, recirculation zones are formed at the maximum flow area, as seen in Figure 5-10(c). The colder fluid which bypasses the recirculation zone is accelerated towards the heating wall at near to 0.5λ , thinning the velocity and thermal boundary layer. The heat transfer enhancement effects attributed to the channel constrictions keep the local heat transfer coefficient relatively high, although it is lower than that at 0.5λ . The rotating vortices aid in removing the heat from the heating surface, at the regions where the vortices start to form after passing through the maximum flow area, resulting in an increment in the local heat transfer coefficient. As the hot fluid is brought towards the heating wall from 0.25λ to 0λ , a thicker thermal boundary is formed and can be observed in Figure 5-11(d). This leads to the lowest h_x after the fluid passes through the minimum flow area.

In the serpentine channel, the geometry governs the shift of the maximum velocity from the centreline towards the heated surface after passing the peak, as shown in Figure 5-10(a). This results in a thinner boundary layer as depicted in Figure 5-11(b) and a higher h_x at 0.5λ . Similarly, the maximum velocity is shifted towards the non-heated surface after passing through the trough, resulting in a thicker boundary layer at the heated surface. Therefore, the minimum h_x results at λ .

(a) Serpentine channel

(b) Wavy-insert channel

(c) Wavy-heater channel

Figure 5-10: Flow streamlines in the channels

(a) Straight channel

(b) Serpentine channel

(c) Wavy-insert channel

(d) Wavy-heater channel

Figure 5-11: Temperature distribution in the channels

5.4.3 Hydrodynamic performance

Figure 5-12 shows the pressure loss across the configurations. The pressure losses of the single-wavy-wall channels are similar, which are 3.8 to 7.5 times higher than the straight channel with the increasing flow rate. The serpentine channel, on the other hand, incurs a higher pressure drop of 17.4 to 52.8 per cent with the increment in the flow rate. The presence of recirculation zones in the single-wavy-wall channels effectively reduces the actual flow cross-section after the constriction zones, leading to high head losses at both the maximum and minimum flow cross-sections. In contrast, the streamlined flow in the serpentine channel mimics that in the straight channel. The marginal increase in pressure drop is attributed to the local head losses incurred at the bends and meanders.

Figure 5-12: Pressure loss across various channel configurations against flow rate

The increment in heat transfer, as described in the previous section, is accompanied by the undesirable increment in pressure drop. An evaluation of the thermo-hydraulic performance is needed to measure the heat transfer increment of the channels at the same pumping power. This is described in the subsequent section.

5.4.4 Thermo-hydraulic performance

Figure 5-13 shows the increment in the increment in heat removal capability evaluated at the same pumping power. Considering both the thermal and hydrodynamic performance, the improvement of the serpentine channel is less than 10 per cent as compared to the straight channel.

The negative effect of higher pressure drops incurred by the single-wavy-wall channels is justified by the more dominant improvement in the heat transfer, corroborated by the high thermo-hydraulic performance. The wavy-insert channel attains a minimum improvement of 19.5 per cent and a maximum improvement of 37.6 per cent. The wavy-heater channel which has similar hydrodynamic performance as the wavy-insert channel further enhances the performance to a minimum of 61.1 per cent, justified by the consistently higher high transfer coefficients at the same flow rate and similar pressure drop. The wavy-heater channel has the highest performance index among the channels across the pumping power. This channel can achieve a performance index of 1.88, which is 36.8 and 74.1 per cent higher than the wavy-insert channel and the serpentine channel.

Figure 5-13: Increment in the heat removal capability at the same pumping power

Chapter 6 Wavy microchannel with a varied amplitude

This chapter studies the effects of varying amplitude of the waveform along the flow direction in the single-wavy-wall microchannels on the thermal and hydrodynamic performance. The modifications are realised on the wavy heated surface and wavy non-heated surface. Measured data is collected for the wavy non-heated surface only. The measured data is used to validate the predictions from the numerical model. This validated numerical model is then used to predict the effects of a varied amplitude of the waveform on the wavy-heated surface.

6.1 Introduction

While it has been proven in Chapter 5 that single-wavy-wall microchannels outperform the serpentine and straight channels in terms of thermo-hydraulic performance, this chapter aims to improve the performance of the single-wavy-wall microchannels.

Majority of the works on the wavy microchannels focused on the waveforms which have constant amplitude and wavelength along the flow direction. These studies have been covered in the previous chapters.

Varying amplitude and wavelength have been proven to be able to improve the performance of wavy microchannels. Sui et al. [13] introduced relative waviness, which was defined as the ratio of amplitude to wavelength, and was varied along the flow direction. The relative waviness was increased at regions with high heat flux to mitigate the hot spot problem. However, the study only investigated an increasing relative waviness along the flow direction, as well as an increased relative waviness at the middle of the channel, and their effects on the substrate temperature.

L. Lin et al. [17] examined three different types of design: constant amplitude with varying wavelength, constant wavelength with varying amplitude, and varying amplitude with varying wavelength. Waveforms with an increased amplitude and a decreased wavelength reduce the thermal resistance and increase the Nu, as compared to the straight and uniform wavy profile. Furthermore, the enhancement is more

significant when the difference in amplitude or wavelength increases along the flow direction.

As the variation in amplitude and wavelength along the flow direction has proven to be effective in reducing the thermal resistance, this chapter adopts this methodology to enhance the performance of the single-wavy-wall channels. The amplitude of the waveform is varied along the flow direction while keeping the wavelength constant. Both increased amplitude and decreased amplitude are studied, with reference to a straight channel as well as a channel with constant-amplitude waveform. These waveforms have the same average amplitude to gap size ratio.

6.2 Channel design

Figure 6-1 shows the definition of the wave parameters. a represents the amplitude of the waveform at the first half of the channel whereas b represents that at the subsequent half of the channel. There are six wavy channels which are being considered in this study and the corresponding wave parameters are shown in Table 6-1. The straight channel is used as the reference channel for the performance comparison. These channels have the same average gap size of 0.30 mm and an average amplitude of 0.10 mm for the waveforms. These channels are illustrated in Figure 6-2.

(a) Wavy non-heated surface

(b) Wavy heated surface

Figure 6-1: Wave parameters

Table 6-1: Nomenclature of the channels

Modified surface	Waveform	Name	a (mm)	b (mm)	H (mm)
Non-heated surface	Increased amplitude	IA	0.05	0.15	0.30
	Decreased amplitude	DA	0.15	0.05	
	Uniform amplitude	UA	0.10	0.10	
Heated surface	Increased amplitude	IA_heater	0.05	0.15	
	Decreased amplitude	DA_heater	0.15	0.05	
	Uniform amplitude	UA_heater	0.10	0.10	
	Straight	-	-	-	

Metrology is conducted on the fabricated inserts and the actual sizes of the channels are tabulated in Table 6-2.

Table 6-2: Actual minimum and maximum gap size

Channel	Front		Rear	
	Minimum gap (mm)	Maximum gap (mm)	Minimum gap (mm)	Maximum gap (mm)
IA	0.24	0.34	0.14	0.43
DA	0.14	0.43	0.24	0.34
UA	0.19	0.40	0.19	0.40

(a) IA

(b) DA

(c) UA

(d) IA_heater

(e) DA_heater

(f) UA_heater

Figure 6-2: Microchannels

The maximum dimensionless surface roughness is 1.95, computed according to Equations (3-3) and (3-4). This is less than the threshold value of 5 and therefore, these channels are considered as smooth.

6.3 Methodology

The experimental measurements are conducted from the volumetric flow rate of 1.0 to 7.0 l/min at an interval of 0.5 l/min and fluid inlet temperature of 16 °C, for the following channels: UA, IA, DA and straight channel. The nominal heat input into the test module is 800 W, ensuring that the fluid remains in the single-phase condition at the lowest flow rate. The repeatability tests and the maximum uncertainty values with a confidence level of 95 per cent are reported in Table 6-3 and Table 6-4 respectively.

Table 6-3: Repeatability tests

Channel	Flow rate	Difference in h (%)	Difference in Δp (%)
UA	2.50	-2.43	-2.44
UA	7.00	2.17	-0.283
DA	2.50	3.00	0.319
DA	3.00	-2.14	-1.86

Table 6-4: Maximum uncertainties reported with 95 per cent confidence level

Parameter	Uncertainty (%)
h	10.3
Δp	3.03
$\dot{V}$	0.0577

The channels are modelled using the finite volume method to visualise the flow field and heat transfer. The governing equations, stated in Section 3.3.1, are solved using the SIMPLE algorithm implemented in the FLUENT solver. SST transition model is employed for turbulence modelling. Attributable to the symmetry condition, a 1/8th model is employed to save computational resources. The following boundary conditions are assumed:

- A mass flow rate inlet with an inlet temperature of 16 °C.

- b. A pressure outlet at the atmospheric pressure.
- c. Negligible heat loss to the surroundings.
- d. Constant solid properties and variable fluid properties based on the IAPWS IF97 standards [106, 115].
- e. Conservative heat flux at the solid-fluid interface.

Mesh independence tests are conducted on the channels and Figure 6-3 shows the results for the IA channel. It shows that increasing the number of elements by 46 per cent, from 9.2 to 13.4 million, yields a maximum of 2 per cent difference in the predicted heat transfer coefficient and pressure drop across the channel, as compared to the model with the finest mesh. Similar results are obtained for the other channels.

Figure 6-3: Mesh independence test for IA channel

The scaled residuals are at the order of 10^{-6} for the energy equation and, below 10^{-3} for the other parameters. Furthermore, the key quantities are monitored so that the values remain constant for more than 50 iterations.

6.4 Results and discussion

6.4.1 Validation of the numerical model

Figure 6-4 (a) to (c) show the validation of the predicted data with the measured data, for IA, DA and UA respectively. In general, the numerical model overpredicts the heat transfer coefficient and underpredicts the pressure loss data. For UA and DA, the average

numerical difference between the predicted and measured data is less than 10 per cent. This numerical difference is slightly higher for IA, at 11 per cent.

Although the pressure loss data is also underpredicted in the previous chapters, the numerical difference is higher in this set of data, reporting a maximum difference of 26.6 and 27.2 per cent for UA and DA respectively, at the higher flow rates. On top of the experimental limitations which cause higher measured pressure data, as explained in the previous chapters, this higher numerical difference might imply that this transition numerical model exaggerates the pressure loss across the channel as the flow rate increases.

(a) IA

(b) DA

Figure 6-4: Validation of predicted data using measured data

6.4.2 Effects of varied amplitude on the non-heated wavy wall

Figure 6-5 shows the heat transfer coefficients of the channels with a non-heated wavy wall. DA has the highest heat transfer coefficients for all the flow rates. Below 3 l/min , DA is able to achieve a maximum enhancement in heat transfer coefficient of 58.3 and 34.8 per cent as compared to UA and IA respectively, both are measured at the same flow rate. The difference between DA and IA is consistently more than 21 per cent. However, from 3.5 to 6 l/min , the difference between DA and UA is less than 10 per cent. At lower flow rates, IA and UA have similar performance. Beyond 2.5 l/min , UA outperforms IA with a higher heat transfer coefficient at the same flow rate. Although these channels have the same average amplitude to gap size ratio for the length of 30 mm, the difference in the heat transfer performance can be attributed to the absolute amplitude of the waveform and the position of the waveform with the same amplitude.

Figure 6-5: Heat transfer coefficient against flow rate

6.4.2.1 Effects of amplitude of the waveform on the heat transfer coefficient

As reflected by the correlation proposed in Section 4.4.4, the heat transfer performance increases exponentially with the amplitude of the waveform. As seen from Table 6-5, h increases 19.4 per cent when the amplitude increases from 0.05 to 0.10 mm at the front of the channel, at the same flow rate. This enhancement increases even more rapidly, by 43.6 per cent when the amplitude increases from 0.10 to 0.15 mm.

Table 6-5: Average heat transfer coefficient at the front and rear of the channel at 3 l/min

Position	Channel	a (mm)	h (kW/m ² ·K)
Front (2 nd to 5 th waveform)	IA	0.05	14.4
	DA	0.15	24.7
	UA	0.10	17.2
Rear (7 th to 10 th waveform)	IA	0.15	21.7
	DA	0.05	16.7
	UA	0.10	16.5

6.4.2.2 Effects of position of the waveforms with the same amplitude on the heat transfer coefficient

As illustrated in Table 6-5, the 0.10- and 0.15-mm amplitude waveforms incur higher heat transfer coefficients at the front of the channel. This is because the fresh fluid entering the channel has a higher temperature difference with the wall and thus a higher temperature gradient at the heat transfer surface. However, this phenomenon is not observed for the low-amplitude waveforms which are situated at the front and rear of IA and DA channel respectively. The high h at the front of DA owing to the high-amplitude waveform enables the fluid to absorb more heat as compared to IA, raising the average fluid temperature, as seen in Figure 6-6. With a higher fluid temperature, the fluid entering the second half of the DA possesses lower dynamic viscosity as compared to that at the front of the IA and thereby having higher flow inertia. Figure 6-7 shows that the fluid entering the region with a lower-amplitude waveform has a higher turbulence intensity in DA as compared to that in IA, causing a higher h . The turbulence intensity is defined as the ratio of the root-mean-square of the velocity fluctuations, to the mean flow velocity. On top of that, this effect causes the local h at the rear of DA to be comparable to that of UA, despite a lower-amplitude waveform, contributing to the highest average heat transfer coefficient throughout the flow rates. The same effect is also observed in UA, where the turbulence intensity is higher in the second half of the channel, reducing the difference in the local h between the front and rear of the channel.

Figure 6-5 shows that the positioning of the waveforms with the same amplitude also leads to different flow regime at the same flow rate. While DA does not show an

observable transition in the trend of the h with flow rate, IA which has a reversed positioning of the waveform depicts a transition beyond 5 l/min. UA manifests a change in the trend beyond 2 l/min.

Figure 6-6: Bulk fluid temperature along the flow direction

*The height of the channel is adjusted to a scale of 1:3 for better illustration of the contour.

Figure 6-7: Turbulence intensity of the channels at 3 l/min

6.4.2.3 Pressure loss across the configurations

Figure 6-8 shows the pressure loss across the different channel configurations. It has been shown in Section 4.4.2 that the pressure loss incurred increases exponentially with the increment in amplitude. The exceptionally high pressure loss incurred by the 0.15-mm amplitude leads to a higher pressure drop in DA and IA. However, as half of the flow in IA remains largely undistributed by the 0.05-mm amplitude waveform, the pressure drop in IA is only slightly higher than UA. As the flow fluctuation is significant downstream of DA despite the low-amplitude waveform, DA has a higher pressure drop than IA.

Figure 6-8: Pressure drop across the microchannels against flow rate

6.4.2.4 Thermo-hydraulic performance

Figure 6-9 shows the enhancement of heat removal capability of the channels, as compared to that of the straight configuration, at the same pumping power. At the lower range of pumping power of less than 2 W, DA has the highest heat removal capability at the same pumping power. At the lowest pumping power, DA has a performance of 1.41. This figure is 19.9 and 21.3 per cent higher than that of IA and UA. As the pumping power increases, UA shows an improvement in the thermo-hydraulic performance, outperforming DA beyond 2 W of pumping power. This channel achieves the highest performance index of 1.57, implying 57 per cent more heat removal as compared to the

straight configuration at the same pumping power. At this pumping power, UA can remove 4.0 and 25.6 per cent more heat as compared to DA and IA respectively.

Figure 6-9: Increment in the heat removal capability at the same pumping as the straight configuration

6.4.3 Effects of varied amplitude on the heated wavy wall

Figure 6-10 shows the heat transfer coefficients for the channels with a wavy heated surface. Akin to the previous configurations, DA_heater depicts a clear advantage as compared to IA_heater with different positioning of the high-amplitude waveforms. The largest difference can be as high as 41.1 per cent. The thermal performance of DA_heater and UA_heater is comparable, similar to the previous configurations for this range of flow rate.

Figure 6-10: Heat transfer coefficients against flow rate

DA_heater has a marginally higher pressure drop as compared to IA_heater owing to the flow fluctuations downstream of DA_heater. However, it is noted that the numerical model tends to underpredict the pressure loss across UA and DA configurations more than that of IA configuration, as shown in Figure 6-4. Therefore, it is expected that the pressure loss across these channels with heated wavy channels to depict similar trend to that of the measured data of the non-heated wavy channels. Although DA_heater and UA_heater have comparable heat transfer performance, the latter has a significantly lower pressure loss across the channel. DA_heater incurs a 46.5 to 49.0 per cent higher pressure drop as compared to UA_heater for this range of flow rate. Therefore, UA_heater clearly depicts a superior thermo-hydraulic performance at these flow rates.

Figure 6-11: Pressure drop across the microchannels against flow rate

Chapter 7 Eccentric and skewed annulus

This chapter presents the predicted heat transfer and hydrodynamic performance for three channel orientations: concentric, eccentric and skewed annuli. For eccentric and skewed annuli, three orientations are studied respectively, for two radius ratios. The numerical model is validated using measured data for the concentric configuration and analytic solutions for the eccentric channels. The performances of the eccentric and skewed annuli are contrasted against the concentric configuration.

7.1 Introduction

Although ultra-precision machining is possible with the traditional CNC methods, it is not energy efficient [101]. A loose machining tolerance and deformation during service result in eccentric annular channels. Therefore, this chapter investigates the effect of eccentricity, both uniform and changing eccentricity along the flow direction, on the single-phase convective heat transfer using liquid water in microscale gaps.

The effect of uniform eccentricity on both fluid flow and heat transfer has been extensively studied with a wide range of radius and eccentricity ratios. Existing studies on the eccentric annuli are mainly categorised into Newtonian and non-Newtonian fluids. Studies on eccentric annular non-Newtonian flows, covering analytical solutions, numerical solutions to physical approximations and purely numerical solutions, are crucial as this deformation to concentric pipes can be commonly found in applications like drilling mud flow during drilling operation, cement slurry flow during wellbore cementation, viscous polymer flow during extrusion process and drag-reducing coolant in a heat exchanger [142]. Many works have been published on this topic, as summarised by Escudier et al. [143], according to different boundary conditions. Nevertheless, as this chapter mainly focuses on the single-phase liquid water heat transfer, more details are covered on the works involving Newtonian fluids. Applications of Newtonian fluids in an annular gap can be commonly found in heat exchangers and nuclear reactors.

The flow field in eccentric pipes has been studied analytically. Snyder and Goldstein [18] presented the fully developed laminar flow friction factors in an eccentric annulus. This

study was compared with the experimental results presented by Jonsson and Sparrow [144]. Heyda [145] deduced the point velocity in laminar incompressible flow between eccentric circular cylinders, which was then validated numerically by Redberger and Charles [146]. Cheng and Hwang [41] and Trombetta [19] pioneered the analytical studies involving laminar convective heat transfer in eccentric annuli. Trombetta [19] studied hydrodynamically and thermally fully developed laminar forced convection in eccentric annuli analytically by applying approximate solutions to the energy equations. He developed fundamental solutions of the first, second and fourth kinds.

Manglik and Fang [147] studied numerically the fully developed laminar heat and flow characteristics in an eccentric annulus with $0 < \varepsilon^* < 0.6$ and $0.25 < r^* < 0.75$. This channel configuration is commonly found. The flow tends to stagnate in the narrow section and peak in the wide section, resulting in temperature non-uniformity and lower average heat transfer. Nevertheless, the reduction in the Nusselt number with increasing eccentricity comes with a reduction in friction factor.

Bourne et al. [20] performed experimental studies on liquid water flow in eccentric channels with a diameter ratio from 0.25 to 0.938 and a unit eccentricity, covering laminar, transitional and turbulent flow regimes. Their results indicate that the minimum friction factor is obtained at a diameter ratio of 0.75, for both laminar and turbulent regimes. The critical Reynolds number decreases with the increase in diameter ratio, until a ratio of 0.813. The trend is reversed beyond this ratio.

In another experimental study [148] that involved free convective air flow in annular pipes with a radius ratio of 0.24, imposed with insulated and constant heat flux boundaries, the rate of heat transfer increases up to an eccentricity ratio of 0.5, after which it decreases. Hosseini et al. [149] concluded that the heat transfer rate increases with eccentricity for turbulent forced convective air flow in a vertical eccentric annulus with a radius ratio of 0.24. Whereas for laminar flows, the heat transfer peaks at an eccentricity ratio of 0.5. The effect of eccentricity diminishes with increasing Reynolds number.

Nobari and Mehrabani [150] solved the continuity, momentum and energy equations of a hydrodynamically and thermally fully developed flow in an eccentric curved annulus with a radius ratio of 0.5, using a second-order finite difference method based on the projection algorithm. Their findings show that, in contrast to straight annuli, the presence of eccentricity in curved annuli enhances heat transfer rates at large Dean numbers with an increase in friction factor.

The existing works on the eccentric channels are mainly meant for the heat exchangers and nuclear reactors. The applicability of these works to a microscale gap has not been particularly studied. The flow is expected to be in the simultaneously developing regions and the analysis is rare in the existing literature. This is of interest as the size effect plays a pivotal role in the flow and heat transfer correlations as the scale decreases although the continuum assumption is still valid [57].

Chauhan et al. [151] presented a very detailed fluid flow field in the narrow gap of the eccentric channels at a wide range of radius ratio and eccentricity, ranges from 0.50 to 0.95 and 0.1 to 0.7 respectively, The study was conducted with fluid properties assumed at the inlet temperature for the Reynolds number range of 10^4 to 10^6 . The influence of eccentricity on the mass flux, velocity distribution, friction factor, Reynolds number, Nusselt number and temperature distribution were detailed in their work. The first portion of this study is similar to this but conducted in a laminar flow regime.

The second part of this study investigates a skewed annulus, which is defined as an annulus of varying eccentricity along its length. There is very limited work on a skewed annulus can be found in the literature. Haci and Cartalos [152] first studied the laminar and turbulent fluid flow in a skewed annulus. Heat transfer was not being considered in their study. The peripheral velocities exist across the annular space without any pipe rotation as the fluid favours the wider part of the annulus. The pressure drops for both laminar and turbulent flows across both eccentric and skewed annulus are found to lower than that in the concentric annulus. However, this is not the case for transitional flow.

Two radius ratios of 0.95 and 0.97 are investigated for the Reynolds range of 550 to 2200, for three eccentricity and skewness ratios respectively. The thermal and

hydrodynamic performances are contrasted to that of the concentric configurations. The predicted results for the concentric and eccentric channels are validated using available measured data and analytical solutions respectively to ensure the reliability of the predicted values. Subsequently, the flow fields are extracted to explain the resultant overall thermal and hydrodynamic performances.

7.2 Channel design

Equation (5-1) defines the radius ratio of an annular channel and Figure 7-1(a) shows the front view of the channel. There are two radius ratios which are investigated in this study, namely 0.95 and 0.97. For the concentric channels, the distance between the centres of the cylinders, δ equals to zero. δ is varied to achieve eccentricity ratios of 0.25, 0.50 and 0.75, as shown in Table 7-1.

$$r^* = \frac{r_i}{r_o} \quad (5-1)$$

Figure 7-1 (b) illustrates the side view of the eccentric channels where p and q represent the dimension of the narrowest and widest gap respectively, and their values are tabulated in Table 7-1.

Figure 7-1: Channel design for the eccentric channel

Skewed annular, on the other hand, has a varying eccentricity along the flow direction, as shown in Figure 7-2(b). Akin to the eccentric channels, an offset distance δ is introduced at the front of the channel. Nevertheless, this distance reduces linearly along

the length of the microchannel, until an equal but opposite offset distance is realised at the rear of the channel. This implies that the narrowest gap a , as shown in Figure 7-2 (b), increases to b at the rear of the channel, and vice versa. The eccentricity ratio at the front of the channel is adapted for the nomenclature of the channels. A larger eccentricity ratio results in a more skewed channel where the inner cylinder deviates more from the horizontal position. The values of the parameters as shown in Figure 7-2(b) are summarised in Table 7-2.

Figure 7-2: Channel design for the skewed channel

Table 7-1: Nomenclature of the eccentric microchannels

Microchannel	r^*	ϵ^*	δ (mm)	p (mm)	q (mm)
C_095	0.95	0.00	0.000	0.500	0.500
R095_E025		0.25	0.125	0.375	0.625
R095_E050		0.50	0.250	0.250	0.750
R095_E075		0.75	0.375	0.125	0.875
C_097	0.97	0.00	0.000	0.300	0.300
R097_E025		0.25	0.075	0.225	0.375
R097_E050		0.50	0.150	0.150	0.450
R097_E075		0.75	0.225	0.075	0.525

Table 7-2: Nomenclature of the skewed microchannels

Microchannel	r^*	ϵ^*	δ (mm)	a (mm)	b (mm)
C_095	0.95	0.00	0.000	0.500	0.500
R095_S025		0.25	0.125	0.375	0.625
R095_S050		0.50	0.250	0.250	0.750
R095_S075		0.75	0.375	0.125	0.875
C_097	0.97	0.00	0.000	0.300	0.300
R097_S025		0.25	0.075	0.225	0.375
R097_S050		0.50	0.150	0.150	0.450
R097_S075		0.75	0.225	0.075	0.525

7.3 Methodology

The governing equations, which include the continuity, momentum as well as energy equations in the solid and fluid domains, are solved numerically using ANSYS FLUENT solver. The governing equations are solved using the SIMPLE algorithm and the discretisation schemes are described in Section 3.3.7.

The following boundary conditions are applied:

- A mass flow inlet which corresponds to Reynolds number of 550 to 2200 and inlet temperature of 28 °C.
- An atmospheric pressure fluid outlet.
- A uniform heat flux of 58950 W/m² at the outer copper surface.
- Conservative heat flux at the solid-fluid interface.
- Constant solid properties [106].
- Variable fluid properties based on IAPWS IF97 standards [115].

Figure 7-3: Mesh independence test for R097_S050

The continuity and momentum solutions yield residual values at the order of 10^{-5} whereas the energy equations converge at the order of 10^{-7} . The residual values are observed to remain constant for at least 50 iterations before the solver is halted.

7.4 Results and discussion

7.4.1 Validation of the numerical model

The numerical model is validated in this section to ensure the accuracy and reliability of the predicted data. The predicted data for the straight configuration, with radius ratio of 0.97, is compared against the available measured data, as shown in Figure 7-4. The difference between the measured and predicted Nu is below 10 per cent. On the other hand, the difference between the measured and predicted f falls below 13 per cent.

Figure 7-4: Comparison between predicted data and measured data

The same numerical model is also used to predict a simultaneously developing flow for $Pr = 0.71$ in an eccentric channel with $r^* = 0.50$ and $\epsilon^* = 0.50$ studied analytically by Feldman as reported in [42]. The result is shown in Figure 7-5 which depicts a very good agreement with the analytical solutions.

Figure 7-5: Validation of numerical model for a simultaneously developing flow in an eccentric annular

7.4.2 Effect of eccentricity on the thermal performance

Figure 7-6 shows the Nusselt number of the channels whereas Figure 7-7 illustrates the ratio of the Nusselt number of the eccentric and skewed channels with respect to that of the concentric channel. By comparing Figure 7-6(a) and (b), eccentric and skewed channels with a higher radius ratio depict a more significant deviation of thermal performance from that of a concentric channel. The most significant deviation of thermal performance from the concentric channel is depicted by R095_E075 and R097_E075 respectively for each of the radius ratios. The former shows a deviation which is about 10 per cent whereas the latter shows a deviation close to 20 per cent. This is also obvious in Figure 7-7(a) where most of the data points are within the ± 5 per cent band whereas there are more data points outside of the ± 5 per cent band in Figure 7-7(b).

Figure 7-6: Average Nusselt number for the channels against Reynolds number

In addition, the thermal performance of the skewed channels is generally better than that of the eccentric channels. Figure 7-8 compares the local heat transfer coefficients along the flow direction of the concentric, eccentric and skewed channels for $Re = 550$. While the variation of the h_x of the skewed channel closely resembles that of that concentric channel, the h_x of the eccentric channel shows significant deviation at the rear of the channel.

(a) $r^* = 0.95$

(b) $r^* = 0.97$

Figure 7-7: Normalised Nusselt numbers with respect to the concentric configurations

The planes used to illustrate the flow fields are defined in Figure 7-9. The velocity component in the flow direction, in both eccentric and skewed channels, is plotted in Figure 7-10. The flow in the narrow gap of the eccentric channel decelerates along the flow direction, mainly due to the loss in the inertia of the flow as the viscous force dominates. The fluid favours the wide gap of the channel where the flow resistance is lower, resulting in an acceleration of fluid in the wide gap. This is because, further away from the inlet of the channel, more fluid moves to the wide gap where the flow resistance

is the lowest. This shift of flow is depicted in Figure 7-11 where the 3-dimensional streamlines are plotted.

Figure 7-8: Local heat transfer coefficients of the channels at Re = 550

The velocity contour plotted at the cross-sectional plane at the mid-length of the channel, as shown in Figure 7-13(a), shows that the bulk of the fluid flows in the wide gap, as the local velocity is higher at the wider gap. The slow-down of fluid in the narrow gap results in the build-up of the high-temperature zone right from the mid-channel plane to the rear of the channel, as shown in Figure 7-16(a). Although the fluid in the wider gap accelerates, the overall thermal performance deteriorates as seen in Figure 7-8. This might be because the fluid, which shifts from the narrow gap to the wide gap, is high in temperature, causing the thermal boundary layer to remain thick. Therefore, although the local heat transfer at the wide gap is relatively better than the narrow gap, the resultant copper wall temperature is higher than the concentric configuration, comparing Figure 7-16(a) and Figure 7-17(a).

The flow in the skewed channel depicts a slightly different pattern. As the gap size decreases, the flow first accelerates before decelerating, as shown in Figure 7-10. The opposite pattern is observed in the diverging gap. As seen in Figure 7-12, the fluid shifts away from the converging gap. As compared to the eccentric channel where the fluid begins to shift from the narrower gap at the front of the channel, the flow in the converging gap of the skewed channel experiences higher flow resistance at a later

position, where the flow gap becomes smaller. As seen in Figure 7-10, although the axial velocity in the converging gap experiences a drop, the axial velocity in the diverging gap increases at a lower rate. This is because, within the length of the microchannel, the bulk flow is still in the peripheral direction, causing a hotspot to build up next to the diverging flow gap, as illustrated in Figure 7-17(b). This is clearly seen in Figure 7-13(b) where the bulk fluid remains in the peripheral position. At this position, the flow gap size remains constant throughout the circumferential direction.

Figure 7-9: Nomenclature of the planes

Although this flow non-uniformity leads to the formation of hotspot next to the diverging gap, the higher heat transfer near to the converging flow gap and in the peripheral direction lowers the overall average copper wall temperature. This is shown by the relatively thinner thermal boundary layer in the diverging gap in Figure 7-14 (b).

Figure 7-10: Axial velocity in the eccentric and skewed channels for $r^* = 0.97$ and $Re = 550$

Figure 7-11: 3-D flow streamlines in R097_E550 at Re = 550

Figure 7-12: 3-D flow streamlines in R097_S550 at Re = 550

(a) R097_E075

(b) R097_S075

Figure 7-13: Mid-plane velocity contour at Re = 550

(a) R097_E075

(b) R097_S075

Figure 7-14: Mid-plane temperature contour at $Re = 550$

Furthermore, it can be observed from Figure 7-7 that the deterioration of thermal performance of the eccentric channels improves with Reynolds number. As the Reynolds number increases, the inertia of the fluid becomes more dominant as compared to the viscous force, thereby delaying the slow-down of the fluid. It can be seen from Figure 7-15 that the velocity reaches the minimum value at an earlier position at $Re = 550$ as compared to $Re = 2300$. The rate of velocity reduction is higher for $Re = 550$.

Figure 7-15: Axial velocity along flow direction: $Re = 550$ (left axis), $Re = 2300$ (right axis)

(a) $Re = 550$

(b) $Re = 2300$

Figure 7-16: Copper wall temperature for R097_E075

(a) Concentric configuration

(b) Skewed configuration

Figure 7-17: Copper wall temperature for C097 and R097_S075 at $Re = 550$

Figure 7-18: Axial velocity for eccentric channels with different radius ratio at $Re = 550$

Figure 7-18 compares the change of axial velocity along the flow direction for two radius ratios, i.e. 0.95 and 0.97. At a higher radius ratio, the effect of the eccentricity is more significant, justified by a more drastic drop in axial velocity at the same Re . This explains the observation that the effects of eccentricity on the thermal performance are more pronounced at a higher radius ratio.

7.4.3 Effect of eccentricity on the hydrodynamic performance

Figure 7-19 and Figure 7-20 compare the friction factors of the channels at the same Reynolds number. Majority of the channels experience an increment in the friction factor by less than 10 per cent. However, both R095_S075 and R097_S075 experience a hydrodynamic penalty which is higher than 30 and 20 per cent respectively. This is despite the fact that these channels exhibit very similar heat transfer performance as compared to the eccentric configurations.

On the other hand, as the bulk fluid in R095_E075 and R097_E075 flows in the wide gap, the flow in these channels experiences lower pressure loss. This is significant at

lower Re where the shift of the flow from narrow to wide gap takes place at an earlier position along the flow direction. Although these channels show a deterioration in the heat transfer performance by 10 and 20 per cent respectively at lower Re, these channels incur a lower pressure loss by a similar magnitude.

Figure 7-19: Friction factors of the channels against Reynolds number

(a) $r^* = 0.95$

(b) $r^* = 0.97$

Figure 7-20: Normalised friction factor with respect to the concentric configuration

Chapter 8 Converging microchannel

This chapter evaluates the thermal-hydraulic performance of four converging microchannels and a straight microchannel, with the same average hydraulic diameter. In addition, the convective and conductive heat transfer areas in the substrate are kept constant, contrasting this study with the existing works. The thermal and hydrodynamic performances are quantified using the measurement system for the Re range of 1300 to 3400 and a constant heat flux of 53.0 W/cm^2 . More importantly, the heat removal capability of the converging channels is compared with that of the straight channel at the same pumping power.

8.1 Introduction

In addition to the wavy microchannels, a converging flow channel along the stream-wise direction is another emerging area of channel configuration design. A converging channel is a channel with a decreasing hydraulic diameter. This configuration increases the fluid velocity along the flow direction, increasing the local heat transfer coefficient and reducing the local wall temperature. The overall temperature uniformity and thermal resistance in the heat sink are improved with such channel configuration. This is essential in microchannels where the temperature variation within the heat sink between the inlet and outlet can be huge owing to the small amount of coolant [21].

Hung and Yan [21] studied numerically the effect of tapering ratio on the thermal resistance and temperature distribution of the heat sink, subjected to fixed pumping power. Their study proves that there is an optimum tapering ratio which yields the minimum thermal resistance. Another study conducted by Hung et al. [22] shows the superior performance, in terms of thermal resistance, of a tapered channel over the single-layered and double-layered straight microchannels with the same pumping power.

Dehghan et al. [153] proved that the convective heat transfer coefficients surge downstream of the converging flow channels, instead of being constant for a straight channel as the entrance effect diminishes. The average hydraulic diameter of the channels varies with the tapering ratio. Osanloo et al. [23] concluded through numerical

predictions that, the thermal resistance improves with a higher convergence angle. Nevertheless, this comes with a higher pressure drop. In the lack of an assessment on whether the additional pumping power required is justifiable by the improved thermal performance, Wong and Ang [24] predicted numerically that in a converging double-layered heat sink, there is a minimal increment in the heat transfer performance with a significant rise in the pumping power requirement. Therefore, it was concluded that the thermal-hydraulic performance of a straight microchannel is better than a tapered one.

The conductive heat transfer in the substrate becomes significant as the dimensions of the flow channel scale down. In the existing implementations, a tapered channel with a higher convergence angle has more conductive heat transfer area and higher convective heat transfer coefficient [21]. Therefore, the effect of tapering ratio, sans the effect of convective and conductive heat transfer areas, on the convective heat transfer and fluid flow in the microchannel is not examined. It was proven in [21] the opposing effects incurred by the aforementioned factors can be significant on the overall thermal resistance and the dominant mechanisms rely on the tapering ratio.

Hence, this chapter presents a study in which the thermal performance depends solely on the effect of tapering on the convective heat transfer coefficient. The convective and conductive heat transfer areas are kept constant. The heat removal capability is evaluated against a straight channel at the same pumping power.

8.2 Channel design

Figure 8-1 shows a converging flow channel. The diameter of the copper block, D_2 remains constant at 20.0 mm. Converging flow gaps with four different converging gradients are realised by varying the diameter of the insert according to the dimensions stated in Table 8-1.

Figure 8-1: Converging flow channel

Table 8-1: Nominal flow channel dimensions

Channel	D_1 (mm)	D_2 (mm)
T1	19.3	19.5
T2	19.2	19.6
T3	19.1	19.7
T4	19.0	19.8
Straight	19.4	19.4

The fabricated inserts are measured at six equally-spaced positions, for the external diameter. The average measured values are presented in Figure 8-2. Furthermore, the maximum dimensionless surface roughness computed by Equations (3-3) and (3-4) is 1.19. Therefore, the channels are considered hydraulically smooth.

Figure 8-2: Measured average diameter of the inserts at different position

8.3 Methodology

The temperature and pressure data are measured from the volumetric flow rate of 2.0 to 5.0 l/min, which corresponds to the Re range of 1300-3400. The experimental runs are conducted at an interval of 0.25 l/min with the steady conditions stated in Section 3.2.6. The nominal heat supply and heat transfer area are kept constant at 1000 W and 18.85 cm² respectively. The data is reduced to the parameters of interest as described in Section 3.2.6.

Repeatability tests are conducted at random experimental runs and the results are shown in Table 8-2. A maximum difference of 0.18% and 0.69% in the heat transfer and pressure drop data ascertains the excellent repeatability of the measured results.

Table 8-2: Repeatability tests

Flow rate (L/min)	Channel	Discrepancy in h (%)	Discrepancy in Δp (%)
2.00	T2	0.10	0.48
3.00	straight	0.02	-0.23
3.25	T1	-0.18	-0.69
4.50	T3	-0.07	-0.60

The maximum uncertainty values for the parameters of interest, reported with 95% confidence level are shown in Table 8-3.

Table 8-3: Uncertainty values

Parameter	Maximum uncertainty (%)
h	8.66
$\dot{V}$	0.115
Δp	11.3

8.4 Results and discussion

Figure 8-3 shows the heat transfer coefficients for the microchannels. Akin to the measured results as shown in the previous chapters, the straight microchannel depicts a transition in trend beyond the flow rate of 3 l/min, suggesting a change in the flow regime from laminar to transitional or turbulent. Channel T1, which has the smallest converging gradient, shows a similar transition at a slightly higher flow rate. Channel T2 and T3, on the other hand, show a transition beyond 6 l/min. This suggests that the channel constriction delays the transition of the flow. A higher degree of constriction suggests more significant viscous effects on the fluid along the direction of the flow. The local Reynolds number decreases with the flow direction. The heat transfer data for channel T4 which has the highest converging gradient, is collected up to 5 l/min owing to the significantly higher pressure drop incurred. The current measurement system is not able to measure this channel configuration at higher flow rates.

Prior to the transition in the flow regime for the straight microchannel, channels T3 and T4 which have relatively higher converging gradients, yield higher heat transfer coefficients as compared to the straight configuration at the same flow rate. The trend changes at higher flow rates, supported by the higher gradient of increment in h after the transition in the straight channel.

Figure 8-3: Heat transfer coefficients against flow rate

Figure 8-4 illustrates the pressure loss across the configurations. The straight channel, as well as Channels T1 and T2, incur similar pressure loss across the flow rates. Owing to the higher degree of constriction, channels T3 and T4 incur significantly higher pressure loss. In contrast to the increment in h which is almost linear, the increment in the pressure loss has an increasing gradient. This dictates the need to evaluate the thermo-hydraulic performance of the channels below.

Figure 8-4: Pressure loss across the configurations against flow rate

Figure 8-5 shows the evaluation of the increment in heat removal capability evaluated at the same pumping power. As the ratios are less than one, the channels possess lower thermo-hydraulic performance as compared to the straight microchannel. The increment in h in both T3 and T4 is outweighed by the deterioration in the hydrodynamic performance. The performance drops with the increment in pumping power and the converging gradient. It can be concluded that the straight channel outperforms the converging flow channels in terms of thermo-hydraulic performance.

Figure 8-5: Increment in heat removal capability at the same pumping power

Chapter 9 Conclusions and Recommendations

This chapter concludes the key findings of this Ph.D. research study. These conclusions are made and presented in sequence, based on the measured and predicted data reported from Chapter 4 to 8. After a thorough reflection on the findings, the opportunities for future work are recommended.

9.1 Conclusions

This Ph.D. study examines the thermo-hydraulic performance of the following microchannel configurations: single-wavy-wall channels, eccentric channel, skewed channel and converging channel. These channels are formed by superimposing two macro geometries, resulting in a microscale annular gap. Detailed experimental measurements and validated numerical predictions have been conducted and the key findings are summarised for each of the configurations as follows:

9.1.1 Single-wavy-wall microchannel

9.1.1.1 Non-heated wavy profile

This channel is formed by a heated flat wall and a non-heated sinusoidal wavy wall. This channel configuration results in a periodic expanding and contracting microchannel, introducing the re-entrance effects where the local heat transfer performance is better. The effects of the waveform amplitude and wavelength on the heat transfer and thermo-hydraulic performance are studied through the implementation of three waveform amplitudes and three wavelengths.

A channel with high-amplitude and short-wavelength waveforms achieves constantly higher heat transfer coefficients. This channel records a minimum increment of 100 per cent in h as compared to that of a straight microchannel at the same flow rate. At the highest measured flow rate, the enhanced channel achieves a h of 51.1 kW/m²·K while the straight channel records a h of 25.3 kW/m²·K.

The amplitude and wavelength of the waveforms dictate the heat transfer performance. A high-amplitude waveform results in the formation of recirculation zone in the trough

of the waveform, effectively reduces the actual flow gap and enhances the heat transfer performance. Therefore, the lowest h_x is shifted slightly downstream of the trough of the waveform. However, the recirculation zones are less obvious in channels with the same amplitude but longer wavelength. In the absence of the recirculation zone, the local h experiences a sudden drop after reaching the peak value as dictated by the profile of the waveform.

In overall, the effect of the waveform amplitude on the heat transfer is more dominant as compared to that of the wavelength. Increasing the number of waveforms along the microchannel length from 3 to 5 and 10 achieves a maximum enhancement in h of 51.7 and 96.4 per cent respectively, recorded at the same flow rate. On the other hand, increasing the waveform amplitude from 0.05 to 0.10 and 0.15 mm improves the heat transfer coefficient by a maximum of 55.6 and 151.8 per cent respectively at the same flow rate and wavelength.

The highest thermo-hydraulic performance is recorded by the channel with a high-amplitude and short-wavelength waveform, i.e. sin_015_3. This channel is able to achieve a 64 per cent improvement in heat removal capability when evaluated at the same pumping power as a straight microchannel. However, the thermo-hydraulic performance of sin_015_3 deteriorates as the pumping power increases and the reverse is true for sin_005_3, which has a waveform with the lowest wave amplitude. The difference between these channels drops from 43.6 to 9.5 per cent as the supplied pumping power increases. Noticeably, sin_005_3 has consistently higher thermo-hydraulic performance as compared to sin_010_3 despite the lower waveform amplitude.

The effect of wavelength on thermo-hydraulic performance is less obvious as compared to that of the waveform amplitude. A difference of less than 6.4 per cent is recorded as the same-amplitude waveform is increased from 5 to 10 within the microchannel length. Increasing the number of waveforms from 3 to 10 improves the thermo-hydraulic performance by an average of 25.7 per cent.

Working correlations for the average Nusselt number and friction factor for this channel configuration are proposed as follow:

$$Nu = 0.034 Re^{0.7} Pr^{\frac{1}{3}} \left[1 + \frac{7}{L/D_h} + 0.6 \frac{(a/H)^{1.3}}{(\lambda/L)^{0.7}} \right]$$

$$f = \frac{1}{Re^{0.3}} \left[0.7 + \frac{0.7}{L/D_h} + 2.7 \frac{(a/H)^3}{(\lambda/L)} \right]$$

The applicability range of the correlations are as follows:

Microchannel	Re range
sin_005_3	2800 – 4600
sin_010_3	2200 – 4600
sin_015_3	1380 – 4600
sin_015_6	1550 – 4600
sin_015_10	1710 – 4600

These correlations are based on 92 steady-state measurements. The average deviation from the measured Nusselt number and friction factor is 5 and 4 per cent respectively, while the maximum deviation is 15 and 12 per cent respectively.

9.1.1.2 Comparison among channel with heated wavy profile, channel with non-heated wavy profile, serpentine channel and straight channel

Three wavy channel configurations are compared to a straight microchannel. These include: (i) wavy-heater channel, a channel with a heated sinusoidal wavy wall and a non-heated flat wall; (ii) wavy-insert channel, a channel with a heated flat wall and a non-heated sinusoidal wavy wall; and (iii) serpentine channel, a channel with in-phase sinusoidal wavy walls. These waveforms have the same amplitude and wavelength. As compared to a straight microchannel, the wavy-heater, wavy-insert and serpentine channel achieve a maximum increment in h of 261.8, 194.1 and 79.1 per cent respectively, predicted at the same flow rate.

In the single-wavy-wall channels, i.e. wavy-heater and wavy-insert channel, recirculation zones are found in the troughs of the waveform. In the wavy-insert channel, these recirculation zones effectively reduce the flow gap and therefore improving the local heat transfer coefficient at the maximum flow gap formed by the heated wall and the trough of the waveform. Whereas in the wavy-heater channel, besides reducing the actual flow gap, these rotating vortices aid in the heat removal from the heated wall. Therefore, the wavy-heater channel records a minimum enhancement of 23.0 per cent over the wavy-insert channel.

In the serpentine channel, the channel curvature shifts the maximum velocity from the centreline towards the heated wall after passing the peak and thereby improving the heat transfer coefficient. However, the enhancement is less significant as compared to the single-wavy-wall configurations. The wavy-insert and the wavy-heater channel record a minimum enhancement of 47.1 and 102.0 per cent in heat transfer coefficient as compared to the serpentine channel.

The wavy-heater and wavy-insert channels incur similar pressure drop, ranges from 3.8 to 7.5 times as compared to the straight channel. The serpentine channel, on the other hand, incurs a higher pressure drop of 17.4 to 52.8 per cent.

In terms of thermo-hydraulic performance, these channels have a performance index greater than 1. However, the improvement of the serpentine channel in terms of the heat removal at the same pumping power as compared to the straight channel is less than 10 per cent. The wavy-insert channel achieves a minimum performance index of 1.20. This value increases with the pumping power to a maximum of 1.38. The wavy-heater channel has the highest performance index of 1.88 and the lowest index of 1.61. At the same pumping power with the highest performance index, the wavy-heater removes 36.8, 74.1 and 88.2 per cent more heat as compared to the wavy-insert, serpentine and straight channel respectively.

9.1.1.3 Single-wavy-wall microchannel with varied amplitude waveform

A change in the waveform amplitude along the microchannel length is introduced to improve the heat transfer and thermo-hydraulic performance of the single-wavy-wall microchannels, i.e. wavy-insert and wavy-heater, which have been studied in the previous section. The waveform amplitude is increased, decreased and kept uniform along the flow direction. These channels are named IA, DA and UA respectively for the wavy-insert configurations; IA_heater, DA_heater and UA_heater respectively for the wavy-heater configurations.

Opposite positioning of the same-amplitude waveform in the IA and DA channel yields different h . DA has a h which is consistently 21.0 per cent higher than that of IA. In DA, the fluid disturbance incurred by the higher-amplitude waveforms upstream is brought downstream, incurring a higher local heat transfer coefficient in the low-amplitude zone as compared to that in IA. In IA, these low-amplitude waveforms are positioned upstream of the channel.

Similarly, this phenomenon in DA leads to a similar h at the rear of the channel as compared to that of UA despite the lower waveform amplitude. Therefore, the higher h incurred by the higher-amplitude waveform upstream of DA leads to an enhanced h of 58.3 per cent as compared to UA. As the flow rate increases, the difference diminishes to less than 10 per cent.

At the lowest pumping power, DA has a performance index of 1.41, which is 19.9 and 21.3 per cent higher than that of IA and UA respectively. UA experiences an improvement in the performance index as the pumping power increases, to a maximum index of 1.57. At this pumping power, UA is able to remove 4.0 and 25.6 per cent more heat as compared to that of DA and IA respectively.

For the channel with a heated wavy wall, DA_heater and UA_heater show similar heat transfer performance of within 3 per cent of each other for the range of flow rate examined. However, the former incurs a pressure drop of 44.8 to 49.0 per cent higher than that of the latter.

9.1.2 Eccentric and skewed channel

Loose manufacturing tolerance and deformation during service might lead to eccentric and skewed channels. The effects of the eccentricity and skewness on the heat transfer and hydrodynamic performances are predicted at three eccentricity ratios for annular channels of two radius ratios.

An eccentric channel has a narrow and a wide flow gap which are situated on the opposite side to each other. The narrow gap poses significant flow resistance, and therefore causing the flow to move to the wider gap. This is proven by the presence of the velocity in the peripheral direction as well as, a slow-down of the fluid along the narrow gap and an acceleration of flow in the wide gap despite the constant gap size along the flow direction. This causes hotspots to be formed neighbouring the narrow flow gap. The channel with the highest eccentric ratio shows the most significant deviation from the concentric configuration, exceeding 10 per cent in this case. The deterioration in heat transfer reduces with the increment in Re which signifies an increase of flow inertia over viscosity. These channels have a lower friction factor by a similar order of magnitude, at the same Re .

A skewed channel has a converging flow gap and a diverging flow gap which are located on opposite end to each other. The flow first accelerates in the converging flow gap before decelerating due to the shift of fluid from the converging gap to the diverging gap. Within the length of the microchannel, the flow is mostly flowing in the peripheral direction, causing hotspots to be formed neighbouring the diverging gap. The delay in the shift of fluid from the axial to the peripheral direction in the skewed channels, as compared to the eccentric channels, causes a deviation in heat transfer performance which is less than 5 per cent from that of the concentric configurations. However, the skewed channel with the highest skew ratio incurs more than 20 per cent higher in friction factor, at the same Re .

9.1.3 Converging channel

The heat transfer coefficients and pressure losses across the channel are measured for four converging channels with different converging gradient.

At lower flow rates, the channels with higher converging gradients have a higher h as compared to the straight channel. As the flow rate increases, the flow in the straight channel depicts a transition, showing an accelerated increment in h with the flow rate and possessing a higher h as compared to the converging channels as the flow rate increases. The channels with a higher converging gradient show a delay in the transition of the flow regime.

All the converging channels possess a performance index which is less than 1, implying that the straight microchannel is able to remove more heat given the same pumping power. The performance index of the converging channels decreases with the increment in pumping power and converging gradient.

9.2 Recommendations

The works completed during this Ph.D. study, both experimentally and numerically, have provided good fundamental understanding and quantified performances on a few channel configurations. The author believes that it has paved the way to several exciting research directions as follows:

9.2.1 Extension of the range of the measurement system

The current measurement system is designed for a flow rate up to 10 l/min and a pressure drop of 3.5 bars. These hydrodynamic constraints limit the upper limit of the pumping power measured.

This is because the straight microchannel, when measured at 10 l/min, incurs a relatively low pumping power owing to the low pressure loss across the configuration. Therefore, when this channel is taken as the reference of the other enhanced channels, the upper limit of the pumping power taken for reference is limited by the maximum pumping power incurred by the straight microchannel at 10 l/min. In order to extend the range of pumping power for the evaluation of the heat removal enhancement at the same pumping power, the range for the flow rate measurement has to be extended.

Figure 9-1 is extracted from Chapter 4. It is noted that the performance index of sin_005_3 increases with the pumping power while the reverse trend is observed for sin_015_3 and sin_015_6. Extending the range of comparison would be useful in making a quantified conclusion on the preferred channel with a higher operating efficiency given the actual operating conditions. The maximum pumping power incurred by sin_015_3 at the highest measured flow rate is 41.4 W.

Figure 9-1: Heat transfer increment evaluated at the same pumping power

9.2.2 Proposal of new correlation for the single-wavy-wall configuration

For the single-wavy-wall configurations, it has been proven that the heated wavy profile has higher heat transfer coefficients and similar pressure losses as compared to the non-heated wavy profile, as shown in Figure 9-2 and Figure 9-3. Given the enhanced performances, it is strongly recommended that working correlations are proposed for this configuration so that further adoptions of this configuration are possible.

Furthermore, it has been observed that one of the strongest contributors to the thermal performance of this single-wavy-wall configuration is the strength of the recirculation zone formed in the trough. Chapter 4 also concludes that the strength of this zone is a function of the waveform amplitude and wavelength. Therefore, it is recommended further studies include a parametric study on the waveform amplitude and wavelength when comparing the performances of these different wavy channel configurations.

Figure 9-2: Heat transfer coefficient for various channel configurations

Figure 9-3: Pressure loss across various channel configurations

9.2.3 Measurement with a different heating boundary condition

Figure 9-2 reports the superiority of single-wavy-wall configurations over the serpentine channel in terms of heat transfer performance at this same flow rate. However, it is noted that the performances are evaluated under a heated wall and a non-heated wall boundary

condition. The thermal performances of the serpentine may have been underestimated given that the channel curvature technique shifts the maximum velocity core to the non-heated surface too, as shown in Figure 9-4. The performance of this channel might be improved if the measurements are taken under double heated wall boundary conditions. The overall thermo-hydraulic performance is hypothesised to be improved given the low pressure drops across this serpentine configuration.

Figure 9-4: Temperature distribution in the serpentine channel

9.2.4 Optimisation of waveform amplitude along the flow direction

The variation of waveform amplitude along the flow direction has shown superiority under a low pumping power condition. Given the same pumping power, implementing a variation in the waveform amplitude along the flow direction gives rise to an enhancement of 21.3 per cent as compared to a uniform waveform amplitude configuration, as shown in Figure 9-5.

A variation in the amplitude along the flow direction is recommended given that the fluid disturbances incurred by the high-amplitude waveform can be carried downstream. The waveform amplitude should be allowed to vary to achieve the maximum heat transfer performance while incurring the minimum pressure loss across the configuration. An algorithm for the topology optimisation using numerical methods for the maximum thermo-hydraulic performance can be implemented to achieve this purpose.

Figure 9-5: Increment in the heat removal capability at the same pumping power as the straight configuration

9.2.5 Correction factor for the eccentric and skewed annular

The deviation of the Nusselt number and friction factor of the eccentric and skewed annular from the concentric channel is a function of the microchannel length, radius ratio, eccentricity ratio, skewness ratio and Reynolds number. A correction factor as a function of these factors can be formulated through a parametric study to predict the performances of these channels in the simultaneously developing flow regime, for a relatively short channel.

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Appendix A

Type B Uncertainty Budget

Non-statistical Evaluation	Distribution	15	20	25	30	35	40	60	80	100
PRT calibration uncertainty	k=2	0.00106	0.001414	0.001766	0.00212	0.00248	0.00282	0.00424	0.00566	0.00706
	k=1	0.000530	0.000707	0.000883	0.00106	0.00124	0.00141	0.00212	0.00283	0.00353
PRT drift (per 500 hours)	Rectangular	0.0106	0.0109	0.0111	0.0113	0.0115	0.0117	0.0126	0.0135	0.0144
	k=1	0.00612	0.00629	0.00641	0.00652	0.00664	0.00675	0.00727	0.00779	0.00831
Bath uniformity	Rectangular	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
		0.0115	0.0115	0.0115	0.0115	0.0115	0.0115	0.0115	0.0115	0.0115
Bath stability	Rectangular	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
	k=1	0.0231	0.0231	0.0231	0.0231	0.0231	0.0231	0.0231	0.0231	0.0231
Total standard uncertainty		0.0265	0.0266	0.0266	0.0266	0.0267	0.0267	0.0269	0.0271	0.0273
Expanded uncertainty (k=2)		0.0531	0.0531	0.0532	0.0533	0.0533	0.0534	0.0538	0.0542	0.0547