HEAT TRANSFER INTENSIFICATION IN MICROCHANNEL HEAT SINKS (MCHS) - WAVY CHANNELS EMBEDDED WITH PIN FINS

Anas Mohammad Amin Alkhazaleh

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HEAT TRANSFER INTENSIFICATION IN MICROCHANNEL HEAT SINKS (MCHS) - WAVY CHANNELS EMBEDDED WITH PIN FINS

Anas Mohammad Amin Alkhazaleh

This thesis is submitted in partial fulfilment of the requirements for the degree of Master of Science in Mechanical Engineering

Under the Supervision of Dr. Bobby Mathew

June 2021
Declaration of Original Work

I, Anas Mohammad Amin Alkhazaleh, the undersigned, a graduate student at the United Arab Emirates University (UAEU), and the author of this thesis entitled “Heat Transfer Intensification in Microchannel Heat Sinks (MCHS) - Wavy Channels Embedded with Pin Fins”, hereby, solemnly declare that this thesis is my own original research work that has been done and prepared by me under the supervision of Dr. Bobby Mathew, in the College of Engineering at UAEU. This work has not previously formed the basis for the award of any academic degree, diploma or a similar title at this or any other university. Any materials borrowed from other sources (whether published or unpublished) and relied upon or included in my thesis have been properly cited and acknowledged in accordance with appropriate academic conventions. I further declare that there is no potential conflict of interest with respect to the research, data collection, authorship, presentation and/or publication of this thesis.

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Abstract

This work conceptualizes heat sinks for electronics employing wavy/sinusoidal microchannels embedded with pin fins and subsequently analyzes their performance in terms of thermal resistance, pressure drop, maximum chip temperature, and the associated pumping power due to the pressure drop. The conceptualized wavy Microchannel Heat Sink (MCHS) is mathematically modeled using a combination of governing equations including energy equations, continuity equation, and Navier-Stokes equations. The performance of the wavy microchannel heat sink embedded with pin fins is evaluated based on a parametric study covering multiple parameters; microchannel’s amplitude, frequency, hydraulic diameter, pin fins’ diameter, and location, and Reynolds number. The different mathematical models are solved numerically using Computational Fluid Dynamics (CFD) techniques applying the different operating and geometric parameters. The performance of the wavy MCHS is also compared to the performance of other designs (straight, straight embedded with pin fins, and wavy microchannel heatsinks). The performance of selected cases of the studied wavy MCHS is investigated experimentally and compared with the simulation results to validate the results and obtain a better understanding of the actual performance of these designs. The performance of the wavy MCHS compared to the straight and straight microchannel embedded with pin fins heat sinks shows less thermal resistance for the wavy MCHS at the same operating parameters. Introducing pin fins to the wavy microchannel enhances the thermohydraulic performance achieving less thermal resistance but with a cost of an increase in pressure drop. Increasing both amplitude and frequency shows improvement in the thermohydraulic performance but also with a cost of an increase in pressure drop. The pressure drop associated with increasing the pin fins diameter happens to increase the pressure more significantly than the other geometric parameters. On the other hand, increasing the hydraulic diameter shows good improvement in the thermohydraulic performance, reducing thermal resistance and pressure drop.

Keywords: Liquid cooling, heat sink, sinusoidal microchannel, wavy microchannel, pin fins, thermal resistance, pressure drop.
تمكين انتقال الحرارة في المشتتات الحرارية ذات القنوات الدقيقة باستخدام تصاميم موجة مدمجة مع امتدادات مسمارية

المقدمة

هذا العمل يقدم تصميم مشتت حراري للإلكترونيات بتوظيف تصميم موج ذو قنوات متناهية الصغر مدمجة مع امتدادات مسمارية، وبعد ذلك يحلل الأداء من حيث المقاومة الحرارية، وانخفاض الضغط، ودرجة الحرارة القصوى للرقاقة، وما يرتبط بها من قوة الضخ اللازمة بسبب انخفاض الضغط. المشتت الحراري ذو القنوات المتناهية الصغر (MCHS) المقترح تم تصميمه رياضيًا باستخدام المعادلات الحاكمة للحالة التالية: معادلة الطاقة، معادلة الاستمرارية، ومعادلات نافير ستوك. يتم تقسيم أداء المشتت الحراري بناءً على دراسة بارامترية تغطي العديد من العوامل الهندسية: سعة وتردد الموجة، القطر الهيدرولكسي، قطر وموقع الامتدادات المسمارية، ومعامل التشغيل رقم رينولدز. يتم حل النماذج الوراثية المختلفة باستخدام التشغيل العددي بناءً على تقنيات ديناميكية الموائع الحوسبة (CFD)، بناءً على المعابير التشغيلية والهندسية المختلفة. يتم أيضًا مقارنة أداء المشتت الحراري المتموج بأداء التصاميم الأخرى (مستقيم، مستقيم مدمج مع امتدادات مسمارية، موج). تم فحص أداء بعض الحالات المختارة تجريبيًا ومقارنته بنواتج المحاكاة للتحقق من صحة النتائج والحصول على أفضل الأداء الفعلي لهذه التصاميم. يظهر أداء المشتت الحراري المتموج ذو القنوات المتناهية الصغر متناهية جداً بالقناة الدقيقة المستقيمة والدقيقة المتموجة المدمجة مع امتدادات مسمارية مقاومة حرارية أقل عند استخدام نفس معلمات التشغيل. يعمل إدخال الامتدادات مسمارية في القناة الدقيقة المتموجة على تحسين الأداء التشغيلي مما يحقق مقاومة حرارية أقل، ولكن مع تكلفة زيادة انخفاض الضغط. تظهر زيادة زيادة كل من السعة والتردد تحسنًا في الأداء الهيدرولكسي الحراري، ولكن أيضًا مع زيادة انخفاض الضغط. انخفاض الضغط المرتبط بزيادة قطر الامتدادات مسمارية له تأثير أكبر من المعاملا الهندسية الأخرى. من ناحية أخرى، تظهر زيادة قطر الهيدرولكسي تحسنًا جيدًا في الأداء الهيدرولكسي الحراري، مما يقلل المقاومة الحرارية وتقليل انخفاض الضغط.

مفهوم البحث الرئيسي: التبريد باستخدام السوائل، مشتت حراري، قناة جيبية دقيقة، قناة دقيقة متموجة، امتدادات مسمارية، المقاومة الحرارية، انخفاض الضغط.
Acknowledgements

I would like to thank my committee - advisor and co-advisors - for their guidance and support throughout my journey at UAEU. My special thanks go to my advisor Dr. Bobby Mathew who guided me since day one. He provided me the support and assistance I needed, which helped me to accomplish this work, and for that, I am very grateful. I would also like to extend my gratitude and appreciation to the Mechanical Engineering department members and everyone who helped me throughout my journey. My sincere thanks also go to Prof. Mohsen Sherif and staff at National Water and Energy Center (NWEC) at UAEU for providing the computational resources for completing this thesis. I am especially grateful to my wife, parents, and family, who supported me and shared their love and care through thick and thin.

We raise to degrees whom we will, but over all those endowed with is the All-Knower (Allah). Yusuf - 12:76
Dedication

To my beloved wife, parents, and family
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List of Abbreviations and Symbols

List of Abbreviations

CFD  Computational Fluid Dynamics
CPU  Central Processing Unit
LMTD Log Mean Temperature Difference (K)
MCHS Microchannel Heat Sink
SIMPLE Semi-Implicit Method for Pressure-Linked Equations

List of Symbols

\( A \)  Area (m\(^2\))
\( c_p \) Specific Heat Capacity (J/kg.K)
\( D_{hy} \) Hydraulic Diameter (\(\mu\)m)
\( d \) Diameter of Pin Fin (\(\mu\)m)
\( h \) Heat Transfer Coefficient (W/m\(^2\)K)
\( k \) Thermal Conductivity (W/mK)
\( n \) Direction Normal to The Surface
\( \text{Nu} \) Nusselt Number
\( P \) Fluid Pressure (N/m\(^2\))
\( PP \) Pumping Power (W)
\( R_{th} \) Thermal Resistance (K/W)
\( Re \) Reynolds Number
\( T \) Temperature (K)
\( u \) Velocity in The x-Direction (m/s)
\( \dot{V} \)  Volumetric Flow Rate (m\(^3\)/s)

\( v \)  Velocity in The y-Direction (m/s)

\( w \)  Velocity in The z-Direction (m/s)

\( q'' \)  Heat Flux (W/m\(^2\))

**Greek Alphabets**

\( \Delta P \)  Pressure Drop (Pa)

\( \Delta T \)  Temperature Difference (K)

\( \mu \)  Viscosity (Pa.s)

\( \rho \)  Density (kg/m\(^3\))

\( \chi \)  Uncertainty

**Subscript**

\( \text{avg} \)  Average

\( \text{chip} \)  Microelectronic chip

\( \text{condo} \)  Conduction

\( \text{conv} \)  Convection

\( f \)  Fluid

\( H \)  Heater

\( \text{in} \)  Microchannel Inlet

\( \text{out} \)  Microchannel Outlet

\( s \)  Substrate

\( \text{SD} \)  Standard Deviation

\( w \)  Wall
Chapter 1: Introduction

1.1 Overview

Advancement in technologies related to microelectronic chips has led to them exhibiting increased processing capability and this has come at the expense of increased consumption of electrical power. Moreover, as there is no work output from microelectronic chips, all the electrical power input into the same is converted into heat as dictated by the 1st law of thermodynamics. The advancement in technologies related to microelectronic chips has additionally reduced their footprint and this combined with increased consumption of electrical power has led to the same exhibiting high heat fluxes [1]. Figure 1 shows the growth rate of the number of transistors on-chip for different chips. The increase in the number of transistors on electronic chips is proportional to the electrical power consumption of the same.

Figure 1: The number of on-chip transistors integrated on a chip and evolution of microchip size [2]

The increased heat flux associated with microelectronic chips demands advanced thermal management solutions for the same. Direct and indirect thermal management schemes have been employed for the thermal management of
microelectronic chips. The coolant is in direct contact with the microelectronic chip in the case of direct thermal management schemes, while in the case of indirect thermal management schemes, the coolant is not in direct contact with the microelectronic chip. Direct thermal management schemes have the advantage that there is no intermediary in the heat transfer between the microelectronic chip and coolant. In indirect thermal management schemes, a heat transfer device is used as an intermediary between the microelectronic chip and the coolant. The key disadvantage of direct heat transfer schemes is the limited choice of coolants; only dielectric fluids are recommended in these schemes. On the other hand, the demerit of indirect heat transfer schemes is the additional thermal resistance, in the heat transfer path between microelectronic chip and coolant, due to the presence of the heat transfer device. One of the heat transfer devices used in indirect thermal management schemes is the heat sink.

Heat sinks can be classified into three categories - passive, active, and hybrid [3]. In passive heat sinks, natural convection dissipates the generated heat. The advantage of these heat sinks is that there is no need for a control system or fans/pumps, therefore they are less complicated, but this comes with a price, the efficiency is limited and less than active heat sinks. In the active heat sinks, forced flow increases the convection heat transfer thereby improving heat dissipation. Finally, in hybrid heat sinks, a control system decides whether to use the passive or active approach based on the heat needed to be dissipated.

Microchannel heat sinks can be used for different applications where the devices' performance and functionality are highly affected by the generated heat; one example is the microelectronic chips used in High-Performance Computing (HPC) systems and data centers. Nowadays, even regular workstations and gaming systems
require a proper heat dissipation system to keep their computing capabilities as high as possible. Another application is the dissipation of the heat in the Concentrated Photovoltaic cells so as to improve the overall performance. Other applications of the microchannel heat sinks include cooling high-powered electronic components such as Insulated Gate Bipolar Transistors (IGBTs) and laser diode arrays. Figure 2 shows different applications of microchannel heat sinks.

Figure 2: Different applications of microchannel heat sinks. (a) High-power diode bars cooling [4], (b) Concentrated Photovoltaic cooling [5]
Approaches adopted for enhancing heat transfer coefficient can be classified into two – passive and active [6-7]. Passive techniques do not use external power for operation, while active techniques use external power for operation. In both schemes, the enhancement is achieved by keeping the boundary layer thickness small. The disruption of the boundary layer causes the fluid to have transverse velocities (secondary flow) over a greater portion of the cross-sectional area than otherwise possible thereby leading to greater uniformity of temperature over the same cross-section as well as lowering of the bulk coolant temperature. Figure 3 shows the different approaches of passive and active techniques. One of the passive schemes has been to reduce the cross-sectional dimensions of the flow passage, which has led to the realization of microchannels. Microchannels are flow passages with a hydraulic diameter smaller than 1 mm [8]. Disruption of the boundary layer can be achieved in several ways including passive approaches such as integrating obstructions in the fluid flow path, altering fluid flow direction, increasing surface roughness, etc. and active approaches such as vibration of the device, flow pulsation, etc.

Figure 3: Different enhancement techniques for MCHS (passive and active) [9]
Thermal management systems can either use air or liquid as the working fluid. With the increase in heat flux associated with microelectronic chips, thermal management techniques that employ liquid as the working fluids are being increasingly used/considered/proposed. Thermophysical properties, such as density, specific heat capacity, and thermal conductivity of liquids are superior to that of air, making them better coolants for thermal management. Higher density implies that a greater mass of the coolant can be contained in the same volume, while higher specific heat capacity implies lower temperature rise and with increase in thermal conductivity, it is possible to have higher heat transfer coefficient. Heat sink consists of multiple flow passages through which the coolant passes [10]. Heat sinks are attached on top of a microelectronic chip; heat generated by the underlying microelectronic chip moves through the heat sink structure by conduction towards the flow passages, and at the interface, the heat transfers from the structure to the coolant by convection. The main desired characteristic of heat sinks is low thermal resistance which in turn leads to maintaining the temperature of the electronic chip low. The thermal resistance of heat sinks has three components - conductive thermal resistance, convective thermal resistance, and calorific thermal resistance [11]. The heat transfer through the substrate of the heat sink is by conduction and the resistance associated with this heat transfer path is the conduction thermal resistance. The heat transfer from the heat sink substrate to the coolant is by convection and the thermal resistance connected with this heat transfer path is convection thermal resistance. The thermal resistance associated with the temperature rise of the fluid is called calorific thermal resistance. Moreover, these three thermal resistances exit in series for the heat sink. The desire to create heat sinks with low thermal resistance has led to the employment of microchannels [11]. Heat sinks employing microchannels are constructed in thin substrates and this keeps the
conductive thermal resistance low. The heat transfer coefficient associated with microchannels is very high, and this has led to employing microchannels in heat sinks for realizing the same with low convective thermal resistance. Among the two thermal resistances, i.e. conduction and convection thermal resistance, the convection thermal resistance is higher and thus there is a drive for creating newer design of microchannels for enhancing the heat transfer coefficient in the same.

1.2 Statement of the Problem

This work proposes a new design of the microchannels employed in heat sinks with the sole purpose of reducing the associated thermal resistance. To the best of the author's knowledge, this is the first work to model a heat sink employing wavy microchannels embedded with pin-fins; there are two possibilities regarding the placement of the pin fins and both are considered in this work. The first possibility is the placement of a pin fin at every peak and valley as shown in Figure 4 (a) while the second possibility, shown in Figure 4 (b), places a pin fin between neighboring peaks and valleys. The microchannel has a sinusoidal path, leading to continuous disruption and redevelopment of the boundary layer. Additionally, pin-fins are present within the microchannel, as shown in Figure 4. It is expected that the presence of pin-fins aggravates the disruption of the boundary layers, thereby enhancing the heat transfer coefficient beyond that achievable simply by employing a sinusoidal microchannel. The employment of this particular flow passage in heat sinks will help keep the heat source's temperature below that achievable with straight microchannels, and this is one of the benefits of the proposed heat sink design. A model of both types of the proposed heat sink is developed and subsequently used for parametric study. The influence of geometric parameters on the working of the devices is analyzed in this work. The work
will provide detailed information on the fluid flow field inside the same. The modeling will also provide information about the effect of pin fin on the enhancing of heat transfer coefficient inside the microchannel. Information related to the flow and temperature field inside this microchannel heat sink embedded with pin fins is currently unavailable and will be generated from this study.

Figure 4: Wavy microchannel heat sink embedded with pin fins. (a) pin fins at peaks and valleys, (b) pin fins in the middle between peaks and valleys

1.3 Relevant Literature

At the beginning of the 1980s and for the first time, Tuckerman and Pease (1981 and 1983), introduced their work on Microchannel Heat Sinks (MCHS) [12-13]. The idea was to develop a heat sink to decrease the thickness of the thermal boundary layer; this helped in decreasing the convective resistance [14]. One more advantage of the microchannel heat sinks is the increase of the surface area density; thus, high cooling rates can be achieved for the same maximum temperature of the electronic chip.

After the development of the microchannel heat sink by Tuckerman and Pease (1981), there has been considerable interest in advancing the enhancement of heat transfer in the microchannel [12]. The traditional design arrangement of the MCHS is
the square/rectangular straight channels; in this design, the heat sink consists of multiple microchannels with a rectangular cross-section; the dimensions of the cross-section are constant. Mathew et al. (2011) conducted experimental studies on thermal management of electronic chips using a combination of a thermoelectric cooler and a microchannel heat sink [15]. The thermoelectric cooler is sandwiched between the electronic chip and heat sink. They observed that by operating the thermoelectric cooler, it is possible to keep the electronic chip's temperature below the coolant's exit temperature, which in their experiments was water. They conducted studies for different power inputs to thermoelectric coolers and electronic chips. Ali et al. (2021) conceptualized and subsequently mathematically modeled a heat sink employing convergent-divergent microchannel for purposes of thermal management of a concentrated photovoltaic module [16]. They investigated the thermohydraulic performance of the proposed heat sink under parallel (flow in all channels are in the same direction) and counterflow (flow in adjacent channels are in the opposite direction) arrangements. They observed that counterflow arrangement had better thermal performance than parallel flow arrangement, but it experienced higher pumping power. Wang et al. (2016) did a numerical simulation to investigate the influence of the geometry of the shape of the microchannel heat sink on the performance of the heat sink; they investigated rectangular, trapezoidal, and triangular-shaped microchannel heat sinks [17]. The rectangular heat sinks showed better performance in most scenarios. Lu et al. (2021) proposed rectangle-shaped protrusions on the microchannel sidewall of a heat sink for vortex generation purposes [18]. The presence of vortices leads to enhanced mixing of the fluid, thereby reducing the microelectronic chip's temperature in the vicinity of the protrusions. They considered heat sinks with different orientations for the protrusions and quantified their
performance in terms of thermal resistance and pumping power. Among all the orientations, the protrusion that pushed the flow downwards to the bottom wall of the microchannel showed the best thermal performance. They also found that the pumping power to be independent of the orientation of the protrusions.

Song et al. (2021) conceptualized two heat sinks employing microchannels with cross-sectional areas varying in the axial direction and tested the efficacy of each while operating it in several configurations [19]. In one design, the cross-sectional area of two adjacent microchannels decreased in the same direction, while in the second design, the cross-sectional area of two adjacent microchannels decreased in the opposing directions. Each configuration differed from the other in terms of the direction of flow of the fluid in adjacent microchannels. They conducted numerical studies on all configurations and quantified the performance in terms of thermal resistance and pressure drop. They found that the heat sinks' thermal performance with the flow in opposing directions is better than the heat sinks with the flow in the same direction. In another study, Mathew et al. (2010) investigated the zigzag configuration and the effect of the number of turns on both the Nusselt number and Poiseuille number, which resulted in enhanced performance compared to the straight channel [20].

Researchers investigated multiple possibilities; one of these ideas was to change the cross-section (i.e., non-uniform channel). This idea was tackled in Oevelen et al. (2010) work on the axial non-uniform width microchannels, where they found that a 7.8% reduction in thermal resistance was achieved [21]. Xie et al. (2015) discussed the idea of Y-shaped bifurcation plates within a straight microchannel, Figure 5, which showed thermal performance improvement compared to the traditional one [22]. Wang et al. (2019) studied the effect of adding bidirectional ribs to the
microchannel, which helps to interpret the thermal boundary layer, they realized that the heat transfer has improved, but the cost was higher pressure drop [23].

Soleymani et al. (2020) conceptualized a microchannel heat sink for thermal management of microelectronic chips, including hot spots [24]. The heat sink employed a two-dimensional array of pin fins in the middle and microchannels elsewhere; the microchannels transport liquid in and out of the pin fin array. They considered different shapes of pin fins as well as both straight and wavy microchannels in the proposed heat sink. They found that rectangular pin fins with rounded edges provided better thermal performance than airfoil-shaped pin fins as well as that with an increase in waviness of the channel, the thermal performance improved. John et al. (2010) conceptualized a heat sink employing a straight microchannel embedded with pin-fins and subsequently modeled the same [25]. They studied the effect of different pin-fin shapes as well. From the study, they identified that embedding pin-fins enhanced the thermohydraulic performance of the heat sink. Figure 6 shows that adding pin fins reduced the overall thermal resistance at the same Reynolds number.
Additionally, they identified that the shape of the pin-fin had a negligible effect on the thermohydraulic performance of the heat sink.

![Graph showing thermal resistance comparison]

**Figure 6**: Thermal resistance of traditional straight microchannel vs. straight microchannel embedded with pin fins [25]

Ansari and Kim (2018) conceptualized a heat sink that both employed straight microchannels and circular pin fins, and they called it a hybrid heat sink [26]. The pin fins were employed above the hot spot section of the electronic chip, and the straight microchannels were employed elsewhere; in their design, the hot spot section was assumed to be at the center of the electronic chip. They found that the hybrid heat sink had a better thermal performance than heat sinks with just microchannels with a minimal penalty of increased pumping power. Hotchandani et al. (2021) carried out numerical studies on a heat sink with straight microchannels embedded with circular pin-fins [27]. The heat sink's thermal resistance employing straight microchannels embedded with pin-fins is smaller than that of heat sinks employing just straight microchannels. However, the reduction in thermal resistance came at the expense of additional pumping power. Moreover, they conducted a parametric study to understand
the influence of geometric parameters on the performance of heat sinks with straight microchannel embedded with pin-fins.

Qiu et al. (2020) investigated the effect of pin fins on the performance of the microchannel heat sink experimentally [28]. The study was divided into three parts, traditional, in-line pin fins, and staggered pin fins, their results showed that the thermal performance represented by the Nusselt number was enhanced for the microchannel heat sink with pins, and in the case of staggered pins, the performance was better than the in-line, this improvement was observed for Re higher than 300 since at lower Re the disturbance caused by pin fins is weakened.

Geometrical parameters such as the cross-section and the pattern can play a main role in optimizing the performance of the MCHS. These parameters have a crucial effect on the temperature, heat transfer coefficient, and pressure drop; hence a lot of designs were proposed and investigated. Figure 7 shows different approaches to studying the cross-section and pattern shapes. Another configuration is the wavy channel which has many advantages, such as enhancing convective fluid mixing compared to a straight channel with the same cross-section, which means a better heat transfer performance [29-30]. Lin et al. (2017) numerically investigated the influence of varying wavelength and amplitude of wavy (sinusoidal) microchannels on the thermohydraulic performance of heat sinks employing the same [31]. They observed that wavy microchannels with axially varying wavelength and frequency had reduced thermal resistance than wavy microchannels in which the wavelength and frequency remain constant in the axial direction.
Sui et al. (2010) numerically studied wavy channels, which showed a superior heat transfer performance; this can be explained by the mixing existing in wavy microchannels compared to the straight design [29]. Mohammed et al. (2011) studied the same and found similar results; they found that changing the amplitude and wavelength has a major effect on the performance and can help to enhance the heat transfer process [34]. Sui et al. (2011) investigated experimentally and confirmed the numerically results [35].

Tiwari et al. (2019) studied different shapes of microchannel heat sinks, one of these was a wavy configuration, which showed good performance compared to straight microchannels [30]. Figure 8 shows the difference between the temperature contours, which clearly indicates that the wavy design achieved lower temperature compared to the straight. Sakanova et al. (2015) studied the effect of the amplitude, the wavelength of a wavy microchannel heat sink, on the performance of the heat sink, they found that the performance has improved but at the cost of increased pressure drop compared to straight channels design (Figure 9) [36].
This work will investigate the performance of wavy microchannel heat sinks embedded with pins. The pin fins are expected to produce extra flow field disruption within the microchannel resulting in a better thermal performance.

Figure 8: Temperature contours of straight vs. wavy microchannel heat sinks [30]

Figure 9: Straight microchannel heat sink vs. wavy microchannel - different amplitudes and wavelength - heat sinks in terms of thermal resistance [36]
1.4 Structure of the Thesis

Chapter 1: The first chapter of the thesis introduces and discusses the problem statement followed by a review of relevant literature, which builds the background of this work, and finally, the structure of the thesis is detailed.

Chapter 2: The second chapter discusses the mathematical model of the proposed microchannel heatsinks, including assumptions and the governing equations, as well as approach of determining the solution of the model. This chapter also details the experimental setup to be employed for validating the simulation results.

Chapter 3: This chapter covers the results of the simulations of each of the cases as well as the post-processing analysis. This chapter also discusses the experimental setup and its results compared to the results of the CFD model.

Chapter 4: The last chapter concludes the thesis by listing the important findings of this work and discusses the possibilities of future work.
Chapter 2: Methodology

This chapter covers the methodology, mathematical modeling and experimentations employed for studying the problem statement mentioned in the previous chapter. There are many factors or specifications that need to be taken into consideration when designing a heat sink such as physical features and geometric parameters. This work is done using water (liquid) as the working fluid due to its superior thermophysical properties (density, specific heat capacity, and thermal conductivity) compared with that of air which makes it the best coolant, among the two, for thermal management.

2.1 Geometry

The heat sink’s geometry can be divided into three parts, heat sink as a whole, a repeating unit, and pin fins. Figure 10 shows the schematic of the proposed heat sinks. The microchannel has a sinusoidal path, leading to continuous disruption and redevelopment of the boundary layer. Additionally, pin-fins are present within the microchannel, as shown previously in Figure 4. In Figure 4 (a) the pin fins are located at each peak and valley of the sinusoidal microchannels while Figure 4 (b) shows the heat sink in which pin fins are located between the peaks and valleys of the microchannel. The presence of pin-fins, in both cases, is expected to enhance the disruption of the boundary layers, thereby enhancing the heat transfer coefficient than that possible in heat sinks with just sinusoidal microchannels. All heat sinks consist of similar multiple microchannels, which allows for identifying a repeating unit (Figure 10). For purposes of this study, only the repeating unit is analyzed. The structure of the repeating unit, in both cases of the location of pin fins, can be described using the
mathematical equation $A \sin(Bx)$ where $A$ is the amplitude and $B$ is the frequency, these will be discussed later in this section.

Figure 10: Schematic of the heat sink employing sinusoidal microchannel embedded with pin fins, repeating unit

Figure 11: Repeating unit - geometric properties

The footprint of the heat sink is 1-inch by 1 inch, but as mentioned earlier, it is studied based on a repeating unit, as shown in Figure 11. The repeating unit's geometric characteristics consist of the amplitude and wavelength of the microchannel, the hydraulic diameter, the length, and finally, the depth. The microchannels studied in
this work have square cross-section. Some of these parameters are fixed and will be the same throughout the study, and the rest will vary based on a parametric study. Table 1 below summarizes these parameters. The parametric study will include a variety of combinations of the dimensions mentioned in Table 1. It is clear that the repeating unit's depth and width are fixed, and the only variables are the amplitude, wavelength, and hydraulic diameter. The effect of amplitude (A) and wavelength (B) represented by the equation \( A \sin(Bx) \) is shown in Figure 12. Figure 13 shows the effect of (A) and (B) as functions.

Table 1: Repeating unit - geometric properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>100 µm, 200 µm, 300 µm</td>
</tr>
<tr>
<td>B</td>
<td>1 µm(^{-1}) ((\lambda=6283.2) µm), 2 µm(^{-1}) ((\lambda=3141.6) µm), 3 µm(^{-1}) ((\lambda=2094.4) µm)</td>
</tr>
<tr>
<td>C</td>
<td>150 µm, 200 µm, 250 µm</td>
</tr>
<tr>
<td>D</td>
<td>500 µm</td>
</tr>
<tr>
<td>W</td>
<td>300 µm</td>
</tr>
<tr>
<td>Pin fin diameter</td>
<td>75 µm, 100 µm, 125 µm</td>
</tr>
<tr>
<td>Pin fin location</td>
<td>At peaks and valleys, in the middle between peaks and valleys</td>
</tr>
</tbody>
</table>
Figure 12: The effect of amplitude and wavelength on the shape of the microchannel. (a) amplitude, (b) wavelength

Figure 13: The effect of amplitude, and wavelength, on a sinusoidal function
Now the repeating unit's geometric parameters are defined. The other crucial part of this work is the pin fins. Figure 14 shows typical wavy microchannels embedded with pin fins. The values of these pin fins’ diameter are (75 µm, 100 µm, and 125 µm); they are shown in Figure 15. These diameters will be part of the parametric study, along with the other geometric parameters mentioned in Table 1.

![Figure 14: Wavy microchannel embedded with pin fins](image1)

![Figure 15: Wavy microchannel embedded with pin fins - diameter of pin fins](image2)
2.2 CFD Analysis

The CFD analysis employed to analyze the repeating unit consists of three main elements. The first element is the pre-processor, followed by the solver, and finally postprocessor. Figure 16 shows the main elements of any CFD Analysis. This work is carried out using Fluent module of Ansys Workbench. The geometry is discussed in the previous section. Mesh generation (mesh independence study), the convergence criteria, and the solution analysis will be presented of Chapter 3. This section will cover the material properties, boundary conditions, governing equations, and the solution of the governing equations.

Figure 16: CFD analysis framework, main elements [37]
2.2.1 Material Properties

The heat sink substrate is made of silicon, whereas the working fluid, as mentioned earlier, is water. One of the advantages of using silicon is the high quality and low cost of the final product; also, silicon is a good heat conductor. Another advantage of using silicon is that it is directly integrable into the semiconductor infrastructure [38]. Table 2 list the properties of silicon and water used in the study.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Fluid (Water at 20°C)</th>
<th>Substrate (Silicon)</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho$</td>
<td>1000</td>
<td>2329</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>$\mu$</td>
<td>0.001006</td>
<td>-</td>
<td>Pa.s</td>
</tr>
<tr>
<td>$C_p$</td>
<td>4181</td>
<td>700</td>
<td>J/kg.K</td>
</tr>
<tr>
<td>$k$</td>
<td>0.597</td>
<td>148</td>
<td>W/m.K</td>
</tr>
</tbody>
</table>

2.2.2 Governing Equations and Model Assumptions

The mathematical model consists of several equations. The mathematical model of the substrate of the heat sink consists of the energy equation, while the mathematical model of the fluid consists of the continuity equation, Navier-Stokes equations, and energy equation. The energy equations are provided in Equations (2.1) and (2.2), while the continuity equation and Navier-Stokes equations are provided in Equations (2.3) and (2.4 a,b,c), respectively [39]. Equation (2.4 a) represents the
components of Navier-Stokes equations in the x-direction. Equations (2.4 b) and (2.4
c) represent the components of the same but in y and z directions, respectively.

\[
\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} = 0
\]  
(2.1)

\[
u_f \frac{\partial T_f}{\partial x} + v_f \frac{\partial T_f}{\partial y} + w_f \frac{\partial T_f}{\partial z} = \frac{k_f}{\rho_f c_{p,f}} \left( \frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial z^2} \right)
\]  
(2.2)

\[
u_f \frac{\partial u_f}{\partial x} + v_f \frac{\partial u_f}{\partial y} + w_f \frac{\partial u_f}{\partial z} = -\frac{1}{\rho_f} \frac{\partial P}{\partial x} + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 u_f}{\partial x^2} + \frac{\partial^2 u_f}{\partial y^2} + \frac{\partial^2 u_f}{\partial z^2} \right)
\]  
(2.3a)

\[
u_f \frac{\partial v_f}{\partial x} + v_f \frac{\partial v_f}{\partial y} + w_f \frac{\partial v_f}{\partial z} = -\frac{1}{\rho_f} \frac{\partial P}{\partial y} + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 v_f}{\partial x^2} + \frac{\partial^2 v_f}{\partial y^2} + \frac{\partial^2 v_f}{\partial z^2} \right)
\]  
(2.3b)

\[
u_f \frac{\partial w_f}{\partial x} + v_f \frac{\partial w_f}{\partial y} + w_f \frac{\partial w_f}{\partial z} = -\frac{1}{\rho_f} \frac{\partial P}{\partial z} + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 w_f}{\partial x^2} + \frac{\partial^2 w_f}{\partial y^2} + \frac{\partial^2 w_f}{\partial z^2} \right)
\]  
(2.3c)

The mathematical model is based on several assumptions, which are listed below:

1. The heat sink is operating under steady-state conditions and under continuum
   regime.
2. Fluid flow in the heat sink is laminar.
3. Constant heat flux at the bottom of the substrate.
4. The fluid is incompressible and does not undergo phase change while in the
   heat sink.
5. No heat transfer exists between the heat sink and its surroundings, no viscous
   dissipation occurs inside the microchannel, and no flow maldistribution exits
   in the heat sink.
For purposes of solving the governing equations, the computational domain, i.e., repeating unit, Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm (Figure 17), is used for solving the Navier-Stokes equations for determining the field variables such as velocity, temperature, and pressure.

![SIMPLE algorithm diagram](image)

**Figure 17: SIMPLE algorithm [40]**

Subsequently, the energy equations are solved for determining the temperature profile in the substrate and microchannel (fluid domain). SIMPLE algorithm is
considered a robust method to solve the governing equations; it is basically an iterative process based on the guess-and-correct procedure.

2.2.3 Boundary Conditions

The boundary conditions associated with the fluid domain of the repeating unit include known velocities, temperature, pressure, and temperature gradient, as well as continuity of temperature and heat. The velocity at the inlet in the x-direction is determined from the Reynolds number, as shown in Equation (2.5 a), and studies are done for the Reynolds number range between 250 and 2000. The velocities in the other two directions at the inlet are zero, as shown in Equation (2.5 b). The temperature at the inlet of the fluid domain is known as indicated in Equation (2.6). The gauge pressure at the exit of the microchannel is held at zero, as shown in Equation (2.7).

Additionally, it is assumed that there is no heat transfer in the axial direction after the outlet cross-section of the microchannel, as shown in Equation (2.8). At the interfaces, there is continuity of temperature and heat, and they represent boundary conditions as well, like shown in Equation (2.9) and Equation (2.10). The velocities on all interfaces are zero as per the no-slip condition, and this is shown in Equation (2.11). The microelectronic chip is kept in thermal contact with the bottom surface of the heat sink, due to which the thermal boundary condition on the bottom surface of the repeating unit is the constant heat flux which is equivalent to that of the electronic chip; studies are done in this work for the heat flux of 1,000,000 W/m² as shown in Equation (2.12), although 1,000 kW/m² seems to be a high value, similar or larger heat flux values were investigated and tested in previous publications [41-42].

All remaining surfaces of the substrate are thermally insulated, and they represent the thermal boundary conditions of the substrate as shown in Equation
(2.13); n represents the normal associated with the surfaces of the substrate. Thermal resistance is calculated using Equation (2.14), while the pumping power is calculated using equation eq. (2.15). The component of thermal resistance (convective, and calorific) are shown in equations (2.17) and (2.18) respectively [39,43-44]. Equation (2.19) represents the average Nusselt number. Figure 18 shows a schematic of the microchannel with the boundary conditions.

\[
\begin{align*}
    u_{in} &= Re \frac{\mu_m}{\rho_m D_h} \\
    v_{in} &= 0 \text{ and } w_{in} = 0 \\
    T_{in} &= 293 \text{ K} \\
    P_{out} &= 0 \\
    \left. \frac{\partial T_f}{\partial x} \right|_{out} &= 0 \\
    T_{f,interface} &= T_{s,interface} \\
    k_f \left. \frac{\partial T_f}{\partial n} \right|_{interface} &= k_s \left. \frac{\partial T_s}{\partial n} \right|_{interface} \\
    V_{interface} &= 0
\end{align*}
\]
\[ q''_{\text{bottom}} = 1,000,000 \frac{W}{m^2} \]  

(2.12)

\[ \frac{\partial T_s}{\partial n} \bigg|_{\text{boundaries}} = 0 \]  

(2.13)

\[ R_{th} = \frac{T_{\text{chip,avg,max}} - T_{f,\text{avg,in}}}{q''} \]  

(2.14)

\[ PP = \Delta P \dot{V} \]  

(2.15)

\[ R_{th,\text{conv}} = \frac{\text{LMTD}}{q''} \]  

(2.16)

\[ R_{th,\text{conv}} = \frac{(T_{w,\text{avg,out}} - T_{f,\text{avg,out}}) - (T_{w,\text{avg,in}} - T_{f,\text{avg,in}})}{\ln\left(\frac{T_{w,\text{avg,out}} - T_{f,\text{avg,out}}}{T_{w,\text{avg,in}} - T_{f,\text{avg,in}}}\right)} \frac{1}{q''} \]  

(2.17)

\[ R_{th,\text{cal}} = \frac{T_{f,\text{avg,out}} - T_{f,\text{avg,in}}}{q''} \]  

(2.18)

\[ \bar{Nu} = \frac{\bar{h}D_{hy}}{k} \]  

(2.19)
2.3 Experimental Setup

A schematic of the experimental setup for the proposed work is shown in Figure 19 below. It can be noticed that the experimental setup is an open-loop and consists of a tank, filter, pump, and test-fixture. Water is used as the coolant. Water from the tank will initially pass through the pump and filter before entering the microchannel heat sink. The microchannel heat sink is held in the test-fixture. The coolant exiting the test-fixture is collected in a second tank. In this way, the temperature of water in the first tank remains unaffected.

![Figure 19: Schematic of the experimental setup](image-url)
The temperature of the coolant at the inlet and outlet sections of the wavy microchannel heat sink are measured. Thermocouples are used for measuring the temperatures of the coolant. The heat generated by the electronic chip is simulated using a 1 inch by 1 inch thin-film heater attached to the backside of the wavy microchannel heat sink (Figure 20). The temperature of the thin-film heater at the exit section of the wavy microchannel heat sink is measured using a thermocouple probe (1.5 mm diameter). The temperatures and heat input (from the heater) would be used for calculating the thermal resistance.

The test-fixture holds the microchannel heat sink. The purpose of the test-fixture is to prevent interaction between the microchannel heat sink and the external ambient. In this way, the heat generated by the heater (thin-film heater) will end up in the microchannel heat sink rather than the ambient.

Figure 20: Schematic of the heat sink, and heater attached at the bottom surface
Chapter 3: Results and Discussion

In this chapter, the simulation and experimental results are reported and discussed. The first section will detail the results of the modeling of the geometry introduced in Chapter 2 in addition to a brief on the solution convergence and the mesh independence study. Figure 21 summarizes CFD solution procedure. The second part will discuss the results of CFD simulations covering the different cases mentioned in Chapter 2, in addition to two other cases of the straight microchannel and straight microchannel embedded with pin fins; these two parts were added for the purpose of comparison with the wavy microchannel performance. The performance is evaluated based on different criteria, thermal resistance, pressure drop, chip maximum temperature, and Nusselt number. This section will also show a visual comparison in terms of contour plots of different results and cases to make the comparison more comprehensible. The last section will discuss the experimental work and results, covering few selected cases for validation.

Figure 21: CFD solution procedure [37]
3.1 Geometry Meshing and Convergence

After creating the necessary geometry for all cases they are meshed. The mesh settings are applied based on the design's complexity and curvature to create a uniform domain. The mesh types include either tetrahedral, hexahedral, or quadrilateral. Figure 22 shows few cases shows repeating units near the outlet with and without pin fins based on the studied case, also a cross-section through a pin fin. A mesh independence study is applied for all cases using at least three different mesh settings from coarse to fine, showing a decrease in the difference in the results towards a difference of less than 1.0%.

Figure 22: Schematic of the meshed computational domains of different cases showing different views of repeating units. (a) straight (b) straight embedded with pin fins (c) wavy (d) wavy embedded with pin fins
The importance of mesh independence study is not only to ensure that the correct solution is calculated, but also to find the balance between the mesh settings and the computational time. The number of nodes of every mesh setting is set to be at least 1.3 times the previous one; on average, most of the cases were around two times. Table 3 shows few selected cases showing the number of nodes, outlet temperature, chip maximum temperature, inlet pressure, and their respective differences. It also shows that the differences decrease to less than 1.0% difference with increasing the number of nodes.

The convergence of the solution and results is satisfied by observing and monitoring the main output results such as outlet temperature, chip maximum temperature, inlet pressure, and the balance between input and output mass flow rate, in addition to residuals values. Figure 23 shows one case where all the previously mentioned results reached a steady solution. The same is applied for all other cases with a range of ±1.0% variation in case of a small oscillation in the steady solution.
Figure 23: Convergence and steady solution, wavy with pin fins 0.2sin(2x), $D_{hy} = 150 \, \mu m$, pin fin diameter = 100 $\mu m$ at $Re = 1000$
Table 3: Mesh settings for simulation of straight, straight with pin fins, wavy, and wavy with pin fins microchannels and their associated parameters

<table>
<thead>
<tr>
<th></th>
<th>Straight – (D&lt;sub&gt;hy&lt;/sub&gt; = 150 µm).</th>
<th>Straight with Pin Fins – (D&lt;sub&gt;hy&lt;/sub&gt; = 150 µm, Pitch = 2500 µm, Pin Fin Diameter = 125 µm).</th>
<th>Wavy – (A = 200 µm, B = 1, D&lt;sub&gt;hy&lt;/sub&gt; = 150 µm).</th>
<th>Wavy with Pin Fins – (A = 200 µm, B = 2, D&lt;sub&gt;hy&lt;/sub&gt; = 150 µm, Pin Fin Diameter = 100 µm).</th>
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<td>309.103</td>
<td>309.119</td>
<td>309.104</td>
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<td>-</td>
<td>0.0</td>
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<tr>
<td>T&lt;sub&gt;chip&lt;/sub&gt;,avg,max (K)</td>
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<td>350.513</td>
<td>350.528</td>
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<tr>
<td>Difference (%)</td>
<td>-0.4</td>
<td>0.0</td>
<td>-</td>
<td>0.2</td>
</tr>
<tr>
<td>ΔP (kPa)</td>
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<td>177.287</td>
<td>177.29</td>
<td>228.002</td>
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<td>-</td>
<td>0.9</td>
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<td>309.103</td>
<td>309.103</td>
<td>309.104</td>
</tr>
<tr>
<td>Difference (%)</td>
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<td>0.0</td>
<td>-</td>
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<td>-</td>
<td>0.1</td>
</tr>
<tr>
<td>ΔP (kPa)</td>
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<td>226.026</td>
<td>225.1356</td>
<td>0.4</td>
</tr>
<tr>
<td>Difference (%)</td>
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<td>-</td>
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</table>
3.2 CFD Simulation Results and Visualization

This section will cover the computational results of the mathematical model of the different microchannel heat sinks. In addition to the wavy design introduced previously in this work, it is necessary to test the performance of the traditional straight (with and without pin fins) microchannel heat sinks. This will allow for a better evaluation of the performance of the different wavy designs. Figure 24 shows the results of selected cases of all configurations.

Figure 24: Variation of thermal resistance and pressure drop with Reynolds number for straight ($D_{hy} = 150 \mu m$), straight embedded with pin fins ($D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$, pin fin pitch = 2500 $\mu m$), wavy ($0.3\sin(x)$, $D_{hy} = 150 \mu m$), and wavy embedded with pin fins ($0.3\sin(x)$, $D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$) microchannels

The main observations from Figure 24 are that the thermal resistance is decreasing with increase in Reynolds number for all cases. For the same Reynolds number, it is shown that the wavy microchannel embedded with pin fins has the lowest thermal resistance while the straight microchannel has the highest thermal resistance.
These observations can be explained through Figure 25, which shows the convective and calorific thermal resistances values for the same. The high pressure drop in the microchannels embedded with pin fins is mainly because of the reduction of the area between the walls of the microchannel and the introduced pin fins and this subsequently leads to increase in the velocity which require more power to push the fluid through.

![Figure 25](image)

**Figure 25**: Variation of calorific and convective thermal resistance with Reynolds number for straight ($D_{hy} = 150 \, \mu m$), straight embedded with pin fins ($D_{hy} = 150 \, \mu m$, pin fins diameter = 75 $\mu m$, pin fin pitch = 2500 $\mu m$), wavy ($0.3\sin(x)$, $D_{hy} = 150 \, \mu m$), and wavy embedded with pin fins ($0.3\sin(x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter = 75 $\mu m$) microchannels

Figure 25 shows that both convective and calorific thermal resistances decrease with an increase in Reynolds number. The reduction in the calorific thermal resistance with Reynolds number is due to the increase in the mass flow rate, which leads to a further reduction in the fluid’s outlet temperature and. It is also shown that the calorific thermal resistance for all configurations at the same Reynolds number is the same since
the mass flow rate depends on the Reynolds number. On the other hand, the reduction in the convective thermal resistance is mainly due to the reduction in the average thickness of the boundary layer. In the case of the straight microchannel, increasing the Reynolds number increases the entrance length, which results in a reduction in the average thickness of the boundary layer. The same goes with the straight microchannel embedded with pin fins, but in addition to the increase in the entrance length, the presence of the pin fins introduces flow obstructions which help reduce the average boundary layer thickness even more and keep it smaller than without pin fins.

The wavy microchannel has an advantage over the straight microchannel in terms of flow path. The continuous change in flow direction leads to repeated disruption of the boundary layer, thereby keeping small the boundary layer thickness, and this boundary layer thickness gets reduced with an increase in Reynolds number. Similar to the straight microchannel embedded with pin fins, introducing pin fins to the wavy geometry causes disturbance in the flow, which keeps the average boundary layer thickness small. The performance of these designs can also be compared in terms of Nusselt number, as shown in Figure 26. It shows that the Nusselt number of the wavy microchannel embedded with pin fins is the highest while the straight microchannel has the lowest; these results support the findings shown in Figure 25.

The effect of both wavy structure and pin fins on the thermal performance of the microchannel can be investigated further using a close-up near either a peak or valley (with and without pin fin) to understand the effect of both factors on the hydrodynamic boundary layer. Figures 27 and 28 show the effect of the wavy geometry and pin fins, respectively, using cross-sections in different locations along the microchannel. In Figure 27, three cross-sections are used to understand the effect of the wavy geometry on the flow direction within the microchannel.
Figure 27 (a) shows the in-plane average velocity (YZ plane) somewhere before a valley of a wavy microchannel without pin fins. These transverse components of the velocity are responsible for the enhanced thermal performance in wavy microchannels than straight microchannels. As the fluid goes forward, it starts to change direction due to the wavy structure, as shown in Figure 27 (b), which illustrates how the flow starts to change direction from one side to another. After it leaves the peak or valley, the fluid starts to adjust itself again, as shown in Figure 27 (c), until it reaches the next peak or valley. As shown in Figure 27 (a, b, c), this repetitive process of changing direction is the main reason for the enhanced thermal performance compared to the traditional straight microchannel.

The disruption of the boundary layer caused by the continuous change in direction as discussed in Figure 27 (a, b, c) can be further enhanced by adding pin fins as shown numerically in Figures 24 and 25. Four cross-sections, two before and two after a pin fin, are shown in Figure 28 (a, b, c, d). Figure 28 (a) shows almost similar behavior compared to Figure 27 (a). The transverse velocity components are going from one side to another due to the microchannel geometry. The in-plane average velocity just before the pin fin located at the valley of the microchannel is shown in Figure 28 (b). Also, similar behavior is noticed compared to Figure 27 (b) as the fluid starts to change direction in the (YZ plane). The difference here is that the fluid is about to hit a pin fin which is supposed to cause further disruption.

The effect of the pin fin is illustrated in Figure 28 (c), which clearly shows a significant disruption in the velocity vectors compared to the effect of the wavy geometry as discussed in Figure 28 (a) and (b). This combined effect of changing in direction due to the wavy geometry and the significant disruption due to the pin fins results in better thermal performance, as discussed in Figure 24. The fluid then starts
to adjust itself again as it travels away from the pin fins, as shown in Figure 28 (d). An estimation of the hydrodynamic boundary layer (average velocity vectors in the flow direction) for both cases (wavy with and without pin fins) is shown in Figure 27 (d, e, f) and Figure 28 (e, f, g, h). Both figures show that the hydrodynamic boundary layers are continuously developing but never fully developed, thus a greater disruption to the thermal boundary layer, hence better thermal performance.

Figure 26: Variation of Nusselt number with Reynolds number for straight ($D_{hy} = 150 \mu m$), straight embedded with pin fins ($D_{hy} = 150 \mu m$, pin fins diameter = 75 µm, pin fin pitch = 2500 µm), wavy ($0.3\sin(x)$, $D_{hy} = 150 \mu m$), and wavy embedded with pin fins ($0.3\sin(x)$, $D_{hy} = 150 \mu m$, pin fins diameter = 75 µm) microchannels
Figure 27: The effect of the wavy structure of the microchannel on the velocity profile of the flow, $0.2\sin(x)$, $D_{hy} = 150 \, \mu m$, Re = 1000
Figure 28: The effect of adding pin fins on the velocity profile of the flow, $0.2\sin(x)$, $D_{hy} = 150$ µm, pin fins diameter = 75 µm, $Re = 1000$.
Figure 29 shows the pressure drop in the axial direction within a wavy microchannel with and without pin fins. The vertical lines represent how the pressure varies in the vicinity of the pin fins. The drop in pressure is due to the obstruction of flow by the pin fins (due to the stagnation point and the reduction in area between the pin fins and channel walls) and the following increase in pressure is due to pressure recovery associated with increase in area. Figure 30 shows velocity and temperature contours of the fluid at the outlet section of the microchannel, which shows improvement in the temperature distribution of wavy with pin fins ($T_{SD\_out} = 2.37$ K for wavy microchannel embedded with pin fins compared to $T_{SD\_out} = 15.12$ K for straight), while Figure 31 shows streamline at the midplane of the fluid for all cases near pin fins. Overall, the wavy microchannel embedded with pin fins show promising performance in terms of thermal resistance. Another important observation is that although the straight microchannel embedded with pin fins show good performance compared to the straight microchannel, the wavy microchannel showed even more reduction in the thermal resistance for the same Reynolds number as well as less pressure drop than the former.

The performance of the previously mentioned configurations can also be evaluated based on the chip's maximum temperature, which is crucial for the performance and safety of the chip or the device. Figure 32 shows the results for the previously discussed designs, demonstrating the superiority of wavy microchannel embedded with pin fins.
Figure 29: Pin fins influence on pressure drop of wavy microchannel embedded with pin fins, (D_{hy} = 150 \, \mu m, \text{pin fins diameter} = 75 \, \mu m)

Figure 30: Velocity and temperature contours at fluid's outlet surface for straight (D_{hy} = 150 \, \mu m), straight embedded with pin fins (D_{hy} = 150 \, \mu m, \text{pin fins diameter} = 75 \, \mu m, \text{pin fin pitch} = 2500 \, \mu m), wavy (0.3sin(x), D_{hy} = 150 \, \mu m), and wavy embedded with pin fins (0.3sin(x), D_{hy} = 150 \, \mu m, \text{pin fins diameter} = 75 \, \mu m) microchannels , Re = 1000. (a) temperature, (b) velocity
Figure 31: Streamlines for straight ($D_{hy} = 150 \mu m$), straight embedded with pin fins ($D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$, pin fin pitch = 2500 $\mu m$), wavy ($0.3 \sin(x)$, $D_{hy} = 150 \mu m$), and wavy embedded with pin fins ($0.3 \sin(x)$, $D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$) microchannels, $Re = 1000$

Figure 32: Maximum substrate temperature vs. Reynolds number, for straight ($D_{hy} = 150 \mu m$), straight embedded with pin fins ($D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$, pin fin pitch = 2500 $\mu m$), wavy ($0.3 \sin(x)$, $D_{hy} = 150 \mu m$), and wavy embedded with pin fins ($0.3 \sin(x)$, $D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$) microchannels
3.2.1 Wavy Microchannels without Pin Fins

It is necessary to understand the effect of the wavy geometry, the amplitude and frequency of the wavy microchannel, on the overall performance before and after adding pin fins. Figures 33 and 34 show the effect of amplitude and frequency on the results of the thermal resistance and pressure drop, respectively, without pin fins. It is shown that in both cases, the thermal resistance is decreasing with an increase in Reynolds number, and this can be explained similar to the previous section by looking at the plots of the convective and calorific thermal resistance.

![Graph showing variation of thermal resistance and pumping power with Reynolds number for wavy microchannel (D_{hy} = 150 \mu m) – effect of amplitude (A)](image-url)

Figure 33: Variation of thermal resistance and pumping power with Reynolds number for wavy microchannel (D_{hy} = 150 \mu m) – effect of amplitude (A)

The calorific thermal resistance decreases with an increase in Reynolds number due to the increase in the mass flow rate (for the same hydraulic diameter). On the other hand, the convective thermal resistance decreases due to the reduction in the average thickness of the boundary layer, which is caused by the nature of the wavy geometry, leading to continuous disruption of the boundary layer within the
microchannel. The other observation is that increasing both the amplitude and frequency help decreasing the thermal resistance for the same Reynolds number. The decrease is found to be more significant with the increase in the frequency than with amplitude. This is due to the geometry aspects of both factors as shown in Figure 12, which shows that increasing the amplitude increases the curvature of the microchannel structure which helps increasing the disruption within the microchannel itself resulting in decreasing the average thickness of the boundary layer even more (Figure 35). Increasing the frequency increases the number of peaks and valleys within the same length, making the fluid changing direction significantly, causing further reduction in the average thickness of the boundary layer (Figure 36).

Figure 34: Variation of thermal resistance and pumping power with Reynolds number for wavy microchannel (Dxy = 150 µm) – effect of frequency (B)
Figure 35: Temperature contours at fluid’s outlet and velocity contours at midplane of wavy microchannel ($D_{hy} = 150 \, \mu m$) – effect of amplitude ($A$), $Re = 1000$. (a) temperature, (b) velocity

Figure 36: Temperature contours at fluid’s outlet and velocity contours at midplane of wavy microchannel ($D_{hy} = 150 \, \mu m$) – effect of frequency ($B$), $Re = 1000$. (a) temperature, (b) velocity
3.2.2 Wavy Microchannels Embedded with Pin Fins

The previous section showed that the performance of the wavy configurations is indeed affected by altering the amplitude and frequency. This section will show and discuss the results of the parametric study of the previously mentioned geometrical parameters of the wavy microchannel embedded with pin fins.

3.2.2.1 The Effect of Amplitude (A) – Pin Fins at Peaks and Valleys

Increasing the amplitude of the wavy microchannel shows an enhancement of the thermal performance of the heat sink, as can be seen from Figures 33 and 34. This enhancement is mainly due to the geometrical features, which help to reduce the average thickness of the boundary layer. Adding pin fins, as stated before, shows a further reduction in the thermal resistance of the wavy design. Figure 37 shows that the thermal resistance at the same Reynolds number decreases with an increase in the amplitude.

Figure 37: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (Dhy = 150 µm, pin fins diameter = 75 µm) – effect of amplitude (A)
The reduction in the thermal resistance at the same Reynolds number is due to two factors. One is the increase in the curvature of the microchannel structure, which increases the disruption within the microchannel. The other one is related to the decrease in the convective thermal resistance due to the existence of the pin fins which help increases the disruption of the flow. Figure 38 shows a comparison between one case with embedded with pin fins and another without, the other parameters are fixed as stated in the caption of the same figure. The calorific thermal resistance is the same for a specific Reynolds number and decreasing with increasing the same due to the increase in the mass flow rate. Figure 39 shows the effect of increasing Reynolds number on the temperature profile of the fluid at the outlet of the microchannel. Adding pin fins has a noticeable impact on the convective thermal resistance; the average reduction is around 32% over the given Reynolds number range, as shown in Figure 38.

Figure 38: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannels embedded with pin fins at peaks and valleys (0.2sin(𝑥), \(D_{hy} = 150 \mu m\), pin fins diameter = 75 \(\mu m\)) – effect of amplitude (A)
Figure 39: Temperature contours at fluid’s outlet for wavy microchannels embedded with pin fins at peaks and valleys \((0.1 \sin(x), D_{hy} = 150 \, \mu m,\) pin fins diameter = 75 \, \mu m), – effect of amplitude \((A)\)

Figure 40 shows contours of outlet temperature for the same cases discussed in Figure 38; it shows that adding pin fins improves the temperature distribution, and that can also be quantified in terms of the standard deviation of temperature (without pin fins \(TSD_{out} = 5.13 \, K\) vs. with pin fins \(TSD_{out} = 2.42 \, K\)). The maximum substrate temperature for all the cases studied in this section, as well as the average Nusselt number plots, are shown in Figures 41 and 42, respectively. Figure 41 shows that increasing Reynolds number decreases the temperature for all designs, it also shows that increasing the amplitude reduces the temperature at the same Reynolds number. In Figure 42, the average Nusselt number increases with increase in Reynolds number. At the same Reynolds number, the largest amplitude value has the highest Nusselt number value.
Figure 40: Temperature contours at fluid’s outlet for wavy microchannels $(0.2 \sin(x), D_{\text{hy}} = 150 \mu m)$ and wavy microchannels embedded with pin fins at peaks and valleys $(0.2 \sin(x), D_{\text{hy}} = 150 \mu m, \text{pin fins diameter} = 75 \mu m)$, Re = 1000 – effect of amplitude (A)

Figure 41: Maximum substrate temperature vs. Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys $(D_{\text{hy}} = 150 \mu m, \text{pin fins diameter} = 75 \mu m)$ – effect of amplitude (A)
Figure 42: Variation of Nusselt number with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys ($D_{hy} = 150 \, \mu m$, pin fins diameter = 75 \, \mu m) – effect of amplitude (A)

The effect of the increase in the surface area of the microchannel walls due to the embedded pin fins is shown in Table 4, representing the following parameters ($0.2\sin(x)$, pin fins diameter = 75 \, \mu m). The surface area increased from 11.41531 mm$^2$ to 11.69806 mm$^2$, around a 2% increase. In terms of thermal performance, this increase in the surface area also showed a similar increase (around 2%). The increase seems to be very limited compared to the effect of the wavy geometry itself and the effect of pin fins on the disruption of the boundary layer on the overall thermal performance of the wavy configuration. These results vary from one case to another based on the different parameters, but in general, the effect is expected to be similar.
Table 4: Effect of pin fins surface area on the overall thermal performance of wavy microchannel

<table>
<thead>
<tr>
<th>Re</th>
<th>$hA$ (Without surface area of pin fins) (W/K)</th>
<th>$hA$ (With surface area of pin fins) (W/K)</th>
<th>Difference %</th>
</tr>
</thead>
<tbody>
<tr>
<td>250</td>
<td>0.4513</td>
<td>0.4624</td>
<td>2</td>
</tr>
<tr>
<td>500</td>
<td>0.6612</td>
<td>0.6776</td>
<td>2</td>
</tr>
<tr>
<td>750</td>
<td>0.8223</td>
<td>0.8426</td>
<td>2</td>
</tr>
<tr>
<td>1000</td>
<td>0.9683</td>
<td>0.9923</td>
<td>2</td>
</tr>
<tr>
<td>1250</td>
<td>1.0994</td>
<td>1.1266</td>
<td>2</td>
</tr>
<tr>
<td>1500</td>
<td>1.1911</td>
<td>1.2206</td>
<td>2</td>
</tr>
<tr>
<td>1750</td>
<td>1.2634</td>
<td>1.2947</td>
<td>2</td>
</tr>
<tr>
<td>2000</td>
<td>1.3301</td>
<td>1.3631</td>
<td>2</td>
</tr>
</tbody>
</table>
3.2.2.2 The Effect of Frequency (B) – Pin Fins at Peaks and Valleys

The effect of increasing the frequency has been shown to enhance the thermohydraulic performance of the heat sink, as discussed earlier in this chapter. Further simulation on the impact of introducing pin fins showed additional improvement in the performance. Figure 43 shows the effect of increasing the frequency in the wavy microchannel embedded with pin fins; the thermal resistance decreases with an increase in Reynolds number for the same frequency and it is also decreasing for the same Reynolds number with an increase in frequency. This behavior can be explained by discussing both calorific thermal resistance and convective thermal resistance (Figure 44) and geometry (Figure 45).

![Graph showing thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (D_hv = 150 µm, pin fins diameter = 75 µm) – effect of frequency (B)](image)

Figure 43: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (D_hv = 150 µm, pin fins diameter = 75 µm) – effect of frequency (B)

Figure 44 shows that for the same Reynolds number, the convective thermal resistance is decreasing with an increase in the frequency, and that can be explained by Figure 45, which shows that within the same length, increasing the frequency
increases the number of peaks and valleys as well as the number of pin fins. This increase causes the fluid to face more resistance and causes larger disruption and, as a result, a decrease in the thermal resistance due to the reduction in the average thickness of the boundary layer. The calorific thermal resistance is the same for all cases at the same Reynolds number since the mass flow rate is the same. The increase in pressure drop with an increase in frequency is much larger than the change in amplitude. This is mainly due to the increase in the number of pin fins and the increase in the change of direction of the flow.

Figure 44: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys ($D_{hy} = 150 \, \mu m$, pin fins diameter = 75 $\mu m$) – effect of frequency ($B$)

Figure 45: The geometrical effect of the frequency ($B$) on the wavy microchannel embedded with pin fins at peaks and valleys ($D_{hy} = 150 \, \mu m$, pin fins diameter = 75 $\mu m$) – effect of frequency ($B$)
It is also shown in Figure 44 that the effect of the pin fins, as well as the increase in frequency, becomes very minimal at high Reynolds numbers. At high Reynolds numbers, the repeated change in flow direction has a significant impact rather than the pin fins and the change in frequency. The increase in frequency is shown to have a different effect for change in \( B \) from \( B = 1 \) to \( B = 2 \) than for change from \( B = 2 \) to \( B = 3 \); the former has more impact. Changing the frequency from \( B = 2 \) to \( B = 3 \) has less impact due to the peaking of the degree of disruption of the boundary layer for a specific Reynolds number. Figure 46 shows velocity and temperature contours at fluid’s outlet for the effect of changing the frequency on the performance; the standard deviation of temperature for \( \text{Re} = 1000 \) at \( A = 1 \) is found to be \( T_{SD,\text{out}} = 2.86 \text{ K} \), \( A = 2 \) is found to be \( T_{SD,\text{out}} = 2.03 \text{ K} \) for \( A = 3 \) is found to be \( T_{SD,\text{out}} = 1.42 \text{ K} \).

The maximum substrate temperature for all the cases covering the effect of the frequency as well as the Nusselt number plots are shown below, Figures 47 and 48, respectively.
Figure 46: Temperature and velocity contours at fluid’s outlet surface for wavy microchannel embedded with pin fins at peaks and valleys ($D_{by} = 150 \text{ µm}$, pin fins diameter = $75 \text{ µm}$) – effect of frequency ($B$), $Re = 1000 \& 500$. (a,b) temperature, (c,d) velocity
Figure 47: Maximum substrate temperature vs. Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys ($D_{hy} = 150 \mu m$, pin fins diameter = $75 \mu m$) – effect of frequency (B)

Figure 48: Variation of Nusselt number with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys ($D_{hy} = 150 \mu m$, pin fins diameter = $75 \mu m$) – effect of frequency (B)
3.2.2.3 The Effect of Diameter of Pin Fins – Pin Fins at Peaks and Valleys

The influence of the pin fins diameter on the thermohydraulic performance of the microchannel is shown in Figure 49. The existence of pin fins, as shown previously, causes more significant disruption of the boundary layer than that without pin fins, and subsequently increase in diameter of pin fins brings about greater disruption of the boundary layer. With the increase in Reynolds number, the disruption of boundary layer due to changes in the flow direction, associated with the structure of the microchannel, increases, and the increase of the diameter of pin fins causes even more disruption, and these factors lead to a reduction in the thermal resistance. On the other hand, the pressure drop associated with increasing the pin fins diameter is relatively higher than changing the amplitude and frequency. The increase in the pin fins' diameter reduces the area available for fluid flow and enhances the disruption of the hydrodynamic boundary layer, but this also increases the pressure drop.

![Figure 49: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (0.2sin(2x), D_{hy} = 150 µm) – effect of pin fins diameter](image)
Figure 50 shows the influence of the pin fins diameter on the pressure drop along the length of the wavy microchannel. The vertical line represents the pressure drop and recovery due to the pin fin; it also shows the effect of pin fins diameter on the velocity contours at the same location. Figure 51 shows how the temperature profile at the fluid’s outlet is affected by increasing the pin fins diameter, $T_{SD\_out} = 1.85 \text{ K at } 75 \mu m, T_{SD\_out} = 1.24 \text{ K at } 125 \mu m.$

![Figure 50: Pin fins diameter influence on the pressure drop in wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2x), D_{hy} = 150 \mu m$) – effect of pin fins diameter](image)

![Figure 51: Temperature contours at fluid’s outlet surface for wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2x), D_{hy} = 150 \mu m$) – effect of pin fins diameter, Re = 1000](image)
The streamlines near pin fins are shown in Figure 52. It can be noticed that the largest diameter causes the fluid to decrease its velocity significantly when it first hits the pin fins, which causes a significant pressure drop. The maximum substrate temperature as well as the Nusselt number plots for all the cases of pin fins are shown in Figures 53 and 54, respectively.

Figure 52: Velocity streamlines at midplane of wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2x)$, $D_{hy} = 150 \mu m$) – effect of pin fins diameter, Re = 1000
Figure 53: Maximum substrate temperature vs. Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2\pi x)$, $D_{by} = 150 \mu$m) – effect of pin fins diameter

Figure 54: Variation of Nusselt number with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2\pi x)$, $D_{by} = 150 \mu$m) – effect of pin fins diameter
3.2.2.4 The Effect of Hydraulic Diameter – Pin Fins at Peaks and Valleys

In Figure 55, it is shown that increasing the hydraulic diameter reduces the thermal resistance significantly while also reducing the pressure drop in contrast to the influence of the previously studied parameters. Increasing the Reynolds number decreases thermal resistance, while for the same Reynolds number, increasing the hydraulic diameter decreases both the thermal resistance and pressure drop. An increase in hydraulic diameter is associated with an increase in the cross-sectional area between the pin fins and the microchannel walls and a lower velocity of flow, Figure 56, which leads to a reduction in pressure drop with the same.

![Figure 55: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (0.2sin(2x), pin fins diameter = 75 µm) – effect of hydraulic diameter](image)

For a specific Reynolds number, an increase in hydraulic diameter leads to an increase in the mass flow rate. This leads to a reduction in calorific thermal resistance with an increase in the same. Simultaneously, an increase in hydraulic diameter leads to a decrease and an increase in heat transfer coefficient and heat transfer surface area,
respectively; however, the decrease in heat transfer coefficient is greater than the increase in the surface area, thereby increasing convective thermal resistance with an increase in hydraulic diameter. The reduction in calorific thermal resistance with an increase in hydraulic diameter dominates the increase in convective thermal resistance with an increase in hydraulic diameter (Figure 57). This leads to a reduction in the overall thermal resistance with an increase in hydraulic diameter.

Figure 56: Cross section of pin fin, effect of hydraulic diameter – pin fins at peaks and valleys

Figure 57: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (0.2\sin(2\pi x), pin fins diameter = 75 \mu m) – effect of hydraulic diameter
The maximum substrate temperature for all the cases covering the effect of the hydraulic diameter as well as the Nusselt number plots are shown below, Figures 58 and 59, respectively. The effect of hydraulic diameter on the velocity streamline near pin fin is shown in Figure 60, and the temperature profile at the fluid’s outlet is shown in Figure 61.

Figure 58: Maximum substrate temperature vs. Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (0.2sin(2x), pin fins diameter = 75 µm) – effect of hydraulic diameter

Figure 59: Variation of Nusselt number with Reynolds number for wavy microchannel embedded with pin fins (0.2sin(2x), pin fins diameter = 75 µm) – effect of hydraulic diameter – pin fins at peaks and valleys
Figure 60: The effect of hydraulic diameter on the velocity streamline near pin fin for wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2x)$, pin fins diameter = 75 µm) – effect of hydraulic diameter, Re = 1000

Figure 61: Temperature contours at fluid’s outlet surface for wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(2x)$, pin fins diameter = 75 µm) – effect of hydraulic diameter, Re = 1000
3.2.2.5 The Effect of Pin Fins Shape – Pin Fins at Peaks and Valleys

The shape of the pin fins has similar impact on the performance of the microchannel heat sink due to the unique geometrical properties of each shape. The suggested shapes are square, triangle, and circle. Figure 62 shows the thermohydraulic performance of these shapes, while Figure 63 shows the convective and calorific thermal resistance of the same. Both figures show that the performance in terms of reduction of thermal resistance is almost the same for all different shapes with a slight advantage observed in the case of the square shape. The overall performance shows that the circular pin fins have a little advantage if the performance is set to be determined in terms of both thermal resistance and pressure drop. At high Reynolds numbers, the pressure drop associated with square pin fins is about 1.8 times the circular pin fins, and the pressure drop associated with triangular pin fins is about 1.2 times the circular pin fins. Figure 64 shows the effect of the pin fins shapes on the velocity profiles.

Figure 62: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys (0.2sin(x), D_hy = 150 µm, pin fins diameter/side = 75 µm) – effect of pin fins shape
Figure 63: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannel embedded with pin fins ($0.2\sin(x)$, $D_{hv} = 150\ \mu m$, pin fins diameter = 75 $\mu m$) – effect of pin fins shape – pin fins at peaks and valleys

Figure 64: Velocity contours of fluid at midplane of wavy microchannel embedded with pin fins at peaks and valleys ($0.2\sin(x)$, $D_{hv} = 150\ \mu m$, pin fins diameter = 75 $\mu m$) – effect of pin fins shape, $Re = 1000$
The maximum substrate temperature as well as the Nusselt number plots for all the cases of the pin fins shape are shown below, Figures 65 and 66, respectively.

Figure 65: Variation of Nusselt number with Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys \((0.2\sin(x), D_{hy} = 150 \, \mu m, \text{pin fins diameter} = 75 \, \mu m)\) – effect of pin fins shape

Figure 66: Maximum substrate temperature vs. Reynolds number for wavy microchannel embedded with pin fins at peaks and valleys \((0.2\sin(x), D_{hy} = 150 \, \mu m, \text{pin fins diameter} = 75 \, \mu m)\) – effect of pin fins shape
3.2.2.6 Pin Fins in The Middle Between Peaks and Valleys

The influence of the amplitude (A) on the performance of the wavy microchannels embedded with pin fins in the middle between peaks and valleys is shown in Figure 67. It can be noticed that increasing the amplitude has a greater impact at lower Reynolds numbers than at higher Reynolds numbers. An increase in amplitude causes further disturbance of the boundary layer in the low Reynolds number range, resulting in a decrease in thermal resistance. While at high Reynolds numbers, disruption is almost independent of the amplitude, resulting in an almost similar thermal resistance behavior. Simultaneously, increasing the amplitude causes pressure drop for a specific Reynolds number.

Figure 67: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins between peaks and valleys (D\text{hy} = 150 \mu m, pin fins diameter = 75 \mu m) – effect of amplitude (A)
Figure 68 shows the influence of changing the frequency on the thermohydraulic performance of wavy microchannel embedded with pin fins in the middle between peaks and valleys. An increase in the frequency of the sinusoidal microchannel decreases the thermal resistance while increasing the pumping power. Increasing the frequency reduces the thermal resistance for the same Reynolds number while increasing the pressure drop simultaneously.

![Graph showing thermal resistance and pressure drop](image)

Figure 68: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins between peaks and valleys ($D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$) – effect of frequency (B)

Figure 69 shows the effect of the pin fins diameter on the performance of the wavy microchannels embedded with pin fins in the middle between peaks and valleys. The impact of changing the diameter on the thermal resistance is more apparent at low Reynolds numbers, while it is almost negligible at higher Reynolds numbers. Changing pin fins diameter at low Reynolds number causes greater disruption of the boundary layer resulting in a reduction in the average boundary layer thickness. On the other hand, at a higher Reynolds number, the impact of changing the flow direction
rapidly has a superior influence than the impact of changing the pin fins diameter. Increasing pin fins’ diameter significantly impacts the pressure drop; the pressure drop increases almost ten fold when increasing the diameter from 75 $\mu$m to 125 $\mu$m at $Re = 2000$. This increase in the pressure drop is caused mainly by the reduction in the area between the pin fin and the walls, but this also enhances the disruption of the flow and subsequently decreases the thermal resistance.

Figure 69: Thermal resistance and pressure drop with Reynolds number for wavy microchannel embedded with pin fins between peaks and valleys ($0.2\sin(x)$, $D_{hy} = 150 \mu m$) – effect of pin fins diameter

The performance of the pin fins at peaks and valleys compared to pin fins in the middle between peaks and valleys is shown in Figure 70. The former has a slight advantage compared to the latter. The reduction of convective thermal resistance is larger when the pin fins are at peaks and valleys, as shown in Figure 71. Another factor that gives the advantage to the design with pin fins at peaks and valleys is shown in Figure 72. By comparing the geometry of the entire microchannel, it can be observed that based on the selected length of the microchannel, as well as the frequency, some
cases of either of the two design can have more pin fins which increase the disruption of the flow resulting in a slightly better performance. In this case, the wavy microchannel with pin fins at peaks and valleys has one more pin fin, which adds more overall to the disruption of the flow.

Figure 70: Thermal resistance and pressure drop with Reynolds number for wavy microchannels embedded with pin fins \((0.1\sin(2x), D_{hy} = 150 \mu m, \text{pin fins diameter } = 75 \mu m)\) – pin fins at peaks and valleys vs. pin fins in the middle

The effect of the location of the pin fins, as well as the design without pin fins on the fluid’s outlet temperature profile, is shown in Figure 73. The velocity contours of a midplane across the fluid domain of the different pin fins locations and with no pin fins are shown in Figure 74.
Figure 71: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannels embedded with pin fins ($0.1 \sin(2x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$) – pin fins at peaks and valleys vs. pin fins in the middle.

Figure 72: Wavy microchannels embedded with pin fins ($0.2 \sin(2x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$), pin fins at peaks and valleys vs. pin fins in the middle, full length.

Figure 73: Temperature contours at fluid’s outlet of wavy microchannel embedded with pin fins ($0.2 \sin(2x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$), no fins ($T_{SD, out} = 2.48 \, K$), fins in the middle ($T_{SD, out} = 1.421 \, K$), fins at peaks and valleys ($T_{SD, out} = 1.311 \, K$), $Re = 1000$. 
Figure 74: Velocity contours at fluid’s midplane of wavy microchannel embedded with pin fins (0.2\sin(2x), D_{hy} = 150 \text{ m}, \text{pin fins diameter} = 75 \text{ m}) – no pin fins vs. pin fins at peaks and valleys vs. pin fins in the middle, \text{Re} = 1000

The Appendix in the end of this thesis contains the full list of the other figures covering all the cases discussed in this chapter that were not included in this chapter. In addition to the pumping power per channel calculated based on the obtained pressure drop. The caption of each figure can be used to follow any specific case.
3.3 Experimental Results

In this section, the experimental work necessary for model validation is discussed and the results are compared with the same from the CFD model. Figure 75 shows the important aspects of validation. Two designs were selected, Figure 76, 0.1sin(\(x\)) \([D_{by} = 150 \, \mu m, \text{pin fins diameter} = 75 \, \mu m]\), and 0.2sin(2\(x\)) \([D_{by} = 171.429 \, \mu m, \text{pin fins diameter} = 75 \, \mu m]\). Figure 77 shows one of the fabricated devices 0.2sin(2\(x\)), the zoom in section clearly shows the pin fins located at the peaks and valleys of the sinusoidal wave.

![Figure 75: Experimental validation of computational model [45]](image)

![Figure 76: Selected configurations for experimental validation](image)
In order to seal the heat sink from the upper side, Polydimethylsiloxane (PDMS) was prepared as shown in Figure 78 (a, b). After preparing the PDMS layers, they were bonded with the heat sinks through plasma oxidation process. The result is shown in Figure 78 (c).

Figure 78: PDMS preparation and bonding with heat sinks. (a,b) PDMS preparation, (c) heat sink bonded with PDMS
The experimental setup is shown in Figure 79. This figure shows the main components from the inlet to the outlet tanks. A list of the main components is shown below:

1- Power Supply: TTi EL302D Dual Power Supply.

2- Micropump: “TCS Micropumps - Diaphragm Pump D250S-L-SV”.

3- Syringe Filter.

4- Heater: “Polyimide Thermofoil™ Heaters: HK6909”.

5- Thermocouple: Type J – Omega.

The heat applied to the heat sink is generated through a heater attached to the bottom of the heat sink, the foam acts as the insulation (the heater is not in contact with the foam). The temperatures inlet, outlet and bottom surface (two thermocouples connected, number two in Figure 79 and another one through the foam from the bottom side) are measured through thermocouples connected to a data logger; measurements are taken after the temperature reached steady state. The heater and pump are controlled through a power source. Two different voltages are provided to the heater which means each configuration is tested twice (20V and 30V), while the pump is controlled up to 6V (the micropump max. voltage). The setup is run using different conditions and the results are noted and compared with CFD results for validation. To reduce the hysteresis effect, different input parameters are used for each case and each sub case.
The results of the selected data points are shown in Figures 80 and 81, respectively. The results show good match between the experimental and CFD results confirming the validity of the mathematical model used to solve the system, hence the results obtained from the CFD based study are indeed accurate and reliable. The uncertainty of the thermal resistance calculation based on the experimental measurements can be calculated using Equation (3.1), the uncertainties of the voltage, current, and temperature are obtained from the devices’ specifications.

\[
\chi_{R_{th}} = \sqrt{\left( \frac{\Delta T}{\text{Power}_{H} \chi_{\Delta T}} \right)^2 + \left( -\frac{\Delta T}{\text{Power}_{H} \cdot \text{Voltage}_{H}} \chi_{\text{Voltage}_{H}} \right)^2 + \left( -\frac{\Delta T}{\text{Power} \cdot \text{Current}_{H}} \chi_{\text{Current}_{H}} \right)^2}
\]  

(3.1)
Figure 80: Validation, experimental vs. CFD results, $\Delta T$ (K)

Figure 81: Validation, experimental vs. CFD results, $R_{th}$ (K/W/cm$^2$)
Chapter 4: Conclusion

The thermohydraulic performance of different configurations of heat sinks employing wavy microchannels embedded with pin fins is studied using a parametric study through CFD simulation supported by experimental investigation of few cases for validation. Several parameters are studied covering different aspects of the heat sink. The performance of the wavy microchannels heat sinks embedded with pin fins is compared to other traditional heat sinks to understand the improvement and impact of pin fins on the wavy heat sink's performance. The microchannel geometry covers the amplitude, frequency, and hydraulic diameter; additionally, the pin fins' geometry, including the diameter, shape, and location of pin fins, are also studied. The study is done for the Reynolds number ranging between 250 - 2000. The thermohydraulic performance is evaluated mainly based on thermal resistance and pressure drop supported by other findings, including the maximum chip temperature and Nusselt number. The following summarizes the main findings and shows the effect of each parameter on the performance.

1- The wavy microchannels heat sinks embedded with pin fins show a better thermal resistance reduction at the same Reynolds number than other configurations (straight, straight embedded with pin fins, and wavy). The structure of the wavy microchannels design introduces a continuous change in the flow direction, where the pin fins enhance the disruption within the microchannels. Simultaneously, the reduction in thermal resistance related to the existence of pin fins is associated with increases in the pressure drop; the wavy microchannels heat sinks embedded with pin fins show the highest pressure drop among all other configurations.
2- Altering the microchannel amplitude and frequency is found to have a considerable impact on the overall performance. Increasing both helps increase the disruption in the flow, resulting in a reduction in the average thickness of the boundary layer, decreasing the overall thermal resistance. The frequency is found to have more impact than the amplitude due to the change in geometry of the microchannel itself, affecting the number of peaks and valleys (also the number of pin fins accordingly) in the former than the degree of curvature in the latter.

3- Increasing the pin fins diameter improves the thermal performance but at the cost of considerable pressure drop compared to altering the amplitude and frequency. This is due to the reduction in the cross-sectional area between the pin fins and walls (the area available for the fluid flow).

4- The effect of increasing the hydraulic diameter shows a reduction in both thermal resistance and pressure drop. The reduction in the thermal resistance is mainly caused by the increase in mass flow rate and, as a result, a reduction in the calorific thermal resistance is observed. The decrease in pressure drop associated with the increase in hydraulic diameter is due to the increase in the cross-sectional area between the pin fins and the microchannel walls, which also reduces the velocity of flow.

5- Among the different shapes of the pin fins, the circular (which was used in the previous simulations) has the lowest pressure drop and almost similar behavior as of the square shape with regards to thermal resistance.

6- Changing the pin fins' location from peaks and valleys of the sinusoidal microchannel to the middle between peaks and valleys shows similar behavior in most cases when changing the other different parameters. The location of pin fins at peaks and valleys shows a slight improvement in the low Reynolds number
range; however, at the high Reynolds number range there is little difference in the thermal performance.

This work discusses and investigates a new type of heat sink employing wavy microchannels embedded with pin fins. The work shows the potential of the conceptualized designs and overall improvement in the thermal performance. Further work can be carried out to investigate and expand the potentials of the suggested design. One approach can be by employing nanofluids to enhance the heat transfer process and overcome related side effects such as the increase in the pressure drop. The other area of improvement can be the optimization of the earlier mentioned parameters. The impact of these parameters is found to be limited in some scenarios and significant in others; an optimization study can find the balance between the improvement and the associated costs.
References


Figure 82: Pumping power with Reynolds number for straight ($D_{hy} = 150 \, \mu m$), straight embedded with pin fins ($D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$, pin fin pitch = $2500 \, \mu m$), wavy ($0.3 \sin (x)$, $D_{hy} = 150 \, \mu m$), and wavy embedded with pin fins ($0.2 \sin (x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$) – effect of amplitude (A) – pin fins at peaks and valleys

Figure 83: Pumping power with Reynolds number for wavy microchannel ($0.2 \sin (2x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter/sides = $75 \, \mu m$) – effect of pin fins shape
Figure 84: Pumping power with Reynolds number for wavy microchannel ($D_{hv} = 150 \, \mu m$) – effect of amplitude (A)

Figure 85: Pumping power with Reynolds number for wavy microchannel ($D_{hv} = 150 \, \mu m$) – effect of frequency (B)
Figure 86: Pumping power with Reynolds number for wavy microchannel embedded with pin fins ($D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$) – effect of amplitude (A) – pin fins at peaks and valleys

Figure 87: Pumping power with Reynolds number for wavy microchannel embedded with pin fins ($D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$) – effect of frequency (B) – pin fins at peaks and valleys
Figure 88: Pumping power with Reynolds number for wavy microchannel embedded with pin fins (0.2sin(2.x), D_{hy} = 150 µm) – effect of pin fins diameter – pin fins at peaks and valleys

Figure 89: Pumping power with Reynolds number for wavy microchannel embedded with pin fins (0.2sin(2.x), pin fins diameter = 75 µm) – effect of hydraulic diameter (D_{hy}) – pin fins at peaks and valleys
Figure 90: Pumping power with Reynolds number for wavy microchannel embedded with pin fins (Dh = 150 µm, pin fins diameter = 75 µm) – effect of amplitude (A) – pin fins in the middle between peaks and valleys

Figure 91: Pumping power with Reynolds number for wavy microchannel embedded with pin fins (Dh = 150 µm, pin fins diameter = 75 µm) – effect of frequency (B) – pin fins in the middle between peaks and valleys
Figure 92: Pumping power with Reynolds number for wavy microchannel embedded with pin fins ($0.2\sin(x)$, $D_{hy} = 150 \, \mu m$) – effect of pin fins diameter – pin fins in the middle between peaks and valleys

Figure 93: Pumping power with Reynolds number for wavy microchannel embedded with pin fins ($0.1\sin(2x)$, $D_{hy} = 150 \, \mu m$, pin fins diameter = 75 $\mu m$) – performance of wavy microchannel with pin fins at peaks and valleys vs in the middle between peaks and valleys
Figure 94: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannels ($D_{hy} = 150 \, \mu m$) – effect of amplitude (A)

Figure 95: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannels ($D_{hy} = 150 \, \mu m$) – effect of frequency (B)
Figure 96: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannels embedded with pin fins at peaks and valleys (0.2\sin(2x), D_{hy} = 150 \mu m) – effect of pin fins diameter

Figure 97: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannel (D_{hy} = 150 \mu m, pin fins diameter = 75 \mu m) – effect of amplitude (A) – in the middle between peaks and valleys
Figure 98: Variation of calorific and convective thermal resistance with Reynolds number for wavy microchannel ($D_{hy} = 150 \, \mu m$, pin fins diameter = $75 \, \mu m$) – effect of frequency (B) – in the middle between peaks and valleys.

Figure 99: Maximum substrate temperature vs. Reynolds number for wavy microchannels ($D_{hy} = 150 \, \mu m$) – effect of amplitude (A).
Figure 100: Variation of Nusselt number with Reynolds number for wavy microchannels ($D_{hy} = 150 \mu m$) – effect of amplitude (A)

Figure 101: Maximum substrate temperature vs. Reynolds number for wavy microchannels ($D_{hy} = 150 \mu m$) – effect of frequency (B)
Figure 102: Variation of Nusselt number with Reynolds number for wavy microchannels ($D_{hy} = 150 \, \mu m$) – effect of frequency (B)

Figure 103: Maximum substrate temperature vs. Reynolds number for wavy microchannel ($D_{hy} = 150 \, \mu m$, pin fins diameter = 75 $\mu m$) – effect of amplitude (A) – in the middle between peaks and valleys
Figure 104: Variation of Nusselt number with Reynolds number for wavy microchannel ($D_{hy} = 150 \, \mu\text{m}$, pin fins diameter = $75 \, \mu\text{m}$) – effect of amplitude (A) – in the middle between peaks and valleys

Figure 105: Maximum substrate temperature vs. Reynolds number for wavy microchannel ($D_{hy} = 150 \, \mu\text{m}$, pin fins diameter = $75 \, \mu\text{m}$) – effect of frequency (B) – in the middle between peaks and valleys
Figure 106: Variation of Nusselt number with Reynolds number for wavy microchannel ($D_{hy} = 150 \mu m$, pin fins diameter = 75 $\mu m$) – effect of frequency (B) – in the middle between peaks and valleys

Figure 107: Maximum substrate temperature vs. Reynolds number for wavy microchannel ($0.2 \sin(x)$, $D_{hy} = 150 \mu m$) – effect of pin fins diameter – in the middle between peaks and valleys