EFFECT OF VORTEX GENERATORS ON THE OVERALL PERFORMANCE OF MICROCHANNEL HEAT SINK

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Submitted by

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June, 2022

CERTIFICATE

This is to certify that **Mr. Meraj Ansari** (Enroll. No. 1900104041) has carried out the research work presented in the thesis titled **"Effect of vortex generators on the overall performance of microchannel heat sink"** submitted for partial fulfilment for the award of the **Degree of Master of Technology in Mechanical Engineering** from **Integral University, Lucknow** under my supervision.

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DECLARATION

I hereby declare that the thesis titled **EFFECT OF VORTEX GENERATORS ON THE OVERALL PERFORMANCE OF MICROCHANNEL HEAT SINK** is a record of research work carried out by me under supervision of Er Mohammad Nawaz Khan, Assistant Professor, **Department of Mechanical Engineering, Integral University, Lucknow.** No part of this thesis has been presented elsewhere

I declare that I have faithfully acknowledged and referred to the work of other researchers wherever their published work has been cited in this thesis. I further certify that I have not willfully taken other's work, text, data, results etc reported in the journals, books, magzines, reports etc.

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TABLE OF CONTENTS

Contents	Page No

LIST OF FIGURES

ABSTRACT

Microchannels are of current interest for use in compact heat exchangers, micro biochips, micro reactors, VLSI system where very high heat transfer performance is desired. This electronic equipment are virtually synonyms with modern life applications such as appliances, instruments and computers. The dissipation of heat is necessary for the proper functioning of these instruments Microchannels provide very high heat transfer coefficients because of their small hydraulic diameters. Here, an investigation of fluid flow and heat transfer in microchannels is conducted. Fluid flow and heat transfer experiments were conducted on a silicon microchannel heat exchanger (MHE) .A three-dimensional Computational Fluid Dynamics (CFD) model was built using the commercial package, FLUENT, to investigate the conjugate fluid flow and heat transfer phenomena in a siliconbased rectangular microchannel heat sink. This work focused on laminar flow (Re < 200) within rectangular microchannel with hydraulic diameter 86μm for single-phase liquid flow. The influence of the thermophysical properties of the fluid on the flow and heat transfer, are investigated by evaluating thermophysical properties at a reference bulk temperature. The micro-heat sink model consists of a 10 mm long silicon substrate, with rectangular microchannels, 57μm wide and 180μm deep, fabricated along the entire length. water at 293K is taken as working fluid. The results indicate that thermophysical properties of the liquid can significantly Influence both the flow and heat transfer in the microchannel. Assumption of hydrodynamic, fully developed laminar flow is valid here on basis of Langhaar’s equation. The local heat transfer coefficient and averaged Nusselt number is calculated and plotted for pressure drop of 50kpa, 30kpa and 10kpa.also result is verify for heat flux 50W/cm2 , 90W/cm2 and 150 W/cm2 .

Chapter 1

INTRODUCTION

In the last few decades, the integrated level of the electronic devices has been rapidly increasing. Along with this, the weight of the component has been on the decrease. It has led to higher heat flux produced by those parts due to less area of contact with atmospheric air. There has been also an increase in the failure rate of the element due to overheating. In order to reduce the failure rates, the effective and feasible method of thermal management is required immediately with less weight to maintain safe temperature limits of the components with compact size. These requirements led to the invention of the microchannel heat sink by Tuckerman (1981). They proposed the practical feasibility of integrated electronic cooling devices using water as working fluid in a rectangular shaped microchannel heat sink. Finally, they achieved high heat transfer coefficient between water and substrate.

The need for removal of high heat flux with small area attracted many researchers to focus on Micro-Channel Heat Sink (MCHS). Currently, the MCHS is used for high heat flux applications like electronic cooling, because of its less weight, compactness and high heat transfer rate when compared to other microelectronic cooling systems. Figure 1.1. shows that the microchannel heat sink geometry used in the microprocessor by

Kandlikar (2005), where L is the length and W is the width of the microchannel. The microchannel comprises the family of the heat exchanger, which is used to transfer heat from high-temperature medium to low temperature medium. A microchannel can be defined as channels whose hydraulic diameter is less than 1 millimeter and greater than 1 micron. Above 1 millimetre the flow becomes macroscopic flows in which the coolant flows inside the channel to remove the heat by convection.

Microchannels can be fabricated using many materials like glass, polymers, silicon, and metals through various processes including bulk micromachining, surface micromachining, embossing, moulding, and conventional machining with micro cutters. Heat transfer can be increased by means of decreasing the thermal boundary layer, increasing flow interruption, increasing the velocity gradient of the fluid flowing inside the channel varying the channel cross sections and also by placing the obstacles. The rectangular geometries are used mainly for industrial applications. A proper understanding of fluid flow and heat transfer in these microscale systems is therefore essential for their design and operation.

1.1 NEED FOR THE MICROCHANNEL

A channel is mainly used to bring the fluid to contact with the channel walls and also for transportation process. The rate of the fluid flow depends on the surface area, and Channel dimensions are employed based on various systems. For example, the biological systems use much smaller sizes, whereas larger channels are used for fluid transportation in mechanical systems. From the engineering point of view, we can reduce the diameter from larger value to smaller value. For the diameter less than 25 microns, they are used in Henle's and tubules loop capillaries. If the diameter is between 25 microns to 2.5 mm, then they are used in electronics cooling application. The diameter greater than 2.5 mm is used in boilers and power condensers tube. The early microchannel has the problem of non-uniform temperature distribution along flow direction which becomes unfavourable for electronic cooling. For that purpose, the channel has a different geometric configuration to overcome the problem. The fabricators improved the design with various parameters varying the thermal boundary layers of the fluid. In addition, they used high thermal conductivity material to decrease the thermal resistance for increasing heat transfer rate. Also, they reduced the thickness of the material to enhance heat transfer rate by convection.

The present electronic cooling systems primarily use air as the coolant. Air has the advantages of good compatibility with the microelectronic circuit environment, low auxiliary system support requirements, high reliability of the cooling system, low initial cost, low operating and maintenance cost and long development history and experience. But the main concern for air cooling systems is their low heat dissipation potential because of low specific heat value of air. The low heat transfer coefficient in air 5 cooling necessitates the use of heat spreader to increase heat transfer surface area. With the spreader, the air cooled heat sink experiences three thermal resistance components. They are: thermal resistance of the bonding material which bonds the spreader with electronic chip, thermal resistance of the spreader and the thermal resistance due to convection between the fin and the air. The spreader resistance can be reduced by using spreader of good conducting material. The convective resistance can be reduced by increasing the air flow rate. But reducing the resistance of the bonding material still poses problems, because of the requirement of a thick base of high thermal conductivity material for the heat spreader and a good bond between silicon and heat spreader. Another factor which limits further reduction in the bonding material thickness is the difference in thermal expansion coefficients for copper and the silicon substrate. Reducing the thickness of the bonding material causes much higher thermal stresses in the silicon substrate, leading to its mechanical failure. In such situations, microchannels may be directly etched on the back of the silicon substrate, which eliminates the contact resistance and associated thermal expansion problem. In addition to this, the high aspect ratio of microchannels further increases the convective heat transfer surface area and convective heat transfer coefficient, which in turn reduce the convective resistance. Equation (1.1) shows Newton's law of cooling which governs convective heat transfer from a surface. $Q = hA(T_s - T_a)$ (1.1) Generally, the temperature limits may be fixed and therefore the heat transfer rate can be increased by increasing the product hA. 6 Microchannels are channels whose characteristic dimensions are in the range of 10 μm - 1000 μm (Gad-el-Hak 2006). Thus, deep narrow microchannels etched at the backside of a silicon substrate, increase the surface area for heat transfer. The flow in the microchannels is generally laminar in nature, because of the smaller characteristic dimensions involved. For a fully developed internal flow, the Nusselt number (Nu) is constant. The Nusselt number is given in Equation (1.2) as Nu = hD/k (1.2) Rearranging for h results in Equation (1.3) h = Nu.k/D (1.3) It is seen that h is inversely proportional to the diameter of the microchannel, at constant Nusselt number. The microchannels possess the advantage of very high h value (of the order of few thousand W/m2 oC) because of their smaller diameter values. Hence for microchannels, the product hA is very large which results in larger convective heat transfer. This makes microchannel a suitable candidate among various high heat removal options of the order of 100 W/cm2 or more.

1.2 APPLICATIONS OF MICROCHANNELS

The removal of high heat flux from a microelectronic circuitry is one of the most important applications of microchannels. Current flow through any electronic component is always associated with some heat dissipation, for example, I2R losses in a resistor where I is the current and R is the resistance. The heat dissipated by the electronic circuitry must be removed and the component should be maintained at a reasonably low temperature to ensure its safety and reliability of operation. Gordan Moore, one of the founders of Intel Company predicted in 1965, that the number of transistors on a substrate would double roughly in every 2 years. The predictions are still applicable today, thanks to the emergence of micro and nano-fabrication techniques. Cramming more and more transistors into small volumes may be viewed as a boon from the electronics point of view. But it is really a challenge from the view point of thermal management. The heat dissipation from a single tiny transistor is not a problem, but it is undoubtedly an issue when it comes to heat dissipation from a few million transistors put together. Future electronic components are expected to dissipate heat in the range 100 W/cm2 or even more. Removal of such high heat fluxes and maintaining the temperature within the safer limit, using the present-day cooling technology (forced air cooled heat sink) is not possible. Therefore, it is essential to devise an innovative and effective cooling scheme to meet the cooling requirement of the electronic circuitry of the forthcoming years. One such promising option is 2 the use of microchannels with water as coolant, because microchannels can handle such high rates of heat flux. Apart from the cooling of microelectronics, microchannels are employed in several other applications. The Heatric Company of Australia developed the well-known commercial micro heat exchangers, using a chemical milling process analogous to the manufacture of electronic circuit boards. The heat exchanger is named as Printed Circuit Heat Exchanger (PCHE). PCHE have a hydraulic diameter between 700 µm to 1.5 mm. The PCHE can support pressures of 500-1000 bar and temperatures of 900oC. The biggest market for the PCHE has been in offshore gas processing owing to its high compactness, typically about one-fifth the size and weight of conventional heat exchangers, for the same thermal duty and pressure drops. Other applications include the manufacturing of LNG (Liquified Nitrogen Gas), ethylene oxide and sulphuric acid, naphtha reforming, fuel cell systems and caustic soda plants. The same technology is used by Heatric to make Printed Circuit Reactors (PCR), which extends the application of microchannel devices into chemical processing, chemical reaction and fuel processing. The PCR can combine mixing, reaction and heat transfer. MarBondTM heat exchanger comprises a stack of bonded plates that are etched photochemically to form a series of slots. The compact diffusion bonded structure facilitates high heat transfer surface areas per unit volume, high heat transfer coefficients and flexibility of design. Gas turbine component cooling is another area employing pin type micro heat exchangers, which are electroformed using a low-stress nickel sulfamate solution. The pins are 500 µm in diameter and 500 µm in height, spaced in a square array 1.25 mm apart. Compressed air bled from the compressor is directed through the core of the turbine blades and into micro-pin fins via coolant holes located at several chord locations on the blade surface. Micro heat exchangers are also preferred for meeting the heat 3 load in critical areas such as nuclear high temperature reactors. A small-scale plate type compact heat exchanger (CPCHX) for nuclear high temperature reactors is designed, which consists of concavo-convex plates (CPs) welded by solid-state diffusion and made of nickel-based superalloy Hastelloy XR, with a channel width about 1000 μm. The feasibility study confirmed that the diffusion-welded CPCHX can attain a heat capacity of 18.75 MW/m3 with the unit size of 1 m (width) × 1 m (length) × 4 m (height), and it can be integrated in the next-generation high temperature reactor. In addition to the above-mentioned applications microchannels are used in fuel cells, automobiles, liquid rocket engines and commercial heating/ cooling. (Fan and Luo 2008)

1.3 ADVANTAGES AND DISADVANTAGES OF MICROCHANNEL HEAT SINKS

A high surface area density substantially reduces the volume of heat exchanger needed for the same thermal power. As a result, the space and cost of material associated with construction and installation could be reduced significantly. Moreover, the fluid holdup is small in a micro heat exchanger; this is important for security and economic reasons when expensive, toxic, or explosive fluids are involved. The relatively enormous overall heat transfer coefficient of microchannel heat sinks makes the heat exchange procedure much more effective. In addition, the development of micro fabrication techniques such as LIGA (LIthografie Galvanoformung Abformung), stereolithography, laser beam machining, and electro forming allows the design of a micro heat exchanger with more effective configuration and high pressure resistance. The quick response time of a microchannel heat sink provides a better temperature control, for relatively small temperature differences between fluid flows. The quick response (small time constant) is connected to the small inertia of the heat transfer interface (the small metal 4 thickness that separates the two fluids). On the other hand, the exchanger as a whole, including the ìperiphericî material, usually has a greater inertia than conventional exchangers, entailing a large time-constant. Thus the response of one fluid to a temperature change of the other fluid comprises two ìtemperature-change wavesî, with very distinct time-constants. In conventional exchangers, it is possible that the two responses are blurred into one. Though microchannels are advantageous in many ways, they are not without limitations. Because of the extremely small size of channels, the pressure drop of the system not only requires high mechanical power for running the system but also make strict demands for accessories such as leakfree connections and pumping devices. Fine channels are sensitive to corrosion, roughness, and fouling of the surfaces. Channel walls can erode due to chemical etching or the physical wearing out of the inner wall surfaces. For this reason, only very clean fluids could be employed in a microchannel heat sink with protection filters, as mechanical cleaning and maintenance in general are not possible. Moreover, certain effects such as fluid maldistribution, axial direction heat conduction, and external losses, which may not be so significant in conventional scale heat exchangers, may also be present in microchannels.

1.4 FUNDAMENTAL ISSUES RELATED TO MICROCHANNELS

As the channel size becomes smaller, it becomes comparable to the molecular mean free path. The granular nature of gas flow in microchannels at this level of channel dimension causes departure from the continuum assumption. This is known as the rarefaction effect. A measure of departure from continuum model due to rarefaction is given by the Knudsen number (Kn), which is defined as the ratio of molecular mean free path to the hydraulic diameter of the microchannel. The flow is assumed to obey continuum laws for $Kn < 0.001$. Situations with higher values of Knudsen 7 number are treated as slip flow, transition flow or free molecular flow depending on the Kn value. For gas flows in microchannels, the large pressure drop gives rise to large changes in the gas density. In turn, some times, high velocity and high Mach number ($M \sim 1$) values are attained, especially towards the channel exit. For such situations, the density variations of gases cannot be ignored, and consideration of compressibility effect becomes important. Also, the fully developed flow assumption cannot be applied, as flow acceleration occurs over almost the entire channel length. Most of the surfaces have neutralized electrostatic charges at the free surface. When a liquid with even a small amount of ions flows over such a surface, the electrostatic charges on the non-conducting surfaces attract counter ions from the fluid. The balancing charge in the liquid is known as the ìElectric Double Layerî, whose thickness is of the order of a few nanometres. Therefore, for liquid flows in microchannels of very small sizes in the range of 10 microns or less, it is essential to consider the electrical double layer effect. Generally, microchannels are fitted with large headers for the purpose of distributing the flow equally among various channels. The turning of flow from the header region to the microchannels will bring in additional pressure drop which should be properly accounted for. Because of the very small size of microchannels, the pressure drop for flow across the channel will be extremely large, which demands the use of shorter channels (to reduce the pressure drop). In such situations, the developing flow length effect needs to be considered. Manufacturing channels of identical dimensions is very difficult; therefore, variations in the channel dimensions and surface roughness effects should be considered. The header design plays a major role in providing uniform flow through all channels. 8 Choosing proper flow and thermal boundary conditions at the channel wall is very essential in any modelling exercise of microchannel heat sinks, as these significantly influence the flow/thermal characteristics. The heat loss from the setup and heat transfer in the header may some times become important, while obtaining the performance characteristics of the heat sink. In microchannels, because of the high effectiveness, the temperature difference between the inlet and the outlet may be very small (less than or equal to 1oC). Therefore, accurate measurement of temperature is very essential. Sometimes, small holes are made on the walls of the channels to connect with the pressure taps. Such holes should be carefully made to avoid excessive interference with flow field. On the other hand, such tap holes also increase the response time of the setup to attain steady state. Care should be exercised to acquire data after the steady state is attained. The sizes of surface roughness elements are often comparable to the sizes of the channel, which can significantly alter the flow area of the microchannel. To overcome this difficulty, the effect of roughness should be properly accounted for. Above all, generally, microchannel heat sinks with multiple parallel channels will be employed for practical applications. If there are a large number of channels, then achieving uniform flow distribution becomes a big challenge. This highlights the importance of proper design for the flow headers. The channel to channel size and shape variation should be minimal; otherwise variations in channel shape and size will further affect the uniform flow distribution. Hence, it is essential to properly design the manifolds and to manufacture identical microchannels in multiple channel applications to avoid flow maldistribution. Among the various factors mentioned above, flow maldistribution is found to have a strong influence on the overall thermal performance of a microchannel based heat sink device (Kandlikar 2003).

Throughout the past 50 years, cooling and thermal management have played a key role in accommodating increases in power while maintaining component temperatures at satisfactory levels to satisfy performance and reliability objectives. Thermal management will play a pivotal role in the coming decade for all types of electronics products. Increased heat fluxes at all levels of packaging from chip to system to facility pose a major cooling challenge. To meet the challenge, significant cooling technology enhancements will be needed in each of the following areas:- Thermal interfaces, Heat spreading, Air cooling , Indirect and direct water cooling Immersion cooling Refrigeration cooling Thermoelectric cooling Data center cooling So the thermal design requirements to meet the growing demands are as follows, it is here Traditional Thermal Design Requirements are explained and categorized as follows:- Design for Performance , Design for Reliability, Design for Serviceability, Design for Extensibility , Design for minimal Cost and Design on minimal Impact on User

1.5 **New Thermal Design Technology**

Design for improved cool ability at the package level via optimized internal thermal conduction paths. · Design for direct air cooling at the product level via enhanced convection process over the packages. · Design for special cooling needs at the module level via spot cooling devices attached to the packages. · Design for low temperature applications- Sub ambient to cryogenic. · Design for low cost via computer aided thermal engineering (CATE) and improved manufacturability.

Current Methods Used in Industry: Here are the various types of methods used in electronics (Computer) industry to cool Modules, Systems, and Data centres

Module Level Cooling Processor module cooling is typically characterized in two ways: cooling internal and external to the module package and applies to both single and multi-chip modules. Fig (1.4) 14 illustrates the distinction between the two cooling regimes in the context of a single-chip module.

1.6 Internal Module Cooling

The primary mode of heat transfer internal to the module is by conduction. The internal thermal resistance is therefore dictated by the module's physical construction and material properties. The objective is to effectively transfer the heat from the electronics circuits to an outer surface of the module where the heat will be removed by external means which will be discussed in the following section. Fig (1.4):- Cross Section of a typical Module Denoting Internal Cooling Region and External Cooling Region 1.3.1.2 External Module Cooling Cooling external to the module serves as the primary means to effectively transfer the heat generated within the module to the system environment. This is accomplished primarily by attaching a heat sink to the module by Wu.and Little . Traditionally, and preferably, the system environment of choice has been air because of its ease of implementation, low cost, and transparency to the end user or customer. This section, therefore, will focus on air-cooled heat sinks. Liquid-cooled heat sinks typically referred to as cold plates will also be discussed. 15 1.3.1.3 Immersion Cooling Immersion cooling has been of interest as a possible method to cool high heat flux components for many years. Unlike the water-cooled cold plate approaches which utilize physical walls to separate the coolant from the chips, immersion cooling brings the coolant in direct physical contact with the chips. As a result, most of the contributors to internal thermal resistance are eliminated, except for the thermal conduction resistance from the device junctions to the surface of the chip in contact with the liquid. Direct liquid immersion cooling offers a high heat transfer coefficient which reduces the temperature rise of the heated chip surface above the liquid coolant temperature. The magnitude of the heat transfer coefficient depends upon the thermo-physical properties of the coolant and the mode of convective heat transfer employed. The modes of heat transfer associated with liquid immersion cooling are generally classified as natural convection, forced convection, and boiling. Forced convection includes liquid jet impingement in the single phase regime and boiling (including pool boiling, flow boiling, and spray cooling) in the two-phase regime.

1.7 System-Level Cooling

Cooling systems for computers may be categorized as air-cooled, hybrid cooled, liquid-cooled, or refrigeration- cooled. An air-cooled system is one in which air, usually in the forced convection mode, is used to directly cool and carry heat away from arrays of electronic modules and packages. In some systems air-cooling alone may not be adequate due to heating of the cooling air as it passes through the machine. In such cases a hybrid-cooling design may be employed, with air used to cool the electronic packages and water-cooled heat exchangers used to cool the air. For even higher power packages it may be necessary to employ indirect liquid cooling. This is usually done utilizing water-cooled cold plates on which heat dissipating components are mounted, or which may be mounted to modules containing integrated circuit chips. Ultimately, direct liquid immersion cooling may be employed to accommodate high heat fluxes and a high system heat load.

1.8 Air Cooling

Although liquid forced convection and boiling offer the highest heat transfer rates, air cooling has been and continues to be the most widely used technique for heat rejection. The principal advantages of cooling with air are its ready availability and ease of application. Prior to 1964, all computers were cooled solely by forced air. Air moving devices took in room air and provided a serial flow of air over columns of boards carrying printed circuit cards with single chip modules. In many cases air moving devices at either the bottom or top of a column of boards provided sufficient cooling air flow. A push–pull airflow arrangement with air moving devices at both the bottom and top of the column of boards was used for those cases requiring higher pressure drop capability. Forced air-cooled systems may be further subdivided into serial and parallel flow systems. In a serial flow system, the same air stream passes over successive rows of modules or boards, so that each row is cooled by air that has been preheated by the previous row. Depending on the power dissipated and the air flow rate, serial air flow can result in a substantial air temperature rise across the machine. The rise in cooling air temperature is directly reflected in increased circuit operating temperatures. This effect may be reduced by increasing the air flow rate. Of course, to do this requires larger blowers to provide the higher flow rate and overcome the increase in air flow pressure drop. Parallel air flow systems have been used to reduce the temperature rise in the cooling air. In systems of this type, the printed circuit boards or modules are all supplied air in parallel. Since each board or module is delivered its own fresh supply of cooling air, systems of this type typically require a higher total volumetric flow rate of air.

1.9 Hybrid Air-Water Cooling

An air-to-liquid hybrid cooling system offers a method to manage cooling air temperature in a system without resorting to a parallel configuration and higher air flow rates. In a system of this type, a water-cooled heat exchanger is placed in the heated air stream to extract heat and reduce the air temperature. The cooling system incorporated an airto-water finned tube heat exchanger between each successive row of circuit boards. The modules on the boards were still cooled by forced convection with air, however; the heated 17 air exiting a board passed through an air-to-water heat exchanger before passing over the next board. Approximately 50% of the heat transferred to air in the board columns was transferred to the cooling water. Ultimately air-to-liquid hybrid cooling offers the potential for a sealed, recirculation, and closed-cycle air-cooling system with total heat rejection of the heat load absorbed by the air to chilled water. Sealing the system offers additional advantages. It allows the use of more powerful blowers to deliver higher air flow rates with little or no impact on acoustics. In addition, the potential for electromagnetic emissions from air inlet/outlet openings in the computer frame is eliminated. Another variant of the hybrid cooling system is the liquid-to-air cooling system. In this system liquid is circulated in a sealed loop through a cold plate attached to an electronic module dissipating heat. The heat is then transported via the liquid stream to an air-cooled heat exchanger where it is rejected to ambient air. This scheme provides the performance advantages of indirect liquid cooling at the module level while retaining the advantages of air cooling at the system or box level. Most recently, a liquid-to-air cooling system is being used to cool the two processor modules in the Apple Power Mac G5 personal computer shipped earlier this year.

1.10 Liquid-Cooling Systems

Either the air-to-water heat exchangers in a hybrid air–water-cooled system or the water-cooled cold plates in a conduction-cooled system rely upon a controlled source of water in terms of pressure, flow rate, temperature, and chemistry. In order to ensure the physical integrity, performance, and long-term reliability of the cooling system, customer water is usually not run directly through the water-carrying components in electronic frames. This is because of the great variability that can exist in the quality of water available at computer installations throughout the world. Instead a pumping and heat exchange unit, sometimes called a coolant distribution unit (CDU) is used to control and distribute system cooling water to computer electronics frames. The primary closed loop (i.e., system) is used to circulate cooling water to and from the electronics frames. The system heat load is transferred to the secondary loop (i.e., customer water) via a water-to-water heat exchanger in the CDU. Within electronics frame a combination of parallel-series flow networks is used to distribute water flow to individual cold plates and heat exchangers. Water flow in the primary loop is provided at a fixed flow rate by a single operating pump, with a stand-by pump to provide uninterrupted operation if the operating pump fails. The temperature of the water in the primary loop is controlled by using a mixing valve to regulate the fraction of the flow allowed to pass through the water-to-water heat exchanger and forcing the remainder to bypass the heat exchanger. A CDU is also required for direct immersion cooling systems. In addition, because of the relatively high vapor pressure of the coolants suitable for direct immersion applications (e.g., fluorocarbons), the cooling system must be both "vapor-tight" and "liquid-tight" to ensure against any loss of the relatively expensive coolant.

1.11 **Refrigeration Cooled Systems**

The potential for enhancement of computer performance by operating at lower temperatures was recognized as long ago as the late 1960s and mid-1970s. Some of the earliest studies focused on Josephson devices operating at liquid helium temperatures (4K). The focus then shifted to CMOS devices operating near liquid nitrogen temperatures (77 K). A number of researchers have identified the electrical advantages of operating electronics all the way down to liquid nitrogen temperatures (77 K). In summary, the advantages are: · Increased average carrier drift velocities (even at high fields); · Steeper sub-threshold slope, plus reduced sub-threshold currents (channel leakages) which provide higher noise margins; · Higher transconductance; · Well-defined threshold voltage behaviour; · No degradation of geometry effects; · Enhanced electrical line conductivity; · Allowable current density limits increase dramatically (i.e., electro migration concerns diminish).

CHAPTER 2

LITERATURE REVIEW

The literature review is presented on the early works of a microchannel with different cross-sections and various base fluids such as water and nanofluids which exhibits various heat transfer rates. The manufacturing methods and their effects on the experimental results are also highlighted. The literature survey on microchannel reveals that there are a few important factors to be considered while evaluating the heat transfer and flow characteristics. One important factor is the flow distribution taking place in the microchannel that is the primary and the secondary flow which depends on the cross-section of the microchannel. The literature survey also discusses the different aspects of cross-sectional areas, fluids, and flow configuration over the microchannel. The numerical and experimental study on microchannel to determine the heat and flow characteristics are explored.

2.1 LITERATURE ON MICROCHANNEL MATERIALS AND GEOMETRY

In this literature survey, the microchannel is mainly concentrated based on the area of the cross-section and materials used which influence the performance of microchannel.

2.2 Influence of Microchannel Materials

The various materials used in fabricating the microchannel and their properties which results in significant heat transfer performance. 20 Wan et al. (2012) presented the flow and heat transfer analysis along with a porous microchannel. They concentrate on the Nusselt number and thermal resistance where the thermal resistance decreases with the increase in the heat flux that is being supplied. Xiang-feiYu et al. (2012) took a new concept in a fractal tree-like structure of microchannel heat sink. The rectangular sized microchannel heat sink was fabricated from three major materials such as silicon wafer, Pyrex glass and Polymethyl methacrylate (PMMA) for the base plate.

It was found that the vortexes generated to increase the heat transfer rate by comparing with the straight microchannel. Finally, they concluded that both hydraulic and thermal characteristics exhibited more significant impact on aspect ratio. The results denote that the concept of fractal structure is much more than that of straight fin microchannels.

Yan Fan et al. (2013) introduced an enveloping jacket-type cylindrical oblique fin mini-channel heat sink. It was fabricated in copper material for better heat transfer. It was pragmatic that the pressure drop in the channel was better than the straight fin mini-channel heat sink. They achieved negligible pressure drop penalty and higher heat transfer rate by decreasing the thermal boundary layer thickness which undergoes flow-mixing.

Navin Raja et al. (2013) investigated the heat transfer enhancements in the trapezoidal cross-section with laminar flow characteristics. They initially considered various base fluids like water, Ethylene Glycol (EG), oil and glycerin with nanosized diamond particles of 2% substrate materials like copper, aluminum, steel, and titanium. They conducted percentage enhancement test in trapezoidal cross-section, and they obtained 98.15% results in thermal resistance which were found to be the mixture of glycerin with diamond nanoparticles. It could naturally increase the cooling performance of the MCHS assisted by the high viscosity glycerin.

2.3Effect of Cross-Sectional Area

The cross-sectional effect on microchannel shows the numerical and experimental heat transfer variations due to change in dimensions and cross-sectional area.

Garcia-Hernando et al. (2009) performed an experimental analysis of the hydrodynamic and thermal performance of micro heat exchangers sized 100 x 100 and 200 x 200 µm square cross-sections with deionized water used as a working fluid. The obtained results were compared with the forecasts of the classical viscous flow and heat transfer theory

Melanie Derby et al. (2012) performed an experiment on condensation heat transfer. The comparison of the square, triangular, and semi-circular channel geometries with the same hydraulic diameter of 1 mm was made. From the parametric study, the mass flux, saturation pressure, and heat flux were found to have significant effects on condensation whereas the channel shape had no significant impact on condensation.

Jingru Zhang et al. (2013) investigated on the fluid flow to remove heat from two different cross-sections, a straight and a U-shaped channel. Both are made of silicon material as a substrate with the microchannel base plate made out of polydimethyl siloxane (PDMS). They found that the thermal resistance and pressure drop are better in U-shaped channel than the straight and also the heat transfer rate is better in the U-shaped channel.

Ngoctan Tran et al. (2015) heat Transfer of aluminum microchannel heat sinks with different channel depths by investigated numerically and experimentally. The channel width of 500 µm, the channel length of 33 mm, and channel depth varying from 200 µ m to 900 µ m was considered.

Maximum heat transfer rate of 143.8 W was achieved for the channel depth of 900µm at the mass flow rate of 213 g/min. The experiment confirmed that the heat transfer increases with increase in mass flow rate and a decrease in channel depth.

Assel Sakanova et al. (2015) performed a study with the new shape of the microchannel. The study dealt with the unique shape of the microchannel by bringing out the wavy type of microchannel with 25µm, 50µm, and 75µm amplitude. The wavy structure was further investigated by varying the volume flow rate with three different concentrations of nanofluid. The performance was increased by increasing the amplitude and decreased wavelength which provides low thermal resistance and comparable pressure drop. However, the replacement of water fluid with Nanofluid effect is not affected by the wavy channels

Lei Chai et al. (2016) studied numerically the heat transfer in microchannel heat sink with offset ribs on sidewalls considering laminar flow. The different offset ribs used were rectangular, backward triangular, forward triangular, semi circular and isosceles triangular. It was smaller velocity difference in y-direction which caused better mixing of hot water near the wall and the cold water in the center of the microchannel. Heat sink with forwarding triangular, semi-circular and isosceles triangular offset ribs had slightly better heat transfer performance than the heat sink with rectangular and backward triangular ribs.

Also, a heat sink with semi-circular and isosceles triangular showed the least pressure drop. They also investigated the laminar flow and heat transfer characteristics in the interrupted microchannel heat sink with ribs in the transverse microchannels. The different rib configurations in the experiment were rectangular, backward triangular, diamond, forward triangular and ellipsoidal.They found that periodic thermal layer developing 23 flow was responsible for the significant heat transfer enhancement. Also, study the heat transfer characteristics of silicon microchannel with fan-shaped ribs .The obtained results show that the height and spacing of fan-shaped ribs have more heat transfer characteristics than width variation. The increase in height and decrease in the spacing of fan-shaped ribs significantly improve the local and average heat transfer coefficient.

Anurag et al. (2016) proposed a new analysis of different inlet and outlet flow arrangements. The cross sections taken were rectangular, rectangular with semicircle and divergent-convergent. All of these were microchannels are made up of copper material. The CFD analysis found that the divergent-convergent had better heat transfer than the other two cross sections. Comparing those two, the rectangular with semicircle had better heat transfer than a rectangular section.

2.4 Influence of Flow Configuration on Microchannel

Influence of flow configuration on microchannel shows the difference in pressure drop and the characteristic of nanofluids in various flow distributions.

Mushtaq I Hasan (2011) studied heat transfer characteristics of Microencapsulated Phase Change Material (MEPCM) suspension in the counter flow microchannel heat exchanger. The MEPCM consists of microcapsules as a phase change material, shell material, and the capsules are concentrated in water (0-20%). Finally, in thermal point of view the MEPCM 24 leads to increase the effectiveness, but in hydrodynamic point of view, it increases the pressure drop

Pawan K Singh et al. (2012) investigated experimentally and numerically the hydrodynamics of nanofluids in the microchannel. Alumina nanofluid particles of water and ethylene glycol were allowed to pass through three microchannels. The transition to turbulence occurred at 211 and 300μm channels due to high surface roughness. The result confirmed that nonuniform distribution of particles due to shear-induced migration was the main reason behind the differential behaviour of nanofluids.

Mohammad Kalteh et al. (2012) investigated experimentally and numerically nanofluid forced convection inside a wide microchannel heat sink. Alumina was used as the nanofluid and water as the base fluid in a wide rectangular heat sink. The two-phase method was found to be more appropriate than the homogeneous method to simulate the nanofluid heat transfer.

Matthew D Byrne et al. (2012) conducted an experiment varying concentration values to the volume of CuO and the second factor was the use of surfactant CTAB as suspension enhancer. In this experiment, the hydrodynamic and thermal performances of seven fluids were determined. The microchannel heat transfer evaluation showed an increase in the performance of pure water to 10% by adding a surfactant and a particular increase of smaller concentration values. Even the surfactant also caused smaller improvements of heat transfer. So it provided a dispersion of nanoparticles in the fluids. The absence of a suspension, enhancer led to agglomeration and settling.

Mat Tokit et al. (2012) investigated the optimized interrupted microchannel for the thermal performance. Interrupted Microchannel Heats Sink (IMCHS) using nanofluid as working fluids were analyzed numerically to increase the heat transfer rate. The results were found to be 1.23% to 0.34% of penalty on the pressure drop in the IMCHS for Reynolds number ranging from 140 to 1034.

Mohammad Kalteh et al. (2012) found that the Nusselt number and nanofluid concentration are some of the important characteristics that influence the heat transfer. There is a significant figure of 12.61% for water and 7.42% in the 0.1% volume concentration alumina-water nanofluid for the deviation between experimental and numerical.

Selvakumar, P & Suresh (2012) did a study on the impact of CuO/water nanofluids in a thin copper channel heat sink with constant heat flux condition with volume fractions 0.1% and 0.2%. The reduction of 1.15oC was measured as the interface temperature of the water block. As a result, the increase in heat transfer coefficient was determined as 29.63% of the nanofluid having a concentration of 0.2% compared with the deionized water for the same flow rate.

Tu-Chieh Hung et al. (2012) numerically investigated the laminar forced convection in the partially porous channel using different porous blocks such as rectangular outlet enlargement, trapezoidal, thin rectangular, block and sandwich distribution. The sandwich distribution had the lowest pressure drop, and most of the thermal performance was improved. It was found that the sandwich and trapezoidal designs have best heat transfer efficiency and coding performance. This is due to the factor pumping power would not overcome the larger pressure drop induced by lower pumping powers

Mohammed et al. (2013) focused on different nanofluids and compared them with various other working fluids. The paper described the inlet cross-section with the trapezoidal being utilized in the experiment and Numerical analysis.

Navin Raja Kuppusamy et al. (2013) investigated numerically thermal and flowed fields in a Trapezoidal Grooved Microchannel Heat Sink (TGMCHS) using nanofluids. Here Al2O3/H2O was used as the nanofluids which showed better thermal performance because the thermal conductivity was more dominant than the heat capacity for thermal enhancement in MCHS. The increment of the maximum width (a) and reduction of the minimum width (b) of the trapezoidal groove gave the maximum thermal performance.

Manu Mital (2013) attempted to enhance the heat transfer without taking the pumping power into consideration.T hey found smaller the nanofluid particle size, higher its heat transfer rate. Its volume fraction was kept between 1% and 7%, and this factor of ratio defined heat transfer coefficient of nanofluid and to water keeping the pumping power constant. In this study, using a lower level, Reynolds number often predicted higher results of improvement in heat transfer rate.

Yue-Tzu Yang et al. (2014) executed the numerical study of flow and heat transfer characteristics of alumina-water nanofluids in a microchannel. The simulation was done on the basis of the lattice Boltzmann method . The simulations were conducted at low Reynolds number ($Re \leqq 16$). They observed the fluid temperature distribution was uniform with the use of nanofluid than that of pure water.

Benjamin Herrmann-Presents et al. (2016) studied numerical simulation of cooling microchannel through the spiral radial flow of fluid. The factors such as height, inlet angle, and fluid were varied to study the effects. The rotational flow of fluid induced the fluid to move over the heat exchanging surface. When the microchannel height was reduced, it resulted in the merging of boundary layers and entrainment effect.

Si-Ning Li et al. (2017) presented a 3D numerical simulation of the flow characteristics and heat transfer effect on non-Newtonian fluid flow in a traditional microchannel heat sink and manifold microchannel heat sink. The effects of non-Newtonian fluid on heat transfer performance included the uniformity of temperature distribution and heat transfer enhancement, as well as the influence of inlet /outlet configurations on heat transfer and fluid flow. The heat transfer performance in the form of pseudo-plastic fluid flow significantly improved the heat transfer enhancement holding to the generation of secondary flow due to the shear thinning property. As a result, the temperature distribution was uniform on MMC while using pseudo-plastic fluid whereas the maximum temperature shown in the study was subjected to be in the corner near the inlet and outlet.

Sung-Min Kim & Issam Mudawar (2017) focused on the thermal design of MCHS with an inlet in saturated conditions. Here pressure drop and heat transfer coefficient were calculated using predictive methods. The predictive tools such as dry out incipience and premature heat flux and twophase critical flow limited considerable heat sink performance. A parametric study was conducted based on these tools to track the change of maximum heat flux corresponding to volumetric flow rate. Using the heat transfer coefficient pressure drop and maximum bottom wall temperature was evaluated. Deep microchannel increased the heat flux and decreases pressure drop and producing an unfavorable effect on the bottom wall temperature.

To improve the heat transfer still more efficient, the nanoparticles came into the field of thermal due to the better heat transfer rate. Thus there were a lot of studies based on nanoparticles used in fluid

2.5 Literature on fluid flow and heat transfer in microchannels

Single Channel Experiments

Wu and Little (1983) fabricated trapezoidal channels on glass and silicon and conducted experiments for friction factors using nitrogen as the fluid. They found that the friction factors for glass channels are 3 to 5 times larger than smooth pipe predictions. They observed that the transition from laminar to turbulent occurs at Re of 400. For the case of smooth channel the theoretical values in both the laminar and turbulent regimes are in good agreement and the transition from laminar to turbulent in this case was found to occur at Re of 2000. From this it is understood that, for single microchannel the friction factor and transition Re values are not different from that of conventional channels, provided that the channel is smooth enough. These values are found to vary for rough channels because of the large relative roughness. They also provided correlations for friction factor for different range of Reynolds number.

Wu and Little (1984) conducted heat transfer experiments with the same setup as discussed above. They found that, in turbulent regime the heat source did not affect the heat transfer. At the same time, in laminar regime the position of heat source appeared to have some effect. The roughened channels enhanced the heat transfer coefficient, but also increased the friction. The Reynolds analogy is not valid for the rough channels tested by them. Based on heat transfer study, they suggested a Nusselt number correlation for Reynolds number less than 3000.

Pfahler et al (1989) conducted experiments on rectangular silicon microchannels using npropanal and found that channels with larger cross sectional areas showed better agreement with theoretical predictions for the 15 friction factor. The authors identified that there appears to be a critical channel size where the general behavior of the experimental observations deviates from the NavierñStokes (N-S) predictions. They concluded that there are insufficient data to state that the discrepancy results from the breakdown of the N-S equations, but their results certainly warranted additional and more precise investigations. They also proposed C value through graph between C and Reynolds Number, which can be used in determining friction factor in the laminar region using f = C/Re. They showed that for channels with relatively large cross-sections, the fluid roughly behaves in accordance with the N-S predictions

Choi et al (1991) conducted experiments on friction factor and heat transfer in silica microtubes using nitrogen and suggested correlations for friction factor and Nusselt number for both laminar and turbulent region.

Harms et al (1997) conducted experimental and theoretical study in a silicon rectangular microchannel using de-ionized water. They identified the critical Reynolds number as 1500.

Their analysis showed that for the same pressure drop and pumping power, thermal resistance was smaller for deeper channels.

Xu et al (2000) carried out experiments in microchannels with hydraulic diameter varied in the range 30 µm to 344 µm at Reynolds number ranging from 20 to 4000. Results obtained showed that characteristics of flow in microchannels agree with conventional behaviors predicted by NavierStokes equations for the range investigated. They opined that the deviations found in the literature could be due to dimensional errors. They recommended that the microchannels etched on silicon wafer and covered with pyrex glass by anodic bonding are suitable for conducting flow characteristics experiments.

Liu and Garimella (2002) carried out experiments to examine the validity of conventional theory in predicting flow through microchannels. They found that the surface roughness did not play any role. Two types of microchannel test sections are fabricated. For the short microchannel the pressure taps were placed at the inlet and outlet header, therefore losses in the inlet and the exit header should be taken in account, while calculating the friction factor. For the long microchannel the pressure taps are machined along the length of the microchannel far away from the headers. They found that the experimental results of long microchannel agree with the theoretical predictions. Also, it was observed that the flow transition takes for a Reynolds number of 2000 confirming to the conventional critical Reynolds number. They strongly put forward that in microchannels there is no possibility of early flow transition. This is because in microchannels the characteristic length scale of the flow is smaller than Kolmogorov length scale and the flow field will be dominated by viscous stresses which will suppress the turbulence. They also verified their experiments against numerical predictions and suggested that conventional CFD analysis can be used in flow maldistribution

Kohl et al (2005) conducted experiments to investigate discrepancies in previously published data for the pressure drop in microchannels. Straight channel test sections with integrated pressure sensors were developed for channel hydraulic diameters ranging from 25 to 100 µm. Compressible flow results using air as the working fluid, for Reynolds number between 6.8 and 18,814 and incompressible flow results with water for Reynolds number between 4.9 and 2068 have been obtained. The result suggested that friction factors for microchannels can be accurately determined from data for standard large channels. They concluded that large inconsistencies in previously published data are probably due to instrumentation errors and/or improper accounting for compressibility effects.

Steinke and Kandlikar (2005) reviewed nearly 150 papers on single phase liquid friction factors in microchannels to identify the discrepancies in reported literature. A database of over 5,000 data points has been generated. Their work includes a Reynolds number range of $0.002 < Re < 5,000$ and a hydraulic diameter range of $8 < Dh < 990$ mm. According to the authors the work which does not consider the entrance and exit losses or the developing flow in the microchannel, are the ones, which deviate from the theory. The remaining data sets showed agreement with conventional theory. They also demonstrated that the uncertainty in Poisuellie number is dominated by the microchannel width and height. They emphasized that even a very accurate pressure drop measurement will often be overshadowed by the geometry measurement uncertainties. The components that contribute to the total pressure drop occurring across the microchannel heat exchanger are identified as the inlet and outlet losses, the developing flow losses and the frictional flow losses. They generated a new experimental setup to demonstrate the procedure for correcting the measured pressure drop. The corrected data showed good preliminary agreement with the conventional theory for fluid flow with some errors remaining that could be contributed to the simplistic treatment of inlet and outlet losses, using conventional restriction and expansion area constants derived empirically for large diameter channels.

Vijayalakshmi et al (2009) conducted detailed experimental study in long microchannel of hydraulic diameter ranging from 60.5 to 211 μm. They could clearly differentiate the incompressible and compressible nature of flow in microchannels. They found that the developing length is substantial at higher Reynolds numbers, when water is used as the fluid. The authors insisted that the pressure loss due to flow development is to be considered along with the inlet and exit pressure losses, in calculating the friction factor. For gas flow through the channel, it is observed that when the channel exit 18 Mach number approaches a value of 0.3, the flow is no longer incompressible, which conforms the conventional theory and hence the incompressible correlations could not be used. For a Mach number less 0.3 at the channel exit, both nitrogen and water exhibited incompressible flow behaviour, with Poisuellie number of 53. As the channel diameter reduces, compressibility effects appear much earlier due to flow acceleration resulting from large pressure drops, at the same time transition to turbulence occurs in the Reynolds number range 1600 to 2300, which is slightly lower than the conventional range of 2000 to 3500. This is due to the relative roughness or the edge effect. It is found that the deviation of Poisuellie number from a constant value is not an indicator of onset of turbulence. The deviation of slope of the friction factor vs Reynolds number curve and the curve becoming parallel to Blasius solution are suggested as the better indicator of onset of turbulence. According to them the deviation from constancy of Poisuellie number in most cases is an indicator of onset of compressible flow. They also suggested that an axi-symmetric model can be used to analyse the pressure profile all along the length of the microchannel including port region. They concluded that the conventional theory based on Navier - Stokes equations with viscous dissipation term and no slip boundary conditions at the wall appear to be appropriate for the range of hydraulic diameters and Reynolds number considered.

2.6 Multiple Channel Heat Sink

Tuckerman and Pease (1981) demonstrated in the early 80's that the ultra high speed VLSI circuits can be cooled very effectively using a series of microchannels 50 µm wide and 300 µm high in the back of the substrate and covering them with a plate to confine the fluid flow within the channels. They were able to dissipate 790 W/cm2 , with a corresponding substrate temperature rise of 71 oC above the inlet water temperature. They measured a thermal 19 resistance of 0.090 oC/W. They also suggested that the lower limit of the channel size is set by the viscosity of the fluid used. Tuckerman and Pease (1982) discussed the details of microchannel fabrication and implementation details. In their paper, the selection of coolant, packaging and headering, micro structure selection, fabrication and bonding issues are discussed. The fabrication processes, etching and precision sawing are compared. The expression for Coolant Figure of Merit (CFOM) is provided. They also tested the use of micro pillars and found that micro pillars yield slightly lower heat transfer coefficient when compared to the micro fins, but the micro pillars possess the advantage of being less vulnerable to clogging. Micro pillars are also preferable in cooling of local hot spots.

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Mahalingam and Andrews (1987) fabricated rectangular microchannel heat sink and used air as the working fluid. They established a linear relationship between the pressure drop and air flow rate for the narrow channel heat sink. They calculated the pressure drop for air flow by using the incompressible flow analysis. They predicted a lesser pressure drop than the experimental value due to fact that the measured value included entry and exit losses. They compared the performance of their heat sink with the conventional heat sinks based on correlations and found that microchannel heat sinks are attractive compared to the conventional air circulation heat sinks. They noticed that the convection coefficient for the tangential flow silicon heat sink (parallel channel) is 13 mW/cm2 oC which is less than the convection coefficient (26 mW/cm2 oC) of impingement flow silicon heat sink (micro pins). They attributed the same to the flow maldistribution in the tangential flow silicon heat sink.

Phillips et al (1987) considered rectangular channels in silicon and used water as coolant. They performed theoretical modeling for fully 20 developed and developing flows and concluded that turbulent flow designs showed equivalent or better performance compared to laminar flow designs. Rahman and Gui (1993) conducted experiments in rectangular channels etched on silicon and found that Nusselt numbers are higher than those predicted from analytical solutions for developing laminar flow.

Peng and Wang (1993) conducted single phase forced convection experiments on stainless steel, rectangular microchannels using de-ionized Water. They observed a steep increase in wall heat flux with the wall temperature in single phase convection. They observed that the heat flux for microchannels is higher than that for normal size tube.

Wang and Peng (1994) conducted forced convection flow and heat transfer experiments in rectangular stainless steel microchannel heat sink. They noticed that the fully developed turbulent regime starts at Re of 1000 to 1500. They also observed that the heat transfer is augmented as liquid temperature was reduced and as liquid velocity was increased.

Peng et al (1994) conducted experiments on frictional behaviour in stainless steel, rectangular microchannel heat sink. They noticed the early transition occurred at Reynolds number between 200 and 700 and proposed correlations for friction factor for both laminar and turbulent flow.

Peng et al (1994 a) conducted heat transfer experiments on the same heat sink used as above. They found that fully turbulent convective conditions reached at Re of 400 to 1500. They also found that transition Reynolds Number diminished with a reduction in microchannel dimension. Peng and Peterson (1995) conducted experiments on effect of thermofluid properties and geometry on convective heat transfer in rectangular 21 stainless steel microchannel heat sink. They found that the transition zone and heat transfer characteristics in laminar and transition flow are influenced by liquid temperature, velocity, Re number and microchannel size.

Peng and Peterson (1996) conducted experiments on single phase flow and heat transfer. They plotted ratio of experimental to theoretical friction factor at critical Reynolds number as a function of Z (where Z = min [dc , wc]/max [dc , wc]. They also proposed Nusselt correlation in laminar and turbulent regions.

Peng and Peterson (1996 a) conducted experiments in a stainless steel rectangular microchannel heat sink using water-ethanol mixture. The laminar heat transfer was found to be ceased for Re of 70 to 400 depending on flow conditions. The fully developed turbulent heat transfer achieved at Re of 200 to 700 depending on Dh. Transition Re is found to be reduced with a reduction in microchannel size. Cuta et al (1996) conducted both single phase and two phase experiments in a microchannel heat exchanger with 54 parallel microchannels with rectangular cross section using Refrigerant 124 as the working fluid. The range of Reynolds number considered was 100 to 570. They found that the average friction coefficient across the channels is smaller than the expected conventional single phase data. The average liquid side heat transfer coefficient showed a significant increase over the expected value in macroscopic flow at the same Reynolds numbers. Single phase tests, for Nusselt number values ranging from 5 to 12 indicated that Nusselt number increases with Reynolds number. They concluded that though the data is considerably scattered, higher heat transfer coefficients can be obtained. Zhuang et al (1997) conducted experiments in impingement on 2-D rectangular microchannels. They suggested new empirical correlation for Nusselt number for the two liquids. Adams et al (1998) conducted turbulent single phase flow experiments in a circular microchannel made of copper using water as the operating fluid. The flow passage was produced using the electrode discharge machining technique. The entry length is sufficiently long to assure fully developed conditions at the heated section inlet. The experimental Nusselt numbers are higher than those predicted by large channel correlations. They modified Gnielenski correlation for Nusselt number in circular microchannels. Heat transfer coefficients and Nusselt numbers for water flowing through circular channels of diameters 0.76 and 1.09 mm were experimentally determined and found to be higher than the predicted values. The data suggested that the extent of enhancement (deviation) increases as the channel diameter decreases and Reynolds number increases.

Mohiuddin and Li (1999) experimentally investigated water flow through fused silica (FS) and stainless steel (SS) based microtubes for diameters ranging from 50 to 254 μm. The experimental results indicated significant departure of flow characteristics from the predictions of the conventional theory for microtubes with smaller diameters. On the other hand for microtubes with large diameters, the experimental results are in rough agreement with the conventional theory. For lower Re, the required pressure drop is approximately same as predicted by the Poiseuille flow theory. But, as Re increases, there is a significant increase in pressure gradient compared to that predicted by the Poiseuille flow theory. For the fixed flow rate and diameter, an FS microtube requires a higher pressure gradient than a stainless steel microtube, which may be due to either an early transition from laminar 23 flow to turbulent flow or the effects of surface roughness of the microtubes. A roughness-viscosity model is proposed to interpret the experimental data.

Harms et al (1999) carried out experiments using two different configurations, a single channel and a multi-channel setup. They concluded that for fully developed laminar flow the thermal resistance is independent of pressure drop. An inverse relationship between pressure drop and thermal resistance is noted for developing laminar flow. Their result indicate that decreasing the channel width and increasing channel depth provide better flow and heat transfer. Importantly they found that the transition in multiple channels occurs at a Reynolds number of 1400 which is lower than the conventional transitional Reynolds number and they also noted that the friction factors for low flow rates are lower than the predicted ones. They concluded that both of this result reflects the importance of manifold design in microchannel systems. The Nusselt number in multiple channels strongly deviated from theory which was attributed to the flow bypass in the manifold. With these they strongly felt that the classical relations for local Nusselt number are appropriate and accurate for modeling microchannel systems, given the right manifold design.

Adams et al (1999) performed water-based experiments on turbulent convection in non-circular microchannel. They found that experimental Nusselt number is well predicted by Gneilenski Nusselt number. They proposed Dh of 1.2 mm as the reasonable lower limit for applicability of standard Nusselt type correlations to non-circular channels.

Tso and Mahulikar (1998) conducted dimensional analysis of the variables influencing forced convection, based on the experimental data from the literature for rectangular channels with laminar and transitional flow. According to them the Nusselt number increases with Reynolds number, if the 24 increase in Reynolds number is brought by increase in velocity. On the other hand, if the increase in Reynolds number is caused by decrease in viscosity caused by viscous heating, then the Nusselt number will decrease. They observed that the Nusselt Number may decrease with increasing Reynolds Number and may remain unaffected in transition regime. They proposed that the Brinkman number may better correlate convective heat transfer in microchannels. With the help of elementary dimensional analysis using inputs from the survey, they showed that in addition to Reynolds number and Prandtl number, a dimensionless geometric parameter of the microchannels, Nusselt number may correlate with Brinkman number.

Tso and Mahulikar (1999) found from the experimental data in the literature, that there are unexplained unusual behaviours in heat transfer and flow in microchannels. They felt that the geometry of the microchannels and Reynolds number alone do not determine the flow regime boundaries and the transition range. They processed the reported experimental data on convection in microchannels based on the correlation with the Brinkman number, from which they obtained the Reynolds and Brinkman numbers at the flow transition points for fixed microchannel geometry. They discovered that in addition to the Reynolds number, the Brinkman number is also needed to determine the flow regime boundaries from laminar to transition and from transition to turbulent flow. The transition range is found to vary due to the difference in the extent of the role played by the Brinkman number in determining the two flow regime boundaries.

Tso and Mahulikar (2000) performed experiments using water flow in microchannel specimens, with thermocouples mounted axially along the flow for local wall temperature measurement. The data obtained exhibited unusual behaviour of the local Nusselt number decreasing with increasing local Reynolds number along the flow. Single-phase forced convective heat transfer 25 in the laminar regime in microchannels correlated well with the Brinkman number even when the experimental data is obtained locally and along the flow, for the constant wall heat flux boundary condition.

Rahman (2000) presented new experimental measurements for pressure drop and heat transfer coefficient in microchannel heat sinks. Tests were performed with devices fabricated using standard silicon 100 wafers. Two different channel patterns were studied. The parallel pattern distributes the fluid through several parallel passages between the inlet and the outlet headers located at two ends of the wafer. The series pattern carried the fluid through a longer winding channel between the inlet and the outlet headers. Channels of different depths were studied. Tests were carried out using water as the working fluid. They found that the Nusselt number is larger near the entrance than the exit, which was attributed to the developing flow near the entrance. They observed that the measured value of Nusselt number is larger for microchannels when compared to that of conventional channels, which is attributed to breaking of boundary layer due to relatively larger roughness. They noticed that the transition from laminar to turbulent takes place smoothly because of smaller size of channels.

CHAPTER 3

3.1 Work Objectives

1. Comparative study of micro channel with vortex generators and plain micro channel on the basis of Nusselt number, pressure drop and overall thermal performance.

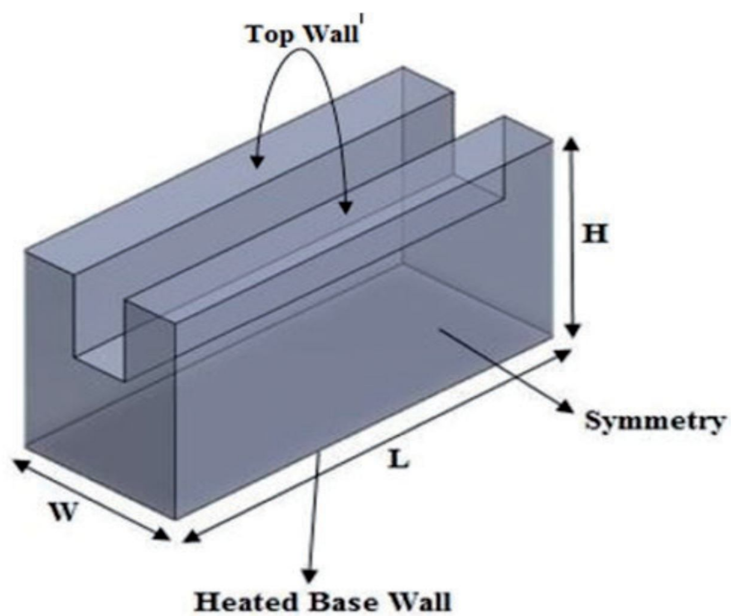
2. Compare the effect of angle of vortex generator on the Nusselt number, pressure drop and overall thermal performance

3. Analysis of change of value of Nusselt number with the change of Reynolds number.

4. Analyze the effect of vortex generators on the overall performance of microchannel heatsink.

3.2 Computational Modelling

The base model of rectangular microchannel are taken from the exploratory work of Nawaz and Karimi, who construct the microchannel with 200µm width and 300 µm height by etching the copper base. Deionized ultra-filtered water is used as a fluid in the microchannel. Figure shows the isometric view of rectangular microchannel with length (10 mm), A uniform heat flux of 100 w/cm2 is applied on the heated base wall and the side walls of rectangular microchannel are symmetric.



3.2 Assumptions:

Following are the assumptions which are to be considered for the present analysis

- Laminar and steady flow throughout the microchannel.
- Single phase flow.
- At walls, no slip boundary conditions are considered.
- At inlet developing flow is assumed.
- Adiabatic boundary conditions are assumed for the top walls of substrate and fluid.

Thermo-physical property of substrate and fluid remains constant

3.3 Boundary Conditions

The set of governing equations is solved using Ansys Fluent by adopting the appropriate boundary conditions at various positions throughout the domain

At z = 0 (inlet),

- Constant fluid temperature, $T = T_{wf,in} = 288\ K$
- Constant fluid velocity, $u = u_{wf,in},\ v = w = 0$

At z = L (outlet),

- Atmospheric Pressure, $P_{wf,out=}\ P_{atm}$

At y = 0 (heated base wall),

a constant flux of 10^6 W/m^2 is provided.

3.4 CASE 1: Angle of vortex Generator is 30°

Program Case 1

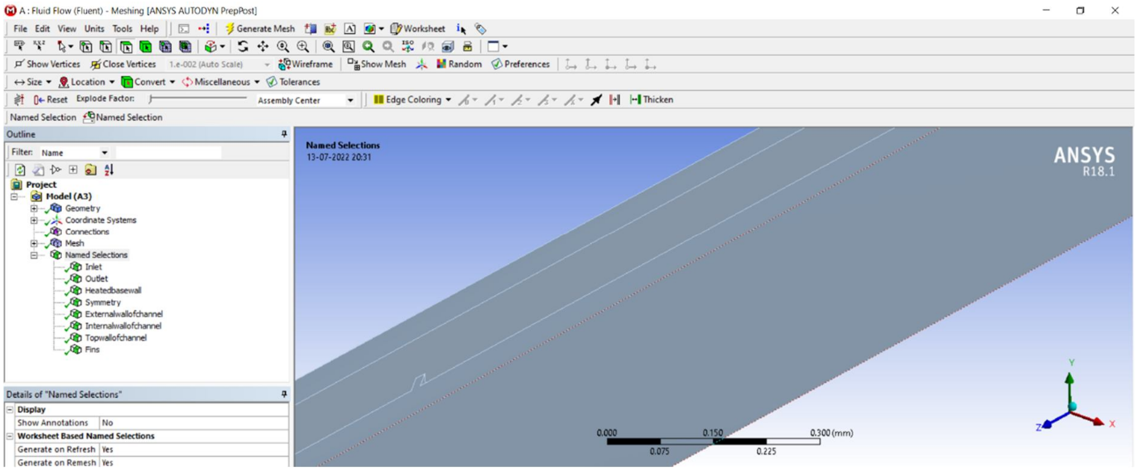


Figure 3.1

Geometry Case 1

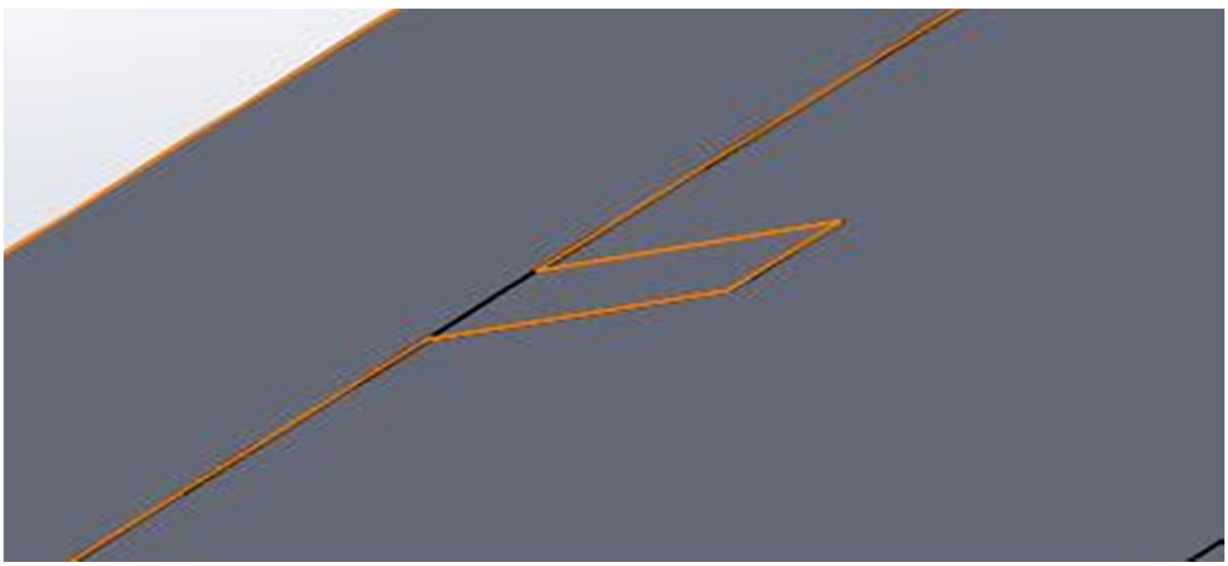


Figure 3.2

Mesh Case 1

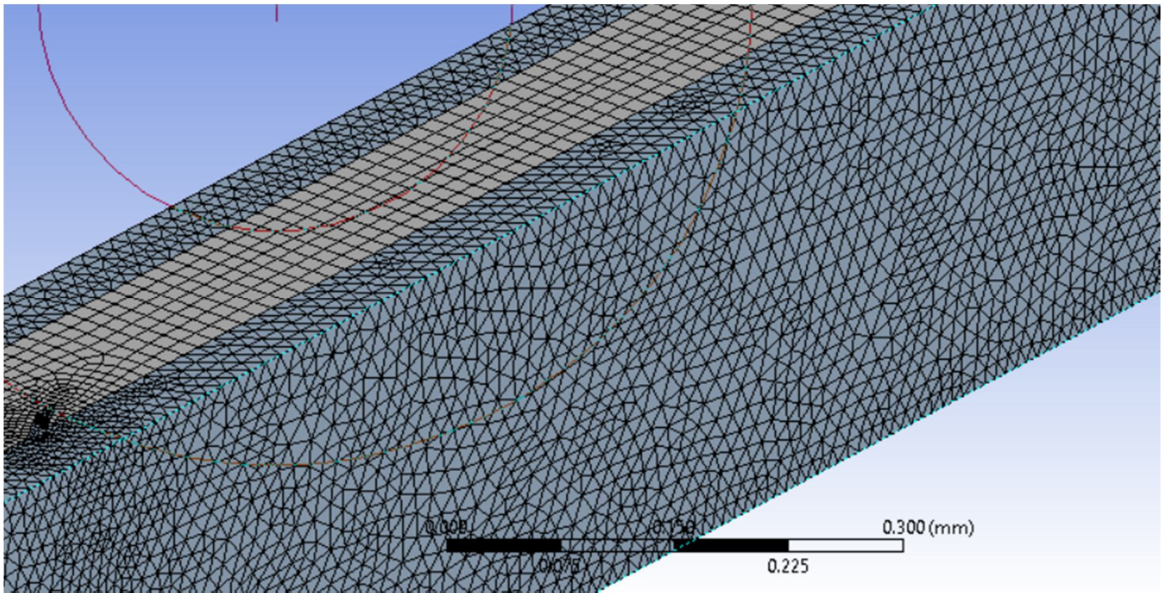


Figure 3.3

In this case the vortex generator placed at 30 degrees to the direction of inlet providing restricted passage to the flow of fluid in creating vortex and lateral shifting of stream lines which enhance the rate of heat transfer and hence greater value of Nusselt number

Also, the pressure drop is more as compared to the case of plain channel because of the restricted passage due to presence of vortex generator which reduces the areas of flow.

Over all thermal performance is more than one when the value of Reynolds number is 250, 300, 350 which reflects that the effectiveness of vortex generator is good when the velocity is high

3.5 CASE 2: Angle of vortex Generator is 45°

Program Case 2

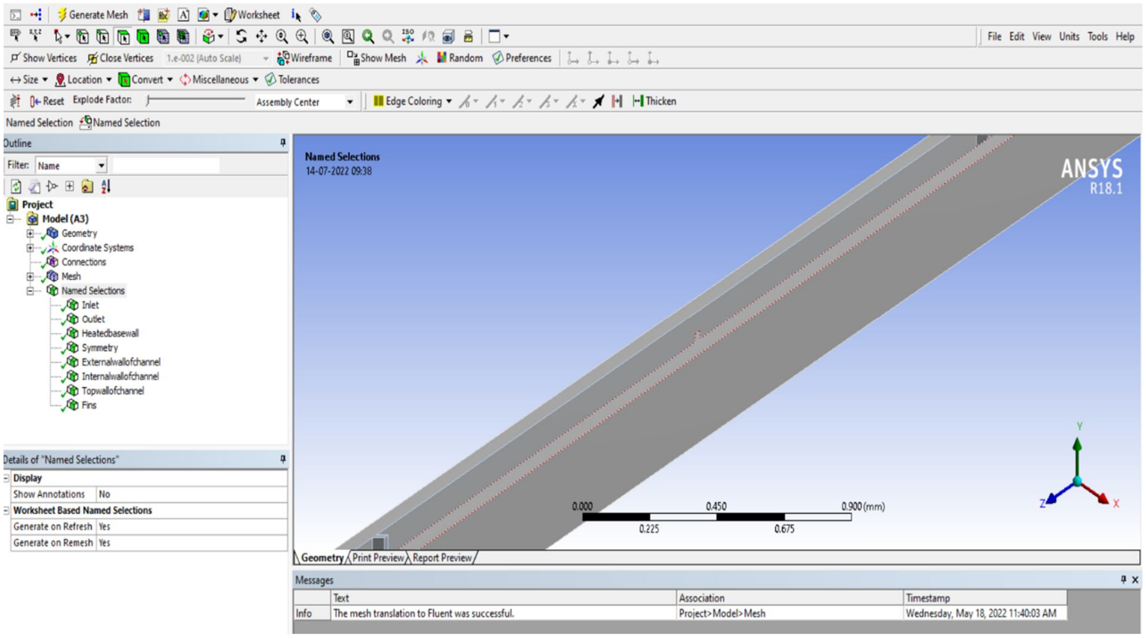


Figure 3.4

Geometry case 2

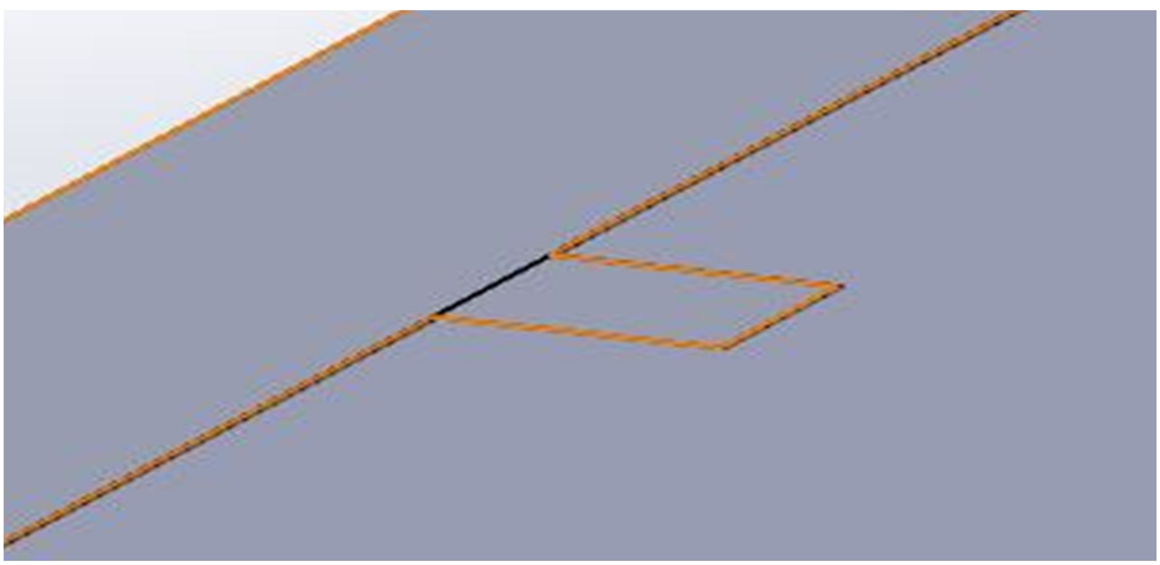


Figure 3.5

Mesh case 2

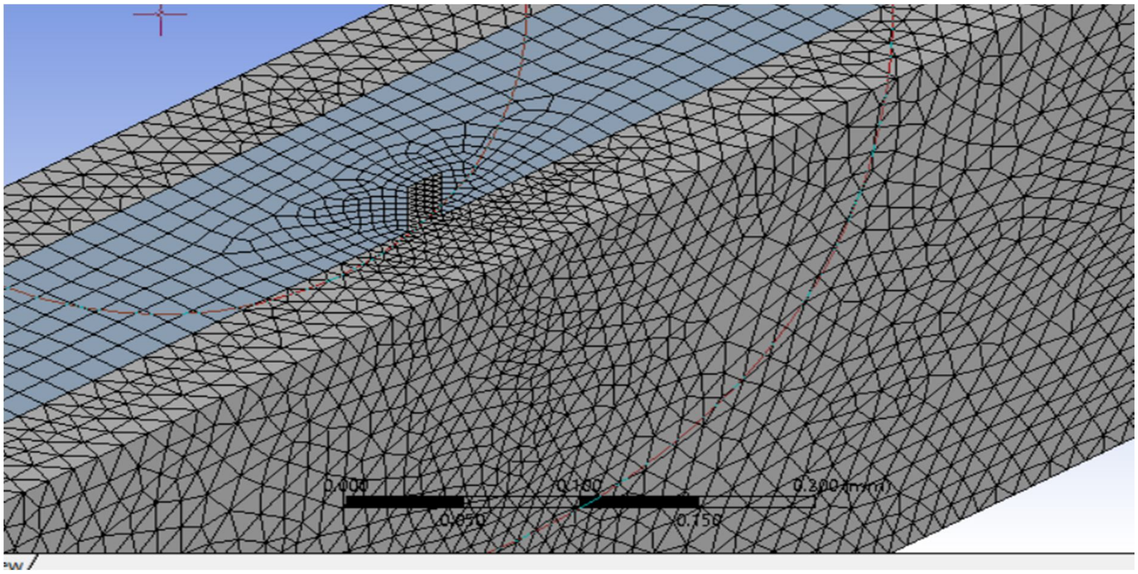


Figure 3.6

In this case the vortex generator placed at 45 degrees to the direction of inlet providing more restricted passage to the flow of fluid as compared to 30 degrees in creating vortex and lateral shifting of stream lines and vortex generation is more in this case which enhance the rate of heat transfer and hence greater value of Nusselt number which ranges from 7.6 to 9.78 for different values of Reynolds number such as 150, 250, 350

Also, the pressure drop is more as compared to the case of plain channel and case of 30 degree because of the restricted passage due to presence of vortex generator which reduces the areas of flow.

Over all thermal performance is more than one when the value of Reynolds number is 200, 250, 300, 350 which reflects that the effectiveness of vortex generator is good when the velocity is high

3.6 CASE 3: Angle of vortex Generator is 60°

Program case 3

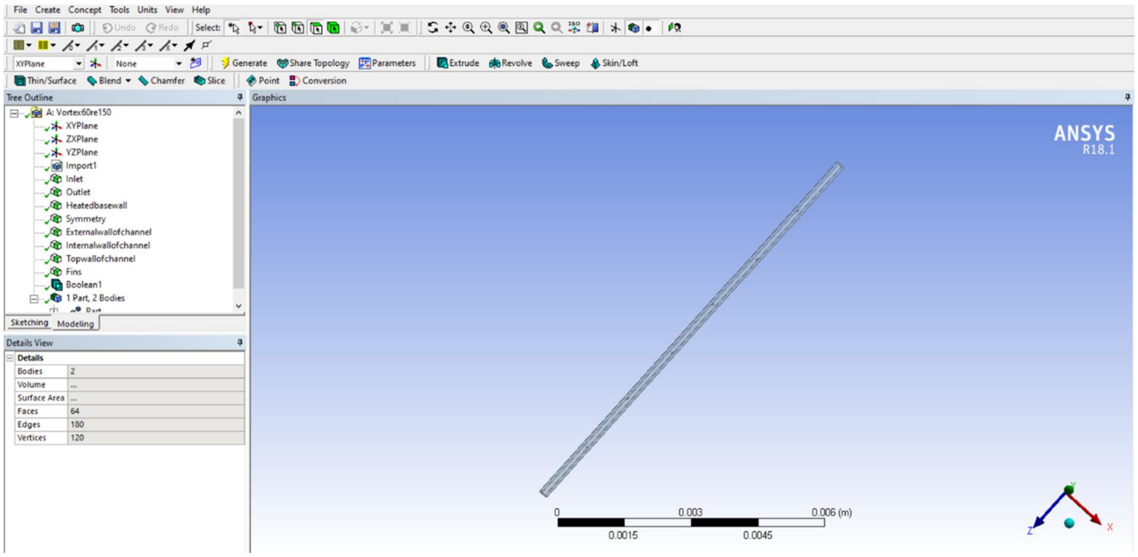


Figure 3.7

Geometry case 3

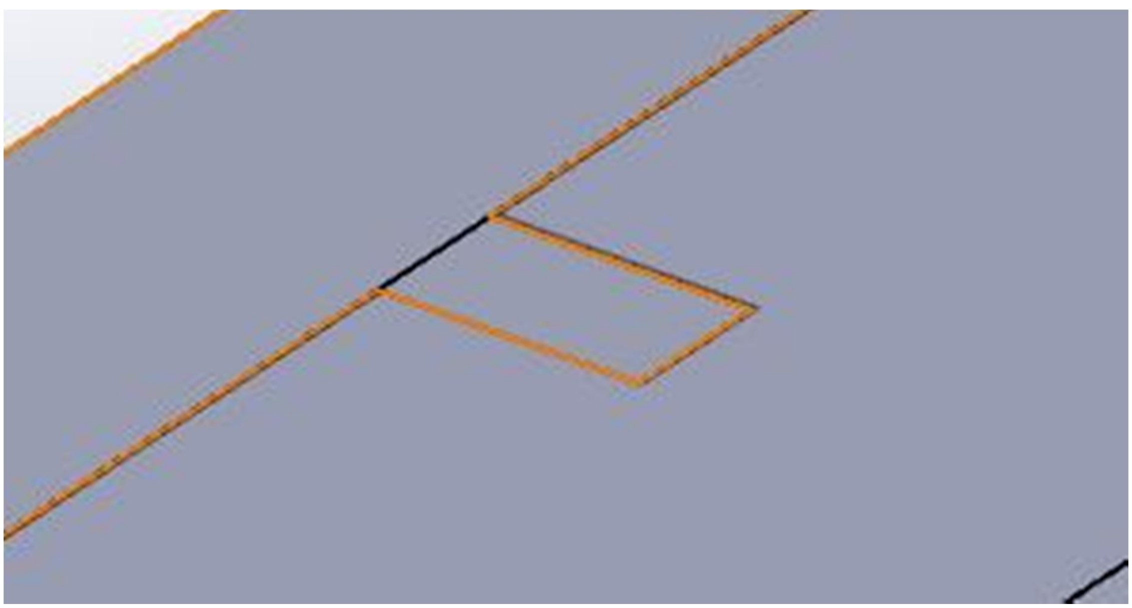


Figure 3.8

Mesh Case 3

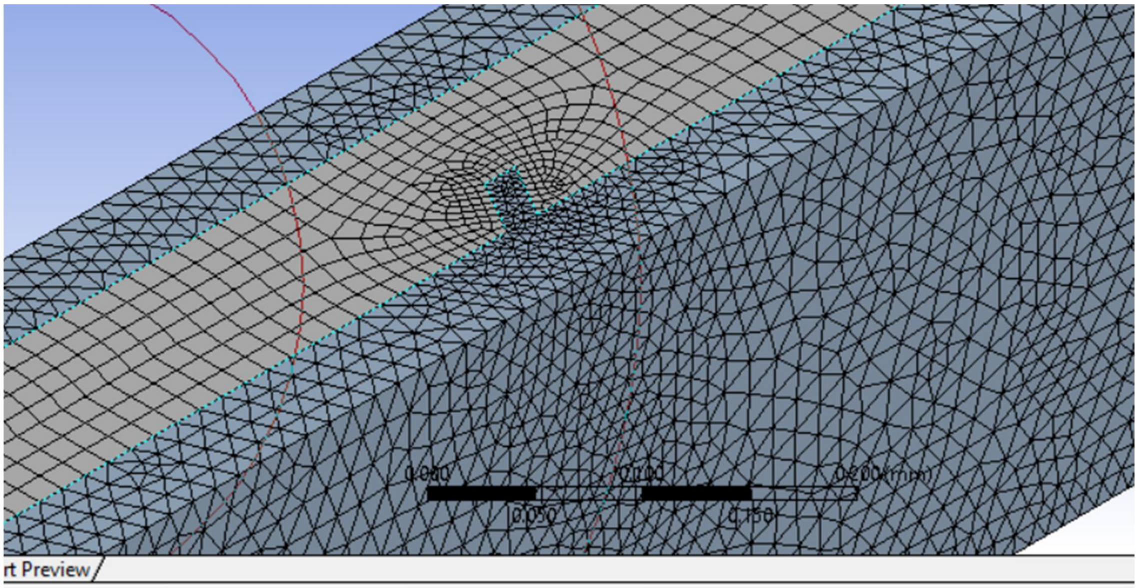


Figure 3.9

In this case the vortex generator placed at 60 degrees to the direction of inlet providing more restricted passage to the flow of fluid as compared to 30 degrees and 45 degrees in creating vortex and lateral shifting of stream lines and vortex generation is more in this case which enhance the rate of heat transfer and hence greater value of Nusselt number which ranges from 7.6 to 9.78 for different values of Reynolds number such as 150, 250, 350

Also, the pressure drop is more as compared to the case of plain channel, case of 30 degree and case of 45 degrees because of the restricted passage due to presence of vortex generator which further reduces the areas of flow.

Over all thermal performance is more than one when the value of Reynolds number is 150, 200, 250, 300, 350 which reflects that the effectiveness of vortex generator is good when the generation of vortices is more pronounced due to high effective area perpendicular to the direction of flow

3.7 CASE 4: Angle of vortex Generator is 120°

Program Case 4

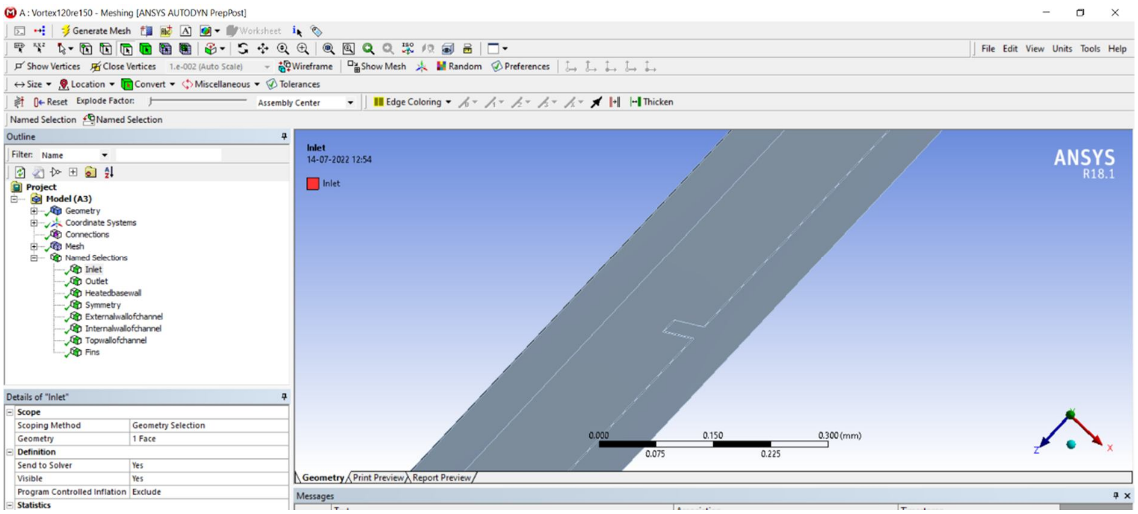


Figure 3.10

Geometry Case 4

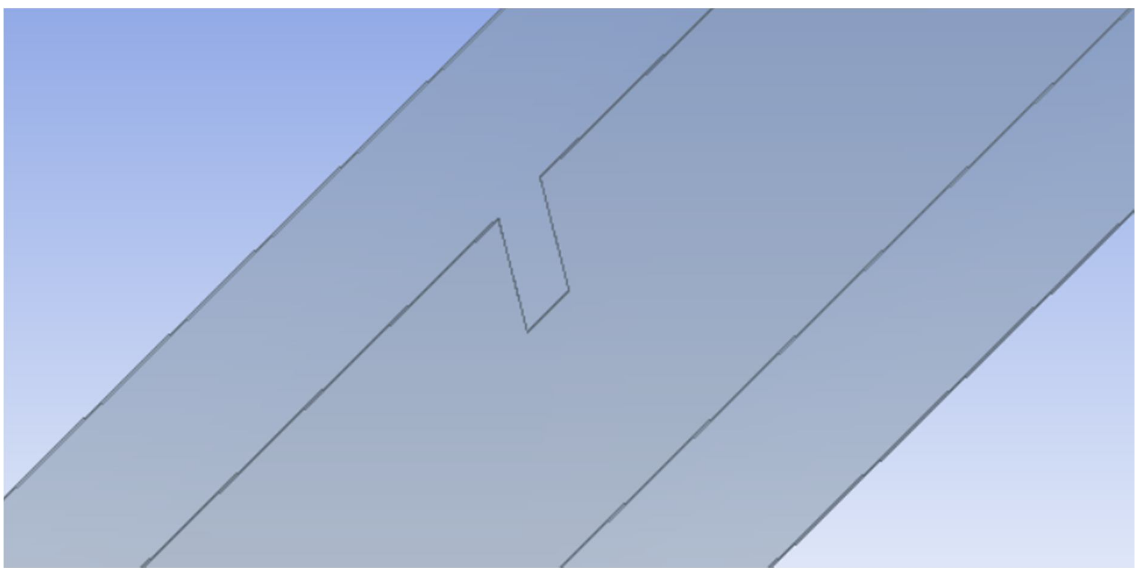


Figure 3.11

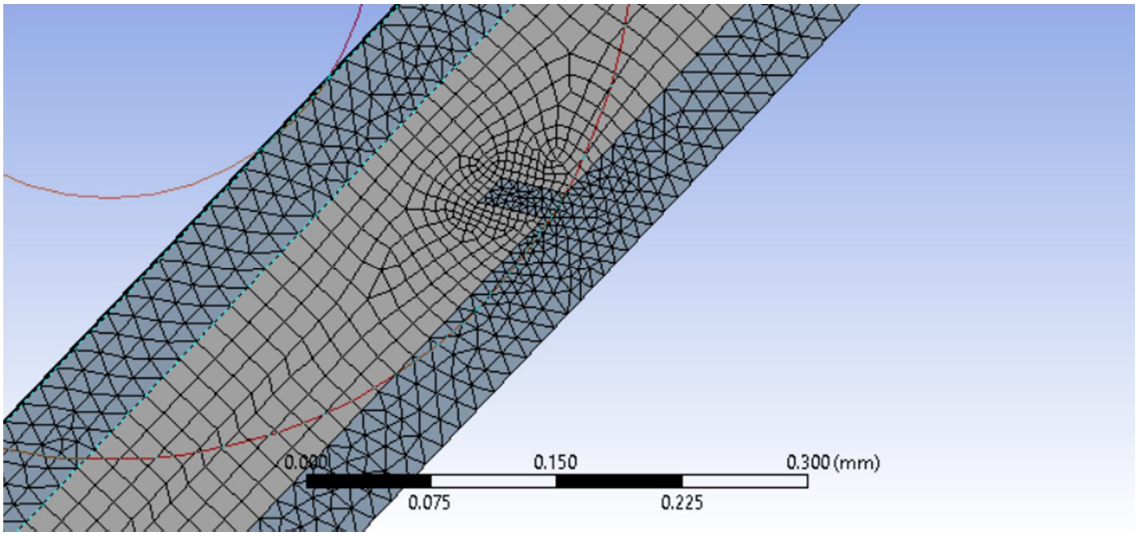


Figure 3.12

In this case the vortex generator placed at 120 degrees to the direction of inlet or 60 degrees with the direction of outlet providing restricted passage to the flow of fluid as compared to 30 degrees and 45 degrees in creating vortex and lateral shifting of stream lines and vortex generation is more in this case which enhance the rate of heat transfer and hence greater value of Nusselt number which ranges from 8.4 to 10.4 for different values of Reynolds number such as 150, 250, 350

Also, the pressure drop is more as compared to the case of plain channel. case of 30 degree and case of 45 degrees because in this case the vortex generator covers more cross-sectional area providing less area for floe however the value of heat transfer is also high due to high intensity of vortex generation.

Over all thermal performance is more than one when the value of Reynolds number is 150, 200, 250, 300, 350 which reflects that the effectiveness of vortex generator is good when the generation of vortices is more pronounced due to high effective area perpendicular to the direction of flow,

3.8 CASE 5: Angle of vortex Generator is 135°

Program Case 5

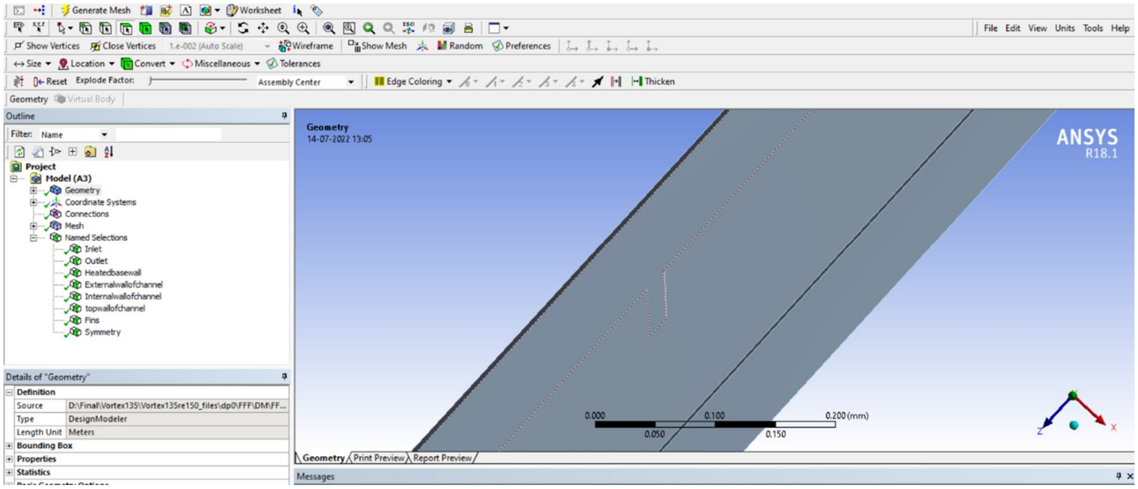


Figure 3.13

Geometry Case 5

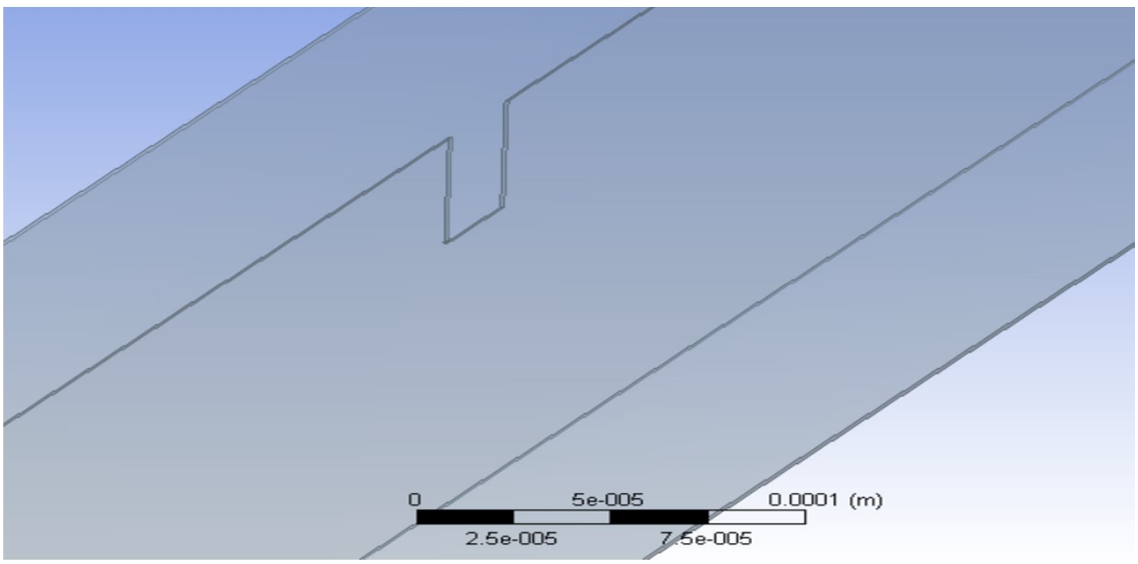


Figure 3.14

Mesh Case 5

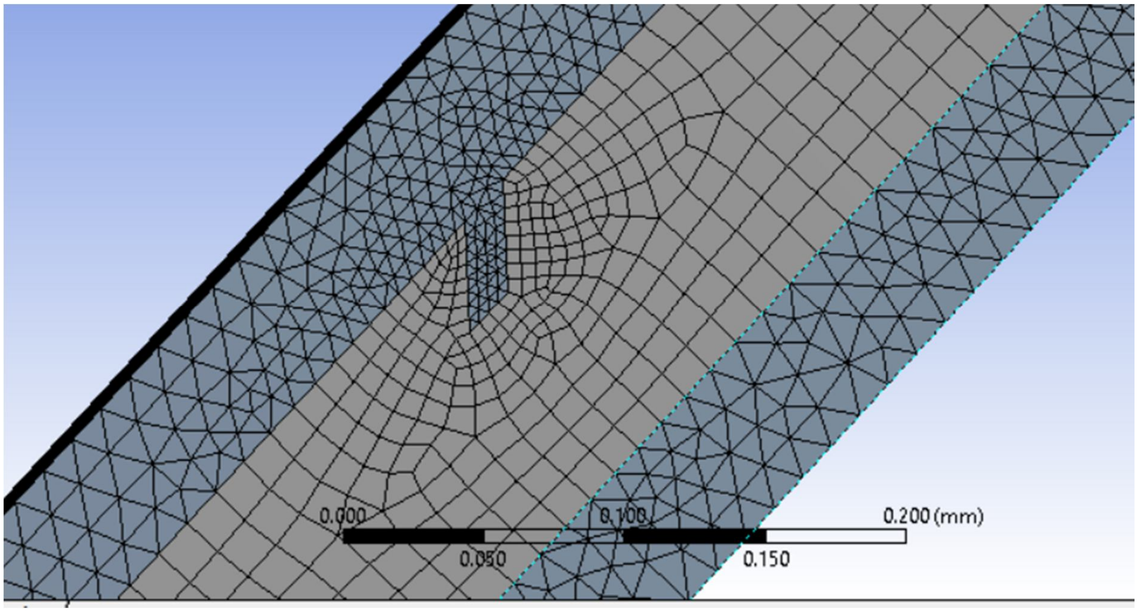


Figure 3.15

In this case the vortex generator placed at 135 degrees to the direction of inlet or 45 degrees to the direction of outlet providing more restricted passage to the flow of fluid as compared to 30 degrees and 45 degrees in creating vortex and lateral shifting of stream lines and vortex generation is more but less than case 4 as effective area of flow has been increased due to greater angle of 135 degrees in this case which enhance the rate of heat transfer and hence greater value of Nusselt number which ranges from 7.8 to 9.8 for different values of Reynolds number such as 150, 250, 350

Also, the pressure drop is more as compared to the case of plain channel. case of 30 degree and case of 45 degrees but resistance to flow is less as compared to case 1, case 2 and case 3 because the vortex generators are directed downstream of flow which provide gradual increase in area and less frictional resistance of the restricted passage due to narrow angle.

Over all thermal performance is more than one when the value of Reynolds number is 150, 200, 250, 300, 350 which reflects that the effectiveness of vortex generator is good when the generation of vortices is more pronounced due to high effective area perpendicular to the direction of flow

3.9 CASE 6: Angle of vortex Generator is 150°

Program case 6

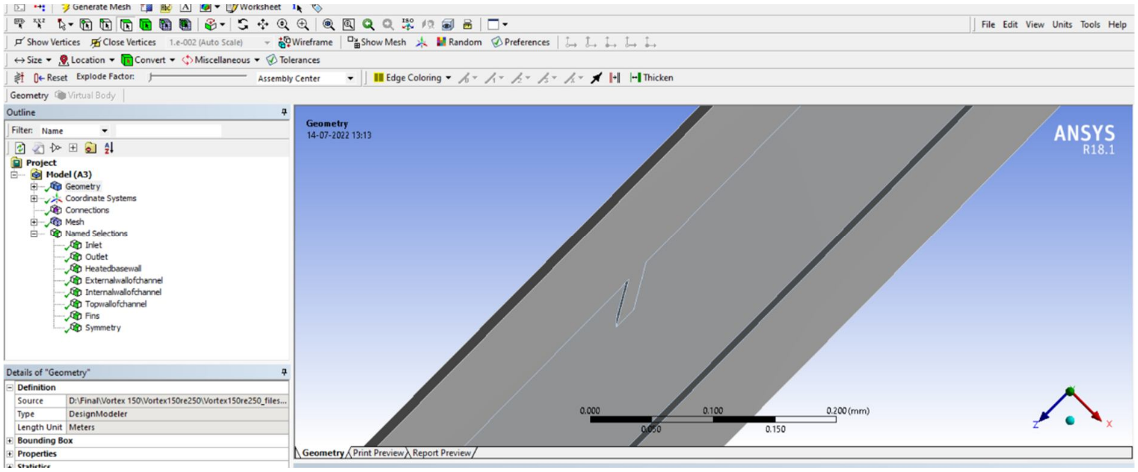


Figure 3.16

Geometry case 6

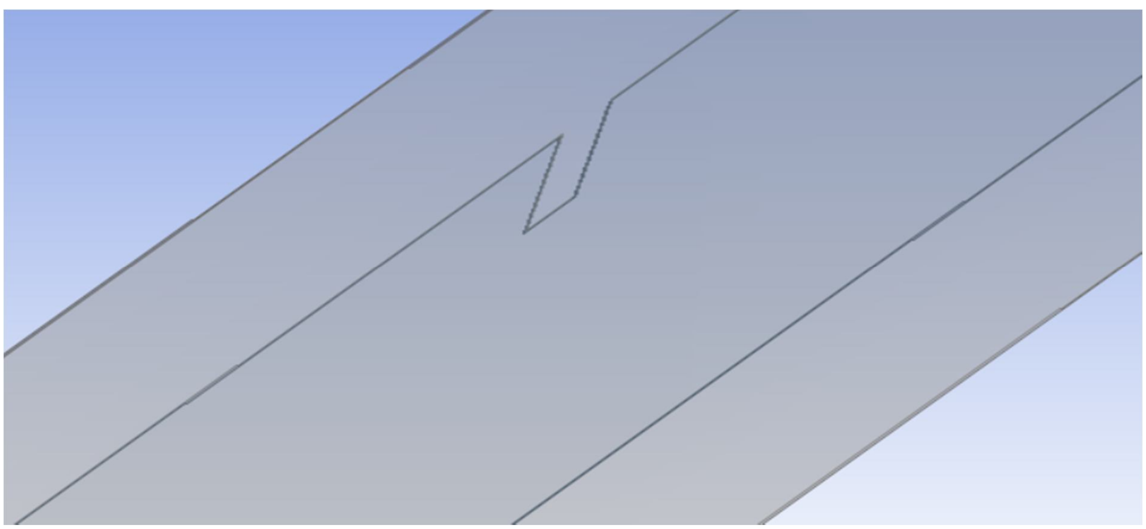


Figure 3.17

Mesh Case 6

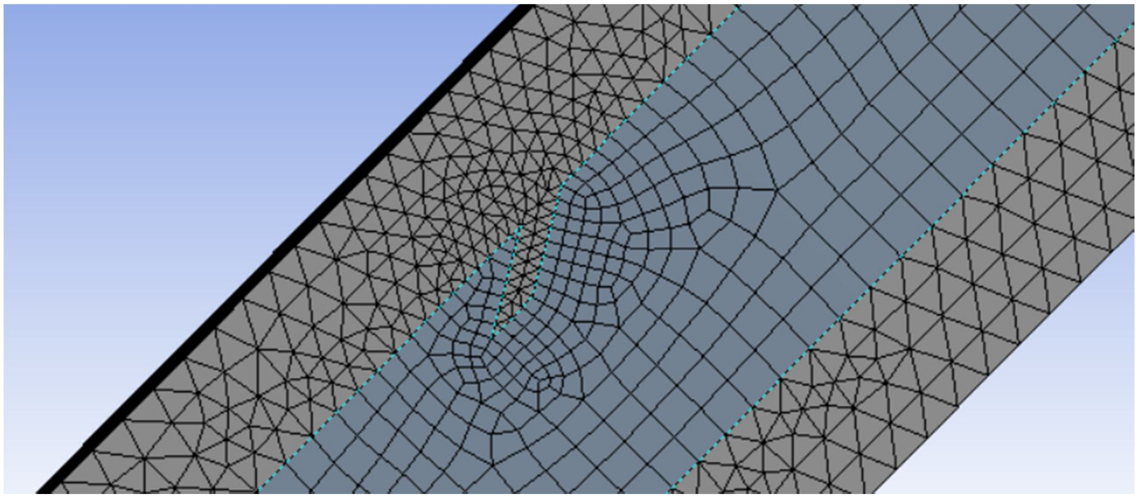


Figure 3.18

In this case the vortex generator placed at 150 degrees to the direction of inlet or 30 degree to the direction of outlet providing less restricted passage to the flow of fluid as compared to 45 degrees and 60 and 120 degrees in creating vortex and lateral shifting of stream lines and less frictional resistance to flow vortex generation is more in this case which enhance the rate of heat transfer and hence greater value of Nusselt number which ranges from 7.6 to 9.78 for different values of Reynolds number such as 150, 250, 350

Also, the pressure drop is less as compared to the case of plain channel, case of 45 degree and case of 60 degrees because of the more area available for flow and also the vortex generators are placed downstream to the flow which provide gradual reduction of area which makes the flow smooth and disturbance is less.

Over all thermal performance is more than one when the value of Reynolds number is 150, 200, 250, 300, 350 which reflects that the effectiveness of vortex generator is good when the generation of vortices is more pronounced due to high effective area perpendicular to the direction of flow

Data for Overall thermal performance,Nusselt number and Pressure drop for different cases at different reynolds number

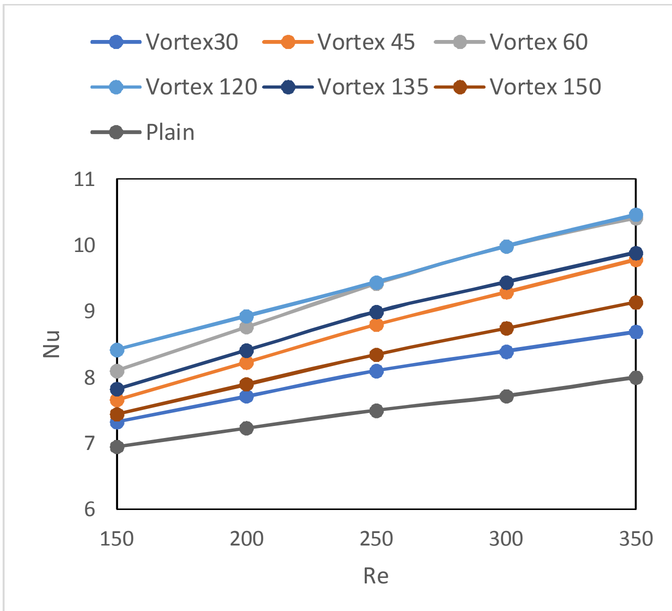
Table 3.1

200	0.9917	1.0255	1.05	1.06	1.0463	1.0135	
250	1.0169	1.065	1.09	1.1	1.0853	1.0459	
300	1.02	1.0885	1.115	1.141	1.10695	1.066	
350	1.03488	1.112	1.135	1.179	1.1286	1.0861	
			Nu				
Re	Vortex30	Vortex 45	Vortex 60	Vortex 120	Vortex 135	Vortex 150	Plain
150	7.324	7.657	8.102	8.419	7.822	7.444	6.95
200	7.711	8.227	8.7595	8.929	8.408	7.894	7.23
250	8.099	8.797	9.417	9.44	8.994	8.345	7.5
300	8.3935	9.288	9.984	9.986	9.44	8.74	7.72
350	8.6885	9.78	10.41	10.461	9.886	9.136	8
			Pressure drop				
Re	Vortex30	Vortex 45	Vortex 60	Vortex 120	Vortex 135	Vortex 150	Plain
150	23881.328	25704.000	28176.000	27981.000	25696.000	23973.000	18463
200	32699.000	35957.000	40172.000	39960.000	35841.000	32855.000	26423
250	41517.000	46210.000	52169.000	51940.000	45987.000	41737.000	34764
300	51045.000	57876.000	63841.000	63919.000	57386.000	51333.000	43436
350	60574.000	69542.000	74984.000	75899.000	68786.000	60929.000	52408

CHAPTER 4

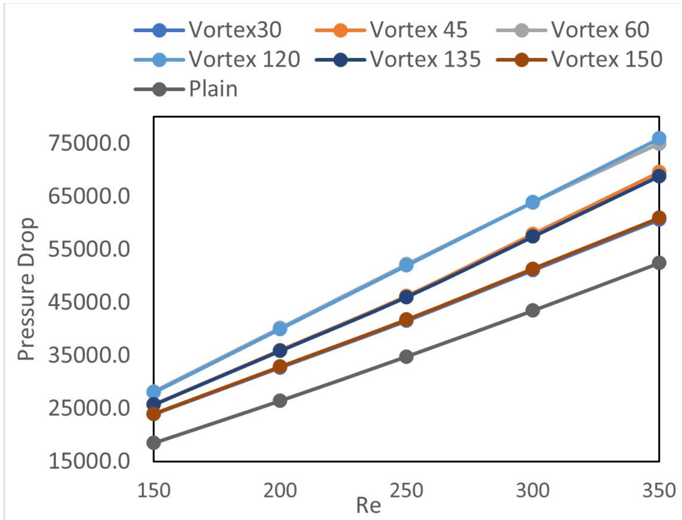
4.1 Results and Discussion: Overall thermal performance, pressure drop and Nusselt number recorded for different cases at different Reynolds number. In all cases the the values are in acceptable range as compared to plain microchannel

4.2 Analysis of Nusselt number



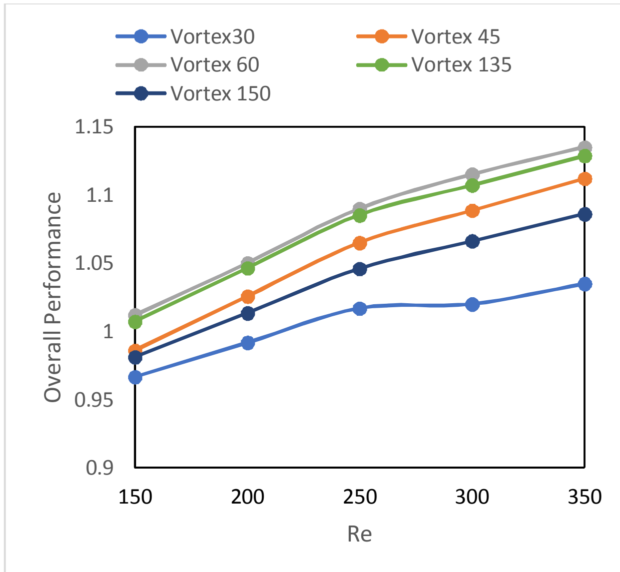
- ❖ Numerical simulations have been performed for these cases with Reynold's number ranging between 150 to 350 with a constant heat flux of 100 W/cm2 and the results are compared with the plain channel.
- ❖ Average Nu is increased with Reynold's number for all the cases.
- ❖ The reason for this increase is that as Reynold's number rises, the discharge of the water rises, assisting to extract a considerable amount of heat.

4.3Analysis of Pressure drop:



- The minimum pressure drop is recorded for the plain microchannel as it has the least heat transfer area which offers minimum flow resistance and the highest pressure drop is attributed to case 1.
- After case 1, the maximum value of pressure drop is recorded for case of 120° followed by 30°,45° 150° 135°.
- This curtailing trend of pressure drop is the outcome of the compound effect of reduction in resisting areas and the shape of the slots in each case respectively

4.4 Analysis of Overall thermal performance:



- ❖ Overall thermal performance portrays the mutual effect of heat transfer and pressure drop on the performance of the microchannel heat sink.
- ❖ Overall thermal performance is greater than 1 for all cases except for the case of 30° at Reynolds number greater than 250
- ❖ The value of overall thermal performance is maximum for the case of 60° followed by case of 135° 45° 120° whereas case of 30° and 150°
- ❖ Hence micro channel with vortex generator may replace plain microchannel in future studies of heat transfer enhancement

4.5 Conclusion

Following are the important concluding points drawn from the analysis:

- ❖ As the flaps acts as vortex generator and also acts as extended surface which eventually enhances the rate of heat transfer as results shows greater value of heat transfer in each case The maximum Nu is recorded for case of 120° followed by case of 60° whereas the minimum is for case of 30°

- The maximum pressure drop is recorded for case of 60° due to confined space available for flow of working fluid. and the minimum pressure drop recorded in case of 30° as the net cross sectional area for the flow is maximum in this case of 30°.
- Overall thermal performance is greater than 1 for all the cases except case of 30° at Reynolds number less than 250. The value of overall thermal performance is maximum for case of 60° followed by 135°.

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