

**STUDIES ON LIQUID FLOW & HEAT TRANSFER FOR
COOLING OF INTEGRATED CIRCUITS USING
MICROCHANNELS**

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requirements for the award of the degree of

**Doctor of Philosophy
In
Mechanical Engineering**

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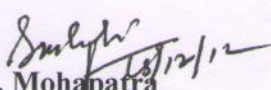


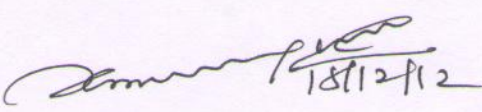
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CERTIFICATE

Certified that the thesis entitled, "**STUDIES ON LIQUID FLOW & HEAT TRANSFER FOR COOLING OF INTEGRATED CIRCUITS USING MICROCHANNELS**" which is being submitted by **MR. SATBIR SINGH SEHGAL** to Thapar University, Patiala, India in fulfilment for the award of degree of Doctor of Philosophy in Mechanical Engineering is a record of bonafied research work carried out by him under our guidance and supervision. The matter presented in this thesis has not been submitted either partially or fully to any other University or Institution for the award of any other degree to the best of our knowledge.

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ABSTRACT

The past couple of decades has witnessed a rapid progress in the applications of Integrated Circuits (ICs) forcing the size of these circuits to decrease drastically with high demand on increased operating speeds and package densities. These factors have lead to high die temperatures which are detrimental to circuit performance and reliability. A need has therefore arisen for new and innovative techniques for the development of embedded cooling solutions, IC-level integration of thermal sensors and heat sinks, and systematic synthesis techniques for ICs that contain embedded heat dissipation mechanisms. Microchannels and minichannels based heat sinks, which provide large surface to volume ratio, are the obvious choice for this task as they provide a large heat transfer surface area per unit fluid flow volume. The large surface to volume ratio leads to a high rate of heat transfer, making these micro devices as excellent cooling systems. Microchannels provide an efficient means to remove heat from a surface but pose certain challenges in bringing the fresh coolant to the heated surface and returning it to the cooling system before the coolant reaches the stringent temperature limits; however, this can be achieved at the cost of higher pressure drop per unit length. This problem is interlinked with the performance issues of the systems being cooled, which dictate a lowering of the surface temperature to increase reliability, speed of the processor units, or other system considerations.

Various researchers have studied the option of using liquid flow based microchannel heat sink (MCHS) as a solution for thermal management. An MCHS has an inlet plenum, from where the fluid is supplied to an array of micro-channels, and an outlet plenum, where it is collected after being made to pass through the microchannels. In order to meet specific requirements for various applications, the fluid inlet and outlet of the microchannel can be arranged in a number of ways. Owing to the small size of the channel dimensions, the entrance and exit effects will significantly affect the heat transfer characteristics of the flow field in the channels. In order to understand the performance of the heat sink as a single unit, it is necessary to study the effect of inlet and outlet flow arrangements on heat transfer characteristics of the microchannels.

A detailed literature review indicates that most of the research works available have focused on heat transfer and fluid flow analysis for the flow field along the direction of the fluid within the microchannels. However, to the best knowledge of the author, no detailed experimental investigation has been carried out to date to study the effect of flow arrangements and size of the inlet and outlet plenums on the performance of MCHSs. These

studies also need to be researched in detail under various heat input and different sizes of inlet and outlet plenums under different flow arrangements to suit practical demands in cooling applications of electronic devices. With these points as objectives in the present dissertation work, an attempt is made to understand the effect of different flow arrangements on the performance of MCHS. As the different flow arrangements are expected to affect the fluid flow and heat transfer characteristics within the heat sink, it is important to investigate the effect of inlet/outlet plenum flow arrangements to gain further knowledge on the performance of MCHS. It is also observed that flow variations at the entrance and exit regions of the microchannel need to be understood fully for efficient design of MCHSs.

As part of the present dissertation work, microchannels with different dimensions were fabricated to study the effect of channel aspect ratio and plenum aspect ratio on the thermal performance of MCHSs. Though channel aspect ratio is a well established parameter in microchannels research, a new aspect ratio called the plenum aspect ratio has been proposed in the present work to understand the influence of plenum dimensions on convective heat transfer within the MCHS. In the present research work, three different types of flow arrangements namely P-type, U-type and S-type were selected, where P stands for perpendicular, U stands for U-shape and S stands for serpent shape entry and exit of the liquid flow. In order to evaluate the thermal performance of the MCHS, experimental test runs were carried out according to the test matrix made with three flow arrangements, P-, U- and S-types; three heat input values, 125 W, 225 W and 375 W; six Reynolds numbers, 224.3, 336.5, 448.7, 673.0, 897.4 and 1121.7; three inlet and outlet plenum aspect ratios 2.5, 3.0 and 3.75, and two channel aspect ratios 4.72 and 7.57 giving rise to a total of 324 experimental trial runs. Based on the experimental data obtained heat transfer coefficient, Nusselt number and pressure drops were computed. Correlations for Nusselt number have also been developed, which can be readily used for estimating the convective heat transfer for a given MCHS. The experimental analysis is also supported by CFD simulation carried out with the help of Solidworks Flow Simulation software, in order to obtain a detailed temperature and pressure distributions within the MCHS which otherwise will be very difficult to obtain by experimental method also.

Experimental analysis shows that the reduction in channel width (increase in aspect ratio, defined as depth to width of channel) for the studied cases has shown about 126 to 165% increase in Nusselt number, whereas increase in plenum length (reduction in plenum aspect ratio defined as width to length of plenum) has resulted in 18 to 26% increase in Nusselt number. For a given flow arrangement, the Nusselt number increases when the

plenum aspect ratio is reduced and at a given range of Reynolds number, with increase in plenum aspect ratio there is decrease in the pressure drop. The minimum pressure drop is for P-type flow arrangement followed by U-type and S-type flow arrangements. As the channel aspect ratio is increased, the fluid velocity through the microchannels and the wall shear stress within the microchannels increase, resulting in higher pressure drop. The maximum pressure drop occurs at the higher channel aspect ratio, $AR_c=7.57$ and the lowest plenum aspect ratio, $AR_p=2.5$ for S-type flow arrangement, whereas the minimum pressure drop is noted for the case of P-type flow arrangement for $AR_p=3.75$.

The results of the CFD analysis of the complete MCHS unit carried out for various heat inputs, flow arrangements, plenum sizes and microchannel widths were found to be in line with the results of the experimental analysis. The heat transfer coefficients computed by CFD analysis were observed to be 2 to 19% higher as compared to the experimental values for the entire range of flow arrangements and heat inputs.

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NOMENCLATURE

A	Area of channel, [m ²]
AR	Aspect ratio, dimensionless]
C	Specific heat, [J/kg K]
D _c	Channel depth, [m]
D _h	Channel hydraulic diameter, [m]
D _i	Channel inner diameter, [m]
f	Friction factor, [--]
g	Acceleration due to gravity, [m/s ²]
h	Convective heat transfer coefficient, [W/m ² K]
k	Thermal conductivity, [W/mK]
L	Test section length, [m]
m	Mass flow rate, [kg/s]
N	Number of microchannels, [--]
Nu	Nusselt number, [--]
P	Pressure, [bar]
Pr	Prandtl number, [--]
q	Heat, [W]
Q	Volumetric flow rate, [m ³ /s]
RTDs	Resistance Temperature Detectors
Re	Reynolds number, [--]
R	Radius, [mm]
T	Temperature, [°C]
V	Velocity, [m/s]
W	Width of microchannel, [m]
x	Horizontal distance, [m]
y	Vertical distance, [m]
<i>Greek symbols</i>	
ρ	Density, [kg/m ³]
μ	Dynamic viscosity, [Ns/m ²]

Subscripts & superscripts

<i>c</i>	Channel
<i>cu</i>	Copper
<i>f</i>	Fluid
<i>i</i>	Inlet
<i>m</i>	Mean
<i>o</i>	Outlet
<i>p</i>	Plenum
<i>w</i>	Width

CHAPTER 1

INTRODUCTION

1.1 Thermal management of microelectronic devices

With the increase in heat dissipation from microelectronics devices and the reduction in overall form factors, thermal management becomes more and more important element of electronic product design. Semiconductor industry has benefited from several decades of growth following the Moore's law. Today's high-performance Integrated Circuits (ICs) have more than one billion transistors. The past couple of decades has witnessed a rapid progress in the applications of Integrated Circuits (ICs) forcing the size of these circuits to decrease drastically with high demand on increased operating speeds and package densities. These factors have led to high die temperatures which are detrimental to circuit performance and reliability. It is becoming increasingly difficult to sustain supply and threshold voltage scaling in order to provide the required performance increase, limit energy consumption, control power dissipation, and maintain the reliability. These requirements pose several difficulties across a range of disciplines spanning process technology, manufacturing, circuits, testing, systems, and architecture. One of these critical challenges is the thermal and power management of high speed ICs which has been subject of extensive research over past several years. Moreover, reliability screening and burn-in at elevated voltage and temperature further exacerbate this problem. Furthermore, researchers have looked into finding ways to cool ICs efficiently by accounting for the cooling cost in total system power. It may be possible to use efficient cooling to improve performance, leakage, and reliability of ICs, and thus continue on the path of scaling

1.1.1 Need for cooling of integrated circuits

Electronic equipment relies on the flow and control of electrical current to perform a fantastic variety of functions, in virtually every major industry throughout the world. Whenever electrical current flows through a resistive element, heat is generated in that element. An increase in the current or resistance produces an increase in the amount of heat that is generated in the element. The heat continues to be generated as long as the current continues to flow. As the heat builds up, the temperature of the resistive element starts to rise, unless the heat can find a flow path that

carries it away from the element. If the heat flow path is poor, the temperature may continue to rise until the resistive element is destroyed and the current stops flowing. If the heat flow path is good, the temperature may rise until it stabilizes at a point where the heat flowing away from the element is equal to the heat generated by the electrical current flowing in the element.

It is seen that over the past five decades the number of transistors that could be fixed on a silicon die has risen exponentially. This trend came to be known as the Moore's law, named after the Intel founder who observed and proposed in his paper [Moore G.E. (1965)] which describes that the number of transistors on a chip would double every year. The law has continued to be valid until very recently except that the transistor doubling happening every two years instead [ITRS update 2010]. The International Technology Roadmap for Semiconductors (ITRS) which comprises of experts from the semiconductor industry who periodically forecast the technology trends in silicon manufacturing related to microprocessors, memories etc. Figure 1.1 shows the trend as projected by ITRS until the year 2025. ITRS forecast that doubling the transistors will follow Moore's law until 2014, after that the trend will slow down with doubling of transistors after every 3 years.

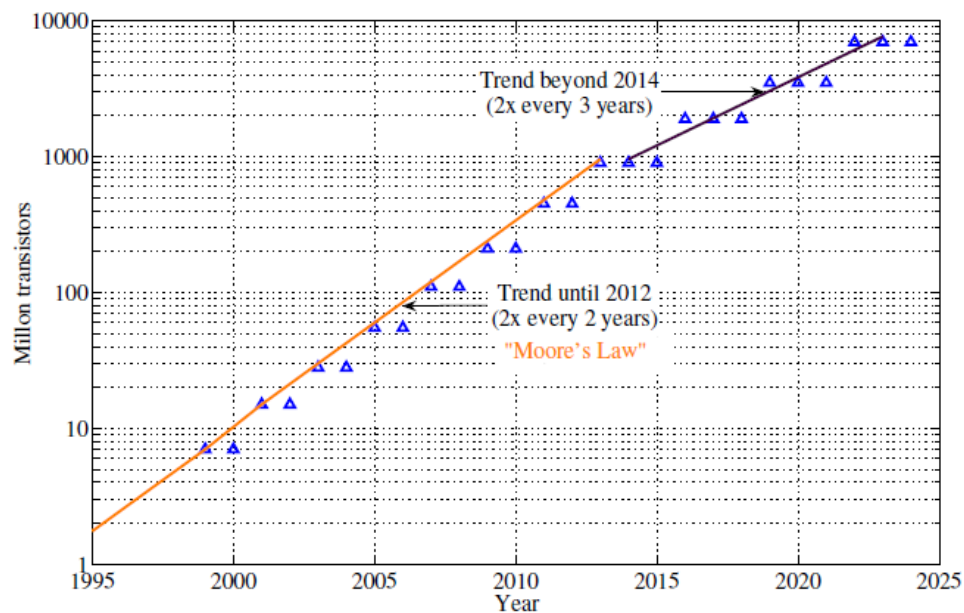


Fig.1.1: Semiconductor Technology Road Map [Image source: ITRS update 2010]

1.1.2 Challenges for integrated circuit cooling

In most of the semiconductor devices, failure mechanisms are temperature dependent (Figure 1.2), the failure rate of an electronic device accelerates with every increase in chip junction temperature. According to the ITRS for future generation of technologies, the junction temperature of the semiconductor devices must be kept at 85°C or lower to ensure the long term reliability of devices. Table 1.1 lists the projected values for the maximum junction temperature, ambient temperature, and the power consumption (heat dissipation) in the near term and long term for cost performance and high performance market segments. One can compute the required junction to ambient thermal resistance as the ratio of the difference between junction and ambient temperatures and the chip power. Computed thermal resistance values are listed in the last two rows of the Table 1.1 These resistance values represent the required overall junction to ambient thermal resistance. For a given heat dissipation rate, a lower chip to ambient thermal resistance ensures a lower package operating temperature and therefore a longer failure free operating period.

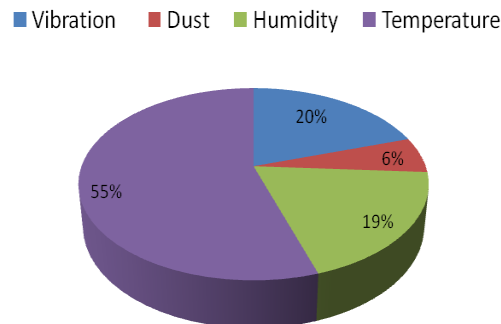


Fig.1.2: Major causes of electronic failure [Data source: Electronic Failure Analysis Handbook, Publisher: McGraw-Hill]

1.2 Electronic cooling solutions

1.2.1 Conduction and heat spreading

In all cooled applications, the heat from the device's heat sources must first arrive via thermal conduction at the surfaces exposed to the cooling fluid before it can be transferred to the coolant. For example, as shown in Figure 1.3, it must be conducted from the chip through the lid to the heat sink before it can be discharged to the ambient air. As can be seen, thermal interface materials (TIMs) may be used to facilitate this process. In many cases a heat spreader in the form of a flat plate with high thermal conductivity may be placed between the chip and the lid. Heat

spreading is a very effective means to alleviate the need for sophisticated high-heat flux cooling options. It is generally observed that the thermal benefit of reducing the heat flux by increasing the area, increase the cost due to inclusion of additional layer in the path of heat stream.

Table 1.1: ITRS sort and long term power and thermal projections for the years 2003 to 2016

Year of production	Term							
	2003	2004	2005	2006	2007	2010	2013	2016
	Maximum Junction Temperature (°C)							
Cost Performance	85	85	85	85	85	85	85	85
High Performance	85	85	85	85	85	85	85	85
	Ambient Temperature (°C)							
Cost Performance	45	45	45	45	45	45	45	45
High Performance	45	45	45	45	45	45	45	45
	Power (W)							
Cost Performance	81	85	92	98	104	120	138	158
High Performance	150	160	170	180	190	218	251	288
	Required Thermal Resistance(°C/W)							
Cost Performance	0.49	0.47	0.43	0.41	0.38	0.33	0.29	0.25
High Performance	0.27	0.25	0.24	0.22	0.21	0.18	0.16	0.14

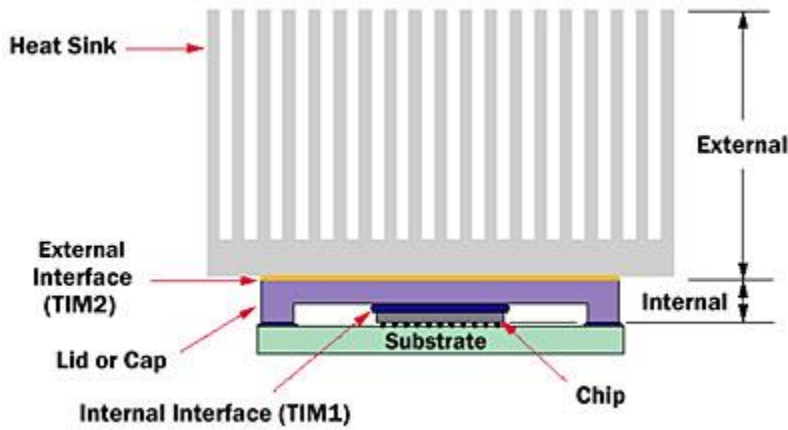


Fig.1.3: Chip package with thermal conduction path to heat sink via TIMs [Lasance and Simons (2005)]

1.2.2 Air cooling

It is generally acknowledged that traditional air-cooling techniques are about to reach their limit for high-power applications. With standard fans a maximum heat transfer coefficient of about $150 \text{ W/m}^2 \text{ K}$ can be reached with an acceptable noise level, which is about 1 W/cm^2 for a $\Delta T = 60 \text{ K}$ temperature difference. Using “macro-jet” impingement, theoretically we may reach

900W/m² K, but with unacceptable noise levels. Non-standard fan/specialized heat sink combinations for CPU cooling are expected to have a maximum of about $q = 50\text{W/cm}^2$.

New Strategies for air cooling:

Piezo fans

Piezoelectric fans are small, low-power, relatively low-noise, solid-state devices that provide viable thermal management solutions for a variety of portable electronic applications, including laptop computers and cellular phones. In these fans piezoceramic patches are bonded onto thin, low-frequency flexible blades driven at resonance frequency, thereby creating an air stream directed at the electronics components.

Synthetic jet cooling

An approach using periodic micro-jets called “synthetic jets” is still in the initial stages of study. Due to the pulsating nature of the flow, the jets create stronger entrainment than conventional steady jets with the same Reynolds number, and more vigorous mixing between the wall boundary layers and the rest of the flow. A test set-up is shown in Figure 1.4. The jet entrains cool air from the environment, impinges on the top hot surface and returns the heated air to the environment. Radial current counter flow is created in the gap between the plates, with hot air dispersed along the top and the ambient air entrained along the bottom surface. The idea was further advanced by the development of flow actuators using micro-electromechanical systems (MEMS) technology.

Recently, a new initiative “nanolightning” has extended traditional cooling option.

1.2.3 Nanolightning

A new approach for increasing the heat transfer coefficient, known as “nanolightning,” is also studied now-a-days. Nanolightning is based on “micro-scale ion-driven airflow” using very strong electric fields created by nanotubes. As shown in Figure 1.5, the ionized air molecules are moved by another electric field, thereby inducing secondary airflow. Cooling a heat flux level of 40 W/cm² has been reported. This technology is being commercialized through a start-up company (Thorn).

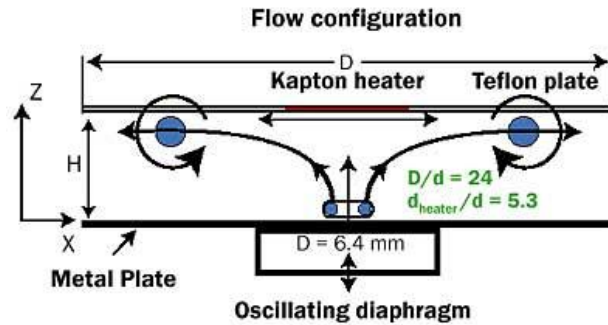


Fig.1.4: Flow dynamics of normal jet impingement with an oscillating diaphragm [Lasance and Simons (2005)]

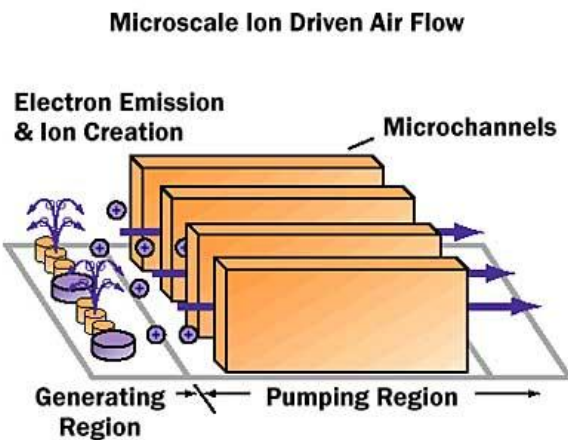


Fig.1.5: Nanolightning sketch [Lasance and Simons (2005)]

1.3 Need for liquid cooling

The continuing push towards more densely packed microchips will soon require greater thermal dissipation than that can be provided by simple forced air cooling. It is generally seen that manufacturers of electronic devices usually specify the rate of heat dissipation and the maximum allowable component temperature for reliable operation. These two parameters help the researcher to determine the cooling techniques that could be appropriate for the device under consideration. Figure 1.6 [Krause and Bar-Cohen (1983)] shows the heat fluxes attainable at specified temperature differences for some common heat transfer mechanisms. Figure 1.6 indicates suitable heat transfer mechanism that can be employed for cooling, based on the selected surface heat flux and the temperature difference between the surface of the device and the surrounding medium. The figure also indicates that at heat fluxes greater than 1 W/cm^2 , even forced convection with air will be inadequate (subjected to maximum allowable temperature difference of typically under 80°C), and we must use a sufficiently large heat sink or switch to a different cooling fluid such as water. Forced convection with water can be used effectively for cooling electronic components with high heat fluxes. Currently, air-cooled fan based heat sinks are the tools of choice for cooling most of the power electronics because of their low cost and low maintenance. Obviously, air cooled heat sinks cannot continue to grow in size in order to meet the demands of technology for adequate cooling of electronic devices.

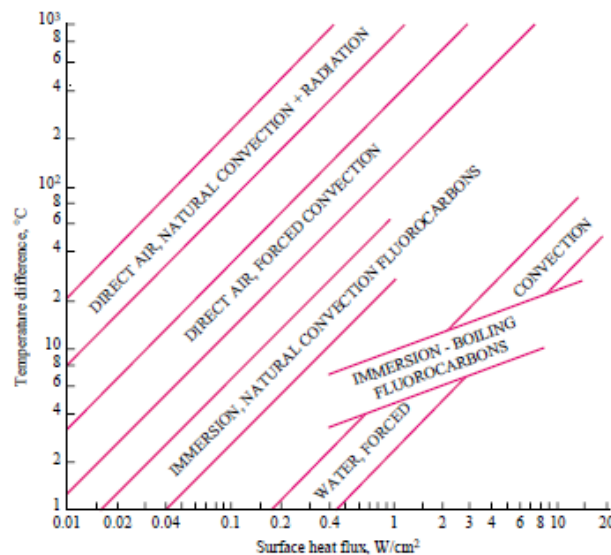


Fig.1.6: Heat fluxes at specified temperature differences with various heat transfer mechanisms and fluids [Krause and Bar-Cohen (1983)]

1.4 Liquid cooling

It is seen that liquids normally have much higher thermal conductivities than gases, and thus much higher heat transfer coefficients associated with them. Therefore, liquid cooling is far more effective than gas cooling. However, liquid cooling comes with its own risks and potential problems, such as leakage, corrosion, extra weight, and condensation. The pioneer water cooled thermal management product was introduced by IBM in 1964. The building chilled water was supplied directly to the system where it was used to cool interboard heat exchangers to reduce the temperature rise across multiple stacks of boards having number of cards. Figure 1.7 shows the schematic of such a liquid cooling arrangement. Improvements to IBM based water cooled systems continued over the next 30 years with many other companies in the mainframe business employing water cooling as a solution to thermal management..

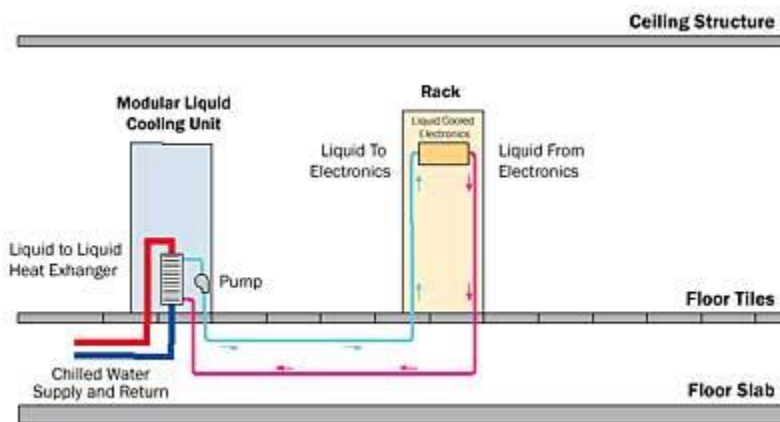


Fig.1.7: External liquid cooled rack with remote heat rejection [Schmidt (2005)]

Most of the companies focused their research on thermal management primarily on the use of fans, blowers, and heat sinks. In mid 90's companies shifted from bipolar technologies to CMOS where the power dissipation was at less magnitude and the traditional air cooling technique was the most cost effective choice. Figure 1.8 shows that module heat flux trend which continued to rise at an alarming rate, despite the employment of enhanced air cooling technologies that utilized heat pipes, vapor chambers, high performance interface materials, and high performance air moving devices.

Liquid cooling in electronics is generally divided into two main categories, indirect and direct, according to the type of contact between the coolant and the cooled components. The

following sections discuss the two categories, represented by heat pipes and cold plates, and immersion cooling and jet impingement, respectively.

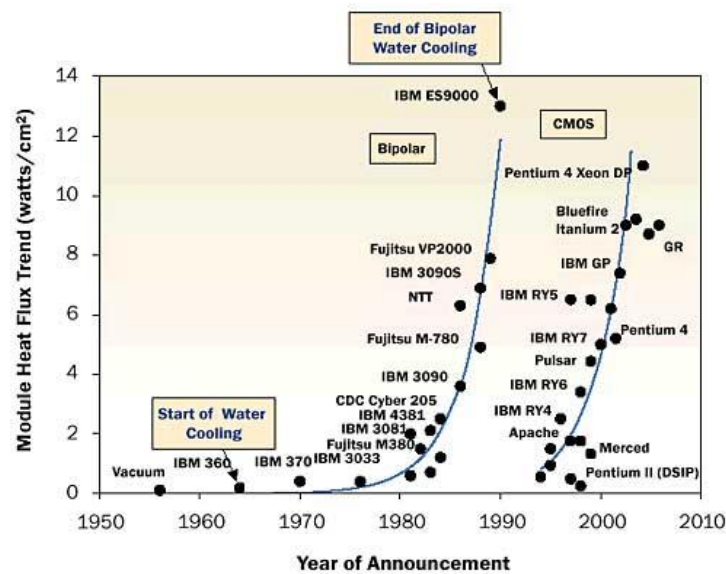


Fig.1.8: Module heat flux trend [Schmidt R.R. (2005)]

1.4.1 Heat pipes

A heat pipe is a heat-transfer device that combines the principles of both thermal conductivity and phase transition to efficiently manage the transfer of heat between two solid interfaces. The heat pipe has a simple configuration with a closed, evacuated cylindrical vessel with the internal walls lined with capillary structure or wick that is saturated with a working fluid.

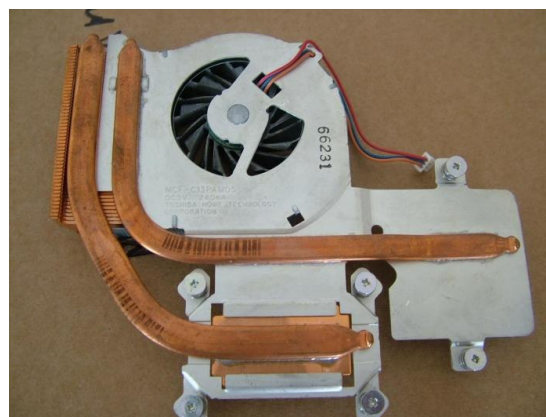


Fig.1.9: Heat Pipe (Actual photograph)

Figure 1.9 shows a typical heat pipe which has sealed and vacuum-pumped vessels partially filled with liquid. When heat is applied to one side of the pipe the liquid begins to vaporize. The

pressure gradient causes the vapor to flow towards the cooler regions, where it condenses and is transported back by the wick structure, thereby closing the loop. Heat pipes are considered as a low cost solution to cooling problem as they have most effective and have great potential when power levels and volume requirements increases.

1.4.2 Cold plates

A cold plate is considered as a fluid flow space that is contained within bounding metallic walls. Cold plate is considered by the thermal management researchers as one of the primary tools for cooling electronic equipments. While designing the fluid flow passages within a cold plate, the key task is to enable heat dissipation by the electronic equipment while maintaining temperatures at a level compatible with high performance.

Kandlikar and Hayner (2009) classified the configuration based on substrate and fluid flow channels within cold plates into four major parts:

- Formed Tube Cold Plate
- Deep Drilled Cold Plate
- Machined Channel Cold Plates
- Pocketed Folded-Fin Cold Plates

1.4.3 Immersion cooling

Technically the term ‘immersion cooling’ implies that the device is plunged entirely into a fluid. In immersion cooling, boiling heat transfer takes place because the temperature of the material to be cooled is usually higher than the boiling temperature of the liquid coolant. Direct liquid immersion cooling provides higher heat transfer, as there is no wall separating the heat source from the coolant, the heat can be removed directly from the chip. Lasance and Simons (2005) observed heat flux up to 100 W/cm^2 can be achieved by surface enhancement technique of the heat source, Figure 1.10 show such a device which was submerged in a pool of dielectric liquid. Vapor bubbles are produced by heat dissipation in the device and are driven by buoyancy forces into the upper region of the container; after the vapor condenses it drips back into the liquid pool.

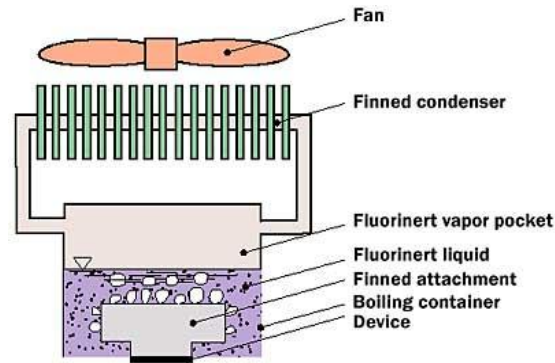


Fig.1.10: Pool boiling (thermosyphon) cooling [Lasance and Simons (2005)]

1.4.4 Indirect liquid cooling

Indirect liquid cooling of electronic devices is considered as one, in which the liquid is not in contact with the microelectronic chips, nor the substrate upon which the chips are mounted. The challenge of this system is to design a good thermal conduction path from the microelectronic heat source to the liquid cooled cold-plate attached to the module. The major advantage of this indirect liquid cooling is that, as there is no direct contact with the electronic device, even water can be used as the coolant. Figure 1.11 illustrates three possible implementations of indirect liquid cooling systems as suggested by Ellsworth and Simons (2005). Figure 1.11a shows the simplest implementation, where a separable cold plate is attached to a lidded module. For larger thermal performance of the system, the interface between the cold plate and the module is eliminated (as shown in Figure 1.11b) by integrating the cold plate with the module. Figure 1.11c shows the highest performance of the system when cold plate is attached directly to the chip.

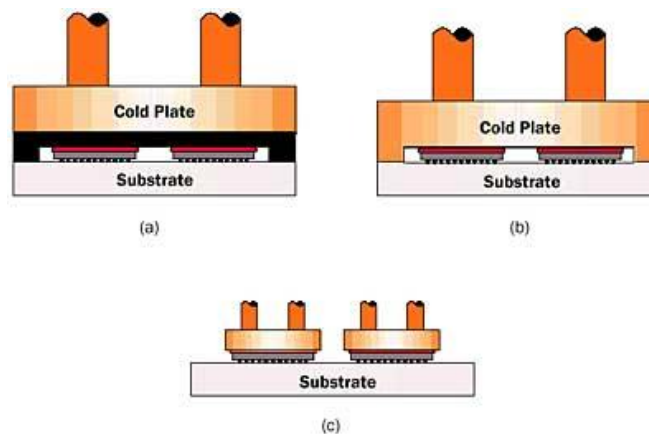


Fig.1.11: Indirect liquid cooling schemes: (a) separable module cold plate, (b) integral module cold plate, and (c) individual chip-scale cold plates [Ellsworth and Simons (2005)]

1.4.5 Single phase and two phase liquid cooling

Due to the higher heat transfer coefficient, liquid coolant can be used as compared to traditional air cooling. Depending on the phase of the coolant (single phase and two-phase) heat transfer and liquid flow can be studied for electronic cooling. Single phase liquid cooling includes dielectric and non-dielectric fluids. The major disadvantage observed in single-phase liquid cooling is that it requires either high flow rates or smaller hydraulic diameters, which consequently results to larger pressure drops.

Two-phase cooling methods are appealing than single-phase methods as the working fluid is vaporized in one section and then condensed in another, leading to much higher heat transfer coefficients. It is observed that latent heat of most of the fluids is at least one or two orders larger than the sensible heat, a two-phase heat transport system can transport considerably more heat per fluid mass resulting in a lower system mass and smaller cooling system. Another important but less well-known advantage is the high temperature stability of two-phase heat transport systems. General disadvantages of two-phase systems are: Gravity dependence, Inflexibility of pay-load interface (evaporator) and use of toxic working fluids (ammonia, ethanol, methanol, etc).

1.5 Heat sink

Heat sink is an object, system or an environment that absorbs and dissipates heat from another using thermal contact (either direct or radiant). It is seen that heat sink transfers thermal energy from a higher temperature source to a lower temperature fluid medium. Air is frequently the fluid medium, but water, refrigerant, oil and other liquids can also be used. Heat sinks are used in integrated circuits and power handling semiconductors to reduce their temperature through increased thermal mass and heat dissipations (primarily by conduction and convection but to a lesser extent by radiation). The most common heat sink materials are aluminium alloys and copper. Sometimes, diamond and other composite materials are also used for thermal management of electronic devices.

1.6 Cooling of integrated circuits using microchannel heat sink (MCHS)

It is observed that a number of cooling solutions for thermal management of microelectronic devices have been developed which include extended surface (fins), highly parallel air and liquid

impingement systems, modular internal conduction enhancement, indirect and direct liquid cooling with water and dielectric coolants.

It is seen in integrated circuits as the size becomes smaller, removal of heat becomes a more complicated since the same (or better) processing power is packed into a smaller package. It is necessary to remove this heat as effectively as possible, preferably using some cooling fluids. The efficient heat removal in the miniature scale (sub-mm) is difficult since convection is relatively weak. The discovery of microchannels as effective cooling solution and their implementation in a miniaturized heat sink is a potential enabler to continued miniaturization. A microchannel heat sink consists of micron sized channels and fins in a parallel arrangement (Figure 1.12).

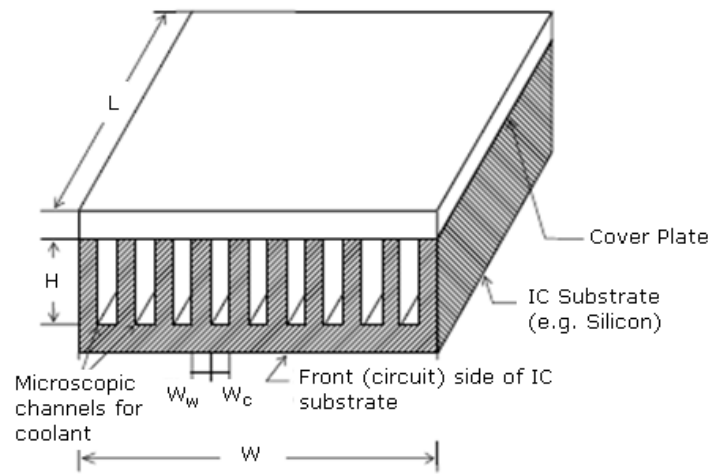


Fig.1.12: High performance microchannel heat sink (without plenums)

For laminar, fully developed single phase flow at constant Nusselt number these small channel size (D_h) in microchannel heat sink gives high heat transfer coefficient.

$$Nu = \frac{hD_h}{k} = \text{constant} \Rightarrow h \propto \frac{1}{D_h} \quad (1.1)$$

Due to heat transfer enhancement, increased heat flux dissipation in microelectronic devices and materialization of microscale devices that require cooling, the flow passage dimensions in convective heat transfer applications have been shifting towards smaller dimensions. It is also seen that though smaller channel dimensions results in higher heat transfer performance, it is at the same time also accompanied by a higher pressure drop per unit length.

1.7 Scope of work

The microchannel heat sink as a whole consists of array of microchannels, the arrangement by which the fluid enters and exits the channels. Like in the case of heat exchangers, where the direction of fluid flow affects the performance, the MCHS performance will also vary with different flow orientations. There are different arrangements for fluid flow before and after it enters the microchannels for the same microchannel heat sink. A detailed literature survey indicates that earlier researchers introduce the fluid directly over the microchannels, which in turn introduced thermal and fluid distributions. Experiments by various researchers were focused mainly on microchannels and only limited numerical research had studied the MCHS as a whole unit. The region before the microchannel region is called inlet plenum and the region after the microchannels are called outlet plenum. Depending upon the dimensions of these plenums, entry and exit flow variations are possible. Numerous flow arrangements are possible for the fluid to enter and exit the MCHS. The type of entry and exit will decide the flow distribution inside the channels and the performance at large. Though some numerical works have been carried out on the study of inlet/exit conditions, there is a need to take up a detailed experimental investigation to study how different flow entry/exit will affect the performance and whether these flow arrangements will alter its influence at different flow conditions and heat inputs. Literature has suggested that the dimensions of the microchannels have affected the thermo-hydraulic performance. Similarly the dimensions like length, width and depth (introducing the concept of aspect ratio) of plenums can have an effect on the performance. In order to have better understanding of the physics of fluid flow problem within the plenums, microchannel and heat sink as a whole, computational fluid dynamics (CFD) analysis can be carried which also can be verified with experimental observations. Thus microchannel heat sink, which is to be used as thermal management solution for cooling of integrated circuits needs to be studied exhaustively alongwith the entry and exit flow conditions of liquid.

1.8 Organization of thesis

The whole thesis is divided into six chapters with reference section at the end. The present research work includes both experimental and computational analysis of the microchannel heat sink.

Chapter 1

This chapter gives a brief introduction to thermal management of microelectronic devices and the need for cooling of integrated circuits. The thermal management solutions available in market for cooling of integrated circuits are also discussed. This chapter introduces the concept of microchannel heat sink with single phase liquid cooling.

Chapter 2

It is related to literature review and problem formulation. Extensive literatures on experimental, numerical and computational work by various researchers are reviewed. Later in this chapter gaps in literature, problem formulation and objectives are described. An overview of research work in the field of liquid cooling using microchannels is presented in the chapter.

Chapter 3

This chapter deals with experimental setup and procedure adopted for the analysis of the performance of microchannel heat sink. This chapter describes the flow arrangements which are generally available in microchannel heat sink (MCHS). The inlet/ outlet flow effects, heat transfer and fluid flow performance parameters in MCHS are also studied. Fabrication procedure of test piece, geometrical parameters which affect the performance of MCHS are also included in this chapter. Details about the instruments used and procedure for heat transfer and fluid flow analysis are also presented in this chapter.

Chapter 4

This chapter deals with the experimental analysis of the MCHS. Design of experiments and test matrix for the MCHS is discussed in brief. The temperature distributions/ variations are discussed for different flow arrangements, channel and plenum aspect ratios. Pressure distributions for the whole MCHS including inlet and outlet plenums are also presented. Correlations for the Nusselt number variation for the studied three flow arrangements are also proposed in this chapter.

Chapter 5

In this chapter CFD modeling of thermal performance of MCHS has been formulated. The governing equations which constitute the thermal and hydrodynamic behaviour of the system are explained. The geometry of the MCHS which is under consideration is also discussed in detail. Boundary and initial conditions required for simulation studies are described in this chapter.

MCHS which has the same dimensions as that of the one used in the experimental test setup was modeled and simulated using Flow Simulation commercial software.

Chapter 6

This chapter describes the summary of the present research, important experimental and computational conclusions. The scope for future work is also narrated in this chapter.

CHAPTER 2

LITERATURE REVIEW AND PROBLEM FORMULATION

2.1 Literature review

Cooling of integrated circuits using microchannel has been studied by many researchers throughout the world. An extensive literature survey has been carried out to study the research works available for a single/ array of microchannel(s) as well as for complete microchannel based heat sinks. The literature survey subsection in this chapter is divided into three broad categories:

- Experimental study
- Computational/ Numerical study
- Studies of the effects of manifolds/flow arrangements

The objective of this chapter is to review all the literature related to the subject extensively. Consequently, limitations of literature review are explored and problem is formulated.

2.1.1 Experimental study

A lot of experimental studies have been carried out by various researchers to find out an effective way of cooling miniature or micro devices. The microchannel heat sink is considered one of the most promising cooling methods because it is adequate for the micro fabrication process. Experimental results of friction factors, pressure drops, convective heat transfer coefficients and Nusselt numbers in single or array of microchannels show that there is a great deal of heat and fluid flow parameter combinations possible. Tuckerman and Pease (1981) started the pioneer work in this area of thermal management by experimentally demonstrating the high-performance heat sinking for very-large-scale integration (VLSI) systems. They optimized the dimensions of the channels in terms of width and height for single-phase flow of water under the constraint of maximum allowable pressure drop and substrate surface temperature. Their work involves testing and modeling single phase heat transfer in channels of 56 μm wide and 364 μm deep. A power density of 790 W/cm^2 with maximum substrate temperature rise of 71°C above the input water temperature was obtained. Another important experimental investigation on single phase forced flow convection of water/ methanol flowing through microchannels with rectangular cross section was carried by Wang and Peng (1994). They observed a fully developed turbulent

convection regime for Reynolds number range, 1000-1500. Peng et al. (1995) experimentally studied the effect of methanol as a coolant on heat transfer characteristics and performance of rectangular-shaped microgrooves of different aspect ratios on stainless steel plates. In their work, influence of liquid velocity, sub cooling, property variations and microchannel geometric configuration was analyzed. The flow nucleate boiling attributes indicated an increased heat transfer rate and different behavior from larger tubes or channels, due to the relatively large portion of the surface area associated with the thin film region. Tso et al. (1998) introduced the relationship of Brinkman number, a dimensionless number related to heat conduction from a wall to a flowing viscous fluid, as applicable to microchannels and the role played in convective heat transfer. Using the experimental data reported in different literatures for the laminar regime correlations were developed by them using Brinkman number. Another experimental and numerical investigation by Weilin et al. (2000) on heat transfer characteristics of water flowing through trapezoidal silicon microchannels with a hydraulic diameter ranging from 62 to 169 μm was studied. They considered the problem of heat transfer as conjugate and numerically solved it to determine temperature fields in both the solid and the fluid regions. They attributed the reason for lower experimental Nusselt number as compared to numerical one, to the effects of surface roughness of the microchannel walls.

The use of micron-resolution particle imaging velocimetry (μPIV) with epifluorescent microscopy was made by Zeighami et al. (2000) to experimentally study the turbulent transition in microchannels. Silicon channels with dimensions $150\ \mu\text{m} \times 100\ \mu\text{m} \times 1\ \text{cm}$ were fabricated using deep reactive ion etching and sealed using a glass plate. Their experimental approach and different Reynolds number transition as measured previously for larger channels with similar shapes, provides the groundwork for a detailed study of turbulence transition. Another important finding was reported by Qu and Mudawar (2002) who proved experimentally and computationally that the conventional Navier-Stokes (NS) equations can accurately predict the liquid flow in microchannels. They investigated the adiabatic single-phase water flow in a single rectangular microchannel at Reynolds numbers ranging from 196 to 2215. Their findings lead to the use of Navier-Stokes equations in numerical analysis for the design and analysis of microchannel heat sinks intended for electronic cooling. The experimental results by Yin et al. (2002) confirmed that the Moody chart can be applicable for the sub millimeter channel flows.

They developed a pressure drop model for the whole heat exchanger, and predicted the pressure and mass flow rate distribution inside the heat exchanger. Bergles et al. (2003) presented a critical review of design based processes which include flow distribution issues in single, parallel, and split flows; flow instability in parallel passages; manufacturing tolerance effects; single- phase heat transfer; nucleation processes; boiling heat transfer and pressure drop; and wall conductance effects. This paper provides an insight about the use of these design factors for practical engineering systems used in micro channel based heat exchangers. Qu and Mudawar (2003) identified two types of two-phase hydrodynamic instability mechanisms: severe pressure drop oscillation and mild parallel channel instability, when they experimentally studied water-cooled two-phase micro-channel heat sink. They also proposed a new and better correlation incorporating the effects of both channel size and coolant mass velocity. Garimella et al. (2004) presented a review on pumping requirements for flow through microchannels, and showed that none of the micropumps in the literature is truly suitable for electronic cooling application. They reported another important finding that the conventional correlations for fluid flow and heat transfer adequately predict the behavior in microchannels of hydraulic diameters as small as 250 μm . Their work also involved optimizing the size of the microchannels for the minimum pumping requirement.

Morini (2004) presented a review on the convective heat transfer through microchannels. He critically analyzed the main results obtained on the friction factor, laminar-to-turbulent transition and Nusselt number in channels having a hydraulic diameter less than 1 mm. He proposed rarefaction and compressibility, viscous dissipation, electro-osmotic (EDL), property variation effects, channel surface conditions (relative roughness) and experimental uncertainties as reasons to explain the anomalous behaviour of the transport mechanisms through microchannels.

Using experimental and theoretical methods, Hetsroni et al. (2005) investigated the effects of geometry, axial heat flux due to thermal conduction through the working fluid on single-phase heat transfer in micro-channels. A continuum model, corresponding to small Knudsen number was considered for the heat transfer analysis in circular, triangular, rectangular, and trapezoidal micro-channels with hydraulic diameters within the range 60 μm to 2000 μm . The need for appropriate channel geometry configuration was studied by Kandlikar and Upadhye

(2005). They found that as the heat flux increases beyond 2 MW/m^2 , the heat transfer and pressure drop characteristics of the plain channels dictate the use of turbulent flow through the channels. Klein et al. (2005) investigated the effects of surfactant solution flows through a micro-channel heat sink which consists of inlet and outlet manifolds. The experimental results had shown that there exists an optimal solution concentration and mass flux for enhancing heat removal. Their work has shown the use of surfactant solutions at low mass flux to enhance the CHF when compared to regular water flows. Lee et al. (2005) also proved experimentally the validity of classical correlations based on conventional sized channels, for predicting the thermal behavior in single-phase flow through rectangular microchannels. Their numerical predictions obtained based on a classical, continuum approach were found to be in good agreement with the experimental data, suggesting that a conventional analysis approach can be employed in predicting heat transfer behavior in microchannels. Liu et al. (2005) investigated the onset of nucleate boiling in the flow of water through a microchannel heat sink when the channels were $275 \mu\text{m}$ wide and $636 \mu\text{m}$ deep. They also developed an analytical model to predict the incipient heat flux and bubble size at the onset of boiling. Rathnasamy et al. (2005) investigated the flow of air, nitrogen, oxygen gases and ethanol, methanol, and their mixtures as liquids in long microchannels. They analyzed three test sections of different dimensions of serpentine shape on a stainless steel. The analysis of friction factor with Reynolds number showed a perceptible dependence on channel dimensions.

Bavière et al. (2006) analyzed the hydrodynamics and heat transfer characteristics in two dimensional microchannels. An important feature of their test section was the provision to vary the channel height and to set a quasi-constant heat flux. It was observed that there was a decrease in the Nusselt number in the laminar regime when the channel spacing was decreased. The role of viscous heating in microchannel flows was studied by Celata et al. (2006). They presented a criterion to illustrate the limit of significance for viscous dissipation effects in microchannel flows. They established experimentally a connection between viscous heating and the friction factor. Chen and Garimella (2006) studied the effects of dissolved air in dielectric liquid FC-77 on flow boiling in a microchannel heat sink which had ten parallel channels, each of $500 \mu\text{m}$ width and 2.5 mm depth. A significant reduction in wall temperature was noted when bubbles were first observed in the microchannels. They noted that wall temperature continued to increase

for the un-degassed liquid when the boiling process was initiated, whereas it remained relatively unchanged in the case of the degassed liquid. The pressure drop was observed to be higher for un-degassed liquid than for degassed liquid once the boiling process became well established in both the cases. They also quantified the flow instabilities induced by the boiling process within the microchannels. Garimella (2006) reviewed the recent advancements in high-performance cooling techniques for emerging electronics applications. The importance of this paper was the analysis of critical thermal management technologies like micro pumps, jet impingement, miniature flat heat pipes, transient phase change energy storage systems, piezoelectric fans, and prediction of interface contact conductance. Steinke and Kandlikar (2006) presented a critical review of available literature on single-phase liquid friction factors in microchannels. Their important finding was the usage of conventional Stokes and Poiseuille flow theories for single-phase liquid flow in microchannels.

Agostini et al. (2007) presented a literature review and compared the merits and drawbacks of three technologies: liquid cooling in copper or silicon micro-geometry heat dissipation elements, impingement of liquid jets directly on the silicon surface of the chip, and two-phase flow boiling in copper heat dissipation elements with numerous microchannels. Hakamada et al. (2007) described in their paper, spacer method as fabrication technique for copper microchannels. Three types of copper microchannels were fabricated using different copper powder sizes and investigated the effect of metal powder size on surface roughness of the microchannel wall. Hrnjak and Tu (2007) investigated fully developed R134a liquid and vapor flow through rectangular microchannels with hydraulic diameters varying from 69.5 to 304.7 μm and aspect ratios from 0.09 to 0.24. Pressure drop data was used to characterize the friction factor in the laminar, transition and turbulent region. They observed that when the channel surface roughness was low, both the laminar friction factor and the critical Reynolds number approached the conventional values.

Lee and Mudawar (2008) proposed a cooling scheme where the primary working fluid flowing through a micro-channel heat sink was pre-cooled to low temperature using an indirect refrigeration cooling system. They measured the Cooling performance using HFE 7100 as working fluid and used four different micro-channel sizes. High-speed video imaging was

employed to help explain the complex interrelated influences of hydraulic diameter, micro-channel width, mass velocity and sub cooling on cooling performance.

Vijayalakshmi et al. (2009) studied experimentally the internal pressure distribution along the length of the microchannels of hydraulic diameter ranging from 60.5 to 211 μm . The effects of compressibility and transition to turbulence occurring in the microchannel flow were analyzed for the local flow behaviour.

Chiu et al. (2011) investigated the effects of geometrical parameters aspect ratio (1.67 to 14.29) and cross-sectional porosity (25% to 85%) of the channels on the pressure drop and heat transfer performance of heat sink. They also found that the aspect ratio corresponding to the lowest effective thermal resistance was altered with respect to the pressure drop. One of the important observations was constant value of effective thermal resistance for cross-sectional porosity in the range of 53% –75% whereas, when cross-sectional porosity deviates from this range effective thermal resistance increases.

Nielsen et al. (2012) investigated the effect of flow mal-distribution on the performance of microchannel parallel plate heat exchangers using an established single blow numerical model and cyclic steady-state regenerator experiments. They found that as the variation of the individual channel thickness in a particular stack (heat exchanger) increases the actual performance of the heat exchanger decreases significantly, deviating from the expected nominal performance. They attributed the reason to both the varying fluid flow velocities in each individual channel and the thermal cross talk between the channels transverse to the direction of the flow. Another experimental investigation by Chu et al. (2012) was carried out to study the behavior of water through curved rectangular microchannels with different aspect ratios and curvature radii for Re numbers ranging from 10 to 600. Similar to earlier work their experimental results indicated that the classical Navier–Stokes equations were applicable for incompressible laminar flow passing through curved microchannels also. Geometrical parameters, such as aspect ratio and curvature ratio, were found to have important effects on fluid flowing through curved rectangular microchannels.

The various researchers who use experimental analysis for the evaluation of heat transfer and fluid flow parameters have used either single microchannel or array of microchannels. The important finding in these experimental studies was the usage of conventional Navier- Stokes

equation and Poiseuille flow theories for single-phase liquid flow in microchannels. In single-phase flow of rectangular microchannels, classical correlations based on conventional sized channels, also accurately predict the thermal behavior.

2.1.2 Computational /Numerical study

It is seen that there are two broad aspects of using computational and numerical studies: first to find a starting point for any laboratory analysis, or to assist in understanding experimental data, second to explore mechanisms/combinations that are not readily studied by experimental means. Numerous works are available who analyze heat transfer and fluid mechanisms for a single or array of microchannels. Fedorov et al. (1999) investigated the flow and conjugate heat transfer in the microchannel-based heat sink for electronic packaging applications. For solving conjugate problems, incompressible laminar Navier-Stokes equations of motion were employed as the governing conservation equations. The combined convection-conduction heat transfer in the microchannel produced a complex three-dimensional heat flow pattern with large, longitudinal, upstream directed heat recirculation zones in highly conducting silicon substrate. The heat sink used in the analysis was compact and capable of dissipating a significant thermal load of heat flux of the order of 100 W/cm^2 with a relatively small increase in the package temperature, if operated at the Reynolds numbers above 150.

Qu and Mudawar (2002) numerically studied the three-dimensional fluid flow and heat transfer in a rectangular micro-channel heat sink using water as the cooling fluid. The heat sink considered in their analysis was of 1cm^2 silicon wafer, with micro-channels width of $57 \mu\text{m}$ and depth of $180 \mu\text{m}$, separated by a $43 \mu\text{m}$ wall. They used a numerical code based on the finite difference method and the SIMPLE algorithm to solve the governing equations. It was observed that the temperature rise along the flow direction in the solid and fluid regions can be approximated as linear. The heat flux and Nusselt number had much higher values near the channel inlet and vary around the channel periphery, approaching zero in the corners. The highest temperature was encountered at the heated base surface of the heat sink immediately above the channel outlet.

Ryu et al. (2002) numerically analyzed the thermal performance of a microchannel heat sink by varying pumping power, channel depth, channel width, and the fin thickness, under the

constraint that the channel aspect ratio should be less than 10. They also concluded that among various design variables, the channel width appears to be the most crucial quantity in dictating the performance of a microchannel heat sink. The important conclusion of their work was the optimal channel shape which remains relatively unchanged even when the channel number variation was large. Zhao and Lu (2002) presented an analytical and numerical study on the heat transfer characteristics of forced convection across a microchannel heat sink. They used two analytical approaches: the porous medium model and the fin approach. They found that whereas the porous medium model predicted the existence of an optimal porosity for the microchannel heat sink, the fin approach predicted that the heat transfer capability of the heat sink increases monotonically with the porosity.

The flow behavior and heat transfer performance of a microchannel heat sink based on two important parameters: Hydrodynamic and Thermal entrance lengths was numerically studied by Azimian et al. (2004). The numerical solution of the problem was based on a finite difference approach using a control volume with staggered grid system. The SIMPLE algorithm was applied to the problem, and convection terms were estimated using QUICK method. The comparison of the Newtonian and non-Newtonian results showed that for shear thinning fluids, the pressure drop could reduce up to 45%, while for shear thickening fluids, it can increase up to 48%. The same comparison for the Nusselt number showed about a 160% increase with shear thinning fluids and a 43% decrease with shear thickening fluids. The numerical simulation of forced convection heat transfer occurring in silicon-based microchannel heat sinks was studied by Li et al. (2004). Their results indicate that thermo physical properties of the liquid can significantly influence both the flow and the heat transfer in the microchannel heat sink. The bulk liquid temperature was shown to vary in a quasi-linear form along the flow direction for high fluid flow rates, but similar results were not obtained for low flow rates. Using the detailed local heat transfer coefficient calculation thermal entrance length was also obtained. Seetharamu et al. (2004) numerically modeled the Double layer counter flow (DLCF) micro-channel heat sinks with rectangular channels using FEM. The dimensions of DLCF such as height of channel, H_c , width of channel, W , and width of fin, W_f were considered as variables selected for optimization in determining minimal thermal resistance using Genetic Algorithms (GA) and Box optimization method for a fixed pumping power. They compared these two optimization

techniques on microchannel heat sinks and observed GA performance marginally better than the Box optimization method. However GA required far less iteration time compared to Box optimization method to complete the optimization.

Analytical and numerical studies on thermal models for cooling microscale electronic processor chips through forced and natural convection heat sinks was studied by Kulkarni and Das (2005). Their prediction of these model equations closely matched the performance of Pentium III heat sink. They used Knudsen number calculation to verify that the continuum theory still held for their designed microchannels. A commercial computational fluid dynamics code Fluent was used by them to refine the analysis. Wang et al. (2005) numerically investigated the friction factors of single-phase microchannel with various roughness elements using rectangular, triangular, and elliptical shape. The two-dimensional numerical solution showed significant influence of surface roughness including the height, size, and spacing of the roughness elements on the Poiseuille number. The fluid temperature effects were also investigated numerically for the rough channels.

Bahrami et al. (2006) investigated the pressure drop of fully developed, laminar, incompressible flow in smooth mini- and microchannels of arbitrary cross-section. A compact approximate model as a function of geometrical parameters of the cross-section, i.e., area, perimeter, and polar moment of inertia was proposed by them to predict the pressure drop for a wide variety of shapes. An excellent agreement of results was experienced between the predicted model and experimental data, collected by several researchers.

The analysis of conjugate heat transfer and two phase flow in a heat sink containing multiple microchannels was investigated computationally by Kelkar and Patankar (2006). They solved one-dimensional momentum and energy conservation equations to find flow and thermal behavior in each microchannel. The important physical effects in the two-phase flow regime: acceleration pressure drop and the boiling enhancement, strongly affect the relative behavior of the individual microchannels. The laminar convective heat transfer in the entrance region of microchannels of rectangular cross-section under circumferentially uniform wall temperature and axially uniform wall heat flux thermal boundary conditions was numerically investigated by Lee and Garimella (2006). Both local and average Nusselt numbers were presented in graphical form as a function of the dimensionless axial distance and channel aspect ratio. Generalized

correlations for the design and optimization of microchannel heat sinks and other microfluidic devices, were also proposed by them for predicting Nusselt numbers. The proposed correlations were compared with other conventional correlations and with available experimental data, and showed good agreement.

Another important numerical investigation was made Bahrami et al. (2007) by analyzing the pressure drop of fully developed, laminar, single-phase flow in singly connected microchannels of arbitrary cross-section. They showed that for constant fluid properties and flow rate in fixed cross-section channels, the Poiseuille number was only a function of geometrical parameters of the cross-section, i.e., perimeter, area, and polar moment of inertia. The model was validated with experimental data for rectangular, trapezoidal, and triangular microchannels. Chen (2007) presented numerical analysis of forced convection heat transfer in microchannel heat sinks based on the Forchheimer–Brinkman-extended Darcy equation for the fluid flow. Analytical expressions for the velocity and temperature profiles for a simpler flow model were also presented. The governing important parameters include the channel aspect ratio, inertial force parameter, porosity, and the effective thermal conductivity ratio. Ngoma and Erchiqui (2007) numerically demonstrate the influence of the slip coefficient, the heat flux and the pressure difference on flow velocity, local temperature and Nusselt number. The combined effect of pressure-driven flow and electro-osmosis was taken into account. Electric potential, liquid flow and thermal characteristics were determined using the Poisson–Boltzmann, the modified Navier–Stokes and the energy equations for a hydrodynamic and thermal steady fully developed laminar flow for an incompressible liquid.

Numerical optimization of microchannel heat sink with the help of surrogate analysis and evolutionary algorithm was investigated by Husain and Kim (2008). They studied design variables related to the microchannel depth, width and fin width. Temperature dependent thermal properties of water (as a coolant) in the silicon microchannels were considered. Using the numerically evaluated objective function, polynomial response surface was constructed for each objective function.

Bharti et al. (2009) investigated numerically the electro-viscous effects in steady, fully developed, pressure-driven flow of power-law liquids through a uniform cylindrical

microchannel by solving the Poisson–Boltzmann and the momentum equations using finite difference method. They observed that electro-viscous resistance reduces the velocity adjacent to the wall, relative to the velocity on the axis. The magnitude of the electro-viscous effect on the apparent viscosity, as measured by the ratio of the apparent and physical consistency indices, decreases monotonically as the power-law index increases. Rij et al. (2009) analyzed numerically the frictional and convective heat transfer characteristics of rarified flows in rectangular microchannels. Numerical results in the form of Poiseuille and Nusselt numbers, were observed to be significant functions of aspect ratio, Knudsen number, slip model parameters, Brinkman number, and Peclet number.

Mathew et al. (2010) analyzed the thermal performance of parallel flow microchannel heat exchangers subjected to constant external heat transfer. It was noticed that irrespective of the heat capacity ratio, for a specific NTU, external heating always decreases and increases the effectiveness of the hot and cold fluid. An opposite trend in the effectiveness of the fluids was observed when subjected to external cooling. It was also observed that at a given NTU, reduction in heat capacity ratio improve the effectiveness of the fluids. Kosar (2010) studied the heat and fluid flow in microchannels of different substrate thicknesses and materials like Polyimide, Silica Glass, Quartz, Steel, Silicon, Copper. A commercial package COMSOL MULTIPHYSICS was used for the analysis. Nusselt number correlation for fully developed laminar flow was also developed as a function of two dimensionless parameters, namely Biot number Bi and relative conductivity k^* , to take the conduction effects of the solid substrate on heat transfer into account. The thermal entrance region of trapezoidal microchannels for hydrodynamically fully developed, single-phase, laminar flow with no-slip conditions was investigated by McHale et al. (2010). A finite-volume approach was used for the Three-dimensional numerical simulations for a wide range of aspect ratios. Results of local and average Nusselt numbers were presented as a function of dimensionless length and aspect ratio. The effect of Prandtl number upon the thermal entrance condition was also explored.

Mohammed et al. (2011) performed numerical simulations to solve the three-dimensional steady and conjugate heat transfer governing equations using the Finite-Volume Method (FVM) for water flow based Micro channel heat sink (MCHS). The performance of MCHS for different channel shapes such as zigzag, curvy, and step microchannels was compared with straight and

wavy channels. Temperature profile, heat transfer coefficient, pressure drop, friction factor, and wall shear stress was used as parameters for analyzing the performance of MCHS. As expected the pressure drop penalty for all the studied channel shapes was higher than the conventional straight MCHS. Another numerical analysis using finite volume method for the geometric optimization of an array of circular, square and isosceles right triangle cross-sections of channels was made by Salimpour et al. (2011). They proposed correlations that can predict the optimal hydraulic diameter and dimensionless heat transfer per unit volume for the microchannels. Among the different geometries used in their study, square cross section offered the maximum efficiency for a given volume and pressure drop.

Shanglong et al. (2012) studied the effects of structures in four different microchannel heat sinks for electronic chips cooling. They classify the structures as TP_A , TP_B , TP_C and TP_D ; TP_A is the ordinary parallel structure heat sink; TP_B is the reticular structure heat sink with vertical and connective microchannel; TP_C is the toroidal heat sink which includes rounding microchannel; TP_D is the tree-like heat sink enlightened from the fractal pattern of foliage's venation with the 45 degree's angle between offshoot and trunk. Numerically analyzed results indicate that the heat sink with tree-like microchannel can take away the most heat at the same inlet flow rate. The temperature of electronic chip cooled by TP_D structure is the minimum and uniform. The minimum heat flux of the fluid is at the center of channel. The heat flux of the fluid lineally increases with the location varies from the channel center to channel wall, which is more than 500%, indicating topological structure in microchannel heat sink has significant influence on electronic chips cooling.

It was observed from the numerical and computational studies performed by various researchers that the conventional Navier- Stokes and energy equation can be used for the analysis of single-phase flow within microchannels. Some of the researchers have even used commercial Computational Fluid Dynamics softwares for the investigation. It was important to see that one of the outcome of these analyzes was developing heat transfer parameter correlations within the studied range.

Table 2.1 Selected literatures for single-phase liquid flow in microchannel passages

Author	Year	Fluid/ Shape*	D_h (μm)	Re	f/Re	laminar flow**	Aspect ratio considered
Baviere et al.	2006	water / R	14–593	0.1–7985	21.5–71.8	Y	Y
Bahrami et al.	2007	water / R,C,Tr,Ti	172–520	2–5272	14.0–51.9	Y	Y
Chein et al.	2009	water / R	100	1.9–3237	41.2–33.3	Y	Y
Hsieh et al.	2004	water / R	146	45–969	14.6–51.2	Y	Y
Judy et al.	2002	water, methanol, isopropyl / C, R	14–149	7.6–2251	12.9–20.3	Y	Y
Lee and Garimella	2006	water / R	318–903	558–3636	ID	Y	Y
Lee et al.	2005	water / R	85	119–989	19.4–43.6	Y	Y
Peng et al.	1995	methanol / R	311–646	1530–13455	ID	N	Y
Qu and Mudawar	2002	water / R	349	137–1670	12.1–33.4	Y	Y
Qu et al.	2003	water / Tr	62–169	94–1491	ID	N	Y
Toh et al.	2002	water / C	728	16.6–37.5	ID	N	NA
Tuckerman and Pease	1981	water / R	92–96	291–638	14.0–20.8	Y	Y
Steinke et al.	2005	water / R	112–520	102–638	ID	Y	Y
Adams et al.	1997	water / Tr	102–1090	2600–23000	131	ID	NA
Warrier et al.	2002	FC-84/R	750	118–1109	ID	Y	NA
Zhuo et al.	2006	water / Tr,Ti	27– 119	17– 392	ID	Y	Y
Mohammed et al.	2011	water / R	232–566	118–889	ID	Y	Y
Chiu et al.	2011	water / R	112–312	137–1113	12.0–27.7	Y	Y
Chu et al.	2012	water / curved R	72– 342	10–600	ID	Y	Y

NA =Not Applicable, ID =Insufficient Data; C =circular, R =rectangular, Tr =trapezoid, Ti =triangle, Y =Yes, N =No

2.1.3 Studies of the effects of manifolds/flow arrangements

Wang et al. (2008) investigated three types of inlet/outlet configurations on flow boiling instabilities in parallel microchannels, having a length of 30 mm and a hydraulic diameter of 186 μm . They classify the three configurations as Type-A, Type-B and Type-C. Fluid flow entering

to and exiting from the microchannels with the Type-A connection was restricted because the inlet and outlet conduits were perpendicular to the microchannels. The fluid flow had no restriction in entering to and exiting from the microchannels with the Type-B connection. In the Type-C connection, fluid flow was restricted in entering each microchannel but was not restricted in exiting from the microchannels. They observed that amplitudes of temperature and pressure oscillations in the Type-B connection were much smaller than those in the Type-A connection under the same heat flux and mass flux conditions. On the other hand, nearly steady flow boiling exists in the parallel microchannels with the Type-C connection under the experimental conditions. They recommend Type-C configuration high-heat-flux microchannel applications. Their finding leads to the conclusion that experimental data of pressure drop were higher and heat transfer coefficients were lower for boiling flow at high vapor quality in microchannels than those predicted from correlation equations for boiling flow in macro channels, due to local dry-out.

Chein and Chen (2009) analyzed the fluid flow and heat transfer in microchannel heat sinks numerically by solving the three-dimensional governing equations for both fluid flow and heat transfer using the finite volume scheme. Due to different inlet/outlet arrangements used in their work, the resultant flow fields and temperature distributions inside these heat sinks were observed to vary under a given pressure drop. In contrast to earlier works, the computational domain taken by them was the entire heat sink including the inlet/outlet ports, inlet/outlet plenums, and microchannels. For all the heat sinks studied by them, the highest heat sink temperature was observed to be at the edge of the heat sink since there was no heat dissipation by fluid convection.

Dang et al. (2010) carried out a numerical and experimental study on the effect of flow arrangement for the heat transfer behaviors of a microchannel heat exchanger. They observed the heat flux obtained from the counter-flow arrangement was always 1.1 to 1.2 times higher than that obtained from the parallel-flow one. The results obtained from the numerical analyses were in good agreement with those obtained from the experiments, with the discrepancies of the heat transfer coefficient estimated to be less than 10%.

2.1.4 Gaps in literature

A detailed literature survey indicates that the microchannels are widely used as a cooling solution for the thermal management of electronics devices, integrated circuits in specific. The heat transfer and fluid flow within the microchannels have been studied in detail using numerical methods and experimental techniques. It is generally seen that all the heat transfer correlations that have been developed correspond to microchannel region only and not for the entire heat sink.

Jones et al. (2008) studied the flow distribution in a silicon microchannel heat sink using infrared micro-particle image velocimetry. The design includes etching of the inlet and outlet manifolds into the substrate. It was observed that at lower flow rates ($Re=10.2$) small flow mal-distribution (only 2.4%) occurs while at higher flow rates, flow mal-distribution becomes significant. The numerical results showed that changing the manifold shape to a rectangular design results in a lower flow mal-distribution (4.7%) compared to the original manifold design (33.5%). The pressure drop between the inlet and exit ports was 32.7% greater for the original manifold compared to the rectangular design.

Chein and Chen (2009) numerically analyzed the fluid flow and heat transfer in MCHSs, consider the entire heat sink as computational domain, including the inlet/outlet ports, inlet/outlet plenums, and microchannels. For all of the heat sinks investigated by them (U-, V-, I-, N-, D-, and S-types), the highest heat sink temperature was observed at the edge of the heat sink. They found that the V-type heat sink has the best performance among the heat sinks studied. For the heat sinks with horizontal fluid supply and collection, i.e., the I-, N-, D-, and S-type heat sinks, they observed velocity mal-distribution to be responsible for greater degradation of performance compared to the heat sinks with vertical fluid supply and collection, i.e., the U and V-type heat sinks. They, however, neither explained the term velocity mal-distribution in detail nor quantified it.

Alfieri et al. (2012) studied a fundamental hydrothermal investigation of the next generation interlayer integrated water cooled three-dimensional (3D) chip stacks, with high volumetric heat generation. They showed that conventional porous medium approach was beneficial in terms of computational cost reduction due to the assumption of fully developed flow, but it results to underestimation of pressure losses in the simulated chip stack, especially

near the channel inlet. They used extended porous medium model which has both developing and fully developed regions, leading to better representation of experiments and therefore improved ability to capture the thermal gradients along the cooling structures. Numerous cooling structure geometries such as microchannels, micro pin-fin arrays and parallel plates were considered, and the correlations were also developed through a detailed modeling of conjugate heat transfer in reduced, representative geometries. The main focus of the work was to understand the complex physics of the entrance region of the cooling microcavities in order to assess its importance.

Some of the above mentioned work has illustrated the importance of a properly designed manifold to maximize the performance of a microchannel heat sink. The microchannels along with the inlet and outlet plenums with different entry and exit flow options have not been studied exhaustively with respect to heat transfer and fluid flow parameters. Similar to heat exchangers where flow patterns have an effect on the performance/effectiveness, the microchannels heat sink's thermo-hydraulic performance also depends on the entry and exit flow conditions. The fluid before entering the microchannels normally has to pass through the plenum so as to have fully developed flow. Researchers have studied the effect of microchannel size (aspect ratio) but the effect of plenum size (aspect ratio) on the performance of the heat sink has not been studied till date. Heat transfer correlations are available in the literature for the microchannels alone, but taking into account the flow (inlet and outlet) arrangements, correlations for the complete heat sink unit have not been developed. The practical option on the basis of total pressure drop for using a particular flow arrangement for the liquid cooling of heat sink also needs to be explored.

2.2 Problem formulation

Based on the literature review, it was decided to investigate the effect of inlet/outlet plenums on fluid flow and heat transfer within a microchannel heat sink, as this type of analysis will help enhance the overall performance of MCHS. The selected heat sink is made up of copper with array of microchannels on it. The fluid before flowing through the microchannels is passed through the inlet plenum and comes out of the MCHS through the outlet plenum. In the experimental set-up provisions are made to achieve different flow paths through the inlet plenum, microchannels and outlet plenum. Different preliminary experimental results and flexibility of using a single top cover of MCHS leads to the selection of three different flow

configurations, called U-type, P-type and S-type. In line with the microchannel aspect ratio, plenum aspect ratio terminology has been introduced for the MCHS as a complete unit. The aspect ratio of the microchannel and plenums under different operating conditions (three heat inputs, six Reynolds number, etc) are proposed to be studied using both experimental and computational techniques. The different arrangements of fluid flow (three in the present research) on the performance of the MCHS are also proposed to be studied using both the experimental and computational approach.

2.3 Objectives

The points discussed in the scope have been carefully analyzed in evolving the following objectives of the present investigation:

- To investigate the thermo-hydraulic performance of microchannel heat sink for P-type, U-type and S-type flow configurations.
- To carry out an experimental study to analyze the effect of different aspect ratios, Reynolds number, and heat input on the thermo-hydraulic performance of MCHS and to develop correlations for Nusselt number for different flow arrangements.
- To analyze flow phenomena at the entrance and exit regions of different types of microchannel configurations on convective heat transfer and pressure drop.
- To simulate convective heat transfer and fluid flow trajectories in microchannel heat sinks of different configurations including the effects of flow phenomena at the entrance and exit regions.
- To study the effect of temperature on variation of viscosity on the flow distributions at the inlet and exit regions of the micro-channel heat sink.

CHAPTER 3

EXPERIMENTAL SET-UP AND PROCEDURE

3.1 Flow arrangements in micro channel heat sink

It is well known that the heat can be a problem in many environments and that the overheating can lead to failures of components such as integrated circuits (e.g. a central processing unit (CPU) of a computer). Heat sinks are common devices used to prevent overheating and mainly rely on the dissipation of heat from the device. However, dissipating heat using gas, such as air, is difficult because of the poor thermal conductivity of gases. Gases also have a low heat capacity, which causes them to heat quickly, which retards the rate of heat absorption by decreasing the temperature difference between the gas and the heat sink. Conventional heat sinks have a limited amount of surface area that can be accommodated in a give volume, and as a result, a conventional heat sink must be large in order to provide the necessary convection surface area. In general, in heat sink designs for cooling a heat source on a substrate, the heat sink dimensions extend substantially perpendicular to the substrate and heat source.

Microchannels with liquid coolants can provide very high convective heat transfer rates. In theory, high convection rates of microchannels can overcome the poor thermal conductivity issue. However, there are two major obstacles for practical implementation of a microchannel concept in a heat sink application. First, microchannels create large resistance to fluid flow and this resistance increases as the length (in the direction of flow) increases. Second, the low heat capacity of gases means that they heat up quickly and become ineffective in dissipating heat. The second obstruction can be eliminated by using liquid flow within microchannels. In a micro channel heat sink, there are arrangements of a plurality of micro-fins spaced apart to form micro channels through which a gas/liquid can flow.

Liquid-cooled micro channel heat sinks (MCHSs) and coolers have been shown to be a very effective way to remove high heat load. A large heat transfer coefficient can be achieved by reducing the channel hydraulic diameter. It is observed that the ability of liquid cooled heat sinks to dissipate heat is determined by conduction in solids and convection in fluids. Generally heat convection is an important factor for reducing thermal resistance, when a high thermal conductive material is used to manufacture a heat sink. With constant wall temperature or

constant wall heat flux, under fully developed laminar flow, the Nusselt number is constant. The heat transfer coefficient ‘h’ can be calculated as:

$$h = Nu \frac{k}{D_h} \quad (3.1)$$

$$\Rightarrow h \propto \frac{1}{D_h} \quad (3.2)$$

Reynolds number characterizes the significance of inertia and viscous effects and is defined as

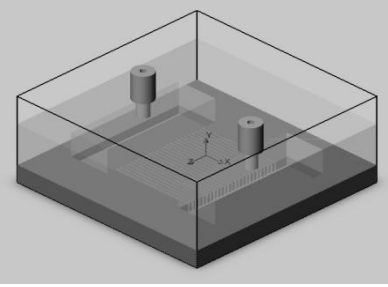
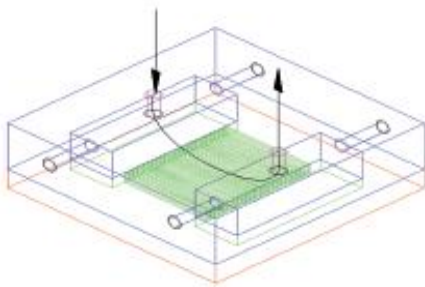
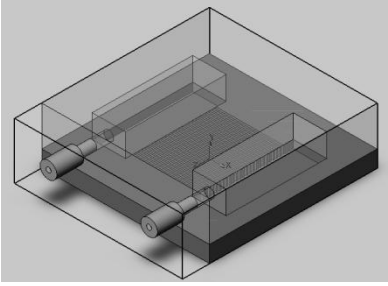
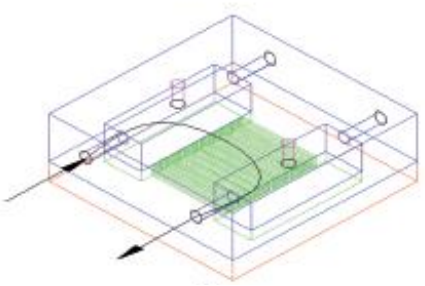
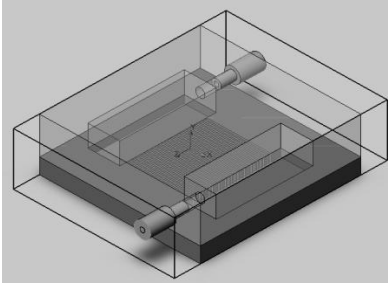
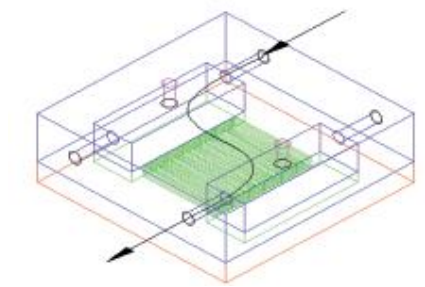
$$Re = \frac{\rho V_f D_h}{\mu} \quad (3.3)$$

In small channels as hydraulic diameter ‘ D_h ’ is less, at uniform density and viscosity of fluid Reynolds number is mostly observed to be laminar. In a confined geometry the small flow rate within microchannels produces laminar (smooth) flow, which results in a heat transfer coefficient inversely proportional to the hydraulic diameter. In other words, the narrower the channels in the heat sink, the higher the heat transfer coefficient.

In order to meet the requirements of various cooling applications of electronic devices, different methods of flow arrangements have been proposed for fluid flow within the inlet and outlet plenums of the MCHS. As the different flow arrangements are expected to affect the fluid flow and heat transfer characteristics within the heat sink, it is important to investigate the effect of inlet/outlet plenum flow arrangements for increased understanding of MCHS performance. It was also observed that flow instability at the entrance and exit regions of microchannel needs to be understood fully for efficient design of MCHSs.

In dealing with liquid flows in minichannels and microchannels in the absence of any wall surface effects, such as the electrokinetic or electroosmotic forces, the flow is not expected to experience any fundamental changes from the continuum approximation employed in macro-fluidic applications. Generally microchannel heat sink has an inlet plenum, from where the fluid is supplied to an array of microchannels, and an outlet plenum, where it is collected after being made to pass through the microchannels. Due to the requirements for various applications, the fluid inlet and outlet of the microchannel can be arranged in a number of ways. Owing to the small size of channel dimensions, the entrance and exit effects will significantly affect the heat transfer characteristics of flow field in the channels. In order to understand the performance of the heat sink as a single unit, it is necessary to study the effect of inlet and outlet flow

Table 3.1: Different flow arrangements within MCHS

P-type flow arrangement		
U-type flow arrangement		
S-type flow arrangement		

arrangements on heat transfer characteristics of the microchannels. Detailed understanding of the inlet and outlet flow arrangements thus becomes necessary to understand the influential parameters that enhance the heat sink performance.

In the present research, experimental and computational results were obtained to study the effect of different entrance and exit flow arrangements within a copper MCHS. Based on the space consideration, generally limited inlet and outlet flow arrangements are practicable. In the present research work, three types of flow arrangements—U-type, S-type and P-type are studied with the main focus on the effect of flow arrangements on heat transfer and pressure drop. Here P stands for perpendicular, U stands for U-shape and S stands for serpent shape. In order to

achieve these flow entries and exits of water to and from the plenum, suitable piping arrangements were employed. The effect of the size of inlet and outlet plenums on thermo-hydraulic performance of micro channel heat sink will also be analysed.

3.2 Inlet and outlet flow effects

As it is observed that counter flow and parallel flow arrangements have different thermal performance in heat exchangers, different inlet and outlet flow arrangements in microchannels have different effects on the heat transfer performance of a heat sink. For the same flow of fluid within the same micro channels, but different inlet and outlet flow conditions, the performance will vary. Based on the same concept, here instead of analysing the performance of only microchannels, the whole heat sink (inlet plenum, array of microchannels, outlet plenum) is chosen. The reasons for instability of flow within the microchannels are also investigated.

3.3 Pressure drop in single phase liquid flow

Kandlikar et al. (2006) in their book described the procedure how one-dimensional flow of an incompressible fluid in a smooth circular pipe forms the basis for pressure drop analysis in internal flows. The force due to pressure difference dp is balanced by the frictional force due to shear stress τ_w at the wall and is given by

$$\left(\frac{\pi}{4}D^2\right)dp = (\pi D dx)\tau_w \quad (3.4)$$

The pressure gradient and the wall shear stress are thus related by the following equation:

$$\left(\frac{dp}{dx}\right) = \frac{4\tau_w}{D} \quad (3.5)$$

For Newtonian fluids, the wall shear stress τ_w is expressed in terms of the velocity gradient at the wall:

$$\tau_w = \mu \left. \frac{du}{dy} \right|_w \quad (3.6)$$

where μ is the dynamic viscosity. The Fanning friction factor f represents the momentum transfer process of fluid flow

$$f = \frac{\tau_w}{\left(\frac{1}{2}\right)\rho u_m^2} \quad (3.7)$$

where u_m is the mean flow velocity in the channel. Rearranging the Eqs (3.5) and (3.6) to calculate the frictional pressure drop Δp over a length L :

$$\Delta p = \frac{2 f \rho u_m^2 L}{D} \quad (3.8)$$

For non-circular flow channels, the D in Eq. (3.8) is replaced by the hydraulic diameter D_h given by:

$$D_h = \frac{4A_c}{P_w} \quad (3.9)$$

where A_c is the flow-channel cross-sectional area and P_w is the wetted perimeter. For a rectangular channel of sides a and b , D_h is given by

$$D_h = \frac{4ab}{2(a+b)} = \frac{2ab}{(a+b)} \quad (3.10)$$

The friction factor f which depends on the flow channel geometry can also be written in the following form:

$$f = \frac{P_o}{R_e} \quad (3.11)$$

where P_o is the Poiseuille number and R_e is Reynolds number.

It was observed that the pressure gradients in small diameter channels are relatively high due to low flow lengths. In case of microchannels the length of channel in the developing region forms a major portion of the flow length. Apparent friction factor f_{app} is used to express the pressure drop for the developing region. It represents an average value of the friction factor over the flow length between the entrance section and the location under consideration and is expressed as:

$$f_{app} = \frac{\Delta p \times D_h}{2 \rho u_m^2 x} \quad (3.12)$$

Kandlikar et al. (2006) expressed the pressure drop in terms of the incremental pressure drop as:

$$\Delta p = \frac{2(f_{app} R_e) \mu u_m x}{D_h^2} = \frac{2(f R_e) \mu u_m x}{D_h^2} + K(x) \frac{\rho u_m^2}{2} \quad (3.13)$$

where $K(x)$ is incremental pressure defect given by:

$$K(x) = (f_{app} - f) \frac{4x}{D_h} \quad (3.14)$$

3.4 Heat transfer and fluid flow performance

The heat transfer and fluid flow performance of a heat sink largely depend on type of cooling fluid used. Figure 3.1 and 3.2 [Kandlikar et al. (2006)] shows the variation of heat transfer coefficient and pressure gradient with channel size for fully developed laminar flow for water and air respectively. Figure 3.1 shows the variation of h for flow of water and air with channel hydraulic diameter and it was observed that there is an increase in h with a reduction in channel size. Figure 3.2 shows that the pressure gradient increases with a reduction in the channel size. It was observed that while designing the coolant flow passages for high flux heat removal, the balance between the heat transfer rate and pressure drop becomes an important issue.

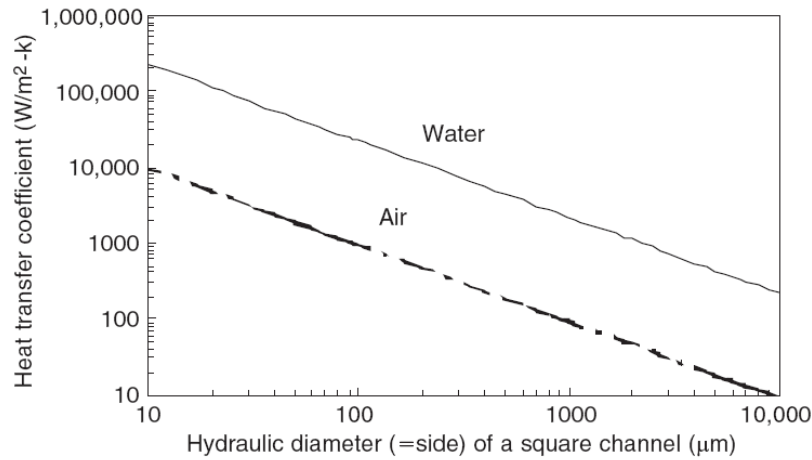


Fig. 3.1: Variation of the heat transfer coefficient with channel size for fully developed laminar flow of air and water.

As predicted by classical theory the Nusselt number in fully developed laminar flow is expected to be constant. The Nusselt number depends on the channel geometry and the wall heat transfer boundary condition. It was observed that in a microchannel region, there is a linear increase in Nusselt number with flow Reynolds number. Due to the relatively short lengths employed in microchannels, the influence of entrance region cannot be neglected. The entrance region effects become more significant at higher Reynolds numbers, in part explaining the increasing Nusselt number trend with Reynolds number.

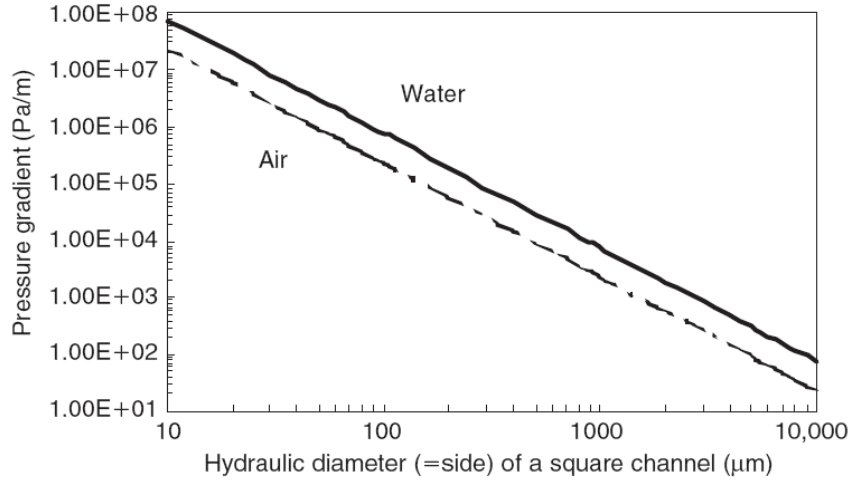


Fig. 3.2: Variation of pressure gradient with channel size for fully developed laminar flow of air and water.

Steinke and Kandlikar (2005) showed that the uncertainty in the Nusselt number calculation is largely dependent on the uncertainties in the experimental measurements for a rectangular channel of base width a and height b :

$$U_{Nu} = Nu \left[\left(\frac{U_{k_f}}{k_f} \right)^2 + \left(\frac{U_I}{I} \right)^2 + \left(\frac{U_V}{V} \right)^2 + x \left(\frac{U_{T_s}}{T_s} \right)^2 + y \left(\frac{U_{T_i}}{T_i} \right)^2 + z \left(\frac{U_{T_o}}{T_o} \right)^2 + \alpha \left(\frac{U_L}{L} \right)^2 + \beta \left(\frac{U_a}{a} \right)^2 + \delta \left(\frac{U_b}{b} \right)^2 + \varepsilon \left(\frac{U_{\eta_f}}{\eta_f} \right)^2 \right]^{1/2} \quad (3.15)$$

where I – current, V – voltage, T_s – surface temperature, T_i – fluid inlet temperature, T_o – fluid outlet temperature, L – flow length, a and b – cross-section dimensions, x , y , α , β , δ , ε are the constants and η_f – fin efficiency. The uncertainties in the measurement of the surface temperature and flow channel dimensions play a critical role in the overall uncertainty. More details are available in Appendix-A

3.5 Geometrical parameters

In a microchannel, the following geometrical parameters are important:

- Microchannel length
- Microchannel width
- Microchannel depth
- Inlet plenum length
- Inlet plenum width

- Inlet plenum depth
- Outlet plenum length
- Outlet plenum width
- Outlet plenum depth
- Length of whole heat sink
- Width of whole heat sink
- Depth of whole heat sink

Effects of geometry on the micro-flow are studied for analyzing the performance. Geometric characteristics that consider the above twelve points are hydraulic diameter and different aspect ratios. The aspect ratio is the ratio of the one dimension of a shape to another dimension. Different researchers have used different definition of aspect ratio and in literature generally the channel aspect ratio effect is studied. It is known that the height-to-width ratio has great effect on the flow friction and heat transfer in the rectangular microchannels.

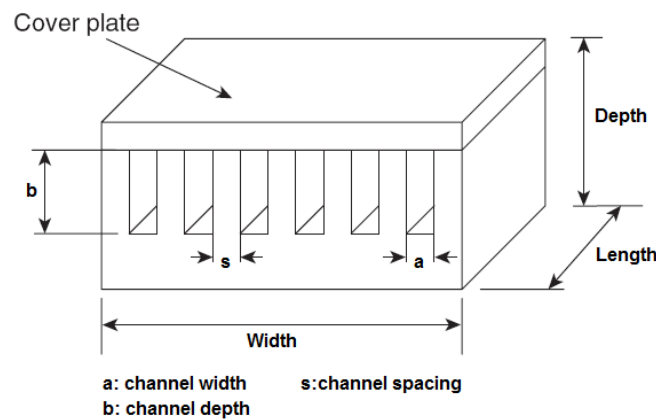


Fig. 3.3: Geometrical parameters used by various researchers (without plenums) [Kandlikar et al. (2006)]

3.6 Design of microchannel heat sink

Design of a heat sink is an important process and one need to evaluate a number of parameters which affect the overall performance of the system along with the thermo-hydraulic performance. Some of the important parameters include: flow condition of fluid, thermal budget and external conditions which surround the heat sink. Lee Seri (1995) presented the following list of design constraints for a microchannel based heat sink:

- induced approach flow velocity
- available pressure drop

- cross sectional geometry of incoming flow
- amount of required heat dissipation
- maximum heat sink temperature
- ambient fluid temperature
- maximum size of the heat sink
- orientation with respect to the gravity
- appearance and cost

The designer can control and optimize the following parameters for analyzing the performance of microchannel heat sink:

- Microchannel height
- Microchannel length
- Microchannel thickness/spacing
- Number/density of microchannels
- Microchannel shape/profile
- Base plate thickness
- Flow arrangements
- Plenum dimensions
- Microchannel heat sink material, etc.

It is seen that in order to achieve an optimum thermal performance of the heat sink, most of the parameters mentioned above are interdependent of the others. It is also observed that the impact of one parameter that has on the performance of a heat sink cannot be generalized, or even foreseen without concurrently considering the consequences exhibited in the other parameters. In order to analyse the thermo hydraulic performance of microchannel heat sink, to be used as thermal management solution for cooling of integrated circuits, a continuous design improvement at every step was carried out in order to achieve the final design.

3.7 Fabrication of the test piece

In order to study the heat transfer and fluid flow within a microchannel heat sink, a copper test piece was manufactured. The top and front views of one of the test pieces are shown in Figure 3.4. The test piece consists of an array of microchannels in parallel having individual width of

330 μm or 530 μm with constant depth of 2.5 mm. The rectangular micro-channels were fabricated using a high-accuracy CNC wire-cut electrical discharge machine (EDM). Wire-Cut EDM from Sodick Technologies India Private Limited, Model: AG600L was used. The test microchannel heat sink has a cover plate which is made from a transparent polycarbonate acrylic sheet and is bolted to the copper heat sink. The copper heat sink was heat insulated at the back by an asbestos sheet covered with glass wool packaging. For the experimental investigation six test pieces were manufactured leading to 18 combinations (Table 3.2).

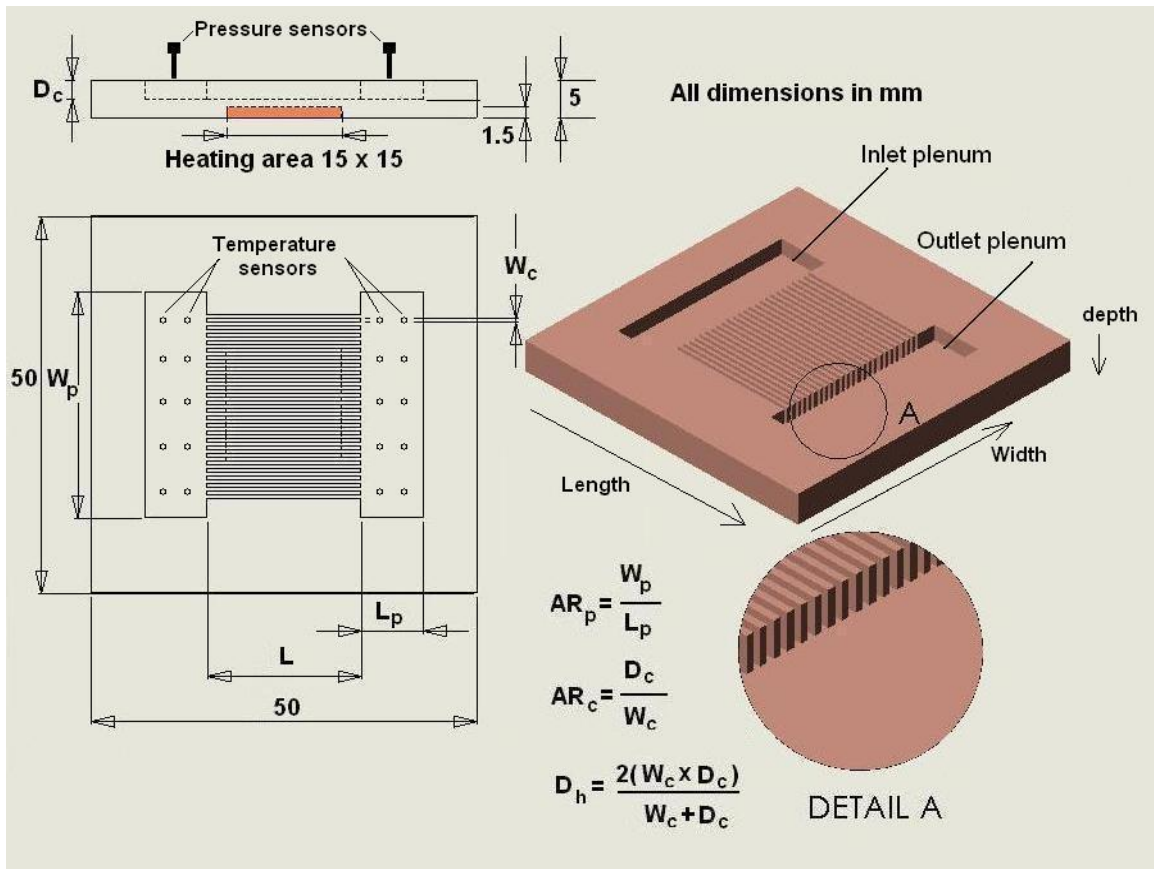


Fig. 3.4: Test piece with dimensions

Table: 3.2: Microchannel geometrical parameters

Test Piece	Microchannel width, W_c (μm)	Microchannel depth, D_c (mm)	Microchannel length, L_c (mm)	Plenum width, W_p (mm)	Plenum depth, D_p (mm)	Plenum length, L_p (mm)	Flow arrangement
1	330	2.5	20	30	2.5	8	P-type
2	330	2.5	20	30	2.5	10	P-type
3	330	2.5	20	30	2.5	12	P-type
4	330	2.5	20	30	2.5	8	U-type
5	330	2.5	20	30	2.5	10	U-type
6	330	2.5	20	30	2.5	12	U-type
7	330	2.5	20	30	2.5	8	S-type
8	330	2.5	20	30	2.5	10	S-type
9	330	2.5	20	30	2.5	12	S-type
10	530	2.5	20	30	2.5	8	P-type
11	530	2.5	20	30	2.5	10	P-type
12	530	2.5	20	30	2.5	12	P-type
13	530	2.5	20	30	2.5	8	U-type
14	530	2.5	20	30	2.5	10	U-type
15	530	2.5	20	30	2.5	12	U-type
16	530	2.5	20	30	2.5	8	S-type
17	530	2.5	20	30	2.5	10	S-type
18	530	2.5	20	30	2.5	12	S-type

In the present research, the hydraulic diameter, aspect ratio for channel (AR_c) and plenum (AR_p) are defined as:

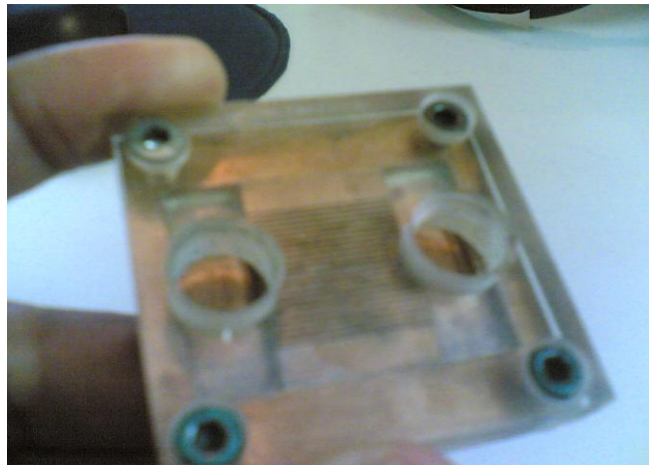
$$D_h = \frac{2(W_c \times D_c)}{W_c + D_c} \quad (3.16)$$

$$AR_c = \frac{\text{microchannel depth}}{\text{microchannel width}} \quad (3.17)$$

$$AR_p = \frac{\text{plenum width}}{\text{plenum length}} \quad (3.18)$$

In order to study the effect of different flow arrangements on the MCHS, the design of whole MCHS was planned in stages. The base plate of MCHS was made of copper as explained above.

In Design-I, two 10 mm circular through holes were made (Figure 3.5a) in such a way that they are at the mid position of the inlet and outlet plenums. The passage of inlet and outlet of the liquid was made through the adapters fixed on these threaded holes (Figure 3.5b). Only one type of flow arrangement (perpendicular type) was possible in this configuration. Apart from the inlet and outlet plenum temperatures, temperatures were also measured at three different locations underneath the microchannels in axial direction. The limitation of the current Design-I was that only one type of flow arrangement can be studied, so Design-II was proposed.



(a)



(b)

Fig.3.5: (a) Design-I microchannel heat sink (b) Design-I position of inlet & outlet adapters

In order to have more flow arrangements in the MCHS, design –II (Figure 3.6) was proposed. In the modified design three flow arrangements are possible. The design-II has the limitation that if the inlet and outlet adapters are positioned in the axial direction of the microchannels, there was no space left for the plenum pressure ports. Also practically the overall size of the MCHS with adapters will increase, and space is an important consideration in thermal packaging. In order to have provision for the pressure ports for analyzing pressure variation within plenums design-III was proposed.

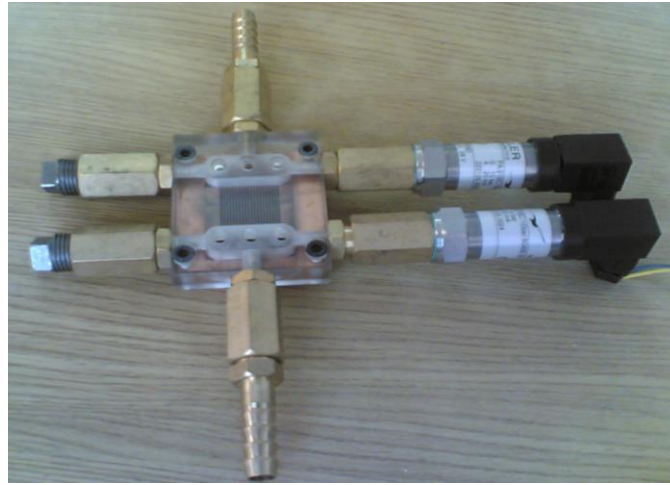


Fig. 3.6: Design-II microchannel heat sink

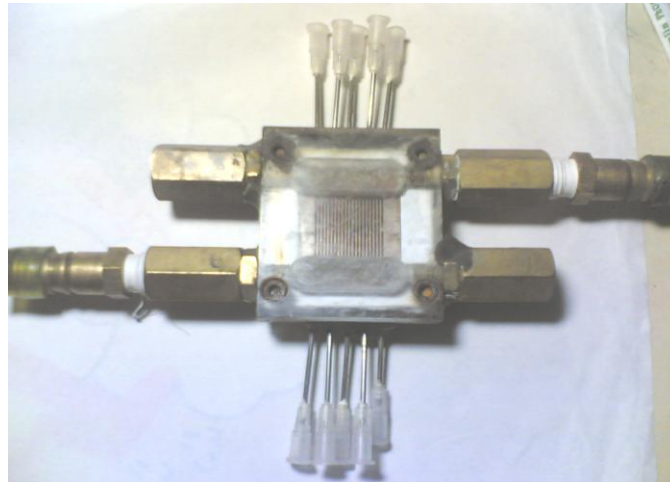
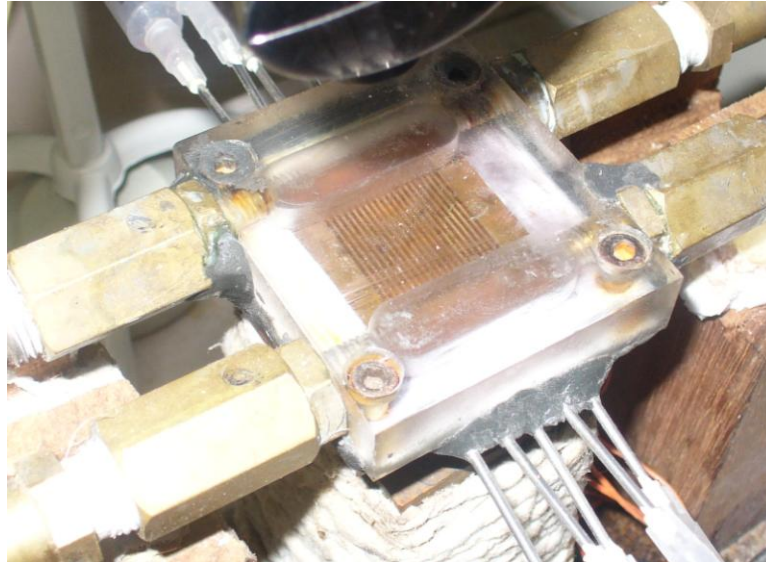


Fig. 3.7: Design-III microchannel heat sink

In order to study the feasibility for the provision of pressure ports for plenum positions, design – III (Figure 3.7) was tested. For the pressure drop measurements, small pressure taps (inner diameter: 0.5 mm) were used so as to ensure static pressure reading. Here total 10 pressure ports are inserted at equal distance within inlet and outlet plenums. The two flow arrangements are

also tested. Figure 3.8a shows the operational condition of the deionised water flow within the MCHS. Figure 3.8b shows the MCHS with camera to capture image, flow and heating arrangement in position. The design-II and design-III were combined to have final design-IV, which has the provision of analyzing 3 flow arrangements, differential pressure drop within inlet-outlet plenums and total pressure drop.



(a)



(b)

Fig. 3.8: (a) View of MCHS with deionised water flow (b) Design-III MCHS with camera, different flow & heating arrangements

Figure 3.9 shows the final design of the microchannel heat sink (non-insulated) which leads to 18 combinations (Table 3.2) of test pieces. The top transparent polycarbonate acrylic sheet cover was designed separately to give three types of flow arrangements. Figure 3.10 (a, b and c) show U-type, S-type and P-type flow arrangements in the top acrylic cover.



Fig. 3.9: Final design of microchannel heat sink

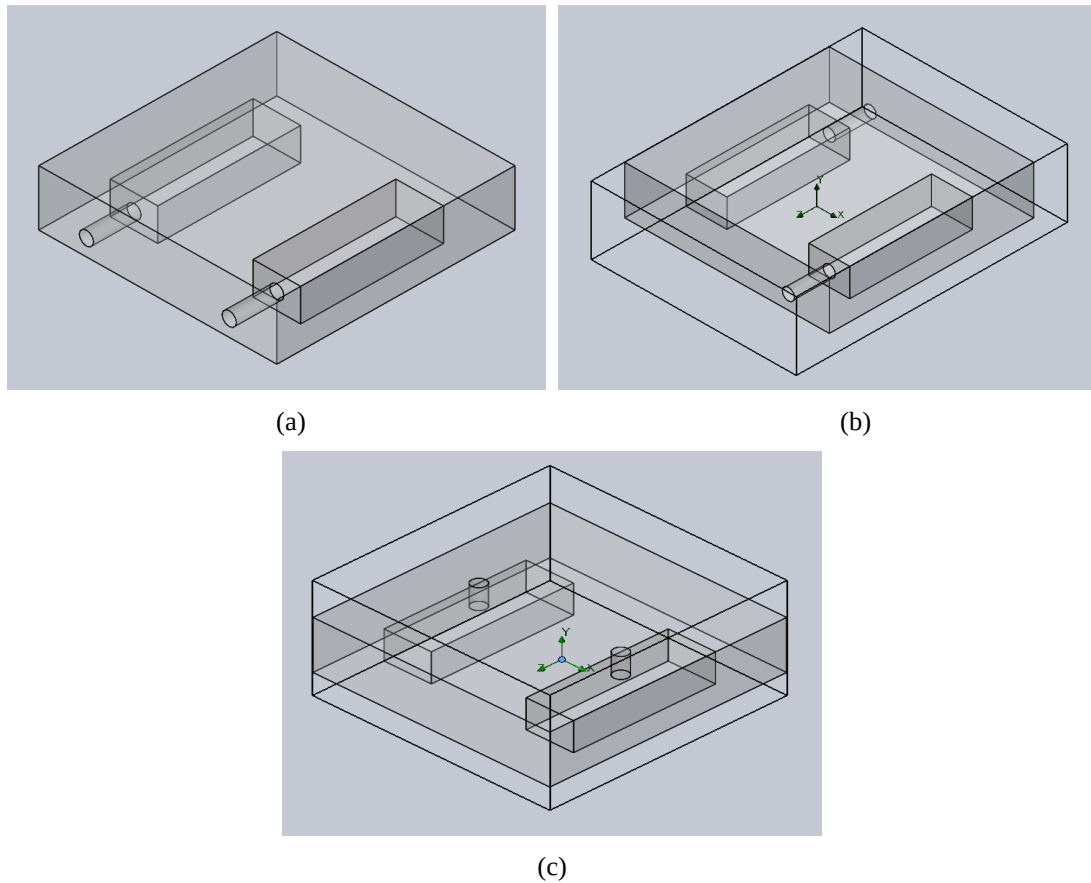


Fig. 3.10: Flow arrangements in top transparent polycarbonate acrylic cover (a) U-type (b) S-type and (c) P-type

There are 21 arrays of parallel microchannels in the MCHS, each having 530 μm or 330 μm width with a constant depth of 2.5 mm giving AR_c of 4.72 and 7.57 respectively. The inlet and outlet plenums have constant width of 30 mm and depth of 2.5 mm. Their length was varied as 12, 10 and 8 mm to have AR_p of 2.5, 3.0 and 3.75 respectively. Twenty T-type thermocouples were inserted in the inlet and outlet plenums of the MCHS, as shown in Figure 3.11a, at equal distances of 5 mm to measure the temperature distribution within the plenums. Pressure sensors were also fixed at ten different locations, as depicted in Figure 3.11b, for measuring the differential pressures within the inlet and outlet plenums of the MCHS.

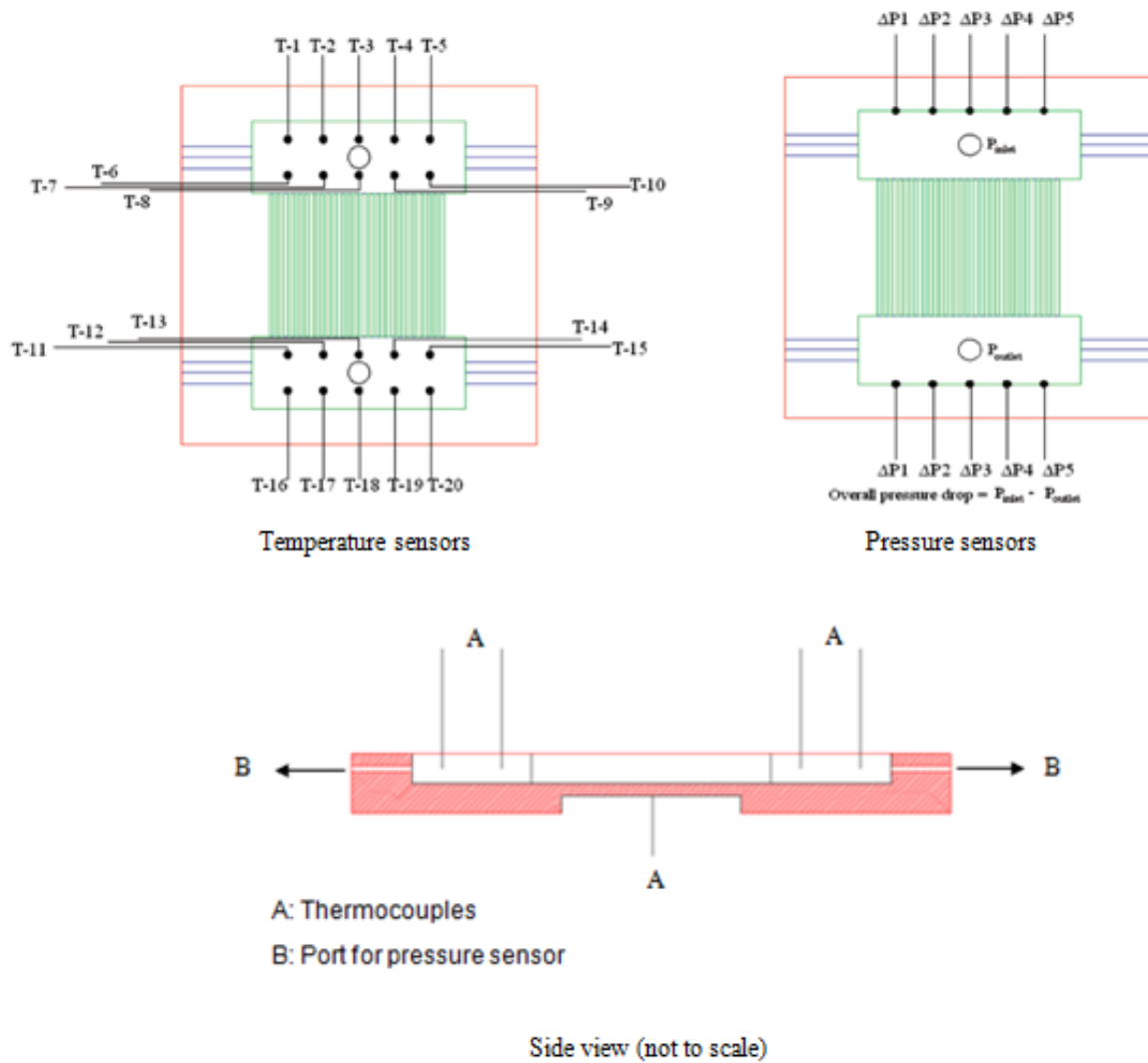
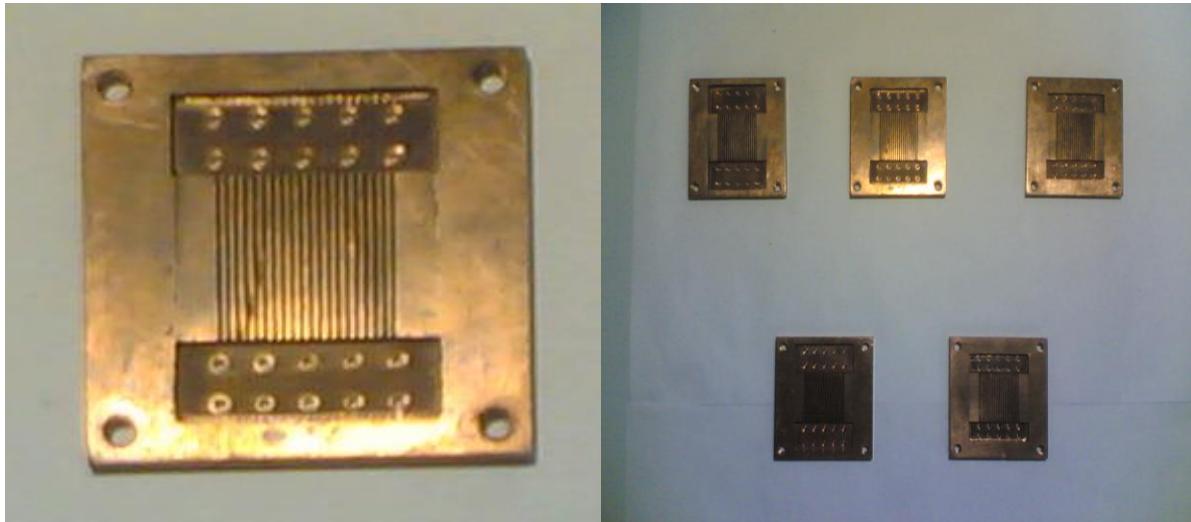


Fig. 3.11: Positions of temperature and pressure sensors within the microchannel heat sink

The actual photographs of the microchannel heat sink without the top transparent polycarbonate acrylic sheet cover are shown in Figure 3.12 (a, b). The position for thermocouples and pressure probes are shown in Figure 3.12a. Figure 3.12 (a, b) shows total six microchannel heat sinks each having different channel aspect and plenum aspect ratios. Figure 3.13 shows the microscopic photograph of microchannels within a heat sink taken by a camera interfaced with computer.



(a)

(b)

Fig. 3.12: Actual photographs of MCHS

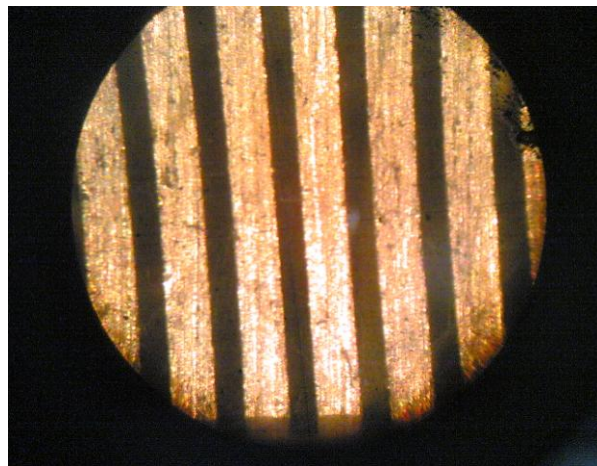


Fig. 3.13: Photograph showing micro channels within heat sink

3.8 Test setup

With constant improvement from Design-I to IV, the experimental setup (Figure 3.14) also sees a change. The final experimental test system is depicted schematically in Figure 3 15. Deionized

water was pumped from the liquid reservoir to the test section through a 15 μm filter. The outlet fluid from the MCHS was cooled by forced convective heat exchanger and returned back to the liquid reservoir, thus forming a closed loop. The 15 μm filter was used to prevent any solid particles from entering the MCHS. The flow meter was calibrated by weighing the outlet liquid over a given period of time using a high accuracy electronic balance with water as the working fluid; the electronic balance has an accuracy of 0.01 g. The inlet flow rate to the test piece was maintained by adjusting the control valve located in the line.

The inlet and outlet fluid temperatures were measured by high-precision T-type thermocouples with a bead diameter of 1.0 mm; the resolution of the thermocouple reader was $\pm 0.1^\circ\text{C}$. All the thermocouples and the data acquisition system were calibrated before use. The response time was estimated to be 80 ms, which is fast enough to trace the dynamic parameters for the present study. A needle valve fitted on the flow meter downstream of the pump allows for fine adjustment of the flow rate, which corresponds to a Reynolds number range of 224–1,121. The measurement of pressure in the plenums (inlet/outlet) was made by using five differential inclined manometers, which were calibrated against a known standard, and the uncertainty in the pressure measurement was less than 0.1%. All the temperature readings were collected by a high-speed data acquisition system and displayed by a computer monitor and stored in the data base for analysis. The average of pairs of temperatures T1-T6, T2-T7, T3-T8, T4-T9, and T5-T10 were taken at the inlet plenum, and similarly, the average of pairs of temperatures T11-T16, T12-T17, T13-T18, T14-T19, and T15-T20 were taken at the outlet plenum. Six holes of 8.0 mm diameter were drilled in the copper block to accommodate the cartridge heaters. The cartridge heater was powered by a 0–220 V AC variac, and the heating power was measured by a precision digital Wattmeter with an accuracy of 0.1%. The copper heat sink was heat insulated by an asbestos sheet covered with glass wool packaging. Figure 3.9 shows the photographs of the test piece as fitted in the experimental set-up. In order to supply variable power to the cartridge heaters a variac was used to vary the power supply within the range of 100 W to 800 W. Lee et al. (2005) suggests 85–90% of power provided to the cartridge heaters was transferred to water.

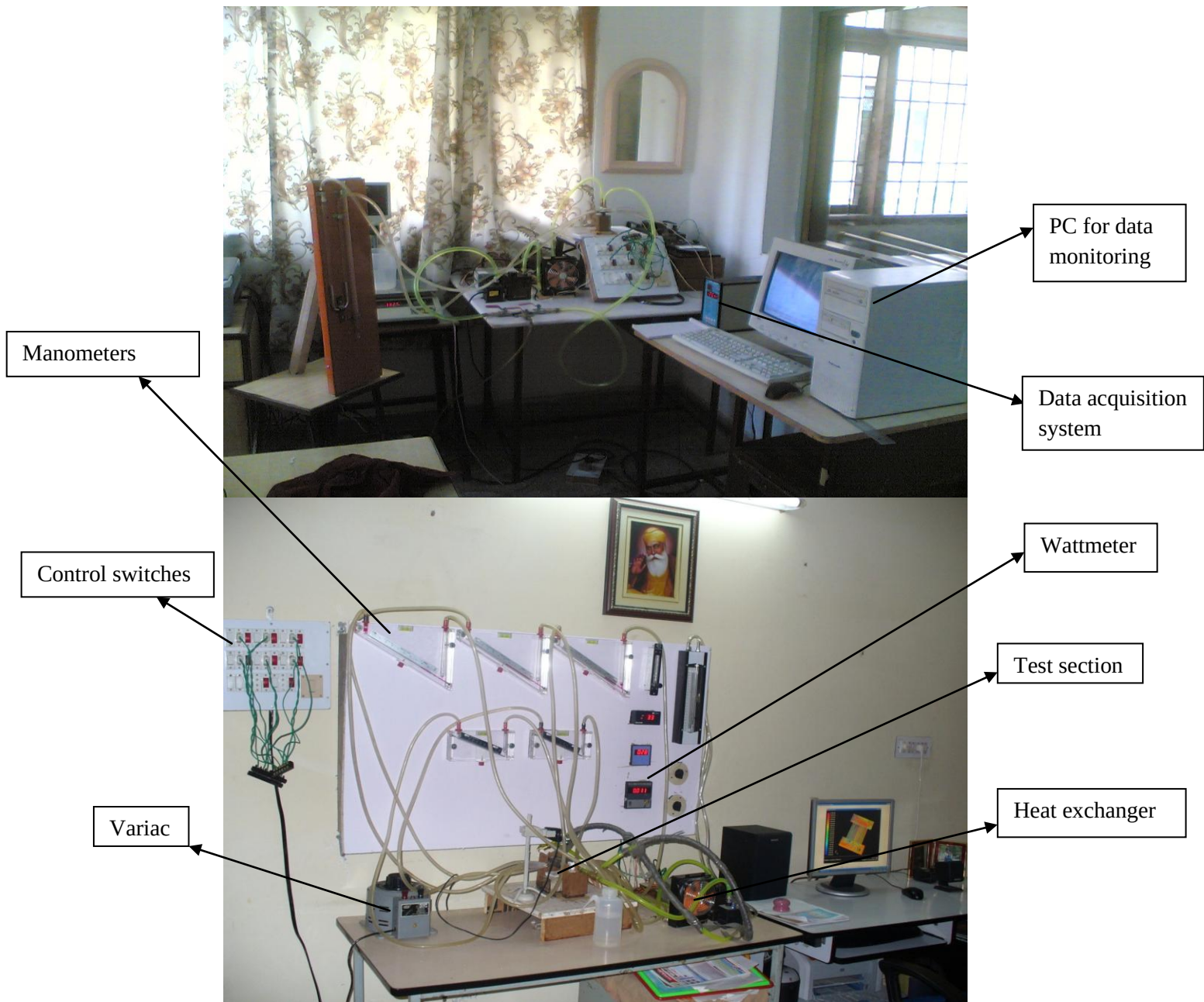


Fig. 3.14: Experimental setup

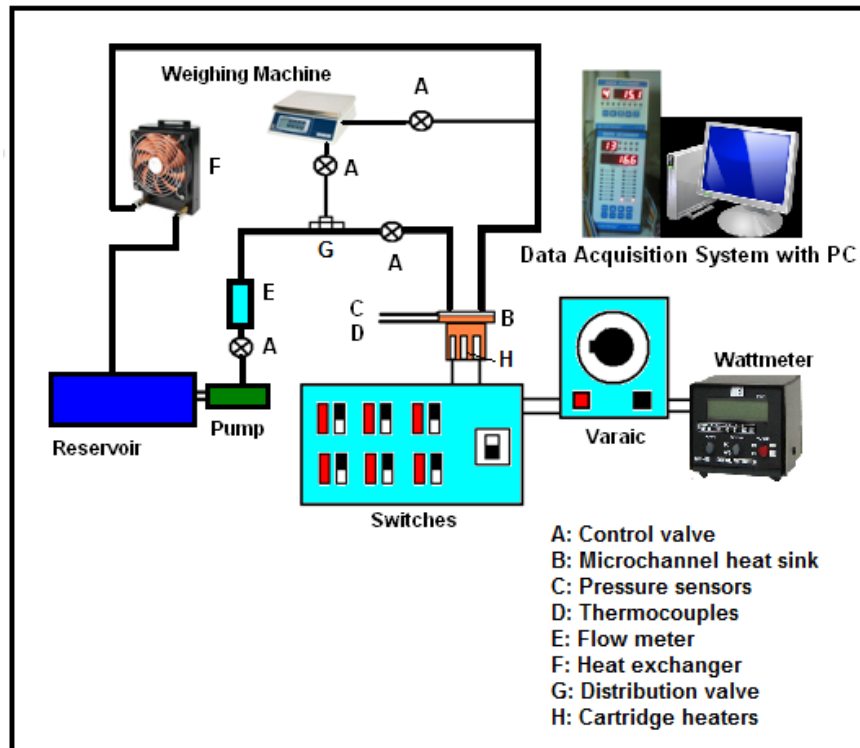


Fig. 3.15: Schematic layout of experimental setup

3.9 Instruments and their calibration

3.9.1 Pump- Make: Micropump, Series 200

Figure 3.16 shows the magnetically driven, suction shoe style gear pump. A suction shoe style gear pump captures fluid in the space between the gear teeth and the adjacent suction shoe to carry it through the pump. This unique suction shoe design provides steady fluid delivery against increasing discharge pressure and self-compensates for wear. The magnetic coupling uses two concentric magnets to drive the pump without direct contact with the motor shaft, eliminating the need for a dynamic shaft seal. Figure 3.17 shows the dimensions of the pump.



Fig. 3.16: Micropump Series 200

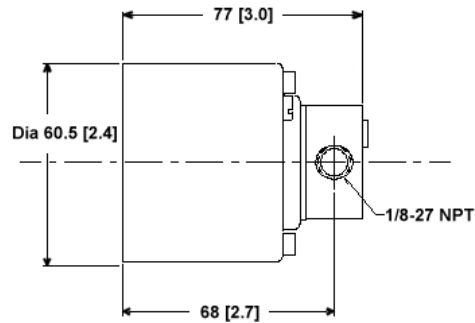


Fig. 3.17: Dimensions of micropump series 200 in mm [in]

3.9.2 Heat exchanger- Make: Thermaltake; Model: TMG1

The technology used in the Thermaltake heat exchanger TMG (Thermal Maximum Grade) is “Dimple Tubes” that create vortex and swirl of the coolant within the radiator. combined with the high fin density of louvered fins, the Dimple Tube Technology greatly enhances the performance of the radiator (Figure 3.18).



Fig. 3.18: Thermaltake TMG-1 heat exchanger

3.9.3 T-type thermocouples- Make: Omega; Model: 5TC-TT-T-36-36 and HYP1-30-1/2-T-G-60-SMPW-M

Thermocouple Model: 5TC-TT-T-36-36 from OMEGA are used to measure the temperature within the inlet and outlet plenum whereas Model: HYP1-30-1/2-T-G-60-SMPW-M from OMEGA (Figure 3.19) are used to measure temperature of deionised water at inlet and outlet

adapter positions. The measurable temperature range is cryogenic to 200°C (400°F). T-Type (Copper-Constantan) grounded-junction probe with PFA-insulated extension wire was used. All the temperature readings were collected by a high speed data acquisition system.



Fig. 3.19: T-type thermocouple

3.9.4 Data acquisition system- Make: Ambetronics; Model: TC-800 and TC-1600F

For measurement of temperatures and pressure at different positions in digital display, we have used a data acquisition system (Figure 3.20) from Ambetronics, Model TC-1600F which has a 16 data channels and Ambetronics, Model TC-800 which has 08 data channels. It can accept J, K, R, S, T thermocouple, PT-100 (3-wire), 4 to 20 mA and 0 to 10 V DC inputs. The J, K, R, S, T thermocouples are field programmable. The 4 to 20 mA linear input range is programmable by users. The unit has built-in channel 'Skip', 'Hold' and fast manual scanning facility. The two independent set points for each channel with common relay 1 and relay 2 are given with separately-1 No/C/NC. Eight feature touch keys for Mode, Shift, Inc, Ent, Hold, Esc and Print, facilitate easy operation. The channel auto scanning time is programmable from 1 s to 999 s. The acquisition system is interfaced with PC by using Windows based Data Log software which plots the real time variation. The data acquisition systems are interfaced with the PC using RS-485 serial port. The data acquisition system has an accuracy of $\pm 0.25\%$ FS and resolution of 0.001% FS.



Fig. 3.20: Data Acquisition Systems

3.9.5 Pressure transmitters- Make: KELLER Druckmesstechnik; Series 33X

The total pressure drop within the MCHS is measured with two KELLER (Figure 3.21) make pressure transmitters. Transmitters were given 24 V DC power supply and for this a voltage regulator was connected between pressure transmitters and data acquisition system. The output of the pressure transmitter is between 4-20 mA and displayed/processed by the data acquisition system. Each pressure transducer has an accuracy of $\pm 0.1\%$ and resolution of 0.002% over the standard pressure range 0–3 bar.



Fig. 3.21: Pressure Transmitter

3.9.6 Flow meter- Make: EGE; Model: SDN 552

EGE's inline sensor SDN 552 (Figure 3.22) with range 0.01-10 litres/min is used for monitoring deionised water flows. SDN 552 has one analog output (4-20 mA) for flow, one analog output for temperature, and a three-digit digital display. SDN 552 monitors the flow when it is between 0 and 90 °C. The sensors are based on thermodynamic principle. SDN 552 has no moving parts

which protrude into the medium. SDN 552 withstands pressures up to 20 bars and feature IP65 protection. The flow meter has a precision reading of 2%.



Fig. 3.22: Flow meter

3.9.7 Inclined manometers- Make: Bombay Instruments; Model: INC 525

The differential pressure (Figure 3.23) drop within the inlet and outlet plenums are measured with the help of inclined manometers from Bombay Instruments, India; Model: INC 525 Acrylic. Total 5 solid 1 inch thick acrylic body inclined manometers are used. INC 525 are thick acrylic plastic solid block, vertically unbreakable, stable and free of distortion. Steel engraved scale, zero adjusting knob, level bottom and panel mounting holes are provided on the inclined manometers.

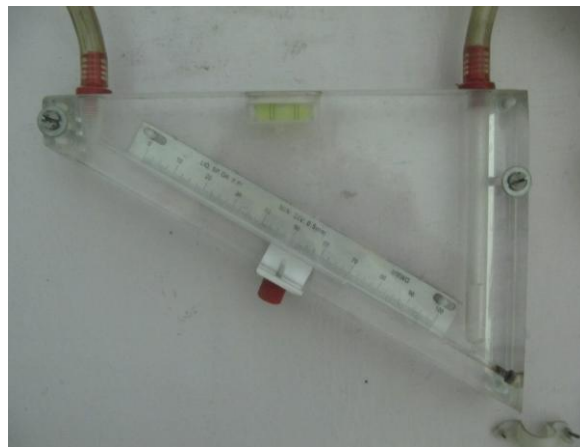


Fig. 3.23: Inclined manometer

3.9.8 Cartridge heaters-Make: BEECO; Model: BE-162

Cartridge heaters are usually tubular and are inserted into drilled holes of metal blocks for heating. For the heating of the microchannel heat sink, 6 electric operated cartridge heaters

(Make: BEECO; Model:BE-162) are inserted in the copper block (Figure 3.24), each having maximum 150 W capacity. BE-162 grounded cartridge heaters has one spiral of Nickel-Chrome 80/20 resistive wire, which is coaxial with 304 Stainless Steel sheath, in compacted MgO environment. The number of spirals per inch varies according to the requested Watt density.



Fig. 3.24: Cartridge heaters fitted in copper block

3.9.9 Wattmeter- Make: SIGMA; Model: SDP-254-A

The digital wattmeter SIGMA SDP-254-A (Figure 2.25) measures up to 1000 Watts, it has LCD display for reading. It comes with inbuilt alarms standard, which can be programmed to activate on low or high set points with any amount of dead band. The panel mounted Wattmeter has 96 × 96 mm size, with auxiliary 230V AC supply. The Wattmeter has the internal circuit of MFJ- 826. The measurement uncertainty associated with the Wattmeter was less than 0.8% (Appendix –A).



Fig. 3.25: Digital wattmeter

3.9.10 Universal Dry Block Temperature Calibrator- Make: Nagman; Model MTC 350

Thermocouples are calibrated by Microprocessor based Low Temperature Calibrator (Figure 3.26) with temperature range from ambient to 350°C. The calibrator has an accuracy of $\pm 0.4^{\circ}\text{C}$ and capability to measure RTDs, Thermocouples, mV / mA & Ohms signals. Tables 3.3 and 3.4 show the specifications and measurement range, resolution and accuracy of the calibrator.



Fig. 3.26: Universal dry block temperature calibrator

3.9.11 Camera- Make: TT; Model: TT1802

CMOS image sensor based webcam/PC camera (Figure 3.27) with 12 Mpixels and 6 LED Night Vision is used to capture the flow within microchannels. The camera has USB 2.0 interface for PC connectivity. The total weight of the camera is 83 g. The camera has a focus range of 5 cm – infinity.



Fig. 3.27: PC camera

3.10 Experimental procedure

The overall performance of the MCHS is estimated in terms of convective heat transfer along with the pressure drop. The pressure drop parameter is important from operating cost point of view. The actual heat input to the heat sink is supplied through the cartridge heaters provided at the bottom section of the heat sink. In the present research, in order to obtain a desired heating load to be carried away by the water, the power input to the cartridge heaters was adjusted. In the absence of viscous dissipation effects, for steady flow through the microchannels, the actual heat convected from the inside surface of the channels to the fluid stream can be equated to the sensible heat carried away by the fluid as

$$q = \dot{m}C_p (T_o - T_i) \quad (3.19)$$

where $\dot{m} = \rho \times Q$

The fluid temperatures T_i and T_o , are measured by the thermocouples located at the entry and exit adapters of the MCHS and the average of these temperatures is taken as the mean fluid temperature, T_m . The density, ρ and specific heat capacity, C_p are calculated based on the mean fluid temperature T_m . For internal flows, the convective heat transfer from the walls of the channels to the fluid stream is calculated from Newton's law of cooling as

$$q = hAN(T_w - T_m) \quad (3.20)$$

where h is the convective heat transfer coefficient, A is the surface area available for heat transfer per channel ($L(W_c + 2D_c)$), N is the total number of microchannels and T_w is the temperature of the channel wall.

As it is difficult to measure the wall temperature of the microchannels directly, it is computed by assuming one-dimensional heat conduction. For this purpose, the temperature closer to the bottom wall of the channel, (T_n) is measured as shown in Figure 3.28.

Assuming that all the sensible heat gained by the water in the microchannel heat sink is equal to the heat conducted through the wall, we can write the heat conducted as

$$q = k_{cu}A_h \frac{(T_n - T_w)}{y} \quad (3.21)$$

in which k_{cu} is the thermal conductivity of the material of the heat sink, A_h is the MCHS bottom heated area over which heating is provided by the cartridge heater, y is the distance between the

bottom wall of the channel and the point at which thermocouple is embedded to measure T_n (Figure 3.28). For microchannels made of copper with dimensions considered in the present work, the temperature gradient along the length and width of the channels is assumed to be very small and hence it is neglected. The wall temperature of the channel, T_w is computed from equation (3.23). However, the mean fluid temperature T_m increases along the length of the channel as the channel wall temperature becomes higher than the mean fluid temperature. For constant uniform surface wall heat flux condition, such as the one considered in the present research, the variation of T_m with the length of the channel is assumed to be linear on a similar ground as considered by Lee et al. (2005).

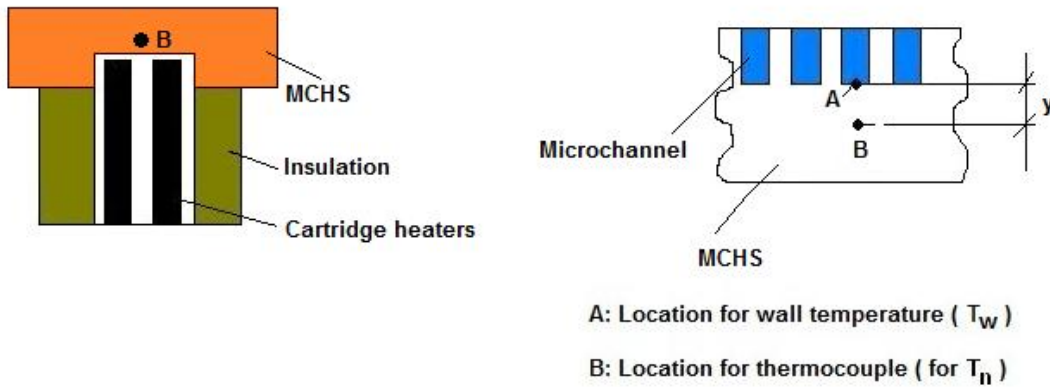


Fig. 3.28: Arrangement for wall temperature measurement (Not to scale)

The fluid temperature at the inlet and outlet plenums are measured from the twenty thermocouples positioned within the MCHS. It was also assumed that the temperature at any location within the microchannel can be computed from the heat balance equation:

$$T_m - T_{mi} = \frac{q}{\dot{m}C} \frac{x}{L} \quad (3.22)$$

where T_{mi} is the fluid temperature at the inlet plenum obtained as the average of the temperatures measured using the thermocouples.

After estimating T_w and T_m , the convective heat transfer coefficient can be computed from Eqn.(3.20). Nusselt number is the heat transfer characteristics that represents the convective heat transfer performance of a MCHS and is defined as follows:

$$Nu = \frac{hD_h}{k} \quad (3.23)$$

where D_h is the hydraulic diameter defined as

$$D_h = \frac{2(W_c \times D_c)}{W_c + D_c} \quad (3.24)$$

where W_c and D_c are width and depth of a microchannel, k is the thermal conductivity of fluid evaluated at the mean fluid temperature, T_m and h is the heat transfer coefficient that can be evaluated. The flow Reynolds number is defined as

$$Re = \frac{\rho V_f D_h}{\mu} \quad (3.25)$$

where V_f is the inlet fluid velocity computed as

$$V_f = \frac{Q}{N \times A_c} \quad (3.26)$$

where $A_c = W_c \times D_c$

Once the system attained steady state conditions, readings from flow meter, thermocouples, pressure transducers and wattmeter were recorded and the Nusselt and Reynolds number were computed using Eqns. (3.23) to (3.26) respectively.

CHAPTER 4

EXPERIMENTAL ANALYSIS OF MICROCHANNEL HEAT SINK

4.1 Design of Experiments and test matrix

Design of Experiments (DOE) is a systematic approach based on statistical methods for studying the effects of multiple variables simultaneously on the required performance of a system or process. A series of structured tests are designed in which planned changes are made to the input variables of a process or system. The effects of these changes on a pre-defined output are then assessed.

DOE helps in:

- Identifying the relationships between cause and effect.
- Providing an understanding of interactions among causative factors.
- Determining the levels at which to set the controllable factors in order to optimize reliability.
- Minimizing experimental error (noise).
- Improving the robustness of the design or process to variation.

The techniques of DOE provide powerful and efficient methods to achieve these objectives. Designed experiments are much more efficient than one-factor-at-a-time experiments, which involve changing a single factor at a time to study the effect of the factor on the product or process. While the one-factor-at-a-time experiments are easy to understand, they do not allow the investigation of how a factor affects a product or process in the presence of other factors. When the effect that a factor has on the product or process is altered due to the presence of one or more other factors, that relationship is called an interaction. Many a times the interaction effects are more important than the effects of individual factors. This is because the application environment of the product or process includes the presence of many other factors together instead of isolated occurrences of single factors at different times. The methodology of DOE ensures that all factors and their interactions are systematically investigated; thus, information obtained from a DOE analysis is much more reliable and complete.

The present designed experiment is usually carried out in five stages planning, screening, optimization, robustness testing and verification.

- **Planning**

It is important to carefully plan for the course of experimentation before embarking upon the process of testing and data collection. With thorough and precise objectives of the present research in mind, and assessment of time and resources available to achieve the objectives and integration, planning of the experimentation was done. A number of factors were identified to investigate the most appropriate response(s) to measure. The factors that could affect the thermal and fluidic performance of liquid flow (water) within microchannels (rectangular shape) heat sink are:

- ❖ Flow field
- ❖ Thermal condition
- ❖ Microchannel width
- ❖ Microchannel depth
- ❖ Microchannel length
- ❖ Number of microchannels
- ❖ Flow configurations within MCHS
- ❖ Arrangement of inlet and outlet fluid flow, etc

- **Screening**

Screening of experiments are used to identify the important factors that affect the process under investigation out of the large pool of potential factors. These experiments are carried out in conjunction with prior knowledge of the process to eliminate unimportant factors and focus attention on the key factors that require further detailed analyses. Since in the present case, important consideration is given to the flow arrangements and its affect on the performance of a MCHS, a detailed listing of parameters were worked out to quantify the following factors:

- a) Microchannel length
- b) Microchannel width
- c) Microchannel depth
- d) Inlet plenum length
- e) Inlet plenum width
- f) Inlet plenum depth
- g) Outlet plenum length

- h) Outlet plenum width
- i) Outlet plenum depth
- j) Length of whole heat sink
- k) Width of whole heat sink
- l) Depth of whole heat sink
- m) Flow configurations within MCHS including space considerations
- n) Fluid flow condition
- o) Heat input condition

The factors (b) and (c) are taken into consideration in hydraulic diameter, D_h and microchannel aspect ratio. A new definition of plenum aspect ratio was introduced to take into account (d), (e), (g), (h) factors. Three types of flow arrangements (P-type, S-type and U-type) are selected for (m) based on the space available generally in practical conditions where integrated circuits such as computer processor are positioned. The flow condition of fluid is taken care by the Reynolds number. The microchannel heat sink experiences a constant heat flux condition simulated by the cartridge heaters as observed in real life applications.

- **Optimization**

Once attention has been narrowed down to the important factors affecting the process, the next step is to determine the best setting of these factors to achieve the desired objectives. The Reynolds number range $224.3 \leq Re \leq 1121.7$ is assumed for the experimentation. Three heat inputs 125, 225 and 375 W are selected as heat inputs to the MCHS. The 225 W value is selected as a factor as in the manufactured MCHS the base heat input area is 2.25 cm^2 , which gives a heat flux value as 100 W/cm^2 , a threshold value as predicted by Moore's Law. A base heat input area of 2.25 cm^2 ($15 \text{ mm} \times 15 \text{ mm}$) is selected as similar dimensions are used in computer processor applications.

- **Robustness Testing**

Once the optimal settings of the factors have been determined, it is important to make the process insensitive to variations that are likely to be experienced in the application environment. These variations result from changes in factors that affect the process but are beyond control. Factors such as humidity, ambient temperature, etc. are referred to as noise or uncontrollable factors. It is important to identify such sources of variation and take measures to ensure that the

process is made insensitive (or robust) to these factors. In the present research the MCHS was heat insulated by an asbestos sheet covered with glass wool packaging and by this provision, repeatability was ensured.

- **Verification**

This final stage involves validation of the best settings and experimental setup by conducting a few follow-up experimental runs to confirm that the process functions as desired and most of the objectives are met.

Based on the discussion in Chapter 3, the geometric parameters are broadly classified into channel aspect ratio (AR_c), plenum aspect ratio (AR_p) and hydraulic diameter. The flow field variation under six different Reynolds numbers, 224.3, 336.5, 448.7, 673.0, 897.4 and 1121.7 are investigated for three flow arrangements, P-, S-, and U-type. As far as the heat input condition is concerned, three heat input values, 125 W, 225 W and 375 W are considered. Tables 4.1 and 4.2 show the microchannel and plenum geometrical parameters leading to different channel and plenum aspect ratios respectively.

Table: 4.1: Channel aspect ratio for MCHS

Test Piece	Microchannel width, W_c (μm)	Microchannel depth, D_c (mm)	Hydraulic diameter, D_h (μm)	Channel aspect ratio (AR_c)
1	330	2.5	583	7.57
2	330	2.5	583	7.57
3	330	2.5	583	7.57
4	330	2.5	583	7.57
5	330	2.5	583	7.57
6	330	2.5	583	7.57
7	330	2.5	583	7.57
8	330	2.5	583	7.57
9	330	2.5	583	7.57
10	530	2.5	875	4.72
11	530	2.5	875	4.72
12	530	2.5	875	4.72
13	530	2.5	875	4.72
14	530	2.5	875	4.72
15	530	2.5	875	4.72

16	530	2.5	875	4.72
17	530	2.5	875	4.72
18	530	2.5	875	4.72

Table: 4.2: Plenum aspect ratio for MCHS

Test Piece	Plenum width, W_p (mm)	Plenum depth, D_p (mm)	Plenum length, L_p (mm)	Plenum aspect ratio (AR_p)
1	30	2.5	8	3.75
2	30	2.5	10	3.00
3	30	2.5	12	2.50
4	30	2.5	8	3.75
5	30	2.5	10	3.00
6	30	2.5	12	2.50
7	30	2.5	8	3.75
8	30	2.5	10	3.00
9	30	2.5	12	2.50
10	30	2.5	8	3.75
11	30	2.5	10	3.00
12	30	2.5	12	2.50
13	30	2.5	8	3.75
14	30	2.5	10	3.00
15	30	2.5	12	2.50
16	30	2.5	8	3.75
17	30	2.5	10	3.00
18	30	2.5	12	2.50

For the purpose of validating the experimental set up and procedure, initial test runs were carried out in the MCHS experimental set up for P-, U- and S-type flow arrangements for six Reynolds number at a power input of 225 W with $AR_c=7.57$ and $AR_p=3$. As the results for the flow arrangements considered in the present work are not available in the literature, attempts are made to compare our results with the case of literature that resemble at least one of our flow

arrangements. Two such works that resemble our P-type flow arrangement are that of Lee et al. (2005) and (2006). The variation of Nusselt number with Reynolds number is considered as the parameter for comparison. Figure 4.1 depicts the comparison between the results obtained for three type of flow arrangements in the present work and the works of Lee et al. (2005) and (2006). The comparison shown in the figure indicates that our experimental results obtained for P-type flow arrangement are in close agreement with the results of correlation proposed by Lee et al. (2006) and the experimental results of Lee et al. (2005) at low Reynolds number range. The average percentage variation between the present experimental results for P-type flow arrangement and those reported by Lee et al. (2005) was found to be 2.9 – 13.5% whereas for Lee et al. (2006) it was 0.6 – 5.8%. The validation of the present work with those by Lee et al. (2005 and 2006) indicates that the experimental facility developed is appropriate for the analysis.

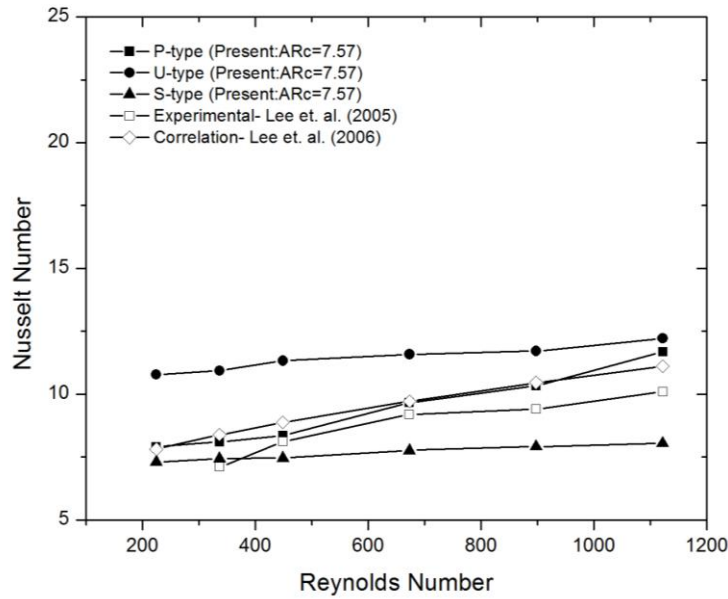


Fig.4.1. Validation results for Nusselt number

After the validation test runs, as shown in Table 4.3, a test matrix was made to evaluate the thermal performance of the MCHS with three flow arrangements, P-, U- and S-types; three heat input values, 125 W, 225 W and 375 W; six Reynolds numbers, 224.3, 336.5, 448.7, 673.0, 897.4 and 1121.7; three inlet and outlet plenum aspect ratios 2.5, 3.0 and 3.75, and two channel aspect ratios 4.72 and 7.57 giving rise to a total of 324 experimental trial runs.

Table: 4.3: Test matrix

Factors	Level-1	Level-2	Level-3	Level-4	Level-5	Level-6
Channel aspect ratio	4.72	7.57				
Plenum aspect ratio	2.5	3.0	3.75			
Heat input (Watts)	125	225	375			
Flow arrangement	P-type	U-type	S-type			
Reynolds number	224.3	336.5	448.7	673.0	897.4	1121.7

4.2 Temperature distributions

4.2.1 Temperature variation for different flow arrangements at constant Reynolds number

In order to understand the convective heat transfer mechanism, initially, the temperature distributions along the inlet and outlet plenums of 10 mm × 30 mm × 2.5 mm (plenum aspect ratio: 3.0) are obtained for the three flow arrangements of the MCHS with an array of parallel microchannels having an individual width of 330 μ m and a uniform channel depth of 2.5 mm (channel aspect ratio: 7.57). Using the temperature measurements obtained at different locations along the inlet and outlet plenums of the MCHS (Section 3.7), temperature variation plots are obtained for the P-, U-, and S-type flow arrangements for three heat input supplies as shown in Figure 4.2 at Re =224.3. The curves in solid and dotted lines show the temperature variations along the inlet and outlet plenum length, respectively. It is observed that the temperature variation along the outlet plenum is always higher than that of the inlet plenum for all the three flow arrangements. A close look at the temperature variations at the inlet plenum indicate that for the P-type flow arrangement, the temperature increases continuously, reaches a maximum at a length of 20 mm, and then decreases. The peak temperature in all the three cases increases with increase in heat input. In the S-type flow arrangement, two peak points are observed at the inlet plenum at 10 mm and 20 mm, and these peaks become more distinct at higher heat inputs. The minimum temperature is found to occur at the center of the plenum. In other words, the P-type

flow arrangement exhibits a single peak, but the U- and S-type flow arrangements exhibit two peaks in the temperature variation.

As far as the outlet plenum temperature variation is concerned, all the three flow arrangements exhibit an almost similar variation. The maximum temperature variation within the inlet and outlet plenum was for the S-type flow arrangement, and the minimum average temperature within the plenums was experienced for the P-type flow arrangement. The maximum temperature peak was observed at a distance of 20 mm within the inlet plenum, while the minimum was at 10 mm from the left edge for the outlet plenum. The mid-position of the inlet plenum, 15 mm from the left edge, corresponds to the inlet of microchannel flow, where the flow is fully developed.

Hydrodynamic entry length X_h as defined by Kandlikar et al. (2006) is :

$$X_h = 0.05 \times D_h \times Re \quad (4.1)$$

In this MCHS, the hydrodynamic entry length for the Reynolds number corresponding to 224.3 calculates as 6.53 mm by using Eq. (4.1). The position corresponding to 5 mm (from the left edge) is within the hydro dynamically developing region, while the position corresponding to 10 mm (from the left edge) is in the fully developed region, so the results are within the conventional laminar flow regime. Within the inlet plenum, there was a drop in temperature from 10 mm to 20 mm for U-type and S-type arrangements, because this position corresponds to the mid-section of the plenum, especially at the entrance regions of the micro-channels. Conventionally, because of the high heat transfer coefficient in the entrance zone of the micro-channels, locally, low temperatures were experienced. Similar types of trends were observed at inlet and outlet plenums for 125W, 225W, and 375W heat inputs. The steep drop and steep increase of temperatures was experienced for the 5–10 mm region and the 20–25 mm region within the plenums, because at the edges and boundary of the heat sink, there is no heat dissipation by fluid convection.

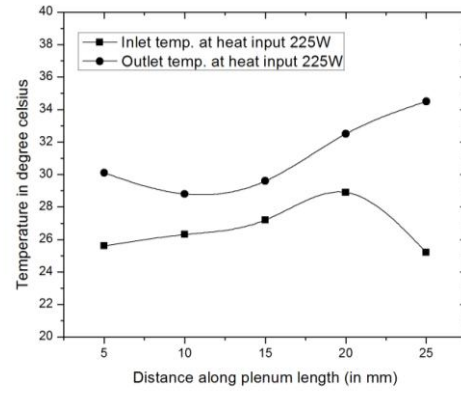
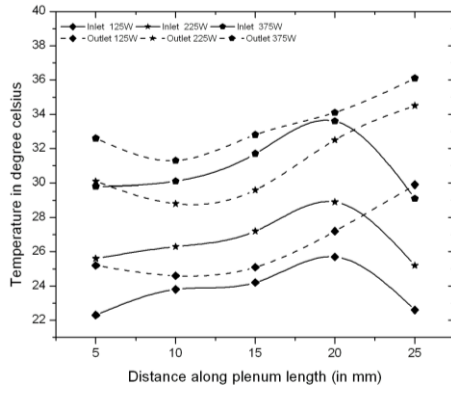
4.2.2 Effect of Reynolds number on temperature distribution

Figure 4.3 shows the variation of temperature within the inlet plenum at various positions for the variation of Reynolds numbers from 224 to 1,121. For low heat input of 125 W, the U-type flow arrangement experiences the maximum temperature variation along the length of the plenum, followed by the P- and S-types of arrangements. But as the heat inputs are increased, the

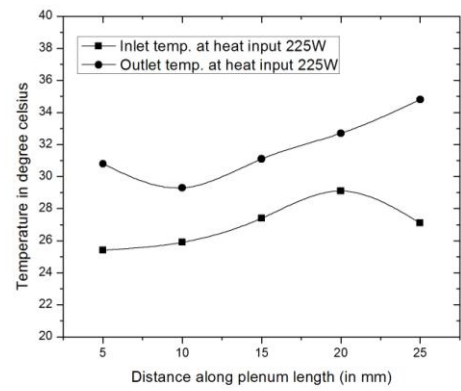
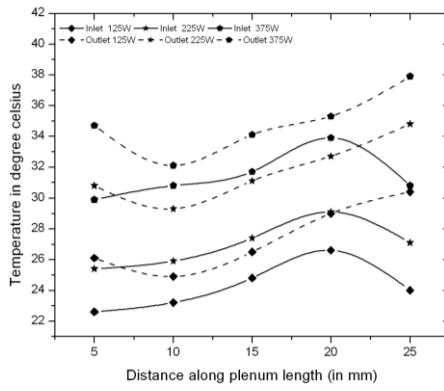
temperature variation changes in nature, with the S-type flow arrangement having the maximum temperature variation followed by the U-type arrangement. The change in the characteristics of temperature variation is due to less effective heat transfer rates for the S-type as compared to the U-type flow arrangement.

In the inlet plenum at the position of 15 mm, the P-type flow arrangement exhibits a higher temperature variation compared to the S- and U-types, which is due to the least retention time taken by the water to flow at the middle position of the plenum. In the P-type flow arrangement, the inlet water tries to fill the entire inlet plenum, thus establishing effective fluid contact with the heat sink bottom wall for convective heat transfer. For low heat input, the S-type flow arrangement always experienced low heating after a distance of 15 mm, whereas this trend gets reversed at higher heat input. As far as the U-type flow arrangement is concerned, it always occupies mid-range temperature variations for all the three heat inputs considered in this work.

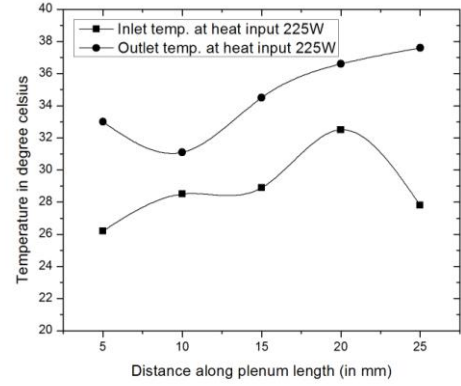
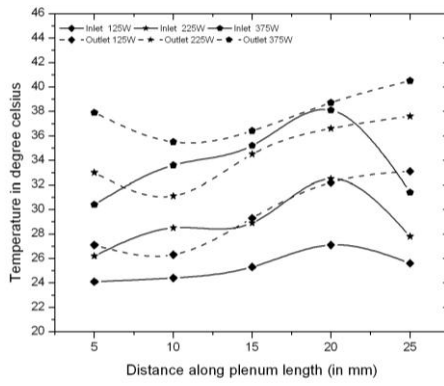
Figure 4.4 shows the variation of temperature along the length of the outlet plenum at various positions of 5, 10, 15, 20, and 25 mm, measured from the left edge for the three heat input values at different Reynolds numbers. The temperature at the outlet plenum for any given case at any given Reynolds number is always higher than the temperature observed at the inlet plenum. As the Reynolds number increases, this temperature rise decreases, and this can be attributed to the increase in convective heat transfer, which is inversely proportional to the temperature difference for a given constant heat input. It can easily be noticed that the S-type flow arrangement always exhibits a higher temperature at the outlet plenum compared to the U-type and P-type flow arrangements. The reason may be that in the S-type flow arrangement, the fluid initially starts filling the inlet plenum by striking the outermost part of the inlet plenum and then starts filling the microchannels to leave from the outlet plenum. But in the case of U-type and, to some extent, the P-type flow arrangement, the fluid starts filling the inlet plenum from the starting point of the plenum and fills the entire plenum in due course of time. It should be noted that once the fluid starts filling the inlet plenum, the fluid starts flowing into the microchannels, thus enhancing the effective fluid contact with the heated bottom wall of the MCHS.



(a)

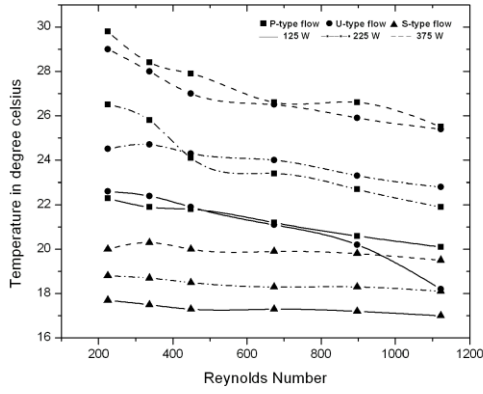


(b)

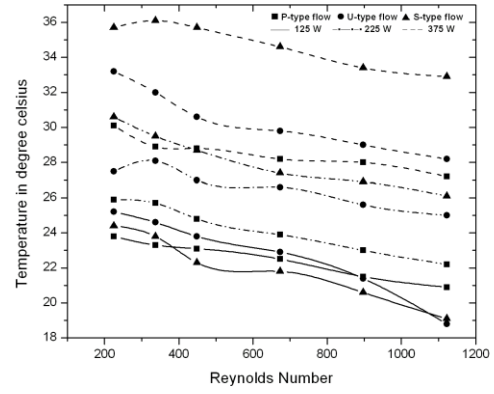


(c)

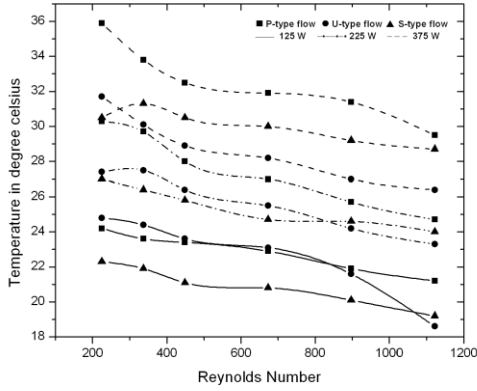
Fig. 4.2: Variation of temperature within plenums for different flow arrangements (a) P-Type (b) U-Type (c) S-Type for $Re = 224:3$



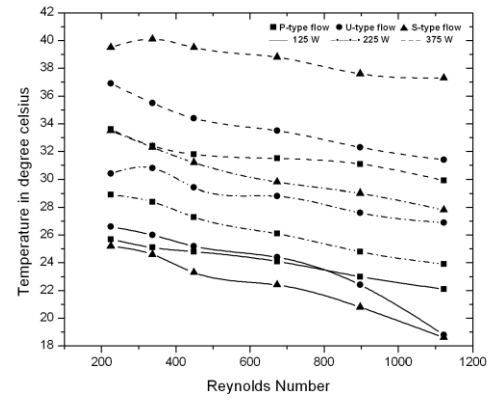
(a)



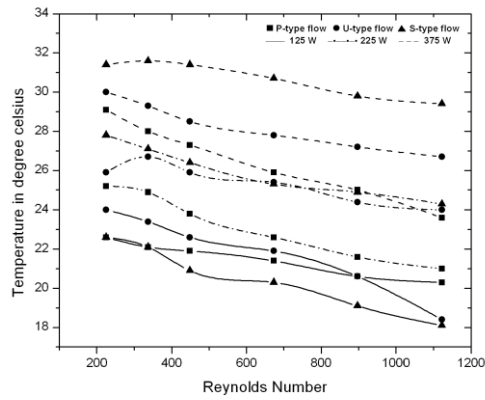
(b)



(c)

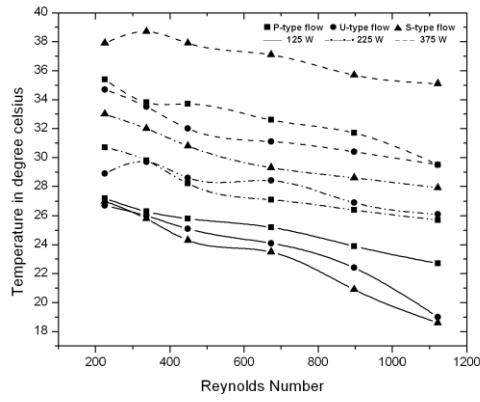


(d)

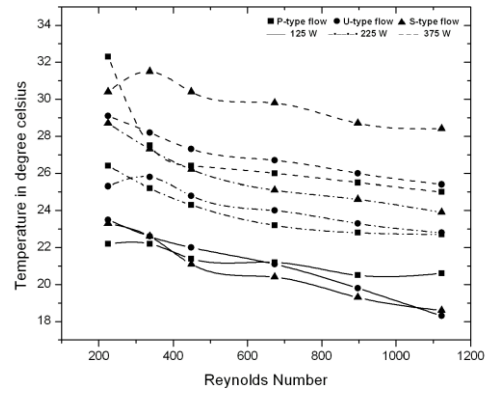


(e)

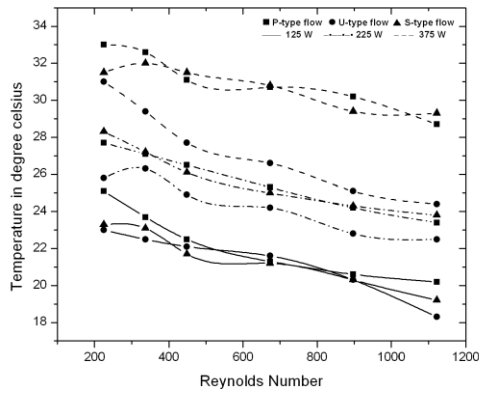
Fig. 4.3: Effect of Reynolds Number on Temperature distribution along the length of inlet plenum at (a) 5 mm (b) 10 mm (c) 15 mm (d) 20 mm (e) 25 mm



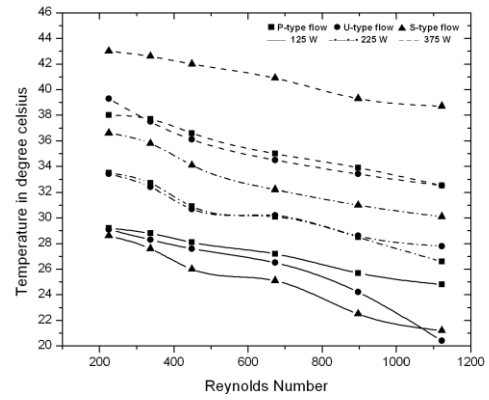
(a)



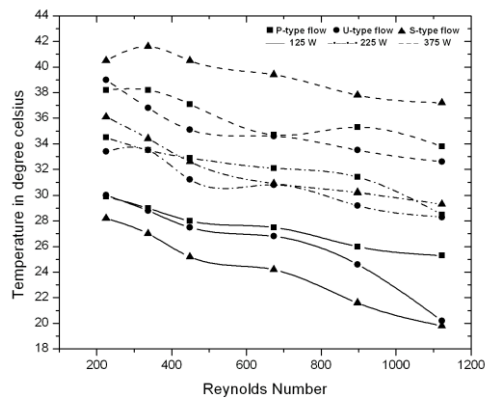
(b)



(c)



(d)



(e)

Fig. 4.4: Effect of Reynolds Number on Temperature distribution along the length of outlet plenum at (a) 5 mm (b) 10 mm (c) 15 mm (d) 20 mm (e) 25 mm

4.2.3 Effect of Reynolds number on Nusselt number for various flow arrangements

The performance of an MCHS can be better characterized by computing the Nusselt number for the given flow situation. Using the average temperature measured at the inlet and outlet plenums and heat sink wall temperature, the average heat transfer coefficient has been computed for all the values of Reynolds number, heat input, and flow arrangements. Figure 4.5 indicates that for a constant heat input and flow arrangement, not much change in Nusselt number was observed with variation of Reynolds number and the reason may be attributed to the fact that the internal flow of fluid within the entire heat sink causes satisfactory velocity and thermal boundary layer approximations, in the region between the plenums. Figure 4.5 shows the variation of average Nusselt number with Reynolds number for all the three flow arrangements (plenum size: $10\text{mm} \times 30\text{mm} \times 2.5\text{mm}$, channel width $330\mu\text{m}$) and heat inputs. The Nusselt number was calculated by taking into consideration the difference between average fluid temperature within the inlet plenum and the outlet plenum. It is seen from the figure that the U-type flow arrangement always has a higher average Nusselt number compared to the P-type flow arrangement, followed by the S-type. The variation of Nusselt number for a constant geometry but at different type of flow arrangements suggests that flow arrangements do play an important role in heat transfer studies, and it is not possible to ignore the effects of inlet/outlet flow arrangements. As considered in heat exchanger analysis, the effect of flow arrangements should also be taken into consideration while developing correlations and performance parameters for MCHSs.

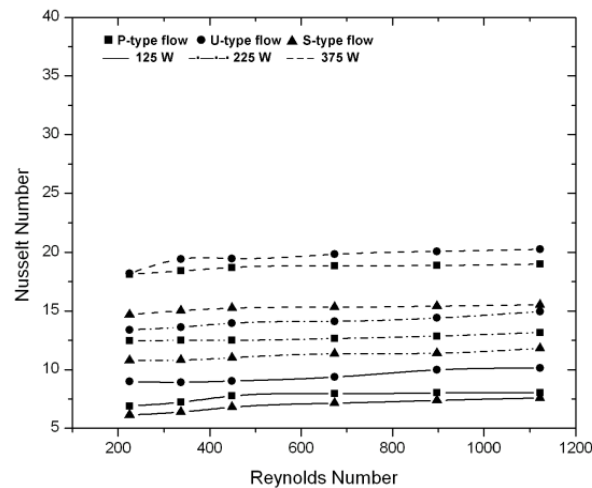


Fig 4.5: Effect of Reynolds number on Nusselt number for various flow arrangements (plenum size: $10\text{mm} \times 30\text{mm} \times 2.5\text{mm}$, channel width $330\mu\text{m}$)

The above discussions on temperature distributions are for a constant plenum size of 10 mm × 30 mm × 2.5 mm and constant microchannel width of 330 μm. The effects of variation of plenum size and microchannel width on the performance of MCHS are studied in the subsequent section.

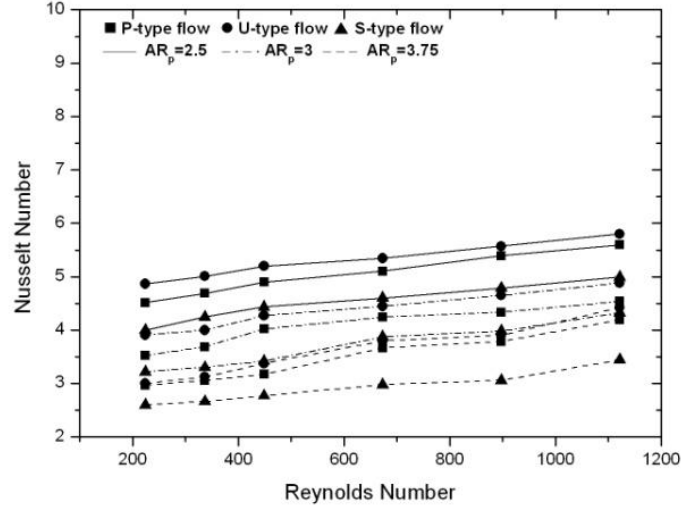
4.3 Effect of aspect ratio and flow arrangements on thermal performance of MCHS

The thermal performance of microchannel heat sink is analyzed by computing the variation of Nusselt number with Reynolds number for various test cases. The effect of channel aspect ratio, plenum aspect ratio and flow arrangements at different heat input values on Nusselt number is discussed in detail in the following sections.

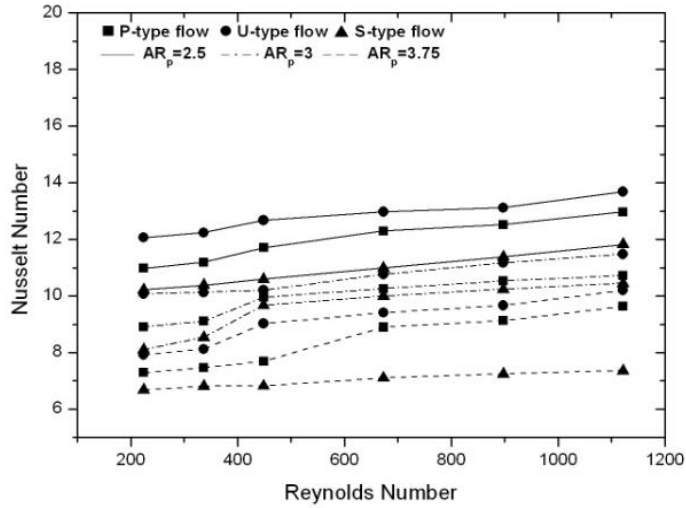
4.3.1 Effect of channel aspect ratio and plenum aspect ratio

Two different channel aspect ratios are obtained by varying the channel width to 330 μm and 530 μm, keeping the channel depth constant, giving rise to $AR_c=7.57$ and 4.72 respectively. Similarly, three plenum aspect ratios 3.75, 3 and 2.5 are obtained by varying plenum length to 8 mm, 10 mm and 12 mm respectively keeping the width constant at 30 mm (Refer Figure 3.4). Nusselt number variation results are presented for three heat input values in Figures 4.6 to 4.8, wherein in each figure results are given for two channel aspect ratios for a given channel aspect ratio: nine plots are shown for the combinations of three flow arrangements at three plenum aspect ratios. In these figures the symbols denote different flow arrangements, P-, U- and S-types, whereas the different line styles indicate different plenum aspect ratios. The variation of Reynolds number from 224.3 to 1121.7 is obtained by varying the fluid velocity from 0.26 to 1.28 m/s for $AR_c=4.72$. With increase in fluid velocity the fluid momentum transport in the axial direction increases, resulting in thinning of boundary layer and this enhances convective heat transfer. As the channel wall receives heat continuously from the cartridge heater, the fluid momentum is also accompanied by the bulk transport of heat from the wall towards the full extent of the fluid stream, thus raising the mean temperature of the fluid stream. Hence increase in fluid velocity gives rise to increase in convective heat transfer, resulting in increase in Nusselt number as seen from Figure 4.6 (a). This trend is observed for all the three flow arrangements at different plenum aspect ratios. However, with regard to the flow arrangement, U-type always records a higher Nusselt number followed by P- and S-types. It is seen from the above figure that

as the plenum aspect ratio is increased from 2.5 to 3.75, the Nusselt number decreases for all the Reynolds numbers. The maximum Nusselt number is obtained for $AR_p=2.5$ for U-type flow arrangement followed by P-type and S-type arrangements and the minimum Nusselt number is obtained at $AR_p=3.75$ for S-type flow arrangement.

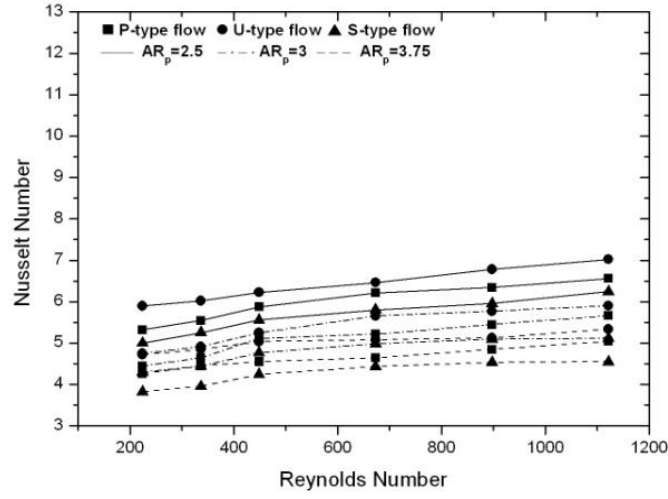


(a) $AR_c = 4.72$

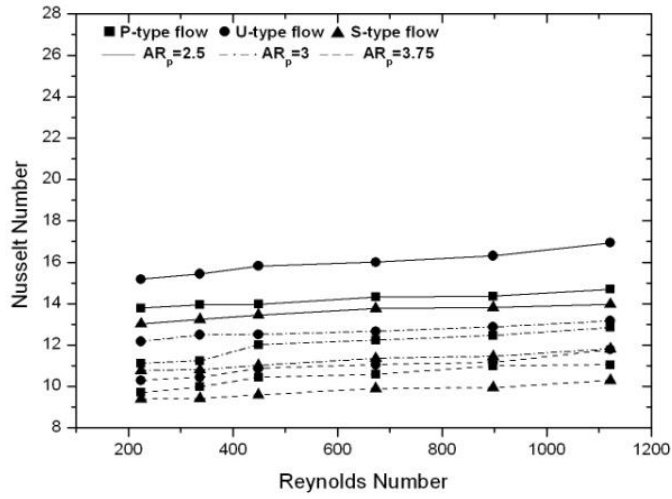


(b) $AR_c = 7.57$

Fig.4.6: Effect of plenum aspect ratios on Nusselt number at constant 125 W heat input and channel aspect ratio

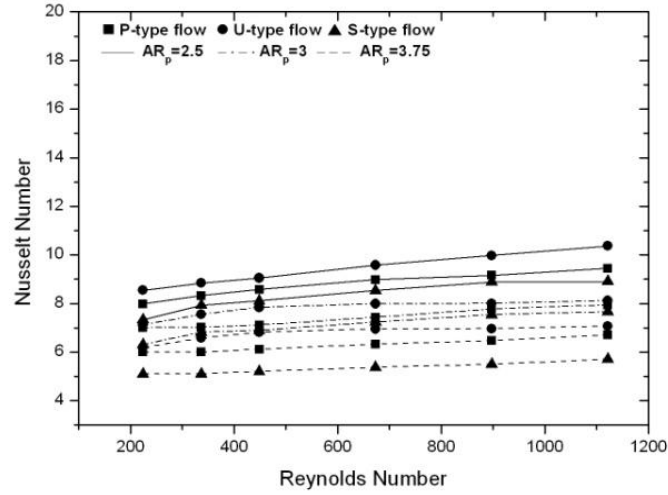


(a) $AR_c = 4.72$

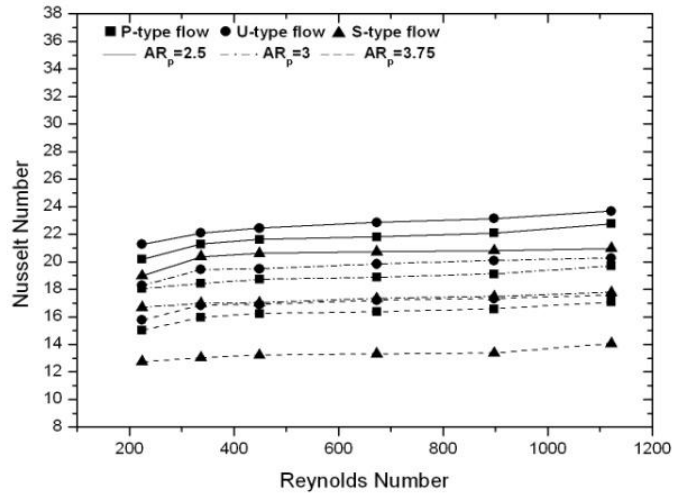


(b) $AR_c = 7.57$

Fig.4.7: Effect of plenum aspect ratios on Nusselt number at constant 225 W heat input and channel aspect ratio



(a) $AR_c = 4.72$



(b) $AR_c = 7.57$

Fig.4.8: Effect of plenum aspect ratios on Nusselt number at constant 375 W heat input and channel aspect ratio

Regarding the selection of critical values of channel aspect ratio, Lee and Garimella (2006) observed that the dimensionless thermal entrance length (defined as the ratio of thermal

entrance length to the product of Reynolds number, hydraulic diameter and Prandtl number) reduces as the channel aspect ratio increases. For a channel aspect ratio of 10, dimensionless thermal entrance length was observed to be around 0.02. For the accurate prediction of heat transfer performance the entrance effects need to be taken into account in the developing section. For an aspect ratio less than 1.0, it is seen that the local Nusselt number increases but the corresponding large pressure drop needs to be examined in the future work.

As far as the plenum aspect ratio is concerned the maximum Nusselt number is obtained only at the lowest aspect ratio for a given flow arrangement. This means that for the fixed value of plenum width, when the length of the plenum (Refer Figure 3.4) is increased the convective heat transfer is increased. This is because with increase in length of the plenum the flow length of flow regime in the inlet/outlet plenum available for the fluid entering/leaving the channels, increases resulting in enhanced convective heat transfer. It is to be noted that the cooling process that takes place in MCHS is a dynamic process and by the time the fluid enters and comes out of the microchannels the heat should have been absorbed by the liquid. This can take place without interruption only when sufficient length of flow regime is maintained within the inlet/outlet plenum and this is well achieved with increased length of plenum. In addition to this, the increased plenum length will also minimize any entry length phenomena associated with hydrodynamic flow conditions within the channels.

4.3.2 Effect of flow arrangements

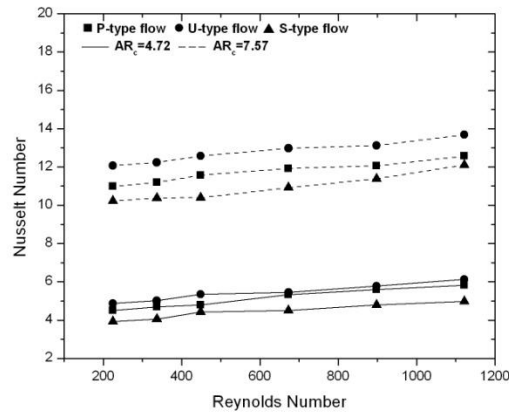
With regard to the flow arrangements, (Refer Table 3.1) one can easily notice that U-type flow arrangement ensures prolonged flow regime in the plenums compared to P-type and S-type flow arrangements. In the P-type flow arrangement as the fluid enters the plenum in the perpendicular direction, it strikes the plenum base directly but this does not ensure perfect fluid distribution within the microchannels. When we look at the S-type flow arrangement, due to its position of outlet just opposite to the inlet position, here again the flow distribution is not fully ensured. However, in the case of U-type flow arrangement the inlet and exit positions of the flow lie on the same side, ensuring longer flow regimes in the inlet/outlet plenum which attributes to the maximum Nusselt number. Figure 4.6(b) shows the Nusselt number variation for the channel aspect ratio, 7.57. The trend is almost very similar to the one observed for the channel aspect ratio, 4.72, except another considerable variation observed at lower Reynolds numbers for P-type

and S-type flow arrangements at $AR_p=3.0$ and 3.75 . However, with increase in channel aspect ratio, there is a overall increase in Nusselt number for all the cases of flow arrangements and plenum aspect ratios but maintaining the same trend of yielding the maximum Nusselt number at lower value of plenum aspect ratio. The reasons for this can be explained as follows: when the channel aspect ratio is increased, that is for the constant channel depth, the width of the channel is decreased, and the flow distribution within the channels is well streamlined so that the resistance for heat transfer decreases. This decrease in width has resulted in increase in flow velocity from 0.38 m/s to 1.92 m/s corresponding to the six Reynolds numbers considered in the present work. When velocity is increased, the wall shear stress also increases resulting in enhanced fluid momentum transport within the channel width at any given Reynolds number. Hence the bulk heat transport from the channel wall is also increased resulting in increase in Nusselt number with Reynolds number. It can be observed from Figure 4.6 that as the channel aspect ratio is increased from 4.72 to 7.57 , the Nusselt number value is almost doubled for all the plenum aspect ratios and flow arrangements.

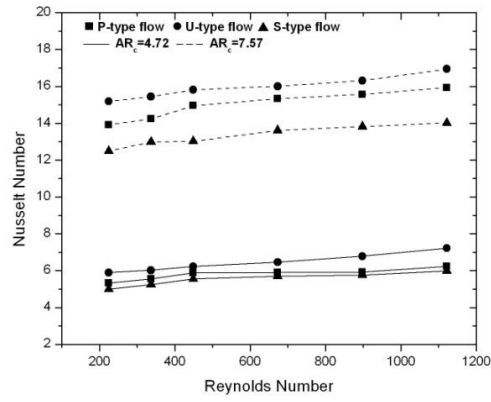
Figures 4.7(a) and 4.7(b) show the Nusselt number variation with Reynolds number for $AR_c=4.72$ and 7.57 respectively for a heat input of 225 W. For this case also, the maximum Nusselt number was observed only at $AR_p=2.5$ and $AR_c=7.57$ for U-type flow arrangement. The increase in Nusselt number almost follows the trend exhibited for the heat input of 125 W. However, the difference in Nusselt number due to the variation in AR_p is more pronounced for $AR_c=7.57$ when the heat input was increased from 125 W to 225 W. A similar trend is also observed for change in flow arrangement as well. It is to be noted that the minimum Nusselt number observed at $AR_c=7.57$ for S-type flow arrangement is much higher than the maximum Nusselt number obtained at $AR_c=4.72$ for U-type flow arrangement. It is clear from these behaviors that the channel aspect ratio plays an important role in deciding the maximum convective heat transfer at a lower plenum aspect ratio. However, proper combination of channel aspect ratio with plenum aspect ratio and flow arrangement will yield the maximum Nusselt number, making the MCHS compact for a given heat input. Nusselt number results obtained for heat input of 375 W is depicted in Figures 4.8(a) and 4.8(b) for $AR_c=4.72$ and 7.57 respectively. The Nusselt number continues to increase with increase in Reynolds number for $AR_c=4.72$ as well as for $AR_c=7.57$. However, a considerable increase is observed for U-type flow

arrangement followed by P- and S-types for the lowest plenum aspect ratio. The reason for this behavior may be due to the increased heat potential available for the fluid stream for the same inlet temperature of water. When the plenum length decreases, that is with increase in plenum aspect ratio, the heat transport from the wall decreases because of the entrance and exit conditions at the plenums. As the channel aspect ratio is increased to 7.57, the values of Nusselt numbers are more than doubled compared to the values observed for $AR_c=4.72$ and the maximum Nusselt number is observed for U-type flow arrangement at $AR_p=2.5$ as observed in the previous cases. This is because, reduced channel width and increased plenum length has favoured the development of high wall shear stress resulting in high fluid momentum transport and the high heat intensity at this favourable flow conditions has resulted in enhanced heat transport from the wall to the bulk of the fluid, resulting in increased convective heat transfer. However, the rate of increase of Nusselt numbers with Reynolds number is not pronounced as noted for the case of $AR_c=4.72$. At this stage the flow field and thermal conditions might have been well established to achieve the maximum convective heat transfer. At higher value of channel aspect ratio, the plenum aspect ratio has significant effect for 375 W heat input.

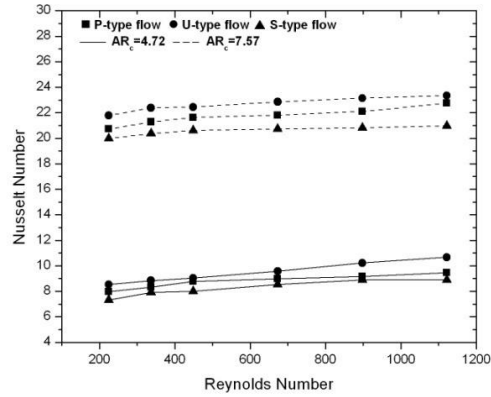
Figures 4.9 to 4.11 shows the variation of Nusselt number with increase in Reynolds number for a constant heat input and plenum aspect ratio. It has been observed that as the channel aspect ratio increases (represented by dashed line) the Nusselt number also increases for all the cases, the reason is attributed due to increase in surface area. It has been observed that as the channel aspect ratio or plenum aspect ratio increases, the performance of the MCHS also enhances, but at the expense of pressure drop. The variation of channel aspect ratio and plenum aspect ratio on the pressure drop characteristics is analysed in the subsequent section.



(a) Heat input=125 W

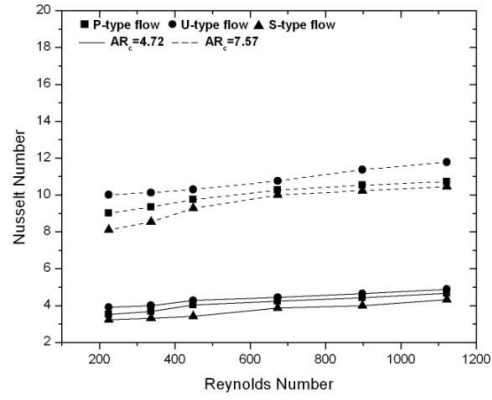


(b) Heat input=225 W

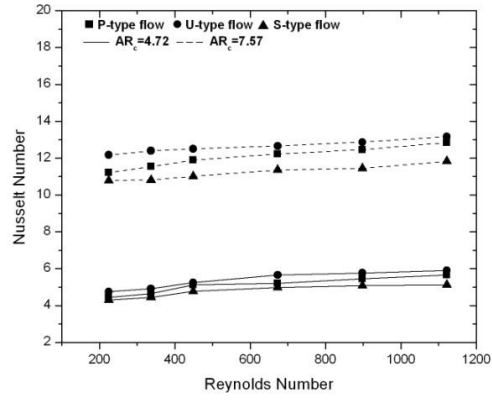


(c) Heat input=375 W

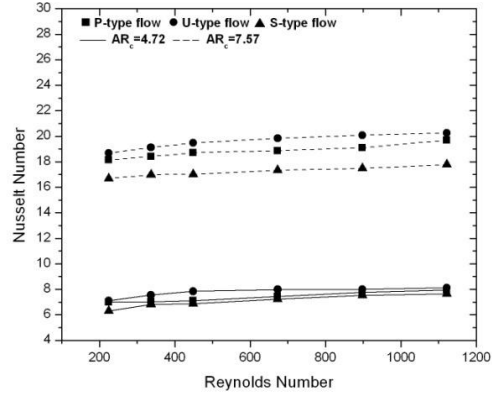
Fig.4.9: Effect of channel aspect ratios on Nusselt number at constant heat input and $AR_p=2.5$



(a) Heat input=125W

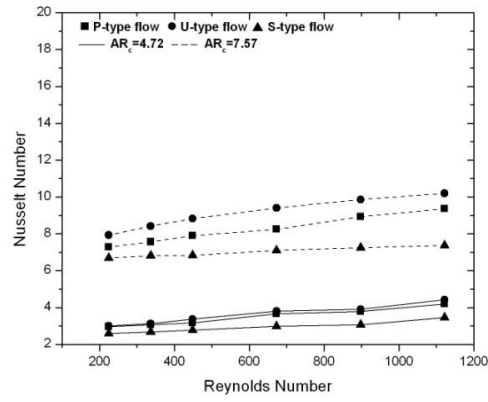


(b) Heat input=225W

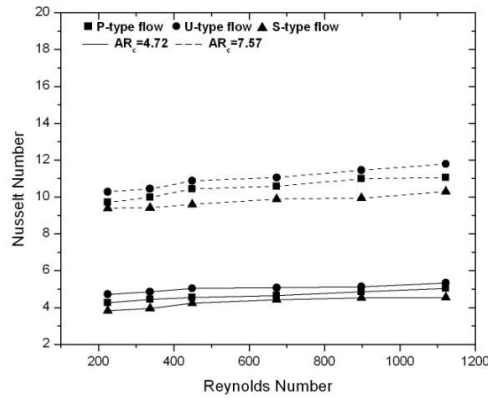


(c) Heat input=375W

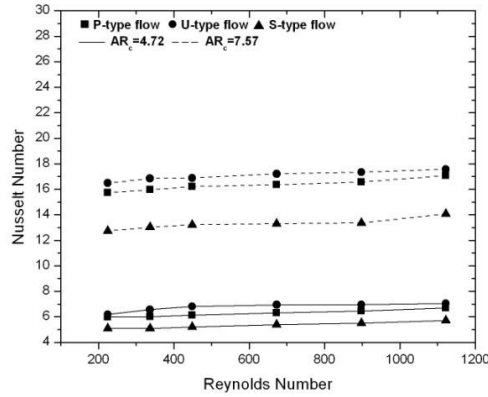
Fig.4.10: Effect of channel aspect ratios on Nusselt number at constant heat input and $AR_p=3.0$



(a) Heat input=125W



(b) Heat input=225W



(c) Heat input=375W

Fig.4.11: Effect of channel aspect ratios on Nusselt number at constant heat input and $AR_p=3.75$

4.4 Pressure Distributions

4.4.1 Effect of Reynolds number on pressure drop variation for various flow arrangements

Figure 4.12 depicts the variation of total pressure drop as a function of Reynolds number for three flow arrangements at three heat inputs. The measured pressure drop is in good agreement with the estimated values. A gradual change in the pressure drop with Reynolds number is attributed to the temperature dependence of fluid viscosity, and the increasing contraction and expansion pressure losses at the inlet and outlet plenum, respectively. The maximum variation of total pressure drop (ΔP) with Reynolds number (Re) is experienced in S-type flow arrangement followed by U-type and P-type flow arrangements.

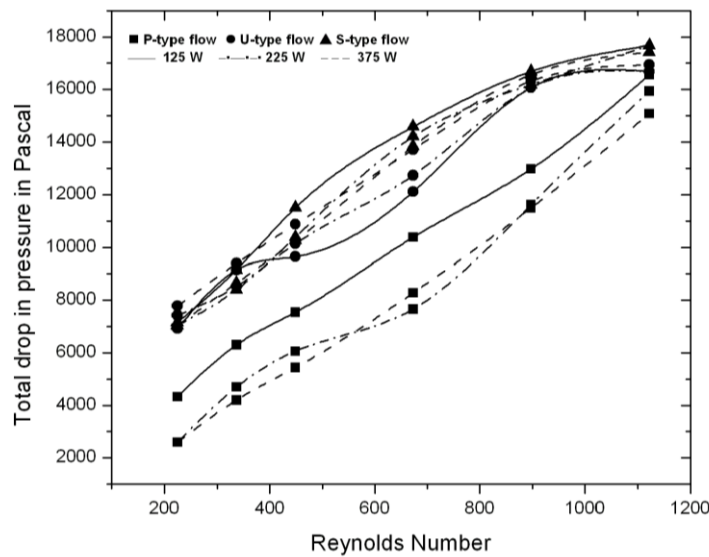


Fig. 4.12: Variation of Reynolds number on total pressure drop (plenum size: 10mm x 30mm x 2.5mm, channel width 330 μ m)

Table 4.4 shows the variation of differential pressure drop at five positions (Refer Figure 3.11b) within the plenums for 125 W, 225 W and 375 W heat inputs. The maximum pressure fluctuations were experienced in P-type of flow arrangement which leads to instability effects. The reason for the maximum fluctuations for P-type flow arrangement may be attributed to the fluid striking the test piece top surface perpendicularly creating more disturbance as compared to the S-type and U-type flow arrangements. The retention time taken by water to flow from inlet plenum to outlet plenum was least for P-type of flow. The CFD flow trajectory analysis

(discussed in Chapter 5) of the particles also validates the physics behind the flow arrangement. For the S-type of flow arrangement, at position 3 (T3-T8 and T13-18) minimum instability was experienced within the plenums because it is at the mid region of the plenum and had crossed the hydrodynamic entry length effect. The pressure drop variation of U-type flow arrangement was in-between the P-type and S-type. Similar types of trends were experienced for 225 W and 375 W heat inputs.

4.4.2 Effect of Reynolds number on friction factor ratio for various flow arrangements

Figure 4.13 shows the variation of friction factor ratio (experimental to theoretical: $f_{\text{exp}}/f_{\text{thy}}$) with Reynolds number. The variation clearly denotes the flow to be laminar and the friction factor is decreasing with increase in Reynolds number. The computed value of $f_{\text{exp}}/f_{\text{thy}}$ in the present work varies from 0.4 to 1.4 which agrees with the trend reported by Chen et al. (2004). Here theoretical and the experimental friction factor are taken as:

$$f_{\text{thy}} = \frac{64}{\text{Re}} \quad (4.2)$$

$$f_{\text{exp}} = \frac{2g_c \Delta P D_h}{L \rho_f V_f^2} \quad (4.3)$$

It is also observed that minimum friction factor has been experienced in P-type flow arrangement as compared to other two arrangements which has lead to its wide practical applications in industry.

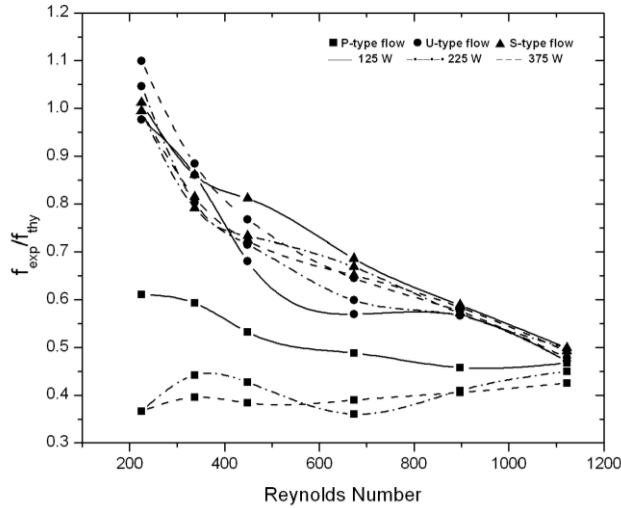
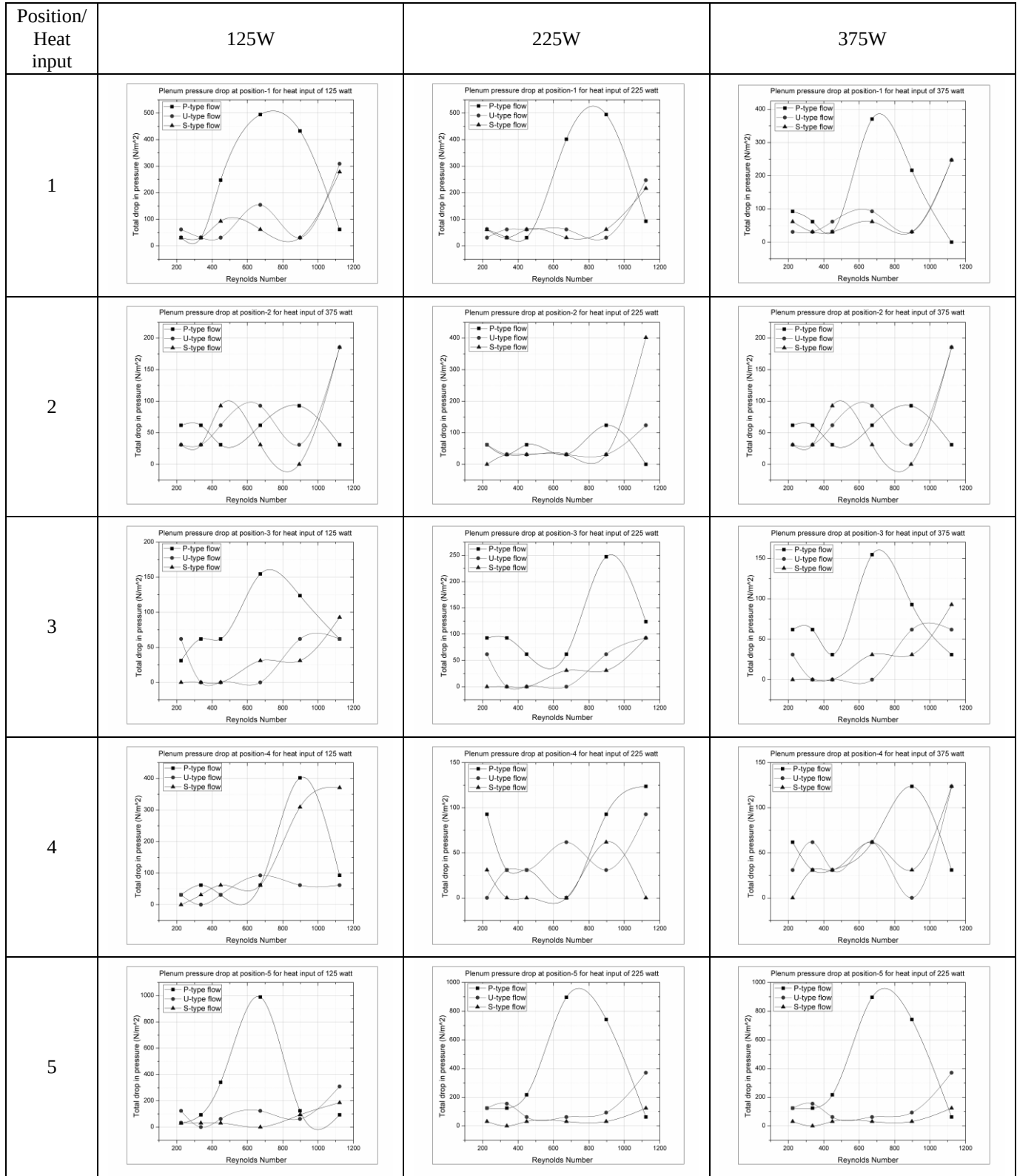


Fig 4.13: Variation of Reynolds number on friction factor coefficient (plenum size: 10 mm × 30 mm × 2.5 mm, channel width 330 μm)

Table: 4.4: Variation of differential pressure drop within the plenums



The above discussions on pressure distributions are for a constant plenum size of 10 mm × 30 mm × 2.5 mm and constant microchannel width of 330 μm. The effects of variation of plenum size and microchannel width on performance of MCHS based on pressure variation are studied in the subsequent section.

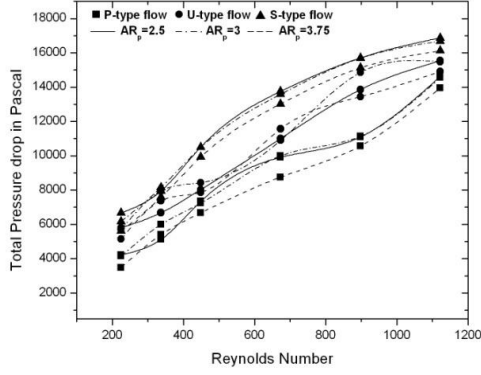
4.5 Effect of aspect ratio and flow arrangements on the performance of MCHS based on pressure drop variation

The total pressure drop between the inlet and outlet plenum are obtained for all the test cases and are plotted as a function of Reynolds number, channel aspect ratio, plenum aspect ratio and flow arrangements for three heat inputs. Figures 4.14(a) and 4.14(b) show the total pressure drop for $AR_c=4.72$ and 7.57 respectively for three values of AR_p and three flow arrangements at 125 W heat input. The maximum pressure drop is observed for higher channel aspect ratio, $AR_c=7.57$ and lowest plenum aspect ratio, $AR_p=2.5$ for S-type flow arrangement, whereas the minimum pressure drop is noted for the case of P-type flow arrangement for $AR_p=3.75$.

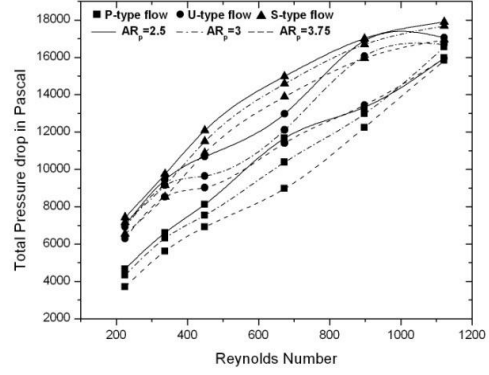
As the channel aspect ratio is increased, the fluid velocity through the channels also increases from 0.26 m/s to 0.38 m/s for the lowest Reynolds number, 224.3 and from 1.28 m/s to 1.92 m/s for the highest Reynolds number, 1121.7. The wall shear stress within the microchannels increases with increase in fluid velocity, resulting in higher pressure drop. The situation is different with plenum aspect ratio, wherein the pressure drop is found to be higher only for the lowest value of plenum aspect ratio. It is observed that the fluid flow travel length at the inlet/outlet plenum of the heat sink decreases with increase in plenum aspect ratio. Hence when the flow length is larger for the case of $AR_p=2.5$, the pressure drop is found to be maximum. When the flow arrangements are considered, the S-type flow arrangement involves slightly longer flow path with two bends and hence gives rise to higher pressure drop compared to the P- and U-type flow arrangements.

In all the cases there is a continuous rise in the pressure drop with increase in Reynolds number as expected. For both the channel aspect ratios the U-type flow arrangement with $AR_p=2.5$ that has given the maximum Nusselt number, takes a middle region of pressure drop as seen from the above figures. For the heat input of 225 W the total pressure drop are shown in Figures 4.14(c) and 4.14(d) for $AR_c=4.72$ and 7.57 respectively. Here again the maximum

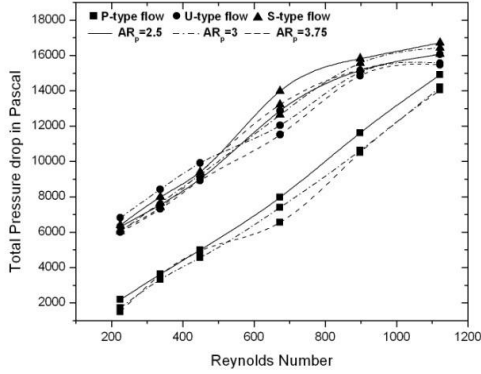
pressure drop seems to take place for S-type flow arrangement for $AR_p=2.5$ for both the channel aspect ratios.



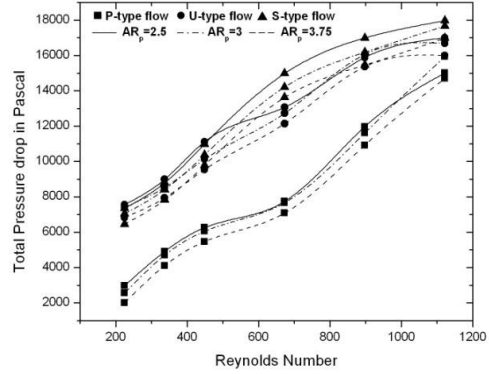
(a) Heat input=125W, $AR_c=4.72$



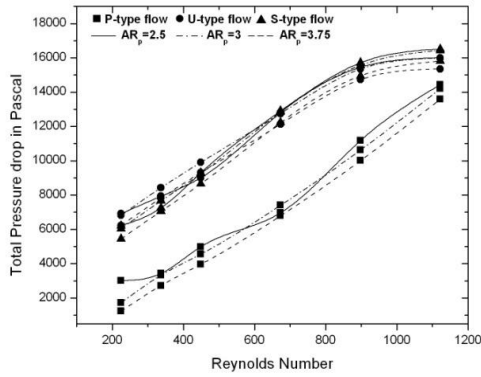
(b) Heat input=125W, $AR_c=7.57$



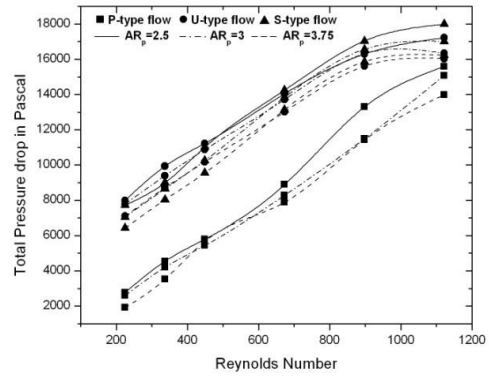
(c) Heat input=225W, $AR_c=4.72$



(d) Heat input=225W, $AR_c=7.57$

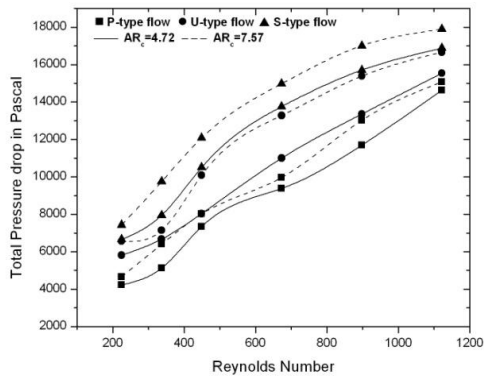


(e) Heat input=375W, $AR_c=4.72$

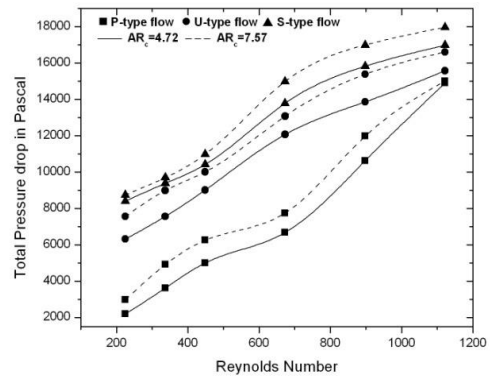


(f) Heat input=375W, $AR_c=7.57$

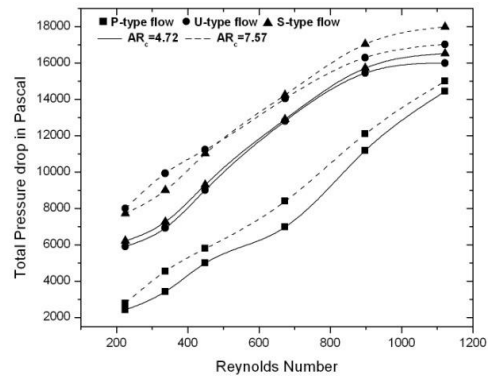
Fig.4.14: Effect of plenum aspect ratios on pressure drop at constant heat input and channel aspect ratio



(a) Heat input=125 W

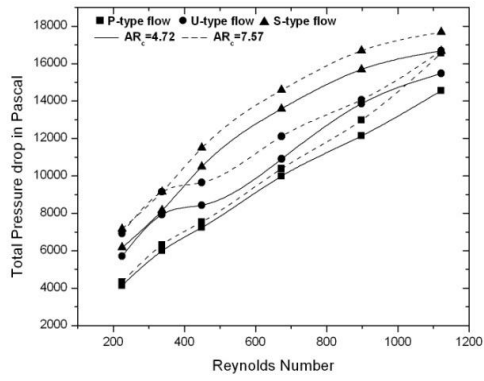


(b) Heat input=225 W

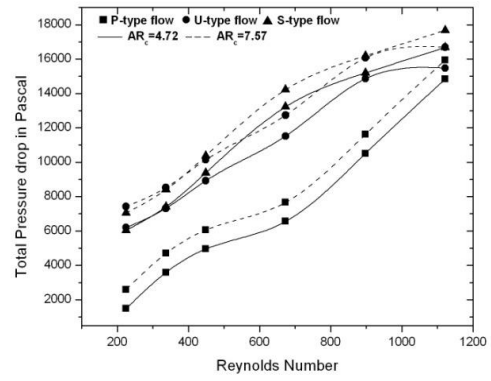


(c) Heat input=375 W

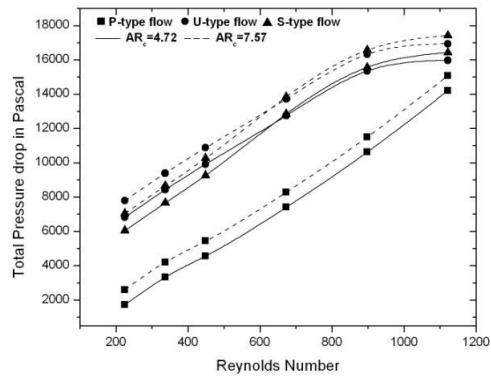
Fig.4.15: Effect of channel aspect ratios on pressure drop at constant heat input and $AR_p=2.5$



(a) Heat input=125W

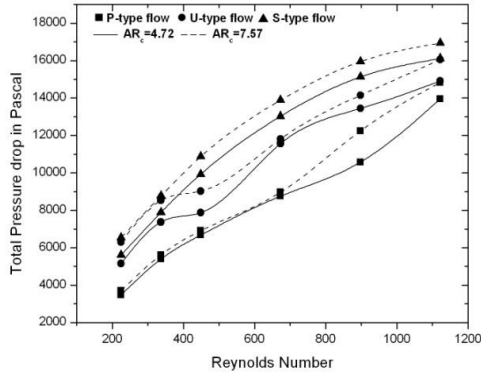


(b) Heat input=225W

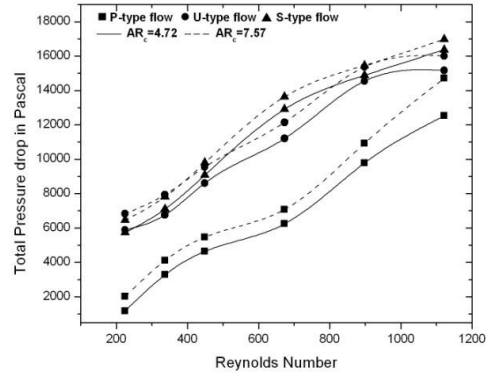


(c) Heat input=375W

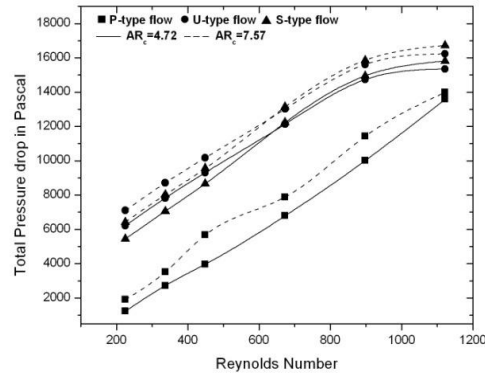
Fig.4.16: Effect of channel aspect ratios on pressure drop at constant heat input and $AR_p=3.0$



(a) Heat input=125W



(b) Heat input=225W



(c) Heat input=375W

Fig.4.17: Effect of channel aspect ratios on pressure drop at constant heat input and $AR_p=3.75$

However, all the pressure drop curves get clustered in two bunches with P-type flow arrangement occupying the lower pressure drop regions. The P-type and U-type flow arrangements at $AR_p=2.5$ give rise to a non-uniform pressure variation with Reynolds number.

Figures 4.14(e) and 4.14(f) show the pressure drop variation with Reynolds number for 375 W heat input for $AR_c=4.72$ and 7.57 respectively. It is seen from the above figures that the pressure drop increase with Reynolds number is more uniform compared to the cases shown in Figures 4.14 (a, b, c and d), except for a slight deviation observed at $AR_c=7.57$, $AR_p=2.5$ for P-type flow arrangement. Comparing Figures 4.14 one can easily notice that as the heat input increases from 125 W to 375 W, the lowest pressure drop obtained for P-type flow arrangement at $AR_p=3.75$ decreases. The reason may be that with increased heat input the temperature of the fluid stream increases, which will result in slight decrease in viscosity of water used as coolant in

the MCHS. The decrease in viscosity with increase in heat input from 125W to 375W was observed to be 14.83%. In addition to this, increased heat input has increased the pressure drop difference observed between the P-type arrangement and the other two types of flow arrangements. Figures 4.14 indicate that for $AR_c=4.72$ at 375 W heat input, the effect of plenum aspect ratio does not have significant effect on the pressure drop as compared to the variation in the pressure drop observed for 125 W heat input. Figures 4.15 to 4.17 indicate the variation of total pressure drop at constant plenum aspect ratio but varying heat inputs and channel aspect ratio. In all the figures (4.14 to 4.17) it was observed that the P-type flow arrangement gives minimum total pressure drop at a particular Reynolds number suggesting this type of flow arrangement is useful for practical applications for cooling of integrated circuits in general and computer processor in particular.

4.6 Correlation for Nusselt number

It is evident that designers need wide ranging correlations that can be used to predict thermal–hydraulic performance or to determine optimum geometric parameters for a particular application.

The dimensionless parameter Prandtl number Pr is used to represent the relative magnitude of momentum and thermal diffusivity of a fluid in convective heat transfer phenomenon.

$$Pr = \frac{\nu}{\alpha} \quad (4.4)$$

In the present work, the influence of channel geometry was considered by using dimensionless geometric parameter AR_c (defined in Equation 3.17). To consider the effect of geometrical influences, flow condition (using Reynolds number) and fluid properties in a functional form of Nusselt number correlation, e.g. $Nu = fn(Re, Pr, AR_c)$ is selected on the basis of simplicity and commonly used power laws. For simple situations these may often be written in power law relationships as

$$Nu = a Re^x Pr^y AR_c^z \quad (4.5)$$

where the constants a , x , y and z will change for different plenum geometries.

For the results to be useful to designers, statistical correlations are presented here for the microchannel heat sink which cover all combinations of the geometric, flow parameters and flow arrangements within the Reynolds number range of 224 - 1121 as a cooling solution for

integrated circuits. As discussed above, Nusselt number varies for different geometric, flow parameters and flow arrangements. The experimental data corresponding to all the eighteen test pieces (in all, 324 trail runs) were used for regression analysis to find a relationship that yields a best fit equation for Nusselt number. Multiple non-linear regression technique is used for developing new correlations. The proposed correlations are:

❑ P-type flow arrangement ($R_e < 2000$; $2.5 \leq AR_p \leq 3.75$)

$$Nu = 0.0056 (R_e)^{0.1190} (Pr \times AR_c)^{1.1743} \quad (4.6)$$

❑ U-type flow arrangement ($R_e < 2000$; $2.5 \leq AR_p \leq 3.75$)

$$Nu = 0.0204 (R_e)^{-0.005} (Pr \times AR_c)^{1.6731} \quad (4.7)$$

❑ S-type flow arrangement ($R_e < 2000$; $2.5 \leq AR_p \leq 3.75$)

$$Nu = 0.0138 (R_e)^{0.0114} (Pr \times AR_c)^{1.6265} \quad (4.8)$$

where R_e is Reynolds number, Pr is the Prandtl number, AR_p is plenum aspect ratio and AR_c is channel aspect ratio.

In order to compare the scatter between Nusselt number (experimental) and Nusselt number (correlation), parity charts are drawn and are shown in Figure 4.18 to 4.20.

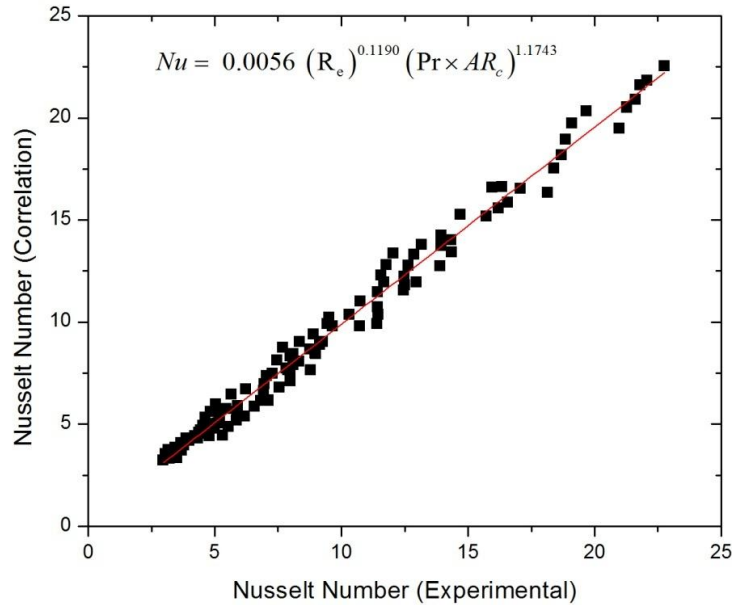


Fig.4.18: Parity plot between Nusselt number (experimental) and Nusselt number (correlation) for P-type flow arrangement

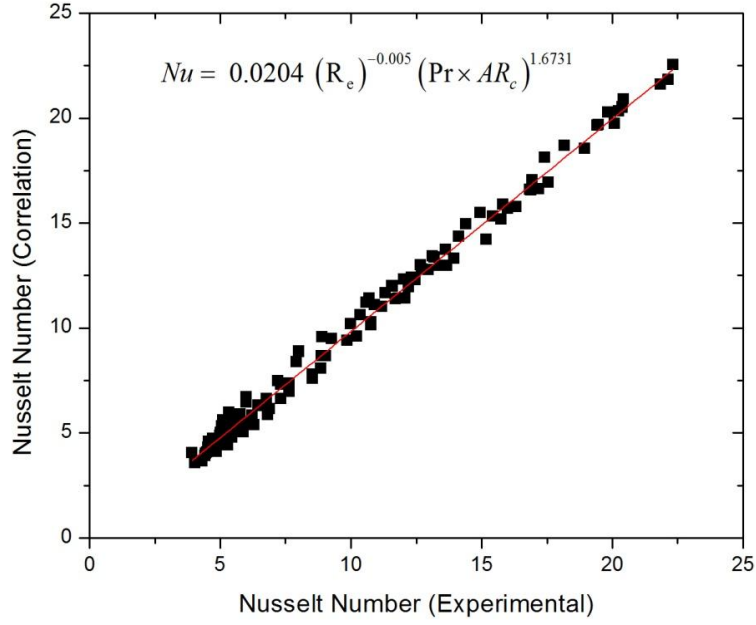


Fig.4.19: Parity plot between Nusselt number (experimental) and Nusselt number (correlation) for U-type flow arrangement

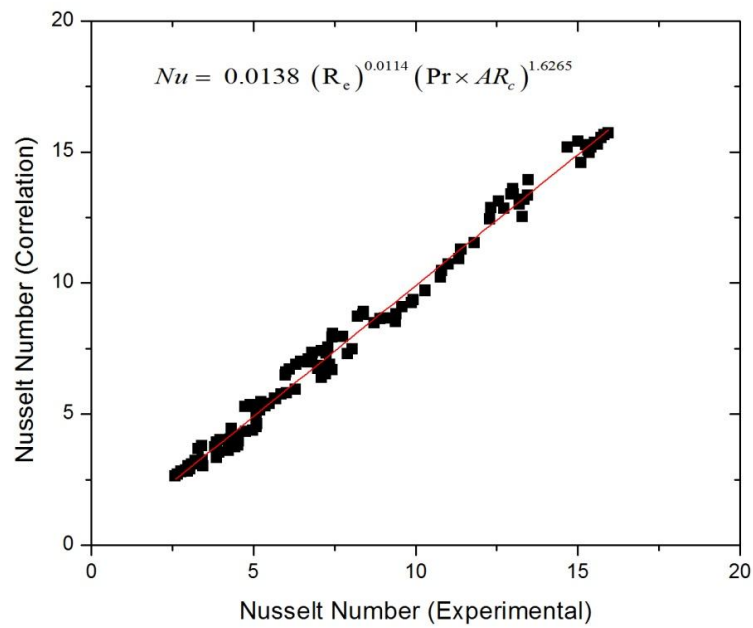


Fig.4.20: Parity plot between Nusselt number (experimental) and Nusselt number (correlation) for S-type flow arrangement

CHAPTER 5

CFD ANALYSIS OF MICROCHANNEL HEAT SINK

5.1 Introduction

In this chapter CFD modelling of thermal performance of microchannel heat sink with deionised water as working fluid has been formulated. The governing equations which constitute the thermal and hydrodynamic behaviour of the system are explained. The geometry of the MCHS which is under consideration is also discussed in detail. Boundary and initial conditions required for simulation studies are described in this chapter. MCHS which has the same dimensions as that of the one used in the experimental test setup was modelled and simulated using Flow Simulation commercial software.

5.2 Description of model

Computational Fluid Dynamics (CFD) commercial software Solidworks Flow simulation was used to study the fluid flow, heat transfer and associated physical phenomena for the MCHS considered in the study. A set of mathematical model equations based on conservation laws are solved using this software in order to obtain the flow variables throughout the computational domain. Validation of the simulation analysis is achieved by comparing CFD results with available experimental results. The three-dimensional microchannel heat sink having same geometrical parameters as experimental set piece is analyzed using deionized water as the cooling fluid. A schematic of the structure of a rectangular microchannel heat sink is shown in Figure 5.1. A uniform surface heat flux is applied at the bottom surface of the heat sink. This heat is being conducted through the heat sink material, thereby reaching the channel regions where the heat flux is removed by convection heat transfer. The heat flux available over the inside surface of the microchannel walls causes an increase in temperature of the fluid. This temperature rise in combination with fluid momentum transport results in the development of hydrodynamic and thermal boundary layers which help to establish heat flow by conduction within the boundary layer and by bulk movement in the free stream of the fluid. These physical phenomena can be represented in mathematical form using continuity, momentum and energy equations as described in Section 5.3.

SolidWorks Flow Simulation software is capable of creating physical domain of the MCHS, discretization (meshing) and analysis. The microchannel inner faces are declared as walls and other walls are declared as walls with insulation boundary but the bottom most region as shown in Figure 5.1(b) is declared as the wall subjected to constant heat flux boundary condition. The MCHS inlet, outlet plenums and microchannels are declared as the fluid zone and the rest is declared as solid zone. This declaration helps to detect and recognize in Flow Simulation for boundary condition and zone assignment. Figure 5.2 shows the geometry of a single microchannel computational model, indicating the direction of fluid flow and heat input. Table 5.1 shows the geometric dimensions of a single microchannel model. For the CFD simulations, three flow arrangements as used in the experimental work are modelled as shown in Figure 5.3.

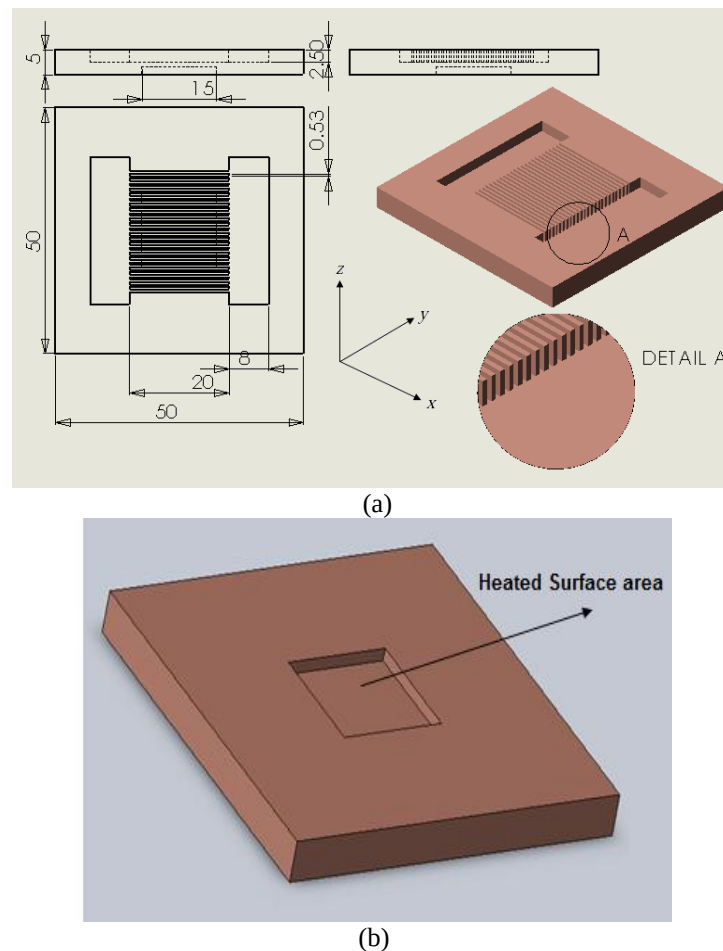


Fig.5.1 (a) Dimensions of computational model of MCHS (b) Back side view of MCHS model

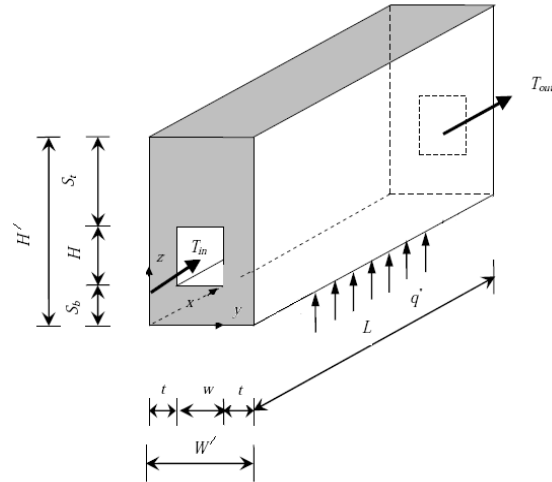


Fig.5.2 Geometry of a single microchannel computational model (without plenums)

Table 5.1 Geometric dimensions of a single microchannel

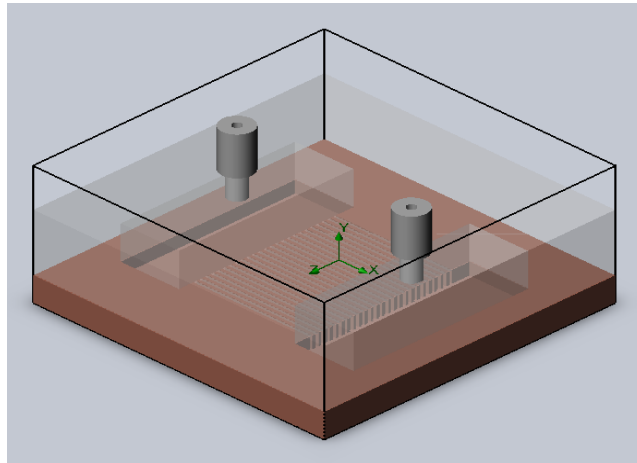
H' (mm)	H (mm)	W (μm)	S _b (mm)	t (μm)	L (mm)
15.0	2.5	530/ 330	2.5	250	20

5.3 Governing equations

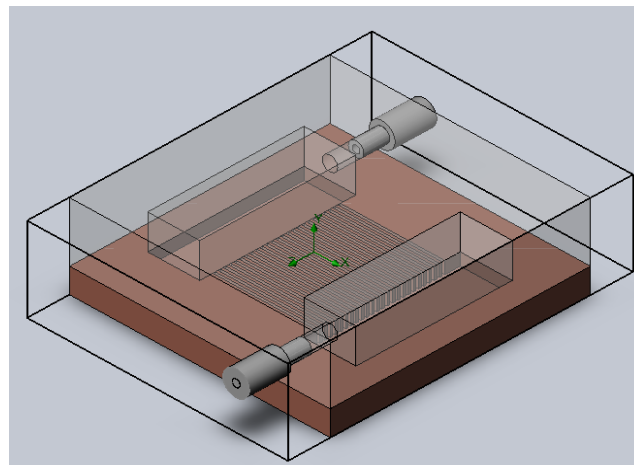
Flow Simulation solves the Navier-Stokes equations, which are formulations of mass, momentum and energy conservation laws for fluid flows. The equations are supplemented by fluid state equations defining the nature of the fluid, and by empirical dependencies of fluid density, viscosity and thermal conductivity on temperature. A particular problem is finally specified by the definition of its geometry, boundary and initial conditions. The following assumptions are made in the simulation of MCHS:

- steady state flow and heat transfer
- incompressible fluid
- laminar flow
- uniform wall heat flux
- constant solid properties (thermophysical properties)
- negligible radiation of heat transfer
- negligible superimposed natural convective heat transfer
- no slip condition.

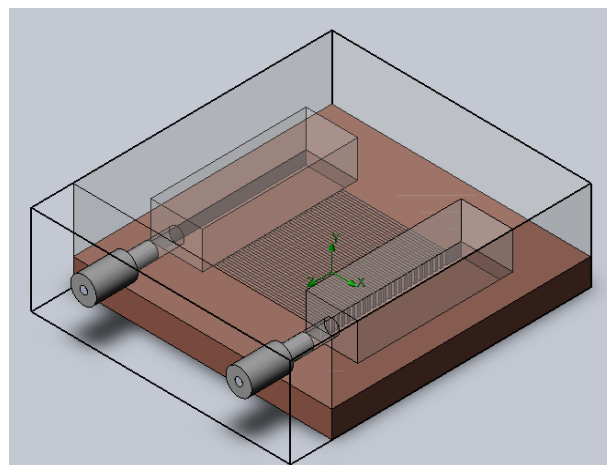
Following are the flow and energy equations used in the simulation.



(a) P-type



(b) S-type



(c) U-type

Fig. 5.3: Different types of simulated flow arrangement

Navier-Stokes equations

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (5.1)$$

Momentum equations

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_j u_i - \tau_{ji}) + \frac{\partial P}{\partial x_i} = S_i \quad (5.2)$$

Viscous stress tensor

$$\tau_{ii} = -\frac{2}{3}\mu D + 2\mu \frac{\partial u_i}{\partial x_i} \quad (5.3)$$

$$\tau_{ij} = \mu \frac{\partial u_i}{\partial x_j} \quad (5.4)$$

Energy equation

$$\frac{\partial(\rho h)}{\partial t} + \frac{\partial(\rho u_i h)}{\partial x_i} = -\frac{\partial p}{\partial t} + \frac{\partial}{\partial x_i} \left[\frac{k}{c_p} \frac{\partial h}{\partial x_i} - \dot{q}_i^R \right] \quad (5.5)$$

The generalized governing equations for variable ϕ can be expressed as

$$\frac{\partial}{\partial t}(\rho \phi) + \frac{\partial}{\partial x_j} \left(\rho u_j \phi - \frac{\Gamma_\phi}{\sigma_\phi} \frac{\partial \phi}{\partial x_j} \right) = S_\phi \quad (5.6)$$

$$\frac{\partial(\rho \phi)}{\partial t} + \frac{\partial(\rho u \phi)}{\partial x} + \frac{\partial(\rho v \phi)}{\partial y} + \frac{\partial(\rho w \phi)}{\partial z} = \frac{\partial}{\partial x} \left[\Gamma \frac{\partial \phi}{\partial x} \right] + \frac{\partial}{\partial y} \left[\Gamma \frac{\partial \phi}{\partial y} \right] + \frac{\partial}{\partial z} \left[\Gamma \frac{\partial \phi}{\partial z} \right] + S_\phi \quad (5.7)$$

where Γ is a diffusion coefficient, σ is Schmidt/Prandtl number, and S is a general source term

For continuity, $\phi = 1$, $\Gamma = 0$, $S = 0$

For momentum, $\phi = \{u, v, w\}$, $\Gamma = f(\mu)$

For energy, $\phi = e$ (or h), $\Gamma = k/c_p$, $S = \text{radiation}$

Using the finite volume method (FVM), Solidworks integrate the generalized transport equation as follows:

$$\int_{\Omega} \frac{\partial}{\partial t}(\rho \phi) d\Omega + \int_s \frac{\partial}{\partial x_j} \left(\rho u_j \phi - \frac{\Gamma_\phi}{\sigma_\phi} \frac{\partial \phi}{\partial x_j} \right) dS = \int S_\phi d\Omega \quad (5.8)$$

The above governing equations (5.1) to (5.7) are solved using SolidWorks Flow Simulation commercial software to obtain results for velocities, u , v , w and temperature T . The fluid flow and heat transfer analysis involves a number of basic steps that are shown in the following flow chart in Figure 5.4.

Initial conditions

$T=290$ K;

pressure at outlet adapter for all the three flow arrangements= 101325 Pa;

velocity at inlet adapter for all the three flow arrangements = 0.41 m/s.

The time dependent governing equations (5.1) to (5.8) are solved assuming the above initial conditions until steady state conditions are reached.

Boundary Conditions

For a steady, laminar flow, $v = 0$ and $w = 0$. If the whole unit cell is chosen as a unitary domain, the boundary conditions can be specified as follows.

Hydrodynamic Boundary Conditions

The velocity is zero at all boundaries except the channel inlet and outlet.

(i) At the inner bottom wall of channel (no-slip condition)

$$u = 0, v = 0, w = 0$$

$$\text{for } z = S_b, t < y < t + W$$

(ii) At inlet

$$p_1 = p_{in}, v = 0, w = 0$$

$$\text{for } x = 0, t < y < t + W \text{ and } S_b < z < S_b + H$$

(iii) At outlet

$$p_1 = p_{out}$$

$$\text{for } x = L, t < y < t + W \text{ and } S_b < z < S_b + H$$

Thermal Boundary Conditions

Adiabatic boundary conditions are applied to all the boundaries of the solid region except the heat sink bottom wall, where a constant heat flux is applied.

$$(i) -k_s \frac{\partial T}{\partial z} = q''$$

$$\text{for } 0 < x < L, 0 < y < W \text{ and } z = 0$$

(ii) At inlet, the liquid temperature is equal to a given constant inlet temperature.

$$T = T_{in}$$

$$\text{for } x = 0, t < y < t + W \text{ and } S_b < z < S_b + H$$

(iii) The flow is assumed to be thermally fully developed at the channel outlet

$$\frac{\partial^2 T}{\partial x^2} = 0$$

$$\text{for } x = L, t < y < t + W \text{ and } S_b < z < S_b + H$$

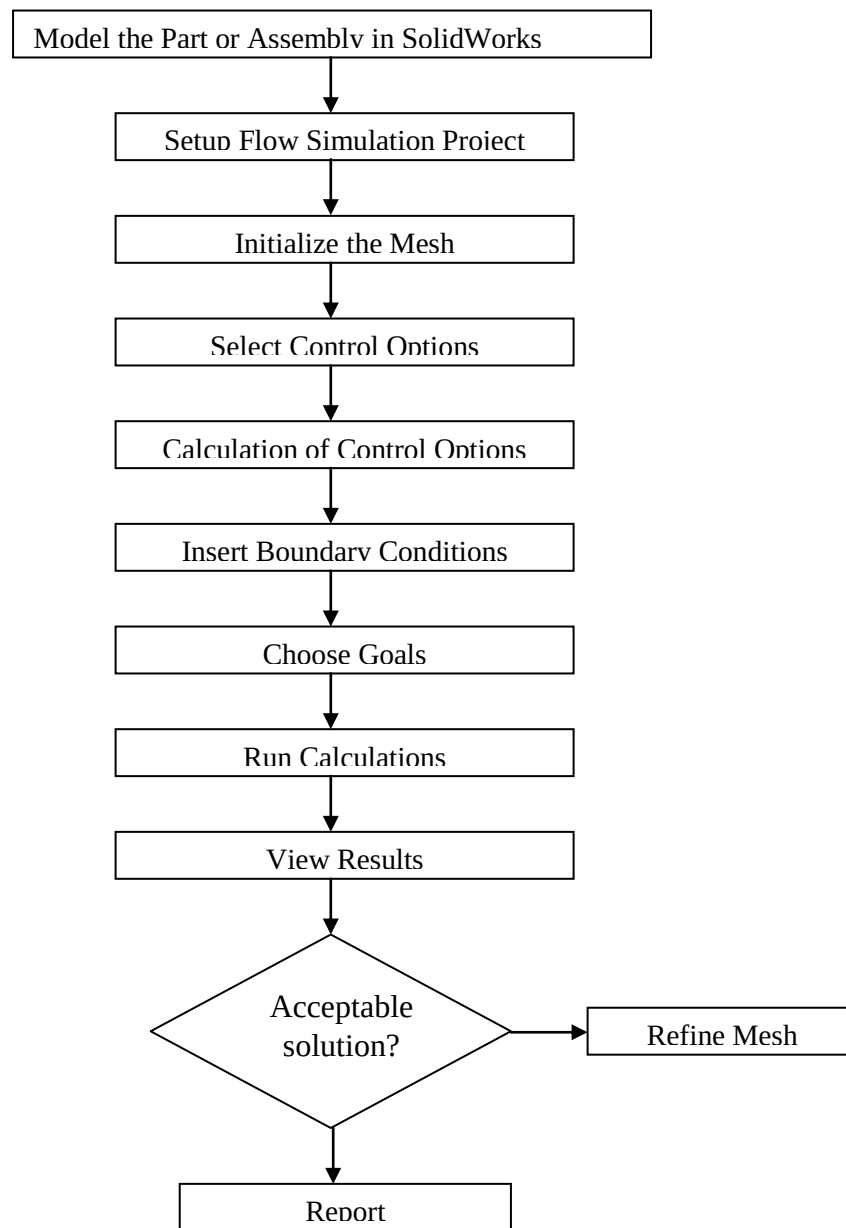


Fig. 5.4: Flow Chart for fluid flow analysis using SolidWorks

5.4 Computational mesh

While performing any numerical analysis of complex processes, the computational mesh generation is an important procedure. The analysis of MCHS was performed using SolidWorks Flow Simulation a Computational Fluid Dynamics (CFD) tool. In SolidWorks Flow Simulation rectangular computational mesh in the computational domain is used, the mesh cells' sides are orthogonal to the specified axes of the Cartesian coordinate system and are not fitted to the solid/fluid interface [Remsburg et al. (2010)]. Due to this, the solid/fluid interface cuts the near-wall mesh cells. As a result these cells in the solid/fluid interface are called partial cells so as to treat the mass and heat fluxes appropriately.

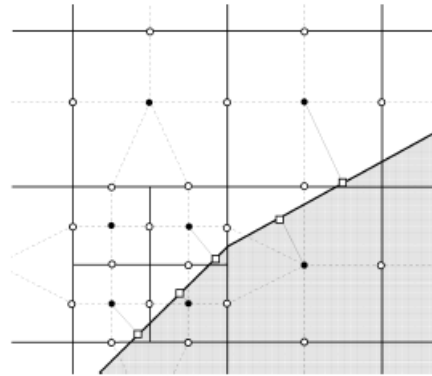
In SolidWorks Flow Simulation [SolidWorks Flow Simulation-Technical Reference (2009)] the computational domain is divided into slices by the basic mesh planes, which are orthogonal to the axes of the Cartesian coordinate system. The basic mesh does not depend on the solid/fluid interface but is calculated by the computational domain used. In SolidWorks Flow Simulation every individual basic mesh cell intersecting with the solid/fluid interface is split into 8 uniform child cells and then every child cells intersecting with the interface is in turn split into 8 cells of next level, and so on, until the specified cell size is attained.

The mesh refinement process in SolidWorks Flow Simulation at the solid/fluid interface continues till the maximum angle between the normal to the surface inside one cell does not exceed certain threshold, otherwise the cell is split into 8 cells again. When a locally refined rectangular computational mesh is obtained, then governing equations are solved on it. In SolidWorks the computational mesh is refined further during the calculation in accordance with the spatial gradients of the solution. It is seen that in the low-gradient regions, the cells are merged, whereas in the high-gradient regions, the cells are split.

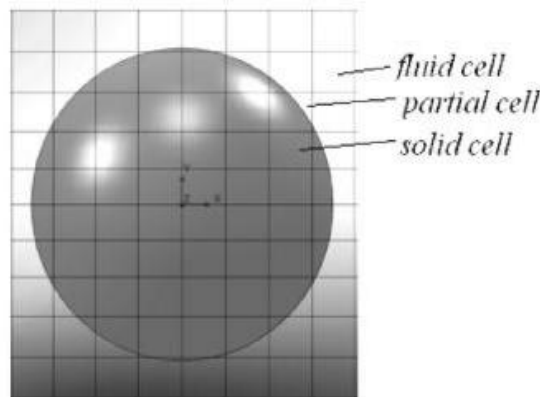
In the present work the computational mesh refinement during the calculation are prescribed manually. SolidWorks use cell-centered finite volume (FV) method to obtain conservative approximations of the governing equations on the locally refined rectangular mesh. Figure 5.5a shows the solid/fluid interface, boundary conditions and geometry taken into account for the CFD tool. A SIMPLE-like approach as defined by Patankar (1980) was used to discrete pressure equations.

5.4.1 Mesh generation

In SolidWorks Flow Simulation, mesh contains basic cells of three different types: fluid cells, partial cells and solid cells (Figure 5.5b). By varying the densities of the computational mesh, simulations are run till the solution converges. Generally it has been observed that fine mesh in fluid flow simulations may require a substantial amount of RAM and calculations take a very long time to reach convergence.



(a)



(b)

Fig. 5.5: (a) Computational mesh cells at the solid/fluid interface (b) Different types of mesh cells (Source: SolidWorks Flow simulation 2009 Technical Reference)

In the CFD simulations, grid sensitivity analysis is performed to determine the mesh size which will produce results independent of the discretization size of the computational domain. In the present research, the MCHS model is meshed number of times, Table 5.2 shows the number of cells for the various cases along with the CPU time taken. It has been observed that there is more deviation between Case 1 and Case 3, while for Cases 4 onwards the deviation decreases. The

percentage variation of temperature for inlet and outlet plenum for Cases 5, 6 and 7 was less than 1% as shown in Figure 5.6a for a plenum size of $30 \text{ mm} \times 8 \text{ mm} \times 2.5 \text{ mm}$. Table 5.3 shows the mesh details for one such analysis along with the values of convective heat transfer coefficient for different number of cells. For the purpose of grid sensitivity analysis, the convective heat transfer coefficients are calculated from the temperature values (details in section 5.5) and compared for different grid densities. It can be seen from the above table that as the mesh density is increased, the percentage error computed with respect to the next denser mesh, decreases. This indicates that the present formulation employed for CFD simulation is capable of producing consistent results for convective heat transfer within the microchannel. These results are also illustrated in Figure 5.6b for better clarity. One can easily notice that from mesh 4 onwards the convective heat transfer coefficient almost becomes independent of the mesh size of the computational domain.

Table 5.2: Simulated Fluid/Solid/Partial Cells with corresponding CPU time

Case	Fluid Cells	Solid Cells	Partial Cells	CPU Time
1	3034	7566	15814	0:8:30
2	5412	17076	26520	0:14:54
3	22812	49586	58338	1:46:44
4	36812	53736	73493	2:12:41
5	58448	78615	111851	3:13:32
6	85260	96084	130998	3:46:35

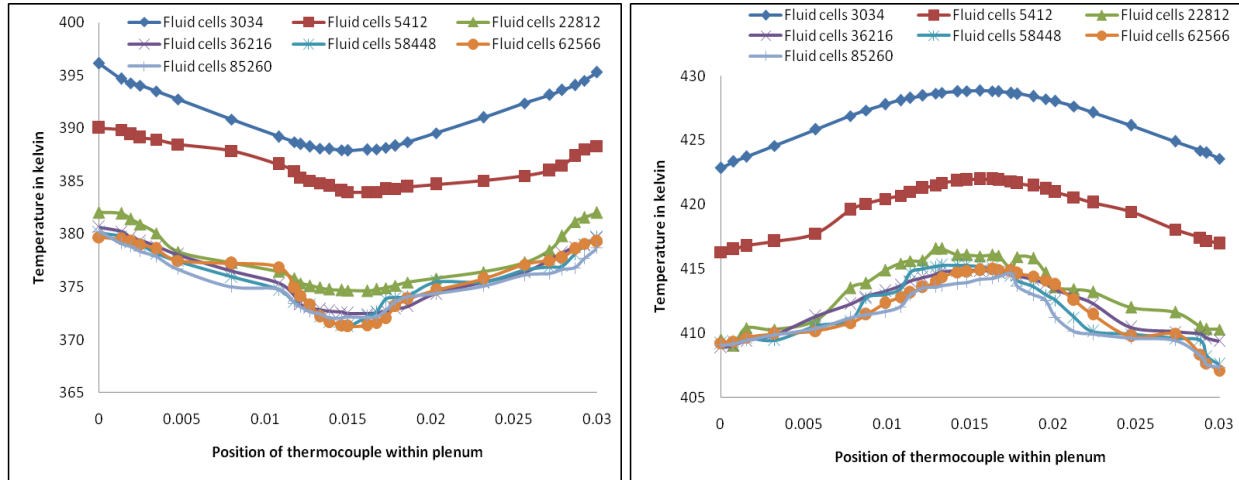


Fig.5.6a: Grid independence examination of the computational solution within inlet and outlet plenum

Table 5.3: Results for mesh sensitivity analysis

Mesh Number	Number of fluid cells	Convective heat transfer coefficient ($\text{W/m}^2 \text{K}$)	% Error
Mesh 1	3034	6821.64	-
Mesh 2	5412	7025.98	2.91
Mesh 3	22812	8869.57	20.79
Mesh 4	36216	9077.37	2.29
Mesh 5	58448	9094.22	0.185
Mesh 6	62566	9108.45	0.156
Mesh 7	85260	9120.12	0.128

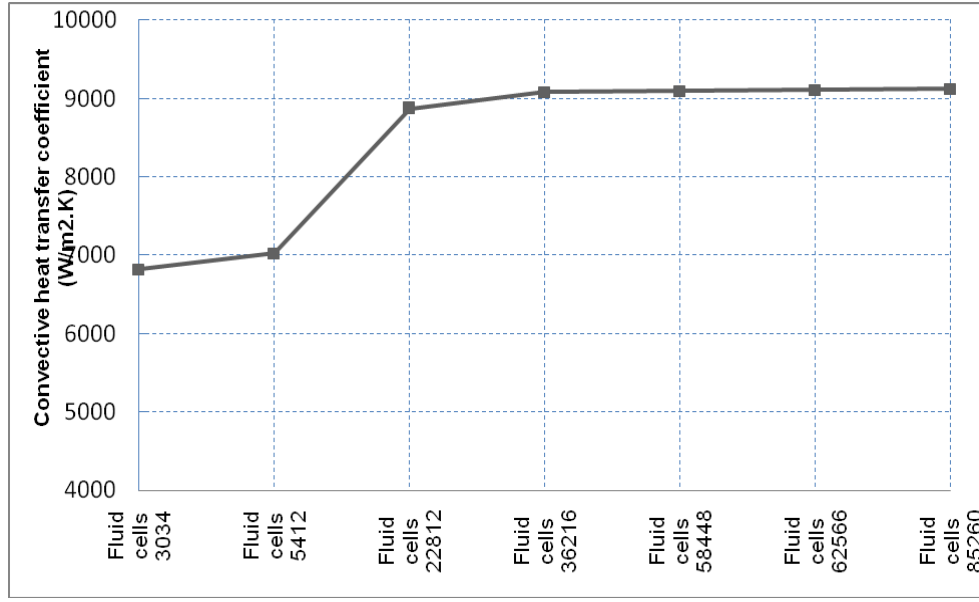
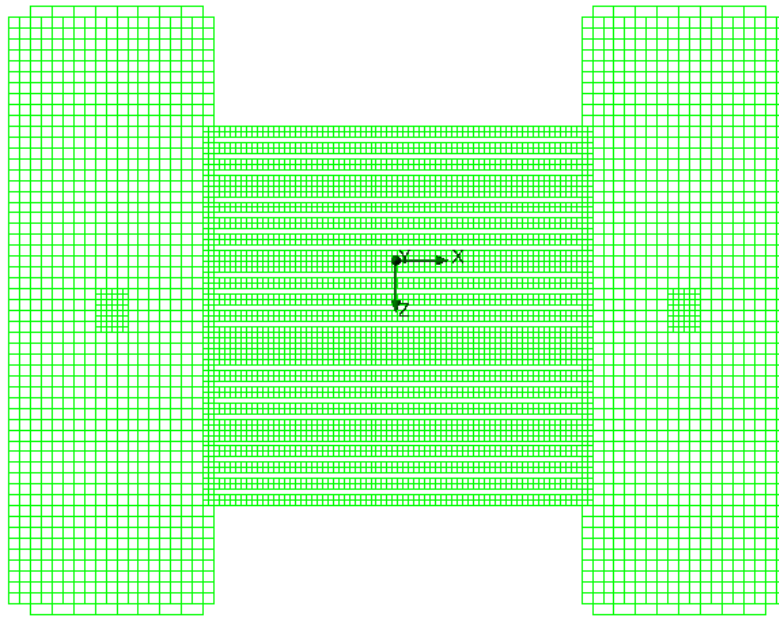


Fig.5.6b: Grid convergence of convective heat transfer coefficient

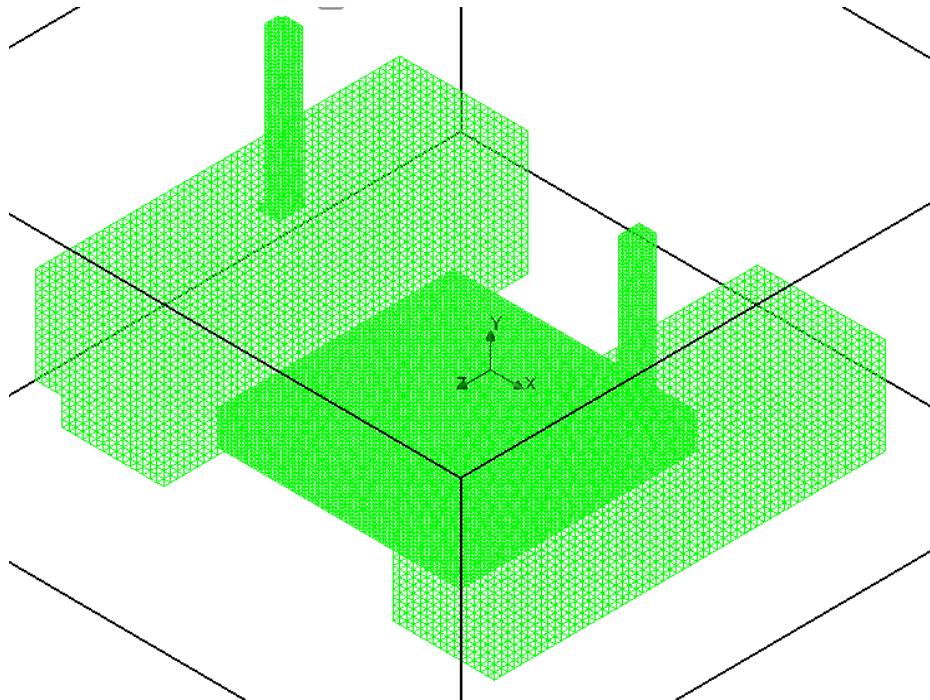
Figure 5.7 shows the views of grid distributions used in the microchannel heat sink. It was ensured that sufficient resolution in mesh generation is maintained especially in the inlet/ outlet plenum and microchannel region for accurate simulations.

5.4.2 Controlling simulations

In order to control the simulations on the present MCHS using SolidWorks Flow Simulation software, average temperature and heat transfer coefficient goals are set. In the value drop down box of the calculation control options in SolidWorks Flow Simulation, it is possible to choose whether calculations will stop when one of the finishing conditions is satisfied or when all of them are satisfied. In the present work, the maximum number of travels (defined as the number of iterations related to the propagation of perturbation) depends on the specific goals that are used in the calculations and resolution level of results.



(a)



(b)

Fig.5.7: Mesh generation in MCHS (a) Top view (b) Isometric view

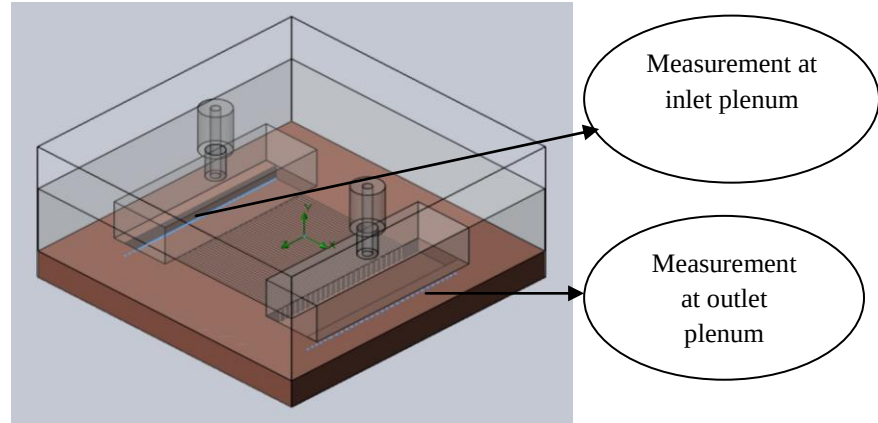


Fig.5.8: Position of temperature probes

The positions of the temperature measurement used in the equation goal(s) are chosen as shown in Figure 5.8. During the initial MCHS simulations, inlet flow velocity of 0.41 m/s, 1.15 m/s and 1.92 m/s are chosen leading to Reynolds number 239, 673 and 1121 respectively. The outlet pressure for all the cases is maintained at 101325 Pa. Similar to the experimental MCHS a surface heat generation source of 125 W, 225 W and 375 W was given to the bottom square slot of 15 mm $\times$ 15 mm. Figure 5.9 shows the simulated locations of boundary conditions of the MCHS.

Due to the variation of temperature within the MCHS, variation of water viscosity (a thermal property) has significant effect on the thermo-hydraulic performance. Therefore, variable temperature-dependent water viscosity based on mean temperature is employed throughout the calculations. In the SolidWorks Flow Simulation software temperature variable properties are selected.

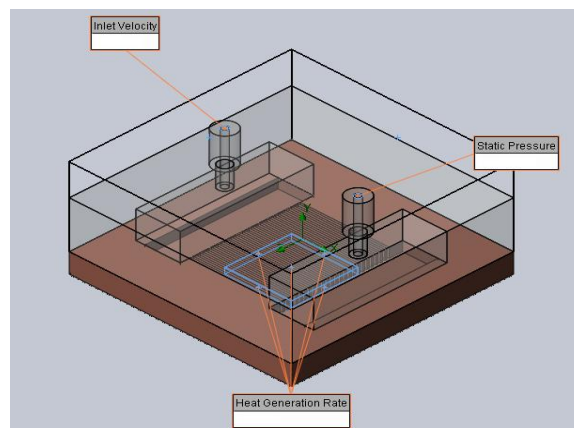


Fig.5.9: CFD Flow simulation boundary conditions on MCHS

5.5 Computational goals

In the MCHS computational goals are criteria that are used to stop the iterative solution process. The goals are chosen from the physical parameters of interest in the Flow Simulation. The use of goals minimizes errors in the calculated parameters of interest and shortens the total solution time for the solver. There are five different types of goals: global goals (GG), point goals (PG), surface goals (SG), volume goals (VG) and equation goals (EG). The global goal is based on parameter values determined everywhere in the flow field whereas a point goal is related to a specific point inside the computational domain. Surface goals are determined on specific surfaces and volume goals are determined within a specific subset of the computational domain as specified by the user. Finally, equation goals are defined by mathematical expressions. In the present case the surface goal of average temperature of fluid inlet and outlet, volume goal of average temperature of solid and equation goal of change in temperature and convective heat transfer coefficient was used as a goal. Using SolidWorks Flow Simulation, in the current analysis, equation goal of change in temperature and convective heat transfer coefficient were used. The software has the provision to calculate the convective heat transfer coefficient using the following equations:

$$\Delta T = \left(VG \, Av \, T_{\text{wall}} - \frac{SG \, Av \, T_{\text{fluid inlet}} + SG \, Av \, T_{\text{fluid outlet}}}{2} \right) \quad (5.9)$$

$$h = \left(\frac{\text{Heat input}}{A \times N \times \Delta T} \right) \quad (5.10)$$

Here ΔT indicates the change in temperature, VG is volume goal, SG is surface goal, Av is average, A is surface area available for heat transfer per channel, N is the total number of microchannels.

5.6 Computational results

The results of SolidWorks Flow Simulation of the selected MCHS can be visualized in a number of ways. In the present CFD simulations, flow trajectories and temperature contours for the MCHS are analysed. Results are discussed for a fixed channel aspect ratio of 4.72 and plenum aspect ratio of 3.75. Initially the flow trajectories are discussed for the three flow arrangements considered in the present research work. These results are analyzed for a given flow arrangement at three different time levels as the flow enters the plenum until it attains steady state conditions.

Apart from the flow trajectories, simulation results for temperature contours obtained over the x-z plane of the MCHS are discussed for the P-, S- and U-type flow arrangements at different heat inputs and Reynolds numbers.

5.6.1 Flow trajectories

Figure 5.10 shows the flow trajectories for the MCHS having inlet and outlet plenum dimensions of $8\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$; channel width of $530\text{ }\mu\text{m}$ and depth of 2.5 mm for all the three flow arrangements. Flow trajectories are important outcome of the simulation, as it helps to understand the physics of the fluid flow phenomena within the microchannel heat sink, when fluid enters and exits with different configurations. Figures 5.10(a) to 5.10(c) show the flow trajectories for P-, S- and U-type arrangements respectively. Under each flow arrangement, three figures are shown to indicate the evolution of flow field from the start of the entry of fluid flow until it attains steady state conditions. P-type of arrangement is generally used in commercial MCHS, because it requires less space. As it is observed in Figure 5.10(a), the water which flows through the inlet adapter comes in contact with the larger surface area of the copper heat sink and then gets transmitted to its left and right region within the plenum before entering the outlet plenum. As the water strikes the inlet plenum initially the water fills the plenum, consequently it fills the microchannels before it fills the outlet plenum. For better clarity the flow field in the inlet and outlet plenums are shown in different colours in the above figure. For the S-type flow arrangement, water comes in contact with the acrylic cover first as shown in Figure 5.10(b) and then flows towards the copper heat sink. In this arrangement the water inlet is located at one of the heat sink and the outlet is located just on the opposite end of the heat sink. However, in the case of U-type flow arrangement shown in Figure 5.10(c), the inlet and exit positions of the flow lie on the same side, ensuring longer flow regimes in the inlet/outlet plenum which attributes to the maximum heat transfer rate as shown by coloured regimes. With regard to the flow arrangements, from Figure 5.10 one can easily notice that U-type flow arrangement ensures prolonged flow regime in the plenums compared to P-type and S-type flow arrangements. In the P-type flow arrangement as the fluid enters the plenum in the perpendicular direction, it strikes the plenum base directly but this does not ensure perfect fluid distribution within the microchannels. When we look at the S-type flow arrangement, due to its position of outlet just opposite to the inlet position, here again the flow distribution is not fully ensured.

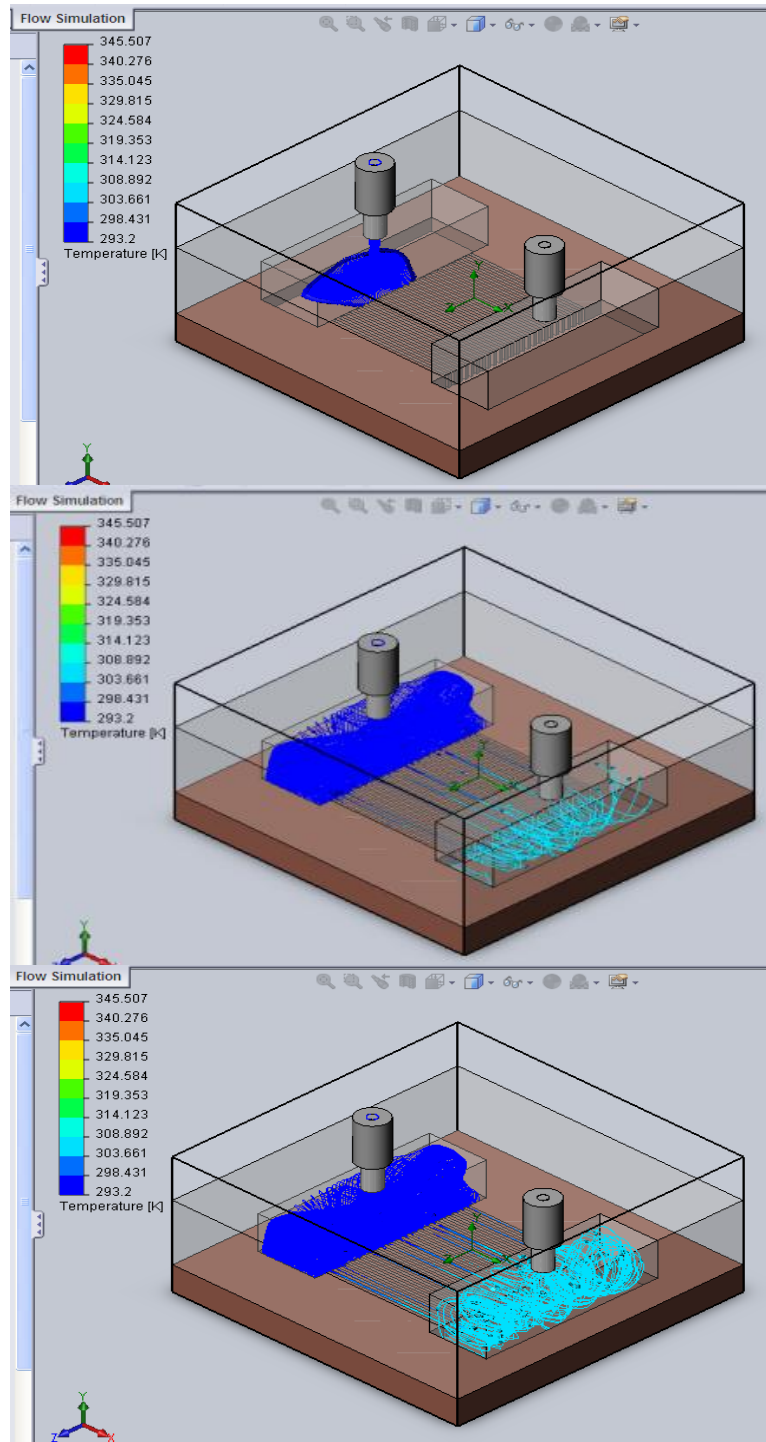


Fig.5.10a: Flow trajectories within MCHS having inlet and outlet plenum dimensions of 8 mm × 30 mm × 2.5 mm for P-type flow arrangement.

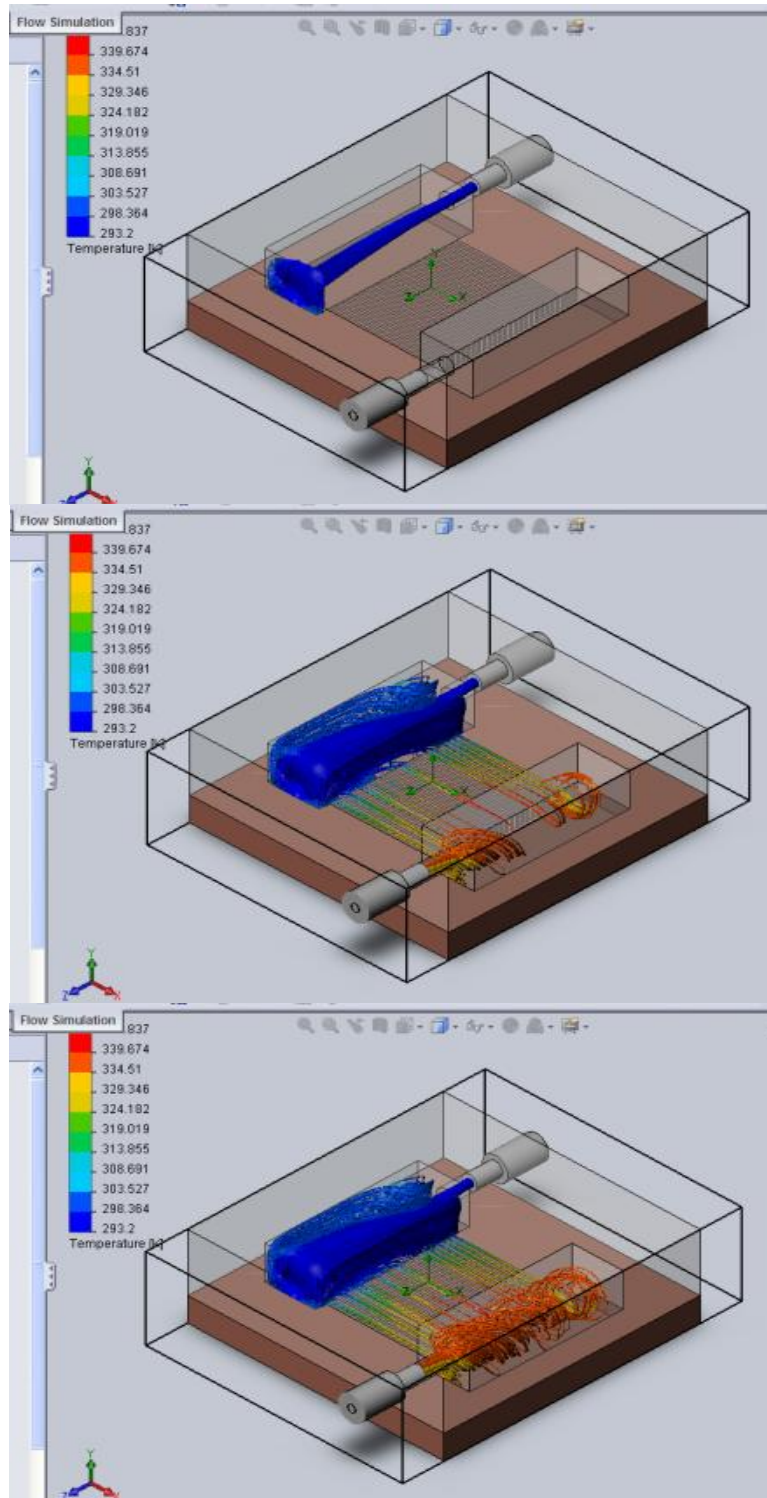


Fig.5.10b: Flow trajectories within MCHS having inlet and outlet plenum dimensions of 8 mm × 30 mm × 2.5 mm for S-type flow arrangement.

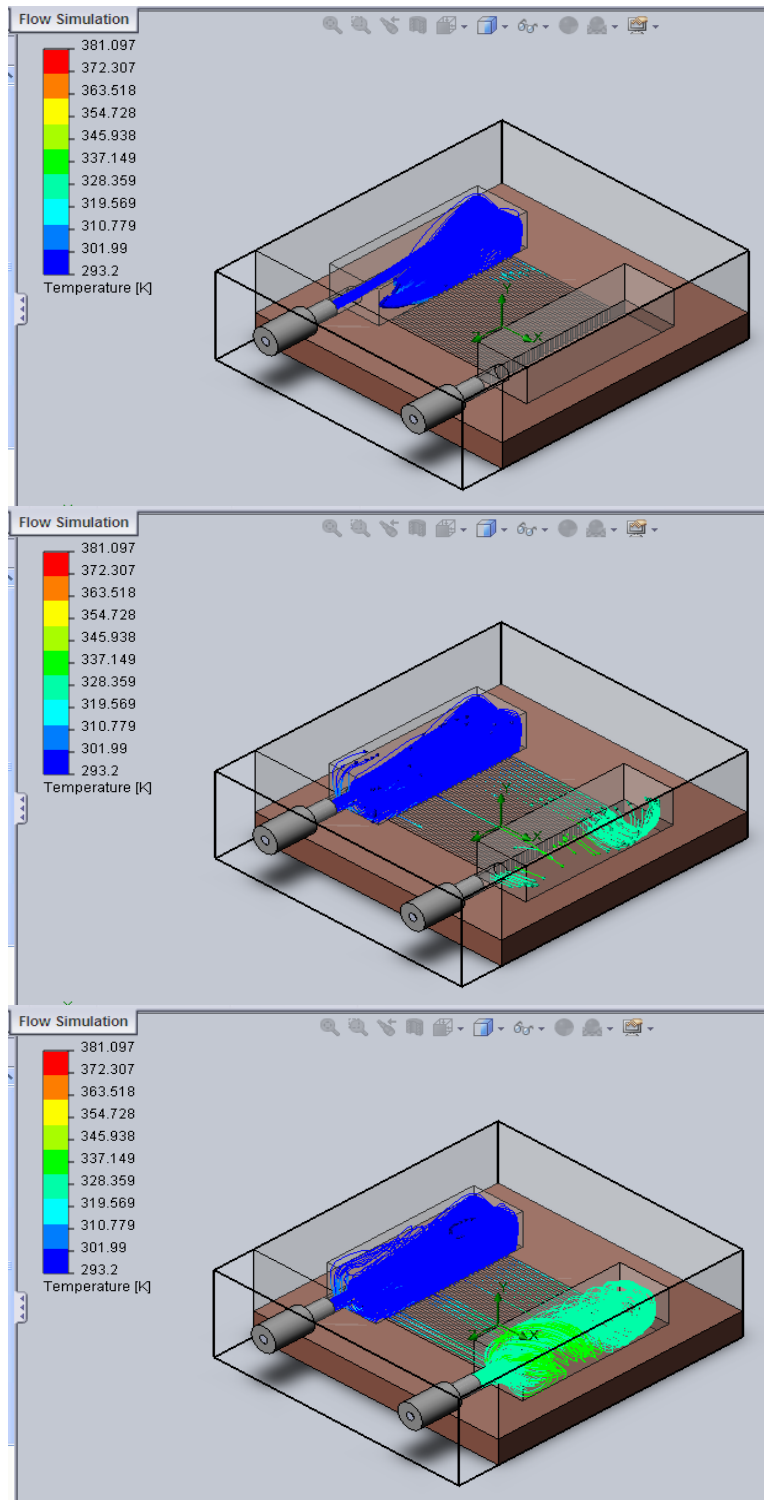
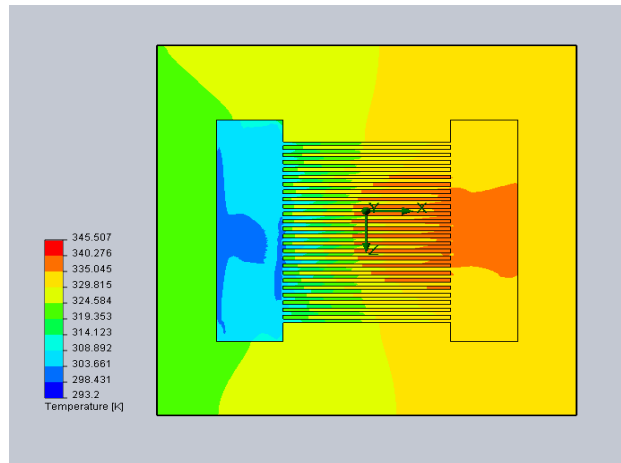


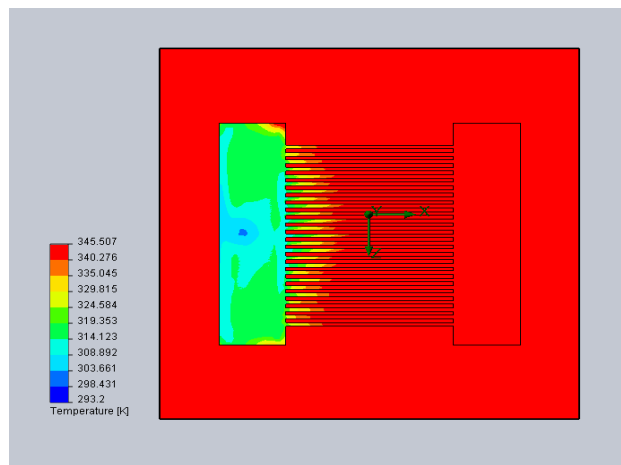
Fig.5.10c: Flow trajectories within MCHS having inlet and outlet plenum dimensions of 8 mm $\times$ 30 mm $\times$ 2.5 mm for U-type flow arrangement.

5.6.2 Temperature variation at different heat input and Reynolds number

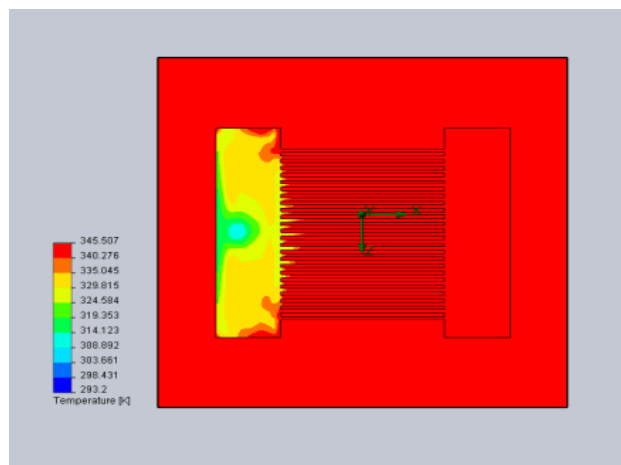
Figures 5.11 to 5.13 show the temperature contour plots of MCHS having inlet and outlet plenum dimensions of $8\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$; channel width of $530\text{ }\mu\text{m}$ and depth of 2.5 mm for P-, S- and U-type flow arrangements respectively. The (a), (b) and (c) sub-divisions of these figures depict the results for Reynolds number, 239, 673 and 1121 respectively for three different heat inputs 125 W, 225 W and 375 W. The red colour indicates the highest value of temperature while the blue colour indicates the lowest temperature. The temperature for any given case in the outlet plenum at any given Reynolds number is always higher than the temperature observed at the inlet plenum. Figure 5.11(a) illustrate the temperature contours for P-type flow arrangement at $\text{Re}=239$ and 125 W heat input. At low heat input there is a slow propagation of heat within the microchannels. In P-type flow arrangement as the flow strikes the inlet plenum, the flow distribution takes some time to become uniform in the inlet plenum and this results in non-uniform distribution of temperature as seen in the above figure. As the heat input increases, the channel fluid also gets heated up as observed in the above figure. One can easily notice that for all the three heat inputs, the temperature non-uniformity is observed at the inlet plenum. When the Reynolds number is increased to 673 (Figure 5.11(b)), the fluid distribution within the microchannel becomes uniform and hence the temperature also becomes uniform as observed in Figure 5.11(b). At this stage, a temperature gradient between the inlet and outlet plenums are being set up for convective heat transfer through the microchannels and this is enhanced with increase in heat input. With further increase in Reynolds number to 1121 (Figure 5.11(c)), there is a distinct temperature gradient being set up between the inlet and outlet plenums for all the three heat inputs, thus ensuring better convective heat transfer across the fluid within the microchannels.



Heat Input=125W

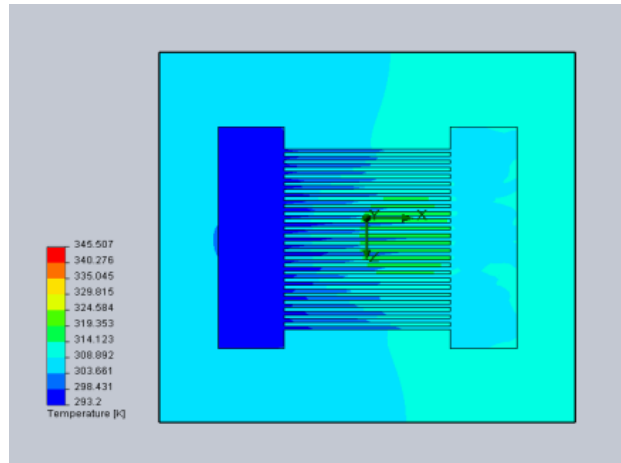


Heat Input=225W

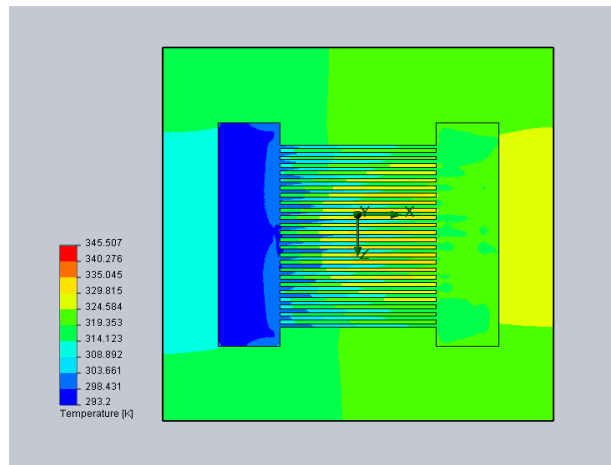


Heat Input=375W

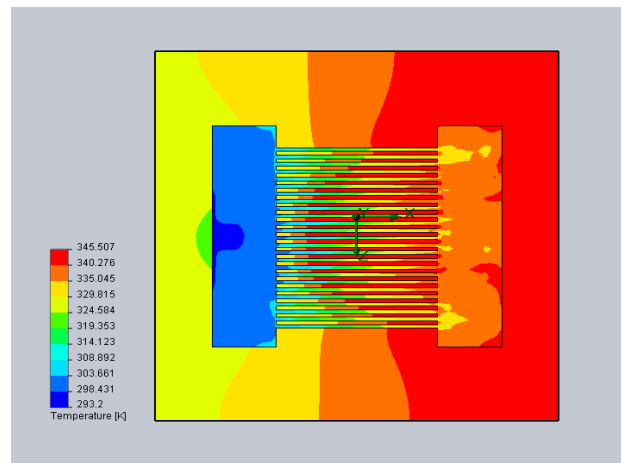
Fig.5.11a: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for P-type MCHS at various heat inputs and Reynolds number=239



Heat Input=125W

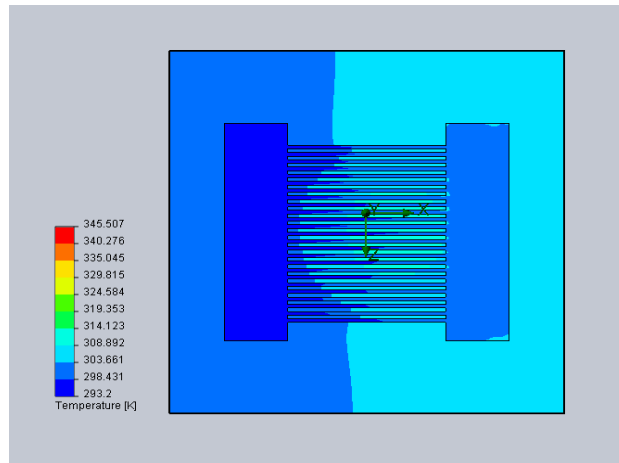


Heat Input=225W

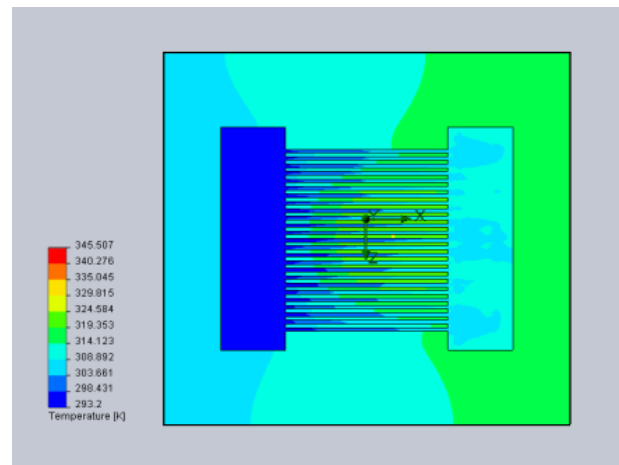


Heat Input=375W

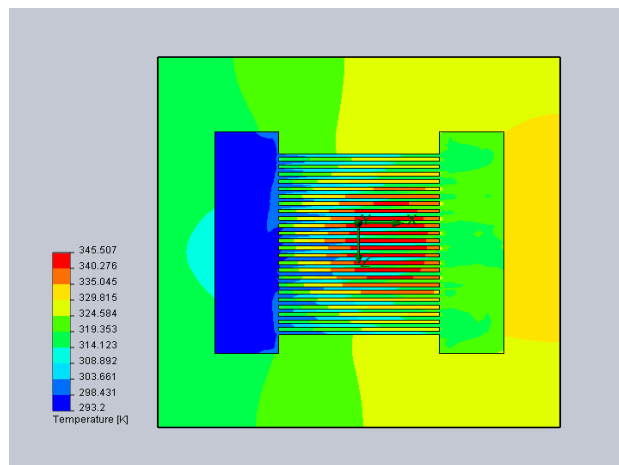
Fig.5.11b: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for P-type MCHS at various heat inputs and Reynolds number=673



Heat Input=125W



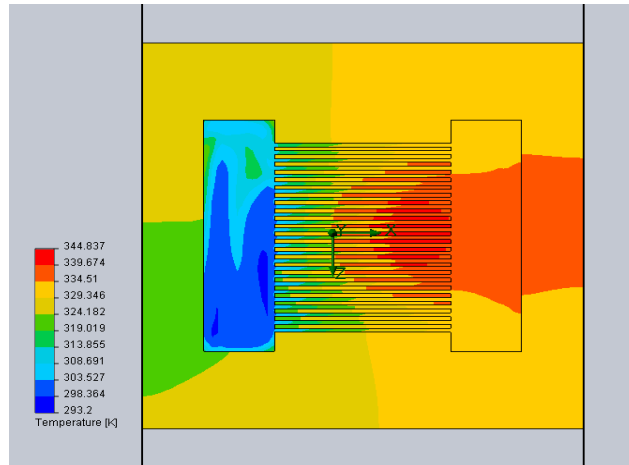
Heat Input=225W



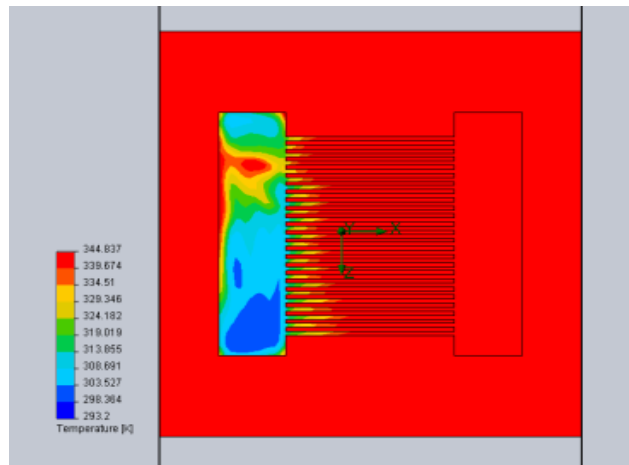
Heat Input=375W

Fig.5.11c: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for P-type MCHS at various heat inputs and Reynolds number=1121

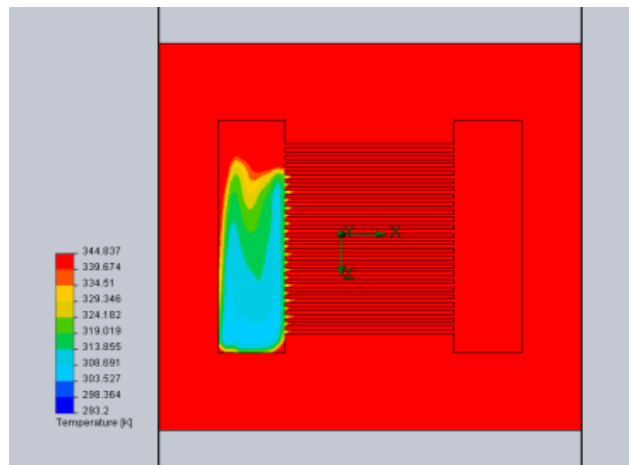
Figure 5.12(a) illustrate the temperature contours for S-type flow arrangement at $Re=239$ for three heat inputs 125 W, 225 W and 375 W. In this flow arrangement the flow regime takes little bit longer time to fill all the microchannels as the fluid has to leave the outlet plenum after passing through all the microchannels. These microchannels offer resistance for the fluid flow which results in slow establishment of flow field within the heat sink. The higher temperature difference between the inlet and outlet plenums at 125 W indicates that the convective heat transfer coefficient will be lower for this case. With increase in heat input, the whole sink tries to attain a maximum temperature, however, maintaining a lower temperature in the inlet plenum, thus demonstrating poor convective heat transfer phenomenon. As the Reynolds number is increased to 673 (Figure 5.12(b)), there is a reduction in the temperature difference between the inlet and outlet plenums, however, at higher heat inputs the temperature difference increases. This shows that the S-type flow arrangement is not able to dissipate the heat from the heat sink to the channel fluid effectively. When the flow Reynolds number is increased to 1121 as depicted in Figure 5.12(c), there is a decrease in the temperature difference indicating an enhanced convective heat transfer compared to the lower Reynolds numbers. One can easily notice the non-uniform temperature distribution in the outlet plenum even at this high Reynolds number. As the flow distribution within the microchannels is not favourable for convective heat transfer, at 375 W heat input, the temperature difference becomes high even at $Re=1121$.



Heat Input=125W

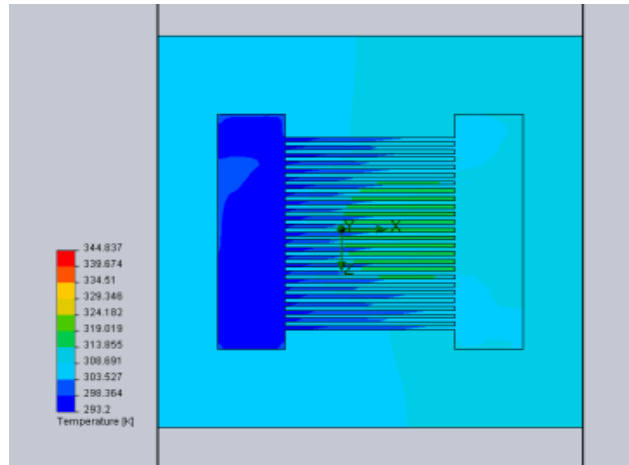


Heat Input=225W

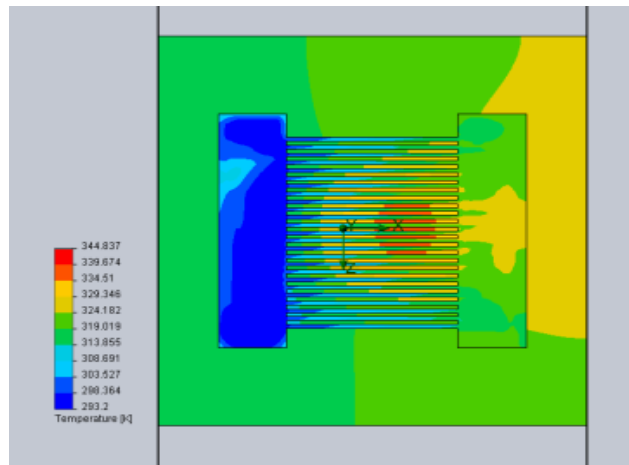


Heat Input=375W

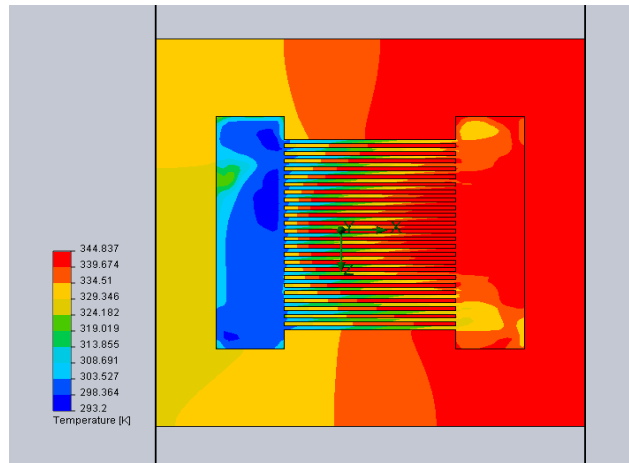
Fig.5.12a: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for S-type MCHS at various heat inputs and Reynolds number=239



Heat Input=125W

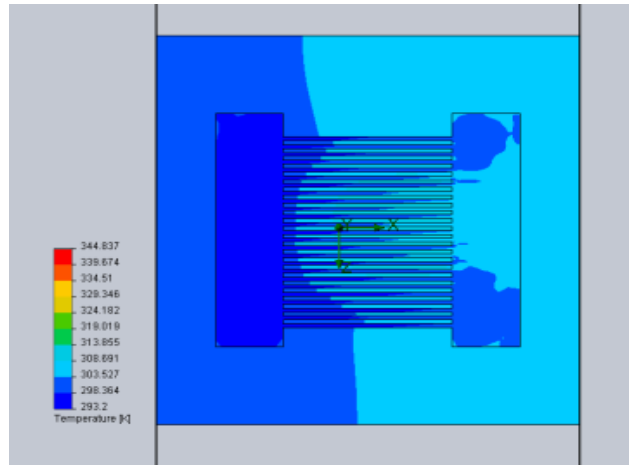


Heat Input=225W

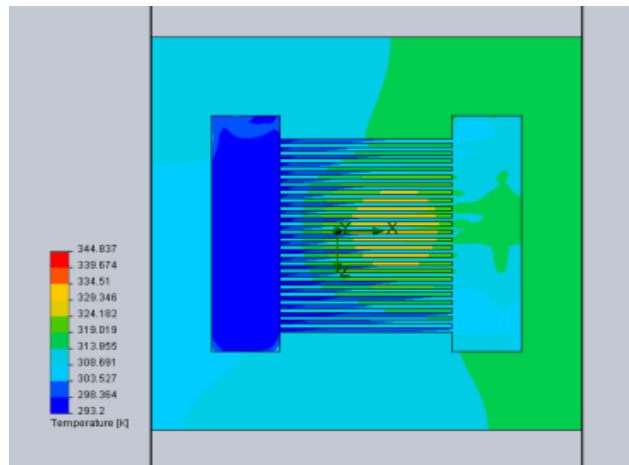


(f) Heat Input=375W, Re=673

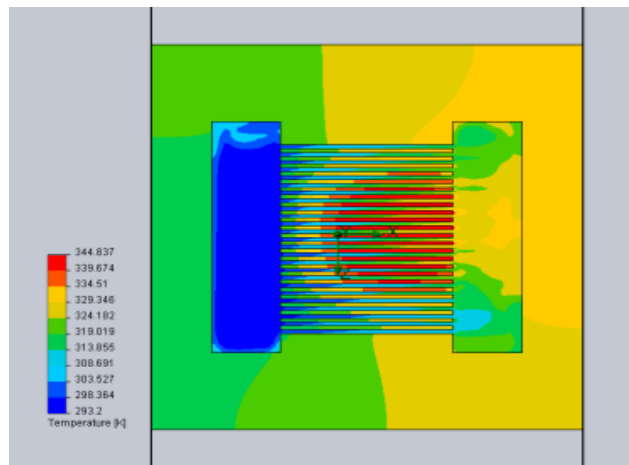
Fig.5.12b: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for S-type MCHS at various heat inputs and Reynolds number=673



Heat Input=125W



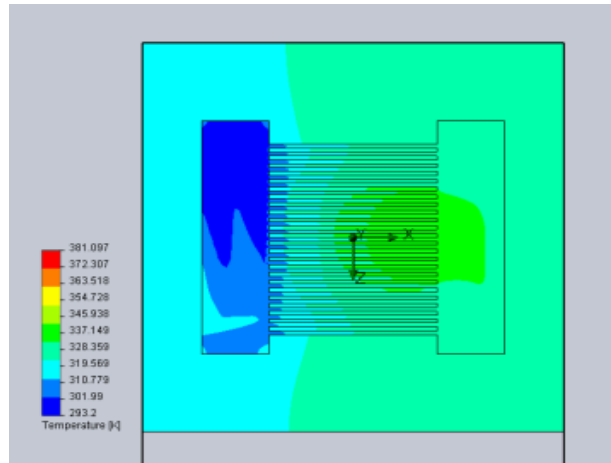
Heat Input=225W



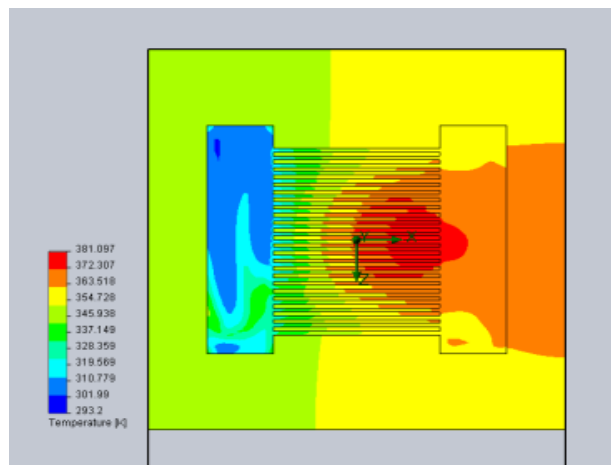
Heat Input=375W

Fig.5.12c: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for S-type MCHS at various heat inputs and Reynolds number=1121

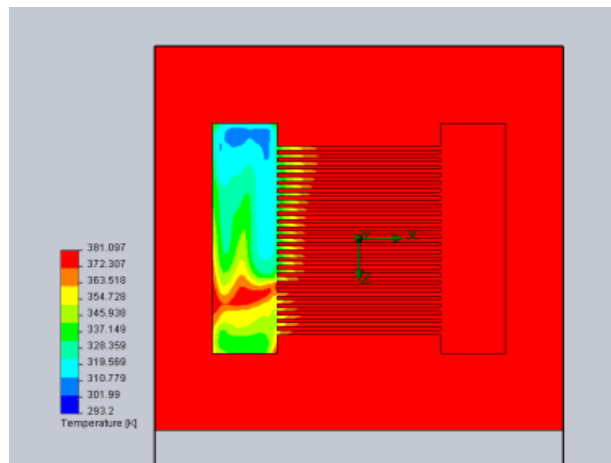
Figure 5.13(a) shows the temperature contours for U-type flow arrangement for $Re=239$ at different heat inputs. In this flow arrangement the fluid inlet and outlet occupy the same side of the heat sink. As the fluid enters the inlet plenum, the fluid also has an opportunity to flow through the microchannels as they are connected to the outlet plenum on the same side of the heat sink. This ensures better fluid distribution within the microchannels as well as in the inlet and outlet plenums. At $Re=239$, the temperature difference between the inlet and outlet plenum is small for 125 W heat input. However, as the heat input increases, this temperature difference also increases as seen in the above figure. With increase in Reynolds number to 673 (Figure 5.13(b)), better flow distribution is ensured in this flow arrangement and hence the heat dissipated by the heat sink is absorbed by the fluid within the microchannel, thus increasing the convective heat transfer within the microchannels. This is indicated by the decrease in temperature difference between the inlet and outlet plenums of the heat sink. It should be noted that even at the highest heat input, 375 W the temperature difference is small in comparison to the difference exhibited by P- and S- types of flow arrangements. With further increase in Reynolds number to 1121, there is a further reduction in the temperature difference as noted in Figure 5.13(c), thus increasing the convective heat transfer within the microchannels.



Heat Input=125W

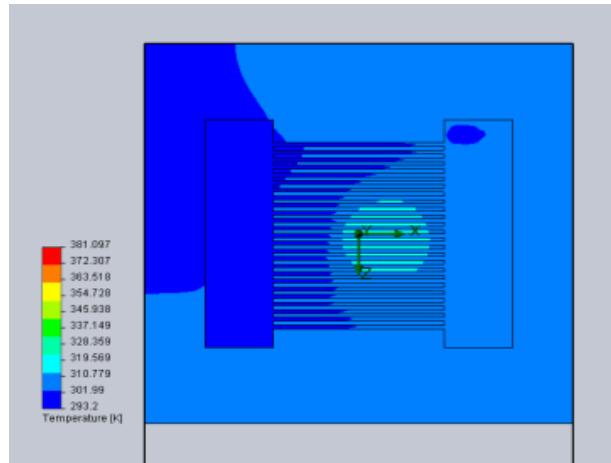


Heat Input=225W

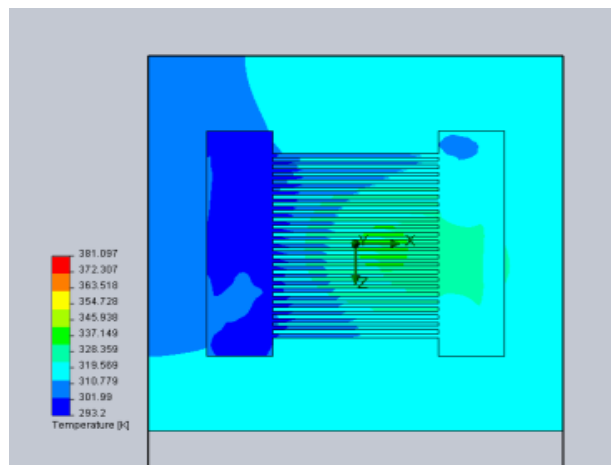


Heat Input=375W

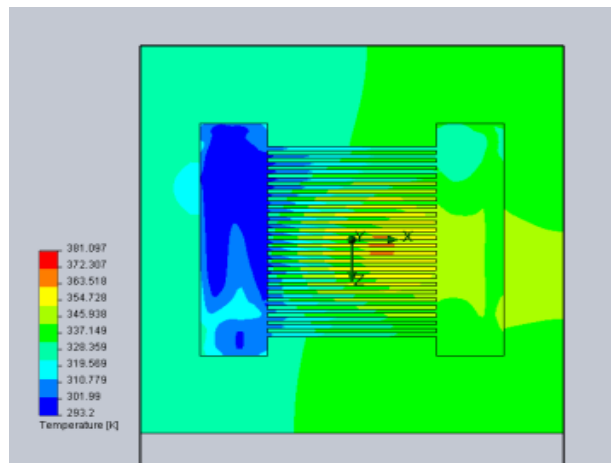
Fig.5.13a: Temperature contours (along inlet, outlet plenum of $8 \text{ mm} \times 30 \text{ mm} \times 2.5 \text{ mm}$ and array of micro channels) for U-type MCHS at various heat inputs and Reynolds number=239



Heat Input=125W

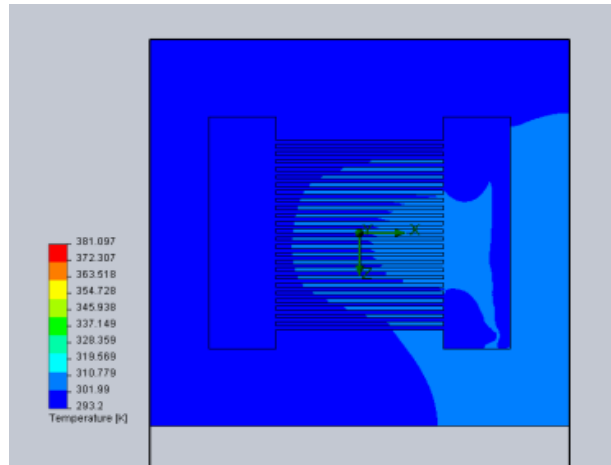


Heat Input=225W

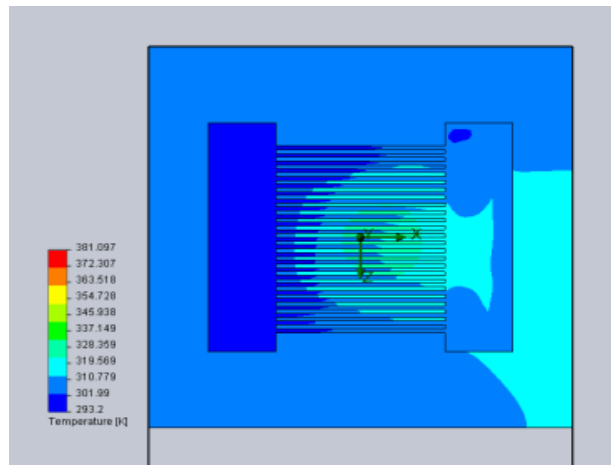


Heat Input=375W

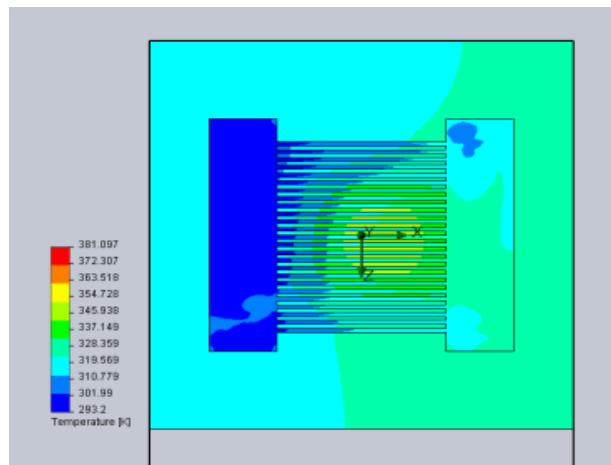
Fig.5.13b: Temperature contours (along inlet, outlet plenum of 8 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for U-type MCHS at various heat inputs and Reynolds number=673



Heat Input=125W



Heat Input=225W



Heat Input=375W

Fig.5.13c: Temperature contours (along inlet, outlet plenum of $8\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$ and array of micro channels) for U-type MCHS at various heat inputs and Reynolds number=1121

As the Reynolds number increases, this temperature rise decreases, and this can be attributed to the increase in convective heat transfer, which is inversely proportional to the temperature difference for a given constant heat input. It can easily be noticed that the S-type flow arrangement always exhibits a higher temperature at the outlet plenum compared to the U-type and P-type flow arrangements. The reason attributed to this is that in the S-type flow arrangement, the fluid initially starts filling the inlet plenum by striking the outermost part of the inlet plenum and then starts filling the micro-channels and later on leave from the outlet plenum. But in the case of the U-type and, to some extent, the P-type flow arrangement, the fluid starts filling the inlet plenum from the starting point of the plenum and fills the entire plenum in the due course of time. It should be noted that once the fluid starts filling the inlet plenum, the fluid starts flowing into the micro-channels, thus enhancing the effective fluid contact with the heated bottom wall of the MCHS.

Similar trends were observed for other MCHSs of inlet and outlet plenum size of $10\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$ and $12\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$ but constant channel width of $530\text{ }\mu\text{m}$ and depth of 2.5 mm as presented in Appendix C (Figures C2 to C7).

5.7 Variation of convective heat transfer coefficient

The computational convective heat transfer coefficient was calculated as described in Section 5.5. Figure 5.14 shows the comparison of experimental and computational heat transfer coefficient values under different flow arrangements and Reynolds number. In Figure 5.14, constant heat input of 225 W is selected for plenum size $10\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$ and channel width of $530\text{ }\mu\text{m}$. The value for the computational convective heat transfer coefficient for P-type flow arrangement was observed to be 3.94 to 4.50% higher than the experimental value, whereas for U-type flow arrangement, it was 3.14 to 8.14%. A minimum rise (1.94 to 4.62%) was observed for S-type flow arrangement among the three flow arrangements. Figures 5.15 and 5.16 show the variation of experimental and computational results obtained for convective heat transfer coefficient with Reynolds number. The heat transfer coefficient in the Flow simulation is computed from the mathematical equation goal for the selected MCHS under various flow arrangements for different heat inputs and Reynolds numbers. Similar to the trends as experienced in the experimental results, the heat transfer coefficient obtained from the simulation work also varies in both the MCHSs when the inlet and outlet plenum size is constant of 10 mm

$\times 30 \text{ mm} \times 2.5 \text{ mm}$ size and channel width is $330 \text{ }\mu\text{m}$ and $530 \text{ }\mu\text{m}$. U-type flow arrangement always has a higher average heat transfer coefficient compared to the P-type flow arrangement, followed by the S-type. By comparing these figures it can be seen, that the computationally evaluated heat transfer coefficient values are numerically higher by 2 to 19% as compared to the experimental values for the entire range of flow arrangements and heat inputs. The reason for variation between computational and experimental values is attributed to the assumptions of steady state flow condition, incompressible fluid, no slip condition and no heat loss to the surrounding as employed in the flow simulations while analyzing MCHS.

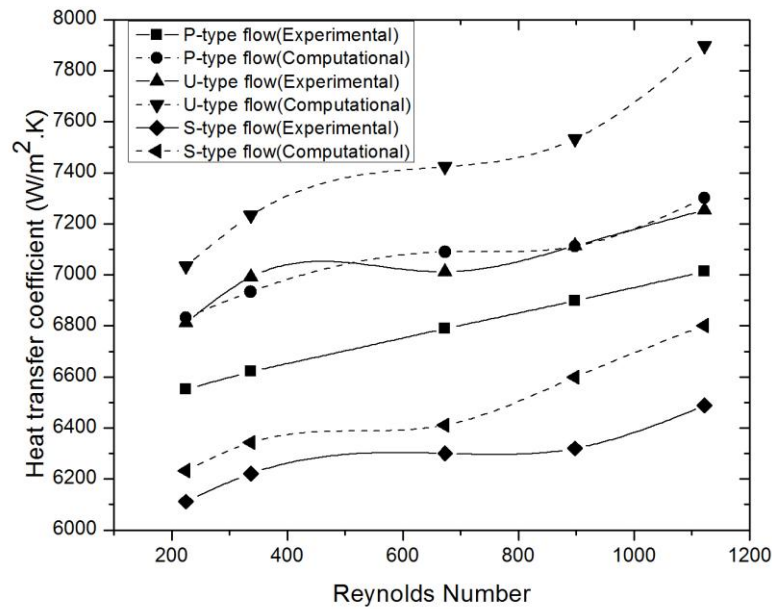
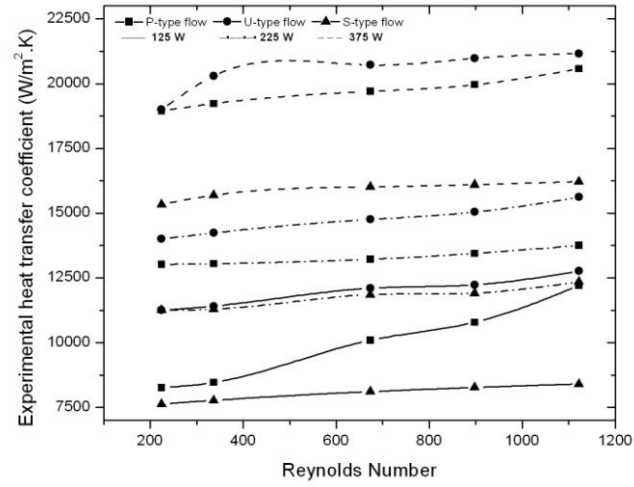
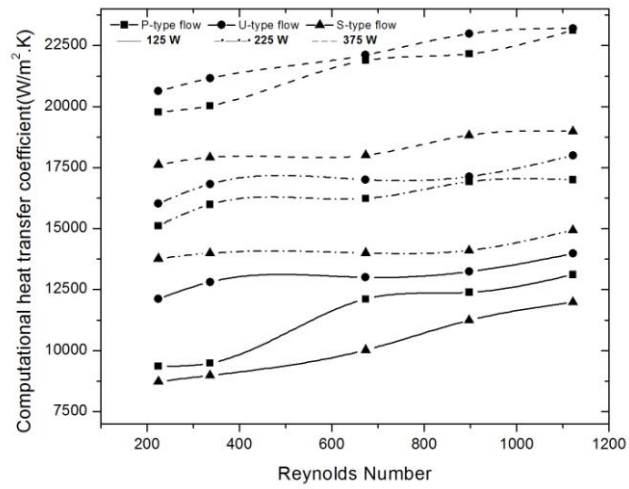


Fig.5.14: Comparison of experimental and computational heat transfer coefficient for various flow arrangements (Heat input: 225W; plenum size: $10 \text{ mm} \times 30 \text{ mm} \times 2.5 \text{ mm}$, channel width $530 \mu\text{m}$)

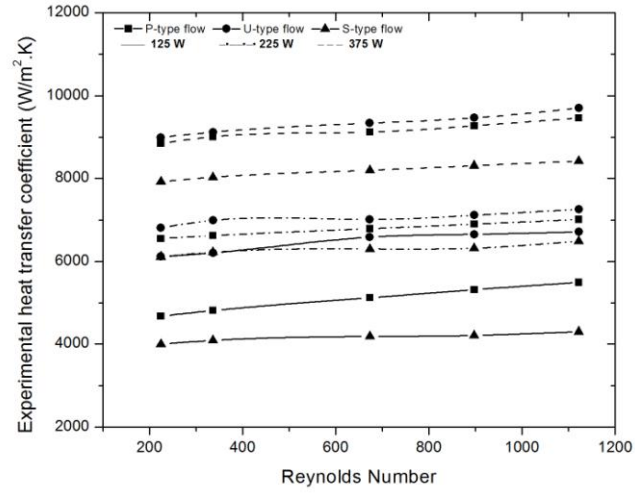


(a)

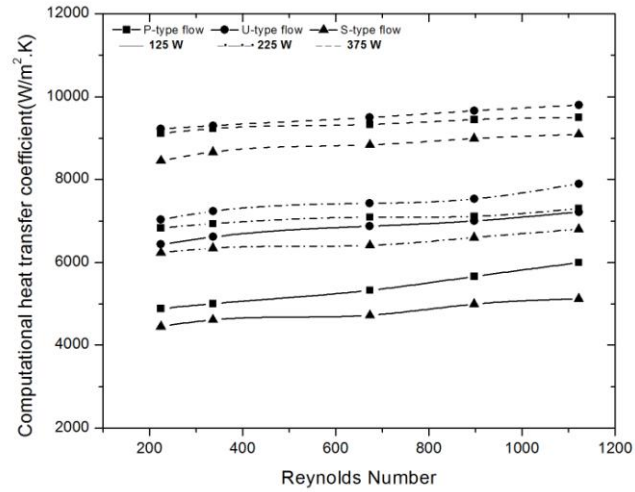


(b)

Fig.5.15: Effect of Reynolds number on (a) experimental (b) computational heat transfer coefficient for various flow arrangements (plenum size: 10mm x 30mm x 2.5mm, channel width 330 μ m)



(a)



(b)

Fig.5.16: Effect of Reynolds number on (a) experimental (b) computational heat transfer coefficient for various flow arrangements (plenum size: 10mm x 30mm x 2.5mm, channel width 530μm)

CHAPTER 6

SUMMARY, CONCLUSIONS AND SCOPE FOR FUTURE WORK

6.1 Summary

The rapid progress in the applications of Integrated Circuits (ICs) has lead to extremely small size of these circuits with high demand on increased operating speeds and package densities. These requirements are associated with a drastic rise in die temperatures which is detrimental to circuit performance and reliability. Liquid cooled microchannel heat sinks (MCHS) are considered as one of the thermal management solutions, as they provide large heat transfer surface area per unit volume of fluid flow. The present work considers not only the heat transfer and fluid flow distribution within the microchannels, but the whole MCHS as an integral unit is analysed which include microchannels and inlet and outlet plenums. The underlying reason is that in practical applications for cooling of processors, ICs, etc., the overall performance of the whole MCHS unit is important. Also depending upon the practical requirements for a given application, the flow arrangements through the plenums of the MCHS critically influences the overall performance of MCHS unit.

The work reported in this thesis may be summarized in two parts: experimental and computational. In both the experimental and computational works, investigations have been carried out to examine the effect of channel and plenum aspect ratios at different flow arrangements on convective heat transfer and pressure drop characteristics of MCHS for various Reynolds numbers at different heat inputs. For this purpose three flow arrangements, P-, U- and S-types have been studied with three levels of heat input. The computational fluid dynamics analysis has also been carried out to understand the physics behind fluid flow and heat transfer within MCHS unit for various flow arrangements, and plenum and channel dimensions. A comparison was also made between the experimental and computational thermo-hydraulic performance parameter of the system under similar conditions.

6.2 Conclusions

The data obtained from detailed experimental and computational studies have been used to compute Nusselt number and heat transfer coefficient, the parameters commonly employed for

evaluating the thermo-hydraulic performance of MCHSs as a function of Reynolds number. The CFD flow simulations were performed in order to understand the physics of the problem. The flow trajectories and temperature contours in the studied MCHS gave an insight into the fluid dynamics and heat transfer characteristics.

6.2.1 Important conclusions from experimental results

Based on the analysis of the experimental results the following general conclusions can be drawn. These are valid for the range of experimental conditions used.

- ❖ A higher Reynolds number is beneficial at reducing both the fluid outlet temperatures and the temperatures within the heat sink but at the expense of greater pressure drop.
- ❖ A good agreement between the present experimental results with the previous work for both pressure drop and temperature data proves that the experimental and numerical simulation methodology adopted in the present work is validated.

Effect of flow arrangement

- ❖ The performance of MCHS is significantly affected by the type of flow arrangement used. Also, for a constant geometry the overall thermal performance of a MCHS varies with varying inlet and outlet flow arrangements.
- ❖ The maximum temperature difference occurs in the S-type, followed by the P- and the U-type flow arrangements. This implies that for a given constant heat input, the average Nusselt number becomes higher for the U-type followed by the P-type and S-type flow arrangements.
- ❖ Minimum friction factor is experienced in the P-type flow arrangement followed by that in the U-type and S-type flow arrangement. This is probably responsible for the popularity of the P-type flow arrangement, in liquid cooled MCHS for practical applications in industry.
- ❖ For any given channel aspect ratio, plenum aspect ratio and heat input the Nusselt number for U-type flow arrangement is higher than the P-type and followed by the S-type flow arrangement.
- ❖ For a given geometry of MCHS a correlation exists for the Nusselt number, Reynolds number and flow arrangements.

Effect of channel aspect ratio

- ❖ For a constant plenum aspect ratio, an increase in channel aspect ratio results in maximum total pressure drop for S-type flow arrangement, followed by U-type and P-type.
- ❖ The Nusselt number increases with increase in Reynolds number. The rate of increase is not significantly affected by the variation in channel aspect ratio, plenum aspect ratio and flow arrangement. However a small variation was noted for $AR_c=7.57$ at 125 W for the Reynolds number range 350 and 450.
- ❖ The variation in Nusselt number with Reynolds number increases with the increase in channel aspect ratio from 4.72 to 7.57 and heat input from 125 W to 375 W.
- ❖ The Nusselt number increases when the channel aspect ratio is increased. This is attributed to increase in fluid velocity which gives rise to effective convective heat transfer.

Effect of plenum aspect ratio

- ❖ With the increase in the plenum aspect ratio, the Nusselt number decreases for all the Reynolds numbers. The maximum Nusselt number occurs at $AR_p=2.5$ for U-type flow arrangement followed by P-type and S-type arrangements, and the minimum Nusselt number occurs at $AR_p=3.75$ for S-type flow arrangement.
- ❖ For a given flow arrangement, the Nusselt number increases when the plenum aspect ratio is reduced.
- ❖ For a given range of Reynolds number, with increase in plenum aspect ratio there is decrease in the pressure drop. The minimum pressure drop is for P-type flow arrangement followed by U-type and S-type flow arrangements.
- ❖ At a constant channel aspect ratio and varying plenum aspect ratio, the total pressure drop in the MCHS is the maximum for S-type flow arrangement followed by U-type and P-type.

Effect on pressure drop

- ❖ A gradual slope change in the pressure drop variation with Reynolds number. This is attributed to the temperature dependence of fluid viscosity and the increasing contraction and expansion pressure losses at the inlet and outlet plenum, respectively.

- ❖ With the increase in Reynolds number, the microchannel heat sink experiences higher pressure drop, thus resulting in more pumping power consumption.
- ❖ As the heat input increases, the pressure drop decreases for all types of flow arrangements.
- ❖ As the channel aspect ratio is increased, the fluid velocity through the microchannels and the wall shear stress within the microchannels increase, resulting in higher pressure drop.
- ❖ The maximum pressure drop occurs at the higher channel aspect ratio, $AR_c=7.57$ and the lowest plenum aspect ratio, $AR_p=2.5$ for S-type flow arrangement, whereas the minimum pressure drop is noted for the case of P-type flow arrangement for $AR_p=3.75$.
- ❖ The pressure drop is higher for the lower value of plenum aspect ratio. It is observed that the fluid flow travel length at the inlet/outlet plenum of the heat sink decreases with increase in plenum aspect ratio. Hence when the flow length is larger for the case of $AR_p=2.5$, the pressure drop is found to be maximum as compared to $AR_p=3.75$ where the flow length is small.

6.2.2 Important conclusions from computational results

The results of the CFD analysis of the complete MCHS unit carried out for various heat inputs, flow arrangements, plenum sizes and microchannel widths were found to be in line with the results of the experimental analysis. The following is a summary of some of the important conclusions drawn from the CFD study.

Effect of flow arrangement

It is possible to analyze the fluid flow through various regions of the MCHS using flow trajectories. In the P-type flow arrangement as the fluid enters the plenum in the normal direction, it first strikes the plenum base and gets non-uniformly distributed among the various microchannels. In the S-type flow arrangement also the flow distribution after striking the plenum regions is non-uniform. This can be attributed to the position of fluid outlet just opposite to the inlet. However, in the case of U-type flow arrangement as the inlet and exit positions of the flow lie on the same side, flow regimes in the inlet/outlet plenum are longer which attributes to higher Nusselt number.

Effect of channel aspect ratio

In the studied MCHS, for a constant plenum aspect ratio, as the channel aspect ratio increases heat transfer coefficient also increases. The computational values of heat transfer coefficient are slightly higher as compared to the experimental ones. The variation can be attributed to the assumptions made in the flow simulations and experimental uncertainties observed during measurements.

Effect of plenum aspect ratio

The maximum heat transfer (as seen from temperature contours) occurs for the lowest plenum aspect ratio (maximum plenum length). The temperature contours also indicate that at constant channel aspect ratio and heat input, as the Reynolds number increases the MCHS experiences higher temperature variations between inlet and outlet plenums.

6.3 Scope for future work

Research in the area of microchannels is receiving increasing attention from both the academic as well as the industrial society. During the last decade, the number of published papers on microfluidic topics has approximately doubled each year. Initially, microfluidics research was focusing on analytical applications and fundamental understanding of fluid flow in microchannels. The liquid cooling of integrated circuits, especially computer processor, has been a prominent research topic but recently these systems were implemented in practical use by companies like Thermaltake, Koolance, Swiftech, etc. Liquid cooled microchannel heat sink has distinct advantage as a thermal management solution but the initial cost, and space considerations, size optimization and closed loop system problems need to be further investigated. In short, more research work may be undertaken in the following directions.

- ❖ The present work of performance evaluation of MCHS can be extended for different materials of the heat sink such as aluminium, brass, bronze etc.
- ❖ It was observed that as the surface roughness in microchannels is increased there is more deviation of the pressure drop from that estimated by conventional laminar theory along with its effect on heat transfer characteristics. To further the fundamental understanding of surface roughness effects in microchannel heat sinks, heat transfer characteristics in the hydraulically and thermally (and simultaneously) developing regions should be

carefully investigated under various flow arrangements. A procedure for incorporating the roughness parameter into studies of microscale heat and mass transfer need to be formulated for the complete MCHS analysis.

- ❖ The effect of different types of fluids like ethylene glycol, Perfluorocarbon (PFC), Hydrofluoroether (HFE) and introduction of nano particles in different types of fluids on the performance of the MCHS can be studied.
- ❖ The present work showed that while using single phase flow, the inlet and outlet flow arrangement has an effect on the thermo-hydraulic performance of MCHS. The flow dynamics inside the small size channel alter when multiphase flow boiling heat transfer occurs with bubble confinement in reduced scale. It was observed that different boiling mechanism based on elongated bubble existence and evaporation of thin liquid layer exists in multiphase flow within microchannels. Much work is still required to produce new insights for a better mechanistic modelling of boiling heat transfer within MCHS under different inlet and outlet manifold conditions. Area worthy of new investigations have been identified and recommended, such as identification of micro-to-macroscale transition, critical heat flux, visualization, flow pattern transition, flow instability, two-phase or multiphase pressure drop, etc.
- ❖ CFD analysis of the liquid cooled MCHS within an operational computer environment under dynamic load conditions can be studied.
- ❖ Optimization of plenum and channel aspect ratio for the wide range of Reynolds number for the complete MCHS unit can be studied experimentally and computationally.

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APPENDIX A

UNCERTAINTY ANALYSIS

In the experimental data, the true values of measured quantities are always unknown. The estimation of the error in these data is called an uncertainty, 'u' usually, the estimated uncertainty, 'u_x', in a given measurement of a physical quantity, x, is made at a 95% confidence level. The statistical suggestions to estimate the uncertainty in single- and multi-sample experiments was described by Kline and McClintock (1953)

The uncertainty of the experimental data was estimated by performing an uncertainty analysis based on the method of sequential perturbations. This method provides a process to estimate an overall uncertainty of a set of data by integrating the uncertainty of each source of error into the data base independently and then using a root sum square method to calculate the overall uncertainty.

Measurement systems and error sources

Measurement systems consist of the instrumentation, the procedures for data acquisition and reduction, and the operational environment. Measurements are made of individual variables, x_i, to obtain a result, X which is calculated by combining the data for various individual variables through data reduction equations

$$X = f(x_1, \dots, x_i, \dots, x_n) \quad (A.1)$$

Each of the measurement systems used to measure the value of an individual variable, x_i, is influenced by various elemental error sources. The effects of these elemental errors are shown as bias errors, Wx_i and precision errors, Sx_i, in the measured values of the variable, x_i. These errors in the measured values then propagate through the data reduction equation, thereby generating the bias, W_X, and precision, S_X, errors in the experimental result, X.

The root-sum-square (RSS), combination is the basic form used for combining uncertainty contributions in both single-sample and multiple-sample analyses of the individual uncertainties.

Bias limits

The bias limit, W. The bias limit is an estimate of the magnitude of the systematic inaccuracy (i.e. fixed and constant error). It is assigned with the understanding that the experimenter is 95%

confident that the true value lies within the limits of the bias error. We can adjust measurements to account for bias errors.

If we assume that all x_i are independent variables, then the bias limit of the result is given by the following expression:

$$W_X^2 = \sum_{i=1}^m \left(\left(\frac{\partial f}{\partial x_i} \right)^2 W_{x_i}^2 \right) \quad (\text{A.2})$$

where $\left(\frac{\partial f}{\partial x_i} \right)$ are the sensitivity coefficients, and W_{x_i} the bias limits in x_i

The bias limit W_{x_i} for each variable is an estimate of elemental bias errors from different types: calibration errors; data acquisition errors; data reduction errors; and theoretical bias. Within each type, there may be several elemental sources of bias. For instance, if for the i_{th} variable x_i there are n elemental bias errors identified as major, and whose bias limits are estimated as $(W_{x_i})_1, (W_{x_i})_2, \dots, (W_{x_i})_n$ then the bias limit for the measurement of x_i is calculated as the RSS combination of the elemental limits.

$$W_{x_i}^2 = \sum_{k=1}^n (W_{x_i})_k^2 \quad (\text{A.3})$$

The bias limits for each element $(W_{x_i})_k$ must be estimated for each variable x_i using the best information available. Bias limits contribution are identified (calibration accuracy, data acquisition, data reduction, or conceptual bias) and combined.

Precision limits (i.e., Random Uncertainties)

To determine the random errors, the measured data must be obtained from a process in a statistically controlled condition. Limits of random errors are stated by means of an estimate of the standard deviation of the mean value ($\bar{x}$) of a series of measurements. In this, the number of measurements (n) is sufficiently large (>20).

The standard deviation (S), which can be calculated from n measured values x_i ($i=1, 2, \dots, n$) as:

$$\left[S_{\bar{x}} = \frac{1}{n \cdot (n-1)} \sum_{i=1}^n (x_i - \bar{x})^2 \right]^{1/2} \quad (\text{A.4})$$

where, the mean value is given by $\bar{x} = \frac{1}{n} \sum_{i=1}^n x_i$

In case the measured data is performed only once, the standard deviation has to be determined from experience, or if the bias error interval in the measured data is known, then the standard deviation can be calculated by:

$$S_{x_i} = \left(\frac{W_{x_i}}{3} \right)^{1/2} \quad (\text{A.5})$$

The $\pm S$ interval about a nominal value is the 95% confidence range within which the mean of many such results would fall, if the experiment were repeated many times under the same conditions using the same equipment. Thus, the limit on the precision of a measurement is an estimate of the non-repeatable inaccuracy caused by an unknown or an uncontrollable influence.

S_X can be estimated by the RSS of the precision limits for the measurements of the individual variables.

$$S_X = \sqrt{\sum_{i=1}^n \left(\left(\frac{\partial f}{\partial x_i} \right)^2 S_{x_i}^2 \right)} \quad (\text{A.6})$$

The value of S_X is determined from n readings over an appropriate and sufficient time interval that includes all factors causing variability in the result.

Overall combined uncertainties

The 95% (20:1) confidence overall combined uncertainty (i.e., the total uncertainty in the result, X ,) is calculated from the RSS of the bias and precision limits.

$$U_X = \sqrt{S_X^2 + W_X^2} \quad (\text{A.7})$$

The $\pm U_X$ interval about the nominal result is the range within which the experimenter is 95% confident that the true value of the result lies.

Computing uncertainty of measurement

Two methods are available to calculate the measurement data uncertainties. Analytical method, and sequential perturbation, both methods are based on the errors RSS estimation.

The analytical method requires deriving a single formula for the measurement uncertainty, which is easy to compute, but the derivations become more difficult as the data reduction procedures become more and more complex.

Applying the sequential perturbation procedure is easy to when the data reduction procedure is conducted using a computer program. The sequential perturbation method gives an approximate of the uncertainty estimation of the measured data, not exact as in the analytical method, but it is simple to apply, and gives the ability to identify the component that consist a major uncertainty in the model of the data reduction.

In this thesis analytical method has been used for finding uncertainty.

Suppose X is function of several independent variables, $X = f(x_1, x_2, x_3, \dots, x_n)$, where, $x_1, x_2, x_3, \dots, x_n$ are independent variables with same kind of odds.

Let w_x be the resultant uncertainty and $w_{x1}, w_{x2}, w_{x3}, \dots, w_{xn}$ be the uncertainty in independent variables $x_1, x_2, x_3, \dots, x_n$ respectively. The uncertainty in result is given by;

$$w_x = \sqrt{\left(\frac{\partial X}{\partial x_1}\right)^2 (w_{x1})^2 + \left(\frac{\partial X}{\partial x_2}\right)^2 (w_{x2})^2 + \dots + \left(\frac{\partial X}{\partial x_n}\right)^2 (w_{xn})^2} \quad (A.8)$$

Uncertainty in temperature measurement

Uncertainty in measuring temperature of test section

The T-Type (Copper-Constantan) thermocouple measurements were captured using a data acquisition system. The thin wire thermocouple implies a faster response time. Temperature range of thermocouple is -50 to 200°C. Tolerance according to specifications is $\pm 0.5^\circ\text{C}$.

Uncertainty in measuring temperature of water reservoir

RTD has been used to measure the temperature of water inside the cylinder. RTDs are among the most precise temperature sensors available with resolution and measurement uncertainties or $\pm 0.2^\circ\text{C}$. Usually they are provided encapsulated in probes for temperature sensing and measurement with an external indicator, controller or transmitter, or enclosed inside other devices where they measure temperature as a part of the device's function, such as a temperature controller or precision thermostat.

Uncertainty in the pressure measurement

Pressure transmitters have been used for the pressure measurement of the test section. These pressure transmitters have an accuracy of $\pm 0.1\%$ and resolution of 0.002% over the standard pressure range 0–3 bar.

Uncertainty in flow measurement

EGE's inline sensor SDN 552 (Figure 3.22) with range 0.01-10 litres/min is used for monitoring deionised water flows. For a maximum flow rate used in the present study, the error in the flow 0.5 % measurement is of the order of 4. The flow meter has a precision reading of 2%.

Uncertainty in the channel dimension measurement

The widths of the microchannels were measured using a microscope with a micrometer eyepiece, while the depth of the channel was measured using a dial gauge. The minimal scale interval of these measurements was 1 μm . Uncertainty in measurement of the dimension is $\pm 0.5 \mu\text{m}$.

Uncertainty in wattmeter

The measurement uncertainty associated with the Wattmeter was $\pm 0.7 \%$.

Uncertainty in Reynolds number

Reynolds number is function of these independent variables,

$$\text{Re} = f(\rho, d, v, \mu), \quad (\text{A.9})$$

where ρ, d, v, μ are density of flowing fluid, diameter of microtube, velocity of fluid and viscosity of flowing fluid.

Let w_{Re} be the resultant uncertainty and $w_{\rho}, w_d, w_v, w_{\mu}$ be the uncertainty in independent variables ρ, d, v, μ respectively. The uncertainty in result is given by;

$$w_{\text{Re}} = \sqrt{\left(\frac{\partial(\text{Re})}{\partial \rho}\right)^2 (w_{\rho})^2 + \left(\frac{\partial(\text{Re})}{\partial d}\right)^2 (w_d)^2 + \left(\frac{\partial(\text{Re})}{\partial v}\right)^2 (w_v)^2 + \left(\frac{\partial(\text{Re})}{\partial \mu}\right)^2 (w_{\mu})^2} \quad (\text{A.10})$$

Uncertainty in Reynold number is found to be $\pm 0.66 \%$.

Computing uncertainty in friction factor

Friction factor is function of these independent variables,

$$f = f(\Delta P, \rho, v, D_i, L) \quad (\text{A.11})$$

where $\Delta P, \rho, v, D_i, L$ are pressure drop across microchannel, density, velocity of fluid, hydraulic diameter and length of microchannel respectively.

Let w_f be the resultant uncertainty and $w_{\Delta P}, w_{\rho}, w_v, w_{D_i}, w_L$ be the uncertainty in independent variables $\Delta P, \rho, v, D_i, L$ respectively. The uncertainty in result is given by;

$$w_f = \sqrt{\left(\frac{\partial f}{\partial(\Delta P)}\right)^2 (w_{\Delta P})^2 + \left(\frac{\partial f}{\partial \rho}\right)^2 (w_\rho)^2 + \left(\frac{\partial f}{\partial v}\right)^2 (w_v)^2 + \left(\frac{\partial f}{\partial(D_i)}\right)^2 (w_{D_i})^2 + \left(\frac{\partial f}{\partial L}\right)^2 (w_L)^2} \quad (A.12)$$

Uncertainty in friction factor is found to be $\pm 11.7 \%$

Computing uncertainty in convective heat transfer coefficient

Convective Heat Transfer Coefficient is function of these independent variables,

$$h = f(Q, A_c, \Delta T), \quad (A.13)$$

where $Q, A_c, \Delta T$ are heat flow through microtube, cross-sectional area and temperature drop in microtube.

Let w_h be the resultant uncertainty and $w_Q, w_{A_c}, w_{\Delta T}$, be the uncertainty in independent variables $Q, A_c, \Delta T$ respectively. The uncertainty in result is given by;

$$w_h = \sqrt{\left(\frac{\partial h}{\partial Q}\right)^2 (w_Q)^2 + \left(\frac{\partial h}{\partial(A_c)}\right)^2 (w_{A_c})^2 + \left(\frac{\partial h}{\partial(\Delta T)}\right)^2 (w_{\Delta T})^2} \quad (A.14)$$

Uncertainty in convective heat transfer coefficient is found to be $\pm 7.3 \%$

Computing uncertainty in Nusselt number

Nusselt number is function of these independent variables,

$$Nu = f(h, L, k), \quad (A.15)$$

where h, L, k are convective heat transfer coefficient, length of microtube and thermal conductivity of the material of microtube respectively.

Let w_{Nu} be the resultant uncertainty and w_h, w_L, w_k , be the uncertainty in independent variables h, L, k respectively. The uncertainty in result is given by;

$$w_{Nu} = \sqrt{\left(\frac{\partial(Nu)}{\partial h}\right)^2 (w_h)^2 + \left(\frac{\partial(Nu)}{\partial(L)}\right)^2 (w_L)^2 + \left(\frac{\partial(Nu)}{\partial(k)}\right)^2 (w_k)^2} \quad (A.16)$$

Uncertainty in Nusselt number is found to be $\pm 13.1 \%$

Summary of uncertainties

The accuracy of the data was assumed by performing an uncertainty analysis based on the analytical method. This method provides a means to estimate an overall uncertainty of data by

integrating the uncertainties of individual sources of error into the data base independently, then using a root-sum-square method to calculate the overall uncertainty.

Error analysis was performed for every test run of each data set and result is given by table A.1

Table A.1 Summary of uncertainties

Uncertainty variable	Uncertainty interval
Temperature, [°C]	±0.5
Pressure transducer, P_{in} [bar]	±0.1%
Mass flow rate, [kg/s]	±0.005
channel diameter, [μm]	±0.5
Digital temperature controller	±0.25%.
Flow velocity,[m/s]	±0.406
Area, [m^2]	±0.056×10 ⁻⁶
Length, [m]	±0.001
Cross section area, [m^2]	±0.265×10 ⁻⁶
Reynold number, [--]	± 0.66 %.
Friction factor, [--]	± 11.7 %
Nusselt number, [--]	± 13.1 %
Convective heat transfer coefficient,[kw/m ² k]	± 7.3 %

APPENDIX B

SPECIFICATIONS OF INSTRUMENTS

B.1 Specifications of micropump Series 200:

Displacement: 1.17 ml/rev

Max. Flow (2 Pole Speed): 3340 ml/min at 2850 RPM (50Hz)

Max. Differential Pressure: 4.2 bar

Max. System Pressure (MAWP): 21 bar

NIPR (Absolute): 180 mbar

Wet Lift (Typical): 51 cm. H₂O (1450 RPM)

Viscosity Range: 0.2 to 1500 cP

Max. Speed: 10,000 RPM

Rotation (Facing Motor Shaft): CW

Weight (Pump head): 0.34 kg

Ports 1/8-27 (F): NPT Side Ports

Driven Magnet (Standard): Ferrite

B.2 Specifications of Heat exchanger- Make: Thermaltake; Model: TMG1

Fan dimension: 120(L) × 120(W) × 35(H) mm

Fan speed: 1300-2400 RPM

Rated voltage: 12 V

Maximum air flow: 38.6 – 93.7 CFM

Noise: 16- 30 dBA

Bearing type: Sleeve

MTBF: 30,000 hours

Connector: 4 pin

Radiator dimension: 153 (L) × 120(W) × 28(H) mm

Material: Aluminium

Tube design: Aluminium, Dimple

Fin design: Aluminium, Louvered

Weight: 530 g

Quick install connector: For 9.5 mm (ID) (3/8'') tubing

B.3 Specifications of Universal Dry Block Temperature Calibrator- Make: Nagman; Model MTC 350

Range	50°C to 350°C
Resolution	0.1°C
Accuracy	±0.4°C
Stability	±0.05°C
Stabilization Time	15 to 20 min
Temperature Readout	°C / °F Switchable
Well Diameter	25 mm
Immersion Depth	110 mm
Heating Time (Ambient to max.)	20 min
Cooling Time (to 100°C)	33 min
Switch Test facility / Supply	Provided / 5 V DC (open)
Interchangeable Thermowells	Single hole to suit 1/4" & 1/2" Probes. Other Sizes / Multihole available as optional
Housing / Mounting	Bench-top
Power Supply / Consumption	230V AC / 50 Hz / 600 W
Instrument Dimensions (L × D × H)	160 × 365 × 350 mm
Instrument Weight	8 kg
Measurement Capability	Thermocouples : ISA Types J, K, T, R, S, N, RTD : Pt100 IEC Std. mA : 0 to 25 mA, mV : 0 to 100 / 1000 mV Volt : 0 to 10 V Resistance : 0 to 500 Ohms / 3.5 kOhms

B.4 Range, Resolution & Accuracy of Universal Dry Block Temperature Calibrator

Type	Range	Resolution	Accuracy
mV (Low)	0 to 100	0.01	±0.05% FS
mV (High)	0 to 1000	0.1	
Volts	0 to 10	0.001	
mA	0 to 25	0.001	
Ohms	0 to 500	0.01	
KOhms	0 to 3.5	0.0001	
TC Type J (°C)	-100 to 1200	0.1	±0.1% FS
TC Type K (°C)	-60 to 1260		
TC Type T (°C)	0 to 350		
TC Type R (°C)	150 to 1760		
TC Type S (°C)	170 to 1760		
TC Type N (°C)	0 to 1300		
RTD Pt100 (°C)	-100 to 800		±0.05% FS

APPENDIX C

COMPUTATIONAL RESULTS

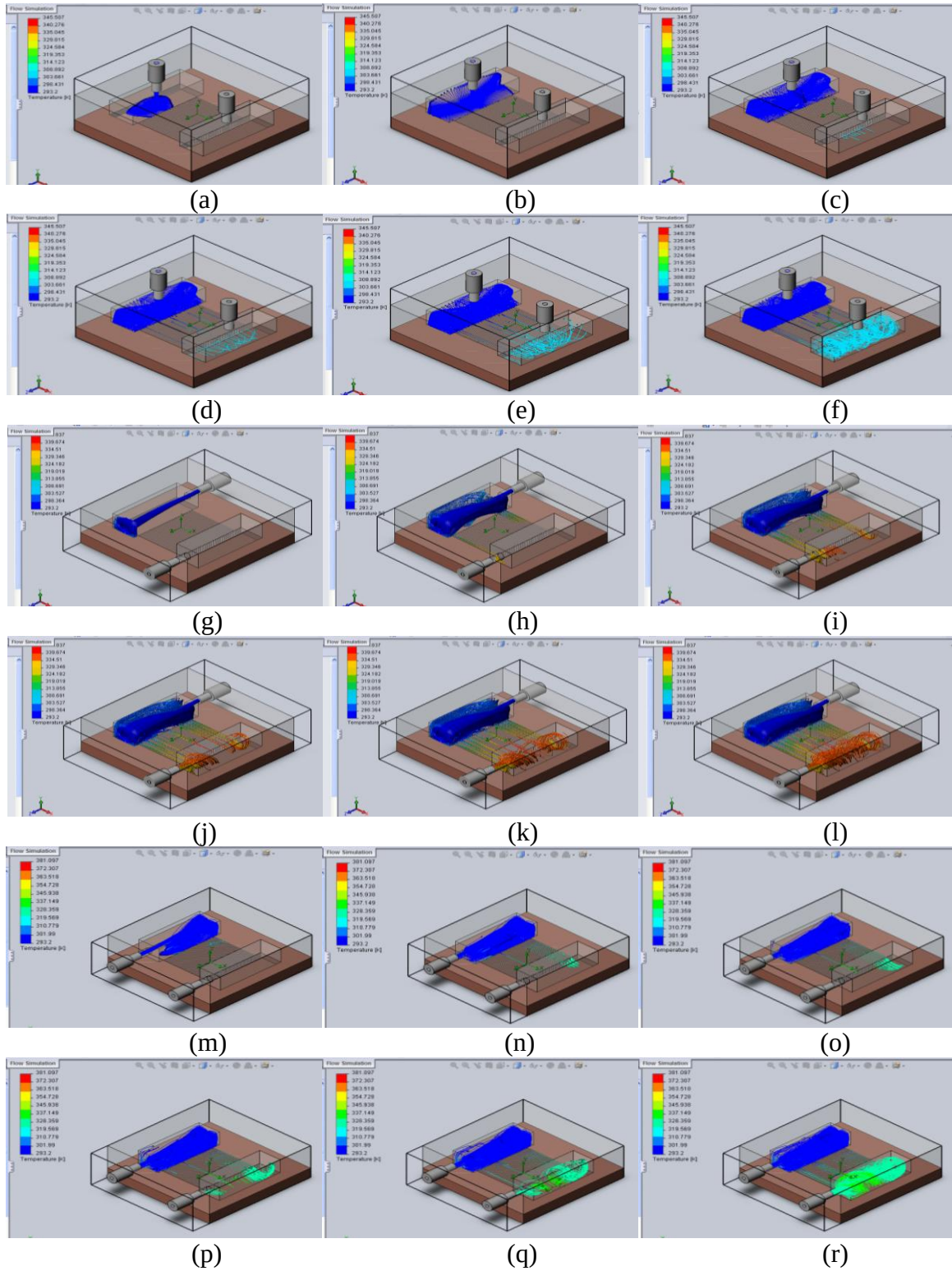


Fig.C1: Flow trajectories within MCHS having inlet and outlet plenum dimensions of $8\text{ mm} \times 30\text{ mm} \times 2.5\text{ mm}$ for P-type, S-type and U-type flow arrangement.

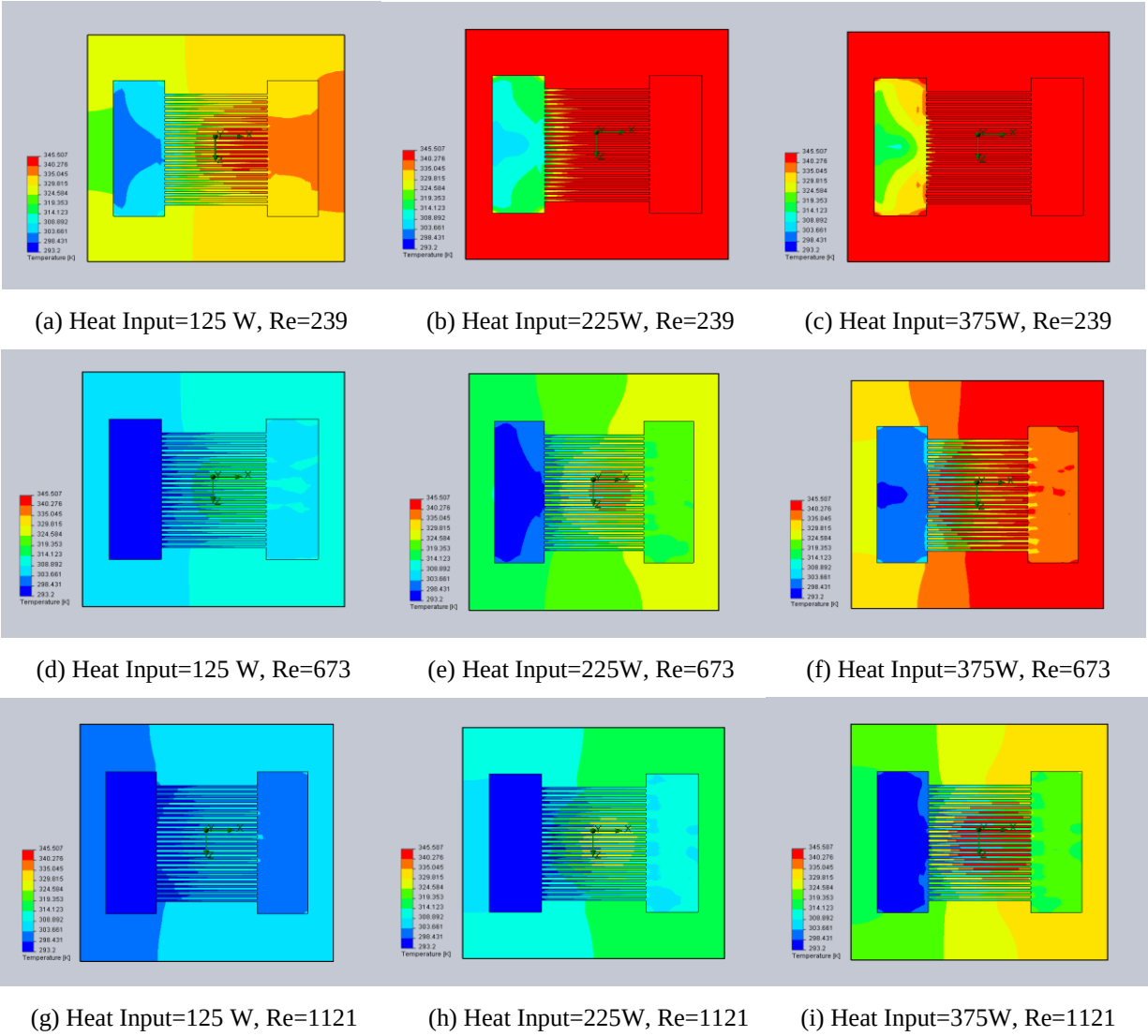


Fig.C2: Temperature contours (along inlet, outlet plenum of 10 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for P-type MCHS at various heat inputs and Reynolds number

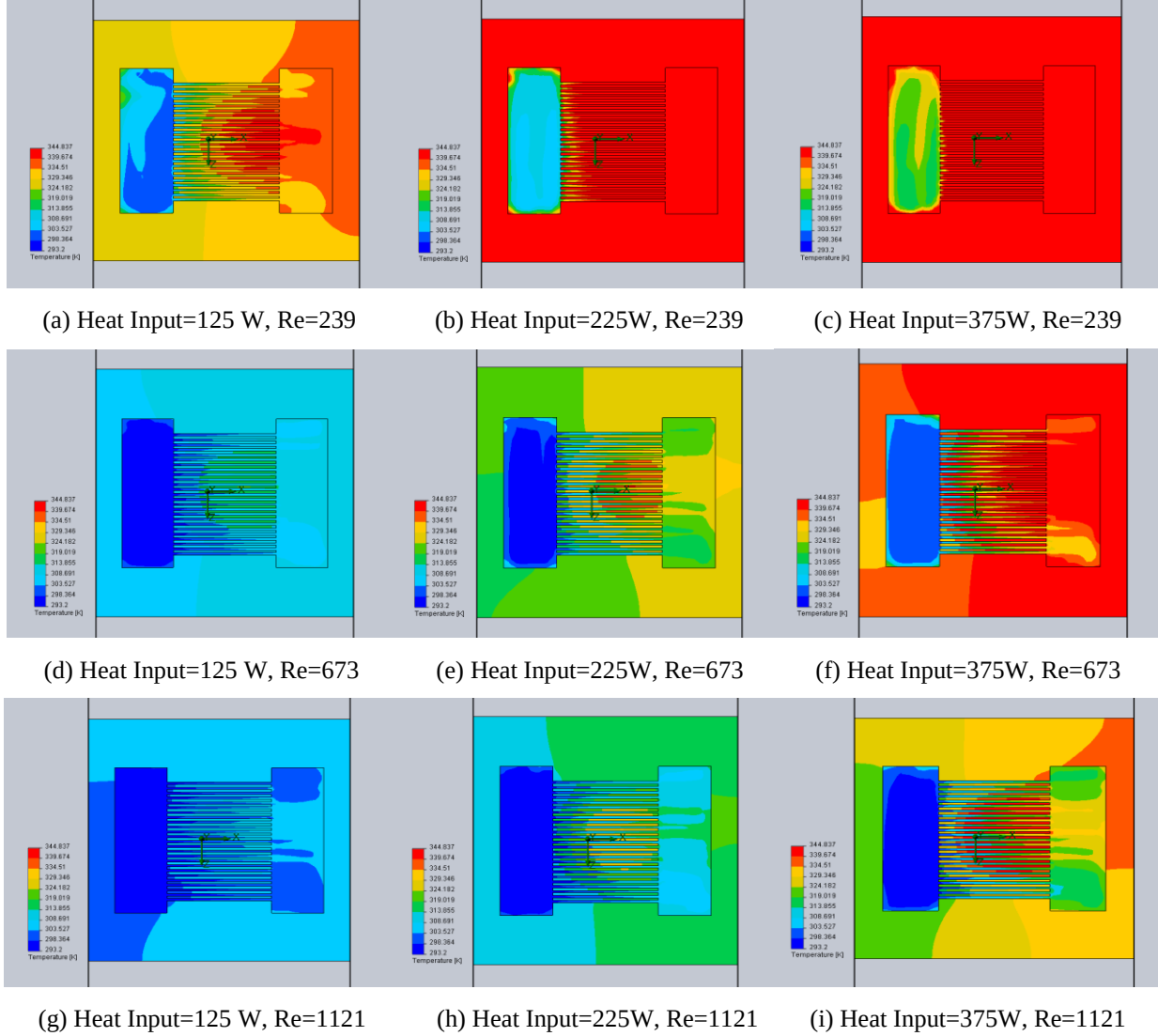


Fig.C3: Temperature contours (along inlet, outlet plenum of 10 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for S-type MCHS at various heat inputs and Reynolds number

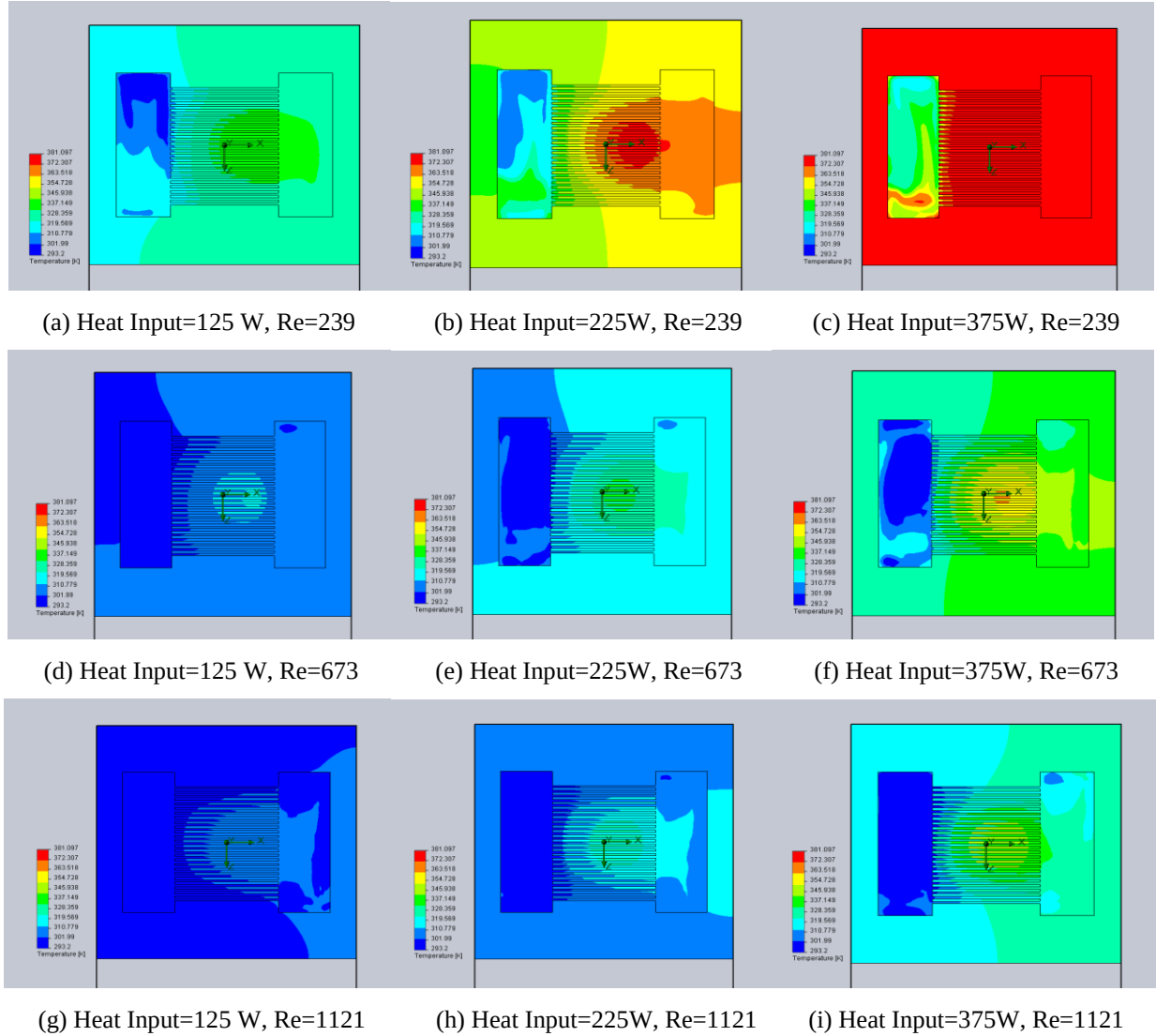


Fig.C4: Temperature contours (along inlet, outlet plenum of 10 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for U-type MCHS at various heat inputs and Reynolds number

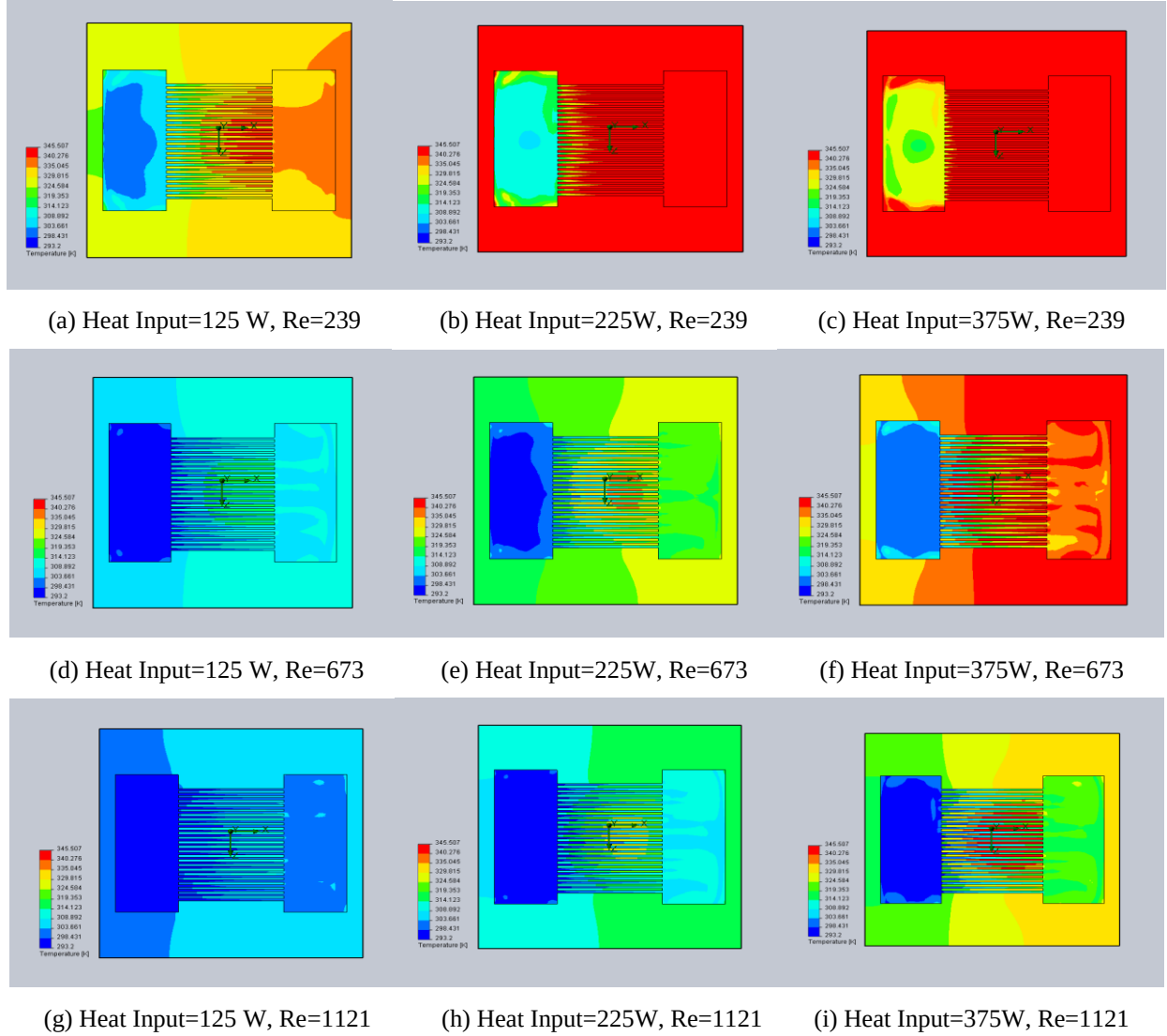


Fig.C5: Temperature contours (along inlet, outlet plenum of 12 mm × 30 mm × 2.5 mm and array of micro channels) for P-type MCHS at various heat inputs and Reynolds number

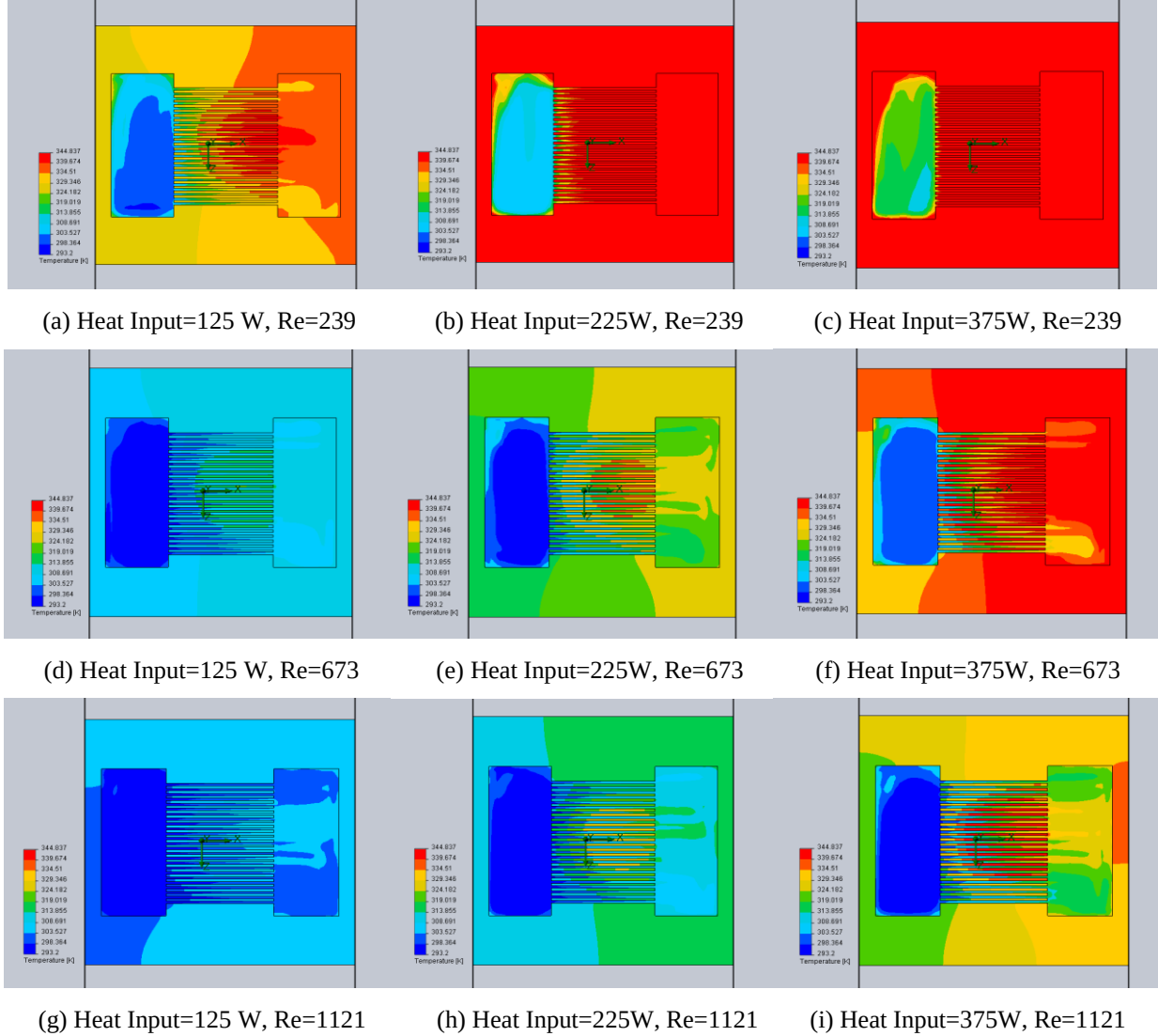


Fig.C6: Temperature contours (along inlet, outlet plenum of 12 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for S-type MCHS at various heat inputs and Reynolds number

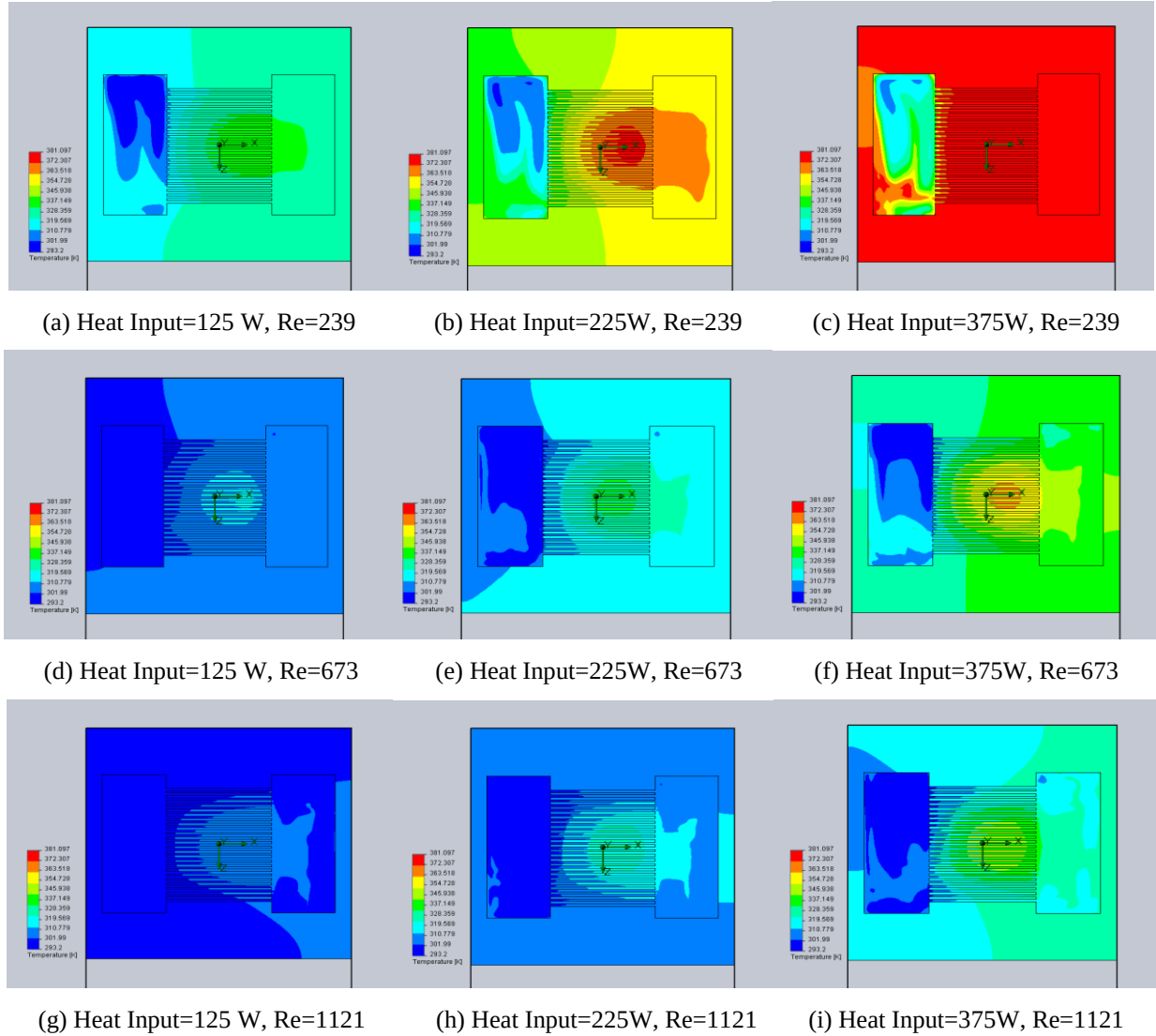


Fig.C7: Temperature contours (along inlet, outlet plenum of 12 mm $\times$ 30 mm $\times$ 2.5 mm and array of micro channels) for U-type MCHS at various heat inputs and Reynolds number

PUBLICATIONS FROM THE RESEARCH WORK

JOURNAL PUBLICATIONS

1. Sehgal S. S., Murugesan K., Mohapatra S. K., Effect of channel and plenum aspect ratios on the performance of microchannel heat sink under different flow arrangements, *Journal of Mechanical Science and Technology*, Vol. 26, No. 9, Publisher: Springer (2012) 2985-2994.
2. Sehgal S. S., Murugesan K., Mohapatra S. K., Experimental investigation of the effect of flow arrangements on the performance of a microchannel heat sink, *Experimental Heat Transfer*, Vol. 24, No. 3, Publisher: Taylor and Francis (2011) 215-233.
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INTERNATIONAL CONFERENCES

4. Sehgal S. S., Murugesan K., Mohapatra S. K., Effect of flow arrangements on thermohydraulic performance of a microchannel heat sink, *Proceedings of MNHMT2009, ASME 2009, 2nd Micro/Nanoscale Heat & Mass Transfer International Conference*, December 18-22, 2009, Shanghai, China (Paper Number: MNHMT2009-18016)
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12. Sehgal S. S., Murugesan K., Mohapatra S. K., Effect of flow arrangements on the performance of a microchannel heat sink for electronic cooling, International Journal of Emerging Multidisciplinary Fluid Sciences, Multi-Science Publishing Company

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