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CFD and Heat Transfer Analysis of Microchannel Heat Sink with Different Inlet and Outlet Flow Arrangements

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Abstract

The overall performance of microchannel heat sink (MCHS) as a system is affected by the inlet and outlet manifold arrangements in addition to array of microcannel. In the present study the CFD analysis is performed w.r.t the heat transfer coefficient, and Nusselt number for three different manifold arrangements. In the present work three different arrangements: Rectangular (R), Rectangle with Semi-Circular (RSC) and Divergent Convergent (DC) of the inlet and outlet manifolds are selected. The result shows MCHS within the Reynolds number ranging from 342-857, the DC type manifold has highest heat transfer coefficient (9%-10%) in comparison to RCS and R type manifold being the lowest.

Keywords: Heat Transfer, Manifold, Micro-channel Heat Sink, Reynolds Number

1. Introduction

Convection is the process of heat transfer seen in fluids or gases when it flows over the surface. When the gas or any fluid flows over the surfaces, it carries away the heat or gives its heat to the surface depending upon the difference in temperature. Micro channels are the mechanism utilized to enhance the convective heat transfer in several engineering applications. In [1] propose the concept of micro channel heat sinks for the first time. When compared with conventional heat exchanger micro channels have a low to moderate pressure drop, lower coolant need smaller geometric size and lower operational cost with higher heat transfer performance. In[2] described micro channel on the base of hydraulic diameter as:

Micro heat exchanger: $1\ \mu m \le D_h \le 100\ \mu m$

Macro heat exchanger: $100\ \mu m \le D_h \le 1$ mm
Compact heat exchanger: $1\ mm \le D_h \le 6$ mm
Conventional heat exchanger: $D_h \ge 6$ mm
where, D_h is a hydraulic diameter.

It has provided the practical means to achieve a large total heat transfer surface area without the use of excessive amount of primary surface area. A numerical study has been borne out [3]. On the modeling the uniformity of manifold with various configurations. In this study a numerical model has been developed to specify the current distribution and pressure drop in the parallel tubes and to validate the answer with the data obtained from experimental setup and set the optimum design of the tapered manifold that can give uniform water distribution through changing the diameter ratio (D1/D2) parametrically. CFD and simulation has performed to validate results. The goal of this investigation is to evaluate the hydraulic parameter of manifold so that same rate of mass outflow can be obtained from the outlet of the manifold. The CFD and simulation and experimental data at different outlets and configurations, a numerical model has used to predict the flow across each lateral for three different Reynolds numbers (i.e., 100,000, 150,000, and 200,000) and the results were found to have the same trend compared with experimental data. In[4] experimentally studied the effect of flow mal distribution on the thermal performance of parallel micro-channel cooling systems. They experimentally analyzed the parameters regarding the flow mal distribution such as channel hydraulic diameter, channel flow configurations (U, Z, I type) across parallel channels. It has found that flow

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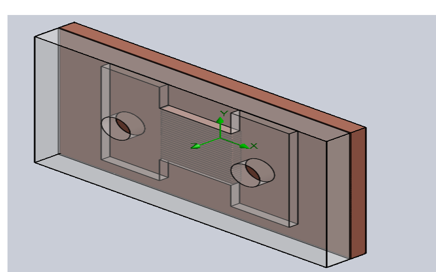
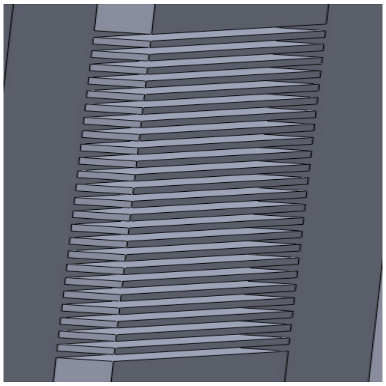
circulation through the channels improves significantly with decrease the hydraulic diameter of channel due to higher pressure drop offered by each individual channels. In[5] experimentally investigated the effect of channel and plenum aspect ratios at different flow arrangements (P, U and S type) MCHS for various Reynolds number at different heat inputs. In this study micro-channel test pieces with two channel aspect ratios, 4.72 and 7.57 and three plenum aspect ratios, 2.5, 3.0 and 3.75 has been tested. In this study the test runs have been performed by taking three heat inputs (125 W, 225 W, 375 W) and the range of Reynolds number ($224.3 \leq Re \leq 1121.7$). Due to decrease in channel width (increase in aspect ratio, determined as the depth to width of the channel) in this work it will show about 126 to 165% increase in Nusselt number, has been observed, where as an increase in manifold length (reduction in the manifold aspect ratio defined as width to length of the manifold) resulted in 18 to 26% increase in Nusselt number. It has been analyzed that for a given range of Reynolds number, the total pressure drop is observed maximum for S-type arrangement. To increase in manifold aspect ratio, there is a decrease in the pressure drop and P-type flow arrangement will have a minimal pressure drop.In[6] experimentally analyzed the flow arrangement effects on the performance of micro-channel heat sink. In this research work the width of each micro-channel is 330 μ m and the depth is 2.5 millimeter. The effects of U, P and S-type of flow arrangements has been studied by varying three heat inputs (125W, 225W and 375W) and Reynolds number range from 224-1121. From this study it has been observed that according to the higher heat transfer coefficient U-type of flow arrangement have better, but due to the lower pressure drop P-type flow arrangement has efficient performance. In[7] analyzed the pressure differences and heat transfer performance of a MCHS by taking Reynolds number 139 to 1672 and 385 to 1289 for different values of heat flux. It has been found that the higher Reynolds number reducing the temperature of outlet water and the heat sink. It results to greater pressure drop.In[8] experimentally investigated heat dissipated by micro channels of different sizes (D_h is 318 to 903 μ m) at different fluid flow rate. It has been observed that heat dissipation will maximum at reducing channel size for a constant fluid flow[9]. In studied the effects of shape of channel on the performance of MCHS. For this study work test pieces have been made of various designs such as (zigzag, curve and stepped) their performance have been studied and checked by wavy and straight channel. It has been found that the pressure drop for all shapes of MCHS will higher than the previous shape of MCHS. The zigzag MCHS experienced a higher pressure drop, friction factor and wall shear stress followed by wavy and curvy and stepped shape of MCHS. In[10] studied the drop in pressure and flow circulation in parallel channel Z-type flow configuration. In this research a model based on mass and momentum equation has been prepared. A special care has been taken on the friction and inertial term. It has been noted that the general polynomial discriminates R2 + Q3 will be applied to define the current distribution and pressure drop. In[11] have reviewed over a micro channel heat sink and they concluded that the various studies have been performed on MCHS. They found that enhancement of heat transfer rate is one of the major concerns in this area. Optimum temperature differences, spacing, width and depth are the most important factor affecting the performance of any device. The performances of the MCHS is also affected by the boundary layer formation. Several experiments have been done and relations were given in parliamentary procedure to count on these parameters. Most of the tests have been performed on aluminum and relations were developed for this. Different parametric studies have also been executed in order to interpret the elements involving the heat transfer process in a MCHS.

2. Methodology

Design of micro channel heat sink showed in Figures1–3 shows the assembly of micro channel heat sink with different manifold arrangements. The material selected for micro channel is of copper while the Cover of Acrylic Sheet. The specification of the micro channel heat sink and Acrylic Cover is given in Table 1. The different parameters taken in consideration for the study have been shown in Table 2. CFD analysis on much have been executed with water as the working fluid. The readings obtained have been recorded. These readings have been used for the calculations so as to find the result.



(a)



(b)

Figure 1. (a) Assembly of MCHS (b) Micro channels.

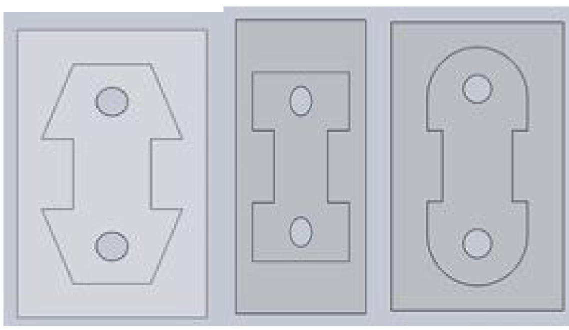


Figure 2. Acrylic cover plates with manifold arrangement.

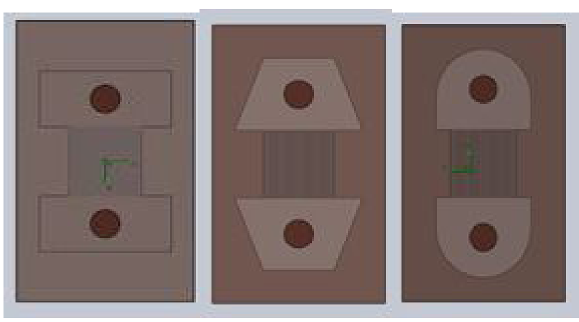


Figure 3. Assembly with rectangular, divergent-convergent and rectangle with semi-circular manifold arrangements.

Table 1. Specification of micro-channels

Width of Channels	500 µm
Spacing Between Channels	500 µm
base Plate Thickness	8000 µm
Length of Channels	24500 µm
Depth of channel	3000 µm
Thickness of Cover	6000 µm

Table 2. Thermo-hydraulic parameters considered

Flow rate (LPH)	4, 6, 8, 10
Wattages	50, 75, 100, 125 W
Base fluid	Water
Design consideration	R, RSC, DC
Number of Experiments Performed	48
Flow type	Perpendicular
Reynolds Number	342-857

The steps of calculation procedure followed are as:

1. Reynolds Number

$$Re = \frac{\rho v d_h}{\mu} \tag{i}$$

2. Average Temperature

$$T_{avg} = \frac{T_1 + T_2}{2} \tag{ii}$$

T_1 Temperature at inlet

T_2 Temperature at outlet

3. Temperature Difference

$$\Delta T = T_w - T_{avg} \tag{iii}$$

T_w is the wall temperature

Constant heat input is applied at the bottom of the MCHS. Other three wall assumed adiabatic. Thermocouple is located bottom of MCHS for observing the wall temperature.

4. Flow Discharge

$$q = m c_p \Delta t \tag{iv}$$

5. Heat Transfer Coefficient

$$h = \frac{q}{A_s \Delta t} \tag{v}$$

6. Nusselt Number

$$N_u = \frac{h d_h}{k} \tag{vi}$$

3. Results and Discussions

3.1 Variation of Heat Transfer Coefficient with Reynolds Number

Through the experiments comparisons have been worked at three different Wattages (50, 75, 100 and 125 W) and at different flow rates. From the result, it has been concluded that the heat transfer coefficient of the Divergent-Convergent is higher among the three followed by rectangular and rectangle with semi-circular Figures 4–7.

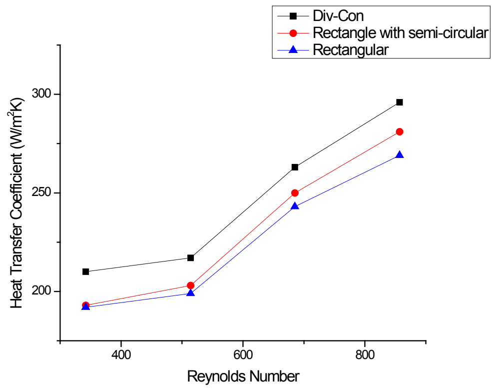


Figure 4. Reynolds Number vs. heat transfer coefficient at 50 W.

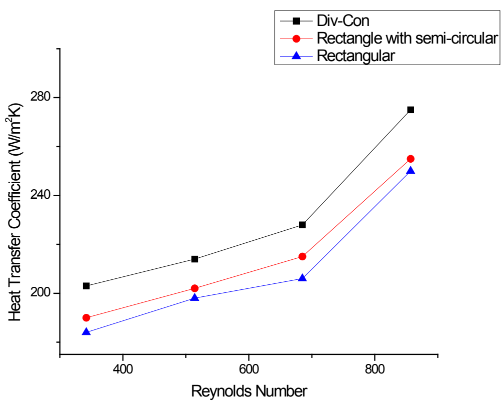


Figure 5. Reynolds Number vs. heat transfer coefficient at 75 W.

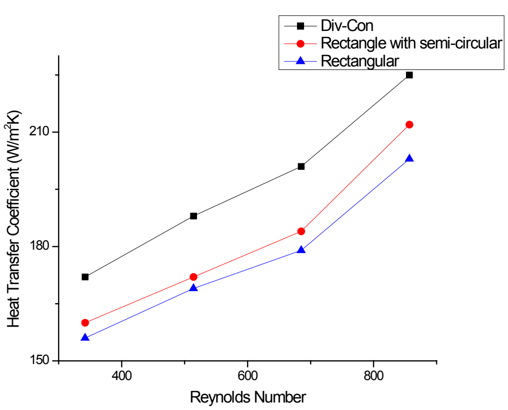


Figure 6. Reynolds Number vs. heat transfer coefficient at 100 W.

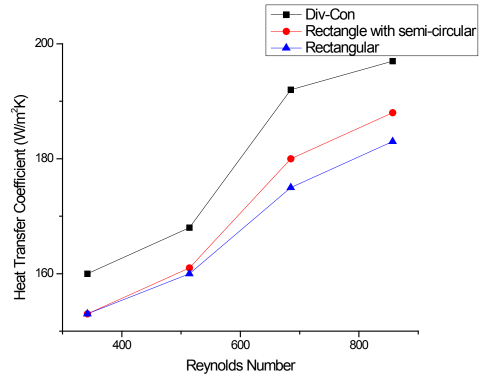


Figure 7. Reynolds Number vs. heat transfer coefficient at 125 W.

3.2 Variation of Nusselt Number with Reynolds Number

Through the experiments comparisons have been made at three different Wattages (50, 75, 100 and 125 W) and at different flow rates. From the result it has been concluded that the Nusselt Number of the Divergent-Convergent is higher among the three followed by rectangular and rectangle with semi-circular Figures 8–11.

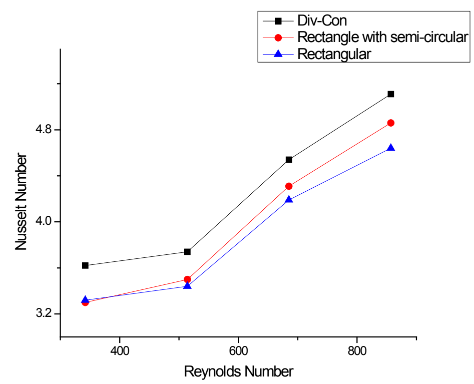


Figure 8. Reynolds Number vs. Nusselt Number at 50 W.

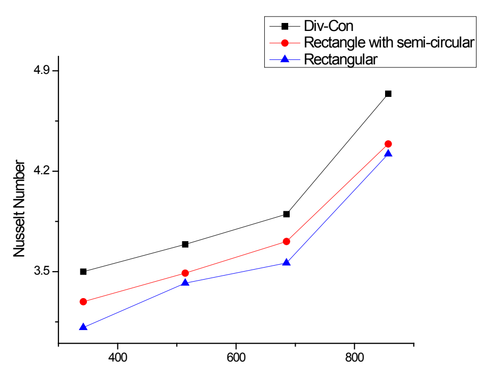


Figure 9. Reynolds Number vs. Nusselt Number at 75 W.

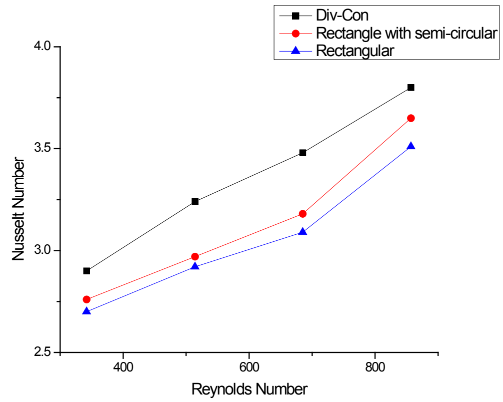


Figure 10. Reynolds Number vs. Nusselt Number at 100 W.

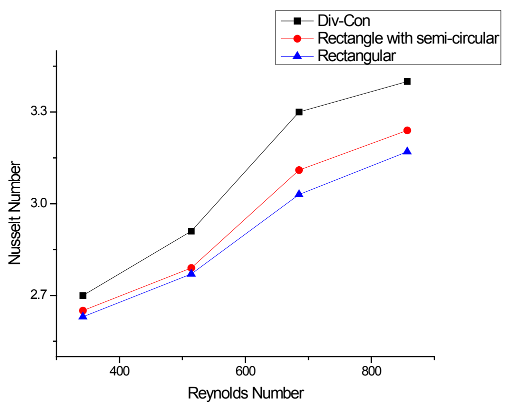


Figure 11. Reynolds Number vs. Nusselt Number at 125 W.

4. Conclusion

In the present work computational analysis have been carried out to calculate heat transfer characteristics on Micro-channel heat sink using water as base fluid.

From the study following conclusions can be drawn:

It has been observed that heat transfer coefficient is increase in Divergent-Convergent manifold than that of rectangular and rectangle with semi-circular manifold by 10 % at 50W, 9.8 % at 75W, 10.2 at 100W and 10.4% at 125 W. Rectangle with semi-circular manifold have more heat transfer than that of rectangle manifold by 2.45% at 50W, 1.81 % at 75W and 1.13 % at 100W but less than the divergent-convergent manifold at different Reynolds number for different heat inputs.

It has been observed that Nusselt number is more in divergent-convergentmanifold than that of rectangle and rectangle with semi-circular manifold 3.21 % at 50W, 3.25 % at 75W and 2.87 at 100W. Rectangle with semi-circular manifold have more Nusselt number than that of rectangle manifold by 2.45% at 50W, 1.81 % at 75W and 1.85 % at 100W but less than the divergent-convergent manifold at different Reynolds number for different heat inputs.

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