

Thermo Hydraulic Studies on Enhancement of Thermal Performance using cylindrical finned Micro channel

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Abstract

Microchannels often provide high heat transfer coefficients due to their tiny hydraulic diameters. Microchannel heat sinks (MCHS) are now thought of as a cooling option for the twenty-first century. The purpose of this numerical research was to investigate the feasibility of using extended surfaces in microchannel heat sinks in order to improve heat transmission. In this investigation, rectangular microchannels with cylindrical fins and rectangular microchannels without fins are used. The experiment included using water as the cooling liquid and copper as the microchannel material. The bottom wall of the heat sink was subjected to two different heat flux levels, $q'' = 100 \text{ W/cm}^2$ and $q'' = 200 \text{ W/cm}^2$. The conjugate fluid flow and heat transfer issue, which involves simultaneously determining the temperature field in both the solid and liquid areas, was investigated using a three-dimensional Computational Fluid Dynamics (CFD) model using FLUENT for analysis. Within a microchannel with a hydraulic diameter of $5.86 \times 10^{-4} \text{ m}$, this study concentrated on laminar flow of single-phase liquids with Reynolds Numbers of 300, 500, 700, and 900 with corresponding velocities of 0.44, 0.72, 1.02 and 1.32 m/s. By measuring these parameters at a known bulk temperature, we may learn how the fluid's thermophysical characteristics affect the flow and heat transmission. It compares and contrasts finned microchannels in terms of their performance factor, pressure drop, average wall temperature, bottom wall temperature, and local Nusslets number.

Keywords: Heat exchangers, finnedchannels,RMC, CFDanalysis.

1. Introduction

Following Tuckerman and Pease's seminal work on VLSI cooling in 1981, other studies have focused on single-phase heat transport in microchannels. First proposed in early 1981, the idea of microchannel heat sinks was advanced by Tuckerman and Pease (1981), who also forecasted that microchannels using single-phase forced convective cooling could remove heat at a rate of $1,000 \text{ W/m}^2$. For many years, industries have relied on liquid injection and forced convection in channels to cool off huge areas quickly. Nevertheless, researchers have shown a growing interest in microchannel heat transfer as a result of its high heat transfer coefficients, which have the potential to surpass those of conventional air and liquid cooled systems while exhibiting low to moderate pressure drops (Philips 1988; Gillot et al. 2000; Hsu et al. 1995; Hahn et al. 1997; Martin et al. 1995; Viday et al. 1993). For instance, in the case of high-power laser diode array cooling, microchannel heat sinks have been shown to be effective, with a 500 W/cm^2 heat flux removal rate (Missaggia et al. 1989; Mundinger et al. 1988; Beach et al. 1992).

The fast development of micro-devices used for cooling of high-power density micro-electronics, supercomputers, plasma facing components, high-powered lasers, and medical devices has made studies on two-phase flow and heat transfer characteristics in microchannel flow passages increasingly important in recent decades. Simple forced air-cooling solutions may not be able to keep up with the increasing need for more

densely packed microchips. As a secondary option, liquid cooling with microchannels and microchips is very appealing. competing option (Schmidt 2003). Extremely high heat fluxes are produced when a device platform has a great number of electrical circuits. Inadequate heat dissipation increases the risk of circuit failure, which in turn increases the risk of device failure due to elevated substrate temperature. For the purpose of dissipating the heat produced by electrical circuits, Tuckerman and Pease first proposed the idea of microchannel heat sinks (MCHS) in 1981.

Fluids may be guided via closed channels called microchannels, which have hydraulic diameters that vary from tens of micrometres to hundreds of micrometres. Microchannels allow for better heat transfer coefficients since their hydraulic diameters are smaller. Compared to millimeter-sized and traditional channels, microchannel heat sinks provide a number of benefits, such as a larger heat transfer surface per volume and a reduced need for coolant. The Nusselt number remains constant in fully developed laminar flow when the heat flux and temperature remain constant. As a result, the correlation between diameter and heat transfer coefficient is inverse. In this case, the heat transfer coefficient is directly proportional to the channel's diameter. In fully formed laminar flow conditions, microchannels may be used to increase heat transfer without raising the coolant velocity. Lee Yong-Jiun et al. [1] He conducted experiments on obliquely finned microchannels, which showed that compared to conventional microchannels, the combination of re-entrance and the secondary flow effect from the fins significantly improved heat transfer performance. The experiments were conducted using copper-based microchannels. Copper microchannel heat sinks operating in water may achieve an average Nusselt number (Nu_{ave}) rise of up to 80%, from 8.6 to 15.8. The overall thermal resistance is reduced by 18% due to the enhanced convective heat transfer, and the maximum base temperature increase above intake fluid temperature drops by 9.3°C, from 50.0°C to 40.7°C. As an interesting aside, this innovative heat transfer enhancement methodology has a

much less pressure drop penalty than the standard enhancement methods. I, Wei, and Joshi [2] Wei and Joshi covered micro channel heat sinks. To enhance the overall accessible flow area, many layers of tiny channels are stacked. At a constant flow rate, the pressure drop decreases at a rate almost proportional to the layer count. Several factors influence the thermal performance of stacked micro channel heat sinks, including the flow rate, base material thermal conductivity, and channel aspect ratio. In order to modify the flow directions and interlayer flow rate ratio for various microprocessor power maps, a basic thermal-resistance-network analysis revealed that, with fixed pumping power, the total thermal resistance of a two-layer microchannel is 30% lower than that of a single-layer microchannel of the same dimensions.

(Chen et al., 2003) The impact of several cross-sectional shapes on the heat transfer performance of a silicon heat sink in the laminar domain ($Re = 50-500$) was studied numerically. The shapes considered were trapezoidal, rectangular, and triangular. They found that channels with a triangular cross section performed better in heat transmission than channels with a rectangular or trapezoidal cross section with the same hydraulic diameter. Harirchian and colleagues [4] Purdue University's Harirchian et al. performed experiments in 2009 using Fluorinert FC-77, a perfluorinated dielectric fluid, and parallel micro-channels with widths ranging from 100 to 5850 μm and a depth of 400 μm . The results demonstrated that for widths over 400 μm , the heat transfer coefficient remained nearly constant regardless of the channel size.

2. Modelling of Microchannel heat sink

The dimension of rectangular microchannels have a width of 700 μm and a depth of 1100 μm as shown in Fig.1. Copper is used as the heat sink material and water is used as the cooling fluid. The electronic component is idealized as a constant heat flux

boundary condition at the heat sink bottom wall, hence a constant heat flux of 100 W/cm^2 and 200 W/cm^2

is applied at the bottom wall of Microchannel (MC).

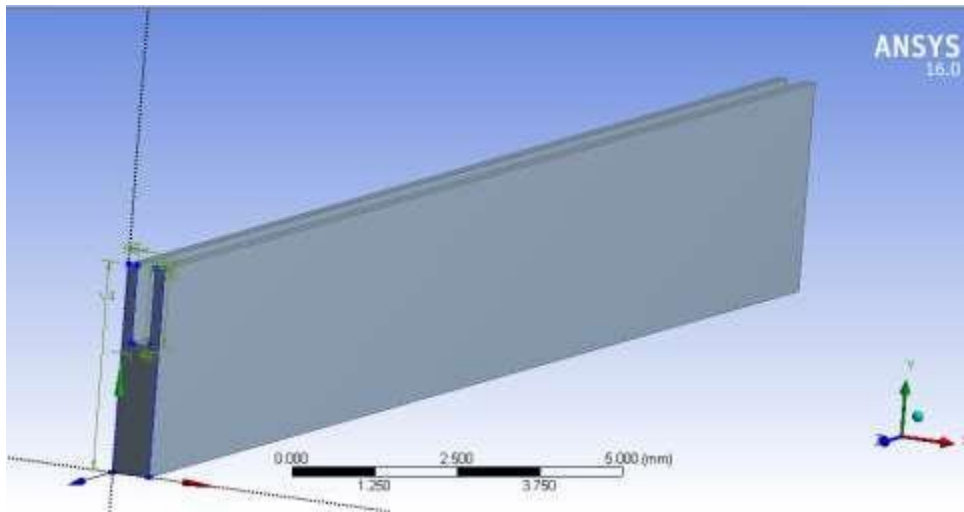


Figure 1. Modeling of Rectangular Microchannel

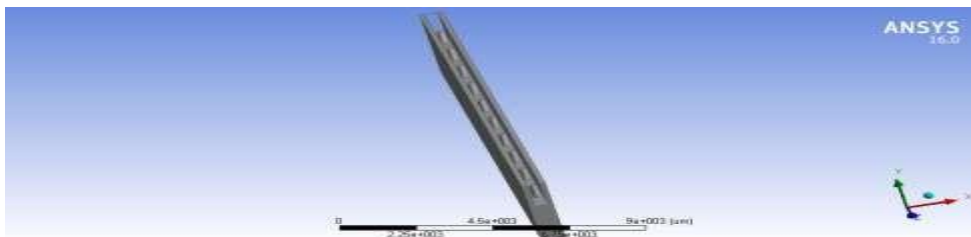


Figure 2: Modeling of Cylindrical finned Rectangular Microchannel

Heat transport in the unit cell is a conjugate problem which combines heat conduction in the solid and convective heat transfer to the coolant (water). Water entering into the channel with a temperature of 300 K . The other wall boundaries of the solid region are assumed perfectly insulated with zero heat flux. The water flow velocities are taken from different Reynolds numbers at 300, 500, 700 and 900 with reference to the hydraulic diameter of $5.8 \times 10^{-4}\text{ m}$. The dimensions of the microchannel are Length of the microchannel = $25000\text{ }\mu\text{m}$. Width of the microchannel = $700\text{ }\mu\text{m}$. Height of the microchannel = $3100\text{ }\mu\text{m}$. Depth of the microchannel = $1100\text{ }\mu\text{m}$.

The following assumptions are considered for the model analysis

- i. The flow is considered as Laminar flow.
- ii. Uniform wall heat flux.
- iii. Gravitational effects are not considered.
- iv. Negligible radiation heat transfer.
- v. The flow of coolant is steady and incompressible fluid.
- vi. Variable thermophysical properties.

2.1

Grid independence

Grid independency test has been carried out to identify the optimum size of the mesh. Simulations have been performed for three different mesh sizes. Table 2 shows the mesh configuration and estimated Nusselt number. Decreasing the mesh size from 0.06 mm to

0.05 mm effecting the nusselt number by 0.87% and further decreases to 0.04 mm size the Nusselt number by 0.11%. Hence further decreasing the mesh size increases the computation time but the variation in nusselt number is insignificant. Hence, it was decided to simulate the present study with mesh size of 0.04 mm.

Table 1: Mesh Configuration and estimated Nusselt number

Mesh Size	Elements	Nu
0.06mm	335437	8.421
0.05mm	458485	8.542
0.04mm	868552	8.522

3. Numerical method

Equations of momentum, energy, and continuity, derived from the laws of heat and fluid dynamics, provide the basis of the governing theory. Simplified Method for Pressure Linked Equation (SIMPLE) algorithm implementation is the goal of these equations. This leads to the solution of the Navier-Stokes equations. The energy, momentum, and pressure discretization equations are based on second-order upwind. Numerical models were developed to depict the concomitant heat transfer and fluid flow inside the microchannel structure. We apply uniform pressure and

temperature conditions at the inlet and outflow by solving the energy, momentum (Navier-Stokes), and steady-state continuity equations. In order to conduct the investigation, the residue convergence conditions for the continuity, momentum, and energy equations were set at 10⁻⁶. Separate tests were conducted for each geometrical arrangement using three different grid sizes (0.04, 0.05, and 0.06). When the change in results with further reduction in grid size was less than 2%, further grid refinement was halted to save computer resources and time. The equations governing the fluid zone's continuity, momentum, and energy are

Continuity equation

$$\nabla \cdot (\rho \vec{v}) = 0$$

Momentum equation

$$\nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla P + \nabla \cdot (\mu \nabla \vec{v})$$

Energy Equation

$$\nabla \cdot (\rho \vec{v} C_p T) = \nabla \cdot (K \nabla T)$$

Energy equation for solid zone is

$$\nabla \cdot (k \nabla T) = 0$$

4. Data Reduction for Microchannel

Reynolds number is given as

$$Re = \frac{\rho U_{avg} D_{mc}}{\mu} \text{-----(1)}$$

Where,

μ = Dynamic viscosity of working fluid,

D_{mc} , U_{avg} and ρ are hydraulic diameter of microchannel, Average fluid velocity and density of working fluid flowing through the microchannel respectively.

Hydraulic diameter is calculated by

$$D_h = \frac{4 * W_{mc} * H_{mc}}{\gamma (W + H)} \text{-----(2)}$$

Pressuredropiscalculatedby

$$\Delta P = \left(\frac{1}{\gamma}\right) * \rho * U_{avg}^2 * \frac{L}{n} \text{-----(3)}$$

Where ,

W_{mc} is width of single microchannel, H_{mc} is Height of the microchannel

L is the length of the

microchannel Nusselts number is find out by using

$$Nu = \frac{h D_h}{k} \text{-----(4)}$$

Where,

h and k represents convective heat transferrate, thermal conductivity of working fluid.

The following equations are used to evaluate Heat transfer coefficient and wall heat flux.

$$h = \frac{q_w}{T_w - T_{avg}} \text{-----(5)}$$

$$q_w = \frac{q * A_b}{N * A_s} \text{-----(6)}$$

Where,

T_w = channel wall

temperature T_{avg} = average fluid temperature

q = *total heat flux applied at the base of the channel*

q_w =

heat convected from inner bottom and side walls of channel A_b = area of bottom wall

A_s = is sum of the surface areas of bottom and side walls of channel and N = Number of channels

5. Results and Discussion

In this part, we will go into more detail on the rectangular micro channel's (RMC) thermo-hydraulic behaviour. Knowing how heat and mass flow affect RMC's thermo-hydraulic behaviour requires running a battery of numerical simulations. For the sake of keeping things simple in the computer model, we only look at one micro channel passage (Fig. 1). Hot air pressure of 100 and 200 W/cm² applied from the floor up. At 300K, the working fluid, which is water, is delivered. The mass flow changed when the Reynolds numbers 300, 500, 700, and 900 changed, in that order. We express the findings in terms of pressure drop, Nusselt number, wall superheat, and wall temperature.

5.1 ThermoHydraulic Studies on Cylindrical Finned Rectangular Microchannel

5.1.1 Average Wall Temperature

Fig. 3 and 4 shows the variation of average local wall temperature across the length of the channel at heat flux of 100 W/cm² and 200 W/cm² respectively. The local average wall temperature is increased across the channel length. However, the local average wall temperatures profiles of the cylindrical finned RMC are significantly differ from RMC. It is also observed that a naverage local wall temperature across the cylindrical finned RMC is lower as compared to RMC. Presence of extended surfaces ruptures the thermal boundary layer and hence the redevelopment of thermal boundary offers better mixing hence the local wall temperature decreases. The cylindrical finned RMC wall temperature contours exhibits linear

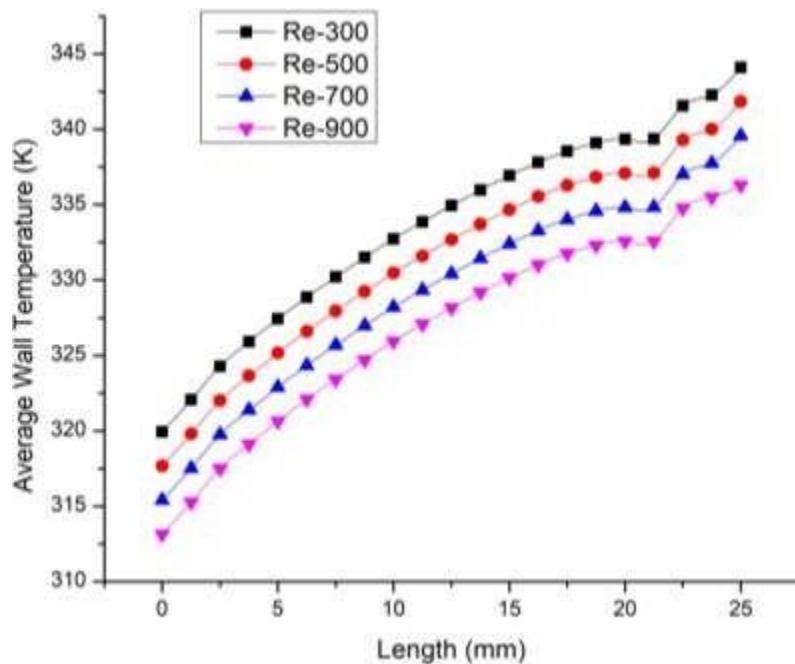


Figure 3. Variation of average wall temperature across the Cylindrical finned RMC for 100W/cm².

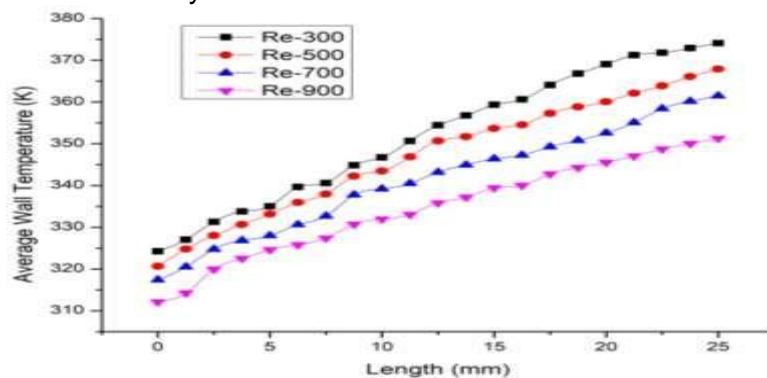


Figure 4. Variation of average wall temperature across the Cylindrical finned RMC for 200W/cm².

5.1.2 Wall Superheat

At 100 W/cm² and 200 W/cm², Figures 4.33 and 4.34 show the effects of cylindrical fins on cylindrical walls, respectively. When calculating the bulk fluid temperature and wall temperature, two points along the fin and in the centre of two consecutive fins are taken into account. Figures 4.33 and 4.34 show that the wall super heat for cylindrical finned RMC follows an oscillating pattern.

Wall superheat was found to be lower at the fin position and higher in the fin's central region. There is a substantial

difference in the local bulk fluid temperature between the fin and the mid-fin regions. The reduction in wall superheat and rise in local bulk fluid temperature are both caused by the greater accessible area at the fin.

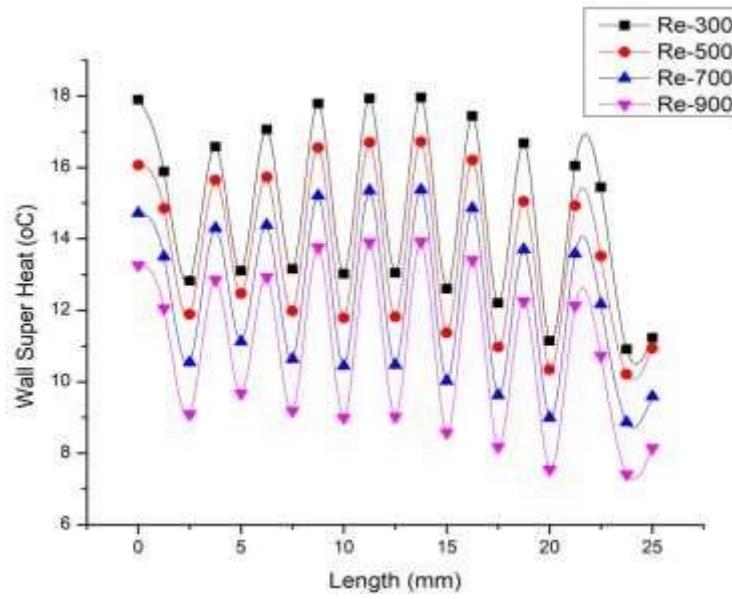


Figure 5. Relation between wall temperature and working fluid temperature for $100\text{W}/\text{cm}^2$.

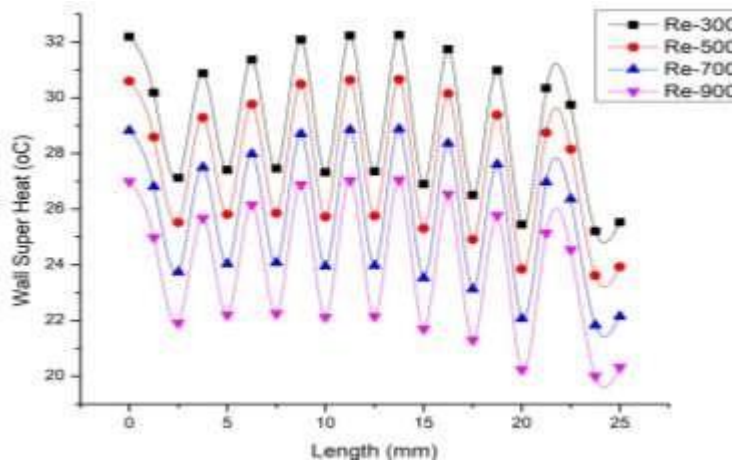


Figure 6. Relation between wall temperature and working fluid temperature for $200\text{W}/\text{cm}^2$.

5.1.3 NusseltNumber

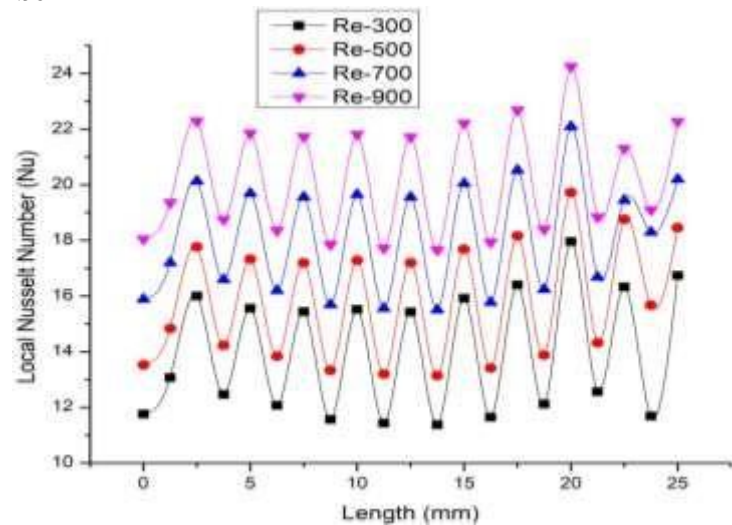


Figure 7. Variation of local Nusselt number across the length of the Cylindrical finned RMC for $100\text{W}/\text{cm}^2$.

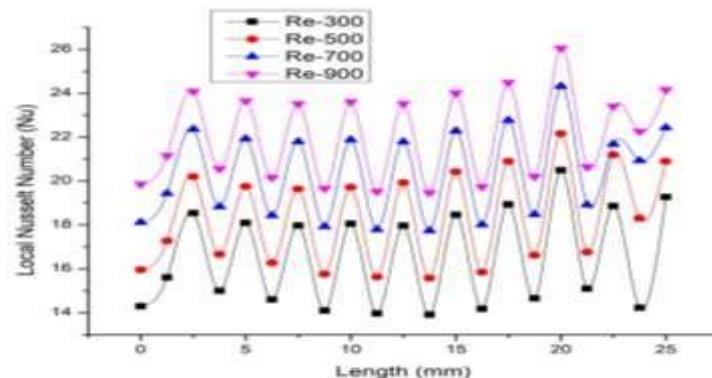


Figure 8. Variation of local Nusselt number across the length of the Cylindrical finned RMC for 200 W/cm^2 .

The influence of cylindrical fin on local Nusselt number is depicted in Fig.7 and 8. It is observed that the local Nusselt number is high at fin locations and low at middle of the two successive fin locations. At the fins more heat transfer takes place hence the wall superheat decreases which results in an increase in local Nusselt number and vice versa. This phenomenon is observed at both heat fluxes.

5.2 Comparative studies on finned RMC's

In order to know the influence of fin shape, a comparison is done for microchannels RMC, cylindrical finned RMC. The average Nusselt number, pressure drop and performance factor are considered in this comparison study.

5.2.1 Nusselt Number

Figure 9 shows the effect of fin shape on the average Nusselt Number at 100 W/cm^2 and 200 W/cm^2 heat flux.

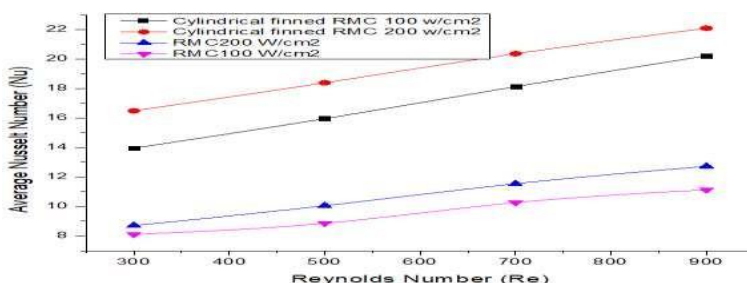


Figure 9. Variation of Nusselt Number with Reynolds Number for different Microchannels for heat flux of 100 W/cm^2 .

Figure 9 shows that the average Nusselt number is higher for finned RMC than for RMC. At all Reynolds numbers, the Nusselt number is greater for cylindrical finned RMCs compared to RMCs without fins. The reason for this is because the heating surface's interface temperature is kept lower by the microchannel's upstream section. At a heat flow of 200 W/cm^2 , we likewise see the same patterns. At 100 W/cm^2 and 200 W/cm^2 , the Nusselt number for cylindrical finned channels increases by 85.55% and 83.52%, respectively. Under certain operational circumstances, a lower wall temperature is produced by a higher Nusselt number, which implies that more heat is transferred down the channel. According to these findings, the upstream finned RMC channel outperforms the other three in terms of heat transfer performance. At a heat flux of 200 W/cm^2 , the same thing happens as well.

5.2.2 Pressure drop

Fig.10 shows the variation of Pressure drop with Reynolds Number for all three microchannels at the heat flux of 100 W/cm². As we know that Pressure drop increases for the finned microchannels due to additional flow resistance offered by extended surface in fluid flow path. From the fig. 4.10 it is observed that the pressure drop for finned channel is more compared to RMC.

The average increase in pressure drop in case of upstream finned RMC is 14.64% at 100W/cm² as compared with RMC. The same phenomenon is also observed at 200W/cm² heat flux.

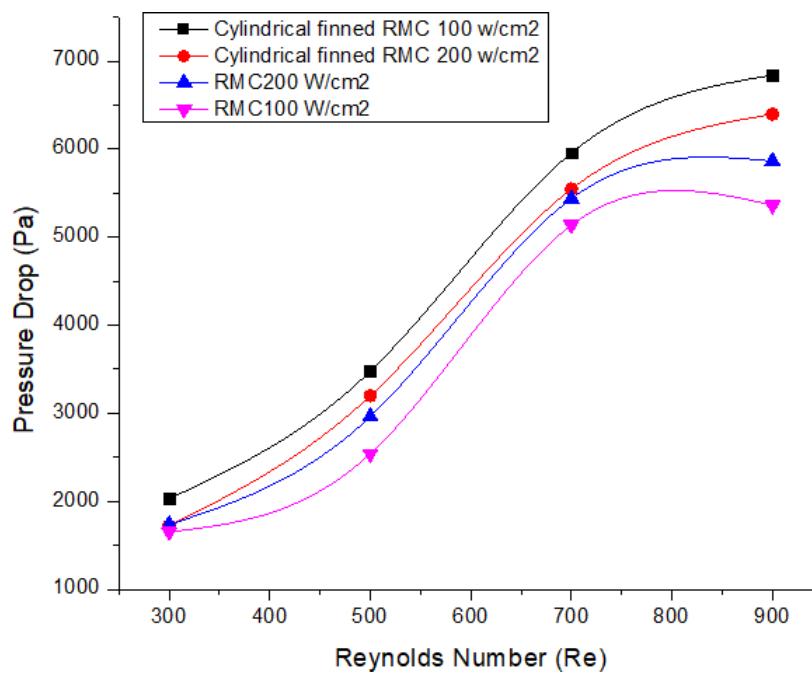


Figure 10. Variation of Pressure drop with Reynolds Number for different Microchannels

5.3.3. Performance factor

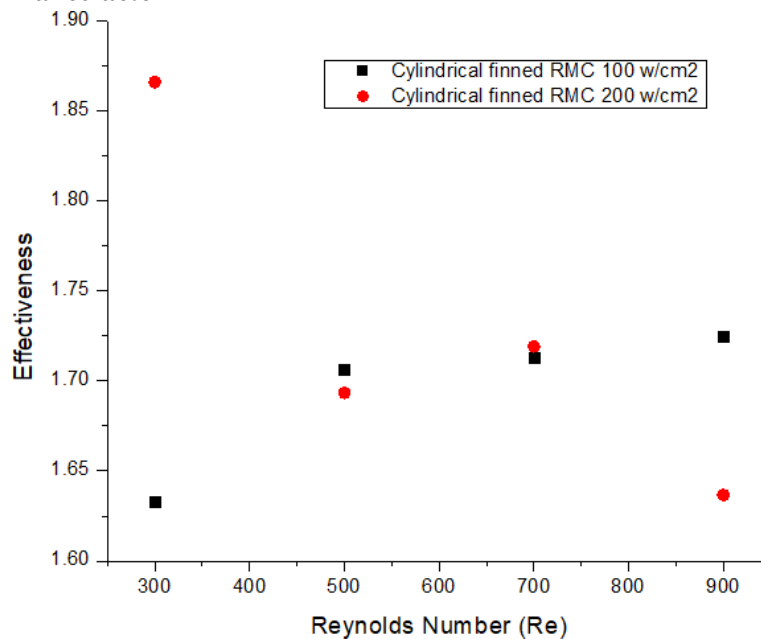


Figure 11. Effectiveness of finned microchannels at 100 W/cm² and 200 W/cm² compared with rectangular microchannel.

Passive methods, including changing the flow channel route, may enhance the microchannel performance. The thermal performance (Nusselt number) and hydraulic behaviour (pressure drop) are affected by these changes in the flow channel, which may be either increased or decreased. By attaching cylindrical fins to the base wall, the flow route is altered in this research. These findings lead to an increase in the average Nusselt number, although they are accompanied by a decrease in pressure drop. We utilise the thermo-hydraulic performance factor to assess the combined effects of finned channels on the Nusselt number and pressure drop.

At 100 W/cm² and 200 W/cm² heat fluxes, Figure 11 demonstrates the fluctuation of the thermo-hydraulic performance factor with Reynolds number. The high thermo-hydraulic performance factor of finned RMC is evidence of the significant impact of the expanded surface (fins) on microchannels. Figure 11 shows that finned RMC outperforms RMC in terms of performance. At a heat flux of 100 W/cm² and 200 W/cm², the average improvement in the thermo-hydraulic performance factor for finned is 69.43% and 72.87%, respectively.

6. Conclusion

The current study presents the results of numerical computational investigations into the thermo-hydraulic behaviour of rectangular microchannels (RMCs). Next, we look at how a rectangular microchannel (RMC) is affected by a cylindrical fin form. An investigation was conducted by altering the mass and heat fluxes (in terms of Reynolds number) in 3D computational fluid dynamics (CFD) models that were created using ANSYS 16.0 software. The current study's findings are as follows.

For the purpose of analysing the thermo-hydraulic behaviour of RMCs, the created computational model was first verified using the experimental work of Yong-Jiun Lee et al.

2. Wall temperatures of finned RMCs are much lower than those of RMC. Cylindrical finned RMC and RMC are the channels with the lowest wall temperatures.

3. RMCs with fins have much better heat

transfer performance. The result is a drop in wall superheat. At all operating circumstances, the upstream finned RMC has reduced wall superheat compared to the other and cylindrical finned RMCs. 4. The Nusselt number is higher for finned RMCs than for RMCs without fins. At a heat flow of 100W/cm², the average improvement in the Nusselt number is 51.9%.

5. RMCs with fins have a greater pressure decrease. When the pressure drop penalty is taken into account, however, the Nusselt number improves significantly. A higher pressure drop and heat flux result in an average Nusselt number that is higher. Cylindrical finned RMCs improve the thermo-hydraulic performance factor by an average of 72.87% at 200 W/cm² and 69.43% at 100 W/cm².

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