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Effect of micro Chambers on Thermal Performance of Microchannel

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ABSTRACT: These days, there is a tremendous increase in cooling demand for the electronic devices for the purpose of heat dissipation and improving the efficiency of the devices. This makes the need to use micro channels in electronic field. These micro channels have desirable geometrical properties such as compact in size and of micro scale with less weight. Hence, this work focuses on examining the effect of geometrical parameters on the performance of a micro channel. The obtained result shows that heat transfer rate, coefficient of heat transfer, and Nusselt Number is increased with increase Reynolds number in a range of 80 to 500. Thermal resistance is decreased by 41.74% when compared with the plain channel. It is also observed that the overall heat transfer effect increase by 45.45% compared with the other channel.

KEYWORDS: Thermal behaviour, microchannel heat sink, heat transfer coefficient, Reynolds number, Nusselt number, etc.

I. INTRODUCTION

The technology improvements in scaling the electronic devices in past few years lead to rapid increase in heat dissipation. This paved ways for research to introduce new and efficient ways of cooling in electronic devices for improving their performance and life. The general methods cannot be accounted as efficient which made researchers to find new cooling methods. In this view, the water cooler micro channels considered as potential technique to enhance the performance in recent times. These micro channels possess effective heat and flow characteristics and better surface to volume ratio. This effect was due to the difference in thermal and hydraulic behaviors. Investigations were carried out to analyze the micro channels' performance. There exist different active and passive methods to improve the heat transfer performance of the micro channels

S.W. Chang et al. [1] proposed a new twisted tape pin fin array for improving the passive heat transfer in micro channel flow. Different parameters were evaluated at the Reynolds number in the range of 5000 to 15000. For the correlation analysis among the numerical flow data and aerothermal data, ANSYS Fluent code was used to explore the turbulent channel flow numerically. With the vertical flows produced by every twisted tape, the parameters fanning friction coefficient and average Nusselt number enhanced by about 39.5 to 38.5 and 5.3 to 4.8 times relative to plain channel with which the thermal performance enhanced by about 1.39 to 1.55. The proposed twisted tape pin fin channel was reported to provide a superior degree of heat transfer and thermal performance relative to various geometries.

Lei Chai et al.[2] analyzed the thermo hydraulic performance of a micro channel with triangular ribs positioned on the side walls. A numerical study of effect of triangular ribs on the heat transfer and local laminar fluid flow characteristics with Re 443 was done. The ribs were arranged as offset or aligned. The effect of various parameters on the heat transfer and fluid flow were investigated. Temperature and pressure distributions and velocity contour were investigated to illustrate the mechanism of heat transfer and fluid flow. It was shown that the temperature rise of heat sink reduces considerably and the local heat transfer coefficient drop was significantly prevented whereas the relatively higher friction factor was reported. For the considered geometry and conditions, the temperature of heat sink base fluctuate between 302 K to 324 K. Based on the arrangement and geometry of the ribs, the heat transfer coefficient and pressure drop varied from one rib to other and their wavelength and amplitude. It was concluded that the proposed design can enhance the average Nusselt number by 2.15 times relative to standard micro channels.

Ji-Feng Zhu et al.[3] compared the performance of a wavy micro channel heat sink with wavy bottom and side ribs. At different width ratios of channel to pitch, channel aspect ratios, wavelengths and amplitudes, maximum bottom wall

temperature variation and global thermal resistance were compared for 2 wavy designed with similar channel cross section and pumping power. The results from simulations illustrated that the wavy designs created a symmetric pair of counter rotating Dean vortices in the cross sectional plain which resulted in improved heat sink performance and coolant mixing. The Dean vortices created by left right and up down wavy designs were located near top & bottom walls and side walls respectively. Independent of the considered parameters the up down wavy design illustrated superior performance for smaller wave lengths.

Fang Li et al. [4] analyzed the performance and field synergy of micro channels added with 3 cavities using 3D simulation of conjugate heat transfer model. It was reported that the Nusselt number and field synergy angle have an influence on investigating the improvement in heat transfer in view of spoiler cavities in channel. In addition to this, the field synergy impact in channel was found to be optimum when the cavities were configured in the form of front loose back compact under laminar flow. This could obtain the optimized performance of heat dissipation in available space without pressure drop. The heat transfer intensity was reported to be proportional to number of cavities.

Mushtaq T. Al-Asadi et al. [5] was assessed the pin fin perforations and vortex generator shapes for improving the performance of heat sink. Two models were developed for analyzing the impact of working fluid and geometry on flow and heat characteristics. First model was designed with perforated pin fin heat sink where as second was designed with various vortex generator shapes such as circular, rectangular and triangular. The Re was maintained in the range of 50 - 2300. The models were subjected to constant heat flux in view of micro electronics water and air cooling. Validations were done using COMSOL Multiphysics® software package and reported to be inline. It was found that there was no considerable improvement in heat transfer with water in first model. Whereas the second model showed considerable potential in fighting the problems of heat transfer existing in those electronic and electric devices which are technologically driven towards lower weight/smaller volume.

Aparesht Datta et al. [6] was studied the conjugate heat transfer cooling performance on rectangular channel having trapezoidal cavities on opposite side walls. The impact of ribs placed symmetrically between the cavities was studied with the rib shapes as diamond, forward and backward triangular and rectangular which lead to composite structures. Considerable enhancement in thermal performance and entropy generation was noted by addition of ribs. It was seen that the channel with trapezoidal cavity diamond shaped ribs performed best. It was found that superior thermal performance was seen at Re 328 and back to back cavities of width 0.75 times of that of plain channel and length of 0.0454 times that of total channel length. It was seen that the fluctuations of silicon and copper performances were comparable with copper achieving superior thermal performance of 2.3 relative to 1.9 for silicon. Also the parameters like maintenance, durability and cost need to be considered for choosing the substrate material.

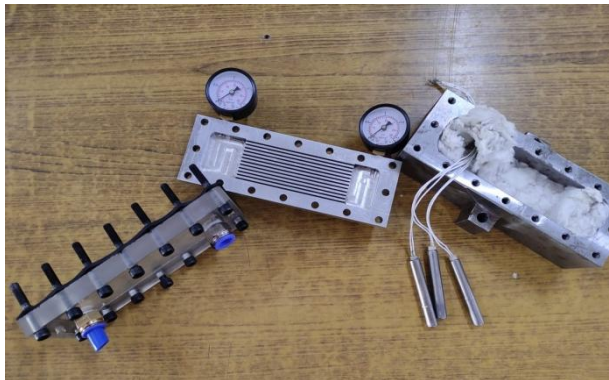
Gaofeng Lu et al. [7] was proposed to combine the dimples and vortex generator on a micro channel for enhancing the heat transfer performance on a cooling surface with a uniform heat flux 100 W/cm². Three different pairs of vortex generators were distributed along flow direction whereas the dimples were distributed downstream of vortex generators with similar intervals. de ionized water was used as working fluid with Re in the range of 167 - 834. It was found that this combination could enhance the performance by about .4% to 59.8% along with 22.1% to 54.4% increased friction factor. The vortices intensity due to vortex generator was stronger than those of dimples. This combination is advantageous for the heat transfer improvement. In addition to this, the vortex generators configured in common flow down was reported to have superior heat transfer than vortex generators configured in common flow up. The superior thermo hydraulic performance was showed by the optimum parameters of attack angle 45 degrees and height ratio 0.6.

Based on the above literature review, it is found that the ribs with cavities are good for enhancements of heat transfer in the exchanger. It is also found that the channel with micro chamber is having better mixing of fluid and leads to improve the heat transfer. In this present work, experiments have been conducted on micro channel to study the effect of micro chamber on thermal performance of channel.

II. EXPERIMENTAL SETUP

A rectangular microchannel heat sink with inline micro chambers is fabricated with entry and exit section by using a spark EDM machine with length, width, and height of 230mm, 80mm, and 15 mm, respectively,. It has consisted of 12 channels with 3 mm (thickness of fin) spacing and a rectangular microchannel with a length, height, and width of 80 mm, 8 mm, and 1 mm, respectively, as shown in figure 1 [6,8,9].

Microchannel is assembled with the following three sections: one adiabatic section at the bottom with thermal wool to reduce the heat losses, one main microchannel plate, and one cover plate with sealing on the top. The above sections are fixed with the Allen socket head cap screw. Cartridge heaters are arranged at the bottom of the channel and kept in the adiabatic section. Fluid Inlet and outlet ports are given at each end of the plate and also thermocouple locations, as shown in the figure.



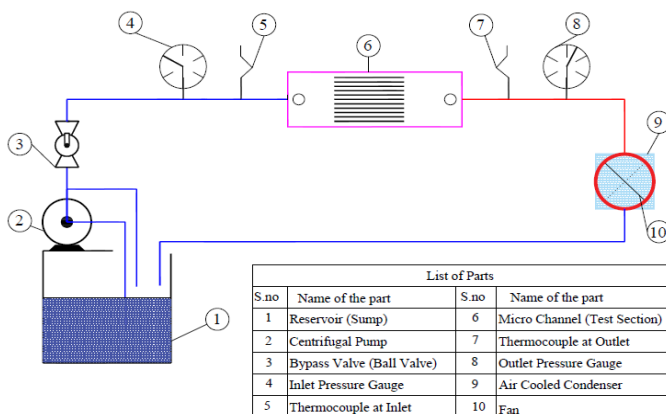
a)



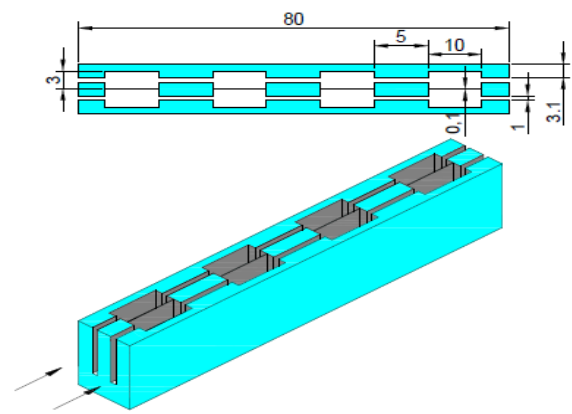
b)

Fig: 1 a) All three sections (adiabatic, main channel, and cover plate) and (b) Experimental setup.

A monoblock 0.5 HP centrifugal pump is used for the circulation of the working fluid in the loop. The control and measurement of fluid flow in the test section are done by a flow meter with a control value. The constant heat is provided to the channel by the three cartridge heaters connected with the voltmeter and ammeter to measure heat supply. A dimmer is used to attain the uniform heat flux condition. Temperature and pressure are measured with K-type Thermocouples and pressure gauges at inlet and outlet ports [10, 11, 12].



a)



b)

Fig: 2 a) Schematic diagram of a setup and b) Micro channel with inline micro chambers

III. METHODOLOGY

The flow rate of working fluid in a loop is set from 0.1lpm to 1lpm. The constant heat flux (70 W/cm^2) is maintained. The inlet temperature of the deionized water is kept at 300K (room temperature). Prior to each test, a steady-state is established for 8 min when the measurement system showed no significant change in the working fluid properties. Consequently, the temperature and pressure are recorded every 5 sec during n intervals of 5min [13]. All experiments are conducted at atmospheric conditions.

IV. DATA ANALYSIS

In the current study, experimental data recorded, such as average temperature, wall temperature, mass flow rate, voltage, and current, are used to calculate the thermal behavior of the microchannel [14, 15, 16].

The heat energy supplied to the channel by the electric heaters is given as

$$Q_{heater} = V \times I \text{ (W)} \quad (1)$$

Here V is voltage, and I is current.

$$Q = mC_p(T_{out} - T_{in}) \text{ (W)} \quad (2)$$

Where m is the rate of mass flow, T_{in} and T_{out} are the inlet and outlet temperatures, respectively, and C_p is the fluid's specific heat.

The convective heat transfer coefficient of a fluid can be calculated by using Newton's law of cooling

$$h = \frac{Q}{(T_w - T_f)A_{ch}} \left(\frac{W}{m^2K} \right) \quad (3)$$

Where, T_w and T_f are the average wall temperature along the channel, average fluid bulk temperature.

$$T_f = \frac{T_{in} + T_{out}}{2} \text{ (}^\circ\text{C)} \quad (4)$$

$$T_w = \frac{T_{wtc1} + T_{wtc2} + T_{wtc3} + T_{wtc4} + T_{wtc5}}{5} \text{ (}^\circ\text{C)} \quad (5)$$

The total heat transfer area of the micro-channel is calculated by the fin analysis method as follows

$$A_{ch} = A_b + \eta_{fin} A_{fin} \text{ (m}^2\text{)} \quad (6)$$

A_b and A_{fin} are the base surface area at the bottom of the channel and area of the fin, respectively. The fin efficiency is denoted by the symbol η_{fin} . The fin condition is assumed as a shortfin adiabatic tip condition and calculated the fin efficiency by using the below expression [17,18,19]

$$\eta = \frac{\tanh(mH_{fin})}{mH_{fin}} \quad (7)$$

Where m is the fin parameter, given by

$$m = \sqrt{\frac{h_{sp}P_{fin}}{k_m A_{fin}}} \quad (8)$$

Where, P_{fin} is the perimeter of the fin, A_{fin} area of the fin.

Nusselt number is calculated by the average heat transfer coefficient as follows.

$$Nu = \frac{hD_h}{k_f} \quad (9)$$

Where k_f is the working fluid thermal conductivity, D_h is the hydraulic diameter of the channel.

Hydraulic diameter is defined as the ratio of four times the channel area to the wetted perimeter and expressed as

$$D_h = \frac{2W_c \times H_c}{W_c + H_c} \text{ (mm)} \quad (10)$$

Here, H_c and W_c are the height, width of the channel

The Reynolds Number is defined as the inertial force ratio to viscous force and is used to determine the channel's type flow. It is expressed by

$$Re = \frac{\rho u D_h}{\mu} \quad (11)$$

Here, ρ , μ are the density and dynamic viscosity of the working fluid, respectively. u is the average velocity of the liquid, which is expressed by

$$u = \frac{V}{NA_c} \left(\frac{m}{s} \right) \quad (12)$$

Here, V is the fluid flow rate, and A_c is the cross-section area of the fluid flow in the channel. N represents the number of channels

Thermophysical properties of the working fluid are taken at bulk mean temperature in the above calculations.

Thermal resistance is the material property, which can resist a heat flow. It is expressed.

$$R_t = \frac{T_{max} - T_{in}}{Q} \left(\frac{K}{W} \right) \quad (13)$$

T_{max} is the maximum temperature in the five thermocouples connected to the channel wall, T_{in} fluid inlet temperature, and the q is the heat extracted by the fluid.

The total pressure drop is measured by the pressure gauges or differential manometer at the channel's inlet and outlet. It includes the pressure drop in the channel, in the manifolds at the inlet and outlet, and the loss of pressure at the channel's contractions and expansions. It is calculated by using the following expression. [20,21]

$$\Delta p_{ch} = \frac{3\mu_f L_{ch} V_{ch}}{b^2 F \left(\frac{a}{b} \right)} \text{ (Pa)} \quad (14)$$

V. EXPERIMENTAL RESULTS

The series of experiments are conducted on micro channels with micro-chamber at constant heat flux conditions for various Reynolds numbers (80 to 500) while keeping the working fluid's inlet temperature at 300K. Figure 2 shows that the effect of heat transfer rate with Reynolds number in a range of 80 to 500. It is observed that the rate of heat transfer increased with Reynolds Number from 142 to 512 W. Heat absorbed increased by the working fluid is due to the mixing of fluid in the micro chambers. It is also depicted that the heat transfer rate is higher in the microchannel than the other channel. Reported average values of Q in channel with micro chambers, microchannel, and plain channels are 352.34, 315.56 and 242.01 Watts, respectively. Figure 3 depicts the variation of heat transfer coefficient with various ranges of Reynolds Numbers. It is seen that the heat transfer coefficient increased with an increase in Reynolds number. It is observed that the variation of heat transfer coefficient is the same as the rate of heat transfer in other microchannels, and the reason is also the same. It is identified that the recorded values of heat transfer coefficients are 19.2, 13, and 10.4 $\text{KW/m}^2\text{K}$. It is also seen that the variation of curve with others at higher Reynolds numbers is due to mixing of fluid and formation of vortices.

Figure 4 shows that the effect of Nusselt number with respect to Reynolds Numbers in both the channels. It is observed that the Nusselt number is increased with Reynolds Number. It is also seen that the trend of the curve of Nusselt number is the same as the heat transfer coefficients curves. It identified that the percentage increase in Nusselt number between the microchannel and plain channel is 45.45%. Figure 5 indicates the effect of thermal resistance with a variation of Reynolds number in the plain and microchannel heat exchanger. It observed that the thermal resistance decreases with increasing Reynolds Number. Also, it is observed that the thermal resistance is very low in a microchannel when compared with the plain channel. The deviation of thermal resistance is observed as 41.74% between micro and plain channels. It is also observed that the thermal resistance depends on the wall temperature and flow rate in channels.

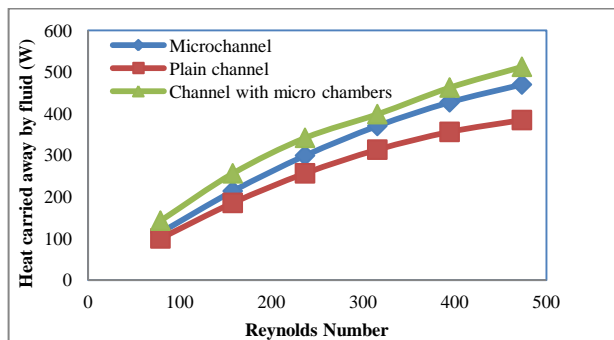


Fig: 3 Heat carried away by working fluid

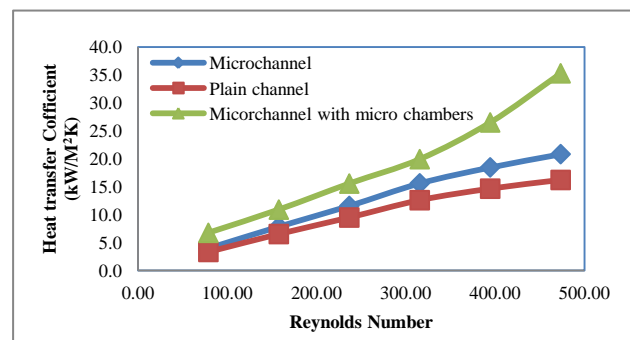


Fig: 4 effect of Heat transfer Coefficient

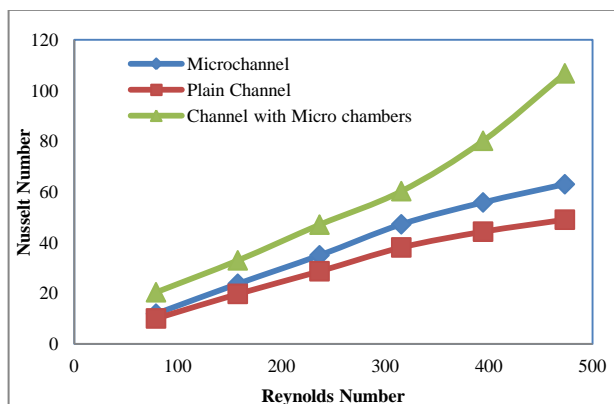


Fig: 5 Effect Nusselt Number

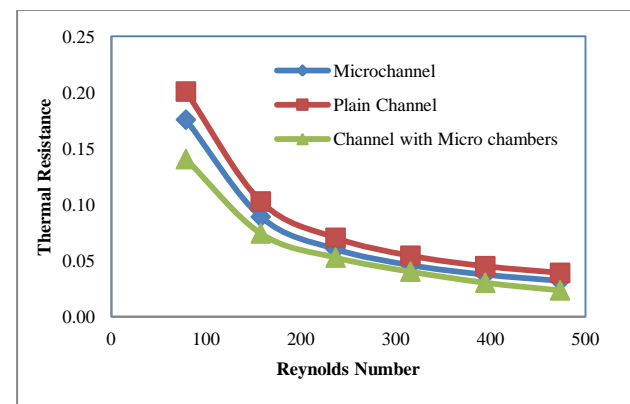


Fig: 6 Variation of Thermal Resistances conclusion

In the present study, experiments are conducted at various ranges of Reynolds number (80 to 500) at constant heat flux. The thermal behaviour of the microchannel is investigated experimentally. From the experiments, the following observations are drawn:

- The heat transfer rate increased with increase Reynolds number linearly in the range of 142 to 512 W. It is due to working fluid film thickness and the mass flow rate of working fluid in the channels.
- It is seen that the heat transfer coefficient increase with increasing Reynolds number. Also observed is that the increase in heat transfer coefficient with micro chambers is 45.45%. It is due to the surface-to-volume ratio and formation of eddies in the chambers.
- Nusselt number is increased with Reynolds number. It is also observed that the trend of the curve of Nusselt Number is the same as heat transfer and heat transfer coefficient curves.
- It is seen that the thermal resistance is decreased by 41.74% with an increase Reynolds number. It is the effect of temperature distribution in the sidewalls of the channel.
- It is concluded that heat is enhanced by 45.54% in the microchannel with micro chambers.

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