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## **Experimental investigation on Thermal Behaviour of Microchannel**

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**ABSTRACT**: The increasing demand for cooling electronic devices for extracting heat and enhancing the reliability of devices leads to micro-channel heat sinks to serve this purpose. These microchannels are compact and micro in size, have a larger surface by volume ratio, and are lightweight. This work is intended to investigate the influence of geometry on the thermal performance of a microchannel. The obtained result shows that heat transfer rate, coefficient of heat transfer, and Nusselt Number increase with increase Reynolds number in a range of 80 to 500. Thermal resistance is decreased by 16.56% when compared with the plain channel. It is also observed that the overall hear transfer effect increase by 19.64% compared with the plane channel.

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

#### I. INTRODUCTION

With the emerging developments in minimizing electronic devices in recent times, heat generation was also increased rapidly. It made the researchers develop new and practical techniques for cooling to enhance the life of appliances. The traditional cooling methods can no longer be considered effective ways now, and there is a need to find new ways. In this contrast, the water cooling microchannels were seen as a potential technology to improve heat transfer in recent years. These microchannels have better heat and flow characteristics, possessing an excellent surface-to-volume ratio. Because of the difference in thermohydraulic behavior, work was conducted to understand the behavior of these microchannels. Different methods exist to improve the heat transfers, which are categorized as active and passive methods. Techniques such as liquid spray, electrodynamics, etc., come under active methods, whereas extended fins, twisted tape, etc., come under passive methods.

Abdullah Al-Sharafi et al. [1] was analyzed the effects of micro post arrays present on the top wall of the channel and the air trapped on the microchannel flow. The analysis was done both numerically and experimentally. The bottom plate of the channel was considered to be smooth moving with uniform velocity. The top plate was supposed to be stationary and arranged with the micro post arrays. The experimentation was carried out for validating the temperature and velocity predictions. The authors were reported that in the top wall vicinity, the flow field was impacted by micro post arrays, which resulted in wavy flow. It was found that the flow circulation of trapped air impact the heat transfer rate. The Nusselt number was reported to be inversely proportional to the pitch size of the array. It was also to the height of the micro-channel. The array was impacted the pressure drop also. The pressure drop was also reported to be inversely proportional to the pitch size up to 12 micrometers.

Zhenfei Feng et al. [2] were studied the effect of geometry such as diameters, height, and spacing of circular pin fin on heat transfer and fluid flow in the interrupted microchannel heat sink. The pros of both circular pin fins and microchamber were combined in the designed microchannel. The effects of geometry on local heat transfer characteristics and fluid flow at Re 398 were analyzed. Also, the variation of the average heat transfer characteristics and fluid flow at Re 398 were analyzed with the help of numerical simulation. The performance was compared with that of the interrupted microchannel with one circular pin fin in every chamber, interrupted microchannel, and smooth microchannel. It was found that the rate of heat transfer improved with double circular pin-fins in every microchamber. It decremented the degradation of local wall temperature and obtained uniform wall temperature. It was reported that the spacing impacts the uniformity of wall temperature, and the pin fin height affects the reduction of pressure drop and improvement of heat transfer. The optimum values of geometry for the channel were,



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at Re 530, 0.55 mm spacing, 0.12 diameters, and 0.08 heights which gives about 42% improvement relative to the smooth channel.

Rajalingam A et al. [3] were investigated the impact of arrangement and shape of microstructures on the thermal and hydraulic performance of a microchannel heat sink. They have introduced micro blind holes and micropillars in the microchannel heat sink to reduce the pumping power, interference wall temperature and improve the heat transfer coefficient. Three computational analyses in Ansys 19.3 were used to optimize these microstructures. Water was used as the working medium in a microchannel of  $0.4 \times 0.4 \times 50$  mm<sup>3</sup> volume. The analyses selected a blind hole, selecting an optimum position and selecting a pillar with an optimized cross-section. The overall thermal performance, temperature gradient, interface temperature, pressure drop, and heat transfer coefficient of all considered cases were compared with those of simple plain channel maintaining the Re in the range of 500 - 1000. It was found that the performance of the channel with circular blind holes was about 5% superior to the plane channel, being highest among all. The heat transfer coefficient was about 1.6 to 1.7 times of the plain microchannel with the advised arrangement. And also, about 12.4% - 16.5% improvement in thermohydraulic performance was seen. The superior performance of about 15% - 19% was seen for the microchannel with elliptical pillars.

Fang Zhou et al. [4] investigated numerically and experimentally the improvement of heat transfer of a microchannel heat exchanger with various micropillars. Numerical simulation was used to obtain the velocity, temperature, and vortices contour distribution. The overall thermal performance was compared in a range of Reynolds numbers. It was seen that the micropillars of square fan-shaped and circular shapes, the ratios Nu/Nu<sub>0</sub> for channels between 1.65 to 1.9, and drop-shaped and irregular drop-shaped pillars were found between 1.6 to 1.7. The last pillars indicated an increased pressure loss of 20% to 40%, whereas the former pillars indicated more than 60%. The drop-shaped and irregular drop-shaped pillars shown superior overall thermal performance, which means excellent heat transfer performance.

Yuting Jia et al. [5] developed a 3D model of a combined microchannel with cone-shaped micro pin fins to numerically investigate the fluid flow and heat transfer characteristics. The inlet Reynolds number was maintained in the range of 147-637. The initial study was on the microchannels with micro pins arranged in upstream, downstream, and middle and distributed throughout the channel. Then the effecting parameters such as relative fin height, diameter, and spacing were studied. It was shown that the microchannel with cone-shaped micro pin fin leads to enhanced heat transfer with a more significant pressure drop relative to the smooth channel. The microchannel with uniformly distributed fins indicated a high overall performance factor and low bottom wall temperature. When the Reynolds number was less than 500, the overall thermal performance of the downstream case was found to be better than that of the upstream case. On the other hand, when the Reynolds number was more significant than 500, a contrary result was seen.

Lei Chai et al. [6] were analyzed the performance of a microchannel heat sink with triangular ribs arranged on sidewalls. The impact of arrangement and geometry of ribs on the thermohydraulic performance was studied with varying the Nusselt number and average friction factor with a range of Reynolds number 187-715. The microchannels used in this study had a similar width and depth of 0.1 mm and 0.2 mm in a uniform cross-section region. The geometric variables of offset <sub>or</sub> aligned triangular ribs were in the range of 0.025 mm to 0.4 mm for width, height, spacing, and width ratio of converging region to single rib were 0.005 mm to 0.025 mm, 0.2 mm to 5 mm, and 0 to 1 mm respectively. Depending on all the cases, the correlations were proposed for Nusselt number and average friction factor. In the considered range of Reynolds number and geometry, the channel with aligned triangular ribs was shown 1.03 to 2.01 and 1.06 to 9.09 times most significant average Nusselt number and friction factor respectively and with offset arrangement, 1.01 to 2.16 and 1.04 to 7.43 times greater average Nusselt number and average friction factor respectively relative to the straight channel heat sink.

The above literature review found that there is a need to study the effect of the geometry of a channel on thermalhydraulic behaviour. A series of experiments have been repeated on a microchannel at various Reynolds numbers in the present work.

#### II. EXPERIMENTAL SETUP

An aluminium rectangular microchannel heat sink is fabricated with length, width, and height of 230mm, 80mm, and 15 mm, respectively, with entry and exit section by using a spark EDM machine. 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].



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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.



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].

#### III. METHODOLOGY

The flow rate of working fluid in a loop is set from 0.11pm to 11pm. The constant heat flux  $(70 \text{ W/cm}^2)$  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. EXPERIMENTAL RESULTS**

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].

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

$$Q_{heater} = (1)$$

$$V \times I(W)$$
Here V is voltage, and I is current.
$$Q = mC_P(T_{out} - T_{in}) \quad (W)$$
(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

$$h = \frac{Q}{(T_w - T_f)A_{Ch}} \left(\frac{w}{m^2 K}\right)$$
(3)  
Where,  $T_w$  and  $T_f$  are the average wall temperature along the channel, average fluid bulk temperature.  

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

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

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



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

(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  $\eta_{fin}$ . The fin condition is assumed as a shortfin adiabatic tip condition and calculated the fin efficiency by using the below expression

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

Where m is the fin parameter, given by

m

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

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

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

$$Nu = \frac{hD_h}{k_c}$$
(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

$$D_h = \frac{2W_C \times H_C}{W_C + H_C} \quad (mm)$$
(10)

Here, Hc and Wc 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

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

Here,  $\rho$ ,  $\mu$  are the density and dynamic viscosity of the working fluid, respectively. u is the average velocity of the liquid, which is expressed by

$$u = \frac{V}{NA_c} \quad \left(\frac{m}{s}\right)$$
(12)
Here V is the fluid flow rate and A is the gross section area of the fluid flow in the sharped. N represents the number

Here, V is the fluid flow rate, and A<sub>c</sub> 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.

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

 $T_{max}$  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.

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

#### V. EXPERIMENTAL RESULTS

The series of experiments are conducted on both plain and microchannels 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 ranges of 80 to 500. It is observed that the rate of heat transfer increased with Reynolds Number from 114 to 470 W. The increase of heat absorbed by the working fluid is due to the increase in the mass flow rate of liquid in a microchannel, and the film thickness of the fluid in the channel is minimal. It is also depicted that the heat transfer rate is higher in the microchannel than the plain channel. Reported average values of Q are 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 is the same as the rate of heat transfer in the



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microchannel and the plain channel, and the reason is also the same. It is identified that the recorded values of heat transfer coefficients are 20.8 and 16.7  $KW/m^2 K$ .

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 19.64%. Figure 5 indicates the effect of thermal resistance with a variation of Reynolds number in the plane 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 plane channel. The deviation of thermal resistance is observed as 16.56% between micro and plain channels. It is also observed that the thermal resistance depends on the wall temperature and flow rate in channels.



Fig:4 Nusselt Number Vs. Reynolds Number

Fig: 5 Thermal Resistance Vs. Reynolds Number

#### VI. CONCLUSION

In the present study, the microchannel heat exchanger is fabricated on an aluminum block and conducted experiments for various ranges of Reynolds number (80 to 500) at constant heat flux. The thermal behavior 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 114 to 470 W. It is due to working fluid film thickness and the mass flow rate of working fluid in the channels.
- It is seen that there is linear variation in the heat transfer coefficient with an increase in Reynolds number. Also observed is that the increase in heat transfer coefficient between the plain and microchannel is 19.64%. It is due to the surface-to-volume ratio.
- Nusselt number is increased by 19.64% of a microchannel compared with the plain channel. 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 16.56% with an increase Reynolds number. It is the effect of temperature distribution in the sidewalls of the channel.



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• It is concluded that heat is an enhanced by 30% in the microchannel than in the plane channel. It is suggested that the microchannels can be used in miniaturization applications.

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