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Experimental Investigation Of Fluid Flow And Heat Transfer In Internally Finned Minichannels

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ABSTRACT

The increase in the circuit density is driving advanced cooling systems for the next generation microprocessors. Minichannels heat sink is a method that receives considerable attention. Experimental investigation of laminar flow and heat transfer characteristics and behaviour of circular shaped smooth minichannels heat sink is compared with internally finned minichannels. Reynolds number ranged from 100 to 1300 at the heat flux of 50 W is used for experimental investigation. The Minichannels heat sink performance is evaluated in terms of heat transfer coefficient, pressure drop, friction factor, thermal resistance and pumping power. The result reveals that when there is an increase in the contact surface area of the flowing fluid the convective heat transfer coefficient is also increased while the thermal resistance is decreased.

Keywords

Minichannel, Internally finned, Electronic cooling, and Liquid cooling.

1. INTRODUCTION

The growing demand for product miniaturization in all industrial sectors, coupled with global competition for more reliable, faster, and cost-effective products, has led to many new challenges for design and operation of thermal management systems. The rapid increase in the number of transistors on microchips, with increased functionality/power and consequently higher heat fluxes, is one such great challenge in the electronics packaging industry. Minichannel heat and mass exchanger technologies are finding new applications in diverse industries and emerging as a promising solution to game changing technologies in the way design and operate next-generation, high performance thermal management systems

1.1 Minichannel

From Newton's law of cooling, it is know that for a fixed temperature difference, heat flux depends on the product of hA, where h is the heat transfer coefficient and A is the heat transfer surface area. So, in order to fulfill the requirement of high heat flux removal the product of hA has to be increased, and since the heat transfer coefficient h is related to the hydraulic diameter, increasing surface area is one option. The heat transfer area can be increased by using minichannels on the heat generating body (chip surface). The flow behaviour of water inside the channels is determined by the channel hydraulic diameter and the channel cross-sectional area. To obtain high heat transfer, a smaller hydraulic diameter and a larger heat transfer area of the channel are preferred, so a number of narrow channels with high depth are suitable. Smaller hydraulic diameter and larger cross-sectional area increase the pressure drop and consequently require higher pumping power. On the other hand, increased cross-sectional area of the heating surface enhances the heat transfer rate. These requirements can be adjusted with next-generation minichannels that will have larger hydraulic diameter but provide larger cross-sectional area as well as high heat transfer coefficient.

Tuckerman and Pease [1] first made use of miniaturization for the purposes of heat removal, within the scope of a Ph.D. study in 1981. Their publication titled "High Performance Heat Sinking for VLSI" is credited as the first study on microchannel heat transfer. Their pioneering work has motivated many researchers to focus on the topic and microchannel flow has been recognized as a high performance heat removal tool ever since. Before proceeding with minichannel flow and heat transfer, it is appropriate to introduce a definition for the term "minichannel". Kandlikar [7] adopted a different classification based on the rarefaction effect of gases in various ranges of channel dimensions, "D" being the smallest channel dimension:

$1 \mu m < D \leq 10 \mu m$: Transitional Microchannels

$10 \mu m < D \leq 200\ \mu m$: Microchannels

$200\ \mu m < D \leq 3\ mm$: Minichannels

$3\ mm < D$: Conventional Passages

A simpler classification was proposed by Obot based on the hydraulic diameter rather than the smallest channel dimension. Obot classified channels of hydraulic diameter under 1 mm ($D_h < 1\ mm$) as microchannels, which was also adopted by many other researchers such as Bahrami et al. and Bayraktar and Pidugu. This definition is considered to be more appropriate for the purposes of this project.

A Minichannel heat sinks typically contains a large number of parallel minichannels. Coolant is forced to pass through these channels to carry away heat from a hot surface. In Minichannel heat exchangers flow is typically laminar and heat transfer coefficients are proportional to velocity. It provide very high surface area to volume ratio, large convective heat transfer coefficient, small mass and volume, and small coolant inventory. These attributes render these heat sinks very suitable for cooling devices such as high performance microprocessors, laser diode arrays, radars, and high-energy-laser mirrors. Minichannel heat exchangers could be easier to repair than their conventional counterparts. It offers other benefits, including increased latent capacity for minichannel evaporators. Minichannel heat exchangers improve heat transfer in two ways. First, the smaller dimensions of the refrigerant flow passages increase refrigerant-side heat transfer. Second, the flat tube orientation reduces airside flow resistance, leading to either increased airflow or reduced fan power.

1.2 Objectives of the work

As evident from the diversity of application areas, the study of fluid flow and heat transfer in minichannels is very important for the technology of today and the near future as developments of the following trend of miniaturization in all fields. The present study deals with experimental investigation and 3D numerical simulations of laminar flow and heat transfer characteristics and behaviour of circular shaped smooth minichannels heat sink is compared with internally finned minichannels. Reynolds number ranged from 100 to 1000 at the total heat flux of 50 W is used for experimental investigation. Results of interests such as temperature distribution, heat transfer coefficient and pressure drop, friction factor and thermal resistance as function of pumping power are reported to illustrate the effects of circular shaped smooth minichannels and internally finned minichannels performance.

2. EXPERIMENTAL INVESTIGATION

2.1 Minichannel heatsink geometric configuration

The advancement of various methods in the micro-fabrication technology has eased the manufacture of minichannel heat sinks. The present test samples have been machined using wire electro-discharge machining (EDM) along with CNC milling. Wire EDM technique is appropriate for machining where straight internally finned channels are to be cut. Smooth circular channels have been machined using CNC milling. Table-1 gives the dimensional details and hydraulic diameter of these test samples. The hydraulic diameters have been estimated based on wetted perimeter. Machined surfaces of smooth circular channels and internally finned channels geometry used in this study are shown in Table-1. As per the nomenclature the smooth circular channels and internally finned channels to be tested belong to minichannel category.

Table 1. Dimensions of the test specimens

TEST SAMPLE	DIMENSIONS	MATERIAL
CIRCULAR SMOOTH CHANNELS	length = 37.5 mm width = 37.5 mm height = 5 mm channel dia. = 2.5 mm No. of channels = 7	Aluminium 6063
INTERNALLY FINNED CHANNELS	length = 37.5 mm width = 37.5 mm height = 5 mm fin width = 0.3 mm fin height = 0.74 mm No. of fins = 6 No. of channels = 7	Aluminium 6063

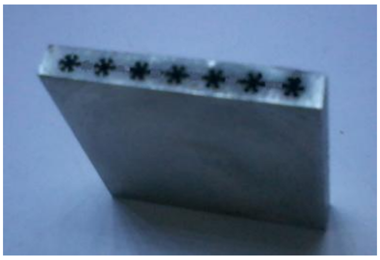

Fig 1: Pictorial view of internally finned channels heat sink

2.2 Experimental setup

The schematic and pictorial view of the experimental setup is shown in Fig.2. The heat sink experimental setup assembly consists of submersible pump which helps to pump the flowing fluid water throughout the circuit. Rotameter is used to control and measure the volumetric flow rate of the flowing fluid. U tube manometer will monitor the pressure difference between the inlet and outlet plenum of flowing fluid in the specimen. The change in pressure can be calculated by using the formula $\Delta P = gh\ (\rho_{hg} - \rho_w)$ where, ΔP is the pressure difference in N/m^2, g is gravity in m/s^2, h is height of mercury above datum line, ρ_{hg} is density of mercury, ρ_w is density of water.

The strip heater of 60W is employed to heat the test specimen. Temperature occurs in the inlet, outlet and base of the test specimen will be monitored by the temperature indicator with the help of thermocouple sensors which placed in inlet, outlet and at the base of the test specimen.

The fluid is pumped and it is supplied to the test specimen through rotameter. The volume flow rate of the fluid will be measured and controlled by the rotameter and the excess fluid will be returned to the reservoir. The height of mercury is noted in the U tube manometer for pressure difference calculation. The height of mercury level will vary based on the volume flow rate of the fluid. Then the strip heater is switched on to maintain the temperature of 60°C at base of the test specimen and its temperature (T_b) will be monitored by temperature indicator. The steady flow of inlet and outlet temperature will be noted by varying the channel of temperature indicator of 1&2.

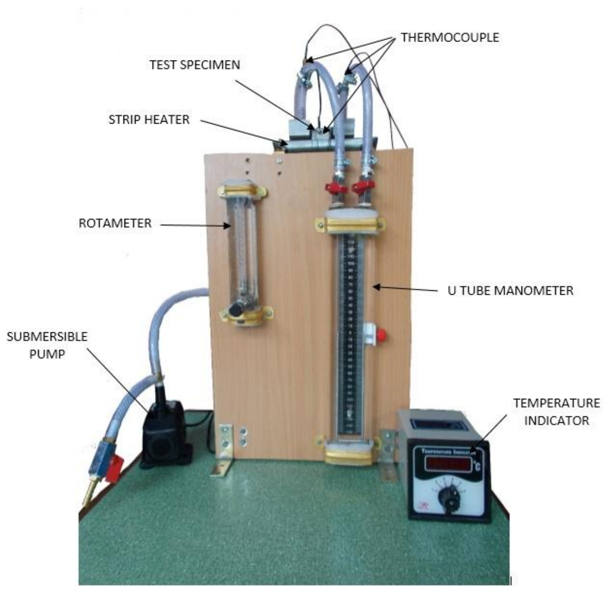

Fig 2: Pictorial view of experimental test set-up for heat sink

The experimental procedure is repeated to the range of 0.05, 0.1, 0.15, 0.2, 0.25 and 0.3 litres per minute and the Reynolds number range of 100 to 1400 is obtained. The conversion of volumetric flow rate from liter per minute to velocity is calculated by using the formula.

$$1 \text{ Lpm} = \frac{1}{(1000 \ X \ 60)} \ m^3/s$$

Volume flow rate = total area of channels x area of channels

$$\text{Area of single channels} = \frac{\pi \ X \text{ hydraulic diameter}}{4} \ m^2$$

$$\text{Velocity} = \frac{\text{volume flow rate}}{\text{total area of channels}} \ m/s$$

Hydraulic diameter was calculated by using the formula

$$D_h = \frac{4A_c}{P}$$

where, A_c is area of single channel, P is perimeter of channel.

For Reynolds number

$$Re = \frac{\rho \ u \ D_h}{\mu}$$

where, ρ is the density of fluid, μ is the dynamic viscosity of fluid, u is the velocity of fluid, D_h is hydraulic diameter of fluid.

From the experimental data, circular smooth channel is compared with internally finned channels. The well-known fact is that by increasing the contact surface area the convective heat transfer coefficient will also be increased.

3. RESULT AND DISCUSSION

3.1 Pressure drop

The Pressure difference is calculated by using the following formula.

Left limb of manometer = Right limb of manometer

Left limb of manometer = $P_1 + \rho_w gh$

Right limb of manometer = $P_2 + \rho_{hg} gh$

$$P_1 + \rho_w gh = P2 + \rho_{hg} gh$$

$$P_1 - P_2 = \rho_{hg} gh - \rho_w gh$$

$$\Delta P = gh\,(\rho_{hg} - \rho_w)$$

Where, ΔP is pressure difference, h is height of mercury above datum line, ρ_{hg} is density of mercury and ρ_w is density of water.

The pressure distribution of the circular smooth channel and internally finned channels versus Reynolds number is shown in Chart-1.

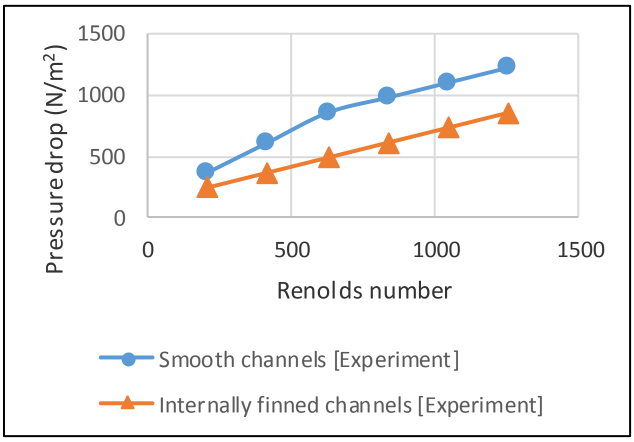

Fig 3: Pressure drop versus Reynolds number

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From Fig.3, it can be seen that from the experiment, the pressure drop rises linearly with the increase of Reynolds number for both smooth and finned channels. This may be considered as one of the advantages of using internally finned channels as heat sink for low Reynolds number range to improve the thermal management systems in many engineering applications. From Chart-1 it is found that compared to smooth channels, internally finned channels has a low pressure drop. Due to that, less pumping power is sufficient for internally finned channels when compared to smooth channels.

Table 2. Pressure drop in smooth and internally finned channels

RENOLDS NUMBER	PRESSURE DROP	
	CIRCULAR SMOOTH CHANNELS (N/m^2)	INTERNALLY FINNED CHANNELS (N/m^2)
210.15	367.875	245.25
420.29	613.125	367.875
630.44	858.375	490.5
840.59	981	613.125
1050.74	1103.625	735.75
1260.88	1226.25	858.375

3.2 Friction factor

The study of laminar flow in rough microchannels has become important and the effect of friction factor is computed using:

Darcy friction factor (f):

$$f = \frac{D_h}{l}\left[\frac{2\Delta P}{\rho v^2} - K_{entry} - K_{exit}\right]$$

$$K_{entry} = 0.8+0.04\sigma-0.44\sigma^2$$

$$K_{exit} = 1-2.4\sigma+\sigma^2$$

where,

l = length of heat sink (m)

v = Fluid velocity (m/s)

ΔP = Pressure drop across the heat sink (Pa)

σ = Ratio of flow area to frontal area

D_h = Hydraulic diameter

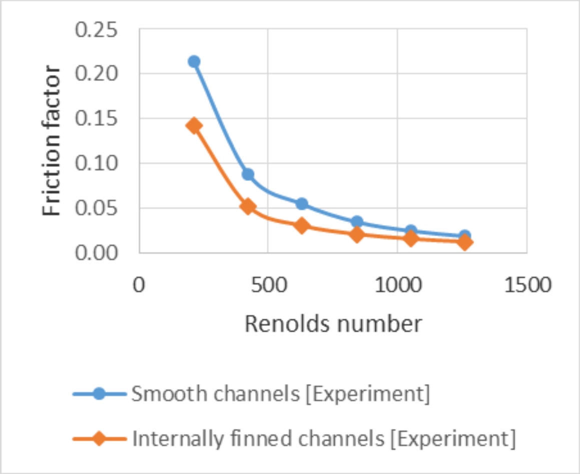

Fig 4: Friction factor versus Reynolds number

The variations of the friction factor with Reynolds number for various particle volume fractions are shown in Fig.4. The results show that the friction factor is similar for both channels where it decreases with the increase of Reynolds number. The smooth minichannels heat sink appears to give a slight rise in the friction factor. Therefore, the friction factor decreases in internally finned channels compared to smooth cahnnels.

Table 3. Friction factor of smooth and internally finned channels

RENOLDS NUMBER	FRICTION FACTOR	
	CIRCULAR SMOOTH CHANNELS (N/m^2)	INTERNALLY FINNED CHANNELS (N/m^2)
210.15	0.2144	0.1426
420.29	0.0888	0.0529
630.44	0.0549	0.0309
840.59	0.0349	0.0215
1050.74	0.0249	0.0162
1260.88	0.0190	0.0130

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3.3 Heat transfer coefficient

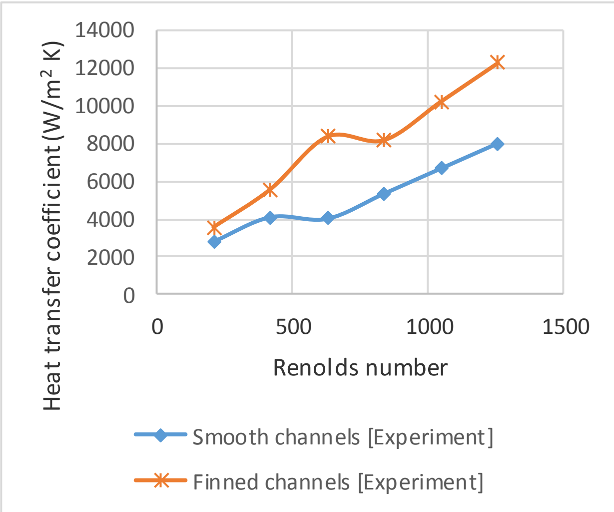

Fig 5: Heat transfer coefficient versus Reynolds number

Table 4. Heat transfer coefficient of smooth and internally finned channels

RENOLDS NUMBER	HEAT TRANSFER COEFFICIENT	
	SMOOTH CHANNELS	FINNED CHANNELS
210.15	2787.57	3563.83
420.29	4091.63	5575.13
630.44	4006.92	8362.70
840.59	5342.57	8183.27
1050.74	6678.21	10229.09
1260.88	8013.85	12274.90

Fig.5 shows the heat transfer coefficient versus Reynolds number of experimental value of smooth and internally finned channels. Therefore, it is clear that the internally finned channel has high heat transfer coefficient than circular smooth minichannels. So this high convective heat transfer coefficient is caused due to the increase in contact surface area of the flowing fluid in channels.

<u>Heat transfer coefficient (h_t):</u>

$$h_t = \frac{q_w}{A_s\,\Delta T_{LMTD}}$$

where,

q_w = *Heat taken by water (W)*

A_s = *Heat sink surface area (m^2)*

ΔT_{LMTD} = *Log Mean Temperature Difference (^oC)*

<u>Heat taken by water (q_w):</u>

$$q_w = \dot{m}\, C_p\, (T_{w,out} - T_{w,in})$$

where,

$\dot{m}$ = *Mass flow rate (Kg/s)*

C_p = *Specific heat of the fluid (J/Kg k)*

$T_{w,out}$ = *Outlet fluid temperature (^oC)*

$T_{w,in}$ = *Inlet fluid temperature (^oC)*

<u>Log Mean Temperature Difference (ΔT_{LMTD}):</u>

$$\Delta T_{LMTD} = \frac{(T_b - T_{w\,in}) - (T_b - T_{w\,out})}{\ln\left[\frac{T_b - T_{w\,in}}{T_b - T_{w\,out}}\right]}$$

where,

T_b = *Temperature of heat sink wall (^oC)*

3.4 Thermal resistances

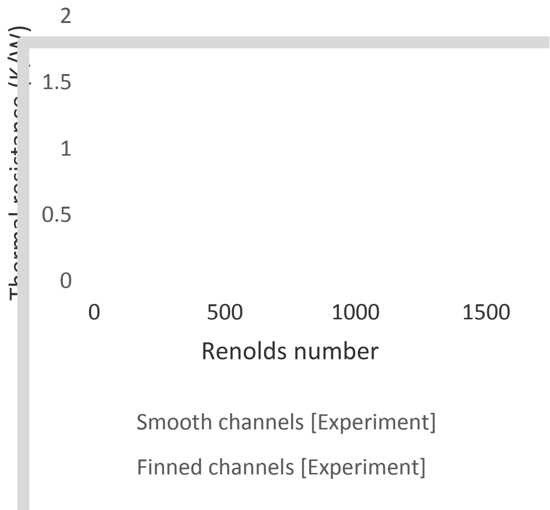

Fig 6:Thermal resistance versus Reynolds number

The cooling performance of Minichannels is assessed by exploring the results for the thermal resistance (R_{th}) based on inlet temperature ($T_{w,in}$), outlet temperature ($T_{w,out}$), base temperature (T_b) and heat taken by water (q_w).

<u>Thermal Resistance (R_{th}):</u>

$$R_{th} = \frac{\Delta T_{LMTD}}{q_w}$$

Thermal resistance versus Reynolds number is shown in Chart-4. From this chart, thermal resistance is lower for internally finned channels compared to smooth channels.

Table 5. Thermal resistance of smooth and internally finned channels

RENOLDS NUMBER	THERMAL RESISTANCE	
	SMOOTH CHANNELS	FINNED CHANNELS

210.15	1.72	1.34
420.29	1.17	0.86
630.44	1.19	0.57
840.59	0.90	0.58
1050.74	0.72	0.47
1260.88	0.60	0.39

3.5 Pumping power

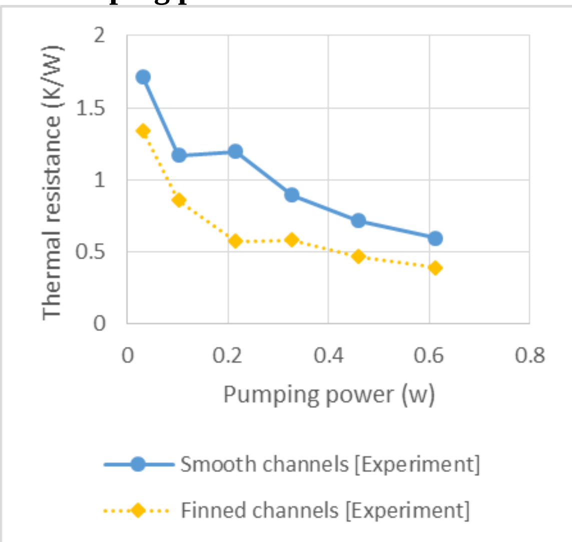

Fig 7:Thermal resistance of heat sink as function of pumping power

Important parameter in the Minichannles heat sink operation is the pressure drop which relates to the coolant pumping power required. The pumping power is required to drive the coolant in Minichannles heat sink operation. It is the product of the pressure drop across the heat sink, ΔP and volume flow rate, Q. Pumping power = QΔP. Where, Q is volume flow rate in m^3/s.

The thermal resistance of minichannles heat sink versus the pumping power is depicted in Fig.7. This great improvement in R_{th} is mainly due to the increase in the thermal conductivity and decrease in the temperature difference between inlet and outlet temperature. The additional reduction in R_{th} is also due to the thermal dispersion of increase in the contact surface area.

4. CONCLUSION

The experimental investigation on laminar flow and heat transfer characteristics in circular smooth minichannels and internally finned minichannels heat sinks are reported in this work. The effects of increase in the contact surface area by using internal fins in minichannels heat are comprehensively studied. Based on the presented results, the following conclusions can be drawn:

- The pressure drop increases with the increase of Reynolds number and compared to smooth channels, internally finned channels has low pressure drop. Due to that, less pumping power is sufficient for internally finned channels when compared to smooth channels.
- The friction factor is similar for both channels where it decreases with the increase of Reynolds number. The smooth minichannels heat sink appears to give a slight rise in the friction factor. Therefore, the friction factor decreases in the internally finned channels when compared to circular channels.
- The heat transfer coefficient for internally finned channels is higher than the circular smooth channels. So, this high convective heat transfer coefficient is caused due to the increase in contact surface area of the flowing fluid.
- The thermal resistance is lower for internally finned channels when compared to smooth channels.

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