U.P.B. Sci. Bull., Series D, Vol. 83, Iss. 2, 2021 ISSN 1454-2358

PERFORMANCE OF A SEMICIRCULAR MICROCHANNEL HEAT SINK IN COOLING OF ELECTRONIC DEVICES

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This paper intends to investigate experimentally the execution of a semicircular microchannel heat sink (MCHS) at a fixed power supply of 72W and mass flow rate (ṁ) varied from 0.006 kg/s to 0.028 kg/s. The working fluid for investigation is deionized water at three temperatures 28.6°C, 23.5°C and 19°C. Experimental results are plotted against the mass flow rate. The MCHS consists of 13 semicircular microchannels of diameter 350µm. Maximum surface temperature of MCHS is found to be 59.7°C at inlet fluid temperature of 28.6°C with a maximum increment of 15.32% which is far below the maximum safe operating temperature of electronic chips. Heat transfer coefficient increases (h) with mass flow rate at a maximum increment of 10.38% at mass flow rate of 0.016 kg/s for the inlet fluid temperature of 19°C due to phase change. Pressure drop rises with the flow rate as viscosity increases while friction factor decreases with mass flow rate of fluid. Hence it may be concluded that semicircular microchannels shows potential for its application in cooling of electronic chips.

Keywords: Semicircular microchannel, heat transfer coefficient, surface temperature, mass flow rate, heat input.

1. Introduction

Miniaturization of electronic devices increases challenge of thermal management because of increased waste heat generation from the electronic devices. The power density in computer systems are projected to be 4.5 MW/m^2 by 2026 [1] and reported to increase beyond 10 MW/m^2 in defense applications [2]. So it is important to blow off the waste heat released from electronic appliances to keep surface temperature of devices within the safe limit and ensure the efficient performance of electronic devices. Generally surface temperature of electronic devices is required to be kept within the range of 85-125℃ [1]. Hence the need for effective cooling technology has been increased. MCHS as an effective tool to dissipate the excess heat from electronic devices was initially studied by Tuckerman and Pease [3]. MCHS can efficiently dissipate heat fluxes

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up to 10^3 W/cm^2 [4]. A channel is a microchannel if its hydraulic diameter (D_h) is less than or equal to 1mm. These microchannels act as a passage for different working fluids which provides the combination of high surface area and heat carrying capability of working fluid to reduce the waste heat. Hence there is necessity to understand different factors affecting the performance of MCHS to enable its successful implementation into real-life applications. MCHS may be either single phase or two phase. Two phase heat sink is most suited for removing excess heat within confined space. Two phase flow occurs when fluid reaches its boiling point while flowing through microchannels. Tong et al. [5] observed that pressure drop curves were affected by mass flux (G), diameter of the tube, and the ratio of length and diameter Mechanism of heat transfer is not same for subcooled and saturated flow conditions due to presence of void fractions. Subcooled flow causes frequent phase change due to bubble formation at the surface of the microchannel. Saturated flow is controlled by either nucleate boiling or forced convection boiling. M. Mirmanto [6] showed that transfer of heat prevails in nucleate boiling where as vapour quality diminishes the local value of heat transfer coefficient (h) after performing experiments in a rectangular microchannel at a range of heat flux (q = 171 to 685 kW/m^2) and (G = 200 to 700 kg/m^{2}s) and with an inlet temperature of fluid (t_{in}) of 98°C. T. Dong et al. [7] investigated that mean value of (h) depends greatly on G and q when R141b was used as a coolant in rectangular MCHS for G ranging from (400 to 980 kg/m^{2}s) and q from (40 to 700 kW/m^2). S. Saisorn et al. [8] observed that the air-water flow improved the heat transfer up to 80% in comparison when only liquid flows in MCHS after conducting experiments at heat supply (Q = 80W). Z. Belkacemi et al. [9] proposed a numerical model to evaluate the impact of surface roughness on hydrodynamic properties for laminar flow in MCHS. It has been observed that there is a linear increment in Poiseuille number with Reynolds number because recirculation arises in fluid. A. A. Alfarjyat et al. [10] investigated rhombus microchannel heat sink applying Al_2O_3 nanoparticles in four different base fluids (deionized water, glycerin, engine oil, and ethylene glycol) at concentration level of 4% and 25 mm diameter. It has been inferred that Al_2O_3 with water has the minimum temperature and thermal resistance while having maximum heat transfer competed to other base fluids at (500 kW/m^2 and 290 K). Qu and Mudawar [11] investigated that the characteristics of heat transfer can be effectively predicted by Navier-Stokes and energy equations. Qu and Mudawar [12] computed the critical heat flux (CHF) in a water cooled MCHS consists of 21 parallel, 215 μm wide and 821 μm deep channels over a range of (G = 86 to 368 kg/m^{2}s), (t_{in} = 30^{0}C and 60^{0}C) and at an outlet pressure (p_{out} = 1.13 bar). It has been inferred that when the flow approaches CHF, then impulsiveness in flow occurs which causes vapour backflow towards the heat sink upstream plenum and this CHF does not depend upon the inlet temperature of fluid but it increases with increase in G. F.J.D et al. [13] found that the quantity (h) depends on (G) for a fixed average vapour quality when R134a used as a fluid in MCHS. Y. Peles et al. and M. Liu et al. [14-15] observed that micro pin fin has very low thermal resistance when competed with conventional MCHS which causes higher heat transfer in micro pin fin heat sink. C. J. Kroeker et al. [16] concluded that circular channels dissipate excess heat in comparison to rectangular micro-channels while having lower thermal resistance than rectangular channels. Prajapati et al. [17] observed that segmented channels have higher value of (h) in comparison to rectangular channels and diverging channels after performing experiments for deionized water at a range of (G = 100–350 kg/m^{2}s) and (q = 10–350 kW/m^2), respectively. Zhang et al. and Hung et al. [18-19] concluded that porous copper heat sink gives higher heat transfer than straight channel. Alfaryjat et al. [20] examined numerically using finite volume method on three different channels (hexagonal, circular, and rhombus) and concluded that hexagonal channels with lesser hydraulic diameter has the maximum value of (ΔP and h) compared to other two shapes. Deng et al. [21] observed that reentrant microchannels (REEM) have higher heat transfer in cases of large inlet subcooling and at medium to higher heat fluxes in comparison to typical rectangular microchannels (RECM) at same hydraulic diameter. A. Sakanova et al. [22] observed that heat transfers in wavy channels are superior to conventional straight channels if purest form of water flows between the values of (0.152 L/min to 0.354 L/min). Kuppusamy et al. [23] revealed that the performance of triangular shaped micro mixer (MTM) depends on the variation of all geometric parameters and overall improvement in performance of MTM is found to be 1.53 times of the simple straight channels while performing numerical solution. Vafai and Zhu [24] suggested a double layer MCHS with counter flow arrangement and found that increase of temperature on the base was reduced when compared to single layer heat sink. Balasubramanian et al. [25] observed that Expanding microchannels are superior to the straight microchannels at same operating conditions because of enhanced flow boiling stability. Mohammed et al. has carried out numerical simulations and applied the finite volume method (FVM) to zigzag, curvy, and step microchannels and compared with straight and wavy channels. Results showed that the zigzag channels have maximum heat transfer coefficient and least surface temperature among various channel shapes. From the above literatures It has been inferred that the performance of MCHS can be enhanced by varying channel shape, size and working fluid flowed also heat transfer is greatly influenced by varying the operating parameters such as (G), (q), (t_{in}) etc. As seen from above literatures and according to my knowledge any experimental investigation had not been informed in the literature to investigate the performance of semicircular microchannel heat sink at a fixed power supply. So, experiments have been performed in this paper to investigate heat transfer of semicircular microchannel heat sink at a fixed power supply of 72 W.

2. Methodology

The experimental set up of MCHS consists of thirteen semicircular microchannels machined on a rectangular copper block insulated well to avoid the leakage. The rectangular copper block comprises of two parts. First part shown in Fig. 1. Integrated with disc heater of 80W at bottom and thirteen semicircular microchannels of diameter 350 µm are machined by wire EDM at the top surface. Second part comprises of inlet and outlet plenum for working fluid flow and four Pt100 thermocouples are inserted at the top of second part to measure the value of (t_{in} and t_{out}) of microchannel. First part and second part of rectangular copper block are clamped together to form a test section. Thermocouples and heater are attached to the control panel as shown in Fig. 2. to display the temperatures and control the heat supply to the test section. Control panel consists of temperature indicator, variac, power meter, heater on and off switch, two thermocouples. One thermocouple is used for the surface temperature measurement of microchannel heat sink and other thermocouple is used to sense the maximum surface temperature for auto cut of disc heater.

2.1 Test module

The test set up of microchannel heat sink consists of following parts:

- Rectangular copper block with semicircular microchannels:

Thirteen microchannels of diameter (d) (350 µm), width (W_{ch}) (350 µm), and Length (L_{ch}) (65.96 mm) as shown in Fig. 1(a) and 1 (c) of semicircular shape were cut on copper block as shown in Fig. 1(b) using wire EDM process. These microchannels act as passage for working fluid and thus transfer heat to the working fluid by convection. Microchannels provide high heat transfer coefficient as it has large heat transfer surface area per unit volume of fluid flow.

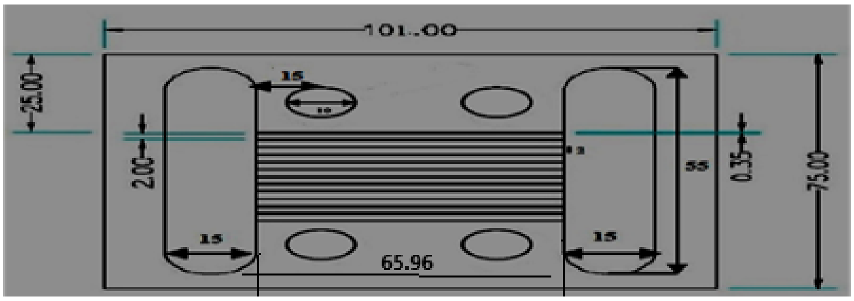

Fig. 1(a). Detail Schematic of Microchannel

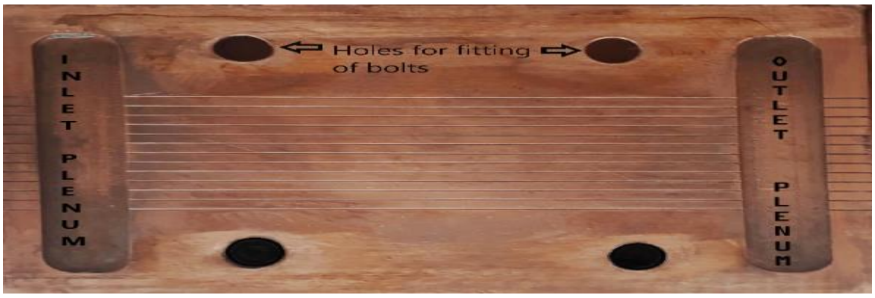

Fig. 1(b). Microchannel heat sink

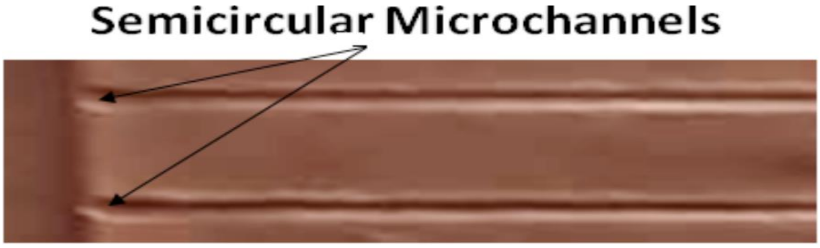

Fig. 1(c). View of Semicircular microchannels

- Cover plate (copper)

  A cover plate with provision for fluid inlet and outlet as shown in Fig. 2 of same size and shape as rectangular copper block has been used to clamp the rectangular block of microchannels.

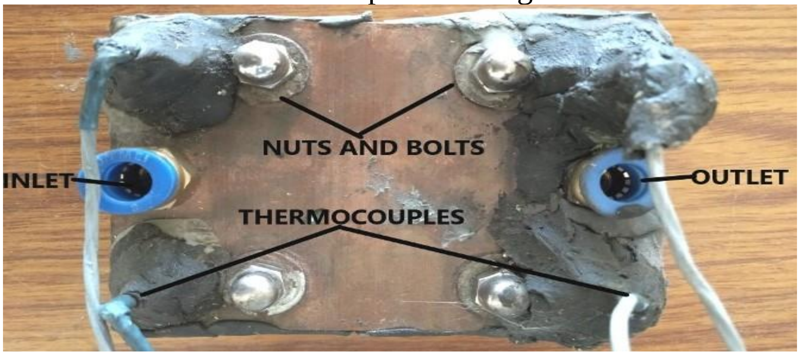

Fig. 2. Microchannel heat sink test section

- Heater (80W)

A disc heater of 80 W as shown in Fig.3 is used as a heating source of the MCHS. Disc heater is attached at bottom of the rectangular cover plate

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Fig. 3. Disc heater 80 W

- Pt 100 thermocouples

Four Pt100 thermocouples (-250°C to 650°C) are used to measure temperatures of working fluid during flow through microchannels. One thermocouple is used to assess the (t_S) of test section and another thermocouple is used of auto-cut of heat source in case of exceed of surface temperature above the prescribed limit.

- Control panel

A control panel as shown in Fig.4 comprises of temperature indicator to indicate the temperature reading of thermocouples, variac to control the voltage and power supply to heater, rheostat to set the prescribed limit of surface temperature of test section for auto-cut the power supply to heater, a power meter to show the heat supply to the test section, heater on-off switch.

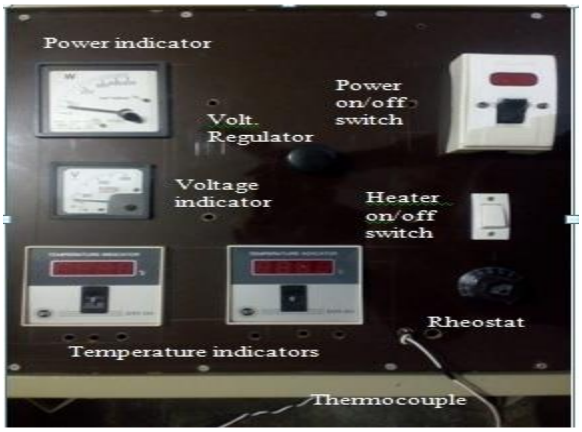

Fig.4. Control Panel

Table 1

Specification of measuring instruments.

Instrument	Range	Least count
Thermocouples (Pt 100)	-250℃ to 650℃	0.1℃
Voltage indicator	0 to 300 V	20 V
Power meter	0 to 500 W	20 W
Digital manometer (HTC 6115)	-103.4 kPa to 103.4 kPa	0.1

2.2 Experimental Procedure

The working fluid (filterized deionized water) is circulated from reservoir 1 as shown in Fig. 5 and Fig. 6 by a mini submersible pump (Inflex, model no: 801) throughout the whole test module. Valve 1 is used for regulating the flow rate while valve 2 and valve 3 are used to measure the pressure difference. Reservoir 2 is used for collecting exit deionized water. Test section is connected with the control panel which indicates temperature and voltage variations. Control panel is switched on during the experiment by pressing the power switch downward also heater switch is switched on by pressing it downward. Temperature indicators shows the temperature variations and required voltage is provided with voltage regulator which can be seen through voltage indicators. A rheostat is provided to fix the maximum required surface temperature of microchannel heat sink test section which can be sensed by a Pt 100 thermocouple connected to control panel. When maximum surface temperature of test section goes beyond the acceptable limit, the heater power is switched off automatically

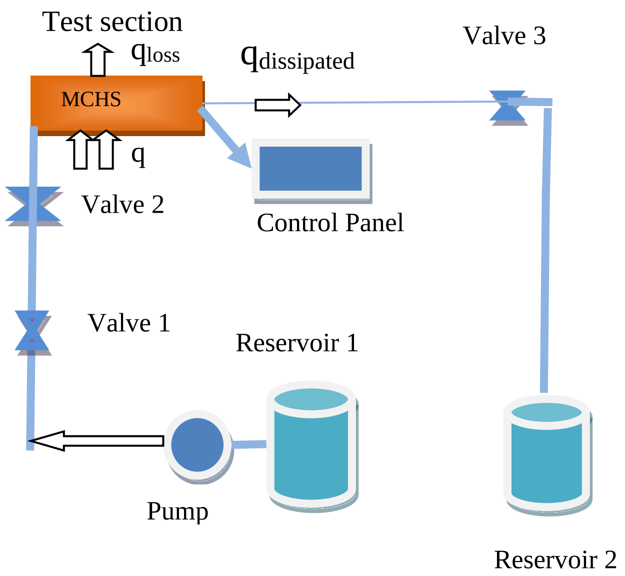

Fig. 5. Schematic diagram of experimental set up.

2.3. Unceratinity analysis

The experimental set up is allowed to be in continuous running condition for about 12 minute to reach a steady state. All the readings are noted after achieving a steady state when the variations in temperature were about 0.1°C. The uncertainties in voltage and power consumption measurements were 0.4%-0.15% due to variation in current. Heat input to the MCHS is assumed to 20% less than power supply due to the heat losses depending on the operating condition. The uncertainties reported during the measurement of pressure difference were 0.001-0.004 bar due to varying experimental condition.

3. Result and Discussion

Experiments have been performed for the heat transfer of a semi-circular MCHS in the range of varying ṁ (0.006 kg/s to 0.028 kg/s) at a fixed power consumption of 72 W.
Temperature at inlet plenum and outlet plenum can be calculated by taking average temperature of thermocouples location.

$$t_{in} = \frac{t_1 + t_2}{2} \tag{1}$$

where, t_1 and t_2 are the temperatures of fluid at inlet location of thermocouples whereas t_3 and t_4 are the temperatures of fluid at thermocouple locations of outlet

$$t_{out} = \frac{t_3 + t_4}{2} \tag{2}$$

Heat input to the microchannel heat sink (Q)

$$Q = h \cdot A \cdot \Delta t \tag{3}$$

where, h= heat transfer coefficient in (W/m^{2}K)
A= surface area of MCHS test section

$$A = (W_{ch} \cdot L_{ch} + \pi/4 \cdot d^2) \cdot N \tag{4}$$

where W_{ch} is the width, L_{ch} is the height d is the diameter and N is the number of microchannels.

$$\Delta t = t_{in} - t_{out} \tag{5}$$

Where Δt is the temperature difference of inlet and outlet fluid.
The pressure (Δp) drop can be calculated from the equation (6)

$$\Delta p = p_{in} - p_{out} \tag{6}$$

Friction factor (f) can be computed from the equation

$$f = \frac{\Delta p \cdot d}{\rho \cdot L_{ch} \cdot U^2} \tag{7}$$

Where, ρ is the density of working fluid (water) and U is the velocity of working fluid

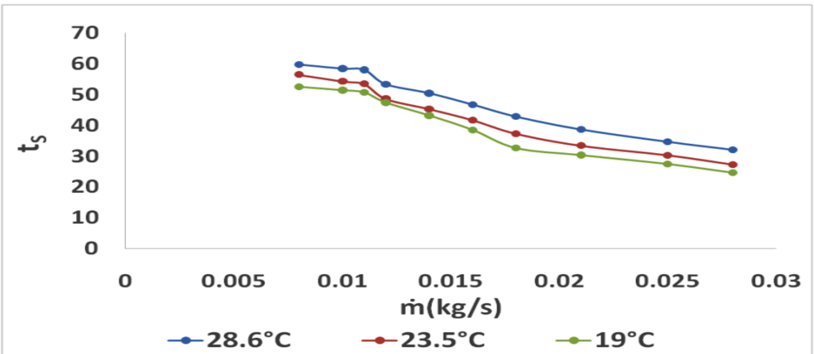

Fig. 6. Effect of mass flow rate on surface temperature

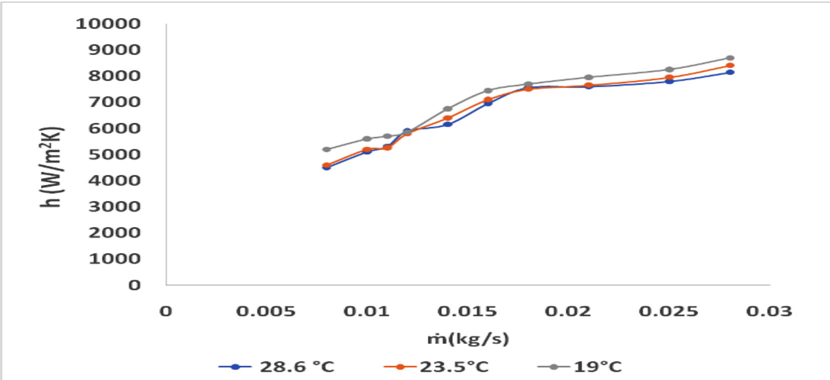

Fig. 7. Effect of mass flow rate on heat transfer coefficient

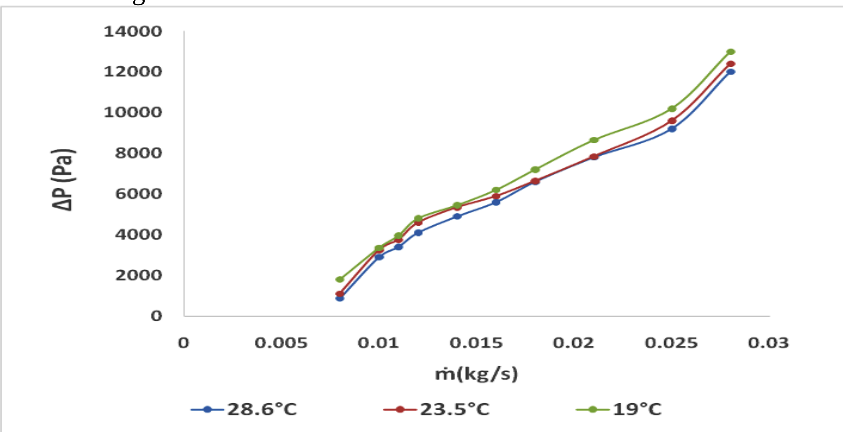

Fig. 8. Variation of Pressure drop with mass flow rate

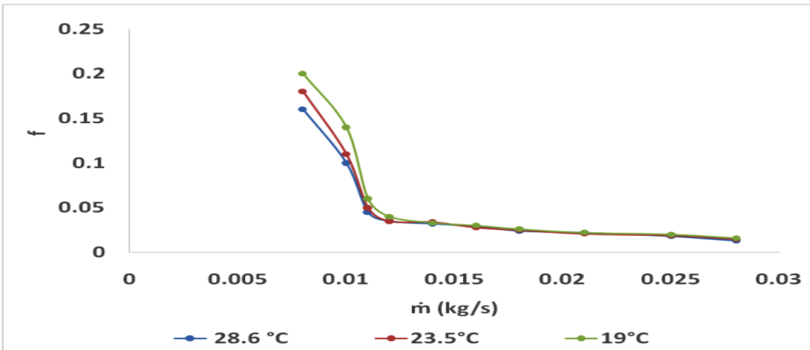

Fig. 9. Effect of mass flow rate on friction factor

As shown from Fig. 6 surface temperature of MCHS declined with increment of mass flow rate. Maximum surface temperature of test section has been observed to be 59.7ºC for inlet saturation temperature of 28.6ºC and maximum decline in surface temperature of test section has been observed to be 15.32% at ṁ (0.018 kg/s). Also at low inlet saturation temperature of fluid surface temperature is less at the same flow rate because smaller temperature inside the test section leads to higher temperature difference and hence surface temperature of the MCHS reduces. From Fig. 7 it has been observed that initially higher value of (h) is achieved with increase in value of (ṁ) and it achieved a maximum value (8700 W/m^2K) at 0.028 kg/s for inlet saturation temperature of 19ºC after this the heat transfer coefficient slope decreases due to phase change at lower value of ṁ for all the cases because of less boiling intensity at higher mass flow rate. It can be understood from Fig. 8 Pressure drop raises with increment in value of ṁ and maximum increment in pressure drop has been found to be 20.13% at 0.021 kg/s. Also when inlet temperature reduces then higher pressure drop is achieved because of higher viscosity. At lower value of ṁ pressure drop increases slightly for all the cases because reduced inlet temperature creates strong condensation effect in core region when boiling occur inside the microchannel heat sink. It can be seen from Fig. 9 friction factor decreases with increase of ṁ for all the three cases of inlet saturation temperature of fluid because when ṁ increases then velocity of working fluid also increased which results in a higher Reynolds number (Re) but smaller value of f at constant pressure drop.

6. Conclusions

Experimental analysis has been done for semicircular microchannels using deionized water at fixed power supply of 72 W and varying (ṁ) ranges from 0.006 kg/s to 0.028 kg/s with exit atmospheric pressure. The results of experimental analysis are summarized below:

I. Higher surface temperature of microchannel has been observed to be 59.7 °C for inlet temperature of 28.6°Cwhich is far below the maximum operating temperature of electronic chips. Maximum decrease in surface temperature is found to be 15.32% at ṁ = 0.018 kg/s because maximum difference in temperature is found at this flow rate.

II. Heat transfer coefficient has a peak value of 8700 W/m^2K at ṁ = 0.028 kg/s for lowest inlet temperature. Maximum increase in coefficient of heat transfer is observed to be 10.38% at the m= 0.016 kg/s for the lowest inlet temperature due to the phase change this flow rate.

III. Pressure drop rises linearly with flow rate while friction factor drops linearly with flow rate because of increase in viscosity with the increasing flow rate and maximum increment in pressure drop is found to be 20.13% at ṁ = 0.021 kg/s.

IV. Semicircular channels show a potential for application in cooling of electronic devices due to its higher heat transfer coefficient and a safe maximum operating temperature.

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