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INFLUENCE THE CHANNEL PATH ON HYDRO-THERMAL PERFORMANCE IN SERPENTINE MINI- CHANNEL HEAT SINK

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Abstract: The current work aims to inspect the influence of formation flow on hydrothermal performance and the uniformity of base temperature in the serpentine mini channel heat sink. The study was conducted by changing the flow formation with the straight channel model [A] and the wavy channel model [B] and by changing the place of inlet and outlet by putting the entrance at the heat sink center and using water as the working fluid. To obtain the numerical results, a 3D (ANSYS Fluent program) is used. A comparison between the numerical findings in the current work and experimental results from the literature review is carried out. The comparison shows moral agreement between the numerical and experimental findings. Moreover, the outcomes display that flow configuration has a great influence on the distribution of temperature and pressure drop. Although the model with straight channels enhances the temperature uniformity and pressure drop. Additionally, the overall performance factor (OPF) for all new models under study is better than traditional model. The average OPF for model [A] is (1.43) and for model [B] is (1.26). In addition, model [A] is better as compared with model [B] by 11.89% due to the efficient OPF and the uniformity of temperature in the base.

Keywords: *Flow formation, Overall performance factor (OPF), Computational fluid dynamic (CFD), Serpentine mini- channel heat sink.*

1. Introduction

The mini- channel heat sink is one of the most efficient devices that removes unwanted heat far away from the automated devices. The miniaturization in electronic devices, as well as the need for cooling technology, are significant impediments to achieving effective heat dissipation from the device and an efficient working device. Many researchers work on enhancing heat transfer and improving the thermal performance and thermal resistance in electronic devices, which leads to enhancing the efficiency of the working devices by changing the flow fragmentation, the shape of the channel, and the type flow laminar or turbulent. Jaffal et. al. [1] conducted a study on the effect of flow fragmentation in serpentine mini- channel heat sinks on the hydro-thermal achievement of the heat sink. The findings indicate that the new models have an obvious effect on enhance the characteristic of serpentine mini- channel heat sink as compared with traditional model. Sajid et. al. [2] explored experimentally the impact of wavy channel on the feature of hydrodynamic and heat transfer by taking TiO_2 Nano fluid as employed fluid. They reported that Nusselt number affected by the wave length more than

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the width of channel and they found that with increasing the power of heating, the improvement of Nusselt number decreased. Awais et. al. [3] accompanied experimental and numerical studies enhancement of hydrothermal performance by presenting geometries with two different types of headers with Nano fluid as the working fluid. The outcomes proved that the optimized mini- channel heat sink with the new header has a great effect on the thermo-hydraulic by decreasing the pressure drop by 43% and increasing the overall heat transfer coefficient by 13% as compared to the traditional model. A study was proposed to investigate the characteristics of fluid flow and heat transfer by using a helical mini- channel heat sink in a novel cylindrical heat sink by Falahat et. al. [4]. The findings provide that mini- channel with helix angle of 45 and 60 have a good enhancement in Nusselt number by 39.5% as compared to heat sink that have straight mini- channel heat sink. Ghasemi et. al. [5] investigated numerically and experimentally the effect of various diameters of channel (4, 6, 8) mm on the hydro-thermal enforcement of heat sink. The outcomes display that mini- channel heat sink with a diameter 4mm have an obvious enhancement in heat transfer coefficient. Thermal and hydraulic investigations have been conducted to test proposed models, namely sandwich distributions, outlet enlargement, block, rectangular, thin rectangular, and trapezoidal in micro- channel heat sink by Hung et. al. [6]**.** The outcomes proved that sandwich distribution is the best model regarding pressure drop and thermal performance. Imran et. al. [7] tested numerically and experimentally four designs in serpentine mini- channel heat sink. The results show an effectual improvement in the thermal hydraulic performance when using serpentine mini- channel heat sink that have 2 inlets and 2 outlets as compared to traditional serpentine that has one inlet and one outlet. The effect of the corrugated mini- channel heat sink with trapezoidal, sinusoidal, and triangular shapes on the thermal hydraulic exhibition has been calculated by Aliabadi et. al. [8]. The outcomes show that model with the trapezoidal corrugated mini- channel heat sink has a great enhancement in pumping power and Nusselt number. The behavior of specific parameters of wavy sinusoidal mini- channel heat sink on the pumping power and heat transfer rate has been conducted by Aliabadi et. al. [9]. They took a wave amplitude (0.5, 1, 2) mm and wave length (10, 20, 40) mm. The results show that the optimum model with wave amplitude and length (2, 40) mm proved its effect on thermal performance. Kumar et. al. [10] experienced the influence of straight, wavy and branch wavy mini- channel heat sink on the thermal and hydraulic performance. The outcomes demonstrated that branch wavy mini- channel heat sink have a superior improvement in the connection temperature and thermal hydraulic performance. Lu et. al. [11] conducted a new geometry to decrease the thermal resistance and pressure drop in a micro-channel heat sink by using a wavy porous fin. They reported that the porous wavy fin gave a respectable enhancement in decreasing both thermal resistance and pressure drop. The temperature uniformity has been improved numerically on the heat transfer surface by using variable height channel by Mu et. al. [12]. The findings show that the distributer with variable height channel and circular turning has the highest efficiency as compared with other models. Ren et. al. [13] conducted the effect of width and non-uniform channel arrangement on the heat sink thermal performance. The arrangement of width non uniform channel show its benefits in reducing the overall thermal resistance by 6.1% and enhancing the distribution of uniform flow. The influence of wavy microchannel has been studied and compared to straight microchannel numerically by Sui et. al. [14], the wavy micro channel proved its advantageous on the heat transfer improvement. The behavior of heat input, Reynolds number, Nano fluid concentration, hydraulic diameter, and type of Nano fluids on serpentine microchannel and its effect on both fluid flow and heat transfer have been studied by Sivakumar et. al. [15-17]. Singh et. al. [18]. examine the influence of interconnecting channels in mini- channel heat sink, the findings demonstrate that the best enhancement of electronic device can be achieved when the angle of the secondary channel is 10°.

Accordingly, there are enormous studies to improve the thermal performance of mini-channel heat sink by studying several parameters and changing the flow fragmentation. The current study aims to fill a small gap in the research's mentioned above by presenting two new models of serpentine mini- channel heat sink, namely model [A] (serpentine with straight channel and inlet at center) and model [B] (serpentine wavy channel with inlet at center). The main goal of putting the inlet at the center is that the core of the heat sink is the hottest region, and by putting the entrance in the center, it leads to cooling it. The new models are investigated under the same boundary conditions, in which the flow is laminar and the heat flux is constant. The hydrothermal performance and the temperature distribution in the base of the heat sink have been studied for model [A] and model [B] and compared with traditional model.

2. Numerical Computation

2.1. Modelling geometry

To investigate the effect of flow fragmentation, two models have been proposed and compared with the traditional model as shown in figure-1-. The three models are the traditional model (serpentine mini- channel heatsink) as presented in figure (1-a), model [A] has a straight channel and the inlet of flow at the middle of the heat sink as depicted in figure (1-b), and model [B] has a wavy channel and the inlet of flow at the center of the heat sink as displayed in figure (1-c). All models under study have been tested for their effects on the distribution of temperature in the base of the heat sink and on the overall performance factor. All the models have the same dimensions with length, width, and thickness (100,86,5) mm respectively. The channel in all models has the same width and height (4,2) mm respectively.

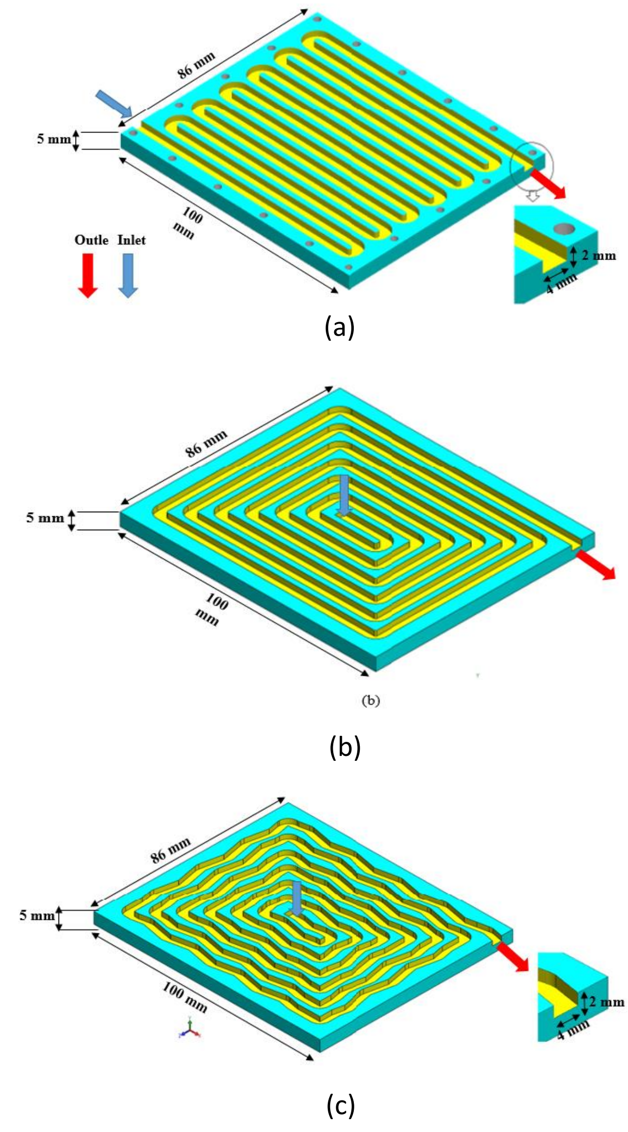

Figure 1. Geometry of serpentine mini- channel heat sink, a) traditional model, b) model [A] and c) model [B]

2.2. Independence of Grid

To establish the accuracy of the results with the minimum expected computational time, mesh independence has been applied. To examine the impact of grid sensitivity, four grids with different types of elements have been tested. The type of mesh in this work is tetrahedron, traditional model with a number of elements of 2600000, Model -A –with a number of elements of 1600000, Model [B] with a number of elements of 889000 as shown in figure -2-. The test has been performed under mass flow rate and heat flux of 0.0025kg/s and 20000W/m^2 respectively. The number of elements is different between shapes due to the complexity of the shape of the channel.

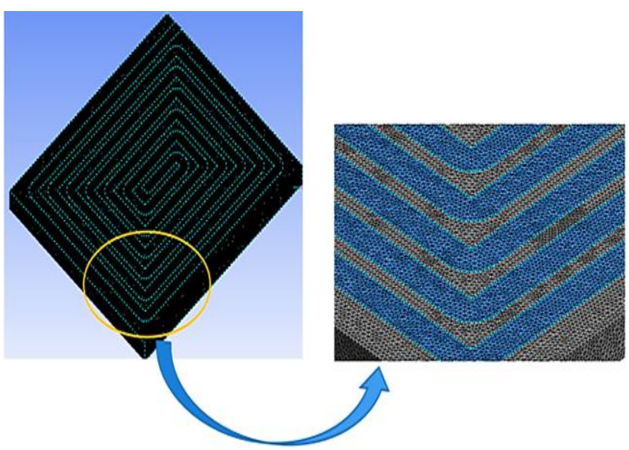

Figure 2. Display grid generation

2.3. Boundary conditions

To investigate the solution of the process, numerous boundary conditions have been applied. The stream is steady, laminar, and incompressible. The outlet pressure is set at zero-gauge pressure. The water is taken as the working fluid, the range of mass flow rate under the present study is 0.0025 and 0.004kg/s. The inlet temperature is equal to 300K with uniform heat flux 20000W/m^2 exposed on the base of heat sink. The other faces of heat sink are adiabatic as shown in figure -3-.

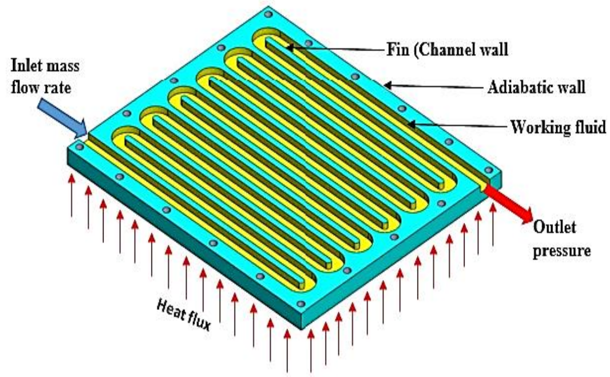

Figure 3. Boundary conditions in the solid and fluid domain

2.4. Governing Equation

The characteristic equation that used in the present study is continuity equation, momentum equation and energy equation [19, 20]. Equation of continuity

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

Equation of momentum

$$\rho_f\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu_f(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}) \tag{2}$$

$$\rho_f\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu_f c(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}) \tag{3}$$

$$\rho_f\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu_f(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}) \tag{4}$$

Equation of Energy

$$\rho_f C_{p,f}\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=k_f(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}) \tag{5}$$

For solid the equation of energy is

$$k_S\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial x^2}\right)=0 \tag{6}$$

Where k_S is the thermal conductivity of copper with 387.6 W/m.K.

3. Calculation Procedure

Several steps have been applied to calculate the overall performance factor as follow.
The following equation is used to calculate heat transfer flow rate.

$$Q = \dot{m}C_p(T_{out} - T_{in}) \quad (7)$$

Where $\dot{m}, C_p, T_{in}$ and T_{out} are presented mass flow rate, specific heat, inlet and outlet temperature of water. The properties of coolant fluid are taken at inlet region with density (ρ) 998.2 kg/m^3, specific heat (C_p) 4182 j/kg-K, thermal conductivity (k) 0.6 W/m-K and viscosity (v) 0.001 kg/m-s.
To evaluate the hydraulic diameter of mini-channel, the following formula have been applied.

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

Where P is the perimeter of channel and A is the area of channel.
To compute the Nusselt number [21], the following relation have been conducted.

$$Nu = \frac{h*Dh}{k_f} \quad (8)$$

$$h_{avg} = \frac{q}{T_{avg}-T_{film}} \quad (9)$$

Where T_{avg} is the based average temperature and T_{film} is the working fluid temperature.

To estimate the overall performance factor [22,23], the relation bellow is used.

$$OPF = \frac{\left(^{Nu_n}/_{Nu_t}\right)}{\left(^{\Delta p_n}/_{\Delta p_t}\right)^{1/3}} \quad (10)$$

Where Δp_n is the pressure drop of new model. Δp_t is the pressure drop of traditional model and Nu_n is the Nusselt number of new model, Nu_t is the Nusselt number of traditional model.

4. Results and Discussion

4.1. Validation of Numerical Computation

To validate the reliability of the proposed numerical approach, a comparison has been conducted between the numerical findings of the traditional model in the present work and experimental findings from Ref. [24] for both Nusselt number and pressure drop. Figures 4 and 5 show the comparison of Nusselt number and pressure drop between the numerical findings of the current study and the experimental results of Ref. [24]. The figures illustrate that the maximum error aberration for Nusselt number is 13.57% and for pressure drop is 12.88%. The difference between numerical and experimental findings was correlated with the accuracy of measuring devices and numerical expectations.

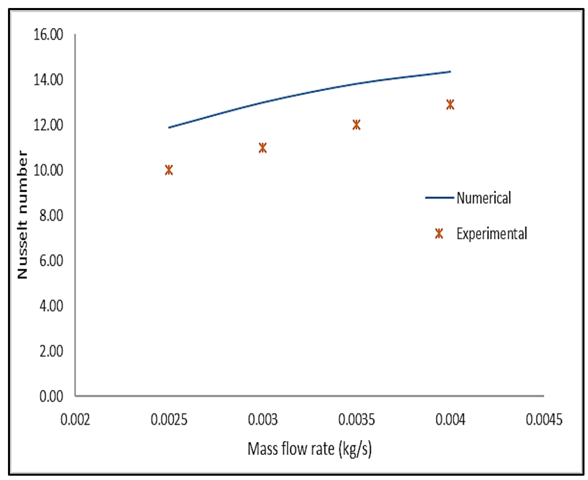

Figure 4. Validation of Nusselt number between experimental and numerical findings.

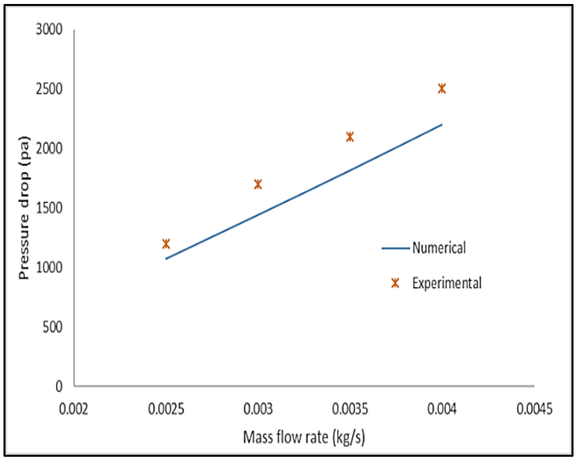

Figure 5. Validation of pressure drop between experimental and numerical findings.

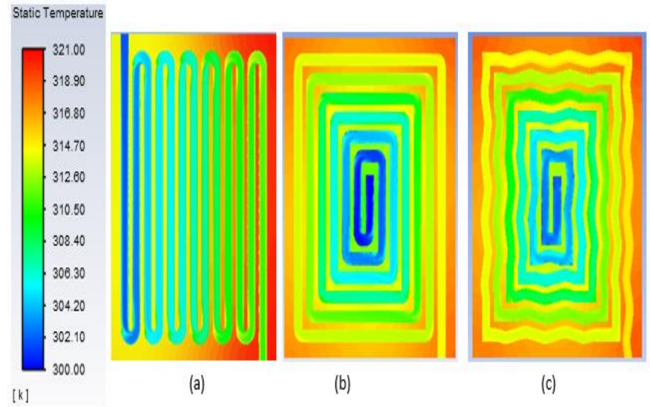

Figure 6. Temperature contour of mini- channel heat sink (a) traditional model, (b) model (A), (c) model (B).

4.2. Effect of flow formation on base temperature and pressure drop

Figures (6 and 7) represent the contour of distribution in temperature of the heat sink and cooling fluid respectively for traditional model and proposed models under study at mass flow rate and heat flux 0.0025kg/s and 20000W/m^2 respectively. For all models under study, it can be noticed that the minimum temperature of the base of the heat sink occurs at the inlet zone of cooling fluid, while the maximum base temperature comes out at the outlet region of cooling fluid. The reason behind that is to increase the cooling fluid temperature as a results of transferring heat from the heat sink to it along the channel. These figures depict that flow fragmentation leads to enhanced thermal achievement and uniformity of temperature at the base of the heat sink. However, these figures demonstrate that model [A] has the better temperature uniformity as compared with traditional model and model [B]. Figure (7) shows that model [A] has the highest outlet temperature according to the large amount of heat transfer.

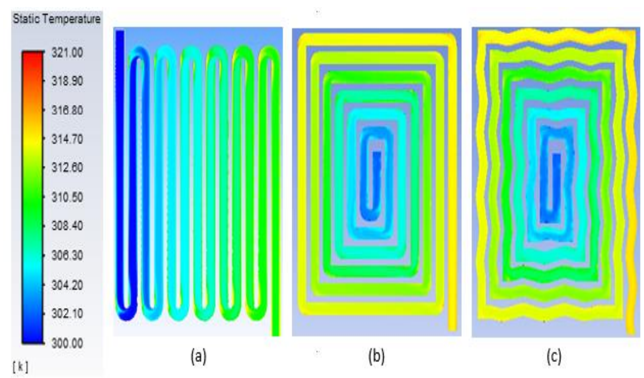

Figure 7. The contours of temperature in the fluid domain (a) traditional model, (b) model (A), (c) model (B).

Figure-8- show the contours of pressure drop in the fluid field in mini- channel heat sink for the traditional model, model [A] and model [B] at mass flow rate and heat flux of 0.0025 kg/s and 20000W/m^2 respectively. The figure illustrates that the maximum pressure drop appears at model [B] and the minimum pressure drop occurs at traditional model. The contours also show that highest pressure drop occur at the inlet region and gradually decrease until reach to the outlet region for all models. This increase in pressure drop is due to the increase in the surface region in the new model by 0.42 % for model[A] and 2.29 % for model [B].

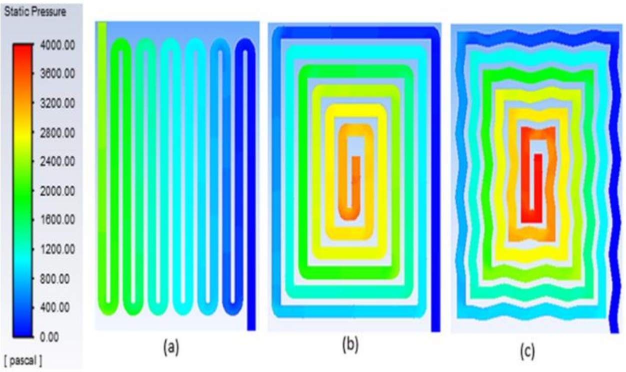

Figure 8. Pressure drop distribution of the fluid domain (a) traditional model, (b) model (A), (c) model (B).

Figure-9- display that pressure drop growths with increasing the mass flow rate for all models. The pressure drop in the new models is higher than the traditional model due to the large amount of curves. The benefit of the new models is the large enhancement in the uniformity of temperature distribution and the large enhancement of hydro-thermal performance. Furthermore, the results show that model [B] has a large pressure drop as compared to conventional model and model [A] due to the increasing of the surface area in model [B].

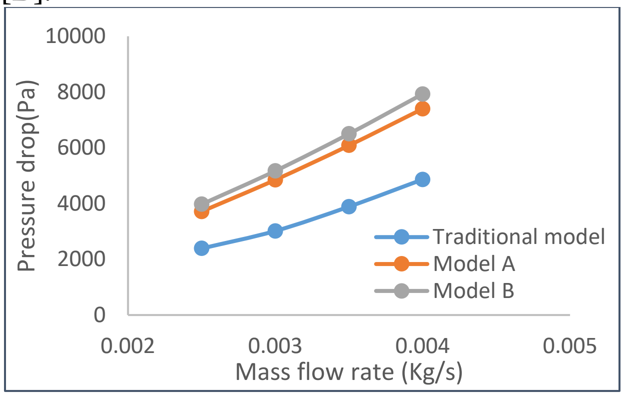

Figure 9. Compression of the pressure drop for different models at different mass flow rates.

Figure -10- depicts variation of the Nusselt number for all models under study at different mass flow rates. The results displayed that Nusselt number increases with growing the mass flow rate according to increasing the mixing rate of cooling fluid and thus increase the amount of heat transfer. In addition, the Nusselt number for the new models is higher than the Nusselt number for traditional model. Moreover, model [A] has a Nusselt number higher than traditional model and model [B]. The average Nusselt number for traditional model, model [A] and model [B] is 10.28,17.135 and 15.28 respectively.

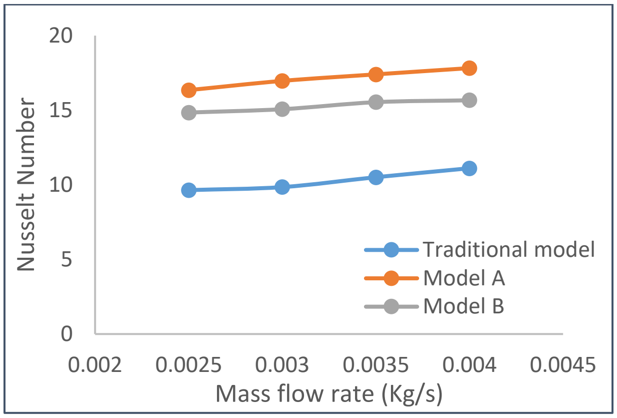

Figure 10. Nusselt number of different mass flow rate for different models.

According to the findings in figure -11-, it is shown that flow formation has a large effect on thermal resistance. According to the large amount of rejected heat, the thermal resistance decreases with increasing the mass flow rate at constant heat flux and thus contributes to the enhancement of thermal performance. The percentage of enhancement thermal resistance is 40 % for model [A] and 34% for model [B] as compared with the traditional model. Furthermore, for wholly mass flow rates model [A] has lesser thermal resistance in comparison with traditional model and model [B]. The central goal of enhancing the thermal accomplishment of mini- channel heat sink is to accomplish the largest amount in the dissipation of heat with the least pressure drop and thus reduce the pumping power required [25].

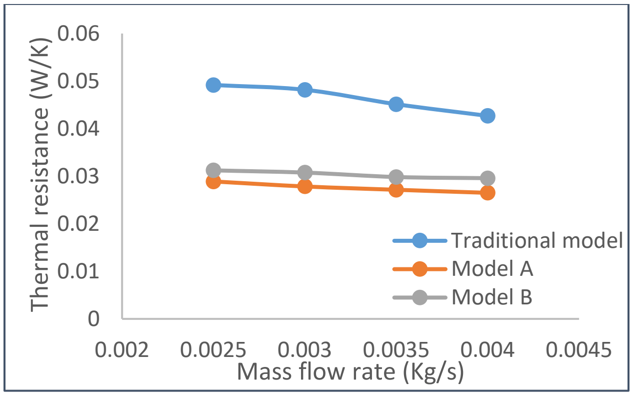

Figure 11. Comparison of different models of different mass flow rate on thermal resistance.

Figure -12- clarifies the behavior of OPF for new models at different mass flow rates. Furthermore, the OPF decreases with increasing the mass flow rate for both model [A] and model [B] this related to that pressure drop increasing with increasing the mass flow rate. Other studies have the same results [26, 27]. The figure depicts that flow fragmentation has a great influence on hydro-thermal performance by the noticeable improvement in the OPF of mini- channel heat sink. In addition, the OPF for model [A] is higher than model [B] by 11.89% due to the high Nusselt number of model [A] as shown in figure (10) by 10.83 % than model B and 39.42% than traditional model.

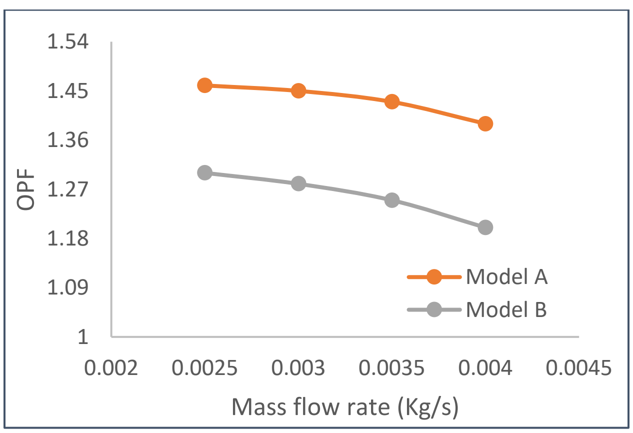

Figure 12. Overall performance factor at different mass flow rate for the new models.

5. Conclusions

This paper is interested in enhancing heat transfer in mini- channel heat sink by studying the influence of both the inlet and the outlet location of cooling fluid, along with the investigation of the influence of different formations of the channel. These parameters' effects are studied on pressure drop, based temperature, Nusselt number, and OPF. In agreement with the results that were found from the current study, the consequent can be summarized to:

1. The average Nusselt number for models [A] and [B] is higher as compared to traditional model by 40% and 32.72% respectively.
2. The average thermal resistance for models [A] and [B] is lower as compared to traditional model by 39.13% and 34.78% respectively.
3. The average OPF for models [A] and [B] is higher as compared to traditional model by 1.43 and 1.26 respectively.
4. The based temperature of the heat sink for models [A] and [B] is more temperature uniform as compared to traditional model.
5. The average pressure drop for model [A] is lower than model [B] by 6.48% and the average OPF for model [A] is greater than model [B] by 11.89 %. According to that it can be concluded that the model [A] is better model among models under study.

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The data value statement

Conflict of interest

The authors confirm that the publication of this article causes no conflict of interest.

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