Performance of Rectangular Fin Heat Sink with Elliptic Bottom Profile Cooled using Air Jet Impingement

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Abstract

Direction of coolant plays an important role in improving the thermal performance of plate fin heat sinks. Perpendicular flows are extensively used for multifarious applications such as electronic equipments cooling, blades of turbine and sheet metal drying due to their enhanced heat and mass transfer characteristics. This paper focusses on numerical analysis of flat and elliptical grooved bottom heat sinks subjected to impinging flow in order to improve their thermal performance characteristics. These include convection coefficient, thermal resistance and pressure drop. Three different configurations of rectangular and elliptical bottom fins were modelled using ANSYS ICEPAK V19 package. The input impinging velocity changed from 0.5 m/s to 1.0 m/s. It was observed that increasing the space between elliptical bottom fins improved the thermal peroformance which was due to the generation of smaller vortex causing smooth flow in the elliptical channels. The thermal resistance of rectangular fins with highest inter space was found moderately high due to high velocity of air causing stronger flow momentum from nozzle exit. In case of elliptical bottom fins, the highest reduction of thermal resistance was found for fins with minimum inter spacing. The pressure drop for all the configuations was vaidated through available experimental results.

Keywords: *Heat sink, Elliptical profile, thermal performance, impingement*

1.0 Introduction

With the advances in semiconductors, power density electronics and microelectronic industries, design of the heat sink for long term reliable operation becomes critical. Heat sink performance is usually measured with respect to thermal resistance, convective heat transfer coefficient and pressure drop. In addition to these characteristics, coolant direction and fin profile also plays an important role in assessing the efficiency of heat sinks. The direction of coolant may be along the heat sink base or

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impinging i.e., perpendicular to base of the heat sink. Impingement flows are extensively used for multifarious applications such as electronic equipments cooling, blades of turbine, continuous drying of sheet metals, and thermal treatment of various materials due to their superior characteristics in terms of heat and mass transfer. The heat transfer coefficient for these applications commonly adopting impingement jet was found to be larger than those using cross circulation dryer for the same amount of flow rate [1]. Type of flow pattern of fluid, whether parallel flow or impingement flow depends upon the type of cavity shapes through which it flows. Heat sink bottom can have different profiles namely rectangular, triangular with different included angles, semi-circular, inverse trapezoidal, trapezoidal, W-shape, rectangular, semi-oval and so on.

Some of the researchers have tried to optimize performance of the heat sink through experimental and numerical methods. Seok Pil Jang et al. [2] explored characteristics of the fluid flow and heat transfer of a microchannel heat sink having flat bottom profile subjected to an impinging jet. It was shown that the cooling attainment of microchannel heat sink with jet impingement got increased by about 21% than parallel jet flow. Li et al. [3] revealed the effect of Reynolds number (5000-25000), impinging distance (4-28), and the fin dimensions on thermal performance of rectangular fin heat sink with flat bottom subjected to air flow perpendicular to the heat sink base. While the heat dissipation from the heat sink was found to increase with the Reynolds number, the influence of the fin height on the thermal resistance was not significant as compared to Reynolds number and width of the fin. Sidy Ndao et al. [4] studied the influence of cross flow area and shapes of pin fin on the single-phase impingement point heat transfer coefficients of jet on micro fins having flat, elliptical, hydrofoil, square and circular bottom with R132a as the working fluid. Among these, the circular and square fins exhibited maximum heat transfer coefficient for a given Reynolds number of the jet. Chuan Leng et al. [5] used three-dimensional solid–fluid conjugate model coupled with a simplified conjugate gradient method to optimize the performance of heat sinks having double-layered microchannel. The enhancement in uniformity of temperature on the bottom wall was attributed to the increasing in height of top channel and decrease in the coolant inlet velocity. Khan et.al. [6] investigated thermal resistance, Nusselt number, and friction factors for various heat sink having rectangular, inverse trapezoidal, triangular, trapezoidal, W-shape, rectangular and semi-oval bottom profiles. While, the inverse trapezoidal shape showed the best thermal performance for Reynolds numbers upto

300, semi-oval showed the worst thermal resistance. Ridvan Yetal. [7] reported flow and thermal characteristics of heat sink with hexagonal fins having flat bottom profile with impinging air jet using ANSYS Fluent CFD program. The correlations developed for Nusslet number, from the experimental and numerical results were very close to each other and had large accuracy. Tang et al. [8] carried out numerical simulations to analyze heat transfer characteristics of jet impingement with a novel single cone heat sink with flat bottom. The authors showed that the cooling effect of fluid impinging on a cone heat sink was superior to that of a conventional flat plate heat sink. Tae et al. [9] developed closed-form solution for the thermal resistance and pressure drop of a rectangular fin heat sink having flat bottom subjected to uniform impingement by air jet. The thermal performance of the heat sink with optimum geometry was approximately 50% larger compared to conventional heat sink. Ammar H[10] et al. analysed the effect of direction of flow and fillet profile on the thermal behavior of rectangular fin heat sinks. It was found that the heat sink resistance and temperature of the base under impingement flow was 5.18% and 3.78 respectively lower compared to those of parallel flow arrangement**.** Bouchenafa et al. [11] investigated the hydraulic and thermal performance of the two heat sinks made of wavy fins having flat bottom. The heat sinks having wavy fins had a substantial pressure drop with reference to rectangular fins heat sinks, because of the increase of friction between the solid walls and fluid. The heat sink having the higher number of waves and the greatest amplitude showed lowest thermal resistance. Ling et al. [12] observed that the multi-walled carbon nanotubes dispersed in the fluid enhanced the Nusselt number and thermal performance of elliptical tubes with helical baffled heat exchanger. Ventola et al. [13] investigated the effect of micro-structured roughness (1 to 25 μm) on thermal performance of heat sinks cooled by air flowing parallel to the heat sink with flat bottom. Laser sintered heat sinks showed 50% higher thermal performance than conventionally manufactured heat sinks. Zhipend Duan et al. [14] conducted investigation to study the pressure drop of a plate fin heat sink subjected to an impinging flow with elliptic bottom and flat bottom profiles. The lower pressure drop for heat sink with elliptical bottom was observed when compared to that of flat bottom heat sink.

From literature review, it was observed that the impinging flow improved thermal performance of heat sinks to higher extent when compared to parallel flow. Many of the studies were mainly focused on fin geometry optimization by considering pressure drop and fin cross-sections as rectangular, square and triangular profiles. Very recently heat sink with elliptical bottom profile has been studied only for pressure drop [14] characteristics. In addition to pressure drop of elliptical profile heat sink, thermal performance parameters such as convective heat transfer coefficient and thermal resistance also becomes important which is not yet been reported so far. Hence the present paper focuses on investigating the effect of impinging flow on heat transfer characteristics such as convection coefficient and thermal resistance of elliptical and flat bottom profile heat sinks using ANSYSICEPAK 2019 R1 package. The resulting pressure drop was compared with experimental data available in literature.

2.0 Thermal analysis of heat sink with flat bottom and elliptical bottom

2.1 Heat sink geometry and Meshing

The heat sink model having fins with flat and elliptical profile was modelled using Solidworks package as shown in Fig.1. In impingement flow, the cooling medium was considered o enterin the direction perpendicular to the heat sink bottom and exit from other sides. Three rectangular (R1, R2 and R3) and three elliptical (E1, E2 and E3) bottom heat sinks were considered for thermal analysis. Table 1 shows plate fin heat sink geometrical parameters. ANSYS Icepak 2019 R1 was adopted to construct the computational grid and discretize the heat sink model. High density hexadecimal structured mesh was generated in fluid and solid domain. The meshed model of heat sink with rectangular bottom is as shown in Fig. 2.

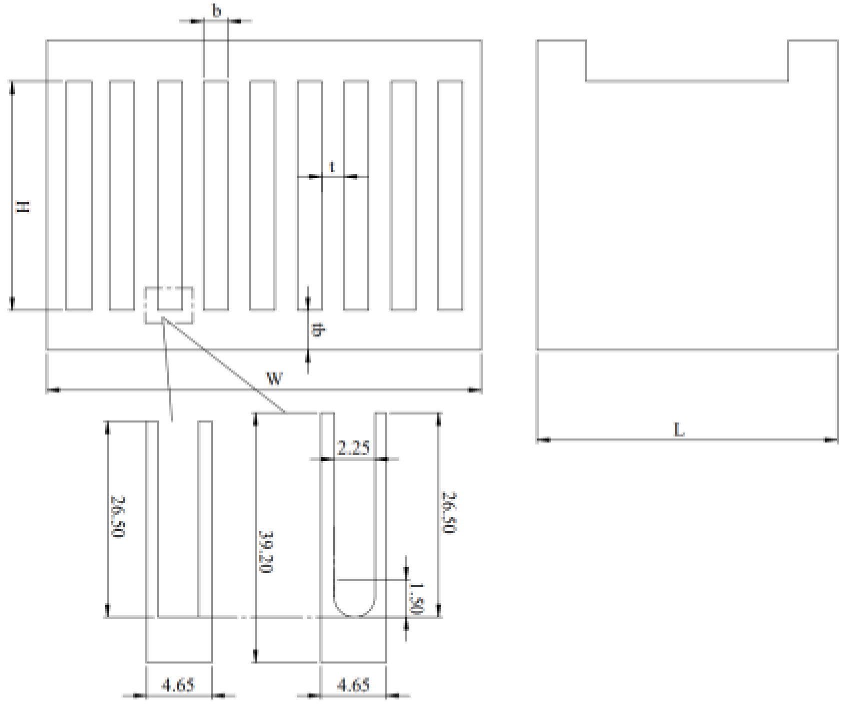


Fig.1. The geometry model of heat sink with rectangular and elliptical bottom

Table 1. Plate fin heat sinks models with various geometrical parameters

Dimension	R1 (Rectangle bottom)	R2 (Rectangle bottom)	R3 (Rectangle bottom)	E4 (Elliptical bottom)	E5 (Elliptical bottom)	E6 (Elliptical bottom)
L (mm)	127	127	127	127	127	127
W (mm)	122	122	122	122	122	122
t_b (mm)	12.7	12.7	12.7	12.7	12.7	12.7
b (mm)	1.2	1.2	1.2	1.2	1.2	1.2
t (mm)	2.25	4.27	6.2	2.25	4.27	6.2
H (mm)	26.5	26.5	26.5	26.5	26.5	26.5
N_f	36	22	19	36	22	19

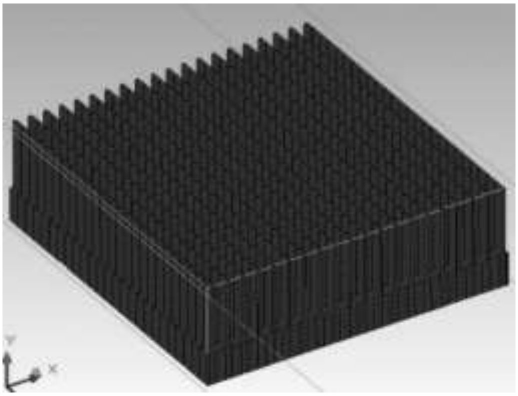


Fig. 2. Meshed model of heat sink with Rectangular bottom

2.2 Boundary Conditions

Heat input of 5W is given to the heat sink base. Impingement velocities of 0.5, 0.6, 0.7, 0.8 and 1.0 m/s at ambient temperature of 25^0C were considered. Aluminium alloy Al6063-T5 grade was adopted as heat sink material having thermal conductivity of 209W/mK. Impinging air was at a height of 26.5 mm from the bottom of the fin.

2.3 Numerical Computation

To assess the heat sink having rectangular and elliptical bottom thermal performance, numerical modeling was carried out. Results were expressed in terms of average Nusselt number, heat sink thermal resistance with Reynolds number. To simulate numerically the convective heat transfer in heat sinks with flat bottom elliptical bottom, continuity equation, Navier-Stokes (N-S) equation and energy balance equations were used. The following assumptions were made during the simulation:

1. The flow of air is considered to be incompressible, steady and turbulent.
2. Three-dimensional solid-fluid conjugate.
3. The properties of coolant air is a function of mean temperature.

The three dimensional Navier-Stokes equations are given by (1) – (3)

$$\nabla(\rho U u) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \quad (1)$$

$$\nabla(\rho U v) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{yx}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} \quad (2)$$

$$\nabla(\rho U w) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} \quad (3)$$

where ρ is the fluid density, τ is the viscous stress tensor, p is the pressure, u, v and w are velocity components considered in x, y and z direction respectively,

The energy equation is expressed as (4)

$$\nabla(\rho HU) = -p\nabla U + \nabla(k\nabla T) + \varphi + S_h \quad (4)$$

where H is the total enthalpy, k thermal conductivity, φ is the dissipation term, T temperature and S_h is the source.
The equation of continuity is expressed by (5)

$$\nabla(\rho U) = 0 \quad (5)$$

The average Nusselt number is as shown in Equation (6)

$$Nu_a = \frac{h_{avg} D_h}{K_a} \quad (6)$$

where D_h is the inlet hydraulic diameter, h_{avg} is considered to be the average convective heat transfer coefficient and K_a is the thermal conductivity of the air considered at the mean temperature, T_m.

The mean temperature, T_m is as shown by Equation (7):

$$T_m = \frac{T_{avg} + T_b}{2} \quad (7)$$

where T_b is the base temperature of fin and T_{avg} is air average temperature given by Eq. (8)

$$T_{avg} = \frac{T_{out} + T_{in}}{2} \quad (8)$$

where T_{in} and T_{out} are air inlet and outlet temperature of air respectively. The average convective heat transfer coefficient, h_{avg}, is as shown in Eq (9)

$$h_{avg} = \frac{Q}{A_s(T_b - T_m)} \quad (9)$$

where Q is the rate of heat transferred to the air, A_s is the total surface area exposed to cooling medium and T_m is the air mean temperature.

The rate of heat transfer to air, Q is given by Equation (10)

$$Q = m_a C_{pa}(T_{out} - T_{in}) \quad (10)$$

where m_a and C_{pa} is the air mass flow rate and specific heat respectively.

For heat sink with flat bottom, the total heat transfer surface area is given by Equation (11)

$$,A_s = (N_f\text{-}1)t_bL\text{-} + 2N_fH(L + t) + B(L + W) \quad (11)$$

For elliptical bottom heat sink, the total surface area is given by Equation (12)

$$A_T = \left(N_f - 1\right)L2\pi\sqrt{\frac{a^2+b^2}{2}} + 2N_fH(L + t) + B(L + W) \quad (12)$$

where W, L, N_f, H, t, B, is the heat sink width, length, number of fins, fin height, fin thickness, heat sink base height respectively, 'a'semi-major diameter and 'b'are semi minor diameter of elliptical groove.

The thermal resistance R_{th} is given by Equation 13

$$R_{th} = 1/(h_{avg}A_T) \quad (13)$$

3.0 Results and Discussions

3.1 Effect of fin geometry and impingement velocity on Nusselt number

The effect of impingement velocity (in terms of Reynold's number) on Nusselt number for rectangular (R1, R2 and R3) and elliptical bottom (E1, E2 and E3) heat sink configurations are as shown in Fig.3 and 4 respectively. It was observed that increase in gap between the fins caused increase in Nusselt number for both rectangular and elliptical models. Increase in impingement velocity from 0.5 to 1 m/s did not cause significant rise in Nusselt number for R1 and R2. But, increase in impingement velocity from 0.5 to 1 m/s caused an increase in Nusselt number from 15.00 to 20.00 for R3 configuration. Increase in impingement velcoity on heat sink having rectangular fins with maximum gap causes increase in maximum velocity as shown in vector plot (Fig.5). This causes the generation of high strength vortex due to decreasing viscous force [15] and hence leads to enhanced convective heat transfer rate. This causes an hike in Nusselt number. Also, increase in impingement velocity caused higher magnitude of Nusselt number for both E2 and E3 models. Hence increasing the space between elliptical bottom fins is more effective than the rectangular one. This may attributed to the generation of smaller vortex causing smooth flow in the channels due to elliptical bottom profile [15].When the gap between the fins changes to 6.2mm, plate heat sink with flat bottom (R3) showed slightly higher Nusset number when compared to plate heat sink with elliptical bottom (E3).This was due to increase in veocity of impinging air flow at the exit of heat sink for R3 when compared to that of E3 as shown in velocity vector plot (Fig.6).Also, the maximum temperature of E3 (26.6ºC) is slightly higher compared to R3 (26.4ºC). This is as shown in temperature contours of R3 and E3 for impinging velocity of 0.5m/s in Fig.7. The rise in the maximum temperature of all configurations was found to be less because of lower magnitude of heat flux, which is equal to 3.22×10^{-4}W/mm^2.

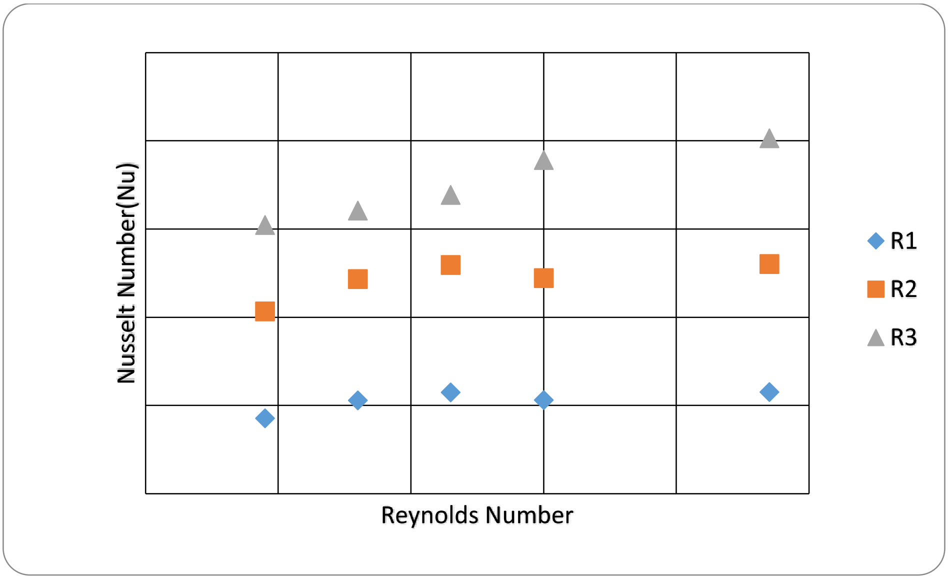


Fig. 3. Variation of Nusselt Number with Reynolds number for rectangular bottom heat sink

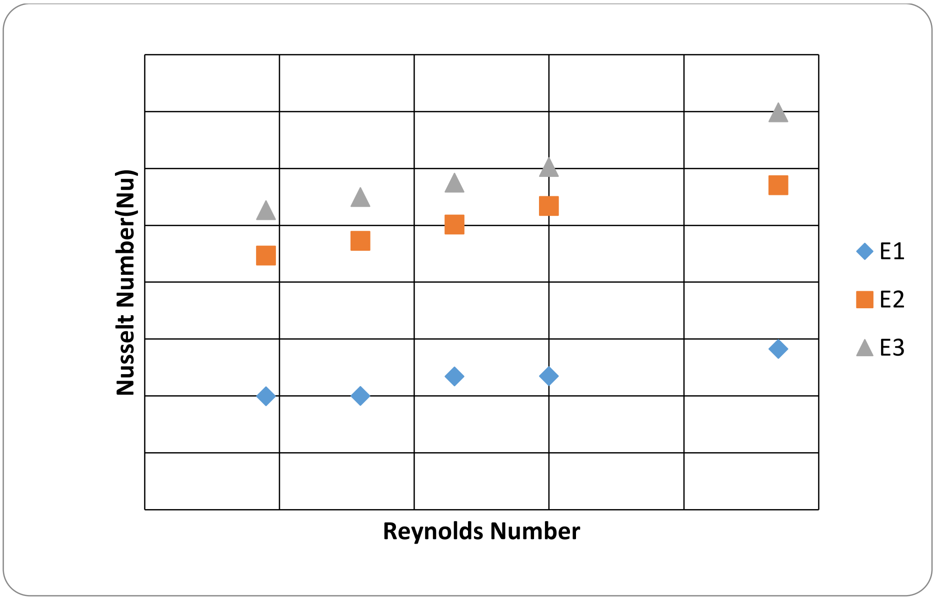
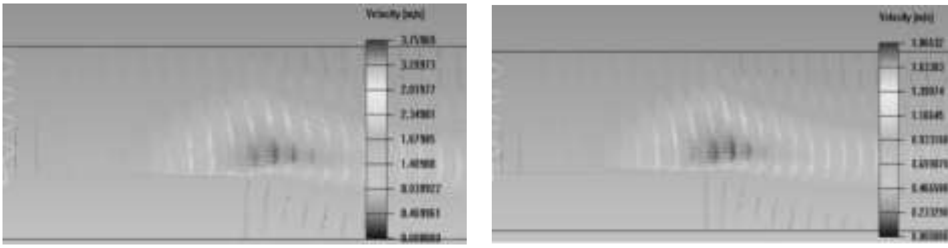


Fig. 4. Variation of Nusselt Number with Reynolds number for elliptical bottom heat sink



(a) (b)

Fig. 5. Vector plot for impinging velocity of a) 0.5m/s b) 1 m/s for R3 configuration

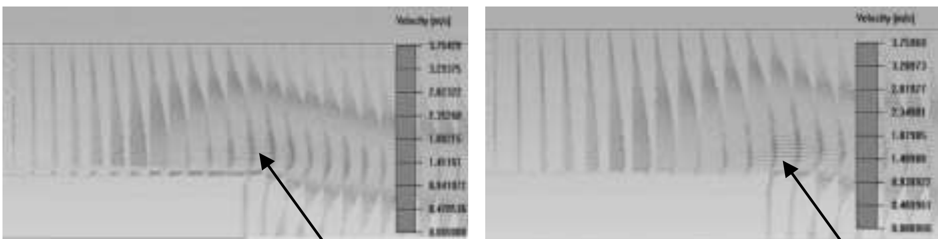
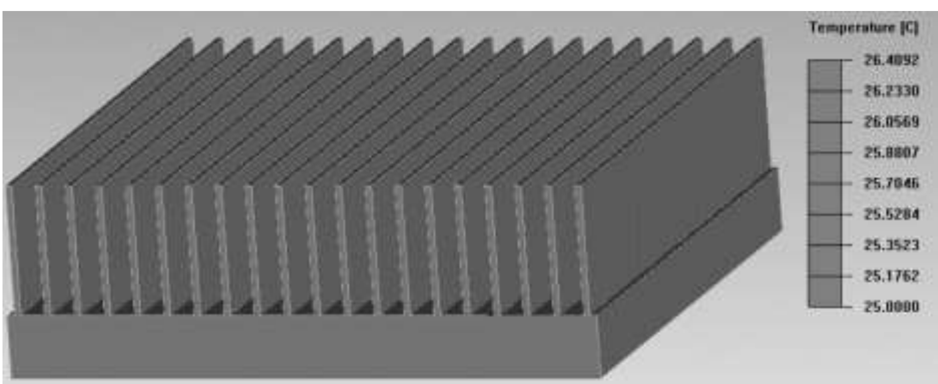
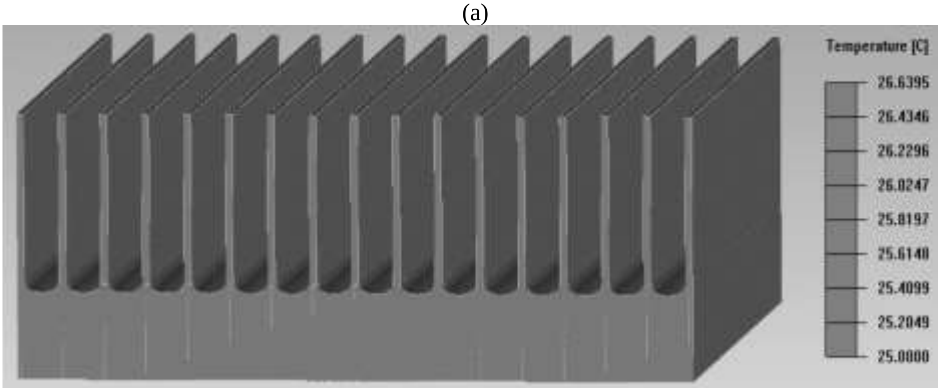


Fig. 6. Vector plot for impinging velocity of 1m/s (a) E3 configuraion, (b) R3 configuration



(a)



(b)

Fig. 7. Temperature contours for a) R3 and b) E3 for impinging velocity of 0.5m/s

3.2 Effect of impingement velocity and geometry of fin on heat sink thermal resistance

The thermal performance of the heat sink subjected jet impingement can be evaluated in terms of heat sink thermal resistance. The effect of impingement velocity (in terms of Reynold's number) on thermal resistance for rectangular (R1, R2 and R3) and elliptical bottom (E1, E2 and E3) heat sink configurations are as shown in Fig.8 and 9 respectively. It is observed that for all heat sink configurations, thermal resistance decreases as Reynolds number increase. This is because convective heat transfer coefficient increase with higher inlet velocity. As impinging velocity increases, source temperature decreases, hence the difference between maximum temperautre (T_b) and inlet air temperature (T_{in}) also decreases. For a given value of heat input, according to expression (11), thermal resistance of heat sink decreases. Additionally, decrease in the thermal resistance with increase in Reynolds number, declines. It means that there is limitation to improve the thermal resistance with increase in Reynolds number [16]. In order to increase the thermal performance of heat sink, an optimal combination of heat sink parameters is required. As the gap between the fin increases, the impinging jet can enter easily the heat sink increasing the overall heat transfer coefficient. This leads to increase in thermal resistance. However, thermal resistance of R3 was between that of R1 and R2 (Fig.8). This may be due to high velocity causing stronger momentum of the flow exiting the nozzle. As a result higher amount of fluids is forced to enter the space between the fins to exchange heat with the heat sink, which decreases the thermal resistance [16]. In case of elliptical bottom fins, highest reduction of thermal resistance can be found for fins with minimum spacing of 2.27 mm as shown in Fig.9.

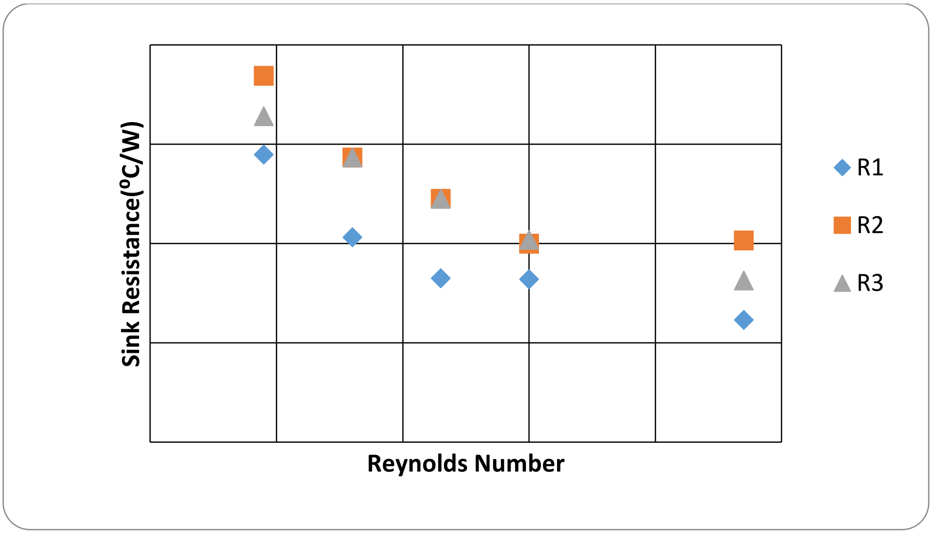


Fig. 8. Variation of heat sink thermal resistance having rectangular bottom

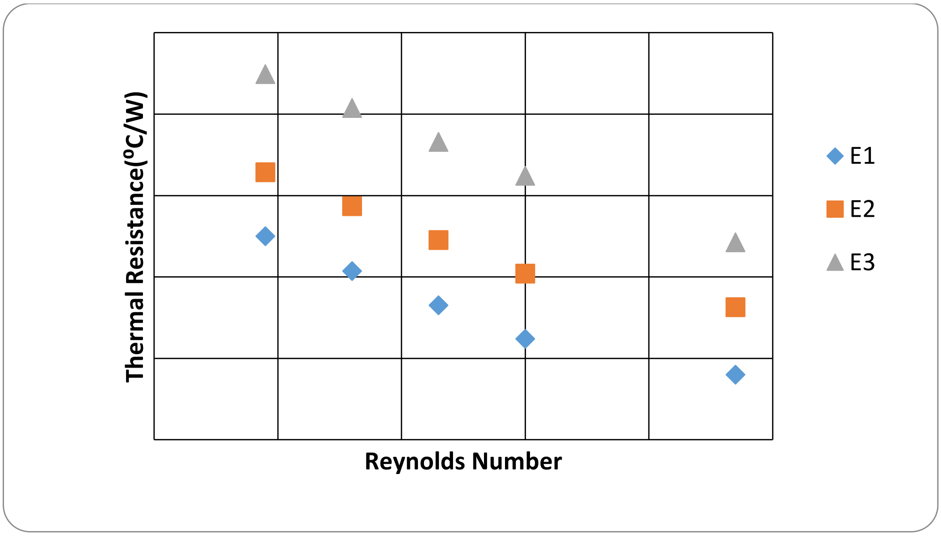


Fig. 9. Variation of heat sink thermal resistance having elliptical bottom

3.3 Validation of Pressure Drop

The pressure drop obtained for plate heat sink with elliptical and flat bottom was reasonably validated with available experimental results of Z. Duan et al[14]. These authors had carried out experimental measurements at seven different velocities in the plenum chamber (0.4 m/ s,0.5 m/ s, 0.6 m/ s, 0.7 m/ s, 0.8 m/ s, 0.9 m/ s, and 1.0 m/ s) and six different impingement inlet widths, 5%L, 10%L, 25%L 50%L, 75%L, and 100%L. Fig.10 predicts the comparison of numerically obtained pressure drop for rectangular fin heat sink (R1) subjected to air flow impingement (100% opening) and that obtained by Z. Duan et al [14]. The pressure drop was found to be in fair agreement with experimental data and within error of ~10%.

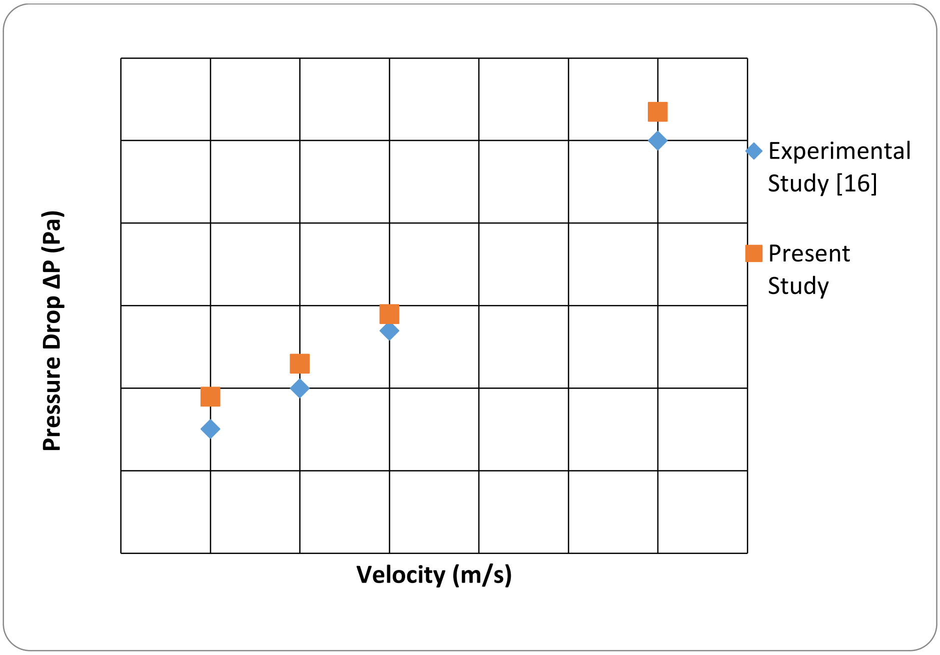


Fig. 10. Validation of Pressure drop with available experimental results [**16**]

Fig.11 predicts the pressure drop for rectangular and elliptical bottom heat sink for all the three configurations. It was observed that pressure drop for elliptical bottom heat sink (E1) was lower when compared to that of rectangular bottom heat sink (R1). With the increase in interspacing within the fins, pressure drop of elliptical bottom heat sink (E2) was found higher for all impingement velocities except 0.7m/s. At the interspacing of 6.2mm, the pressure drop was found to be same.

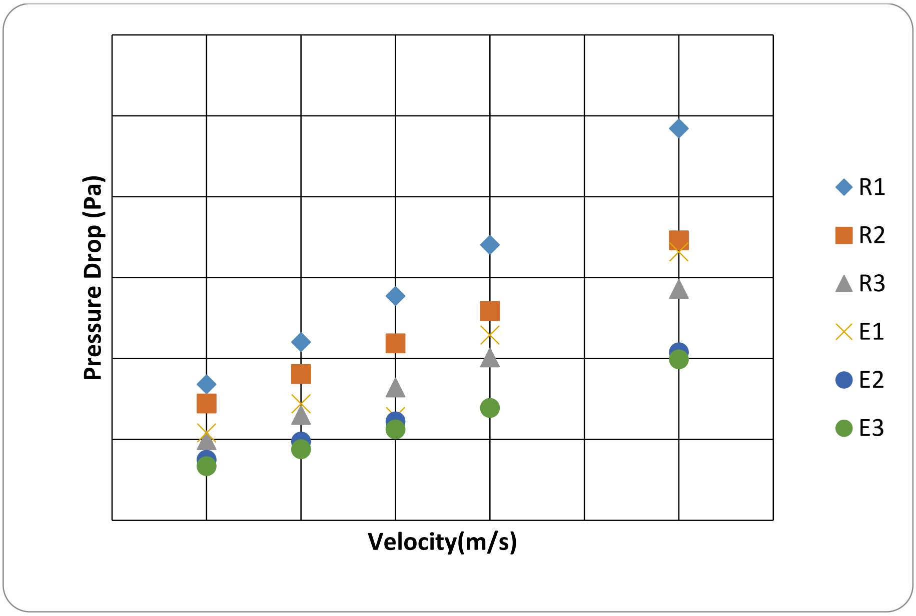


Fig. 11. Comparison of pressure drop for rectangular and elliptical bottom heat sinks

4.0 Conclusions

In this research study, the thermal performance of flat bottom plate fin heat sink was compared with that of elliptical bottom. The effect of fin geometry and impinging velocity on Nusselt number and thermal resistance was reported using ANSYS ICEPAK R19 package. Increasing the space between elliptical bottom fins was found to be more effective than the rectagular one which was attributed to the generation of smaller vortex causing smooth flow in the elliptical channels. The thermal resistance of rectangular fins with highest inter space was found between that of R1 and R2 due to high velocity of air causing stronger flow momentum from nozzle exit. In case of elliptical bottom fins, highest reduction of thermal resistance was found for fins with minimum inter spacing of 2.27 mm. With the increase in interspacing within the fins, pressure drop of elliptical bottom heat sink was found higher for all impingement velocities except 0.7m/s.

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