# A Comparative Analysis of Natural Convection between Horizontal and Vertical Heat Sink using CFD

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*Abstract*— **The Natural convection heat transfer is often augmented by the provision of rectangular fins on vertical or horizontal surfaces in numerous electronic applications, transformers, and motors. The recent trend in the electronic industry is miniaturization, overcoming from overheat issue of component due to reduction in surface area offered for heat dissipation in such way that effective thermal management can be attained. Therefore heat transfer from fin arrays has been examined extensively, both experimentally and analytically.**

**A heat sink helps keep a device at a temperature below its specified maximum operating temperature This work is concerned with the comparative natural convective heat transfer analysis of a Vertical and horizontal heats sink with different number of fins and Fin spacing in which convective heat transfer has been examined and with the help of thermo physical characteristic performance is predicted by using Finite element volume tool ANSYS - Fluent, where simulation is being done. The goal is to carry out evaluating heat transfer within the heat sink using different Density based module such as incompressible and Boussinesq model. The FEV results are validated with well published results in the literature and furthermore with experimentation. The present Finite Element Volume (FEV) and computational results show good agreement.**

***Key Words: Tap Hole, CFD, Heat Transfer,*** *FEV.*

## I. INTRODUCTION

In the present era, the trend to design electronic products becomes thinner, lighter, shorter, & smaller. Due to the actuality that shrinking in the dimension of these electronic components will consequence in a drastic increase in the heat generation rate when evaluating with previous products. For this reason, an efficient cooling system to remove the high heat generation and consequently maintains the reliability and stability of the products, have gained much attention.

The heat sink component is the most common heat exchanger for CPUs and has been extensively exercised in order to provide cooling utility for electronic components. The conventional heat sink module utilized the natural as well as forced convection cooling technique; dissipate heat from CPUs to the ambient air. The combination of the heat sink and fan design usually involved in this forced convection cooling technique.

## II. LITERATURE SURVEY

C.J. Kobus, T. Oshio 2005 investigate the effect of thermal radiation on the thermal performance of heat sink having pin fin array by theoretical and experimental approach. In order to investigate the ability of influence of thermal radiation on the thermal performance a new coefficient effective radiation heat transfer is collaborated. For validation of theoretical model it is matched with experimental data.

Hung-Yi Li et al. 2007 done their investigation on plate-fin heat sinks numerically and experimentally. Impingement cooling is used by amending, the Reynolds number (Re), the impingement distance (Y/D), and the fin dimensions. The results shows that heat transfer is enhance by the heat sink with increasing the impinging Reynolds number

G. Hetsroni et al. 2008, Natural convection heat transfer in metal foam strips with two porosities examined experimentally. Image processing of the thermal maps were used for evaluation of non-equilibrium temperature distribution for surface along with inner area of the metal foam. Augmentation in heat transfer at natural convection was found 18–20 times with respect to the flat plate.

Goshayeshi and Ampofo 2009 conduct numerical studies on vertical fins, attached with the surface. Natural convective heat transfer find out from heated plane, which is kept into air with horizontal and vertical surface. Results show that vertical plate with dimensionless form delivers best performance for the natural cooling.

Dong-Kwon Kimet al. (2009) compared the thermal performances of two types of heat sinks i.e.: plate-fin and the second is pin-fin. By their investigation results propose, a volume averaging approach based model for envisaging the pressure drop and the thermal resistance.

Burak and Hafit 2009 developed expression for prediction of the optimal fin spacing for vertical rectangular fins with rectangular base. The correlation for predicted on the basis of experimental data.

Li and Chao 2009 investigated the performance of plate-fin heat sinks with cross flow. The effect of different parameters like the fin width, fin height, Re. number of cooling air on the thermal resistance and the pressure drop of heat sinks were studied.

Naidu et al.2010 investigates by both experimentally and theoretically to find the outcome of inclination of the base of the fin array on heat transfer rate.

Sable et al. 2010 investigated the natural convection of a vertical heated plate with a multiple v- type fins having ambient air surrounding. The mica gladded Nichrome element is inserted between two base plates.

Mahmoud et al. 2011 conducted an experiment to investigate the effects of micro fin height and spacing on heat transfer coefficient for a horizontally mounted heat sink under steady state natural convection conditions, fin height ranging from 0.25-1.0 mm and fin spacing from 0.5 to 1.0 mm was taken.

Cheng-Hung al. 2011 developed a three-dimensional heat sink design to estimate the optimum design variables. Levenberg–Marquardt Method (LMM) was used and commercial code CFD-ACE+ was developed. Temperature distributions are dignified by using thermal camera for the optimal heat sink modules and results are compared with the numerical solutions to validate the design

Fahiminia et al. 2011investigated the laminar natural convection on vertical surfaces computationally. The CFD simulations are carried out using fluent software. Governing equations are solved using a finite volume approach. Relation between the velocity and pressure is made with SIMPLE algorithm

Ayla dogan et al. 2012 performed numerical investigation to find out the natural convection heat transfer from an annular fin on a horizontal cylinder and present correlation for the optimum fin spacing depending on Rayleigh number and fin diameter.

Mateusz al.2013 used water and copper oxide nano fluids for cooling heat sink of PC Processor. The commercial package ANSYS Fluent 13 was employed to generate a CFD heat transfer simulation. The experimental results were used to validate the numerical model of the analyzed system.

Farhad et al.2013 solved .Navier–Stokes equations and RNG based k- turbulent model for array of solid and perforated fins mounted in vertical flat plates used to predict turbulent flow parameters. Flow and heat transfer features are presented for Re. no. from 2×104 - 3.9 × 104.Prandtl numbers was taken as 0.71. Numerical simulation is validated by compare with experimental results.

Qarnia and Lakhal 2013 using numercail approach investigate the heat transfer by natural convection during the melting of a phase change material. A mathematical model was developed to investigate the thermal performance of PCM based-heat sink.

Mehran al. 2014 examined numerically and experimentally, Steady-state external natural convection heat transfer from vertically-mounted rectangular interrupted fins. FLUENT software is employed to develop a 2-D numerical model of fin interruption effects. An experimental numerical parametric study was performed to investigate the effects of fin spacing, and fin disruption. A new compact correlation is proposed for calculating the optimum interruption length.

Emrana and Islama 2014 performed a three-dimensional numerical simulation is in order to investigate the flow dynamics and heat transfer characteristics in a microchannel heat sink. A commercial CFD code was employing with finite element method to numerical simulation. For the accuracy of results Mesh independence test was performed.

Younghwan and Kim 2015 investigated analytically thermal performance of optimized plate-fin and pin-fin heat sinks with a vertically oriented base plate. A new correlation of the heat transfer coefficient is proposed and validated experimentally to optimize pin-fin heat sinks.

## III. Mathematical Modelling

The governing equations for heat sink problem are Navier-Stokes along with the energy equation. The Navier-Stokes equation is applied to incompressible flows and Newtonian fluids, including the continuity equation and the equations of conservation of momentum on the x and y

According to equations

$$\frac{\partial u_2}{\partial t} + u_1 \frac{\partial u_1}{\partial x_1} + u_2 \frac{\partial u_1}{\partial x_2} =$$

$$-\frac{1}{\rho}\frac{\partial \rho}{\partial x_2} + \nu\left(\frac{\partial^2 u_1}{\partial x_1^2} + \frac{\partial^2 u_1}{\partial x_2^2}\right) + g\beta(T - T\infty)$$

$$\frac{\partial u_1^*}{\partial x_1^*} + \frac{\partial u_2^*}{\partial x_2^*} = 0$$

$x_1$ momentum equation

$$\frac{\partial u_1^*}{\partial t*} + u_1^* \frac{\partial u_1^*}{\partial x_1^*} + u_2^* \frac{\partial u_1^*}{\partial x_2^*} = -\frac{\partial p*}{\partial x_1^*} + \Pr\left(\frac{\partial u_1^*}{\partial x_1^*} + \frac{\partial u_1^*}{\partial x_2^*}\right)$$

$x_2$ momentum equation

$$\frac{\partial u_2^*}{\partial t*} + u_1^* \frac{\partial u_2^*}{\partial x_1^*} + u_2^* \frac{\partial u_1^*}{\partial x_2^*} = -\frac{\partial p*}{\partial x_2^*} + \Pr\left(\frac{\partial u_2^*}{\partial x_1^*} + \frac{\partial u_2^*}{\partial x_2^*}\right)$$

$$+Gr \Pr^2 T*$$

Energy equation

$$\frac{\partial T^*}{\partial t^*} + u_1^* \frac{\partial T^*}{\partial x_1^*} + u_2^* \frac{\partial T^*}{\partial x_2^*} = \left( \frac{\partial^2 T^*}{\partial x_1^{*2}} + \frac{\partial^2 T^*}{\partial x_2^{*2}} \right)$$

Where Gr is the Grashof number given as

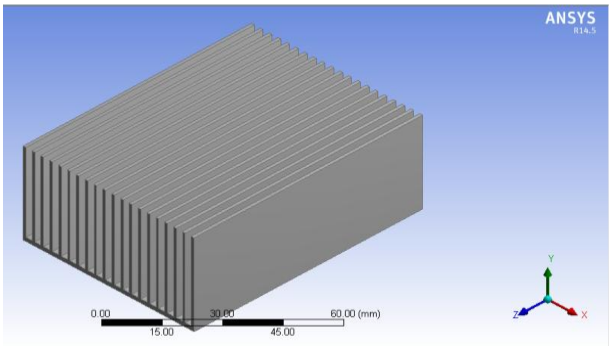
$$Gr = \frac{g\beta\Delta TL^3}{\nu^2}$$

Often, another non-dimensional number called the Rayleigh number is used in the calculations. This is given as

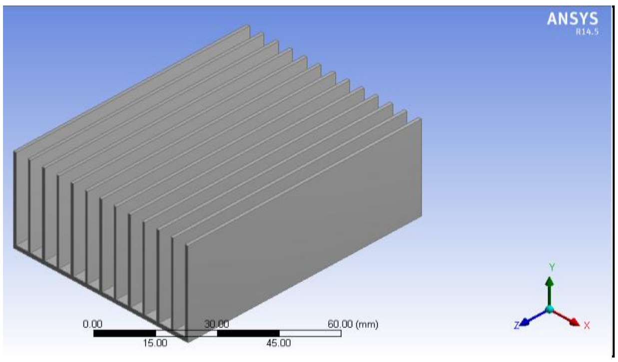
$$Ra = Gr\Pr = \frac{g\beta\Delta TL^3}{\nu\alpha} \qquad Nu = \frac{hl}{k}$$

## IV. METHODOLOGY

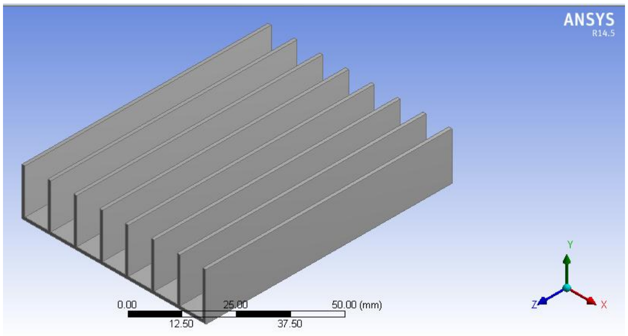
The ANSYS 14.5 finite element program was used to analyze natural convection in differentially heated enclosures. For this purpose, the key points were first created and then line segments were formed. The lines were combined to create a surface. Finally, this surface is provided thickness model is made. We modeled 6 different geometrical configurations of the heat sink with different number of fins and fin spacing configuration. The heat sink was discretized into 21788 elements with 44520 nodes. Heat sink boundary conditions can also be applied in the mesh section through naming the portion of modeled sink i.e. Base, Base Top, Fins, Interior.



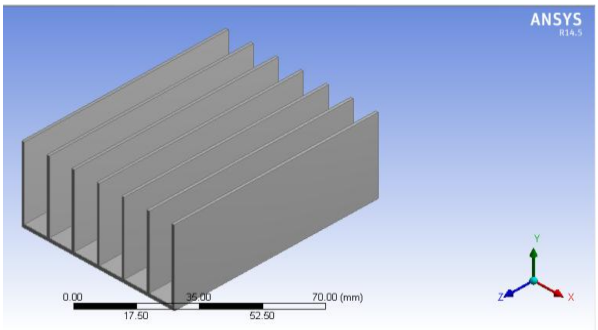

20 Fins



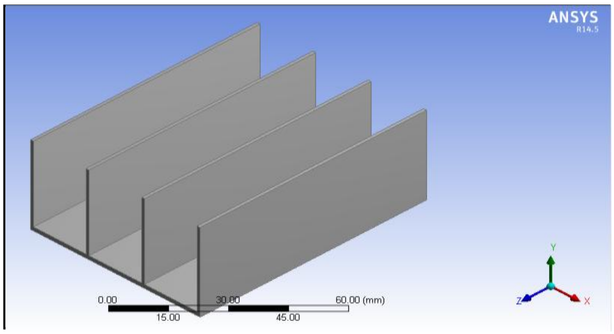

13 Fins



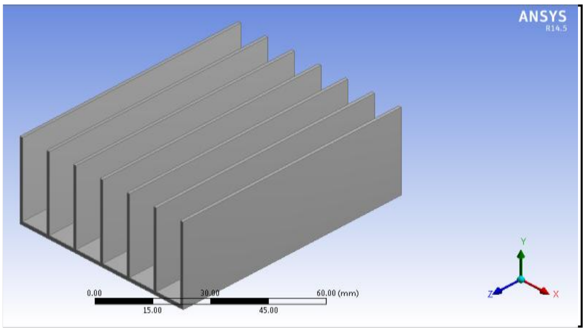

8 Fins




7 Fins

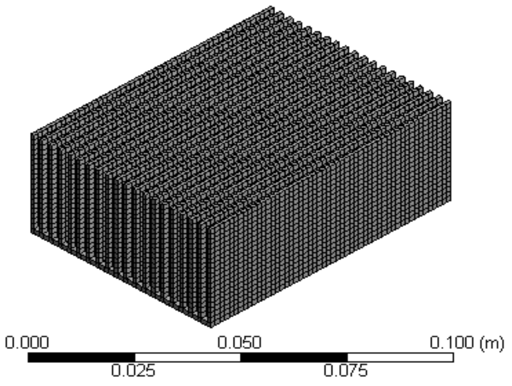



4 Fins

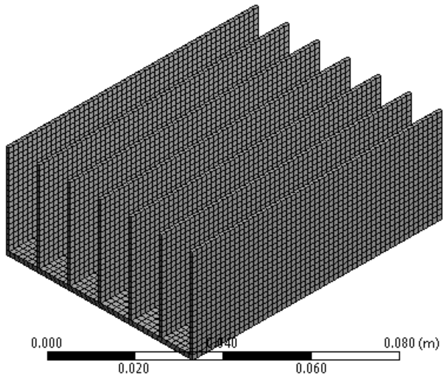



5 Fins

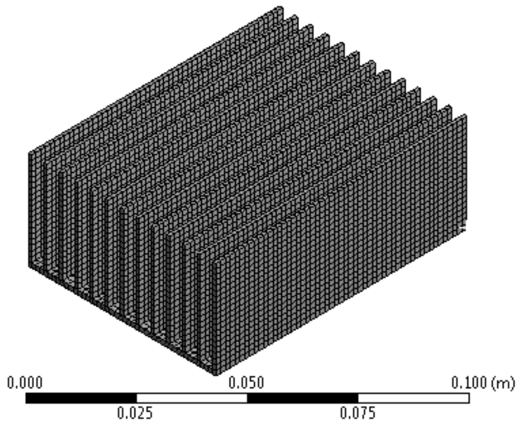
Figure 1 Geometry Modeling of Heat Sink



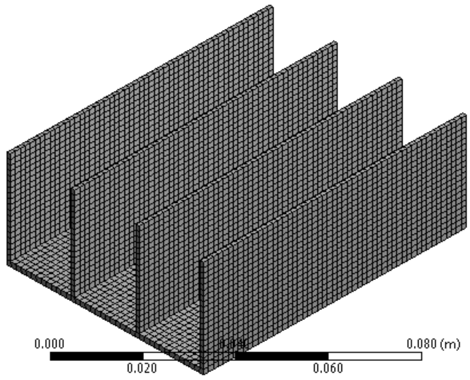

20 Fins



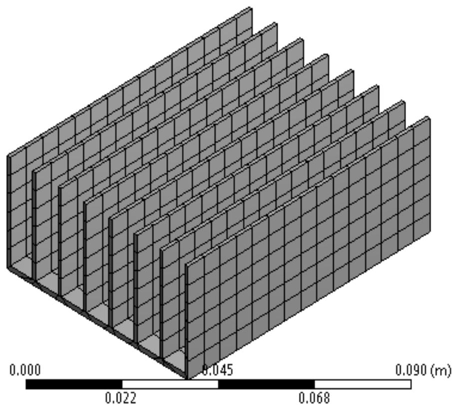

7 Fins



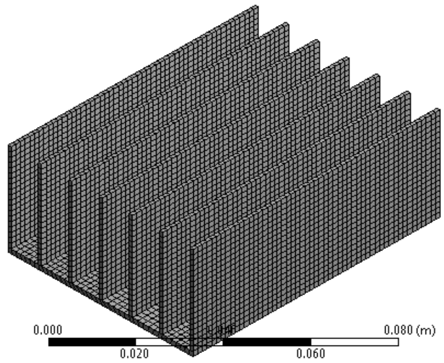

13 Fins




4 Fins




8 Fins




5 Fins

Figure 2 Mesh Model

Table 1 Dimensions of the fin configuration

| Fin Length (L) in mm | Fin width (W) in mm | Fin thickness (t) in mm | Base thickness (d) in mm |
|---|---|---|---|
| 80 | 59.8 | 1 | 1.4 |

| Fin height (H) in mm | Fin spacing (s) in mm | Number of fins (n) |
|---|---|---|
| 29.2 | 2.1 | 20 |
| 29.2 | 3.9 | 13 |
| 29.2 | 7.4 | 8 |
| 29.2 | 8.8 | 7 |
| 29.2 | 13.7 | 5 |
| 29.2 | 18.6 | 4 |

Table 2 The boundary Condition of Heat Sink

| Considered Temperatures for Analysis of Heat Sinks | | | | |
|---|---|---|---|---|
| Heat Sink Base Temperature in K | 350 | 375 | 400 | 430 |
| Ambient Temperature in K | 300 | 300 | 300 | 300 |
| Mean Temperature in K | 325 | 337.5 | 350 | 365 |

| Various properties applied of air for the analysis of Heat Sinks at different temperature differences between base plate of heat sink and temperature of flowing fluid (air) | | | | |
|---|---|---|---|---|
| Temperature difference, ΔT (K) | 50 | 75 | 100 | 130 |
| Kinematic viscosity, ν ($m^2/s$) | 1.81E-05 | 1.94E-05 | 2.06E-05 | 2.0209E-05 |
| Thermal Conductivity, k (W/m-K) | 0.02816 | 0.028836 | 0.03003 | 0.031045 |
| Dynamic viscosity, µ (kg/m-s) | 1.96E-05 | 2.03E-05 | 2.08E-05 | 2.1466E-05 |
| Specific heat capacity, $C_p$ (J/kg-K) | 1006.3 | 1008.4 | 1009.3 | 1010.2 |
| Prandtl Number, Pr | 0.701122 | 0.708566 | 0.697402 | 0.69850067 |
| Coefficient of Thermal Expansion, β ($K^{-1}$) | 0.003077 | 0.002963 | 0.002857 | 0.00273973 |
| Density, ρ ($kg/m^3$) | 1.086 | 1.0475 | 1.009 | 0.96655 |
| Diffusivity, α ($m^2/s$) | 2.56E-05 | 2.74E-05 | 2.92E-05 | 3.2478E-05 |
| Density Module | Boussinesq Model | Boussinesq Model | Boussinesq Model | Boussinesq Model |

| Properties of Heat Sink Material | |
|---|---|
| Material of Heat Sink | Aluminum |
| Density, ρ ($kg/m^3$) | 2719 |
| Specific heat capacity, Cp (J/kg-K) | 871 |
| Thermal Conductivity, k (W/m-K) | 202.4 |

## V. Results and Discussion

### A. Validation

The governing equations of the problem were solved, numerically, using a Element method, and Finite Volume Method (FVM) used in order to calculate the Thermodynamic characteristics of a Heat sink. As a result of a grid independence study, a grid size of $10^5$ was found to model accurately the Thermodynamic performance characteristics as described in the corresponding results. The accuracy of the computational model was verified by comparing results from the present study with those obtained by Fahiminia [10], Goshayeshi [6], Analytical and FVM (Finite Volume Method) results.

Table 3 Grid Independence Test at ΔT=130K for different Heat sink configuration

| Total Convective Heat Transfer in (W) | | |
|---|---|---|
| Number of Fin (n) and Fin Spacing (s) in mm | Mesh Element | Present (FEV) |
| 20, 2.1mm | 44520 | 29.57845 |
| | 36520 | 29.56967 |
| | 16640 | 29.42761 |
| | | |
| 13, 3.9mm | 43750 | 47.35242 |
| | 31752 | 47.34912 |
| | 11328 | 47.21461 |
| | | |
| 4, 18.6mm | 12636 | 30.51392 |
| | 10528 | 30.50927 |
| | 4416 | 30.48760 |

Table 4 Comparative Optimum Fin Spacing at different Temperature Gradient

| Temperature difference (ΔT) in K | Optimum Fin Spacing (s) in mm | |
|---|---|---|
| | Ref.[10] | Present (FEV) |
| 50 | 6.42 | 6.40 |
| 75 | 6.19 | 6.15 |
| 100 | 6.04 | 6.00 |
| 130 | 5.84 | 5.50 |

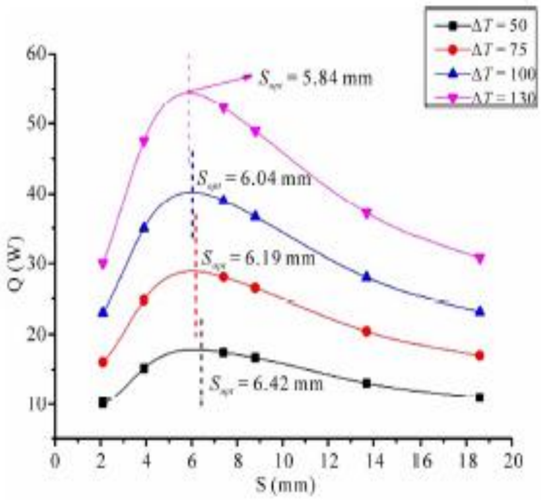



Figure 3 Variation of convective rate with base-to-ambient temperature difference at H = 29.2 mm and L = 80 mm from Ref. [10]

Table 3 shows the Grid independence result of FVM obtained from the ANSYS tool. It has been seen that the obtained result for different mesh element shows good convergence for difrrent number of fins and fin spacing.

figure 3 and 4 shows the Comparative Optimum Fin Spacing at different Temperature difference. It has been observed that the obtained result shows the same trend so that the results are suitably verified and the minute variation in result is due to grid sizing, operating condition, geometrica paramters, etc.

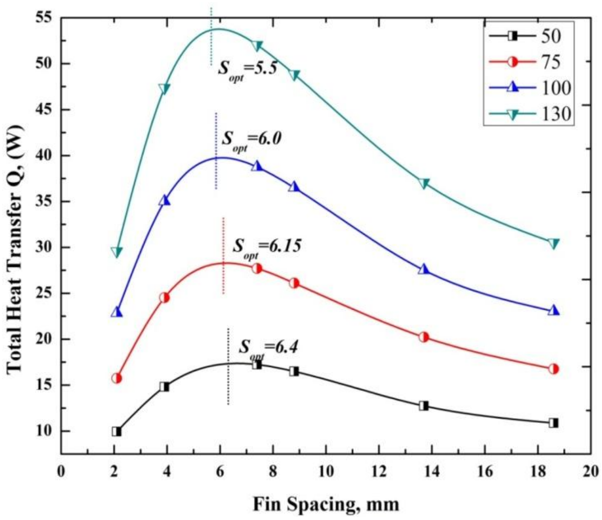



Figure 4 Variation of convective rate with base-to-ambient temperature difference at H = 29.2 mm and L = 80 mm in present work

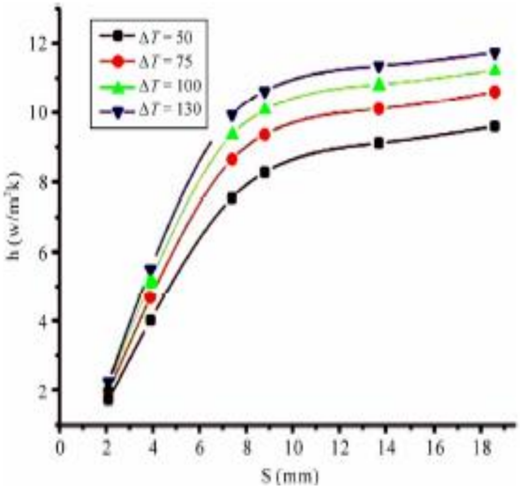



Figure 5 Natural convection heat transfer coefficients for different heat sinks from Ref. [10]

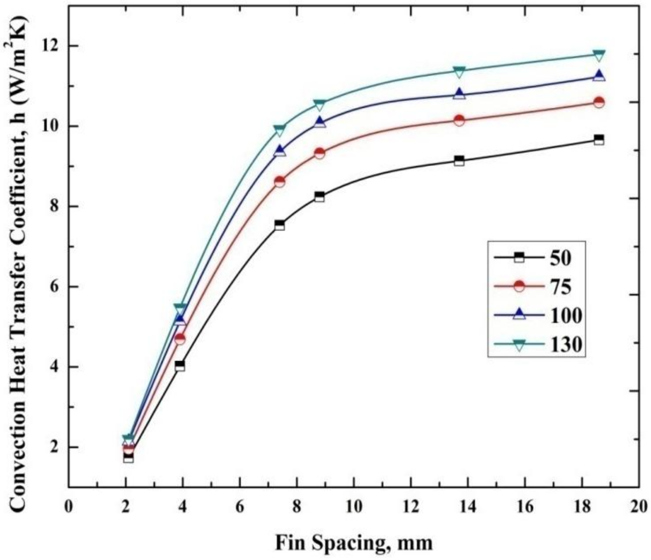



Figure 6 Variation of Convective heat transfer coefficient for different fin Spacing in present work

Figure 6 shows the Variation of Convective heat transfer coefficient for different fin Spacing. It has been observed that on increasing fin spacing as well as temperature gradient convective heat transfer coefficient remarkably increases. This is due to mixing of the boundary layer occurs (that fills up with warm air). However, the obtained results show same trends with the available literature Fahiminia [10] and form the figure 6 also.

Therefore, from this it can be concluded that at high temperature difference between base-to-ambient, leads to higher rates of convective heat transfer coefficient.

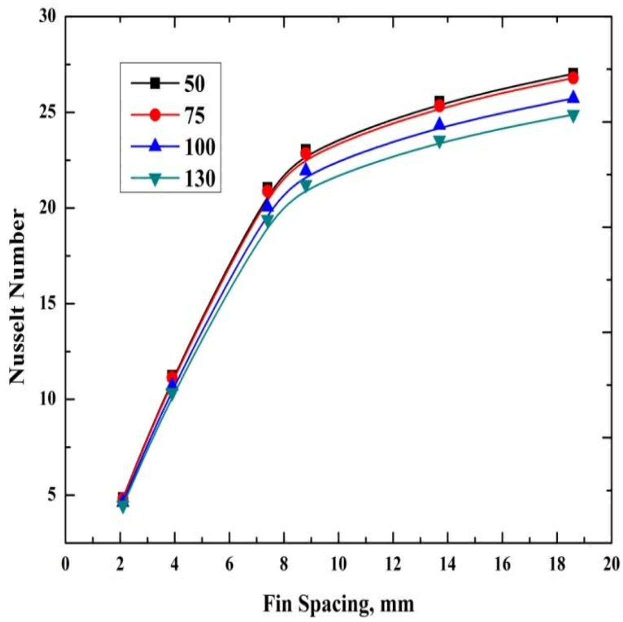


Figure 7 Variation of Nusselt Number for different fin spacing

Figure 7 shows the variation of Nusselt Number for different fin spacing. It has been observed that on increasing fin spacing nusselt number noteworthy increases and monotonous at lower temperature difference. Therefore, widely spaced fins have higher heat transfer coefficient but smaller surface area.

A Case study has been considered in which a heat sink with vertical as well horizontal configuration shown in figure 8 is simulated at same boundary condition and it has been observed that Vertical plate with vertical fins gives the best performance for natural cooling in comparison with a horizontal one this is due to air enters the channel from the lower end, rises as it is heated under the effect of buoyancy, and the heated fluid leaves the channel from the upper end, however this effect can be visualized from the figure 9 to 12 in which closely packed and widely spaced fin are simulated for different temperature difference and the convective heat transfer is significant for vertical configuration at higher temperature difference which is clear from figure 13

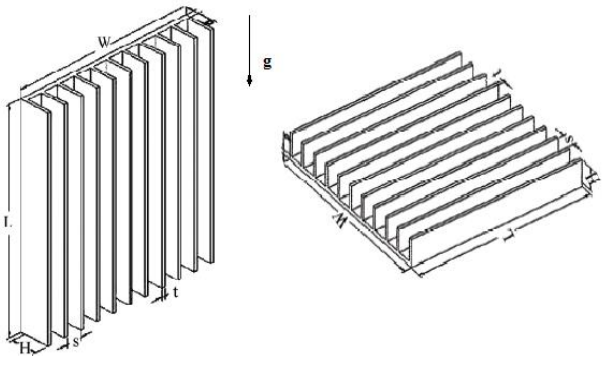


Figure 8 Fin configurations for natural (Free) cooling

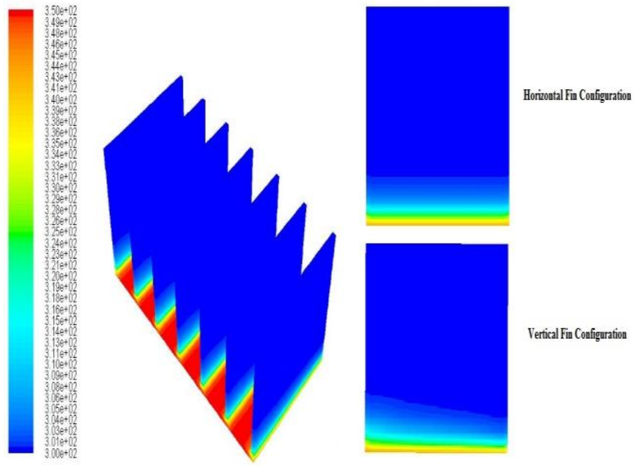


Figure 9 Variation of temperature contour of the heat sink s=8.8 mm and ΔT=50 K

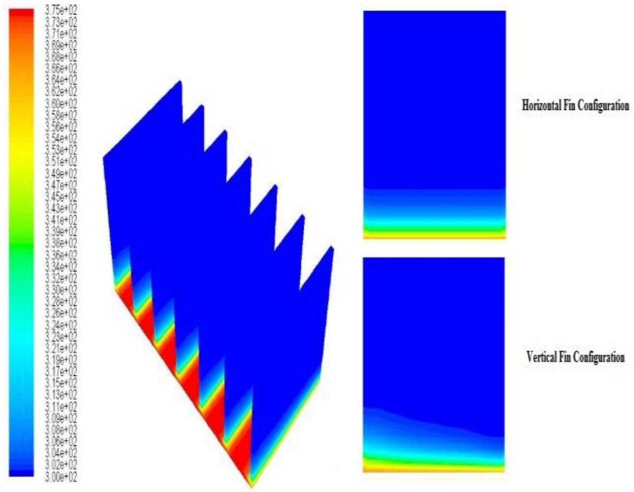


Figure 10 Variation of temperature contour of the heat sink s=8.8 mm and ΔT=75 K

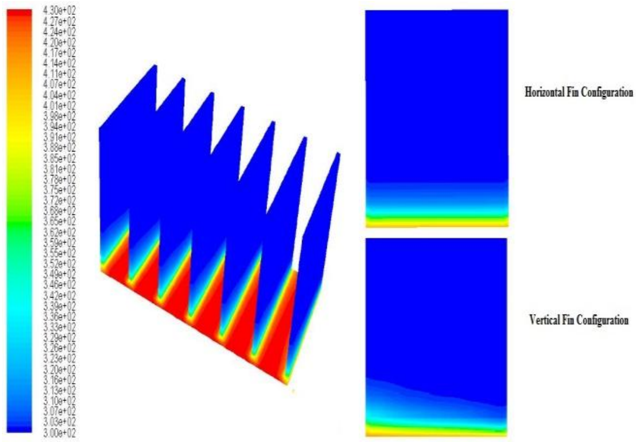


Figure 12 Variation of temperature contour of the heat sink 20 Fins and ΔT=130 K

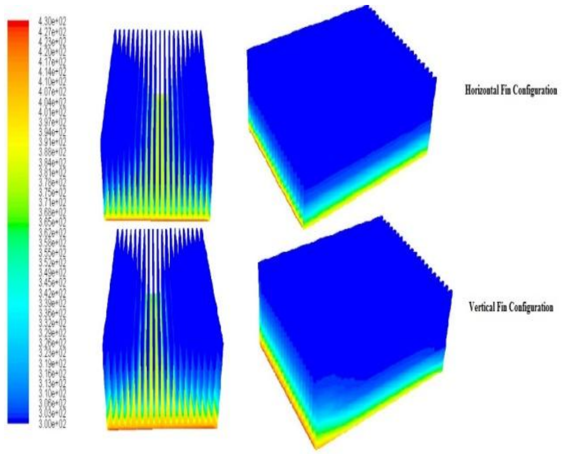



Figure 13 Variation of temperature contour of the heat sink with 20 fins and ΔT=130 K

It has also been observed that an increase in 160% of temperature difference the heat transfer increases by 197.76% and 180% for Vertical and Horizontal fin Configuration respectively at 2.1mm fin spacing (closely packed). Similarly for widely spaced fin configuration i.e. at 18.6 mm fin spacing 180.74% and 164.32% increase in heat transfer for Vertical and Horizontal respectively.

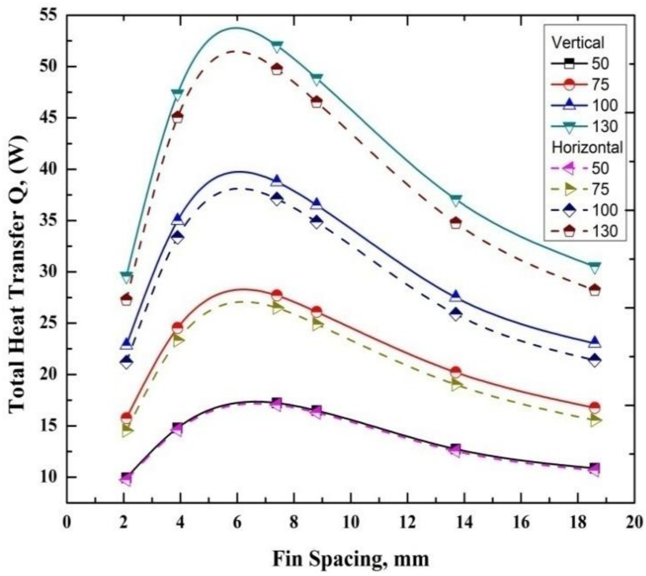



Figure 13 Variation of Convective Heat transfer in both Vertical and Horizontal fin Configuration for Different Fin Spacing

Figure 14, The variation of Convective Heat transfer for different Number of Fins has been observed. And it can be concluded that on increasing number of fins total heat transfer first increases and then start decreasing. This is due to greater surface area for heat transfer but a smaller heat transfer coefficient because of the extra resistance the additional fins introduce to fluid flow through the inter fin Passages.

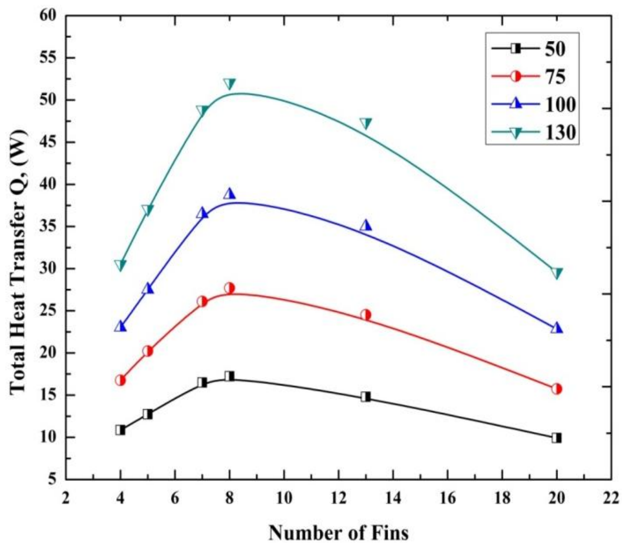



Figure 14 Variation of Convective Heat transfer for different Number of Fins

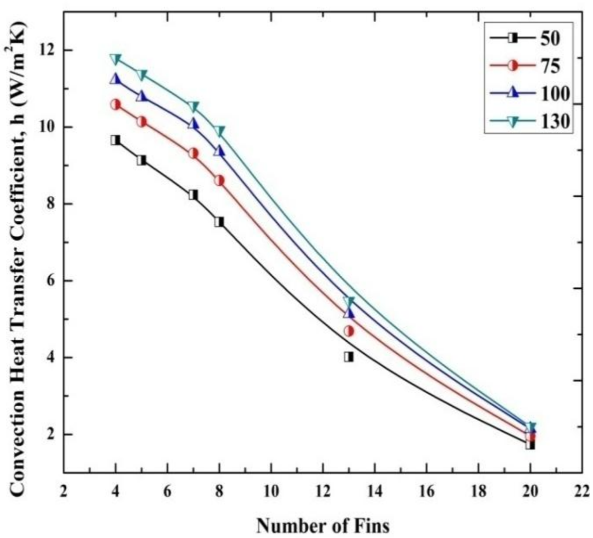



Figure 15 Variation of Convective Heat transfer coefficient for Number of Fins

Figure 15 shows the variation of Convective Heat transfer coefficient for Number of Fins. It can be conclude that on increasing number of fins convective heat transfer coefficient drastically decreases. This is because of the extra resistance the additional fins introduce to fluid flow through the inter fin passages. Therefore, there must be an optimum spacing that maximizes the natural convection heat transfer from the heat sink for a given base area WL, where W and L are the width and height of the base of the heat sink.

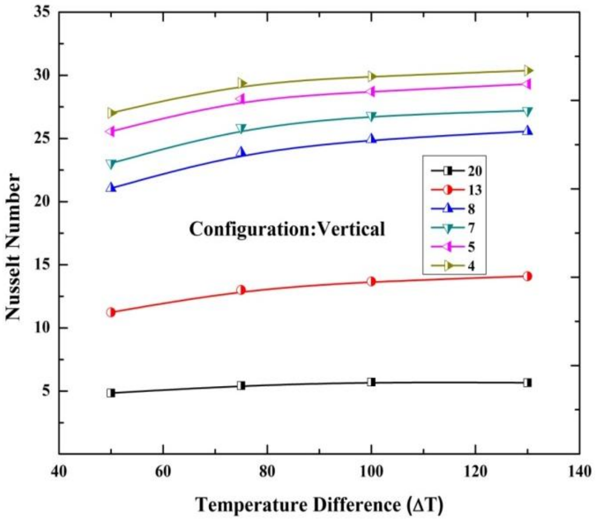



Figure 16 Variation of Nusselt Number in vertical Configuration for different Temperature Gradient

Figure 16 illustrate the Variation of Nusselt number in Vertical configuration for different Temperature Gradient. It has been observed that on increasing temperature difference (Gradient) nusselt number significantly increases and it is monotonous at lower number of fins. This is due to significant spacing between the fins which yields high convective coefficient at high temperature gradient.

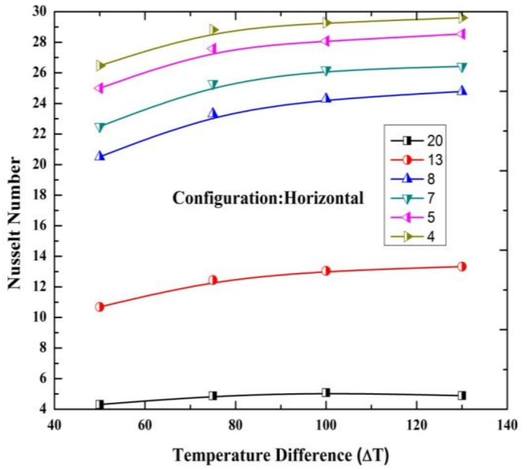



Figure 17 Variation of Nusselt Number in Horizontal Configuration for different Temperature Gradient

Figure17 illustrate the Variation of Nusselt number in Horizontal configuration for different Temperature Gradient. It has been observed that the graph shows same trend for Horizontal one but in lower limit as compared with vertical configuration. From it can be conclude that convective coefficient is a strong function of nusselt number.

## VI. CONCLUSION

From the present analysis of a heat sink following conclusion has been drawn which significantly affects the performance of heat sink

On increasing fin spacing convective heat transfer first increases up to optimum spacing and then starts decreasing. This is due to higher heat transfer coefficient but lesser surface area.

As temperature difference increases convective heat transfer coefficient also increases.

Widely spaced fins have high heat transfer coefficient at corresponding higher temperature difference.

Vertical configuration heat sink has better performance on comparison with horizontal one.

Nusselt number linearly increases as the temperature difference increases.

Increasing fin spacing Nusselt number increases

Convective heat transfer coefficient is a strong function of nusselt number.

At a specified temperature difference and fin height, the convective heat transfer rate increases with increasing fin spacing till it reaches optimum spacing and then with further increasing fin spacing heat transfer rate decreases.

On increasing fin height by employing longer fins, but with a fixed volumetric flow rate performance may actually decrease with fin height.

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Nomenclature

| | |
|---|---|
| g | Acceleration due to gravity, m $s^{-2}$ |
| k | Thermal conductivity, W $m^{-1}$ $K^{-1}$ |
| L | Side of the Wall, m |
| N | Normal direction on a wall |
| Nu | Local Nusselt number |
| P | Dimensionless fluid pressure |
| Pr | Prandtl number |
| Ra | Rayleigh number |
| Re | Reynolds number |
| Gr | Grashof number |
| T | Fluid temperature, K |
| u | x component of velocity |
| U | x component of dimensionless velocity |
| v | y component of velocity |
| V | y component of dimensionless velocity |
| X | Dimensionless distance along x coordinate |
| Y | Dimensionless distance along y coordinate |
| $\tau$ | stress |
| $\tau_p$ | non-dimensional ramp time |
| $\varepsilon$ | Emissivity |
| z | Axial Coordinate |

Greek Symbols

| | |
|---|---|
| $\alpha$ | Thermal diffusivity, $m^2$ / s |
| $\beta$ | Thermal expansion coefficient at constant pressure, $K^{-1}$ |
| $\rho$ | Density, kg $m^{-3}$ |
| $c_p$ | Specific heat coefficient, J/kg-K |
| $c_v$ | Specific heat coefficient, J/kg-K |
| $\varphi$ | Angle |