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Computational Analysis of Heat sink or Extended Surface

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Abstract—This paper focuses on comparative natural convection of an heat sink with vertical and horizontal configuration. The heat transfer characteristics has been examined and compared and the obtained results from finite element approach. The governing equation of heat sink are solved and simulated by using FEV tool namely ANSYS Fluent where parametric analysis has been carried out in which effect of various number of fin configuration has been studied. The validation of heat transfer characteristics has been done with previous published work and the results are within acceptable range.

***Key Words: Heat Sink, CFD, Heat Transfer*.**

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.

Literature Survey

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.

Mehran al. 2014 examines 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.

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.

Farhad et al.2013 solve .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.

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.

Ayla dogan et al. 2012 perform 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.

Fahiminia et al. 2011investigate 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

Cheng-Hung al. 2011 develops 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

Mahmoud et al. 2011 conduct 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.

C.J. Kobus, T. Oshio 2005 investigate the 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 there investigation on plate-fin heat sinks by numerically and experimental. Impingement cooling is used by amending, the Reynolds number (Re), the impingement distance(Y/D), and the fin dimensions. The results appearance 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 experimentaly. Image processing of the thermal maps was used to 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 they 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 investigate 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 investigate 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.

Younghwan and Kim 2015 investigate 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.

Mathematical Modelling

The governing equations for heat sink this problem are those of Navier-Stokes along with the energy equation. The Navier-Stokes equations are 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

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

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

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

x1 momentum equation

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

x2 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

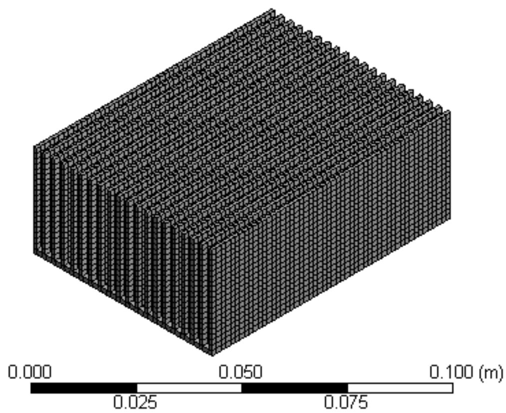
$$Gr=\frac{g\beta\Delta TL^3}{\nu^2}$$

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

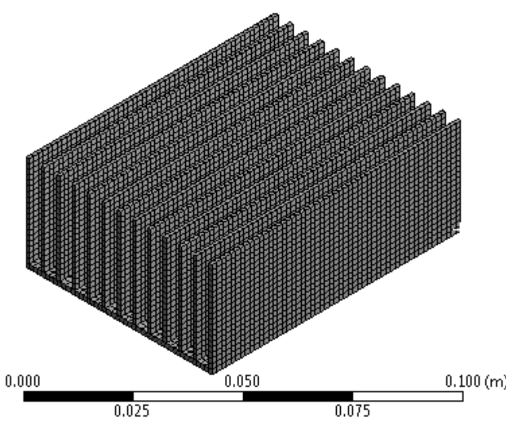
$$Ra=Gr\Pr=\frac{g\beta\Delta TL^3}{\nu\alpha} \qquad Nu=\frac{hl}{k}$$

Methodology

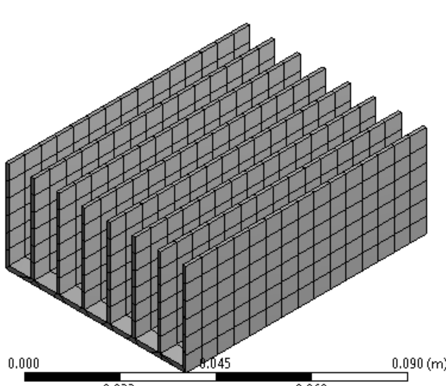
The ANSYS 14.5 finite element program was used natural convection in differentially heated enclosures. For this purpose, the key points were first created and then line segments were formed. The lines were combined to create a surface. Finally, this surface is provided thickness model is made. We modelled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



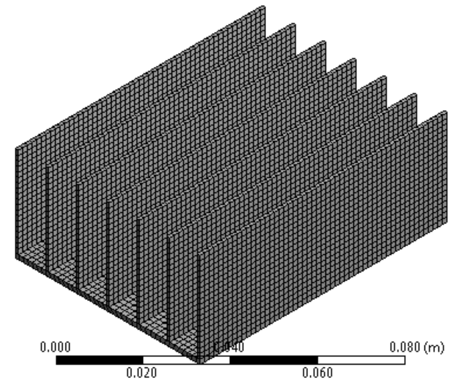
20 Fins



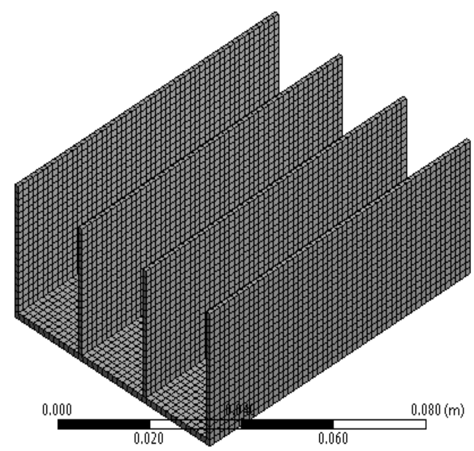
13 Fins



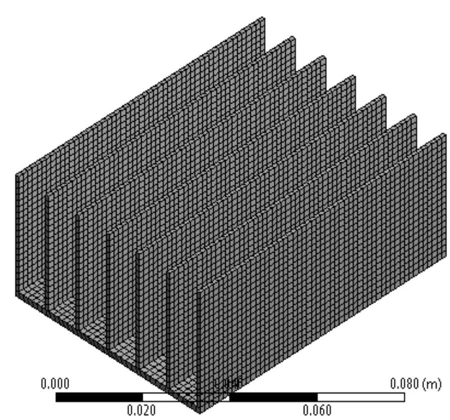
8 Fins



7 Fins



4 Fins



5 Fins

Figure 1 Geometrical and Mesh Model

Table 1 the boundary Condition of Heat Sink

Boundary Condition	Value
Heat Sink Base Temperature, K	350, 375, 400, 430
Ambient Temperature, K	300
kinematic viscosity, ν(m²/s)	1.81E-05
Thermal Conductivity, k(w/mK)	0.02816
Dynamic viscosity, μ(kg/ms)	1.96E-05
Specific heat capacity, Cp(j/kgK)	1006.3
Prandtl Number, Pr	0.701122
Coefficient of Thermal Expansion, β	0.003077
Density, ρ	1.086
Diffusivity, α	2.56E-05
Density Module	Boussinesq Model

Results and Discussion

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 105 was found to model accurately the Thermodynamic performance characteristics are 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 results.

Table 2 Grid Independence Test at ΔT=130K for different Heat sink configuration

Total Convective Heat Transfer in (W)		
Number of Fin and Fin Spacing 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 3 Comparative Optimum Fin Spacing at different Temperature Gradient

Temperature difference (ΔT)	Optimum Fin Spacing 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

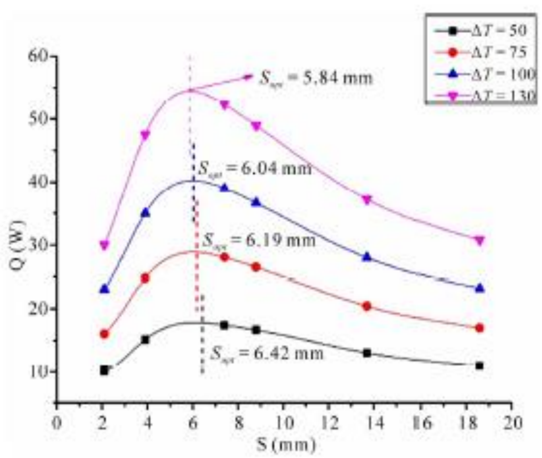


Figure 2 Variation of convective rate with base-to-ambient temperature difference at H = 29.2 mm and L = 80 mm Ref. [10]

In table 2 shows the Grid independence result of FVM result 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 2 and 3 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.

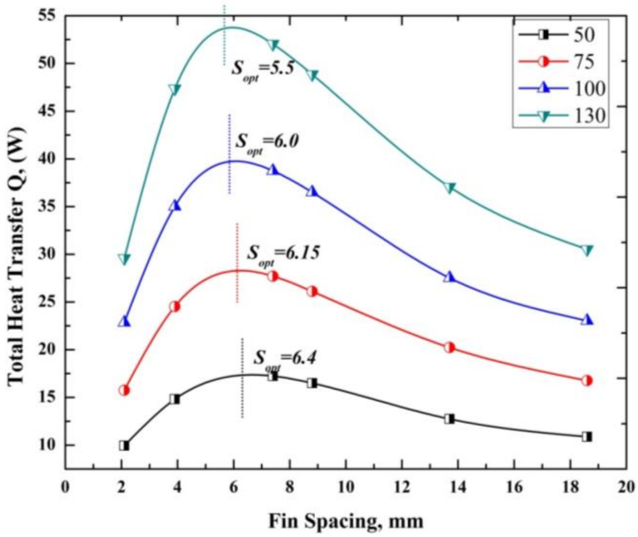


Figure 3 Variation of convective rate with base-to-ambient temperature difference at H = 29.2 mm and L = 80 mm

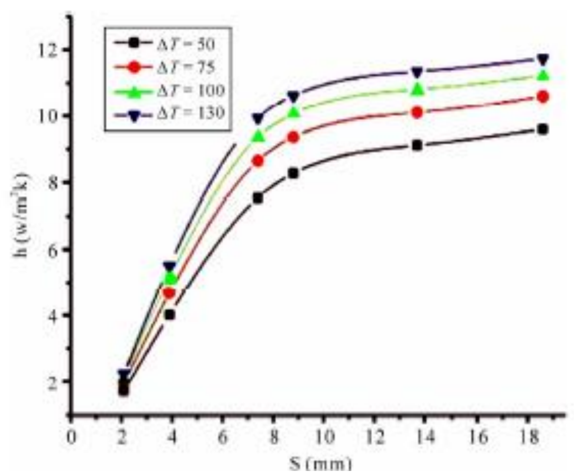


Figure 4 Natural convection heat transfer coefficients for different heat sinks Ref. [10]

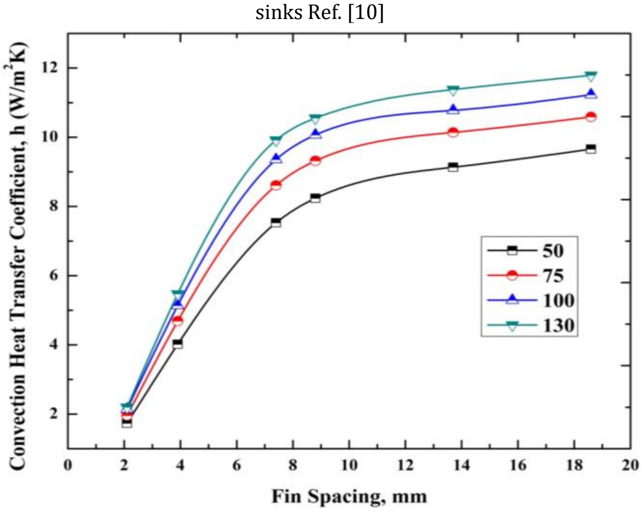


Figure 5 Variation of Convective heat transfer coefficient for different fin Spacing

Figure 5 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 (the fills up with warm air). However, the obtained results show same trends with the available literature Fahiminia [10] and form the figure 6.3 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.

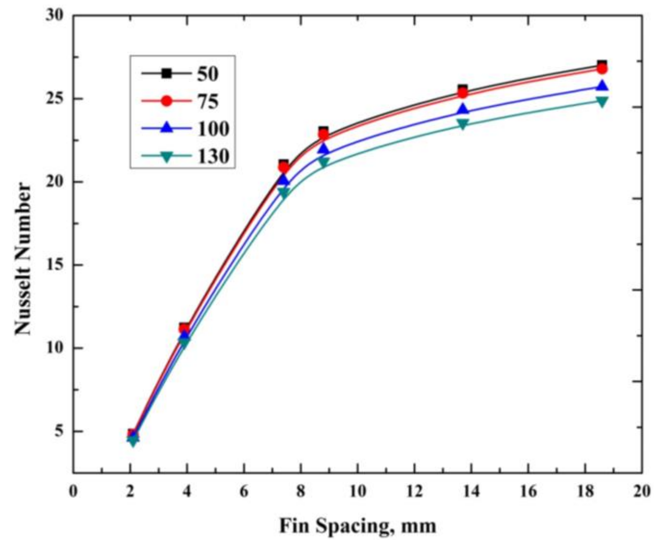


Figure 6Variation of Nusselt Number for different fin spacing

Figure 6 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 show in figure 7 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 8 to 11 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 12

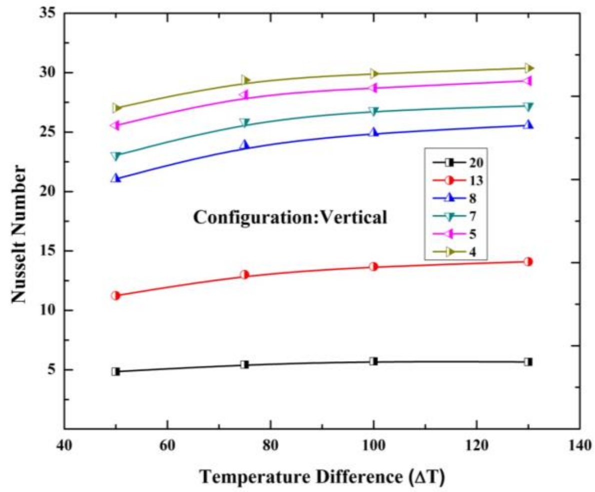


Figure 13 Variation of Nusselt Number in vertical Configuration for different Temperature Gradient

Figure 13 illustrate the Variation of Nusselt number in Vertical configuration for different Temperature Gradient. It has 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.

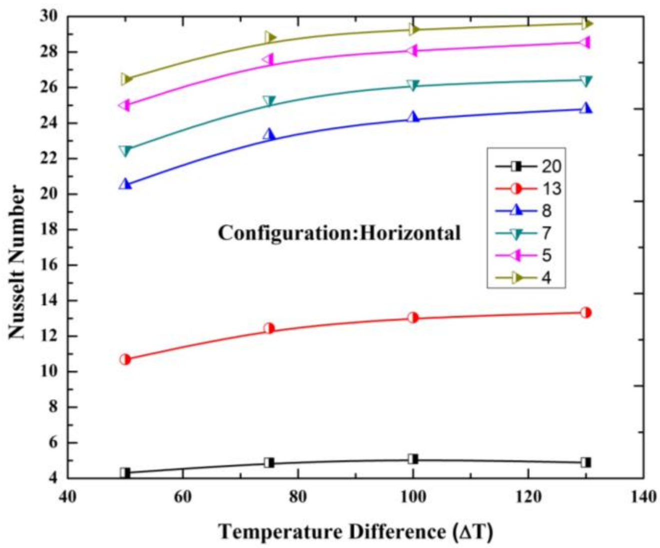


Figure 14 Variation of Nusselt Number in Horizontal Configuration for different Temperature Gradient

Figure14 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.

Conclusion

For Optimization and 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 increases noteworthy.

Widely spaced heat 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 has 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.

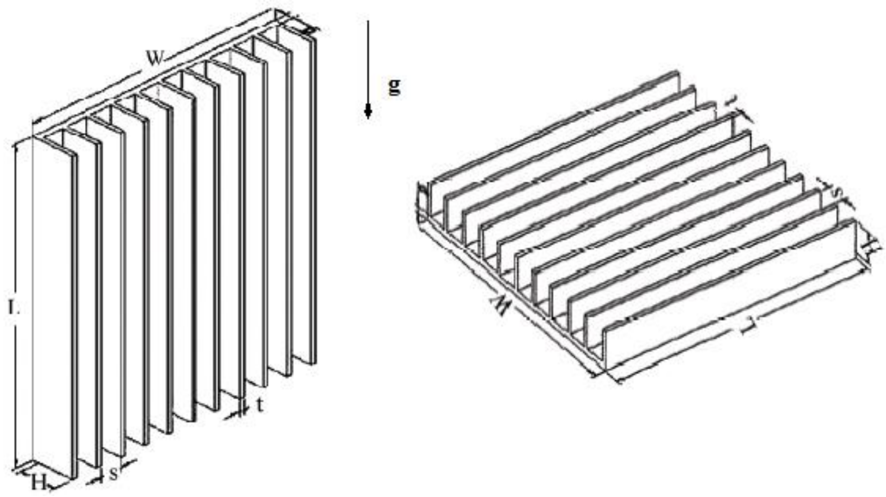


Figure 7 Fin configurations for natural (Free) cooling

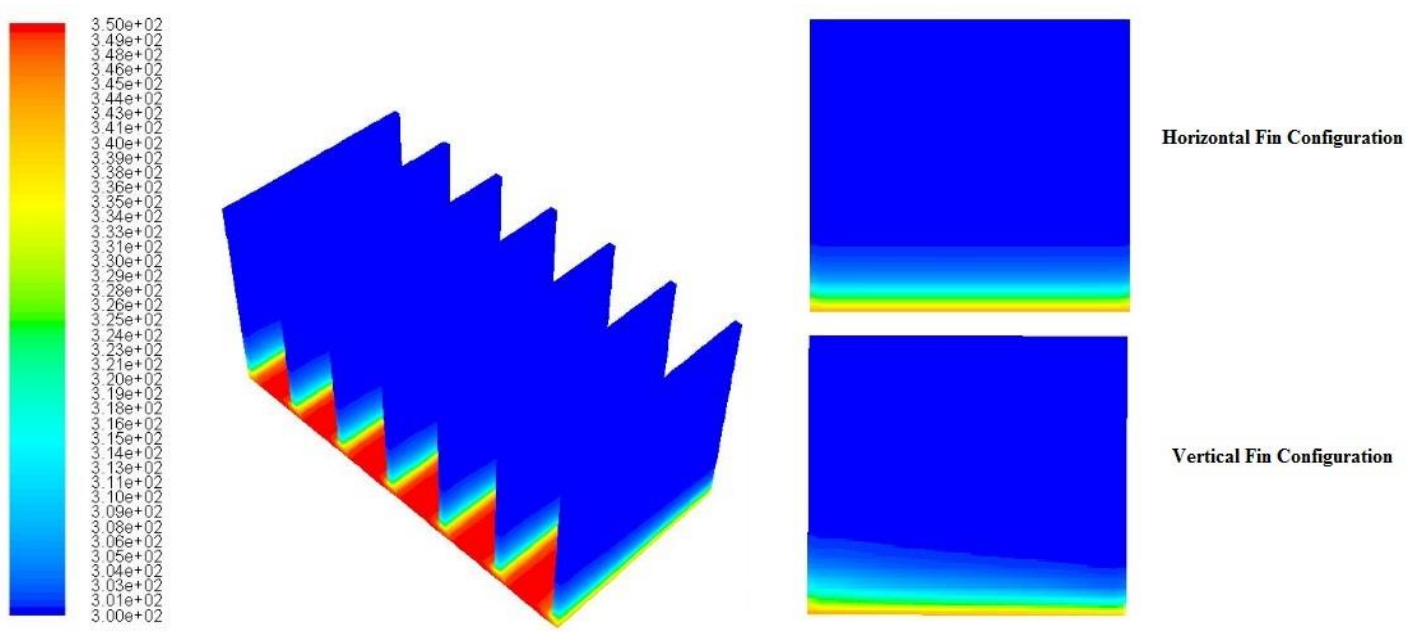


Figure 8 Variation of temperature contour of the heat sink s=8.8 and ΔT=50

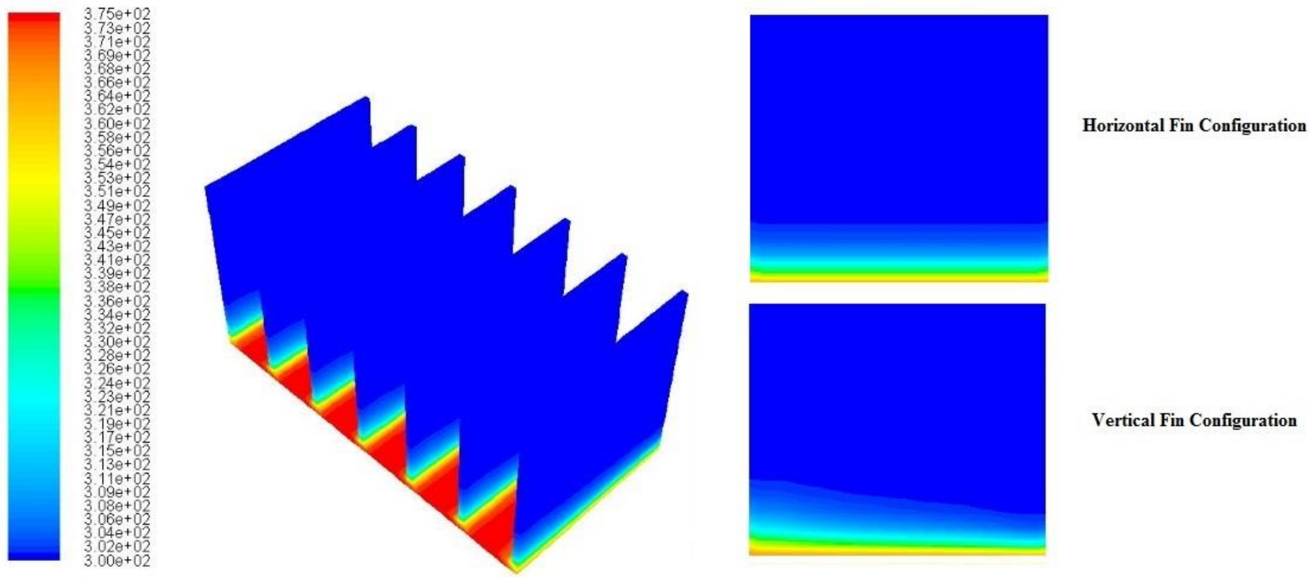


Figure 11 Variation of temperature contour of the heat sink s=8.8 and ΔT=75

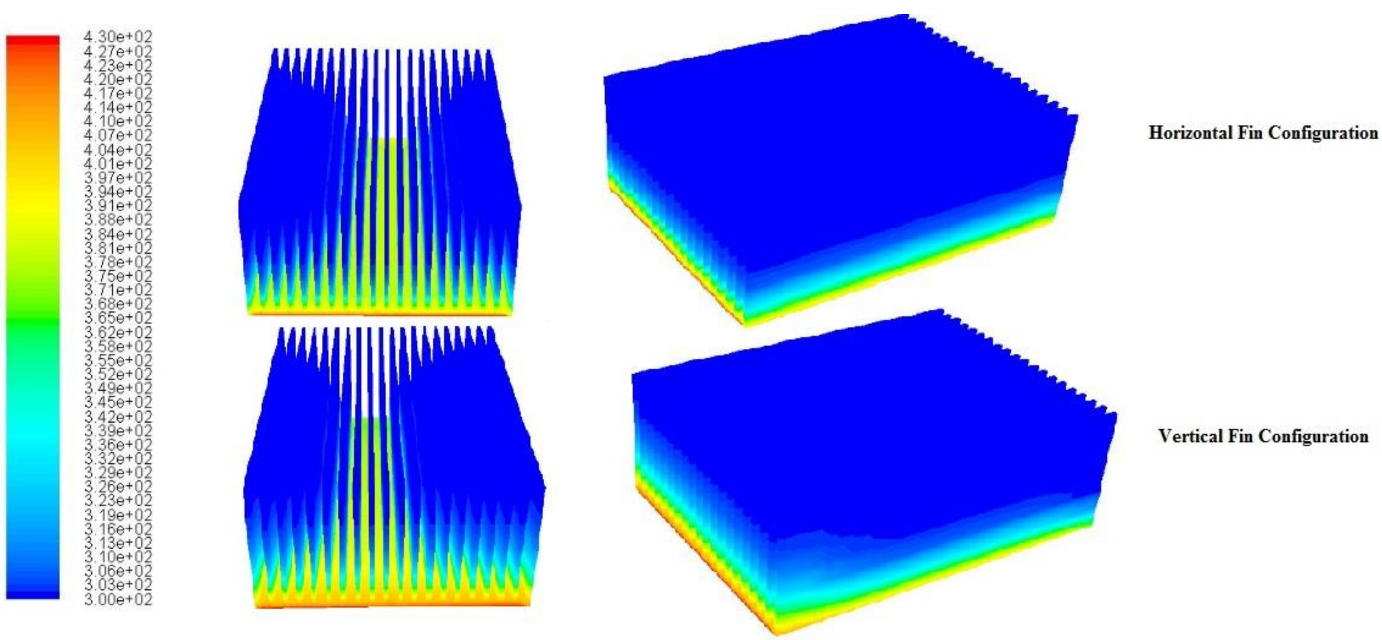


Figure 12 Variation of temperature contour of the heat sink 20 Fins and ΔT=130

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
τ	stress
τ_p	non-dimensional ramp time
ε	Emissivity

z	AxialCoordinate

Greek Symbols

α	Thermal diffusivity, m^2 / s
β	Thermal expansion coefficient at constant pressure, K^{-1}
ρ	Density, kg m^{-3}
c_p	Specific heat coefficient
c_v	Specific heat coefficient
Λ	Thermal Conductivity
φ	Angle