Parametric Analysis of Heat Sink using Finite Element Volume

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Abstract

Fins and extended surface in the form of the heat sink module are the most commonly used various industrial applications such as Nuclear reactor, Mixing tank, and solar air heater and even in electronic field also such in the CPU as heat exchanger. To get effective use of extended surface flat plate is used as fins with small spacing between them with counting proper flow of air on it. Effective heat transfer is there with perfect flatness of the contact area, a flat contact area allow to use a thinner layer of thermal compound, which will reduce the thermal resistance between heat sink and heat source. Mathematical model is developed for the flat plate extended fins which gives relationship between effectiveness and Nusselt Number. Nusselt number is observing for model with experiment and with mathematical expression. At the same time Variation in Nusselt Number is observe with temperature difference in fins and fins spacing which helps to conclude for effectiveness of the fins.

Keywords— *Fins, Heat Sink, CFD, Convection.*

I. INTRODUCTION

The heat sink is a very important component used for the proper and safe operation of electronic equipments by dissipating heat generated in them while working. Heat dissipation is one of the most important aspects to deal with to make them work within safe temperature limit specified by the manufacturer of the electronic equipment.

The heat sink is mounted over electronic devices such as motherboard and processors of PCs and laptops and increases their surface area for proper dissipation of heat. Heat sink conducted heat through the base of electronic device and then converting it to the surrounding fluid flowing around the extended or finned surfaces of the heat sink. For proper convection of heat from finned surfaces to the surrounding fluid moving around, the arrangement and geometry of the fins should be such that, the flowing fluid must make proper contact with the major portion of heat sink.

II. MATHEMATICAL MODELLING

The mathematical model based for a comprehensive, general-purpose model of fluid flow and heat transfer from the basic principles of conservation of mass, momentum and energy. This leads to the governing equations of fluid flow and a discussion of the necessary auxiliary conditions – initial and boundary conditions.

A. *Governing equations of fluid flow and heat transfer*

The governing equations of fluid flow represent mathematical statements of the conservation laws of physics:

- The mass of a fluid is conserved
- The rate of change of momentum equals the sum of the forces on a fluid particle (Newton's second law)
- The rate of change of energy is equal to the sum of the rate of heat addition to and the rate of work done on a fluid particle (first law of thermodynamics)\

$$p-\frac{\partial p}{\partial x}\frac{1}{2}\delta x$$

$$p+\frac{\partial p}{\partial x}\frac{1}{2}\delta x$$

B. *Mass conservation in three dimensions*

The first step in the derivation of the mass conservation equation is to write down a mass balance for the fluid element:

Rate of increase of flow of mass in fluid = Net rate of mass into element fluid element.

$$\left(\rho u-\frac{\partial(\rho u)}{\partial x}\frac{1}{2}\delta x\right)\delta y\delta z-\left(\rho u+\frac{\partial(\rho u)}{\partial x}\frac{1}{2}\delta x\right)\delta y\delta z$$
$$+\left(\rho v-\frac{\partial(\rho u)}{\partial y}\frac{1}{2}\delta y\right)\delta x\delta z-\left(\rho v+\frac{\partial(\rho u)}{\partial y}\frac{1}{2}\delta y\right)\delta x\delta z$$
$$+\left(\rho w-\frac{\partial(\rho w)}{\partial z}\frac{1}{2}\delta z\right)\delta x\delta y-\left(\rho w+\frac{\partial(\rho w)}{\partial z}\frac{1}{2}\delta z\right)\delta x\delta y$$

$$\frac{\partial\rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0$$

C. *Rates of change following a fluid particle and for a fluid element*

The momentum and energy conservation laws make statements regarding changes of properties of a fluid particle. This is termed the Lagrangian approach. Each property of such a particle is a function of the position (x, y, z) of the particle and time t. Let the value of a property per unit mass be denoted by φ. The total or substantive derivative of φ with respect to time following a fluid particle, written as Dφ/Dt, is

$$\frac{D\phi}{Dt}=\frac{\partial\phi}{\partial t}+\frac{\partial\phi}{\partial x}\frac{dx}{dt}+\frac{\partial\phi}{\partial y}\frac{dy}{dt}+\frac{\partial\phi}{\partial z}\frac{dz}{dt}$$

$$\frac{D\phi}{Dt}=\frac{\partial\phi}{\partial t}+u\frac{\partial\phi}{\partial x}+v\frac{\partial\phi}{\partial y}+w\frac{\partial\phi}{\partial z}=\frac{\partial\phi}{\partial t}=u.grad\phi$$

D. Momentum equation in three dimensions

Newton's second law states that the rate of change of momentum of a fluid particle equals the sum of the forces on the particle:

Rate of increase of momentum of fluid particle = Sum of forces on fluid particle

$$\rho\frac{Du}{Dt}=\frac{\partial(-p+\tau_{xx})}{\partial x}+\frac{\partial\tau_{yx}}{\partial y}+\frac{\partial\tau_{zx}}{\partial z}+S_{Mx}$$

$$\rho\frac{Dv}{Dt}=\frac{\partial\tau_{xy}}{\partial y}+\frac{\partial(-p+\tau_{yy})}{\partial y}+\frac{\partial\tau_{zy}}{\partial z}+S_{My}$$

$$\rho\frac{Dw}{Dt}=\frac{\partial\tau_{xz}}{\partial x}+\frac{\partial\tau_{yz}}{\partial y}+\frac{\partial(-p+\tau_{zz})}{\partial z}+S_{Mz}$$

E. Energy equation in three dimensions

The energy equation is derived from the first law of thermodynamics, which states that the rate of change of energy of a fluid particle is equal to the rate of heat addition to the fluid particle plus the rate of work done on the particle:

Rate of increase of energy of fluid particle = Net rate of heat added to fluid particle + Net rate of work done on fluid particle

$$-\frac{\partial(up)}{\partial x}-\frac{\partial(vp)}{\partial y}-\frac{\partial(wp)}{\partial z}=-div(pu)$$

III.CFD (FLUENT) ANALYSIS

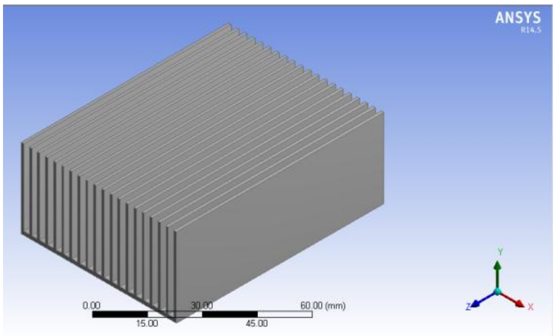

Fig1. Twenty fins

The equation of motion of the heat sink is solved using the FEV tool (ANSYS- Fluent) as the equation of motion for heat sink is difficult to visualize therefore some FEV tool is the only solution method for analysing thermo physical characteristics of heat sink.

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 modelled sink i.e. Base, Base Top, Fins, Interior.

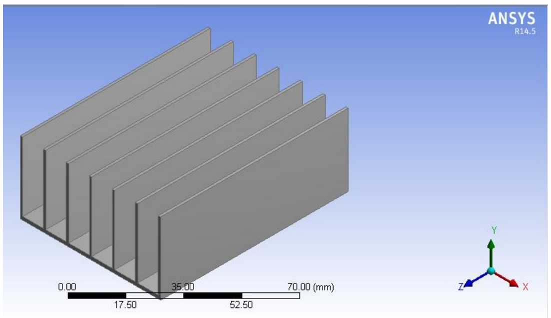

Fig2. Seven Fins

F. Generate Mesh

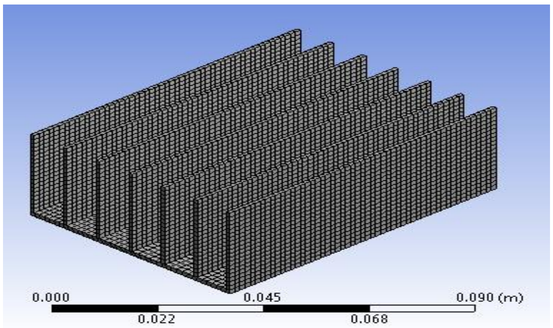

Fig3. Fine Size Mesh Generation

G. The boundary Condition of Heat Sink

Give boundary condition name in mesh module such as Inlet Base, Base Top, Fins, and Interior

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Table1. 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

IV. RESULTS 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.

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

The accuracy of the computational model was verified by comparing results from the present study with those obtained by Fahiminia [14], Goshayeshi [6], Analytical and FVM results.

Table 3. Comparative Optimum Fin Spacing at Different Temperature Gradient

Temperature difference (ΔT)	Optimum Fin Spacing in (mm)	
	Ref.[14]	Present (FEV)
50	6.42	6.40
75	6.19	6.15
100	6.04	6.00
130	5.84	5.50

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.

Table 3 and figure 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.

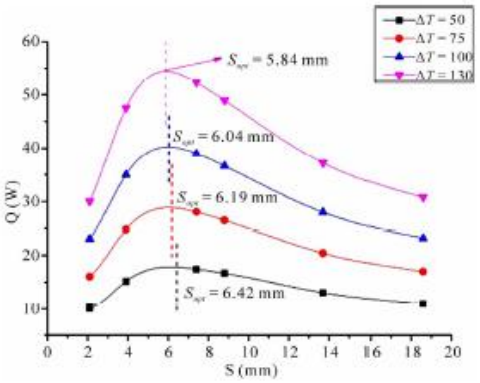

Fig4. Variation of Convective Rate with Base-To-Ambient Temperature Difference At H = 29.2 Mm And L = 80 Mm Ref. [14]

Table 4 and 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 [14]

Table 4. Variation of Convective Rate with Base-To-Ambient Temperature Difference

	Convective Rate			
Fin Spacing	ΔT (50)	ΔT (75)	ΔT(100)	ΔT (130)
2.1	9.93354	15.73346	22.84305	29.57845
3.9	14.79799	24.5269	35.00419	47.35242
7.4	17.23022	27.7075	38.74607	52.02978
8.8	16.48184	26.10237	36.50094	48.84917
13.7	12.73995	20.22373	27.52041	37.06223
18.6	10.86901	16.75603	23.03014	30.51392

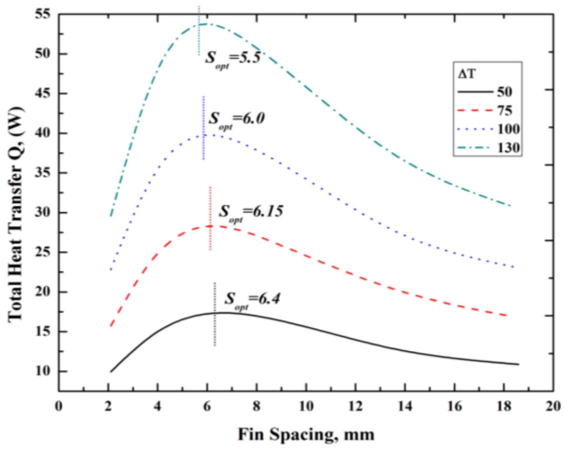

Fig5. Page Layout Variation of Convective Rate With Base-to-Ambient Temperature Difference at H = 29.2 mm and L = 80 mm

V. 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.

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.

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