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CFD ANALYSIS OF RECTANGULAR HEAT SINK OF THE VARIABLE FIN PITCH

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Abstract

In electronic equipments at the data centers where large server are involved, thermal management is one of the important topics of current research. These electronic equipments are virtually synonyms with modern life, for instance appliances, instruments and computer specifications. The dissipation of heat is necessary for its proper function. The heat is generated by the resistance encountered by electric current. Unless proper cooling arrangement is designed, the operating temperature exceeds permissible limit. As a consequence, chances of failure get increased. For more than a decade, investigations have been conducted to better understand the fluid flow and heat transfer characteristics in heat sinks designed for applications in electronic cooling. Heat sink is an environment or objects that absorbs heat and dissipates heat from another using thermal contact (either direct or radiant). And it is one of the commonly used devices for enhancing heat transfer in electronics components. This project is about the of study rectangular heat sink having variable fin pitch using ANSYS WORKBENCH version 14.1. The main objective of our analysis is to study the effect of variable fin pitch on the heat transfer, pressure drop, surface heat transfer coefficient. In this project the optimal fin dimension is taken from[1] and geometry is made, the optimal geometric parameter is as followed fin height, fin thickness, base height and fin pitch 48 mm, 1.6 mm, 8 mm and 4mm,here all the fin parameter is kept constant and only fin pitch is varied as 2mm, 4mm, 6mm, to study the effect. The air velocity is taken as 15 CFM and the heat input of 100w is provided.

INTRODUCTION

Extended surfaces or fins are commonly found on electronic components ranging from power supplies to transformers. The fins serve to increase the surface area thorough which heat is transferred to the surrounding environment by natural convection. Due to the rapid growth of electronic technology, electronic appliances and devices now are always in our daily life. Users prefer computers of higher performance and are therefore more willing to spend reasonably on them. This tendency has motivated the manufacturers to employ overdrive technology to improve their products. With this technology, these devices are capable of processing more data within a given period of time and the system performance is therefore regarded as higher. However, this capability is directly related to its heat generation .The larger the amount of data the system processes at a time, the greater the amount of heat it generates. The performance of these devices is directly related to the temperature; therefore it is a crucial issue to maintain the electronics at acceptable temperature levels. The dissipation and subsequent rejection of potentially destructive self produced heat is an important aspect of electronic equipment design. The most common method for cooling electronic devices is by finned heat sinks made of aluminum. These heat sinks provide a large surface area for the dissipation of heat and effectively reduce the thermal resistance. In order to design an effective heat sink, some criterions such as a large heat transfer rate, a low pressure drop, an easier manufacturing, a simpler structure, a reasonable cost and so on should be considered. As heat sinks are used in most electronics applications, the race for selecting a particular design of heat sink or more specifically a particular fin cross sectional profile remains somewhat undefined. Unfortunately, heat sinks often take up much space and contribute to the weight and cost of the product. Consequently, the need for new design and more effective ways to dissipate this energy is becoming increasing urgent.

THEORETICAL FORMULATION

The energy balance equation for an element having rectangular fins made of material of uniform thermal conductivity is

The rate of heat conduction into the element = rate of heat conduction out of element + rate of heat convection from the element surface

The rate of heat conduction in the element is the function of distance x which can be given as

$$Q(x) = -kA_c \frac{dT(x)}{dx}$$

$$Q(x) = Q(x) + \frac{d}{dx}[Q(x)]dx + h_c dA_s[T(x) - T(\infty)] \quad (1)$$

By using element surface area dAs=pdx, we get

$$\frac{d^2T}{dx^2} - \frac{h_c P}{k}[T(x) - T(\infty)] = 0$$

Let as assume that

$$\theta(x) = T(x) - T(\infty)$$

$$m^2 = \frac{h_c P}{kA_c} \quad (2)$$

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0 \quad (3)$$

The general solution of the equation is

$$\theta(x) = C_1 e^{-mx} + C_2 e^{mx} \quad (4)$$

Let us assume that the fin is of finite length and loss of heat from its tip is convective.

The boundary conditions are

$$-k[\frac{d\theta}{dx}]_{x=c} = h_c\theta_{x=L}$$

By using this boundary condition and rearranging the equation, we get

$$\frac{\theta(x)}{\theta_0} = \frac{\cosh\{m(L-x)\} + \left(\frac{h_c}{mk}\right)\sinh\{m(L-x)\}}{\cosh(mL) + \left(\frac{h_c}{mk}\right)\sinh(mL)}$$

And,

$$Q_{fin} = \sqrt{h_c PkA_c}(T_0 - T_\infty)\frac{\sinh mL + \left(\frac{h}{mk}\right)\cosh mL}{\cosh mL + \left(\frac{h}{mk}\right)\sinh mL}$$

MODELING AND SIMULATION

CFD codes are structured around the numerical algorithms that can be tackle fluid problems. In order to provide easy access to their solving power all commercial CFD packages include sophisticated user interfaces input problem parameters and to examine the results. Hence all codes contain three main elements: In CFD calculations, there are three main steps

1. Pre-processing.
2. Solver
3. Post processing.

PRE-PROCESSING: Pre-Processing is the step where the modeling goals are determined and computational grid is created. Preprocessor consists of input of a flow problem by means of an operator friendly interface and subsequent transformation of this input into form of suitable for the use by the solver.

The user activities at the Pre-processing stage involve: Definition of the geometry of the region:

The computational domain. Grid generation is the subdivision of the domain into a number of smaller, non-overlapping sub domains (or control volumes or elements Selection of physical or chemical phenomena that need to be modeled).

Definition of fluid properties: Specification of appropriate boundary conditions at cells, which coincide with or touch the boundary. The solution of a flow problem (velocity, pressure, temperature etc.) is defined at nodes inside each cell. The accuracy of CFD solutions is governed by number of cells in the grid. In general, the larger numbers of cells better the solution accuracy. Both the accuracy of the solution & its cost in terms of necessary computer hardware & calculation time are dependent on the fineness of the grid. Efforts are underway to develop CFD codes with a (self) adaptive meshing capability. Ultimately such programs will automatically refine the grid in areas of rapid variation.

SOLVER EXECUTION: In the second step numerical models and boundary conditions are set to start up the solver. Solver runs until the convergence is reached. There are three distinct streams of numerical solutions techniques: finite difference, finite volume& finite element methods. In outline the numerical methods that form the basis of solver performs the following steps. The approximation of unknown flow variables are by means of simple functions. Discretization by substitution of the approximation into the governing flow equations & subsequent mathematical manipulations.

POST-PROCESSING: As in the pre-processing huge amount of development work has recently has taken place in the post processing field.

Governing Equations of Fluid Flow

The most general form of fluid flow and heat transfer equations of compressible Newtonian fluid with time dependency used in solver execution is given as follows

Continuity Equation

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

$$\frac{\partial \rho}{\partial t}+\nabla.(\rho\bar{V})=0$$

Momentum Equation

X-momentum equation

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

Y-momentum equation

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

Z-momentum equation

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

Energy Equation

$$\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{1}{\alpha}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)$$

CFD SIMULATIONS

The geometric parameters of the heat sink chosen for the present study are given in Table 1. Considering the various geometric variables present, models are created in Uni-graphics NX-6 and simulation are carried out using Fluent 14.1 to determine the output parameters like maximum temperature, pressure drop, surface heat transfer coefficient .The number of fin is kept constant as increase in the number of fins has an adverse effect of increase in pressure drop.

Table 1 Geometric parameters

Model	Fin height	Fin thickness	Base height	Fin pitch
1	48mm	1.6mm	0.8mm	6mm
2	48mm	1.6mm	0.8mm	4mm
3	48mm	1.6mm	0.8mm	2mm

RESULTS AND DISCUSSIONS

Table.2 Result

Fin pitch (mm)	Overall Heat transfer coefficient (w/m^{2}k)	pressure drop (Pascal)	Max. Temperature (k)
2	34.16299	92.48031	356.636
4	32.23675	18.88845	364.8125
6	26.85737	7.199409	369.0408

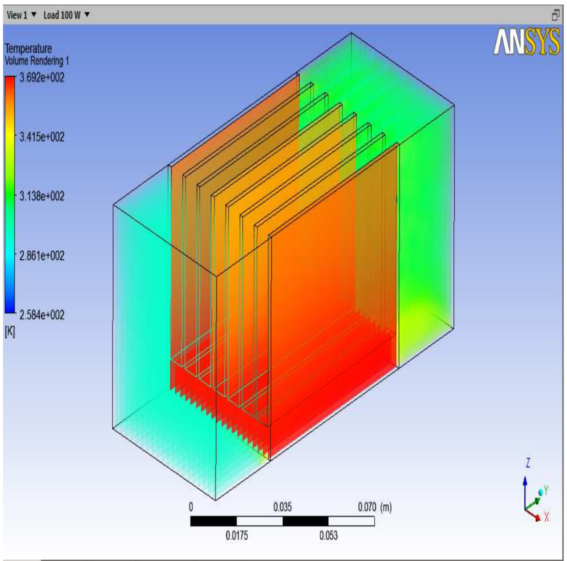


Figure 1 contour of static temperature of heat sink of fin pitch 6mm for 100w heat input

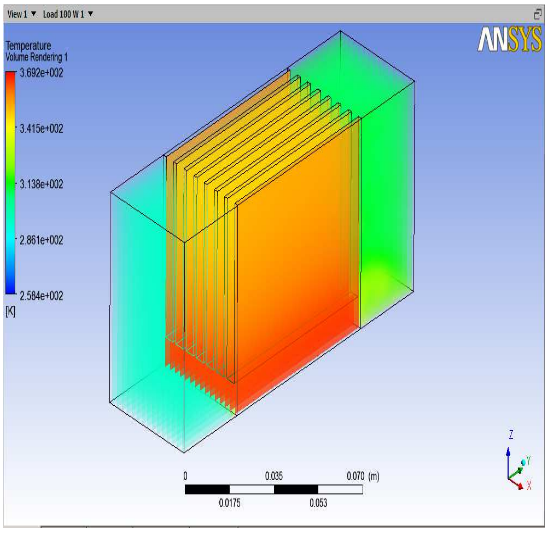


Figure 2 contour of static temperature of heat sink of fin pitch 4mm for 100w heat input

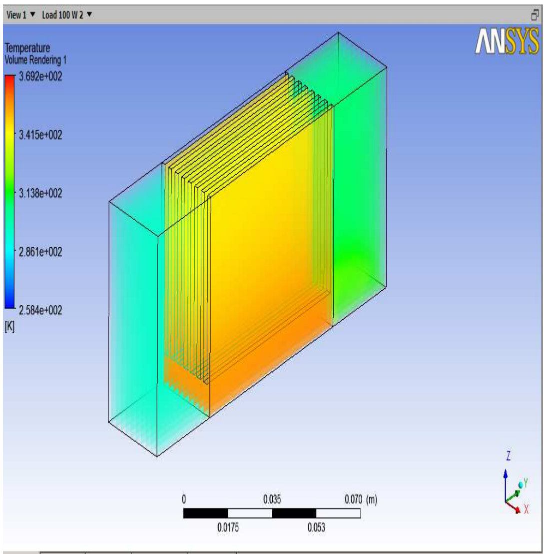
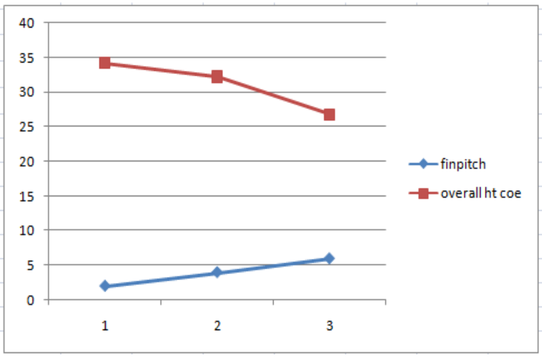
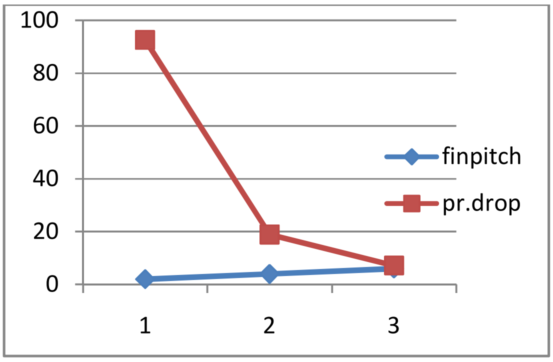


Figure 3 contour of static temperature of heat sink of fin pitch 2mm for 100w heat input

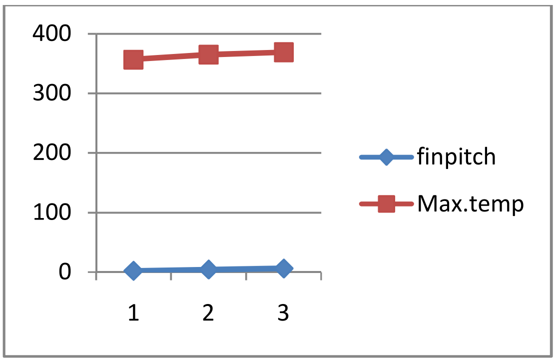
Graph 1 Fin pitch v/s overall heat transfer coef.



Graph 2 Fin pitch v/s pressure drop



Graph 3 Fin pitch v/s Max Temperature



From the above graphs

Graph 1 indicates as the fin pitch increases overall heat transfer coefficient decreases.

As per further studies we know that to acquire minimum temperature the heat transfer coefficient should be more herefrom the above result the heat sink with fin pitch 2 mm has more heat transfer coefficient

Graph 2 indicates as the fin pitch increases

Values of the maximum temperature increases

Similarly here also the heat sink with fin pitch 2 mm shows minimum temperature attained.

Graph 3 indicates as the fin pitch increases

Pressure drop decreases.but for the satisfactory result the pressure drop should be minimum. Here in case of 2 mm fin pitch the pressure drop is 92.4803 pa and with 4 mm pitch 18.88 pa and with 6 mm 7.19 pa so the heat sink with pitch 6 mm satisfies. but it shows the max temp which is un avoidable.

So understanding all the conditions the heat sink with pitch 4mm should be selectes as it has all the optimum values.as we can't select fin with pitch 2mm as it has high pressure drop, though it has minimum temperature attained.And that to we cannot select heat sink with in pitch 6mm as it has maximum temperature though it has least pressure drop and the heat sink with fin pitch 4mm shows the optimum values of all the three parameter and the result found to be in good agreement with conclusion drawn by Arularasan R.and Velraj R[1].

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