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Experimental Analysis of Parallel Plate Heat Sinks with Base Plate for Variable Fan Distance

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ABSTRACT

A numerical analysis of forced convective heat transfer from a parallel plate fin heat sinks are investigated by varying fan locations through Computational Fluid Dynamics (CFD) technique for dealing with the Navier-Stokes equations and energy equations. In this paper the thermal performance of heat sink has been examined for the distance between fan and heat sink which is varied from 5 to 50mm, fin thickness, fin pitch and copper & carbon carbon composite (CCC) base plates at contant Reynolds number. The results predicted in this work are qualified by comparing with Experimental results. The results indicate that the thermal resistance can significantly increased by increasing the fan distance upto 30mm and above 30mm the thermal performance is increased due to the increase of the average top surface temperature of heat sink.The present findings not only set up a numerical heat transfer analysis of desktop computer but also provide a basis for further simulation of the associated heat transfer for more complicated situations.

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INTRODUCTION

Electronic devices and systems are reduced gradually in size while a rapid growth in their functions and complexity has been encountered over the past four decades. In the electronic industry, the trend toward denser and more powerful products has led to the use of electronic devices with higher power densities (Oktay and Hannemann, 1986; Bar-Cohen, 1996). However, high power densities result in increased junction temperatures, which significantly affect the reliability of electronic devices. Therefore, an effective cooling technique is essential for reliably operating electronic components (Incropera, 1988; Nakayama, 1986). Many ideas pertaining to cooling methods have been proposed such as jet impingement cooling (Chung and Luo, 2002; Nishino, 1996) and heat pipe (Kim, 2003; Wang and Vafai, 2000; Zhao and Avedisian, 1997). Among the various types of cooling systems that have been developed, the parallel plate-fin heat sink is the most widely used because of its simple design and ease of fabrication. A good literature review can be summarized by Liu and Garimella (Liu and Garimella, 2005), several studies have focused on optimizing the size of plate-fin heat sinks. Many optimization methods have been proposed based on the fin model (Knight, 1991). Kim *et al* (2003) have conducted the thermal optimization of a plate-fin heat sink with the fin thickness varying in the direction normal to the fluid flow. Kim and Kim (2003) have experimentally studied the effects of cross-cuts on the thermal performance of heat sinks under the parallel flow condition. Hwang and Lui (1999) studied the heat transfer and pressure drop characteristics between pin-fin trapezoidal ducts with straight and lateral outlet flows. The effect of pin arrangement for the ducts of different direction outlet flow was also examined. Moreover, a similarity of the pin Reynolds number dependence of row-averaged Nusselt number was developed. An experimental study was conducted to investigate the heat transfer from a parallel flat plate heat sink under a turbulent air jet impingement by Sansoucy *et al* (2006). The forced convection heat transfer rates from a flat plate and from a flat plated heat sink under an impinging confined jet have been obtained. In addition, the experimental results were compared with the numerical predictions obtained in an earlier study. They concluded that the numerical analysis in a previous study was adequate for appraising the mean heat transfer rate in jet impingement for situations of thermal management of electronics. Linton and Agonafer (1994) simulated an entire desktop PC with one fan using Phoenics code. Lee and Mahalingam (2000) used Flotherm code to

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simulate detailed flow and temperature fields within a computer chassis with two fans. Similar work was also done by Wong and Lee (1996). Yu.C.W and Webb.R.L (2001) analyzed the flow and heat transfer inside a computer cabinet for the high power conditions expected in desktop computers. In this research CFD (Icepak) has been used to identify a cooling solution for a desktop computer. The 40W PCI card, different case fan size and different ducting positions have been studied. Chang, J.Y. *et al* (2000) report the results of CFD analysis to cool the 30W socketed CPU of a desktop computer with minimum air flow rate and minimum heat sink size. In the paper the methodology of CFD analysis for the heat sink, and duct design has been described as well as the experimental procedures to validate the predictions. David Lober (1999) discussed some thermal management considerations involved in choosing an enclosure and demonstrated the use of CFD thermal modeling software which optimally integrated a computer system into an existing enclosure and reduced the design cycle time. Ozturk and Tari (2008) have investigated the flow and temperature fields inside the chassis and also the three different commercial heat sink designs have been analysed by using CFD. The flow obstructions in the chassis and the resulting air circulation that affect the heat sink temperature distribution are studied. It is recommended that a maximum temperature distribution in the heat sink can be reduced by changing the geometry, base thickness and especially, replacing aluminum with copper as the heat sink material. The four copper fins are used at the center of the heat sink for reducing hot spots. Mohan and Govindarajan (Mohan and Govindarajan, 2011; Mohan and Govindarajan, 2011; Mohan and Govindarajan, 2010; Mohan and Govindarajan, 2010; Mohan, 2014) have studied the performance of slot parallel plate fin heat sink, double base plate heat sink, elliptical fin heat sink, pin fin heat sink with various base plate materials and ununiform fin width heat sinks with nano coating. The experimental results are compared with CFD results. In this paper the heat sink with base plate and optimal distance between heat sink and fan for air flow are designed and implemented for better performance. This study investigates a parallel plate heat sink model with various thickness base plates used to cool central processing units (CPUs) of desktop computers

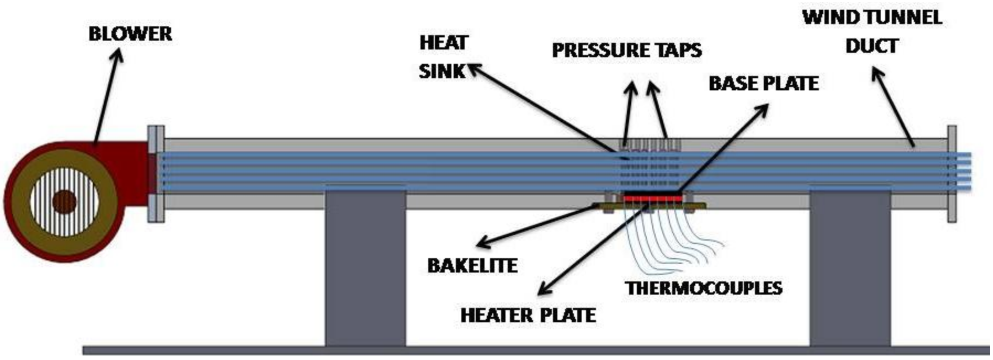

Fig. 1: A Schematic diagram of the Experimental setup.

Experimental setup and procedure:

In order to simplify the experiments, the experimental apparatus is made of a wind tunnel of a rectangular section. An experimental setup is arranged with an electrical heater of size 25 x 25 mm as a heat sources to imitate a processor and it is supplied by DC power supply. A heater strip was mounted on a piece of circuit board which was, in turn, mounted on a piece of insulation. The insulation was machined so that the top surface of the heater lies just below the top surface of the insulation. The insulation was drilled or milled to allow routing of power and thermocouple wires to minimize free stream interference in the vicinity of the heat sink. The heater is fixed with heat sink using anabond compound with thermal conductivity of 0.437 W m/ K and wakefield thermal compound with thermal conductivity of 0.735 Wm/ K is applied to the contact surface of the heat sink to make proper surface contact between heat sink and the heater. The heat sink rejects the heat into air which is enhanced by placing the blower. The area covered by the resistance coils (on the heat strip) is less than the area of the base of the heat sink, so that the bottom surface of the heat sink completely covers the active heater section. This ensures that the entire power output of the heater is dissipated through the heat sink. Fourteen J- type thermocouples were used for temperature measurement. To measure the maximum temperature of the heat sink, seven thermocouples were mounted through 5mm deep holes at the base plate of the heat sink. They are 7 mm apart. The first thermocouple is positioned 8mm from the leading edge of the heat sink and the last one is positioned 8 mm from the end of the heat sink. Figure 1 shows the schematic representation of the arrangement for the experimentation. The 25 x 25 mm as a heat sources to imitate a processor is attached to different heat sinks models and heated with heat loads 80W. Since the test setup is an open domain, the atmospheric temperature is the temperature of the air blown to the heat sink. The atmospheric air is passed around the heat sink which is heated that is exhausted by blower and also the pressure drop has been noted. In this setup other heat sources are not considered for simplify the experiments. In this experimental setup the desired volume flow rate of the air was generated by a suction type blower in steady of CPU fan and exhaust fan. The steady state was assumed the change in the maximum temperature of the heat sink was smaller than ± 0.1 °C for a period of 3min. the temperatures are recorded and used to calculate the thermal resistance of the heat sink.

The wind tunnel is an open circuit type consisting of a variable speed 1/2 HP DC motor powering a squirrel cage blower. Air velocities were measured from ports located on the sides of the test section tube. Velocity measurements were made with a Dwyer Model 471-2 Digital Thermo-Anemometer. Temperatures were measured with thermocouples and temperature data was collected at the locations upstream of the heat sink, and at the base of the heat sink. The maximum base temperature of the heat sink and the bulk mean inlet temperature were used to calculate the thermal resistance of the heat sink.

$$R=\frac{T_{w,max}-T_{bm,in}}{Q}$$

Numerical approach:

The CPU heat sink with base plate is attached to the CPU in conjunction with with a fan. The mainboard and all the other components are enclosed in a chassis. There are many other heat sources in addition to the CPU. Some of them are on the mainboard (e.g., northbridge chip), some of them are attached to the mainboard (e.g., memory modules) and some of them are in the chassis volume (e.g., DVD).

Problem description and Boundary conditions:

The CFD 3D chassis model is shown in Figure 2. The chassis is modeled using standard dimensions of a common ATX chassis by hollow blocks and internal components are represented as lumped objects. During modeling, all the components inside the chassis are standard sized components and exact dimensions are obtained by measurement. The CPU chip is modeled as a 2D area which dissipates 80W. The 30 mm x 30 mm cross sectional area of the CPU chip is taken which is a commercially available AMD CPU. For simplicity, the mother board, and chipset card are modeled as zero thickness with heat generated uniformly. The CPU fan is modeled as a lumped parameter model and does not have blades which produce flow rate of 30CFM. Ram cards are fixed on the motherboard. They are also heat sources and accurate dimensioning of the space between ram cards is difficult. Therefore they are not considered for this study. The power supply (75W) has a very complex geometry which includes lot of electric components, wiring and heat sinks. It is assumed as a lumped media which exerts a resistance on the cooling air flow streams. The SMPS (Switch Mode Power Supply) and a few miscellaneous cards (20 W) are modeled and lots of small electronic components on these cards are not modeled.

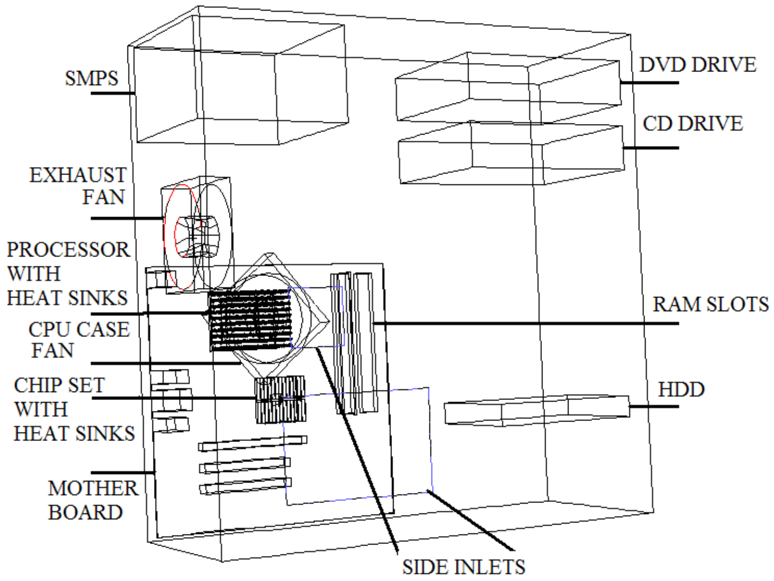

Fig. 2: Computer Chassis mode.

The computer cases have small holes which are used to allow inlet air for cooling and to discharge hot air through the outlet. The modeling of these holes in accurate dimensioning is difficult and computationally expensive. Therefore it is modeled as a zero thickness flow resistance. The flow rate of case fan is considered as 40 CFM. The HDD (Hard Disk Drive, 20W), DVD(15W), and CD(15W) are modeled as solid blocks generating a specified amount of heat uniformly inside the volume. The scope of this study is the investigation of temperature distributions on CPU heat sinks. A total of 225W of heat is dissipated.The fans inside the domain are modeled as circular surfaces which add momentum source to the flow. The added momentum source is given as the pressure rise across the fan versus the flow rate curve. The relationship between the pressure and the flow rate is taken linearly. The boundary condition for the power supply is different. The power supply is geometrically very complicated. Therefore it is modeled by simplifications. The power supply is a rectangular box which is a resistance to flow. The resistance is different in the y-direction. A uniform heat flux is applied on the chip due to the chip is high thermal conductivity silicon material. The chassis walls are assumed as adiabatic. The initial temperature of air was assumed as 27^0C. No-slip boundary conditions are applied to all channel walls. The computations are considered to be converged when the residues for all governing equations were less than 1×10-6. In order to reduce the thermal stress between the heat sink and chip, silicon whose thermal conductivity is 150 W/(m•K), is chosen as the material of heat sink. It's helpful for improving the reliability of chips.

Validation and Grid Independence:

To check the validity of the present study, the experimental results of the thermal resistance of a heat sink with copper base plate under the conditions of constant Re and Q = 70 W , one of the experimental cases, were compared with the numerical results. Figure 3 shows that the experimental results of the thermal resistance of a heat sink are in good agreement with numerical results proposed within 6.5%. Based on these comparisons, it is clear that the experiments in this study were properly conducted. For constant Reynolds number and copper base heat sink with differet fan position, the calculations were performed with four different meshes; the sizes of grids were 287702, 575404, 1150807, and 2301613. The computational grid consisted only of tetrahedron grid.The results of thermal resistance were compared for all four meshes. The deviations of the thermal resistance predicted using meshes with 287702, and 575404 cells are 5.5% and 2.5% from the obtained using a mesh with 1150807 cells. The thermal resistance change by 0.233% from mesh 3 to mesh 4, respectively.It was found that a grid with 1150807 cells is sufficient to accurately predict the basic characteristics of flow and heat transfer.

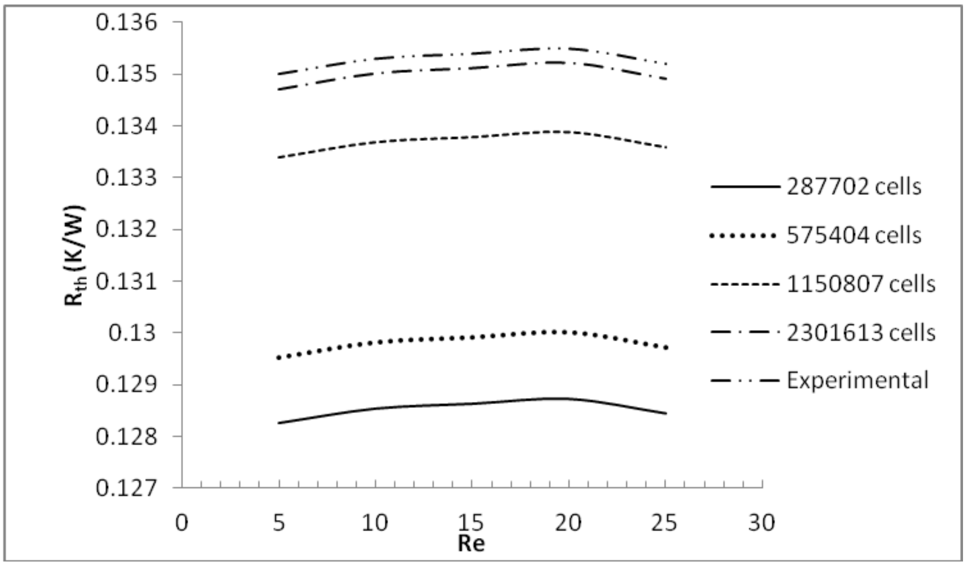

Fig. 3: Verification of Experimental value.

Governing equations for fluid flow and heat transfer:

The thermal and flow fields were calculated numerically with commercial CFD software ANSYS FLUENT 14, according to the following assumptions: the flow was steady state; the flow was incompressible and turbulent; the fluid and the solid properties were constant, and the effects of gravitation and thermal radiation were neglected. The equations governing the fluid are the Reynolds-averaged Navier–Stokes equations and the energy equation. Based on the aforementioned assumptions these equations can be expressed as:

Continuity Equation

$$\nabla\left(\rho\,\vec{V}\right) = 0 \tag{3.1}$$

Momentum Equations

$$\nabla(\rho u\vec{V}) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + B_x \qquad (3.2)$$

$$\nabla(\rho v\vec{V}) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + B_y \qquad (3.3)$$

$$\nabla(\rho w\vec{V}) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + B_z \qquad (3.4)$$

Energy Equation

$$\nabla(\rho h\vec{V}) = -p\nabla\vec{V} + \nabla(k\,\nabla T) + \dot{g} \qquad (3.5)$$

Turbulent kinetic energy k Equation

$$\overline{U}\frac{\partial k}{\partial x} + \overline{V}\frac{\partial k}{\partial y} + \overline{W}\frac{\partial k}{\partial z} = \frac{\partial}{\partial x}\left(\nu + \frac{\nu_t}{\sigma_k}\frac{\partial k}{\partial x}\right) + \frac{\partial}{\partial y}\left(\nu + \frac{\nu_t}{\sigma_k}\frac{\partial k}{\partial y}\right) + \frac{\partial}{\partial z}\left(\nu + \frac{\nu_t}{\sigma_k}\frac{\partial k}{\partial z}\right) + P_k - \varepsilon \qquad (3.6)$$

Turbulent energy dissipation rate ε Equation

$$\overline{U}\frac{\partial \varepsilon}{\partial x} + \overline{V}\frac{\partial \varepsilon}{\partial y} + \overline{W}\frac{\partial \varepsilon}{\partial z} = \frac{\partial}{\partial x}\left(\nu + \frac{\nu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x}\right) + \frac{\partial}{\partial y}\left(\nu + \frac{\nu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial y}\right) + \frac{\partial}{\partial z}\left(\nu + \frac{\nu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial z}\right) + C_{\varepsilon 1}\frac{\varepsilon}{k}P_k - C_{\varepsilon 2}\frac{\varepsilon^2}{k} \qquad (3.7)$$

P_k is the production of the turbulent kinetic energy that is usually modeled as

$$P_k = \nu_t S^2$$

Where S is the modulus of the mean strain rate tensor. The turbulent viscosity is given by

$$\nu_t = C_\mu \frac{k^2}{\varepsilon}.$$

This model contains five constants and the most commonly used values for those are

$$C_\mu = 0.09,\ C_{\varepsilon 1} = 1.14,\ C_{\varepsilon 2} = 1.92,\ \sigma_k = 1.0,\ \sigma_\varepsilon = 1.3.$$

Heat sink selection:

The parallel plate fin heat sinks in two arrays of variable fin thickness and base plates are used to cool the CPU. Unfortunately, significant modeling and run time is needed to represent small pins with complex meshing. In the course of preliminary numerical simulation work, three variable thickness fin geometries of same base area, same fin height, and same fin pitch are simulated. The 54mm x 65mm heat sink base plate size is selected for this work. The different shape of extruded fins and a 3.5mm thick base plate is finished of aluminum materials. In addition to enhance the heat transfer 5 mm thick copper and CCC base plate has been provided as a spreader to conduct heat from CPU processor. For all fin geometries 2.5 mm fin pitch, 40mm fin height, 54mm x 65mm heat sink base plate, 5mm to 50 mm clearance between the power supply and the fin tips of the heat sink at the flow rate of 30CFM. The fin thickness is varied from 3 to 5mm for plate heat sink model. The thermal performance of the heat sink is modeled using gambit. The heat sink solid model in the chassis numerical model is done and the CFD software solves the heat transfer problem for the heat sink. The flow of air is parallel to the heat sinks and vertical flow of air is pulled upward by a fan mounted at the top of the heat sink with clearance. The description of various heat sink models are listed in table 1.

Table 1: Description of heat sinks with base plate materials.

Parameters in mm	Sink A	Sink B	Sink C
Fin material	Aluminum	Aluminum	Aluminum
Fin profile	Plate Fin	Plate Fin	Plate Fin
Fin Dimensions	25x40x3	25x40x4	25x40x5
Fan Distance	5,20.30.40 &50	5,20.30.40 &50	5,20.30.40 &50
Base Plate Thickness	5	5	5
Base Plate material	Copper/ CCC	Copper/ CCC	Copper/ CCC

RESULTS AND DISCUSSIONS

The chassis model with a parallel plate heat sink with base plates are analyzed by CFD simulations. The results have been obtained by varying the heat sink model and keeping the entire computational domain same. For all of the heat sinks, it is viewed that their centers are the hot spots since the heat source corresponds to the closeness of the base center. The fans installed on the heat sinks are identical in dimension. The fans have hubs where air cannot pass through and it makes the center parts hotter. In the current simulations, the swirl of the fan is not modeled since the fans are lumped parameter models. For real cases, the center would not be as hot as the present simulations predict, due to the swirl. It is observed that the upper right and left part of the heat sink has a lower temperature when compared to centre part of the heat sink. This is due to more air flow circulation in the sides of the heat sink and also the exhaust fan sucks the hot air which is nearer to side of the heat sink. The cooling becomes less efficient at other sides of the heat sink. In figure 4, 5 and 6 the thermal resistance of CCC base plate heat sink models is reduced when compare to copper base plate heat sink models. Since for the same heat source CPU the bottom heat sink temperature is decreased for an increasing number of fins. Figure 4, 5 and 6 presents the thermal performance of heat sink of different cases at constant Reynolds number. It is observed that the thermal performance with 5mm fan distance is performed well when compared to other fan distance. This is due to more air flow rate in the vicinity of heat sink as well as higher air velocity As the fan distance increases gradually, the air flow rate de creases and the air velocity decreases with respect to the heat sink position which causes the increase in thermal resistance. The thermal resistance is decreased for lesser fan distance. It is noticed that temperature of heat sink is slightly increased for increasing the fan distance. The performance of heat sink is not differ for increasing fan distance, but top and bottom surface temperature slightly increased upto 0.1 to 0.4°C is shown in figure 7 and 8. The thermal performance of heat sink with CCC base plate increases when compared to copper base plate. The thermal performance of heat sink does not change much as fan distance increases.

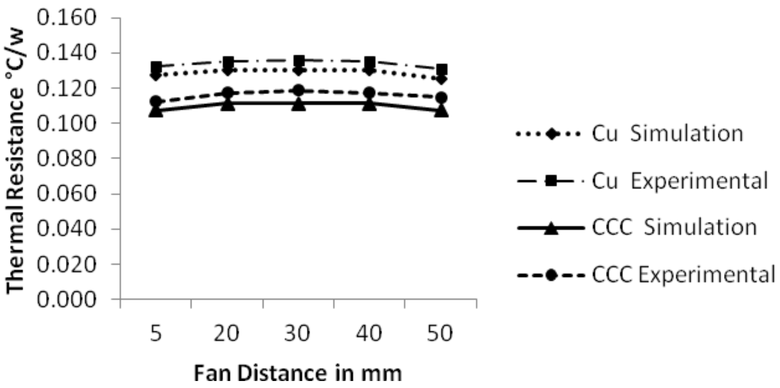

Fig. 4: Thermal Resistance vs Fan distance for Fin thickness 3mm.

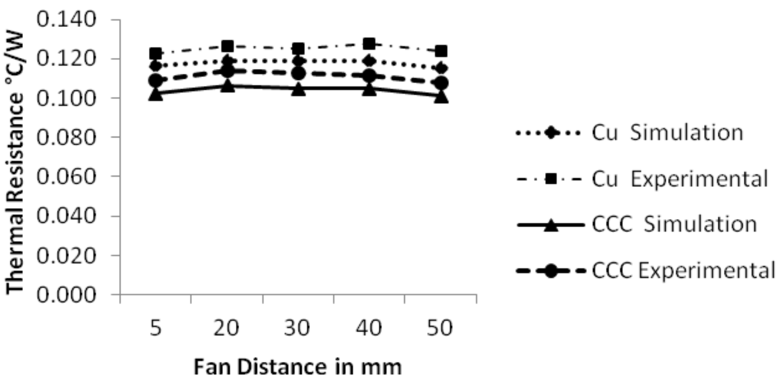

Fig. 5: Thermal Resistance vs Fan distance for Fin thickness 4mm.

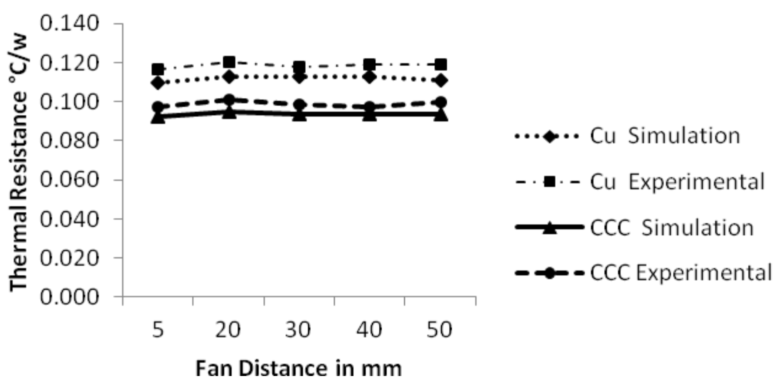

Fig. 6: Thermal Resistance vs Fan distance for Fin thickness 5mm.

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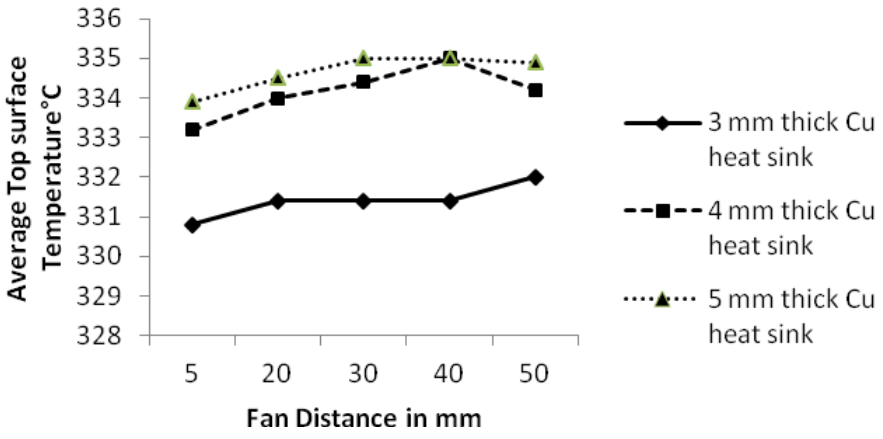

Fig. 7: Top Surface Temperature vs Fan Distance for Cu base plate.

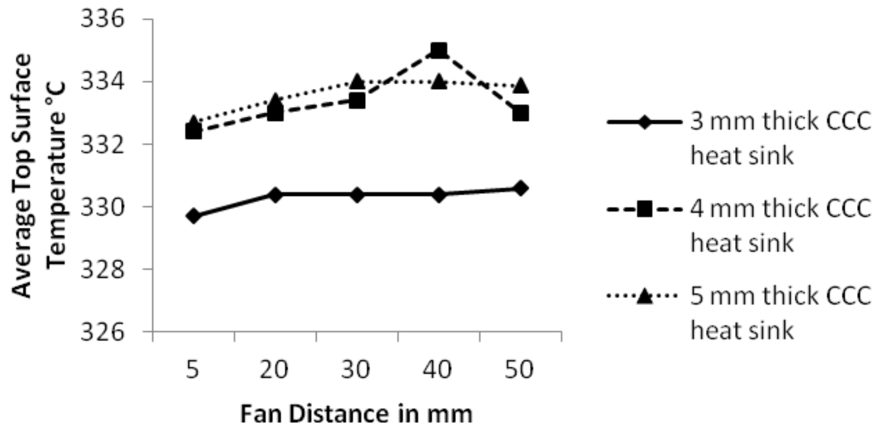

Fig. 8: Top Surface Temperature vs Fan Distance for CCC base plate.

Although the heat sink dimensions are similar, the CCC base plate heat sinks enables higher conduction rates, and heat is conducted to the whole heat sink in a more efficient way is shown in Figure 9and 10.

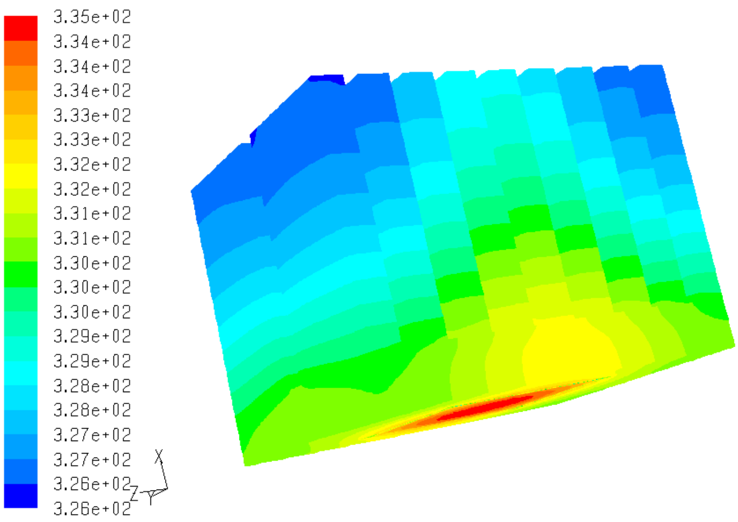

Fig. 9: Temperature distributions on 5mm thick Plate fin Heat sinks with 5mm Cu base plate.

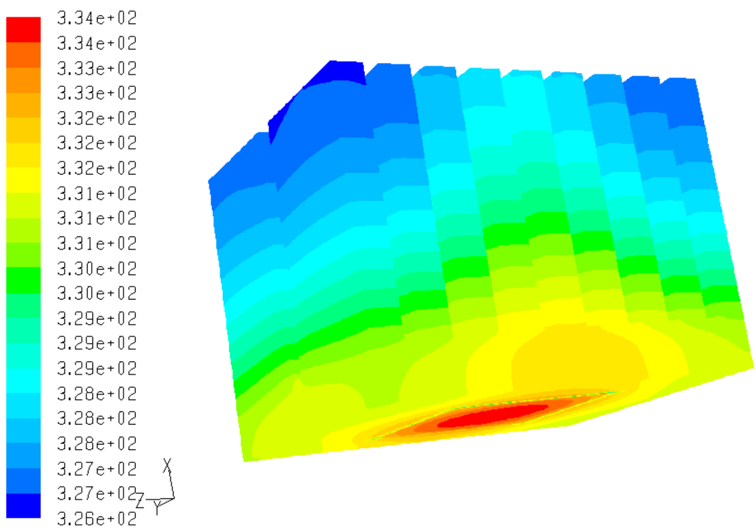

Fig. 10: Temperature distributions on 5mm thick Plate fin Heat sinks with 5mm CCC base plate.

On the other sides of the CPU, air returns to the proximity of the heat sink by hitting the wall, the fan sucks the returning relatively hot air and the cooling becomes less efficient at these sides of the heat sink as can be observed in Figure 11.

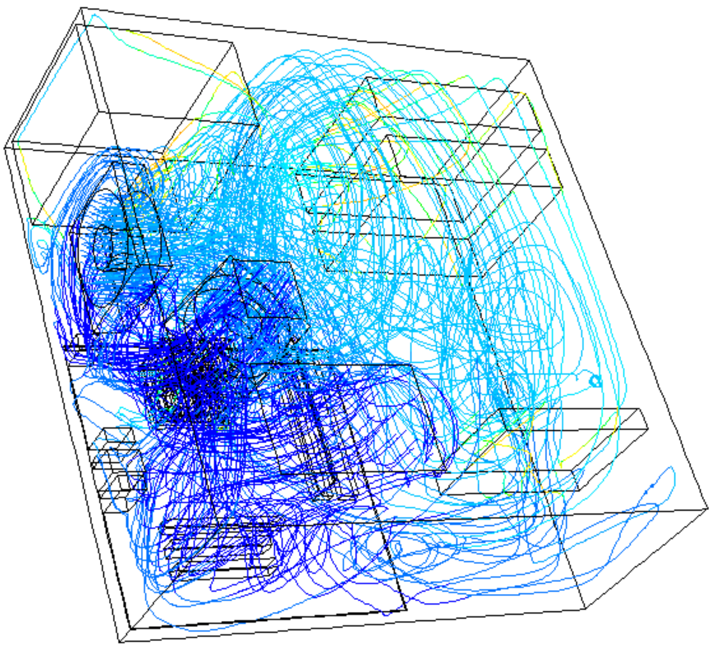

Fig. 11: Path lines and temperature distribution for 3mm thick Plate fin Heat sinks.

It is also observed from these results that modeling not only the CPU- heat sink assembly but the whole chassis which is important for predicting heat sink performance. To investigate this issue further, everything inside the chassis is fixed and the heat sink model is changed. The mesh is kept same, to be able to compare the results with the detailed chassis model. The model with CPU heat dissipation values also resembles the experimental setup. The air can bounce off the chassis walls and recirculate in the chassis, but the temperature distribution is much more symmetric compared to the detailed whole chassis model. it is noticed that the performance of the heat sink is increased by increasing the thickness of fins instead of increasing the number of fins. In the case of a large number fins, it is noticed that the small pitch between fins does not permit air to cool the hottest centre part of the heat sinks. The reduction in heat sink material and weight creates value for the manufacturer. The heat transfer rate is enhanced by increasing the thickness of fin. By increasing the base plate thickness and changing the material of the base plate the performance of heat sink is enhanced. It is also observed that by adding the base plate the heat conduction rate is enhanced in place of increasing the fin height.

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Conclusion:

A hybrid approach was employed to significantly optimize the distance between the fan and heat sink for parallel plate, and based on the experimental results it was found that 5 mm fan distance is the optimum fan distance for getting maximum efficiency. A fan distance does enhance heat dissipation significantly for lower distance. The occurrence of vortices is an adverse factor for heat dissipation. The elimination of the vortex region can be achieved by providing a fan near heat sink. The number of fins and their distribution, fin material, fan distance and base plate thickness are investigated for enhancing the heat dissipation rate from a CPU. Improvements on heat sink designs are made possible by the use of CFD. It is possible to design a new heat sink with a suitable base plate which has better thermal performance and uses less material using CFD simulations. The heat sink base thickness is also a parameter for increasing the performance of heat sink. If the base plate material is selected to be CCC rather than copper or aluminum, then the thermal resistance of the heat sink is decreased. In this paper, three thicknesses of heat sinks with base plate are selected and analyzed. From which the optimal design of heat sink is selected which gives a greater heat transfer rate. It is observed that the velocity field around the heat sink is affected by the presence of the other components inside the chassis as well as the chassis walls which redirect the hot air back into CPU heat sink. If the heat sink is a plate fin type, plate fins can reduce the recirculation. In this study, it is suggested that a parallel plate heat sink model with a CCC base plate will benefit the design engineers involved in electronic cooling.

Nomenclatures:

b	-	Aluminum base thickness (mm)
B	-	Fin width (mm)
B_x , B_y , B_z	-	Components of body force per unit volume of the fluid in x, y and z directions
ccc	-	Carbon carbon composite
Cu	-	Copper
f_p	-	fin pitch (mm)
g	-	Gravitational acceleration (m/s^2)
h	-	Total enthalpy (J/kg)
h	-	Heat transfer coefficient (W/m^{2}K)
H	-	Fin height (mm)
k	-	Thermal conductivity of the fluid (W/mK)
L	-	Heat sink Length (mm)
Nu	-	Nusselt number
P	-	Pressure (bar)
Pr	-	Prandtl Number
Q	-	Power generated (W)
Ra	-	Rayleigh Number
RANS	-	Reynolds averaged navier stokes
R_{th}	-	Thermal Resistance
$T_{bm, in}$	-	Bulk mean inlet temperature (K)
T_∞	-	Ambient temperature (K)
t_b	-	Base plate thickness (mm)
$T_{w,max}$	-	Maximum wall surface temperature (K)
u, v and w	-	Velocity components(m/s)
W	-	Heat sink Width (mm)

Greek Symbols:

ρ	-	Density (kg/m^3)
β	-	Coefficient of cubical expansion
Φ	-	Viscous dissipation
$\dot{g}$	-	Heat generation per unit volume of the fluid (W/m^3)
$\vec{V}$	-	Velocity vector,
k_{eff}	-	Effective thermal conductivity (W/mK)
μ_{eff}	-	Effective viscosity
λ	-	Second viscosity (μm)
μ	-	Dynamic viscosity (kg/ms)

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