

Experimental and Numerical Analysis of Natural Convection and Radiation Heat Transfer in Array of Stepped Bar and Hollow Cylindrical Heat Sinks

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ABSTRACT

Recent development in technologies has placed a greater demand in dissipating the heat produced by the various systems for the optimal performance of the system. The technological development has drastically reduced the sizes of many electronic chips leading to reduction of total heat transfer area available for heat dissipation process making this area very critical for research and development. The heat transfer rate is greatly influenced by various parameters such as materials, temperature gradient, geometry but material type and temperature in most of common applications are fixed but the geometry can be modified to provide the optimal heat transfer rate. In this present paper stepped bar geometry and hollow cylindrical fin geometry is considered and its heat dissipation characteristics, total surface heat flux, nusselt number are calculated by experimental and numerical methods. In this work Ansys 15 is used for numerical computation of the geometry for the respective load and boundary conditions. In the current paper the analysis is focused only on heat transfer due to natural convection and radiation. SIMPLEC algorithm and first order upwind scheme is used for numerical computational analysis of the problem. S2S model is used to model the heat transfer due to radiation.

KEY WORDS: Heat transfer rate, Heat flux, Nusselt number, forced convection, Stepped bar fin, hollow cylindrical fin.

1.INTRODUCTION

Heat sinks are the passive heat exchanger which is used to transfer heat from the surface to the ambient environment. There are two modes of heat transfer from heat sinks forced convective heat transfer and natural convective heat transfer. Radiation also involved in heat transfer from fin but heat transfer due to radiation is negligible in many cases but in this work we are considering the heat transfer due to radiation. Heat sinks are now-a-days commonly found in high power electronic chips to dissipate heat generated to ensure optimal performance of the device. The general principle used in heat sinks is that the heat transfer rate is directly proportional to surface area thus manipulating the surface area is one of the key to increase the heat transfer rate. Air velocity, choice of material, protrusion design and surface treatment are factors that affect the performance of heat sinks.

Numerous researches have been carried out in this domain to explore the possibilities of increasing the efficiency of heat sinks. Leonardo Micheli and his team investigated the thermal performance of natural

convective heat transfer in micro-finned heat sinks and formulated the general correlations among geometry, orientation of heat sinks [2]. Experimental investigation have been carried out by Luigi Ventola [1] and his team to assess the effect of rough surfaces with enhanced heat transfer for electronics cooling by direct metal laser sintering and found that the thermal performance of the heat sinks can be considerably increased by embossing rough surface in heat transfer area of heat sinks.

[3] J.G. Maveety and H.H. Jung investigated the effect of turbulent air impingement over a square pin-fin heat sink both experimentally and numerically and found the optimal dimensional parameters of heat sinks such as fin length and fin cross-sectional area for best cooling performance. The effect of un-uniform fin height on the thermal performance of pin-fin heat sink have been studied by Yue-Tzu[12] and found that there is a potential for increasing the performance of heat sinks by using un-uniform fin length. The coupling of the velocity and the pressure terms of momentum equations are solved by the SIMPLEC algorithm in their work for the numerical computation. Mahdi Fahiminia and his team experimentally investigated natural convection heat transfer of the fin arrangement on a computer heat sink and found the correlation relating the convection heat transfer of fin arrays relative to that for base plate without fins with the relevant non-dimensional parameters. They found the air gap between fins for optimum fin spacing based on which the natural convective heat transfer can be improved [13].

Thermal performance of the heat sinks with base plate for CPU cooling has been analyzed by R.Mohan, P. Govindarajan [7] both experimentally and numerically using commercial software package and found that flow obstructions in the chasis and the resulting air recirculation affect the heat sink temperature distribution. [6] Works have been carried out in system level thermal module in the device that controls dynamic motions in a washing machine called MICOM, this device is accompanied by IPM but the heat produced during it's working is very high causing malfunctioning of the electronic chips STM [6] model also implements analysis of parameters for optimization of heat sinks and reduced material cost. Performance comparison of plate-pin heat sinks and plate fin heat sinks have been investigated by Xiaoling yu and his team [8] and found thermal resistance of a PPFHS is 30% lower than PFHS which results in better performance and better air cooling results. In the article [4] modeling and simulation parallel plate heat sinks have been carried out for the maximum thermal load of 150w. The analysis is carried out by varying fin length, fin base height, fin height, number of heat sinks. Buoyancy effects on heat transfer and temperature profiles in horizontal pipe flows are studied by Gasljevic and his team of scholars and found that temperature profiles of drag reducing fluids are greatly affected by the action of buoyancy under laminar flow conditions.

II. FACTORS AFFECTING THERMAL PERFORMANCE OF HEAT SINKS

- i. Thermal resistance
- ii. Temperature gradient
- iii. Material
- iv. Fin efficiency
- v. Spreading resistance
- vi. Geometry

- vii. Fin arrangements
- viii. Surface color
- ix. Surface roughness

III. OBJECTIVES OF WORK

- To analyze the proposed geometry of heat sinks experimentally and numerically using the Fluent solver.
- To compute surface Nusselt number.
- To compute total surface heat flux.
- To compute total radiation heat flux.
- To measure the thermal performance of the heat sinks.

IV. MATHEMATICAL MODEL AND NUMERICAL ANALYSIS

The schematic diagrams of array of pin-fin heat sink, stepped bar heat sink and hollow cylindrical heat sink are shown below in Fig. 1, Fig. 2, and Fig. 3 respectively and their geometric parameters are listed below in the respective tables.

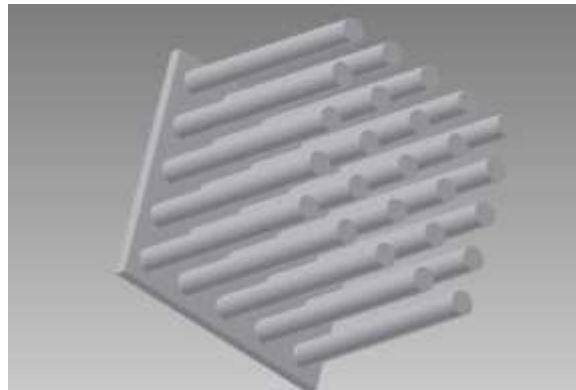


Figure 1.

Table 1. Array of pin-fin parameters

Fin Height H (mm)	Fin Number N(mm)	Fin diameter D(mm)	Base plate thickness T(mm)
90	25	10	10



Figure 2.

Table 2. Array of stepped bar heat sinks

Fin Height H (mm)	Fin Number N (mm)	Fin diameter D (mm)	No. of steps “n”
90	25	10	2

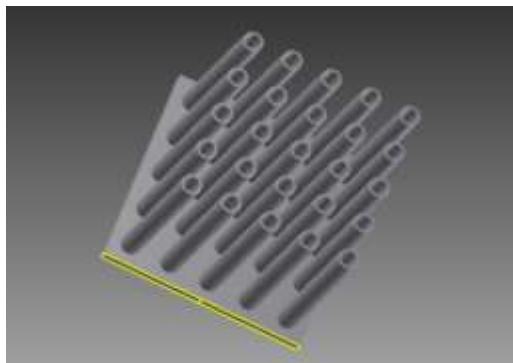


Figure 3.

Table 3. Array of hollow cylindrical heat sinks

Fin Height H (mm)	Fin Number N (mm)	Fin diameter D (mm)	No. of pores per fin “n”
90	25	10	8

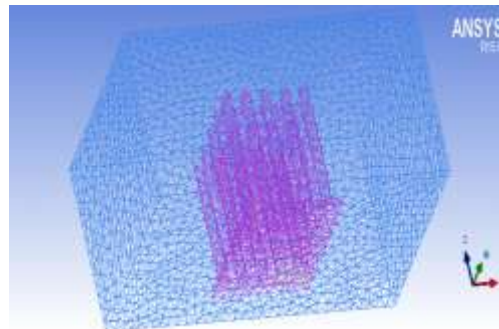


Figure 4. Fluid Domain

The numerical analysis is only carried out for heat transfer due to natural convection and radiation. The surface to surface model is used to model the heat transfer due to radiation in the heat sinks. When heat is added to a fluid and the fluid density varies with temperature, a flow can be induced due to the force of gravity acting on the density variations. Such buoyancy-driven flows are termed natural-convection (or mixed-convection) flows. The heat transfer due to natural convection is solely governed by the action of buoyancy-driven flow theory, the relation between grashof number and Reynolds number is given by,

$$Gr/Re^2 = g\beta dTL/\nu$$

$$Ra = g\beta dTL^3\rho/\mu\alpha$$

Rayleigh number less than 10^8 indicate buoyancy-induced laminar flow, with transition to turbulence occurring over the range of $10^8 < Ra < 10^{10}$.

Where,

$$\alpha = k/\rho c_p$$

The temperature distribution $T(x,t)$ in the heat sinks is given by the following differential equation,

$$dT/dt + dT/dx - \alpha d^2T/dx^2 = 0$$

The algorithm solves the problem by the following Navier-stokes equations for an incompressible 3-D flow.

Continuity equation,

$$du/dx + dv/dy + dw/dz = 0 \quad (1)$$

X-component of the momentum equation,

$$\rho Du/Dt = dp/dx + (d\sigma_{xx}/dx + d\sigma_{xy}/dy + d\sigma_{xz}/dz) + pf_x \quad (2)$$

Y-component of the momentum equation,

$$\rho Dv/Dt = dp/dy + (d\sigma_{yx}/dx + d\sigma_{yy}/dy + d\sigma_{yz}/dz) + pf_y \quad (3)$$

Z-component of the momentum equation,

$$\rho Dw/Dt = dp/dz + (d\sigma_{zx}/dx + d\sigma_{zy}/dy + d\sigma_{zz}/dz) + pf_z \quad (4)$$

The SIMPLE algorithm is used for pressure correction in numerical method, the linearized navier stokes equation written in operator form is given by,

$$U = Au + Bp + C \quad (5)$$

Calculation of the auxiliary velocity for buoyancy driven flow is given by,

$$U^* = Au^* + Bp^{(m-1)} + C \quad (6)$$

Calculation of the pressure correction,

$$DBp' = -Du^* \quad (7)$$

Calculation of the velocity correction,

$$U' = Bp' \quad (8)$$

First order upwind scheme is employed because of major inadequacies of the thermal differencing scheme because of its inability to identify the flow direction. The heat sinks are modeled using Auto desk inventor and Discretization have been carried out using the ICEM CFD-15 software and analyzed in Ansys FLUENT 15 solver. The base plate is subjected to 3 different heat loads and the respective temperature contours and various heat transfer rates required are simulated. The momentums for solving equations are taken as .4. The Tetrahedral mesh is used to discretize the fluid domain because of its accuracy over other types of discretization method.

V. EXPERIMENTAL ANALYSIS





The experimental setup consists of 4 thermocouples of j type which is fixed to 3 equidistant points in the heat sinks and one thermocouple is fixed to the base plate of the heat sinks to measure the input temperature, thermocouples are connected to the temperature indicator to take readings of the temperature in various points in the heat sinks. The heat transfer rate per heat sink is calculated and the value is extrapolated for the total number of heat sinks used for numerical analysis. In the article[8] the pin fin is introduced between the plate heat sinks and the performance of the heat sinks are studied at full scale model but in this study only one heat sink is taken for each profile and the performance is studied and the results are extrapolated for numerical model. The ammeter and voltmeter is used to vary the power input to the heat sinks based on which the temperature of the base plate is controlled. The different profiles of the heat sink are fabricated using the CNC machining process for quality of the heat sink surface. The procedure includes fixing the desired input power in the voltmeter and ammeter and allowing the system to attain steady state condition and after that reading is noted in temperature indicator for various thermocouple position in the heat sinks. And the corresponding base plate temperatures are noted down. The same procedure is carried out for all heat sink models and results are tabulated and calculations have been carried out to determine the heat transfer rates. The base plate dimensions are 100*100 mm and the thickness of the base plate is 10mm. The heat transfer only due to radiation and natural convection are taken into account. Other forms of heat transfer are neglected.

VI. RESULTS

6.1 Temperature contours.

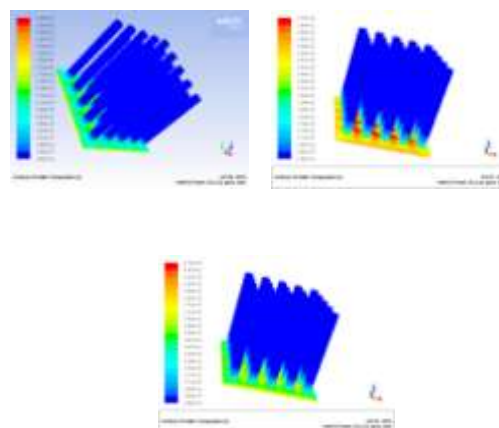


Figure 5. Temperature contours of pin-fin profile for base plate temperature (333k, 373k, and 413k)

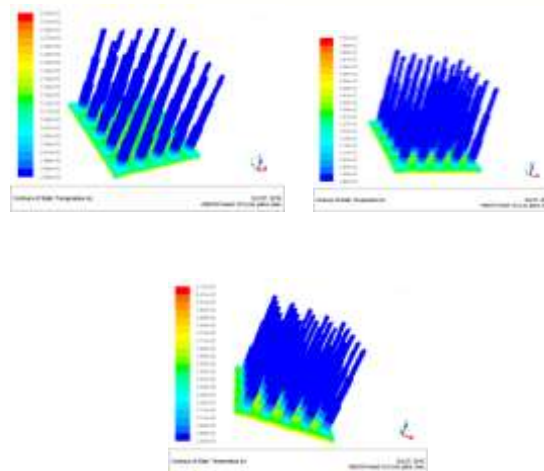


Figure 6. Temperature contours of stepped bar profile for base plate temperature (333k, 373k, 413k)

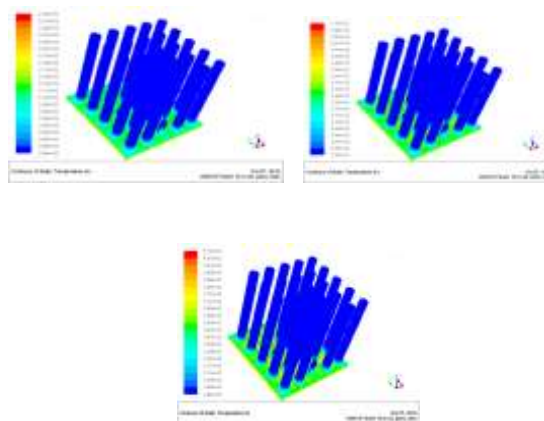


Figure 7. Temperature contours of hollow cylindrical heat sink profile for base plate temperature (333k, 373k, and 413k)

6.2. Experimental and Numerical Analysis Data's

Table 4. Data's for pin-fin heat sinks

Input (T) k	Total heat transfer rate W	Radiation heat transfer rate W	Experimental total heat transfer rate W
333	3.98	2.89666	3
373	6.234	4.1976	5.6

413	9.51	4.19	7.788
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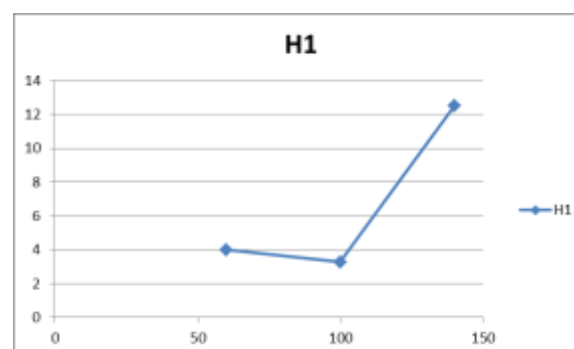
Table 5. Data's for stepped bar profile heat sinks

Input (T) k	Total heat transfer rate W	Radiation heat transfer rate W	Experimental total heat transfer rate W
333	2.7274	1.65206	2.5
373	5.7331	3.6371	5.33
413	8.9545	5.8550	7

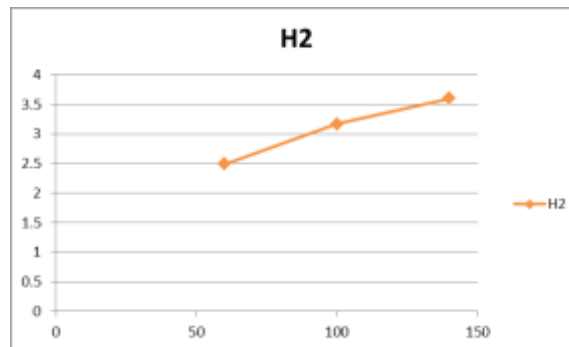
Table 6. Data's for hollow cylindrical profile heat sinks

Input (T) K	Total heat transfer rate W	Radiation heat transfer rate W	Experimental total heat transfer rate W
333	3.007	1.9476	2.7
373	5.35	3.17	5
413	8.889	5.7392	7.22

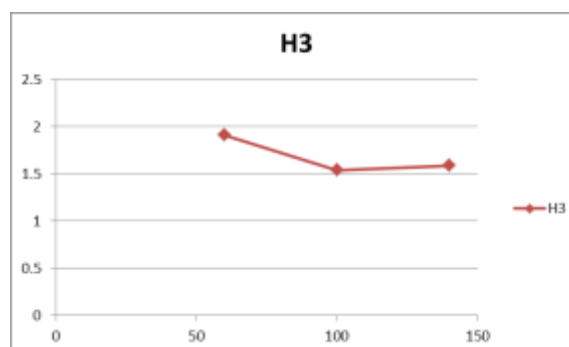
6.3. Graphical results



Graph 1. T vs. H₁

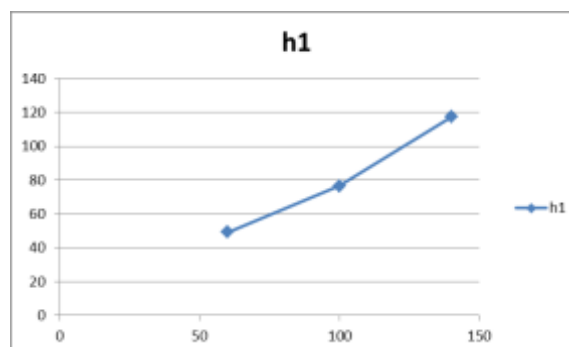


Graph 2. T vs. H₂

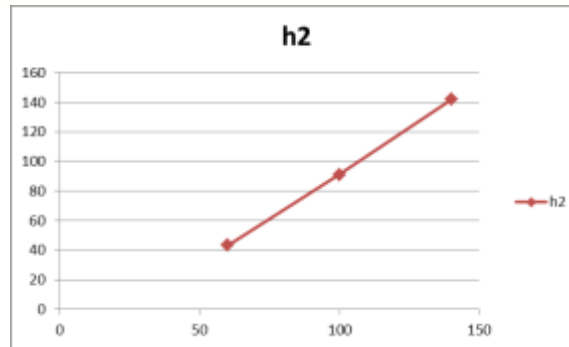


Graph 3. T vs. H₃

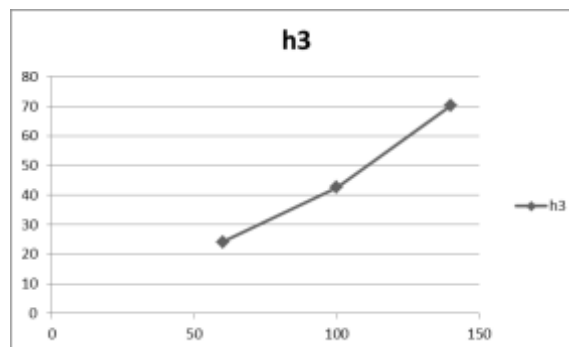
The above set of graph shows the plot between temperature and heat transfer co-efficient, where H₁, H₂, H₃ refers to heat transfer co-efficient of pin-fin heat sink, stepped bar heat sink, hollow cylindrical heat sink.



Graph 4. T vs. h₁



Graph 5. T vs. h2



Graph 6. T vs. h3

The above set of graph shows the plot between temperature and Total surface heat flux, where h1, h2, h3 are total surface heat flux of pin-fin heat sink, stepped bar heat sink, hollow cylindrical heat sink.

VII. CONCLUSIONS

The Experimental and numerical analysis of the heat sinks of three different geometry have been carried out and results are tabulated above. It is evident from total heat transfer rate in Table 1, 2,3, that heat transfer due to radiation dominates the natural convective heat transfer rate. The simulation results are found to be in good agreement with the experimental values. The temperature contours of the heats sinks have been shown in above figures. The significant observation is that the heat transfer rates for pin-fin and stepped bar heat sinks are nearly equal which shows that instead of pin-fins stepped bar heat sinks can be employed which will result less material consumption in large scale. Similarly the heat transfer rates of hollow cylindrical heat sinks are very low when compared to other types of geometry considered. The T vs. H_1 shows that there is sudden increase in surface heat transfer co-efficient when increasing the base plate temperature. The graph T vs. H_2 shows that the surface heat transfer co-efficient is gradually increasing with temperature. It is clear from the observations that hollow cylindrical heat sinks have least performance characteristics. And stepped bar heat sinks have better performance characteristics.

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