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Analysis of Natural Convection around Radial Heat Sink: A Review

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Abstract:- ***This paper presents the details of an experimental and numerical investigation of natural convection in a radial heat sink, composed of a horizontal circular base and rectangular fins. The general flow pattern is that of a chimney; i.e. cooler air entering from outside is heated as it passes between the fins, and then rises from the inner region of the heat sink. Parametric studies are performed to compare the effects of three geometric parameters (fin length, fin thickness, fin height, and number of fins) and a single operating parameter (heat flux) on the thermal resistance and the average heat transfer coefficient for the heat sink array. Find out one optimum design from various fin structure (heat sink) and Compare of the temperature difference between the experimental and numerical. The variables for natural convection cooling with the help of finned surfaces are orientation and geometry where single chimney flow pattern is present, a stagnant zone is created at the central bottom portion of fin array channel and hence it does not contribute much in heat dissipation. Hence it is removed in the form of inverted notch at the central bottom portion of fin to modify its geometry for enhancement of heat transfer. An experimental setup is developed for studying the investigation on normal and inverted notched fin arrays (INFAs). Fin spacing, heater input and percentage of area removed in the form of inverted notch are the parameters. For few spacing, it is verified by computational fluid dynamics analysis (Course Notes on Introduction to Commercial CFD of and the results are well) matching. It is found that the average heat transfer coefficient for INFAs is nearly 30–40% higher as compared with normal array.***

Key word: **Natural Convection, Heat Flux, Heat Transfer Coefficient, Thermal Resistance, Surface Area.**

I. INTRODUCTION

In electronic systems, a heat sink is a passive component that cools a device by dissipating heat into the surrounding air. Heat sinks are used to cool electronic components such as high-power semiconductor devices, and optoelectronic devices such as higher-power lasers and light emitting diodes (LEDs). ight-emitting diode (LED) lights have recently attracted the attention of the illumination industry, due to their lower power consumption, longer life, and smaller, more durable structure compared to other light sources. However, their use presents a thermal problem, since about 70% of their total energy consumption is emitted as heat. An efficient heat sink design is essential to solve this problem. Natural convection heat sinks are appropriate for LED lights, considering their overall advantages. However, natural convection heat sinks commonly have rectangular bases, whereas LED lights are generally circular. It is therefore desirable to investigate natural convection heat transfer via a heat sink with a circular base. (1)

II. MATHEMATICAL MODELING

Fig. 1 shows a radial heat sink consisting of a circular base and rectangular fins. The fins were arranged radially at regular intervals. The heat sink base was oriented horizontally. The heat sink was made of aluminum, whose properties are listed in Table 2.(2)

Governing equations For the numerical analysis, the Following assumptions were imposed.

- The flow was steady, laminar, and three-dimensional.
- Aside from density, the properties of the fluid were Independent of temperature.
- Air density was calculated by treating air as an ideal gas.
- Radiation heat transfer was negligible.

b	spacing between fins, mm
c_p	coefficient of heat capacity, J/(kg °C)
F	view factor
h	heat transfer coefficient, W/m^2 K
H	fin height, mm
k	thermal conductivity, W/m °C
L	fin length, mm
M_w	gas molecular weight, kg/kmol
Nu	Nusselt number, hL/k
n	number of fins in the normal direction
Pr	Prandtl number
p	pressure, N/m^2
$\dot{q}$	heat flux, W/m^2
R_c	universal gas constant
R_{TH}	thermal resistance, °C/W
Ra*	modified Rayleigh number, $\frac{\rho^2 g\beta c_p \pi (r_o^2-r_i^2)\dot{q}L^3}{\mu k^2}$
r	radius, mm
T	temperature, K or °C
t	fin thickness, mm
u	x-component of velocity, m/s
v	y-component of velocity, m/s
w	z-component of velocity, m/s

Greek symbols

ε	emissivity
μ	dynamic viscosity, N/m^2 s
θ	angle, °
ρ	density, kg/m^3
σ	Stefan–Boltzmann constant, 5.67×10^{-8} W/m^2 K^4

Subscripts

avg	average
f	fluid (air)
i	inner
o	outer
s	solid (heat sink)

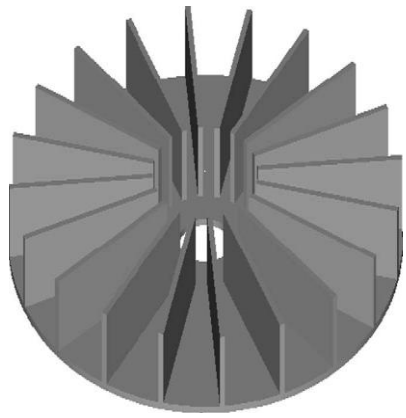

Fig. 1. Radial heat sink with a circular base and rectangular fins

Periodic boundary conditions were adopted in accordance with the geometry of the heat sink (Fig. 1). Because of the number of grids and the computational time involved, only a single fin was considered.(3)

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III. TAGUCHI EXPERIMENT: DESIGN AND ANALYSIS

Essentially, traditional experimental design procedures are too complicated and not easy to use. A large number of experimental works have to be carried out when the number of process parameters increases. To solve this problem, the Taguchi method uses a special design of orthogonal arrays to study the entire parameter space with only a small number of experiments .Taguchi methods have been widely utilized in engineering analysis and consist of a plan of experiments with the objective of acquiring data in a controlled way, in order to obtain information about the behavior of a given process. The greatest advantage of this method is the saving of effort in conducting experiments; saving experimental time, reducing the cost, and discovering significant factors quickly. Taguchi's robust design method is a powerful tool for the design of a high-quality system. [9]

IV. EXPERIMENTAL MODEL

Optimization model by taguchi: - Length of fins, height of fins and number of fins.

Table 1

Model no.	Length of fins	Height of fins	Number of fins
Model 1	35	15	24
Model 2	35	20	28
Model 3	35	25	32
Model 4	45	15	32
Model 5	45	20	24
Model 6	45	25	28
Model 7	55	15	28
Model 8	55	20	32
Model 9	55	25	24

V. NUMERICAL PROCEDURE AND VALIDATION

The numerical simulation was conducted using a commercially available CFD code based on the finite volume method. The grid dependence was investigated by varying the number of grid points. Additional grid points produced a change of less than 0.5% in the average heat sink temperature for the reference model. The numerical results were validated with experimental data by comparing the differences between the ambient and heat sink temperatures. The geometric parameters of the experimental model were n = 24,48 & 32, L =35mm,45mm & 55 mm, H = 15 mm, 20mm &25 mm, r_0 = 80 mm, r_i=10 mm and t = 2 mm. Fig. 3 compares the temperature differences between the experimental and numerical results in terms of the heat flux applied to the heat sink base. This implies that the present numerical model can correctly predict the natural convection flow around a radial heat sink [2].

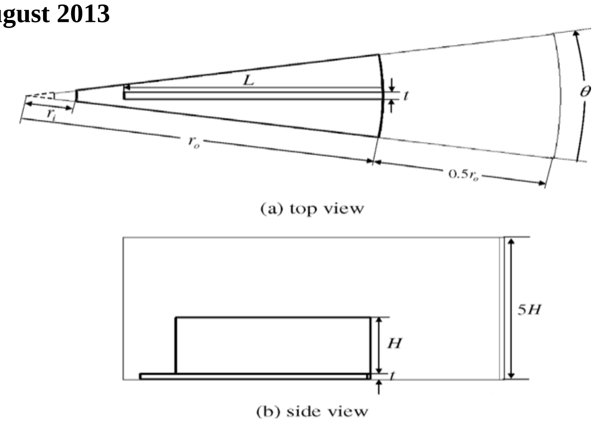

Fig.2 Computational Domain and Dimension.

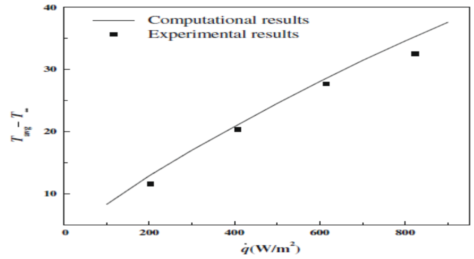

Fig.3. Comparison of the Temperature Differences between the Experimental and Numerical Result.

Table 2.Air and heat sink properties

Material	c_p (J/ kg °C)	μ (N/m^2 s)	k (W/m °C)	ρ (kg/ m^3)
Air	1005.585	1.834×10^{-5}	2.643×10^{-5}	Eq. (7)
Heat sink (aluminum)	2800	-	193	880

VI. EXPERIMENTAL INVESTIGATION

The numerical model is verified with experimental data, by comparing the differences between ambient and heat sink temperatures. The heat sink is made of aluminum (Al2014), with no additional surface treatment. As Fig. 1 shows, the experimental setup consisted of a film heater a heat sink, an insulator, type-k thermocouples a power supply, a wattmeter, and a personal computer. The film heater is attached to the bottom surface of the heat sink. Thermal grease is used to minimize the thermal contact resistance between the film heater and the heat sink. To reduce heat loss, the film heater Section is surrounded by an insulator, heat sink temperatures are measured with eight thermocouples (located at four points of the central region and four points of the outer region of the upper heat sink base), and the ambient temperature is measured with two thermocouples. For the numerical simulation, radiation heat transfer is neglected. In the experiments, however, natural Convection and radiation heat transfer occurs simultaneously.

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VII. METHODOLOGY

Experiments are performed and steady-state observations are recorded. Fin spacing, heater input, and percentage of area removed are the parameters of experimental study. Table 2 summarizes the parameters included in the experimentation. Radiation loss is accounted suitably by the predetermined values of emissivity, which are 0.5 for no blackened and 1 for black array. [11]

VIII. OBJECTIVES

A) Design and manufacture the various fin structure by changing the parameter like number of fin, fin length, fin height.
B) Parametric studies were carried out by numerically investigating the effects of the number of fins, fin length, fin height, and heat flux on the thermal resistance and the heat transfer coefficient.
C) Find out one optimum design from various fin structure (heat sink).
D) Compare of the temperature difference between the experimental and numerical results.

IX. RESULTS AND DISCUSSION

Parametric studies were carried out by numerically investigating the effects of the number of fins, fin length, fin height, and heat flux on the thermal resistance and the heat transfer coefficient. Based on these results, a correlation was proposed to predict the Nusselt number for a heat sink with a horizontal circular base and rectangular fins.

X. THERMO-FLOW CHARACTERISTICS

There are two flows, i.e., vertical and horizontal flows, around the radial heat sink. The vertical flow is in the upward direction, since air is heated by the heat sink (which is maintained at a higher temperature) and becomes lighter than the surrounding air. The horizontal Flow is created by air entering from outside the heat sink to make up for the vertical flow in the inner region. Therefore, the overall flow pattern is chimney-like. The temperature of heat sink maintains almost uniformly high because of high conductivity of aluminum. The heat transfer rate in the outer region of the heat sink was higher than in the inner region. This was because the temperature difference between the air and the heat sink decreased as the cool air proceeded towards the inner region of the heat sink.

XI. PARAMETRIC STUDY

The effects of the number of fins, fin length, fin height, and heat flux on the thermal resistance and the heat transfer coefficient were investigated.. The effect of the number of fins on the thermal resistance and heat transfer coefficient is shown in Fig. 4(a). The average heat transfer coefficient decreased as the number of fins increased, since the flow rate of the cooler air entering the spaces between the fins decreased and the air was heated more quickly on account of the reduced space between fins. However, when the number of fins was less than 36, the thermal resistance of the heat sink decreased with increasing n, since the effect of the increased heat transfer surface area was larger than the effect of the decreased heat transfer coefficient. When the number of fins was greater than 36, the thermal resistance of the heat sink increased with increasing n, since the heat transfer coefficient was very small. Consequently, there exists optimum number of fins that gives the minimum thermal resistance.[3] Fig. 4(b) shows the effect of the fin length. As the fin length increased, the thermal resistance and average heat transfer coefficient decreased. The thermal resistance leveled off and reached a steady value when the fin was longer than 55 mm. This was because the air temperature in the inner region was almost the same as the heat sink temperature, and hence any additional fin length beyond 55 mm did not contribute to the heat transfer rate.[3] Fig. 4(c) indicates the effect of the fin height. A lower thermal resistance resulted from the increased heat transfer surface area created by the incremented fin height. However, the change in the heat transfer coefficient was relatively small, since the velocity of the air entering from outside increased very little with increasing fin height. [3] Fig. 4(d) illustrates the effect of the heat flux applied to the heat sink base. The decrease in thermal resistance due to increasing heat flux resulted in a greater rising air velocity, which in turn increased the flow rate of the cooler air entering from outside. Accordingly, the average heat transfer coefficient increased almost linearly, thanks to the enhanced effect of natural convection.[3]

XII. FLOW VISUALIZATION

Figure 5 shows the photographs of flow visualization by simple smoke technique using for normal and 30% INFAs. From photographs, it is concluded that coalescing of two streams at less height from fin bottom in normal fin array whereas it is at more height in INFAs, giving wider chimney and enhancing the heat transfer rat Single chimney flow pattern is retained in INFAs also with a wider chimney zone, which is the possible reason for heat transfer enhancement. When single chimney flow pattern is present, in mid channel stagnant bottom portion becomes ineffective. The modified array is designed in inverted notched form and that has proved to be successful retaining single chimney together with the removal of ineffective fin flat portion. This is the main contribution of present paper. Limited CFD solutions obtained are in good agreement with experimental work. Radiation contribution is also important and needs further investigation. [11]

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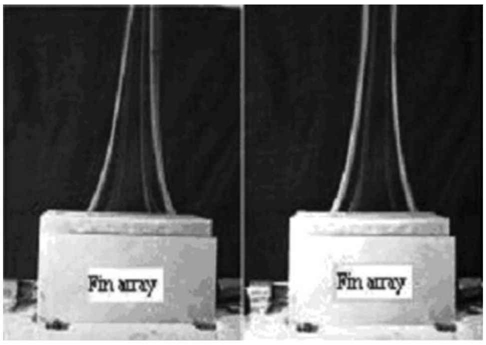

Fig 4 Photograph Showing Simple Visualization by Simple Smoke Technique

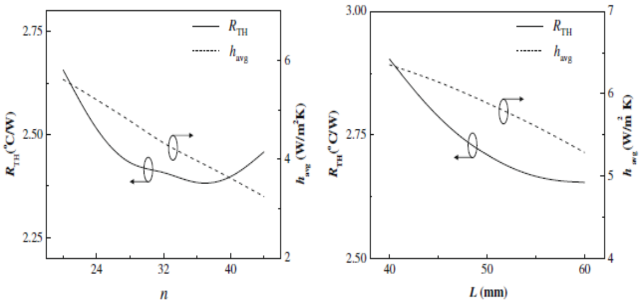

Fig. (a) The effect of number fins. Fig. (b) The effect of fin length.

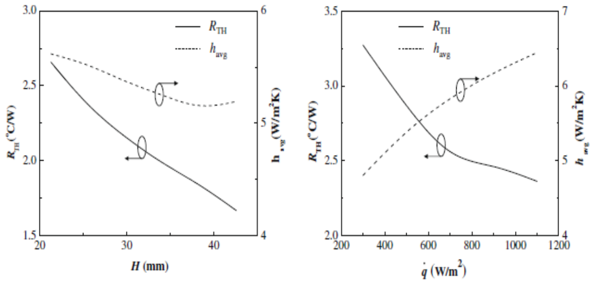

Fig. (c) The effect of fin height. Fig. (d) The effect of fin heat flux.

Fig. 5 Comparison of the temperature differences between the experimental and numerical results.

XIII. CORRELATION

A correlation for predicting the Nusselt number for a heat sink with a horizontal circular base and rectangular fins was derived as a function of the parameters investigated in the previous section, as well as other geometric parameters, and was obtained from numerical data. This formula is based on the correlations for rectangular heat sinks obtained in previous studies, using average fin spacing and the modified channel Rayleigh number and the properties are based on the film temperature. The predicted correlation was consistent with the numerical data, with an error of less than 10%, when t = 2 mm, $20 \leq n \leq 36$, 21.3 mm$\leq$ H$\leq$ 63.9 mm, 75 mm $\leq r_0 \leq$ 102 mm, 40 mm $\leq$L $\leq$ 80 mm, and 300W/m2 $\leq$q $\leq$ 1100W/m2.[3]

XIV. CONCLUSION

Natural convection from a radial heat sink was experimentally and numerically investigated. The general flow pattern was like that of a chimney; i.e., the cooling air entering from outside was heated as it passed between the fins, and then rose from the inner region of heat sink. Parametric studies were performed to compare the effects of the number of fins, fin length, fin height, and heat flux on the thermal resistance and the heat transfer coefficient. As the number of fins, fin length, and fin height increased, the thermal resistance and heat transfer coefficient generally decreased. However, there existed optimal values of the number of fins and fin length to obtain an effective low heat sink temperature. The thermal resistance decreased and the heat transfer coefficient increased in proportion to the heat flux applied to the heat sink base. A correlation was proposed to predict the average Nusselt number for a radial heat sink.

XV. ACKNOWLEDGMENT

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