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THERMAL PERFORMANCE OF RADIAL HEAT SINKS UNDER FORCED CONVECTION

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Abstract

Electronic devices, e.g. projector, high power semiconductors devices, LED, chips, etc., require efficient cooling methods for dissipation of heat in a limited region. These devices should be maintained at low and acceptable temperature for their efficient working. So the heat dissipation from these devices becomes critical factor. One of the solutions to remove the heat from such devices is to use heat sink in natural or forced convection. This study the performance of radial heat sink under forced convection is investigated. The different parameters which are considered for the study of radial heat sink are number of fins, height of the fin, length of the fin, velocity of air and the heat input. To investigate the thermal performance under forced convection of a radial heat sink different models are designed and manufactured using the Taguchi method of design of experiments. The experiments are performed and the data collected is analyzed using analysis of variance (ANOVA) to observe the effect of various parameters on the performance of the radial heat sink. The effects of various parameters have been experimentally investigated and the optimum values are found out for radial heat sink under forced convection.

Conclusion

Experimental investigations are carried out for the thermal performance of radial heat sink under forced convection. The Taguchi method of design of experiments (DOE) is used to determine the influence of various parameters of radial heat sink on the heat transfer

coefficient and Nusselt number under forced convection. The various parameters examined are number of fins, fin length, fin height, heat flux and velocity of air.
On the basis of studies carried out the following conclusions are drawn.

- For low velocity i.e. 3 m/s the number of fins is the dominating parameter which has maximum impact on the performance of radial heat sink.
- For moderate velocity i.e. 6 m/s the length of fins have somewhat more impact on the performance.
- Whereas for high velocity i.e. 9 m/s the height of the fins have highest impact on the performance of heat sink under forced convection.

Key words: ANOVA, Design of Experiments, Forced Convection, heat sink.

INTRODUCTION

Heat sinks are the heat exchanging devices that absorb the heat from the system and dissipate it to the surrounding. So heat sinks are extensively used to increase heat transfer rate from electronic systems and components to the surrounding fluid. Generally the heat sinks are made of metals with base having integrated fins arranged in different fashions. The fins on the base of heat sink increases the surface area to increase the heat transfer rate. Active cooling systems are mechanically assisted. Active cooling method offer high cooling rate. They allow temperature control that can cool even below ambient temperatures. Active cooling methods require external energy and they usually involve higher cost, volume and noise than passive methods. Also active techniques are intrinsically at risk of malfunctioning. Active cooling methods include air/liquid jet impingement, forced liquid convection, spray cooling thermoelectric coolers, refrigeration and thermo ionic cooling. Scott [1] classified all the methods into four broad categories in order of increasing heat transfer effectiveness, for the temperature difference between the surfaces and the ambient is 80°C and also compared the methods as shown in figure[1].

- Natural convection and radiation (155-1550 W/m2),
- Forced air-cooling (800-16000 W/m2),
- Forced liquid cooling (11000-930000 W/m2), and
- Liquid evaporation (15500-1400000 W/m2).

Taguchi method with L25 orthogonal array was used for four operating parameters that are aspect ratio, approach velocity, void fraction and heat flux with five levels by P. A. Deshmukh and R. M. Warkhedkar [2]. The regression analysis was carried out to provide the platform to decide the selection and omission of the parameters for further experimentation by studying the Pareto charts. R. Senthilkumar, et al[3] analyzed natural convective heat transfer of nano coated aluminium fins using Taguchi method. The temperature and heat transfer characteristics were investigated using Nusselt, Grashof, Prandtl and Rayleigh numbers and also optimized by Taguchi method and ANOVA analysis. S. Manivannan, S. Prasanna Devi, et al[4] presented an approach for the multi objective optimization of the flat plate heat sink using Taguchi design of experiments-based Grey relational analysis. Isak Kotcioglu et al.[5] in their paper presented the determination of optimum values of the design parameters in a plate-fin heat exchanger with a rectangular duct by using Taguchi method. The effects of the six design parameters such as the ratio of the duct channel width to height, the ratio of the winglets length to the duct channel length, inclination angles of winglets, Reynolds number, flow velocity and pressure drop were investigated. An L_{25} (5^6) orthogonal array was chosen to conduct the experiments. The analysis of Taguchi method conducted with an optimization process to reach minimum pressure drop (friction factor) and maximum heat transfer (Nusselt number) for the designed heat exchanger. K. Al-Jamal, H. Khashashneh[6] conducted the experiments with triangular and pin fin array at constant heat flux. Nusselt number was determined as a function of Re_{max} at constant Pr = 0.7 for both sets of experiments. From the experiment results an equation is obtained to determine Nu as a function of Re_{max}. Md. Farhad Ismail, Muhammad Noman Hasan and Mohammad Ali[7] investigated the thermal and fluid dynamic performances of perforated heat sinks under turbulent flow. It was concluded that under turbulent flow conditions, the perforations increase thermal performance of heat sinks. It also showed that fin effectiveness value is greater for perforated fin as compared to other fins. There is sharp reduction of total drag force so the power required for the cooling fan is less that the solid fins. Perforation shape is also an important factor for the better fluid dynamic performance. Wayne L Staats and J.G. Brisson [8] investigated the enhancement of convective heat transfer air cooled heat sink using integrated, interdigitated impellers. The close integration of impeller blades with heat transfer surface results in a decrease thermal resistance per unit pumping power compared to conventional forced convection heat sink. The fan design of the impeller had optimized independently for the heat transfer performance. Diego Copiello and Giampietro Fabbri[9] had done the optimization of the heat transfer from wavy fins which are cooled by a laminar flow under the conditions of forced convection. A finite element method was used for computing the velocity and the temperature distributions in a finned conduit cross section under conditions of imposed heat flux. Thereafter, the fin profile was optimized by means of multi-objective genetic algorithm which aims to find geometries that maximize the heat transfer and, at the same time, minimize the hydraulic resistance. The geometry of the fins was parameterized by means of a polynomial function and several orders were investigated and compared.

EXPERIMENTAL METHOD

The main objective of the present work is to perform the experiments to study the effects of various geometrical parameters on the heat transfer.

2.1 Calculation of Thermal Performance for Heat Sink

The radial heat sink is made up of number of individual fins which are rectangular in shape arranged radial to the base of the heat sink. The heat is supplied from the base of the heat sink which is arranged horizontally and radial fins vertically on it.

According to Newton's law of cooling the convective heat transfer rate is given by the following equation [10],

$$Q = h\,A\,(T_{avg} - T_a) \qquad (1)$$

Where the A is the total surface area of the heat sink, Tavg and Ta are the average temperature of the heat sink and ambient temperature respectively. The emissivity of the heat sink material is very less i.e. ε = 0.055, so the radiation heat loss is neglected. Also it is for all the models so if any loss of heat by radiation it will same for all model.

The heat loss to the insulation material by conduction is also very less and it is similar for all the models so it is also not considered. The heat input given to the base of the heat sink is dissipated to the surrounding air. When the heat transfer rate is high then the temperature difference between the system and surrounding will be minimum.

The heat transfer coefficient h can be obtained from equation (2),

$$h = \frac{Q}{A(Tavg - Ta)} = \frac{(V \times I)}{A(Tavg - Ta)} \qquad (2)$$

Where V is the voltage and I is the current. The value of heat transfer coefficient h is used to calculate the dimensional less parameters Nusselt number Nu which gives the effect of convective heat transfer and conductive heat transfer in the fluid. The Nusselt number defines the type of heat transfer in the fluid higher the value gives high rate of convective heat transfer than conductive heat transfer.

$$Nu = \frac{h \times L}{k} \quad (3)$$

Where L is the characteristic length and here for the heat sink it is the height 'H' of the fins. Other dimension less numbers used for the forced convection are number Prandtl number (Pr) and Reynolds number. The equations for Prandtl number (Pr), Reynolds number (Gr) are given below.

$$Pr = \frac{\mu C_p}{k} \quad (4)$$

$$Re = \frac{UH}{\nu} \quad (5)$$

To find out the thermo physical properties of air required in above dimension less number mean temperature of the air is used, which is given as below.

$$T_{mean} = T_{avg} + T_a \quad (6)$$

2.2 Selection of the Geometrical Parameters of Radial Heat Sink

As the effect of the design parameter on the performance of heat sink is the motto of present study so the outer diameter of the base of the heat sink is selected as 160. Then according to the base outer diameter of the heat sink the other design parameters are selected which are given in table 1. As per the present trends of the power usage for the electronics devices the heat inputs for the experimentation under forced convection are selected as 80 W, 120 W, 160 W, 200 W and 240 W with various air velocities 3 m/s, 6 m/s and 9 m/s.

2.3 Experimental Models for the Natural Convection and Forced Convection

The Minimum Number of Experiment (MNE) combinations for conducting experiments are given by Taguchi orthogonal arrays. As per the number of parameters and the number of level here Taguchi L9 orthogonal array is used for the experimentation.

The experimental setup for forced convection is shown in figure 3. Air flow bench is used to supply uniform air on the heat sink for forced convection.

RESULT AND DISCUSSION

3.1 Radial Heat Sink under Forced Convection

The experimental data obtained for heat sink under forced convection is analyzed by ANOVA in Minitab Software. The 9 radial heat sinks were used to perform the experiments under forced convection using airflow bench. The outcome of this analysis is shown in figures 4 to figure 6 for various heat inputs and different velocities. The aim of present study is to find out the effect of different design parameters of the radial heat sink under forced convection heat transfer. Along with the designing parameters of radial heat sink that are length of fin (L), height of fin (H), number of fins (N), heat input another parameter i.e. air flow velocity is added for the analysis. The mean effect plot for all the heat inputs and velocities are plotted and these are useful to find the effect of individual parameters and the performance of the radial heat sink. The effect of the length of fin, height of fin and number of fins on the Nusselt number for 3 m/s velocity in terms of main effect plot for mean is shown in figure 4. Whereas for the 6 m/s and 9 m/s velocities shown in figure 5 and figure 6 respectively. The effect of individual parameters on the performance of heat sink under forced convection is then obtained by using these results of main effect plot for mean.

The summary of the influence of the selected independent parameter on the response i.e. Nusselt number for 160 W heat input and for 3 m/s, 6 m/s and 9 m/s velocities are shown in table 3.

The Pareto charts for different velocities and for 160 W heat input is shown in figure 7 (a) to (c). The Pareto charts gives the significant contribution of different parameters on the air side performance of heat sink under forced convection. All the parameters are having the effect on the performance of heat sink. For 3 m/s velocity the significant parameter is number of fins which is having 50.80% contribution ratio. And this trend is same for other heat inputs for this velocity. For 6 m/s velocity the significant parameter is length 38.24% contribution ratio and other parameters are having contribution ratio near to this ratio. For high velocity i.e. 9 m/s the contribution ratio for height of the fin is more on the performance of heat sink which is about 54.90 %. The air velocity now plays a great role on the performance of heat sink. Different heat inputs also have their own effect on the performance of the heat sink. So the individual effects of these parameters on the performance of heat sink are discussed in detail herewith.

The uncertainty of all the variables, measured and derived values are summarized in the table 2. Conventionally, the engineering community accepts uncertainty estimates at 95% confidence [11, 12] The uncertainty for primary measurement parameters are very less whereas for derived parameters i.e. the heat transfer coefficient, thermal resistance and Nusselt number etc are close to 95% so it is acceptable.

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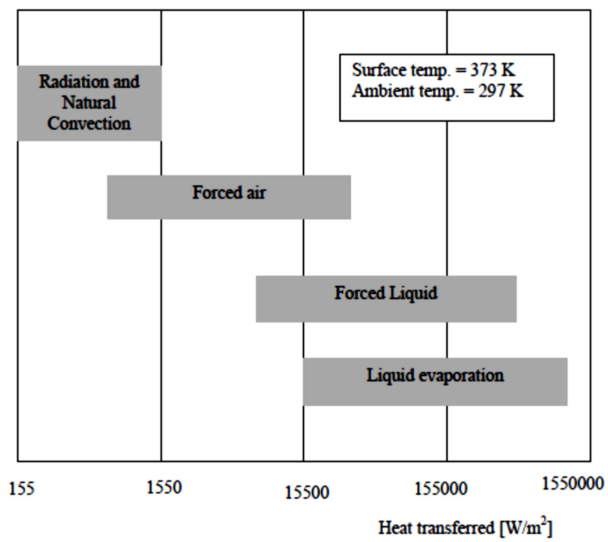
LEGENDS

Figure 1 Range of Conventional Heat Transfer Modes [1]
Figure 2 Radial Heat Sink with a Circular Base and Rectangular Fins
Figure 3 Experimental setup for forced convection
Figure 4 Influences of Length of Fin (L), Height of Fin (H) and Number of Fins (N) on Nusselt Number for 3 m/s Velocity.
Figure 5 Influences of Length of Fin (L), Height of Fin (H) and Number of Fins (N) on Nusselt Number for 6 m/s Velocity.

Figure 6 Influences of Length of Fin (L), Height of Fin (H) and Number of Fins (N) on Nusselt Number for 9 m/s Velocity.
Figure 7 Contribution Ratios of Parameters N, H and N for the Nusselt Number under Forced Convection for 160 W (a) 3 m/s velocity (b) 6 m/s (c) 9 m/s.
Table 1 Heat Sink Parameters
Table 2 Details of Experimentation Matrix for the forced convection
Table. 3 Factorial Effect and Contribution Ratio for Nusselt Number for Forced Convection for 160 W(a) 3 m/s velocity (b) 6 m/s, (c) 9 m/s
Table 4 Uncertainty of Variables



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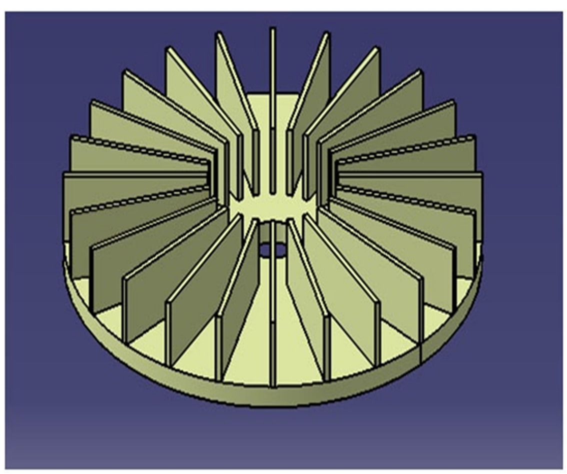


Figure 2 Radial Heat Sink with a Circular Base and Rectangular Fins

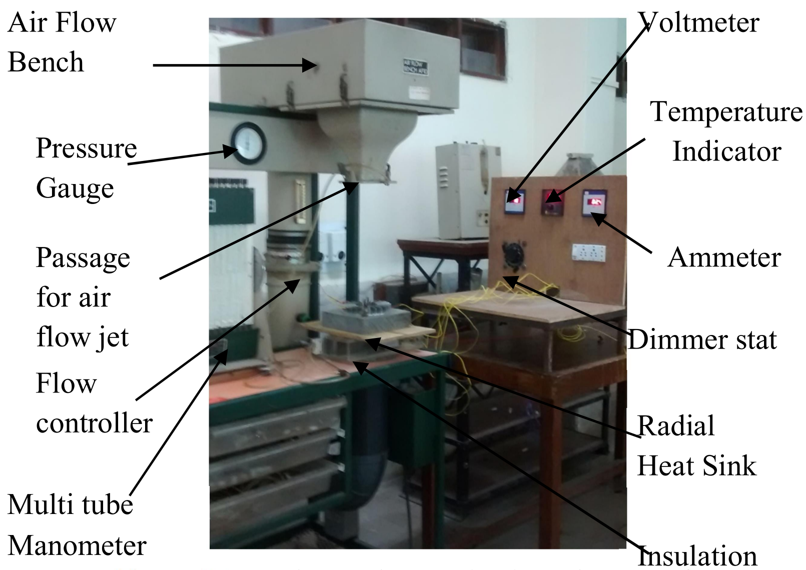
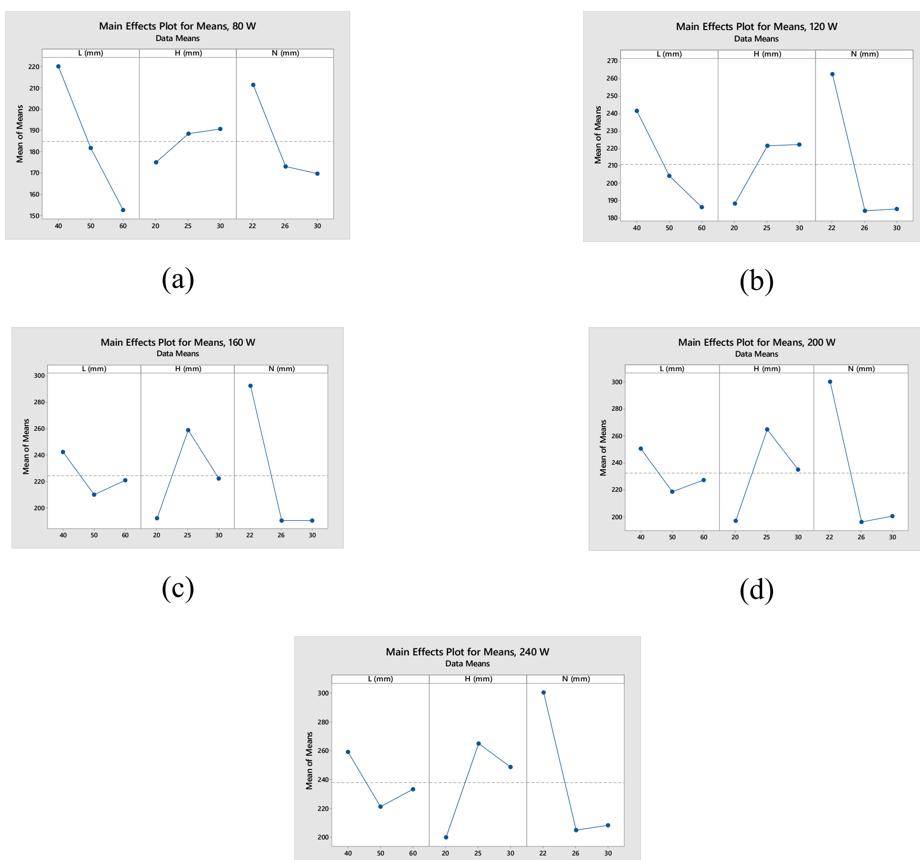
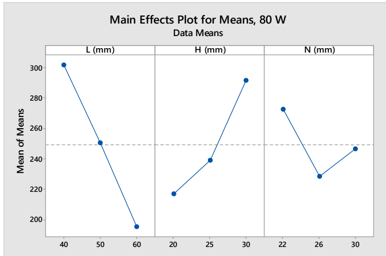


Figure 3 Experimental setup for forced convection

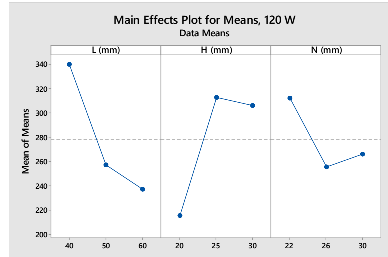


(a) (b) (c) (d) (e)

Figure 4 Influences of Length of Fin (L), Height of Fin (H) and Number of Fins (N) on Nusselt Number for 3 m/s Velocity.



(a)



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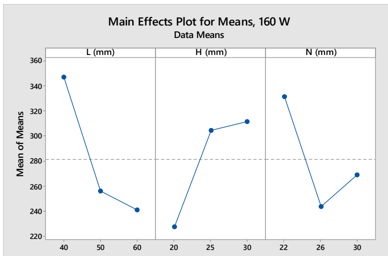
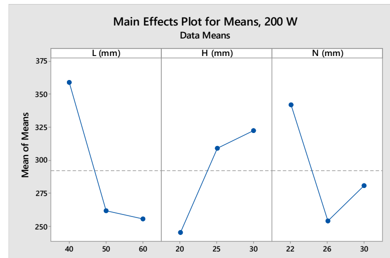


(c)



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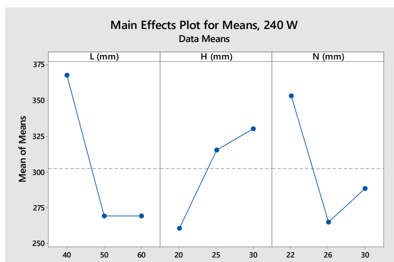
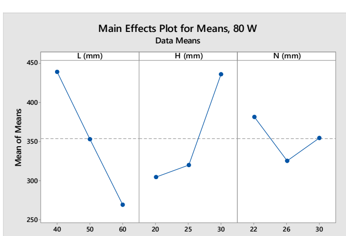
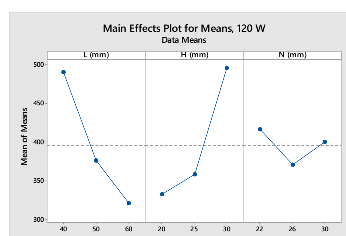


(e)

Figure 5 Influences of Length of Fin (L), Height of Fin (H) and Number of Fins (N) on Nusselt Number for 6 m/s Velocity.



(a)



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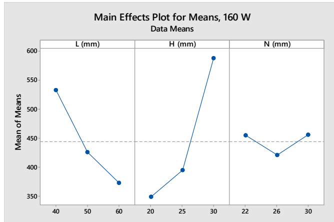

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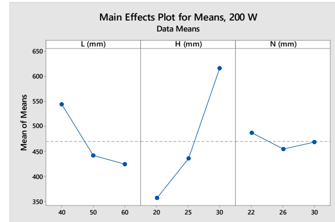

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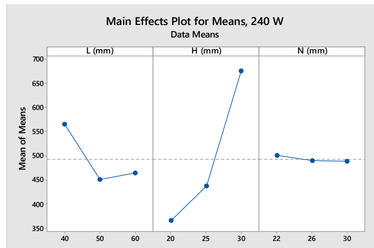
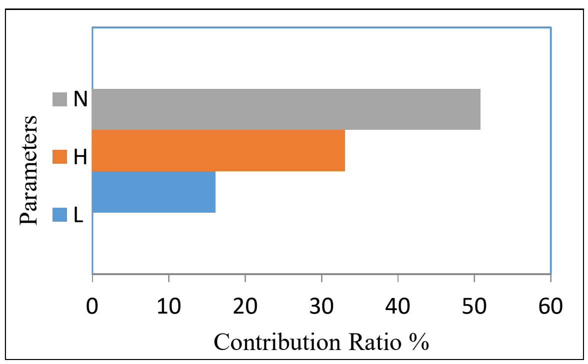


(e)

Figure 6 Influences of Length of Fin (L), Height of Fin (H) and Number of Fins (N) on Nusselt Number for 9 m/s Velocity.



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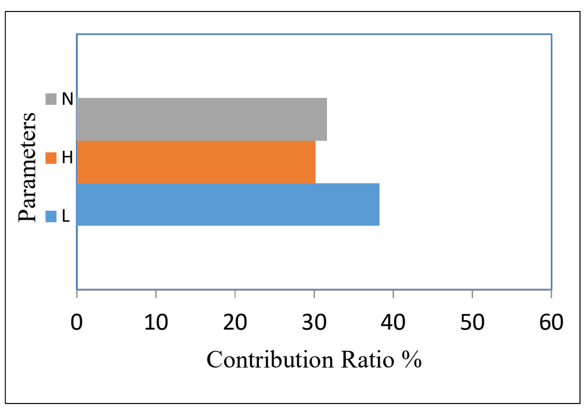

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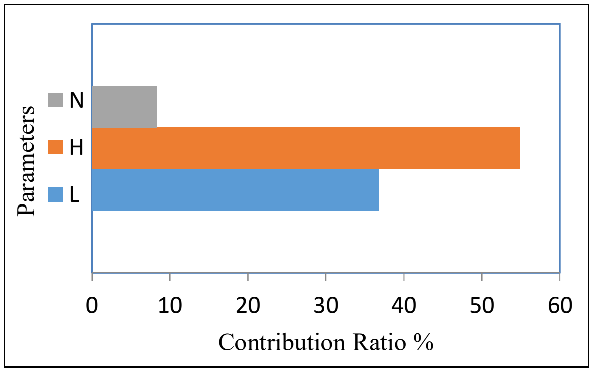

(c)

Figure 7 Contribution Ratios of Parameters N, H and N for the Nusselt Number under Forced Convection for 160 W (a) 3 m/s velocity (b) 6 m/s (c) 9 m/s.

Table 1 Heat Sink Parameters

		Parameter		
		Length of fin	Height of fin	Number of fin
	Code	L	H	N
Levels	1	40 mm	20 mm	22
	2	50 mm	25 mm	26
	3	60 mm	30 mm	30

Table 2 Details of Experimentation Matrix for the forced convection

Sr. No.	Experimentation	Heat Input (W)	Velocity of Air (m/s)
1	Effect of Length of the Fin 40 mm, 50 mm, 60 mm	80, 120, 160, 200, 240	3
2	Effect of Height of the Fin 20 mm, 25 mm, 30 mm	80, 120, 160, 200, 240	6
3	Effect of Number of Fins 22, 26, 30	80, 120, 160, 200, 240	9

Table. 3 Factorial Effect and Contribution Ratio for Nusselt Number for Forced Convection for 160 W(a) 3 m/s velocity (b) 6 m/s, (c) 9 m/s

(a)

Level		L	H	N
Mean	1	242.6	192.5	292.6
	2	210.1	259	190.6
	3	221.1	222.3	190.5
R (max-min)		32.4	66.5	102.1
Rank		2	3	1
Contribution ratio (%)		16.12	33.08	50.80

(b)

Level		L	H	N
Mean	1	346.9	228	331.3
	2	256.2	304.6	243.7
	3	241	311.5	269.1
R (max-min)		105.9	83.4	87.6
Rank		1	3	2
Contribution ratio (%)		38.24	30.12	31.64

(c)

Level		L	H	N
Mean	1	533.2	349.8	455.4
	2	426.5	395.6	420.9
	3	373.5	587.8	456.7
R (max-min)		159.7	238	35.8
Rank		2	1	3
Contribution ratio (%)		36.84	54.90	8.26

Table 4 Uncertainty of Variables

Uncertainty Variable	**Measurement Range**	**Error**
Fin Length	40 – 60 mm	± 2.222 x 10^{-2} %
Fin Height	20 – 30 mm	± 4.00 x 10^{-2} %
Outer diameter	0 – 160 mm	± 0.00625%
Inner diameter	0 – 20 mm	± 0.05%
Thickness	0 – 2 mm	± 0.5%
L/H of the heat sink	1.33 – 3.0	0.03726452%
Temperature	28 – 130 ^{0} C	± 3.2144%
Heat transfer rate	40 – 240 W	2.6925 %
Heat transfer coefficient	13 – 600 W/m^2 K	± 5.833031%
Thermal resistance	0.014 – 0.75 K/W	± 5.8505869%
Nusselt number	7 – 670	± 5. 8197205%

Reynolds number (Jet)	12724 - 42923	± 5%
Reynolds number	3500 - 19500	± 5%