

# Thermal Design of Heat Sink and Optimization of Fin Parameters

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**ABSTRACT:** Radar is an electronic system that is widely used in the field of object detection to determine the range, velocity and angle of objects. Radar systems consist of various components that produce excessive heat during operation. Since usually these components are housed on a thermally conducting material, thermal design and analysis of such systems plays a very important role in ensuring optimal functionality of radar. Air cooled radar transmitter system is considered here. The aim of this paper is to ensure the optimum functioning of a transmitter unit by maintaining the maximum temperature at a specified location on the thermal base plate to within 70°. The environment temperature is considered to be 49° which determines the range of temperature in which the radar system is used. This involves the design of a heat sink that is effective in dissipating the heat that is rejected onto it by the components at this specified location. It is known that around 420Watts of heat is being rejected onto an area of around 71.25cm<sup>2</sup> on the heat sink. The material of the heat sink used is Copper. Commercially available software packages like PTC Creo Parametric 3.0 and Autodesk CFD simulation 2012 are used for modelling and analysis respectively. Analysis is carried out with the operating conditions of air cooled transmitter system given as input. Theoretical validation of the CFD result is carried out and excellent convergence is obtained. MATLAB is used in plotting the various graphs which determine the final fin parameters of the newly designed copper heat sink. Comparisons of the design currently in use in a manufacturing company for the aforementioned scenario versus the new design is presented.

**KEYWORDS:** Triangular fin, Rectangular fin, Efficiency, Effectiveness, fin spacing, Optimum thickness.

## I. INTRODUCTION

For optimum and reliable performance, proper cooling of an electronic system is imperative. This in turn, results in serious design considerations. Popular cooling methods generally employed are water and air cooling. Air cooling is relatively simpler in construction due to ease of availability of coolant and also additional heat exchanger units are not required as in the case of water cooling. Due to the high power usage during the operation of the transmitter device, the current flowing through the components generate heat as a result of resistance offered to its flow. Thus, generally these components are housed onto a base plate made of thermally conducting material like aluminium. At a specific spot, due to extensive heat localization, discontinuity in the base plate is introduced and a copper heat sink is plugged due to higher thermal conductivity of copper when compared to aluminium. Sandhya Mirapalli et al. [1] have carried out various analysis on triangular fins and compared the results with Rectangular fins. These analysis were carried out in two steps .i.e. first the length of the fins were varied keeping the base temperature constant and second the base plate temperature was varied keeping the fin length fixed. R.Mohan et al. [2] have considered the optimal plate fin heat sink design and cylindrical fin heat sink design with variable copper base plate in finding a cooling solution for a desktop which uses an 80 Watt CPU. It was concluded that replacing aluminium with copper as base plate material improved the performance. HoSung Lee [3] provides his insights into the general approach towards Heat Sink Design with derivations of relations between various parameters like Heat flow rate, Thickness of fin, Length of fin, Optimum fin Spacing and Number of fins for both free and forced convection cooling arrangements. M. Thirumaleshwar [4] and R.K.Rajput [5] have provided the practical design considerations on triangular and rectangular fins and the derivations of the expressions like heat flow rate, efficiency and effectiveness for plate fin configurations. Younus A Cengel [6] has provided the expressions for fin parameters of triangular fins which are used to evaluate the performance of the existing design of the heat sink.

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This paper includes the evaluation of the existing design of the heat sink having trapezoidal fin array with thickness of 2.75mm at the base to 1.5mm at the tip. Various parameters like heat flow through the fin, efficiency, effectiveness and number of fins required to dissipate the heat are calculated. Due to space constraints currently only 25 fins are being employed. A new design of the heat sink having rectangular fin array is proposed. Fin parameters of the two designs are compared and tabulated.

## II. METHODS AND PROCEDURE

### A. Existing design evaluation

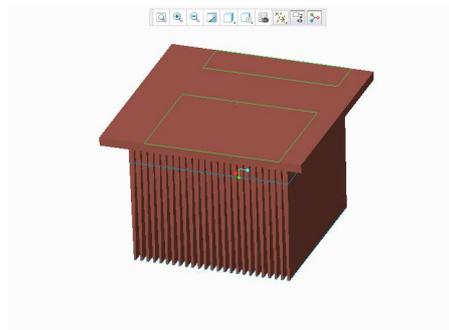


Figure 1. Copper heat sink

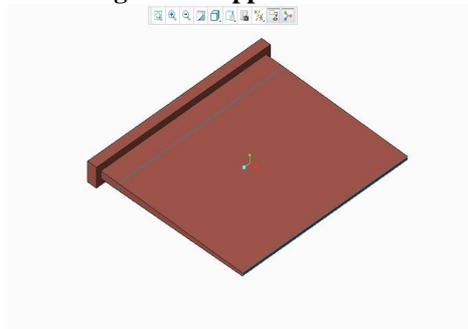


Figure 2. Triangular fin

The above figures show the triangular fin array currently in use. It consists of 25 fins. The heat rejected on the heat sink is known to be 420 Watts on the split line area on the top surface as shown.

#### i. Fin Calculations

The expression for rate of heat transferred through the fin is given by

$$Q_{fin} = 2W\theta_0 \sqrt{hk\delta} \left[ \frac{I_1(2B\sqrt{L})}{I_0(2B\sqrt{L})} \right]$$

Where,

$I_0$  and  $I_1$  are Modified Bessel functions of first and second kind of first order.

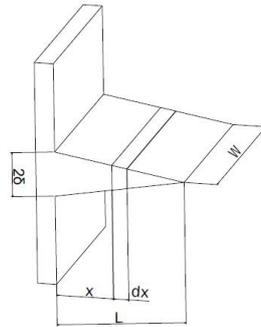
$h$  is the convective heat transfer coefficient.

$k$  is thermal conductivity of fin material. i.e. Copper.

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**Figure 3. Triangular Fin Dimension**

$$\delta = \frac{t_{base} - t_{tip}}{2}$$

$$B = \sqrt{\frac{hL}{k\delta}}$$

The Number of fins required is calculated using the expression

$$N = \frac{Q_{max}}{Q_{fin}}$$

The efficiency of the triangular fin,

$$\eta_{fin} = \frac{2W\theta_0\sqrt{hk\delta} \left[ \frac{I_1(2B\sqrt{L})}{I_0(2B\sqrt{L})} \right]}{2WLh\theta_0}$$

The effectiveness of the fin is given by,

$$\epsilon_{fin} = \frac{2W\theta_0\sqrt{hk\delta} \left[ \frac{I_1(2B\sqrt{L})}{I_0(2B\sqrt{L})} \right]}{hA_b\theta_0}$$

Where

$A_b$  is the base cross sectional area

$$\theta_0 = (T_b - T_\infty)$$

The results obtained for the above relations for the existing dimensions of the fin

Given  $W = 112\text{mm}$  ;  $L = 0.90$  ;  $\delta = 1\text{mm}$  ;  $k_{fin} = 300\text{ W/mK}$  ;

$h = 37.380\text{ W/m}^2\text{K}$  ;  $T_\infty = 49^\circ\text{C}$

For  $T_b = 80^\circ\text{C}$

1.  $Q_{fin} = 15.06\text{W}$

2.  $N \cong 28\text{ fins}$

3.  $\eta_{fin} = 73.12\%$

4.  $\epsilon_{fin} = 35.0968$

For  $T_b = 70^\circ\text{C}$

1.  $Q_{fin} = 10.2014\text{W}$

2.  $N \cong 42\text{ fins}$

3.  $\eta_{fin} = 73.12\%$

4.  $\epsilon_{fin} = 35.0968$

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The existing design consists of 25 fins with 1mm fin spacing which is not sufficient for effective cooling with the current set up of forced air convection at 100CFM.

### B. New design of heat sink

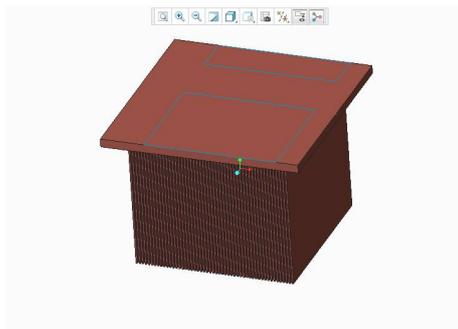


Figure 4. Copper heat sink with rectangular fins

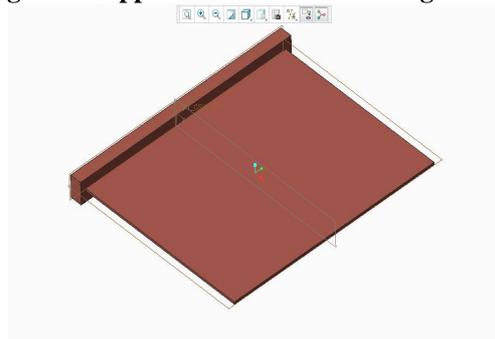


Figure 5. Rectangular fin

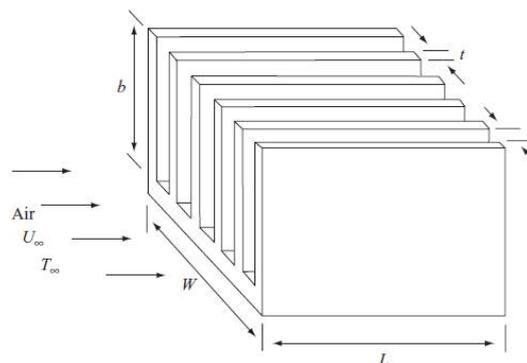


Figure 6. Forced Convection of fin array <sup>[3]</sup>

The following derived relations were obtained from the book Thermal Design by HoSung Lee [3]

1.  $\varepsilon_{opt} = 2.725 \left( \frac{k_{air} \varepsilon_{air} L^2}{\Delta p} \right)^{\frac{1}{4}}$
2.  $\bar{h}_L = \frac{k_{air}}{L} 0.664 Re_L^{1/2} Pr^{1/3}$
3.  $b_{opt} = 1.419 \sqrt{\frac{k_f \varepsilon_{opt}}{2 \bar{h}_L}}$
4.  $Q_{fin\ opt}(total) = \frac{W(2 \bar{h}_L L^2 \varepsilon_{opt} k_f)^{1/2}}{\varepsilon_{opt} + \varepsilon_{opt}} \theta_0 \tanh \beta$

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$$5. \eta_{fins} = \frac{W}{z_{opt} + z_{opt}}$$

*i. Existing conditions of operation*

Width of Cutout for heat sink on Thermal base plate  $W=118\text{mm}$

Length of flow  $L = 112\text{mm}$

Volumetric Flow Rate  $\dot{V} = 100\text{CFM}$

Velocity of Air  $u_{ac}=9.0480\text{ m/s}$

Reynolds Number  $Re_L = 5.04 \times 10^4$  (Laminar Region)

Properties of Air at  $49^\circ\text{C}$

1.  $\rho = 1.093\text{kg/m}^3$
2.  $\mu = 19.61 \times 10^{-6}\text{Ns/m}^2$
3.  $\nu = 17.95 \times 10^{-6}\text{m}^2/\text{s}$
4.  $\alpha = 25.722 \times 10^{-6}\text{m}^2/\text{s}$
5.  $Pr = 0.698$
6.  $c_p = 1005\text{ J/KgK}$
7.  $k_c = 0.02826\text{W/mK}$

Therefore substituting in the above derived relations

$$1. z_{opt} = 1.2\text{mm} \quad 2. \bar{h}_1 = 37.380\text{ W/m}^2\text{K}$$

*ii. MATLAB plots of the derived relations and optimization with heat flux constraint*

For the existing total heat flux constraint of  $420\text{W}$  being ejected onto the heat sink, keeping the flow length  $L = 112\text{mm}$  constant, the resulting optimized values of fin dimensions are derived using the above plots obtained using MATLAB.

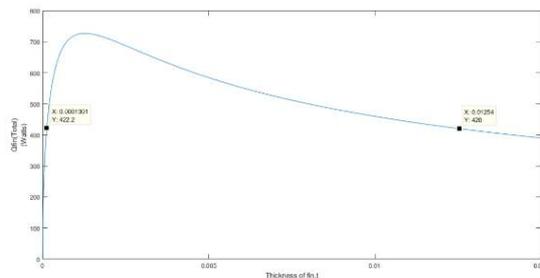


Figure 7. Q(total) vs Optimum thickness of fin

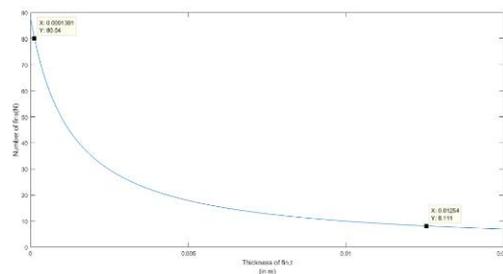
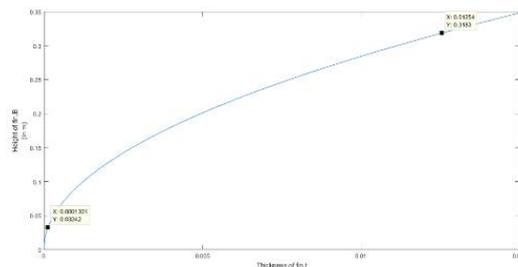


Figure 8. Number of fins vs Optimum thickness

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**Figure 9. Height of fin vs Optimum Thickness**

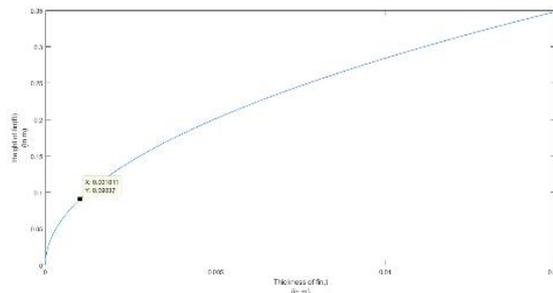
For  $Q = 420W$ ,

$$t_{opt} = 12.54 \cong 13mm; N_{fin} = 8.11 \cong 9; b_{opt} = 318.3mm$$

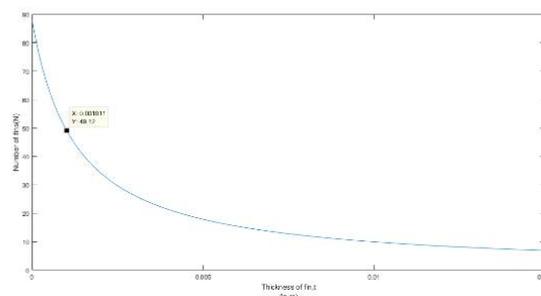
NOTE: The plot consists of 2 data points between which heat transferred by the fin array is more than the required level of 420Watts. The obtained height of the fin ( $b_{opt}$ ) is very large. The height constraint in the design is at a maximum value of 90mm. Thus the above design is to be modified by taking into consideration the height constraint.

### iii. Optimization of fins with constraints using MATLAB plots

Now we want to obtain an optimum heat sink with the constraints which are the profile length  $b = 90mm$  and keeping  $L = 112mm$  constant as before. We seek an optimum thickness  $t$  as a function with the given profile length  $b$  as shown in the plot.



**Figure 10. Height of fin vs Optimum Thickness**

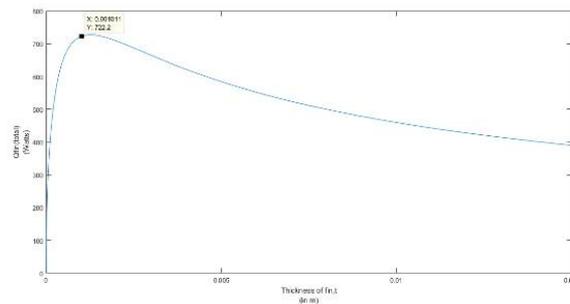


**Figure 11. Number of fins vs Optimum thickness**

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**Figure 12. Q(total) vs Optimum thickness of fin**

For  $b = 90\text{mm}$ , we get  
Optimum thickness of fin,

$$t_{opt} = 1\text{mm}$$

Number of fins required,

$$N_{fin} = 49.12 \cong 50$$

Total heat flux transferred by the obtained fin array,

$$Q_{fin}(total) = 722.2\text{W}$$

Efficiency of the fin,

$$\eta_{fin} = \frac{\tanh(mb)}{mb}$$

$$\eta_{fin} = 62.70\%$$

Effectiveness of fin,

$$\epsilon_{fin} = \sqrt{\left(\frac{k}{h}\right)\left(\frac{P}{A_c}\right) \tanh(mb)}$$

$$\epsilon_{fin} = 113.227$$

### III. RESULTS AND CONCLUSION

The below table gives a comparison of the performance of the two heat sink designs discussed above.

FIN PARAMETERS	EXISTING SINK WITH TRIANGULAR FIN ARRAY	NEW SINK WITH RECTANGULAR FIN ARRAY
Q <sub>total</sub> (Transferred by the fin array)	255.035W - 376.5W while maintaining temperature between 70° - 80° respectively.	722 Watts
Fin spacing (z)	1 mm	1.2 mm
Number of fins ( $N_{fin}$ )	25	50
Fin Efficiency ( $\eta_{fin}$ )	73.12 %	62.50 %
Fin Effectiveness ( $\epsilon_{fin}$ )	35.0968	113.227

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1. It is seen that the rectangular heat sink design dissipates 722 Watts as opposed to the required level of 420 watts in the current configuration.
2. The new heat sink design with rectangular fin array has a spacing of 1.2mm which is the optimum value as calculated and the existing design has a fin spacing of 1mm which is not sufficient for effective cooling.
3. It is seen that the efficiency of rectangular fin is around 10 % less than that of the Triangular fin.
4. It is also observed that the effectiveness of the rectangular fin is more than three times that of the triangular fin.

## IV. THEORETICAL VALIDATION OF NEW DESIGN OF HEAT SINK

The following fin parameters were arrived at during the design of the new copper sink.

1.  $s_{opt} = 1.2mm$  ; 2.  $\bar{h}_2 = 37.380 W/m^2K$  ; 3.  $L = 112mm$  ; 4.  $b = 90mm$  ; 5.  $t_{opt} = 1mm$  ; 6.  $N_{fin} = 49.12 \cong 50$  ;
7.  $Q_{fin}(total) = 722.2W$  ; 8.  $\eta_{fin} = 62.70\%$  ; 9.  $\epsilon_{fin} = 113.227$

We know that the expression for heat transferred by a rectangular fin for the case of end fin being insulated is given by [5],

$$Q_{fin} = \sqrt{hP k_{fin} A_{cs}} \theta_D \tanh mL$$

Where  $m = \sqrt{\frac{hP}{k_{fin} A_{cs}}}$

Therefore substituting the obtained fin parameters in the above equation for dissipation of a total heat flux of 420 Watts, we get the expected base temperature value (the only unknown in the above equation)

$$T_b = 66.57^\circ C$$

Below is a simplified model which was generated to evaluate and compare the performance of the two heat sinks. The actual design consists of fins on the thermal base plate behind the copper sink (bottom side of the red zone in the result below). However in this, these fins are not added in order to reduce the complexity of the model. Hence inferences are made only with respect to heat sink region at the front and the red zone can be neglected.

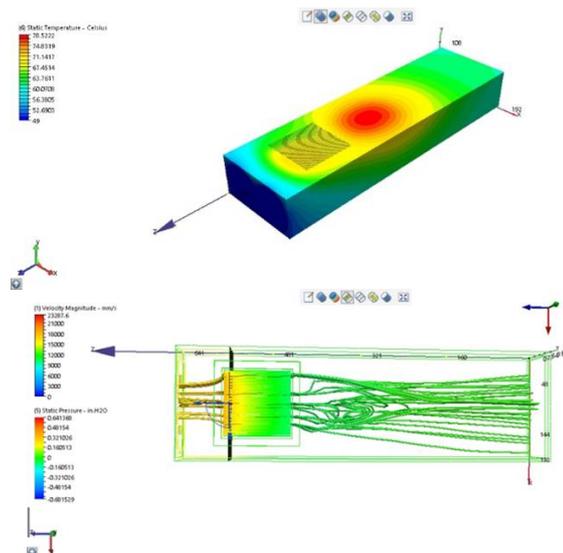


Figure 13. Temperature profile and flow path of air around the new copper sink with rectangular fins

When compared to the above obtained result of around 68° on the heat sink with the new design, the percentage of error that exists between the analytical result and the experimental result is found to be at around 2%. Hence this proves an excellent convergence of the CFD results.



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