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Optimal Design Methodology of Rectangular Heat Sinks for Electronic Cooling under Natural Convective and Radiative Heat Transfer

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Abstract: ***In order to comply with the recent demand for downsizing of the electric equipment, the miniaturization and the improvement in heat transfer performance of a heat sink under natural air-cooling and radiation are increasingly required. In this study steady-state external natural convection and radiation heat transfer for vertically-mounted rectangular heat sinks is investigated numerically and analytically. MATLAB is used for formulating analytical data. CATIA V5 software has been used in order to develop a 3-D numerical model and ANSYS FLUENT software for numerical investigation of designed model. After regenerating, and validating the analytical results for fin spacing, thickness and height a systematic numerical study was conducted on effect of the above configurations. Further study is made by providing interruptions to the vertical rectangular fins. Results show that adding interruptions to vertical rectangular fins enhances the thermal performance of fins. In this study optimum height, thickness and spacing of the rectangular fins for maximum fin performance has been found and correlated.***

Keywords: Convective Heat Transfer, Rectangular fins, ANSYS FLUENT, Fin Parameters, Optimum Fin Spacing, S2S Radiation model

1. Introduction

The rapid advances in micro semiconductor technology have been leading to increase the heat dissipation from microelectronic devices. Passive cooling is a widely preferred method for electronic and power electronic devices since it is a low price, quiet, and trouble free solution. Air-cooling is recognized as an important technique in the thermal design of electronic packages, because its accessibility particularly with respect to safe operation. The use of fins to enhance air-cooling is the simplest and effective heat sink structure under the cost, space and weight constraints. Thus, to develop a systematic air-cooling heat sink design methodology is very important for satisfying the current thermal necessities and successfully removing elevated heat of high-ranking electronic components in the future. Natural convective heat transfer form vertical rectangular fins, and also pin fins is a well-established subject in the literature. It has been investigated analytically, numerically and experimentally. The following paragraphs provide a brief overview on the pertinent literature; the previous studies are grouped into analytical, numerical, and experimental works.

1.1 Analytical Approach

Analytical work in this area was carried out by Elenbaas [1]. He investigated isothermal finned heatsink analytically and experimentally. The analytical study resulted in general relations for natural convective heat transfer from vertical rectangular fins that is not accurate for small values of fin spacing. Churchil [2] developed a general correlation for the heat transfer rate from vertical channels using the theoretical and experimental results obtained by a number of authors. Bar-Cohen and Rohsenow [3] also performed an analytical study to investigate the natural convective heat transfer from two parallel plates. They developed a relationship for Nusselt number in terms of Rayleigh number for isothermal and isoflux plates.

1.2 Numerical Approach

Bodoia and Osterle [4], followed Elenbaas [1] and used a numerical approach to investigate developing flow in the channel and heat transfer between symmetrically heated, isothermal plate in an effort to predict the channel length required to achieve fully developed flow as a function of the channel width and wall temperature. Ofi and Hetherington [5] used a finite element method to study the natural convective heat transfer from open vertical channels

1.3 Experimental Approach

Several experimental works in this area was carried out. Starner and McManus [6], Welling and Wooldridge [7], Chaddock [8], Aihara [9], Leung et al. [10] and Van de Pol and Tierney [11] are some examples, which were mostly focused on the effects of varying fin geometric parameters, the array, and base plate orientation.

Radiation heat transfer plays an important role in the heat transfer from fin arrays. This has been shown by Edwards and Chaddock [12], Chaddock [13], Sparrow and Acharya [14], Saikhedkar and Sukhatme [15]. It has been reported that the radiation heat transfer contributes between 25–40% of the total heat transfer from fin arrays in naturally cooled heatsinks.

In this paper, the aim is the investigation of the optimal fin height, thickness and spacing for the highest natural convective and radiative heat transfer. Further investigation is made by using interrupted rectangular fins. A proper selection of fin spacing and interruption sizes can lead to higher thermal performance. This is a direct result of thermal boundary layer interruption and formation of new boundary layers at the interrupted fin sections. Furthermore, fin interruption leads to significant weight and thus cost reductions in heatsinks. The goal of this study is to investigate the effects of adding interruption to fins and determine an optimum value for different geometrical parameters of the fin array, mainly the height, spacing and thickness of fin.

2. Problem Statement

When a heatsink is heated, thermal boundary layers start to develop at the bottom edges of the opposing surfaces of the neighbouring fins; the boundary layers eventually merge if the fins/channels are sufficiently long creating fully developed channel flow. Interrupted fins disrupt the thermal boundary layer growth, maintaining a thermally developing flow regime, which in turn leads to a higher natural heat transfer coefficient.

A schematic of the considered fin geometry along with the salient geometric parameters is shown in Fig. 1 (a) and (b). A cubical electronic housing (100cm X 100 cm X 100 cm) is taken with a centerline heat load generating 50 W of heat. Fins are to be used on the two side surfaces as shown in Fig. 2. The ambient temperature (Ta) taken is 328 K.

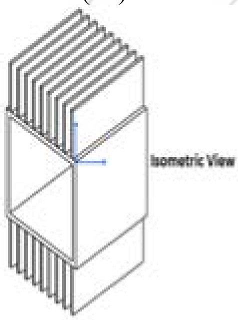

Figure 1 (a)

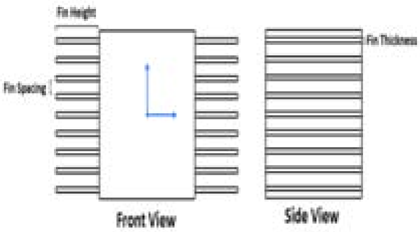

Figure 1 (b)

Figure 1: Geometrical Parameter of the Problem statement

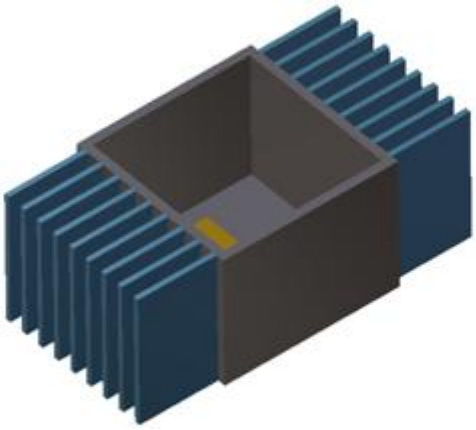

Figure 2: CAD model showing the problem statement

An analytical approach in MATLAB is made to find the surface temperature (Ts) of the above cubical housing without using fins. Then again the analytical approach is used to find the minimum surface temperature using fins by varying the height (z), spacing (ω) and thickness (t) of the fins respectively. This solution is then analyzed in ANSYS FLUENT. Further investigations are made by using interrupted fin by slitting the fin length by 5mm from the center and again repeating the above procedure.

3. Analytical Solution

An algorithm is written in MATLAB. The objective of this algorithm is to know the surface temperature $T_s$for a given heat sink and working conditions.

3.1 Nomenclature

The nomenclature of various parameters used is given in Table 1.

Table 1: Nomenclature of various parameters

Name	*Details*
h_v	Vertical side plate convection coefficient, W/m^{2}K
h_b	Bottom plate convection coefficient, W/m^{2}K
H	Fin efficiency
A	Surface Area of the plate, m^2
A_f	Surface Area of the fin, m^2
A_{wf}	Surface Area of the plate on the fin side not covering fin, m^2
P	Fin parameter, m
N	Number of fins
K	Thermal conductivity of the fin material, for aluminium 202 W/mK
T	Fin thickness, m
Z	Fin height, m
Ω	Fin spacing, m
L	fin length, m
σ	Stefan-Boltzmann constant = $5.67x10^{-8}$ W/m^{2}K^{4}
ε	emissivity = 0.9 for black anodized Aluminium
f	view factor, here it can be taken as 1 since the radiations from the plate are transmitted directly to the surroundings
R	Reduction in radiation heat transfer due to shielding by adjacent fins. For surfaces without fins R=1, the graph of the reduction in radiation corresponding to different fin diameters is given in Fig. 3.
T_a	Ambient Temperature, K
T_s	Surface Temperature, K

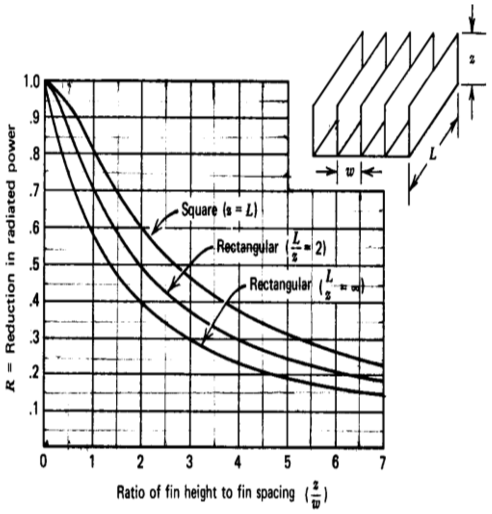

Figure 3: Radiation shielding effect of adjacent surfaces

3.2 Correlations

3.2.1 Convection coefficient

(a) For vertical side surfaces

For vertical side surfaces the natural convention coefficient is given by:

$$h_v = 1.42\left[\frac{\Delta T}{l_c}\right]^{0.25} \quad (1)$$

Where ΔT= T_s-T_a and 'l_c' is the characteristic length of the side vertical surface

Finned surfaces are provided on the two vertical side plates to augment the heat transfer as shown in Fig. 2. The natural convection coefficient must be corrected taking into account the fin efficiency. The fin efficiency is given by:

$$\eta = \frac{\tan(mz)}{mz} \quad (2)$$

Where m can be calculated using the following:

$$m = \sqrt{\frac{h_v P}{k A_f}} \quad (3)$$

Therefore, the effective heat transfer through the finned surface = ηh_v

(b) For bottom surface

For horizontal surfaces facing downwards the natural convention coefficientis given by:

$$h_b = 0.59\left[\frac{\Delta T}{l_o}\right]^{0.25} \quad (4)$$

Where ΔT= T_s-T_a and 'l_b' is the characteristic length of the bottom surface.

3.2.2 Calculation of Radiation Heat Transfer coefficients

Radiation heat transfer coefficient is given by:

For finned surface:

$$h_{rad,f} = R\sigma f\varepsilon(T_s + T_a)(T_s^2 + T_a^2) \quad (5)$$

The values of R can be calculated from the graph in Fig. 3.

For surfaces without fins:

Since R=1 for surfaces without fins

$$h_{rad,wf} = \sigma f\varepsilon(T_s + T_a)(T_s^2 + T_a^2) \quad (6)$$

3.2.3 Calculation of Total Heat Transfer

The total heat transfer is due to natural convection and radiation from the finned surface and surfaces without fins.

$$q_{total} = q_{con} + q_{rad} \quad (7)$$

$$q_{con} = \left[(2h_v + h_b)A + 2\eta h_v Pz + 2h_v A_{wf}\right](T_s - T_a) \quad (8)$$

$$q_{rad} = \left[2h_{rad,f}Pz + h_{rad,wf}\left(2A_{wf} + 3A\right)\right](T_s - T_a) \quad (9)$$

3.3 Methodology for the solution

An algorithm is used to calculate the surface temperature using iteration in MATLAB. Considering the first value of surface temperature T_s=T_a+1. Then iterating such that the value of q_{total}=50∓0.1 W in equation (7).

3.3.1 Methodology for no fins

First the value of T_s is calculated for the case when no heat sink is used.

The equations for this case becomes:

$$q_{con} = [(4h_v + h_b)A](T_s - T_a) \quad (10)$$

$$q_{rad} = \left[h_{rad,wf}(5A)\right](T_s - T_a) \quad (11)$$

The value of $T_s$comes out to be **391.24 K**

3.3.2 Methodology when continuous fins are used

Now considering heat sinks. Taking different values of height, spacing and thickness, and again calculating surface temperatures. The various values of Ts for different values of fin parameters are given in Table 2.

Table 2: Variation of surface temperature (T_s) for different fin parameters

z(mm)	*t(mm)*	*ω(mm)*	*T_s(K)*
15	3	6	368.97
15	3	8	363.68
15	3	10	365.37
15	4	6	365.81
15	4	8	364.11
15	4	10	366.30
15	5	6	368.11
15	5	8	366.68
15	5	10	367.35
30	3	6	358.18
30	3	8	356.76
30	3	10	366.18
30	4	6	359.22
30	4	8	357.92
30	4	10	359.98
30	5	6	361.82
30	5	8	361.33
30	5	10	364.05
45	3	6	370.70
45	3	8	354.75
45	3	10	357.54
45	4	6	373.05
45	4	8	354.84
45	4	10	357.15
45	5	6	373.76
45	5	8	358.10
45	5	10	359.24

Thus it can be seen from Table 2 that at fin thickness 3mm, fin height 45mm and fin spacing 8mm surface temperature attains its minimum value i.e. **354.75 K.** These variations are plotted on graphs in Fig. 4.

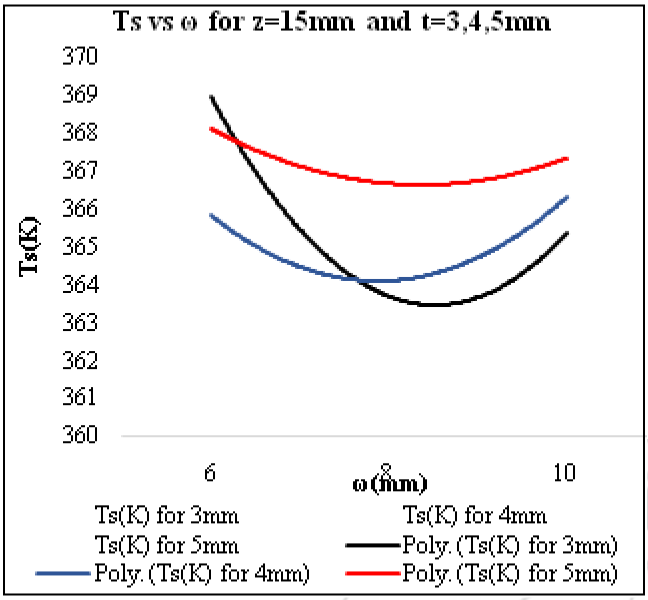

Figure 4(a)

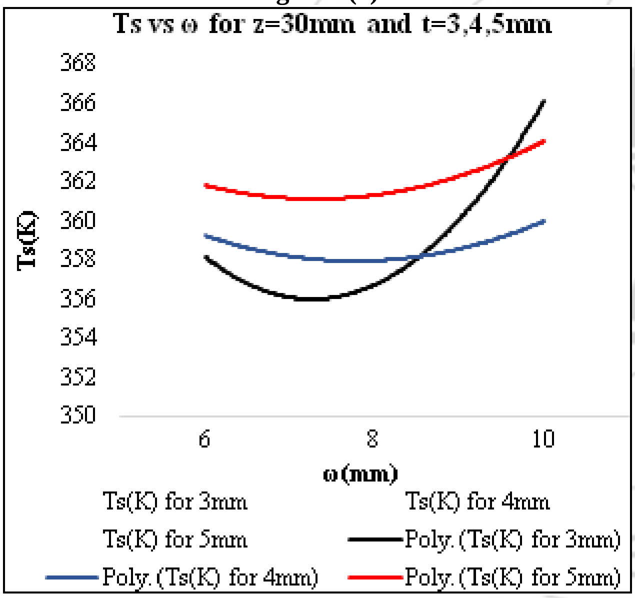

Figure 4(b)

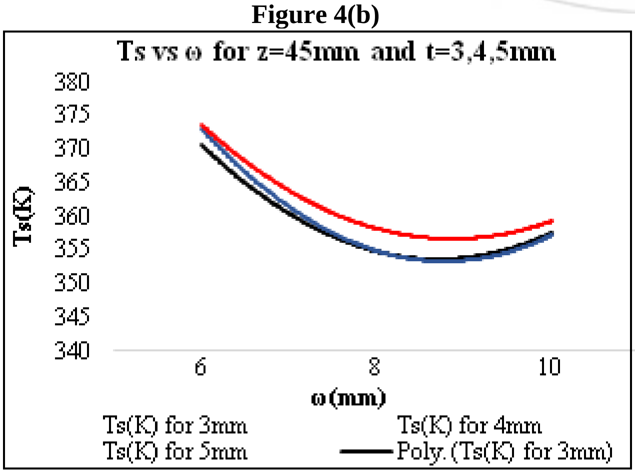

Figure 4(c)

Figure 4: Variation of T_s with the fin parameters

3.3.3 Methodology for Interrupted Fins

Again the same algorithm is used for calculating the surface temperature required to dissipate the heat generated due to the electronic load for interrupted fins. The interruption is made by slitting the fins by 5mm from the center as shown in Fig. 11. The model used for slitting was the same model which gave the minimum temperature for continuous fins i.e. z=45mm, t=3mm, ω=8mm.

The value of the surface temperature for this model came out to be **344.61 K.** The reason behind lower value of the surface temperature in this case is due to the fact that interrupted fins disrupt the thermal boundary layer growth and new thermal boundaries grow, maintaining a thermally developing flow regime, which in turn leads to a higher natural heat transfer coefficient.

4. Numerical Solution

A steady state 3-D model was developed in CATIA and analyzed in ANSYS FLUENT in order to investigate the correctness of the analytical solution. For the domain dimension selection the optimum fin spacing, *ω*, fin height, z, and fin thickness, *t*, is estimated using above analytical solution. The effect of fin parameters was thus decoupled in simulation of the natural convection and radiation model. Surface to Surface (S2S) Model of radiation is used in the analysis. The pressure inlet boundary condition is applied to the channel inlet (the bottom of the channel) which defines the static/gauge pressure at the inlet boundary. This is interpreted as the static pressure of the environment from which the flow enters. For the top of the domain, i.e. the outlet of the channel, outlet boundary condition is applied in which the flow direction should be defined. Gravity equal to 9.81 m/ s^2 has been taken into account. For solving the system of partial differential equations ANSYS-Fluent 14.5.0 have been employed. Using the above methodology three models have been analyzed which are:

1) Electronic box without fins
2) Electronic box with continuous fins having the optimum fin parameters as calculated in analytical solution
3) Electronic box with interrupted heat sinks having the slitting of 5mm through the center of each fin.

4.1 Electronic box without fins

The above methodology was used to check the analytical solution through numerical solutions. Fig.5 shows the CAD model generated in CATIA. Fig. 6 shows the schematic of the domain considered for numerical analysis. Fig. 7 shows the simulation results.

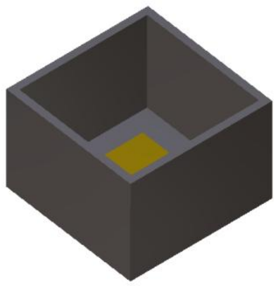

Figure 5: CAD model of the electronic box without fins

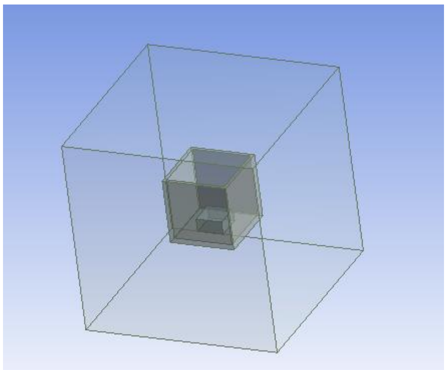

Figure 6: Domain considered in ANSYS FLUENT

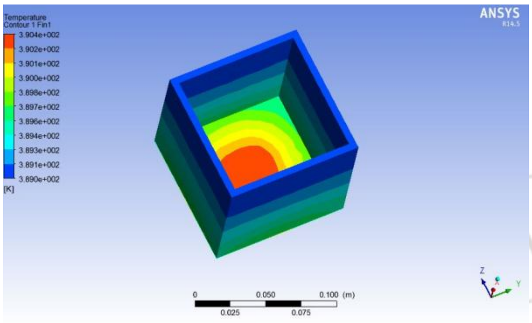

Figure 7(a)

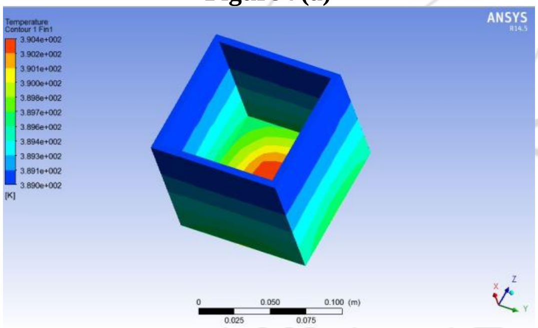

Figure 7(b)
Figure 7: Simulation results

The simulation results show that the average surface temperature is nearly around 390 K, which justifies the analytical result.

4.2 Electronic box with continuous fins having the optimum fin parameters as calculated in analytical solution

Fig.8 shows the CAD model generated in CATIA. Fig. 9 shows the schematic of the domain considered for numerical analysis. Fig. 10 shows the simulation results.

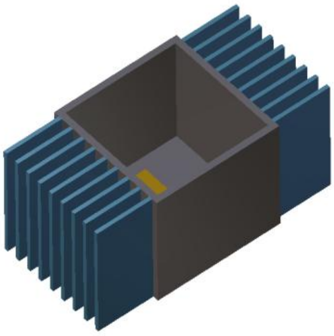

Figure 8: CAD model of the electronic box with continuous fins

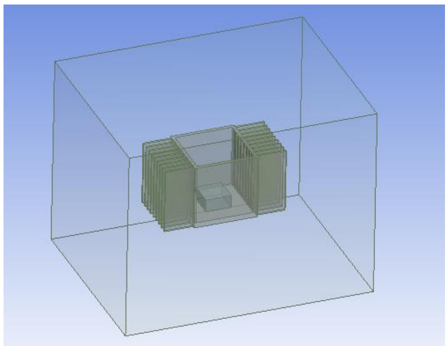

Figure 9: Domain considered in ANSYS FLUENT

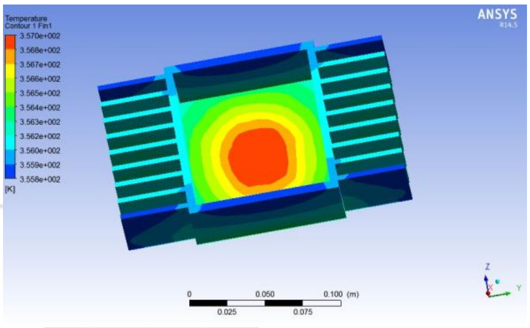

Figure 10(a)

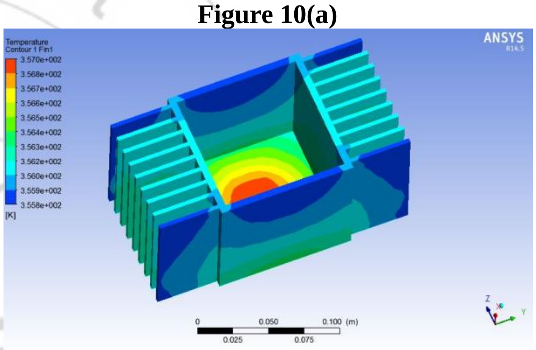

Figure 10(b)
Figure 10: Simulation results

The simulation results show that the average surface temperature is nearly around 356 K, which justifies the analytical result.

4.3 Electronic box with interrupted heat sinks having the slitting of 5mm through the center of each fin

Fig.11 shows the CAD model generated in CATIA. Fig. 12 shows the schematic of the domain considered for numerical analysis. Fig. 13 shows the simulation results.

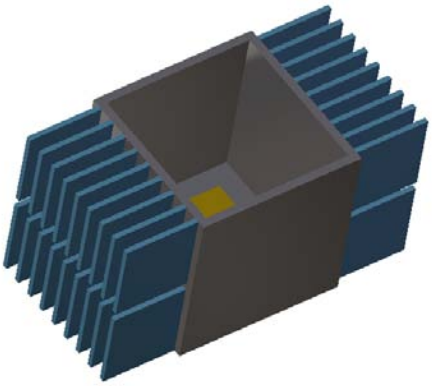

Figure 11: CAD model of the electronic box with interrupted fins

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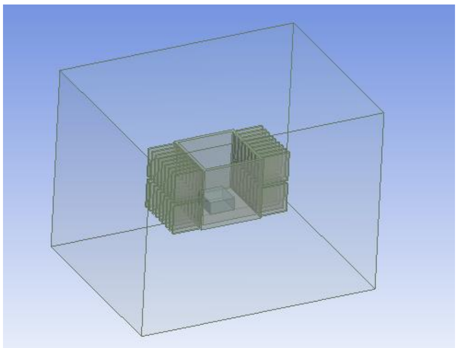

Figure 12: Domain considered in ANSYS FLUENT

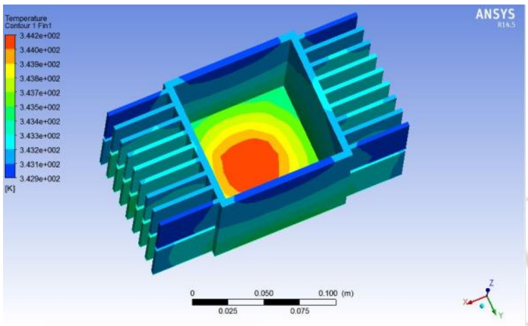

Figure 10(a)

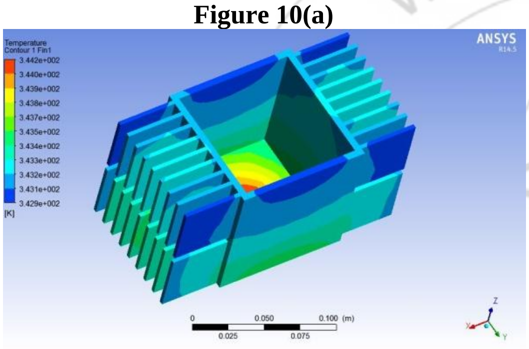

Figure 10(b)
Figure 10: Simulation results

The simulation results show that the average surface temperature is nearly around 344 K, which justifies the analytical result.

5. Results and Discussions

The effects of important parameters on the steady state natural convection and radiation heat transfer from continuous and interrupted rectangular fins are investigated. The effects of fin spacing, ω, fin height, z for continuous fins on the average wall temperature are discussed in detail in the following subsections.

In order to verify the numerical results, the averaged surface temperature was measured for the given problem using the optimum model design. The average surface temperature was also calculated using the existing relationships for natural convection and radiation heat transfer using analytical solution.

5.1 Continuous finned Heat sinks

In the study, different continuous finned parameters were investigated and the results were compared to the analytical solution. Adding the effect of radiative heat transfer to the existing analytical solutions, the optimum value of fin spacing, fin height and fin thickness for continuous rectangular fins were calculated. Surface temperature for different values of the fin parameters are jotted down in Table 1. and plotted in Fig. 4. As it was expected, surface wall temperature depends on fin height, fin spacing and fin thickness; it first decreases with increase in fin spacing up to a minimum value for a particular fin height and fin thickness and then again increases.

5.2 Effects of Interrupted Fins

It can be seen from the analytical as well as numerical solutions that due to interruptions in the fin there is a decrease in the surface temperature by an amount of nearly **10 K** for the same optimum solution as obtained for the continuous finned heat sinks. This is due to fact that the interruptions in the fins resets the hydrodynamic and thermal boundary layers and new boundary layers grow, thus increasing the overall convective heat transfer coefficient.

6. Conclusion

Numerical and analytical approach were employed in order to find the optimized geometrical fin parameters. Analytical results were verified by numerical analysis, and optimum fin spacing was calculated and compared to existing data in literature. A great consistency was observed. In a novel approach, interruptions were added to continuous fins with initial idea of resetting the hydrodynamic and thermal boundary layer in order to decrease thermal resistance. Analytical and numerical results show a decrease in the surface temperature for the same amount of heat transfer from the heatsink.

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