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RESEARCH ARTICLE

STEADY STATE THERMAL ANALYSIS OF PERFORATED HONEYCOMB PLATE FIN HEAT SINKS USING ANSYS®

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Abstract

Heat sink or heat exchanger is a passive cooling device used in electronic components to prolong their longevity, performance and reliability. All electronic components utilize current for the operational purposes and thus become prone to sharp increase in the temperature. The generated heat above the operating level becomes critical in-terms of failure component and hence, appropriate thermal management demands come into act. Finned or extended surface heat sinks are used to cool power electronic devices and components. The comparative results of plate-fin forms on the thermal performance of the heat-sink with 'inline' arrangement is analyzed in this paper. Four forms of fins: 'Rectangular', 'One-side tapered', 'Inverted T section' and 'I section' with and without honeycomb perforations are designed on SOLIDWORKS® and analysed using ANSYS® software to identify a cooling solution for a CPU in terms of temperature and directional heat flux along x, y and z directions. The aluminium alloy 6063-T6 and natural graphite are selected as a base plate and fin materials respectively. The main objective of this paper is to contribute to this improving area of research by studying the effect of honeycomb perforations of plate fin heat sinks under natural convection using steady state thermal analysis at a constant heat flow of 20W and 40W in two different cases with air inlet temperature taken as 37.85° C. A total of 16 specimen were analysed. 8 specimen of plate fin heat sinks without perforations were compared with the rest 8 specimen with honeycomb perforations. It was evaluated from both the cases that inverted T sectional fin with perforations provided an improvement in thermal efficiency and better heat flux removal results among other plate fin profiles where as in terms of density and cost trade-offs one side perforated tapered fin outperformed other fins.

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Nomenclature:-

L_W = length of the base plate of the heat sink (mm)

W_W = width of the base plate of the heat sink (mm)

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t_w= thickness of the base plate of the heat sink, (mm)

L = length of the fin (mm)

W= width of the fin (mm)

t = thickness of the fin (mm)

hc= height of honeycomb core (mm)

θ = angle of inclination of honeycomb core (mm)

tc = thickness of the core (mm)

a = width of the core(mm)

$\dot{q}_x$ = rate of heat transfer through the base plate of the heat sink (W)

$\frac{d\dot{T}}{d\dot{x}}$= Temperature gradient between the surrounding temperature and the heat sink temperature, (°C)

$\dot{A}$= area through which the heat is transferred (mm^2)

$\dot{T}_{(x)}$= local temperature (°C)

$\dot{X}$= distance in the direction of the heat flow (mm)

$\dot{\lambda}$= thermal conductivity (W/mm-K)

$\dot{A}_C$= cross sectional area which may vary with $\dot{X}$ (mm^2)

$\dot{q}_C$= rate of heat transfer by convection (W)

$\bar{h}_c$= convective heat transfer coefficient (W/mm-K)

$\dot{A}_s$= convective heat transfer area/surface area of the differential element (mm^2)

ΔT= difference between the surface temperature ($\dot{T}_s$) and a temperature of the air $\dot{T}_\infty$ (°C)

$\dot{T}_s$= surface temperature (°C)

q_{x+dx}= rate of heat flow by conduction out of element x + dx, (W)

$dq_{convection}$= rate of heat transfer by convection from surface between x and x + dx (W)

θ = excess temperature (above atmospheric temperature) (°C)

$\dot{q}_{x+dx}$ = rate of heat flow by conduction out of element x + dx

P = perimeter of the fin surface (mm)

P $d\dot{x}$= fin surface area between $\dot{X}$ and $\dot{X} + d\dot{X}$ (mm^2)

C_1, C_2 = constants (dimensionless)

$\dot{q}_f$ = fin heat transfer rate (W)

$\dot{q}_b$ = base heat transfer rate (W)

N= number of fins

H = height of fins (mm)

n = number of perforations

d = diameter of perforation (mm)

U = dimensionless velocity component along X direction

V = dimensionless velocity component along y direction

W = dimensionless velocity component along z direction

G = acceleration due to gravity (m/s^2)

ρ = density (kg/m^3)

X = dimensionless distance along 'X' coordinate

Y = dimensionless distance along 'Y' coordinate

Z = dimensionless distance along 'Z' coordinate

Introduction:-

With today's rapid advancement in the field of science and information technology the operating capability and consumption power especially in the electromechanical devices like CPU have been increasing owing the major trend of the industry's prowess and pining to manufacture the miniature products.This mandates reduced dimensions, augmented thermo-physical properties, lighter weight characteristics, robust-mechanical properties, lower hardware cost, processing of data at a faster pace with greater reliability (1-3). Inevitably the heat fluxes of these electronic components evidently spike due the demands of powerful and scaled-down products. This necessities the dissipation of high heat flux at the chip level. Increases in power levels and densities combined with every new packaging design leads to thermal challenges that, if left un-engineered, can significantly shorten the reliability (failure of critical components) and operating performance (user-device interaction discomfort) of microelectronic components. Therefore, definite thermal steps are mandated for improving their thermal life (3-5, 16). To remove heat efficiently from the high-density electronic component, i.e., this trend has necessitated stringent packaging requirements is a promising option to tackle this issue. However, a serious issue in electronic packaging is the thermal management (6-9). This outlays the importance of thermal probe for the semi-conductor industry, however, the cooling of these devices has become increasingly challenging (4-5). The concerning satisfaction for the critical control over the junction temperature requirements improvements in cooling technologies. Various cooling techniques, e.g., 'phase-change' cooling, 'thermoelectric' cooling, 'liquid cooling', 'air and impinging jet' cooling, have been settled to resolve the issues (10-13). Each of these technologies has its benefits and drawbacks, and the challenge is to search out their optimal performance for a given electronics cooling application (14-19). This is achieved by a selected efficient heat sink with the extended surfaces known as fins attached to its base plate. In electronic systems, a ‘heat-sink’ (figure1) is a passive component that cools a device by dissipating heat into the encircling. Heat- sinks are accustomed for cooling such as high-powered semiconductor devices, and optoelectronic devices like higher-power lasers and light-emitting diodes (LEDs) (20-21).By functional means ‘fins’ provide the substantial (total heat transfer) surface area with less excessive (primary surface) area thus, acting as the turbulence booster for additional augmentation of thermal transfer rates, however, for commercial means the usage is based for maintaining the processor temperatures below the designed critical values, keeping in view not to overdesign the cooling systems and unnecessarily complicating the control hardware. So, therefore, the thermal design has become an important contribution in improving the thermal life of an equipment (16, 22-24).

Active-cooling is ‘usually’ performed mechanically by pushing the coolant through the electronic component passages to exhaust away the heat, whereas passive cooling is performed by the aid of natural convection and radiation for the heat transfer to ambient. The efficacy of passive cooling is typically improved by the usage of heat-sinks.Hence, research interest is trending in passive thermal management, that is, a cooling system which requires no any mechanical means for maintaining the temperature and thermal profile within the ‘specified range’ (25-26).The challenges of space utilization, effect of atmospheric constraints, cost have been a difficult task to maintain and as such new innovative heat sink designs, usage of advanced materials and fabrications can lead to better performance of the heat sinks (6, 27-29). For this purpose, in the literature, various types of heat sinks such as ‘Rectangular’, ‘Triangular’, ‘Hexagonal’, Tapered’ etc. have been designed to investigate various cooling techniques such as ‘cold-water’, ‘Heat-pipes’, ‘Semi-conductor’ and ‘Liquid-nitrogen’ from time to time in different research work to explore the heat transfer mechanisms in the most effective way.Every combination of shape, size and material for the specific application area has their own adaptability and effectiveness and thereby designating the foremost workable combination which gives the best efficacy and reliability (30-33). Despite already a lot of research work on the individual cooling technologies has been already done but a few researches are available on comparing the available technologies for their felicity in respective effective necessity required in any respective application. This insufficiency lag is realistically to be related to the difficult assignments of comparing the above mentioned technologies, due to their interactive relationships with the respective design attributes.Yet, a considerable researchers have utmost compared these technologies (34-38). Air cooling with the help of heat sinks is the most cost effective and definite means of electronics cooling. Natural convection air cooling is definitely favorable for low power dissipating of heat sinks since, it offers an energy-free, low cost and noise free operation.In order to improve the thermal performance in heat sinks by the aid of natural convection, one recent method seen is by ‘adding perforations’, this augments the ‘heat transfer area’ and overall ‘heat transfer performance’ which ultimately increase the heat dissipation and the overall rate of ‘heat transfer’ (39-43).An outsized range of studies are already conducted on varied forms of fin modifications by cutting some material from fins to form holes, cavities, slots, grooves or channels through the fin body to extend the heat transfer area and/or the heat transfer coefficient. Bassam and Abu (44-45) conducted the numerical analysis of pervious and the solid fins and concluded that the former fins outperformed the latter fins.Plate-fin heat sink is considered as one of the generally approved heat sinks for its simple-geometry and low-cost (36). Besides the widely used plate-fin heat sinks, various fin modifications have been proposed to improve the natural convective ‘heat transfer’, such as plate-fins with non-rectangular cross-sections (46), sloped plate fins (47), radial plate-fin arrays (13-14, 48-51), perforated pin-fins (52), and metal foams (51-55). The new computational and experimental researches of perforating fins have conjointly shown the vital advantages over non-perforating fins.Shaeri&Yaghoubi (56-57) and Shaeri& Jen (58-59), conducted a study on perforated plate fin variables (number and size of perforation), based on their study they concluded that thermal performance incremented by 80% over the non-perforated fin, while FarhadIsmail et al. (60-61) studied the effect on air in laminar and turbulent flows due to the ‘perforation shape’. It was concluded that by adding more perforation, increase in heat transfer was obtained with decrement in the frictional drag.Dhanawade and Dhanawade (62) conducted an experimentally study to determine the relationship between perforation size and that of heat fluxes. It was concluded that larger size perforations were useful for low heat fluxes and smaller perforations higher for prime heat fluxes.

As extended surface technology continues to grow, new design ideas come ‘back-forth’, together with fins manufactured of anisotropic composites, porous-media as well as perforated and interrupted plates (Bayram and Alparslan, 2008) (63). Due to the requirement for light-weight, small and cost-effective fins, the optimization of fin size is important. Accordingly, fins should be designed to achieve maximum heat removal with minimum material expenditure, thereby facilitating manufacture of the fin (Al-Essa and Al-Hussien, 2004) 64-65). Many studies have investigated optimizing fin-shapes. Other studies have introduced form changes by screwing some material from the fins to form cavities, holes, etc. through the fin body to improvethe heat-transfer area and/or the 'heat-transfer coefficient' (30, 50-52).It can be concluded that a variety of analytical, numerical and experimental research work has been already carried out individually for heat transfer augmentation using honeycomb perforations using different types of heat-sinks. However, numerical and analytical simulation comparison of plate fin heat sinks with diverse fin profiles with and without honeycomb perforations has not been done yet. (66-70). In this research work passive heat sinks (71) to cool CPU of desktop computers are investigated under natural convection. This paper presents a steady-state numerical simulation of the heat transfer performance under natural convection for four different fin profiles of heat sinks with and without honeycomb perforations using Ansys®. The study not only focuses on the comparative analysis but also the miniaturization is kept in mind in terms of weight, cost and reliability due to no usage of any external pumping power and no maintenance requirement. A comparison between the solid fins is done in the first case and later the same comparison is done on same diverse fin profiles with honeycomb perforations under two different heat flow conditions keeping heat transfer coefficient and the dimensions same. In overall this research work is a proof between the cost and material (72-74) trade off in-addition to the heat transfer augmentation in perforated fins as compared to the solid fins. The main objective of this work is to study the temperature variations and the effect of heat flux along x, y and z directions inside the fins made in four profiles (rectangular, I sectional, one side tapered. Inverted T) fins with and without perforations in order to enhance the heat dissipation rate under natural convection. The main aim of this research work is to extend the heat transfer properties by shifting geometry, material and design of fins.

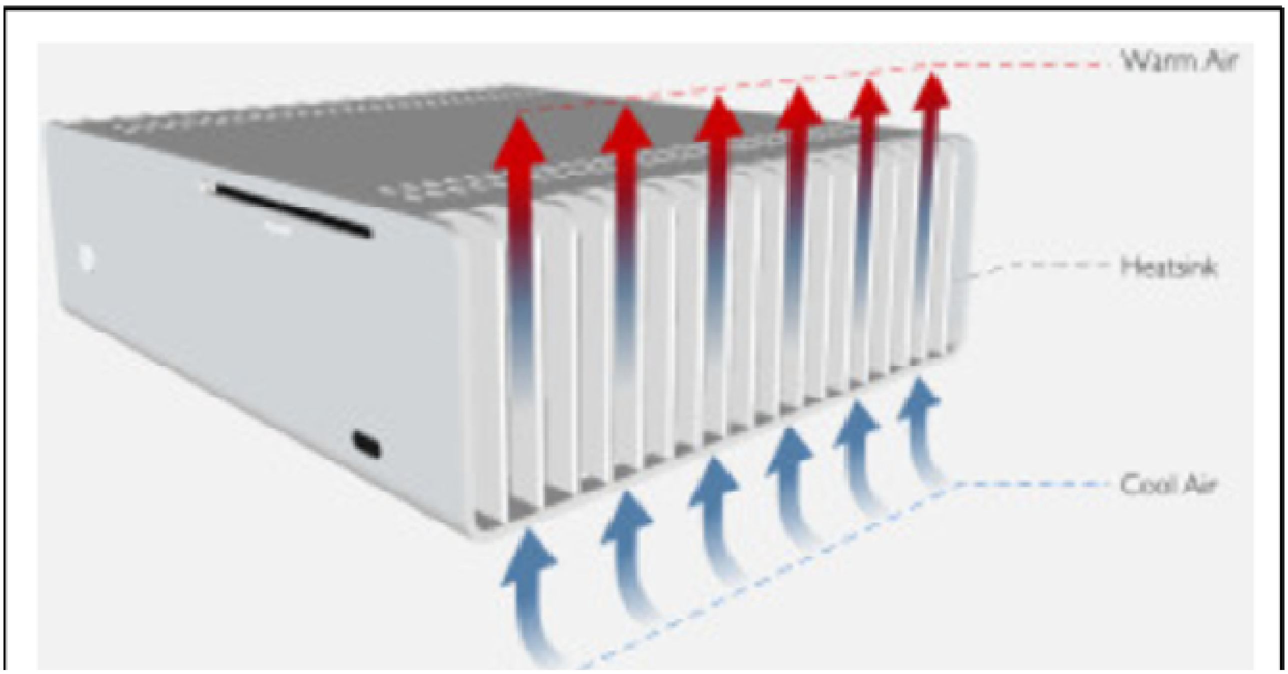


Figure 1:- Principle of Heat Sink (20).

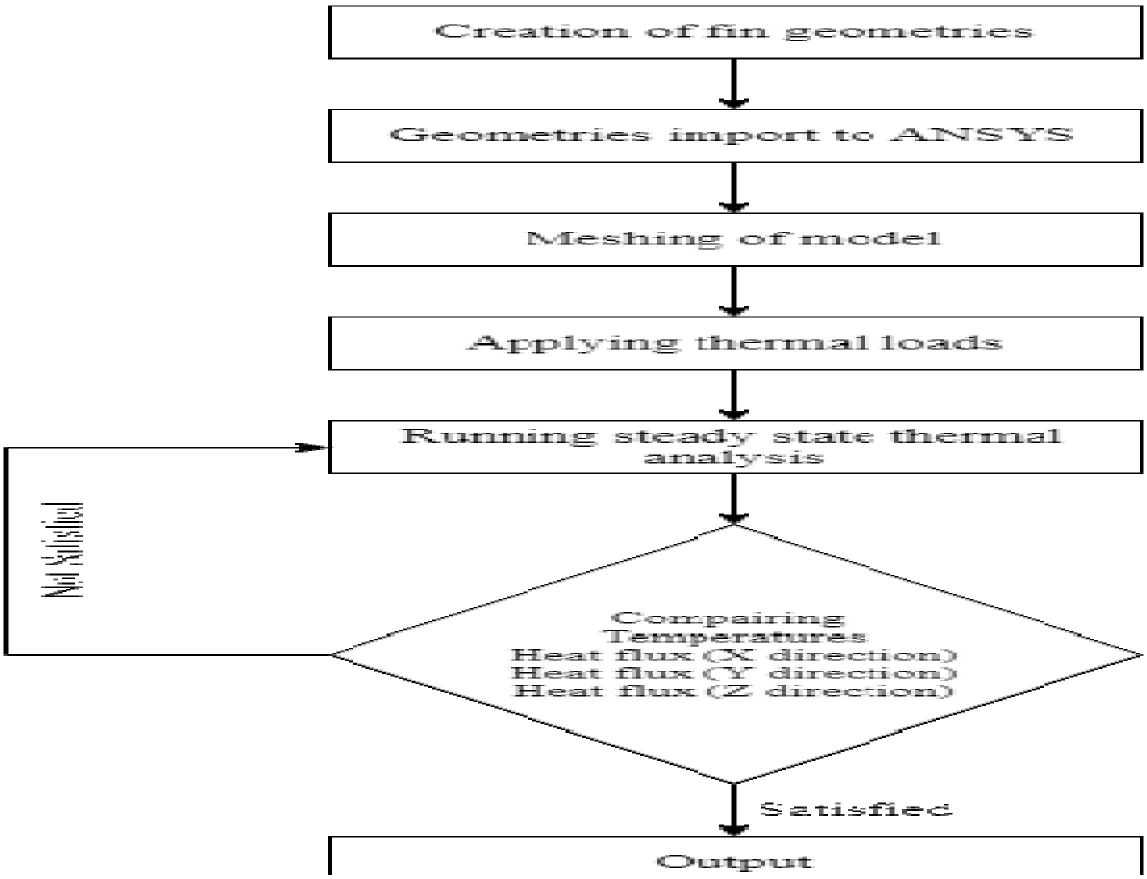


Figure 2:- Methodological flow chart.

Materials selected for the research work and their material properties:

Properties/Parameters	**Units**	**Aluminum alloy 6063-T6**	**Natural Graphite**
Density	g/cm^3	2.70	1.95
Young's Modulus	MPa	69500	15000
Thermal Conductivity	W/mK	201	470
Poisson's Ratio	-	0.3	0.23
Ultimate Tensile Strength	MPa	190	524
Tensile Yield Strength	MPa	160	500

Methodology:-

A lot of literature is in the trend for the comparative analysis of different heat sinks and already a lot has been done but most of the researches have focused on the comparison between pin fin and pate fin heat sinks but it cannot be justified which one is better than the other. All the shapes with different cooling mechanisms have their own importance in their application point of area (75-76). In the present research work 'plate-fin' heat sinks is used due to the cost-effectiveness and easy maintenance and high rate of heat dissipation and ultimately the optimal cooling and heat transfer enhancement. A comparative analysis of 16 specimens in two cases is done between the solid and honeycomb perforated 'plate-fin' heat-sinks to evaluate the heat transfer augmentation (77-78).

Preparation Of The Cad Model

The designs of heat sink with Rectangular fins, One side tapered fins, Inverted T-section Fins, I-section fin, with 'inline' arrangement is prepared in Solid works in IGES format. A flat platform of 40mm X 50mm X 5mm is common in all designs (79-80).

Design of Fins

The design of all fins (rectangular, one side tapered, inverted t section and I section) with and without honeycomb structure has been constructed on SOLIDWORKS®. The detailed construction of all the fins has mentioned below. All the dimensions considered are in mm. The material used for base plate is Al. Alloy 6063 – T6 and for plate fins is Graphite. The dimension for honeycomb core is as follows height of core (hc) is 22mm, θ is 120°, thickness of core (tc) is 0.1mm, width of core (a) is 1mm and number of cores (n) is 108-432.

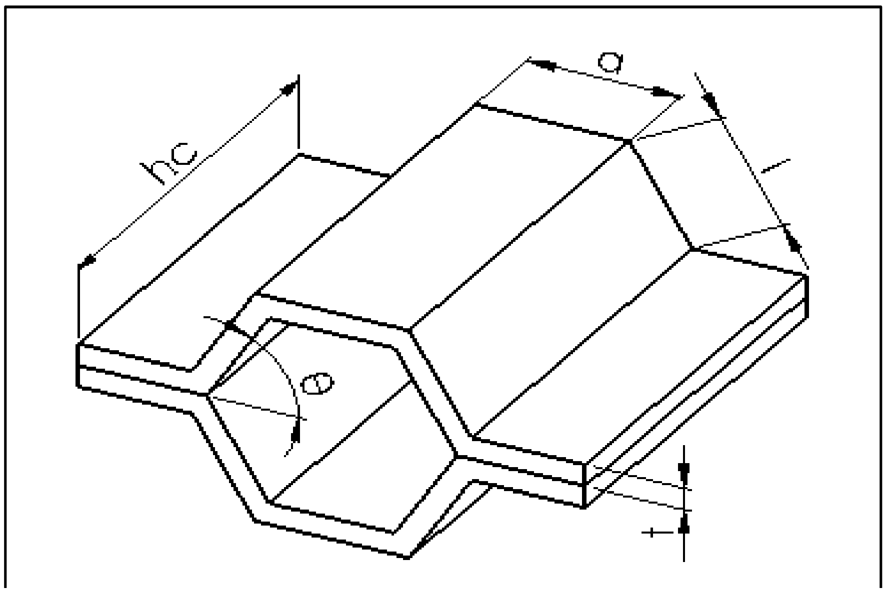
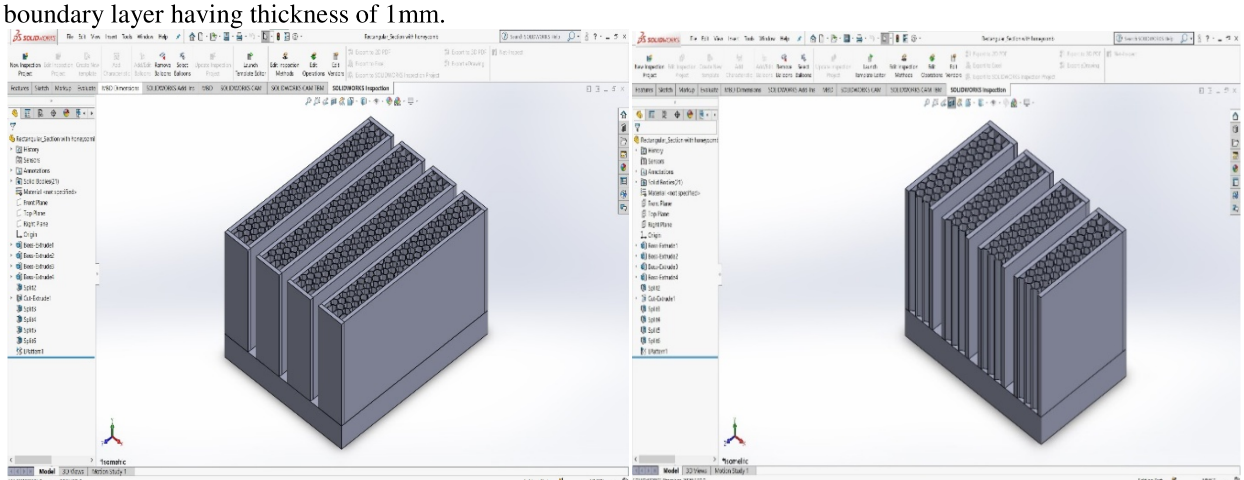


Figure 3:- 3D view of the perforated honeycomb.

Rectangular section

The rectangular section containing with and without honeycomb structure have a base plate of 40mm x 50mm x 5mm. The fins are attached to it having fin top, fin base as 6mm and fin height as 22mm. The fins are covered with a boundary layer having thickness of 1mm.



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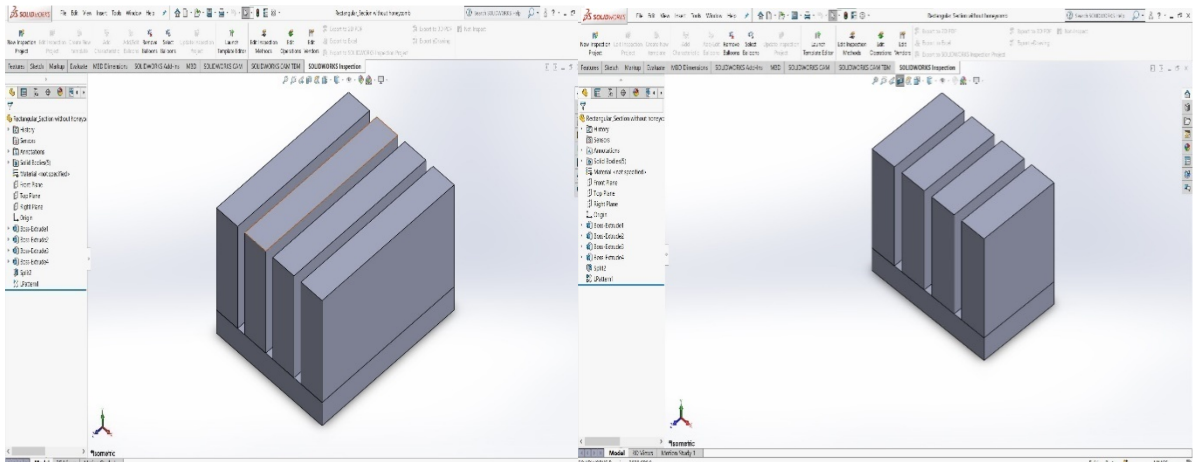
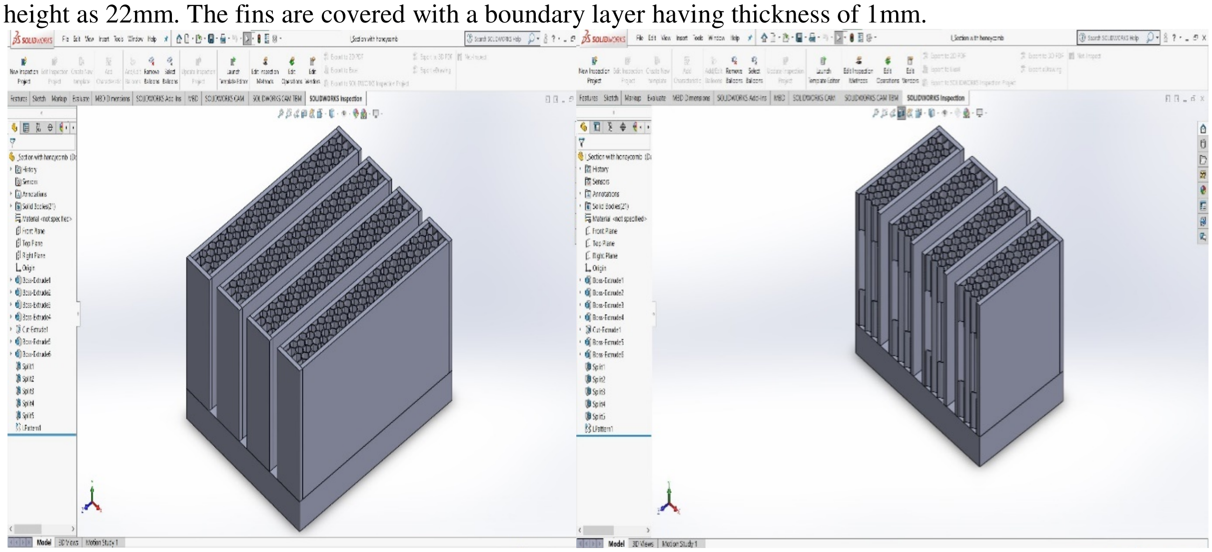


c) d)

Fig 4:- CAD model of rectangular fin using solid works.

I-Sectional fin

The I-sectional fin containing with and without honeycomb structure have a base plate of 40mm x 50 mmx 5mm. The fins are attached to it having fin top as 6mm x 6mm, fin base as 6mm x 6mm, fin web as 10mm x 4mm and fin height as 22mm. The fins are covered with a boundary layer having thickness of 1mm.



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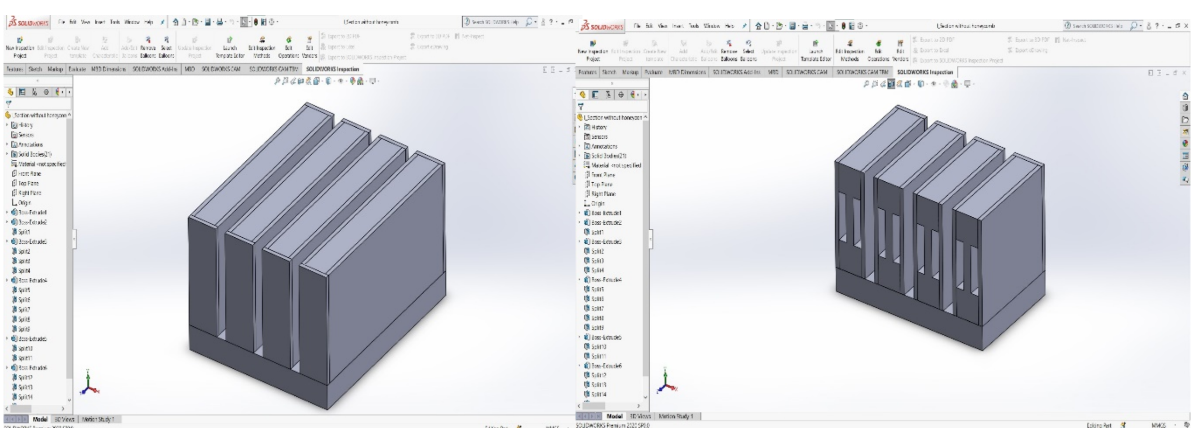
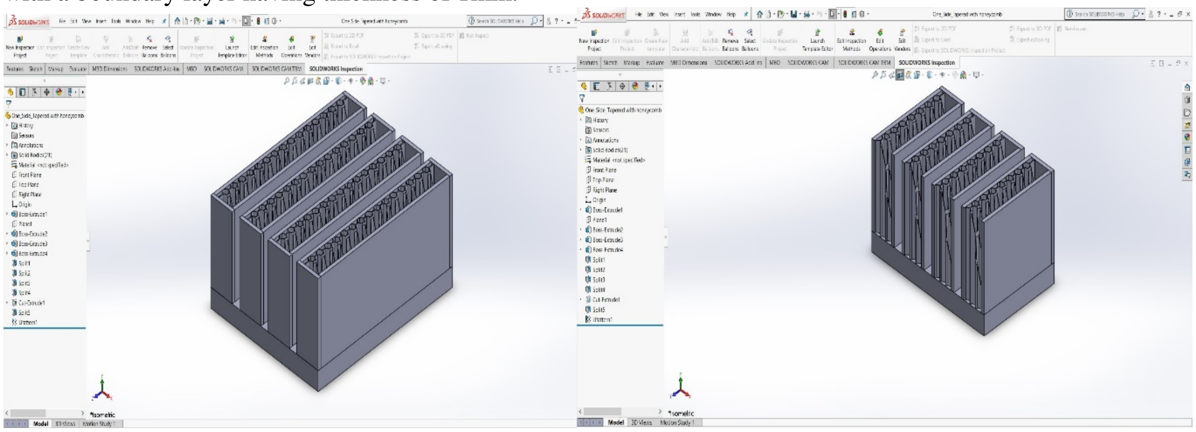


C) d)

Fig 5:- CAD model of I-sectional fin using solid works®

One Side Tapered

The one side tapered section containing with and without honeycomb structure have a base plate of 40mm x 50mm x 5mm. The fins are attached to it having fin top as 2mm, fin base as 6 and fin height as 22mm. The fins are covered with a boundary layer having thickness of 1mm.



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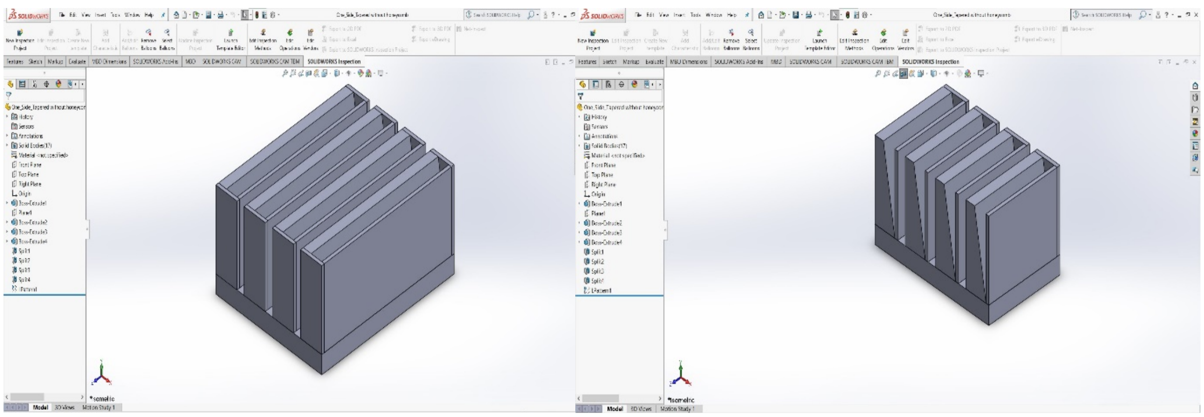
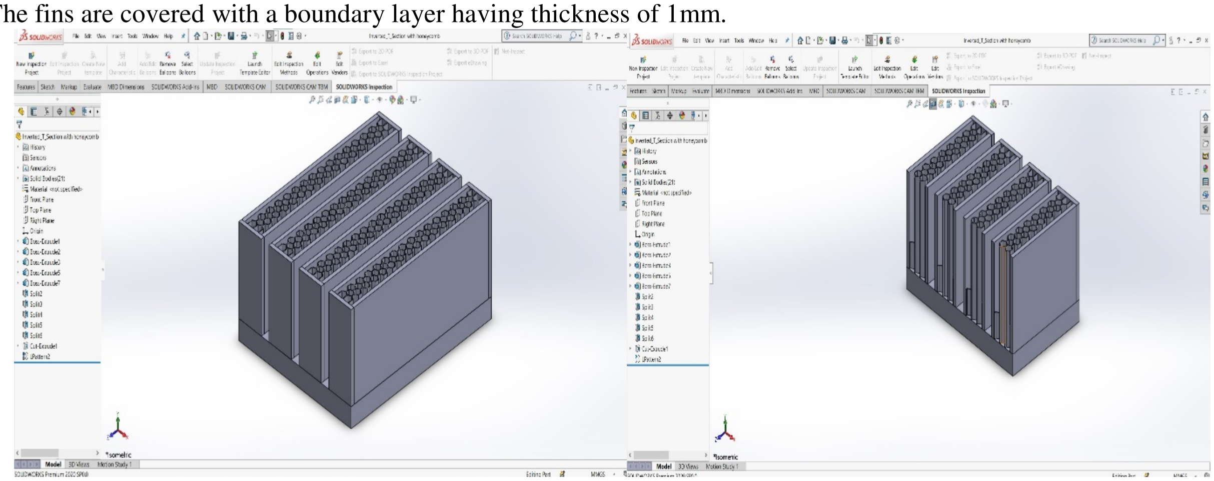


b) d)

Fig 6:- CAD model of one side tapered fin using solid works®.

Inverted T section

The Inverted T section containing with and without honeycomb structure have a base plate of 40mm x 50mm x 5mm. The fins are attached to it having fin base as 6mmx 6mm, fin web as 16mm x 2mm and fin height as 22mm. The fins are covered with a boundary layer having thickness of 1mm.



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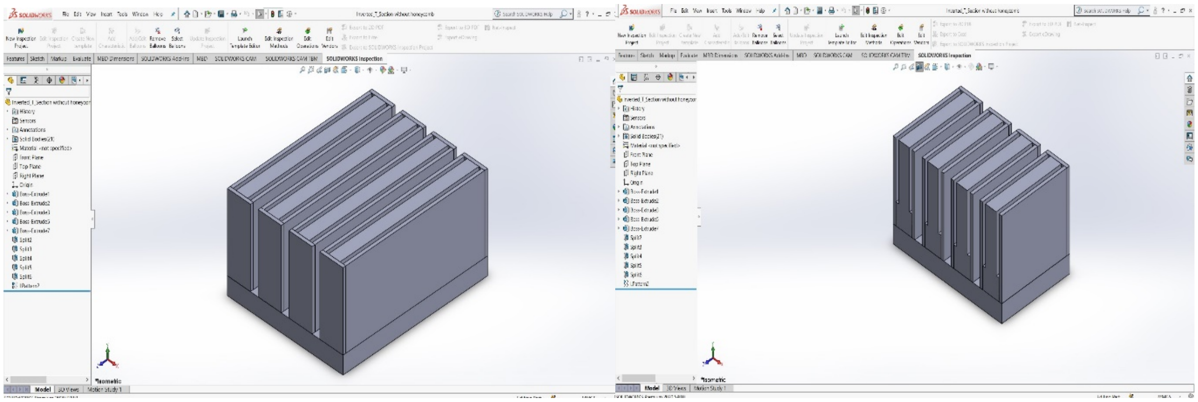


C) d)

Fig 7:- Inverted T sectional fin using solid works®.

Table 1:-

CAD models prepared in Solidworks®					
Fig no.	Type of fin	A	b	C	d
Fig 4	Rectangular	Perforated fin with 4 boundary layers	Perforated fin with 3 boundary layers	Solid fin with 4 boundary layers	Solid fin with 3 boundary layers
Fig 5	I-sectional	Perforated fin with 4 boundary layers	Perforated fin with 3 boundary layers	Solid fin with 4 boundary layers	Solid fin with 3 boundary layers
Fig 6	One side tapered	Perforated fin with 4 boundary layers	Perforated fin with 3 boundary layers	Solid fin with 4 boundary layers	Solid fin with 3 boundary layers
Fig 7	Inverted-T sectional	Perforated fin with 4 boundary layers	Perforated fin with 3 boundary layers	Solid fin with 4 boundary layers	Solid fin with 3 boundary layers

The 3D models of the heat sinks generated in Solid works software are imported into Ansys workbench design modeler in IGES format. Required geometric thermal operations have been carried out to simplify he geometry for improved meshing.

Analysis:

Analysis of the plate fin heat sink is performed in Ansys workbench this software is used for pre and post processing whereas Fluent Solver is used for solving. By proper material selection, fin geometry-shape under natural convection conditions have been analyzed with 16 specimen of the heat sinks. In all the simulations constant heat flux of 20W and later 40W in two different cases has been analyzed and simulation is carried with base of the heat sink as Aluminium alloy 6063-T6 and natural graphite as fin material for all thermal conditions.

Meshing of the domain:

The second part of pre-processing is the mesh generation. After the model is imported to Ansys workbench it is then launched in the meshing module for the mesh generation. In our research work hexahedral brick elemental meshing has been used with course meshing size.

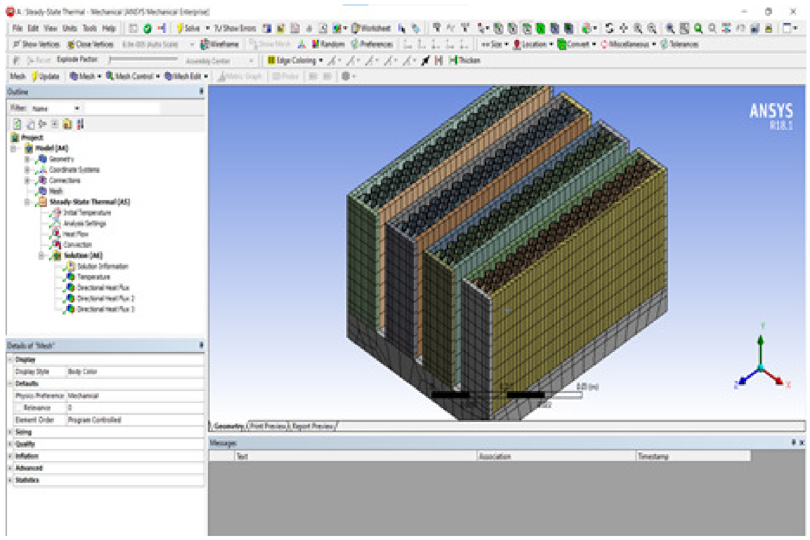


Figure7:- Hexahedral mesh.

Boundary conditions:

In this analysis of the plate fins of different profiles are modeled and analysis is done with a constant heat flow of 20W and 40W from the base plate in the Y direction. In this work heat sink materials considered are aluminum alloy 6063-T6 for base and natural graphite for fins. The analysis is carried out with 37.85^{0}C ambient. A constant heat tansfer coefficient of 60W/m^2 is applied for all the four fins irrespective of the material properties. After boundary conditions, the simulation is performed under 'steady state' thermal conditions for prepossing the data in terms of output variables for heat transfer rate. Similar iterations have also been followed for all the fin designs. The post processing is the last step of finite element analysis and in this research work the output variables are the temperature distribution and the directional heat fluxes along x,y and z directions(79-80).

Numerical method:

The heat is transferred across the base by conduction and through the fins by conducion and convection. Whenever a thermal difference is found in any solid media, flow of heat from high gradient to low gradient occurs.For base plate the rate at which heat $\dot{q}_x$ is transferred is proportional to the temperature gradient of $\frac{d\dot{T}}{d\dot{x}}$ times the area$\dot{A}c$through which heat is transferred.

$\dot{q}_x \propto \dot{A}c\frac{d\dot{T}}{d\dot{x}}$

It is found that the amount of heat which flows actually is dependent on the thermal conductivity $\dot{\lambda}$,(a physical property of the system.

So, $\dot{q}_x = -\dot{\lambda}\dot{A}\frac{d\dot{T}}{d\dot{x}}$………. (1)

The negative sign indicates a heat flow from higher to lower temperature.

Equation 1, is known as Fourier'slaw of conduction.

under steady state: condition$\dot{\lambda}$ is considered a constant and radiation from the surface is negligible.heat generation effects are absent, convective heat transfer coefficient $\bar{h}_c$

between the fluid (air) is uniform.

Similarly, the amoutnt of heat-transfer obtained bu convection between a fluid (air) and a surface is

$\dot{q}_c = \bar{h}_c\dot{A}_s\Delta T$………. (2)

Under Steady state condition

Rate of heat flow by conduction = rate of heat flow by conduction + rate of heat transfer by convection

the element at x of element at x + dx from surface between x and x + dx

$\dot{q}_x = \dot{q}_{x+dx} + dq_{convection}$ ………. (3)

$\dot{q}_{x+dx} = \dot{q}_x + \frac{d\dot{q}_x}{dx}d\dot{x}$

Applying the taylore series of expansion we obtain

$\dot{q}_{x+dx} = -\dot{\lambda}\dot{A}\frac{d\dot{T}}{d\dot{x}} - \frac{\dot{\lambda}\,d}{d\dot{x}}(\dot{A}\frac{d\dot{T}}{d\dot{x}})dx$

And

$dq_{convection} = \bar{h}_c \dot{A}_s \Delta T$

$= \bar{h}_c \dot{A}_s (\dot{T}_s - \dot{T}_\infty)$

So, equation 3 can be written as

$\frac{d}{d\dot{x}}(\dot{A}c\frac{d\dot{T}}{d\dot{x}}) - (\frac{\bar{h}_C d}{\dot{\lambda} d}\frac{\dot{A}}{\dot{x}})((\dot{T}_s - \dot{T}_\infty) = 0$

or

$\frac{d^2\dot{T}}{d\dot{x}^2} + (\frac{1}{\dot{A}_C}\frac{d\dot{A}_C}{d\dot{x}})\frac{d\dot{T}}{d\dot{x}} - (\frac{1}{\dot{A}_C}\frac{\bar{h}_C d}{\dot{\lambda} d}\frac{\dot{A}s}{\dot{x}})((\dot{T}_s - \dot{T}_\infty) = 0$ ………. (4)

For prescribed fin $\dot{A}_C$ is constant so:$\frac{d\dot{A}_C}{d\dot{x}} = 0$ and $\frac{d\dot{A}\dot{s}}{d\dot{x}} = P$ and under steady state thermal analysis $\bar{h}_c$ and $\dot{\lambda} are constant$ equation 4, reduces to

$\frac{d^2\dot{T}}{d\dot{x}^2} - \frac{\overline{h_C}\dot{P}}{\dot{\lambda}\dot{A}_C}(\dot{T}_s - \dot{T}_\infty) = 0$ ………. (5)

To simplify the form of the equation 5, and to find the fin temperature above the ambient temperature: excess temperature (θ) can be defined as:

$$\Theta(\dot{X}) = \dot{T}_{(x)} - \dot{T}_\infty \qquad ……….\ (6)$$

Since

$\dot{T}_\infty$ is constant so,

$\frac{d\theta}{d\dot{x}} = \frac{d\dot{T}}{d\dot{x}}$………. (7)

Put equation 6 in equation 5 we obtain

Where $m^2 = \frac{\overline{h_C}\dot{P}}{\dot{\lambda}\dot{A}_C}$

The general solution of the equation 7 is of the form

$\theta(\dot{X}) = C_1 e^{m}\dot{X} + C_2\, e^{-m}\dot{X}$ ………. (8)

To evaluate the values of C_1 and C_2 boundary conditions in terms of temperature (heat flow) at fin base ($\dot{X} = 0$) and at $(X \doteq L)$ for fin tip

At $\dot{X} = 0$: $\theta(0) = T_b - \dot{T}_\infty = \dot{\theta} b$ ………. (9)

At $(X \doteq L)$: $[\dot{T}_{(L)} - \dot{T}_\infty] = -\dot{\lambda}\dot{A}_C \frac{d\dot{T}}{d\dot{x}} |\dot{X} = L$ ………... (10)

equation 9, presents the concept that the rate that is conducted across the fin must balance the convection from the tip.

substituting equation 8 in equations 9 and 10 we obtain

$\dot{\theta} b = C_1 + C_2$ ………. (11)

And $\bar{h}_c(C_1 e^{mL} + C_2 e^{-mL}) = \dot{\lambda} m(C_2\, e^{-mL} - C_1 e^{mL})$

$$\frac{\theta}{\dot{\theta}_b} = \frac{\cos\bar{h}_c(L-\dot{X}) + (\frac{\bar{h}_c}{m_{\dot{\lambda}}})\sin\bar{h}_c(L-\dot{X})}{\cos\bar{h}_c mL + \left(\frac{\bar{h}_c}{m_{\dot{\lambda}}}\right)\sin\bar{h}_c mL}$$

$\dot{q}_f$ can be obtained by applying Fouriers law at the fin base

$\dot{q}_f = \dot{q}_b - \dot{\lambda}\dot{A}_C \frac{d\dot{T}}{d\dot{x}} |\dot{X} = 0 \quad = \dot{\lambda}\dot{A}_C \frac{d\dot{T}}{d\dot{x}} |\dot{X} = 0$

So, knowing the temperature distribution

$\theta(\dot{X})$, $\dot{q}_f$ may be obtained as

$\dot{q}_f = \sqrt{\bar{h}_C p\dot{\lambda}\dot{A}_C\theta\bar{b}}\,\frac{\sin\bar{h}_c mL + (\frac{\bar{h}_c}{m_{\dot{\lambda}}})\cos\bar{h}_c mL}{\cos\bar{h}_c mL + (\frac{\bar{h}_c}{m_{\dot{\lambda}}})\sin\bar{h}_c mL}$

For fins of non-uniform cross-sectional area

Thermal analysis becomes more complicated. For such cases the second term of equation of equation (4)

$\frac{d^2\dot{T}}{d\dot{x}^2} + (\frac{1}{\dot{A}_C}\frac{d\dot{A}_C}{d\dot{x}})\frac{d\dot{T}}{d\dot{x}} - (\frac{1}{\dot{A}_C}\frac{\bar{h}_C d}{\dot{\lambda} d}\frac{\dot{A}s}{\dot{x}})((\dot{T}_s - \dot{T}_\infty) = 0$ is retained

$\dot{A}_{CS}$ for solid fin = $2NHL_w + (N-1)\,SL$

$\dot{A}_{CP}$ for perforated fin = $WL_w + \pi N\,[(Ht_w) + (ndt_w) - (\frac{nd}{2})]$

Governing equations:

For the numerical analysis

Assumptions taken in this study are:

1. The flow is steady and 3 D with constant properties.
2. The Boussinesq approximation is used for the natural convection flow for density and temperature relation.
3. The flow considered is incompressible and for steady state the density is considered.

The set of governing equations for natural convection are

Continuity equation

$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} + \frac{\partial W}{\partial Z} = 0$ ………. (12) (under steady state and natural convection velocitycomponents are equal to zero)

Momentum equation

$$U\frac{\partial U}{\partial X} + V\frac{\partial V}{\partial Y} + W\frac{\partial W}{\partial Z} = -g - \frac{1}{\rho}\frac{\partial P}{\partial X} + V\frac{\partial}{\partial}\frac{(\partial U)}{(\partial X)} + + (\frac{\partial}{\partial}\frac{(\partial U)}{(\partial Y)})\ (\frac{\partial}{\partial}\frac{(\partial T)}{(\partial Z)}) \text{ ………. (13)}$$

Energy equation

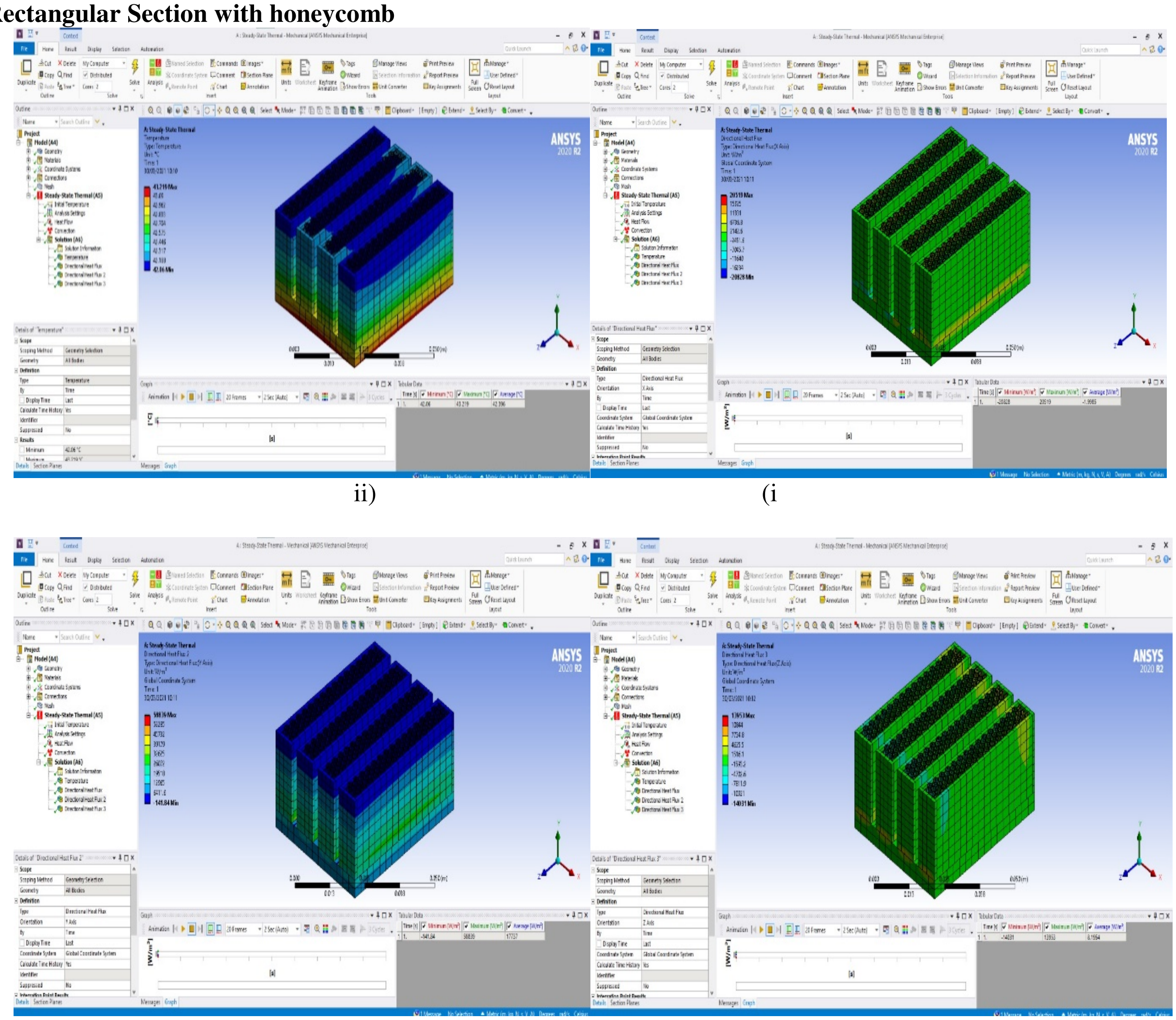
$$U\frac{\partial T}{\partial X} + V\frac{\partial T}{\partial Y} + W\frac{\partial T}{\partial Z} = \alpha\ (\frac{\partial}{\partial}\frac{(\partial T)}{(\partial X)} + (\frac{\partial}{\partial}\frac{(\partial T)}{(\partial Y)} + (\frac{\partial}{\partial}\frac{(\partial T)}{(\partial Z)} \text{ ………. (14)}$$

Results And Discussion:-

Thermal Analysis

Case 1:- Heat Flow with 20W

Rectangular Section with honeycomb

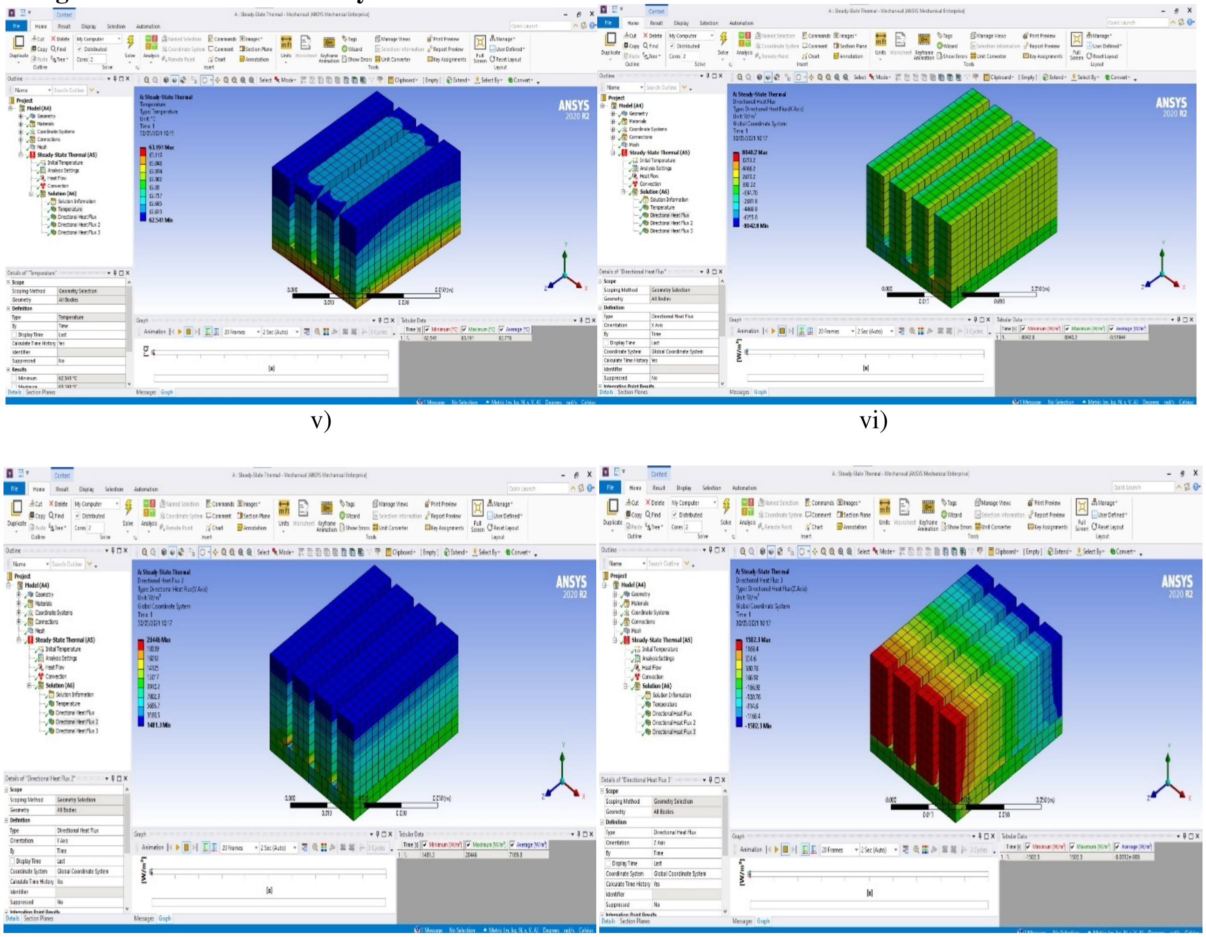


ii) (i

iii) iv)

Figure 8:- Steady state thermal analysis of rectangular perforated fin using Ansys®.

Rectangular section without honeycomb

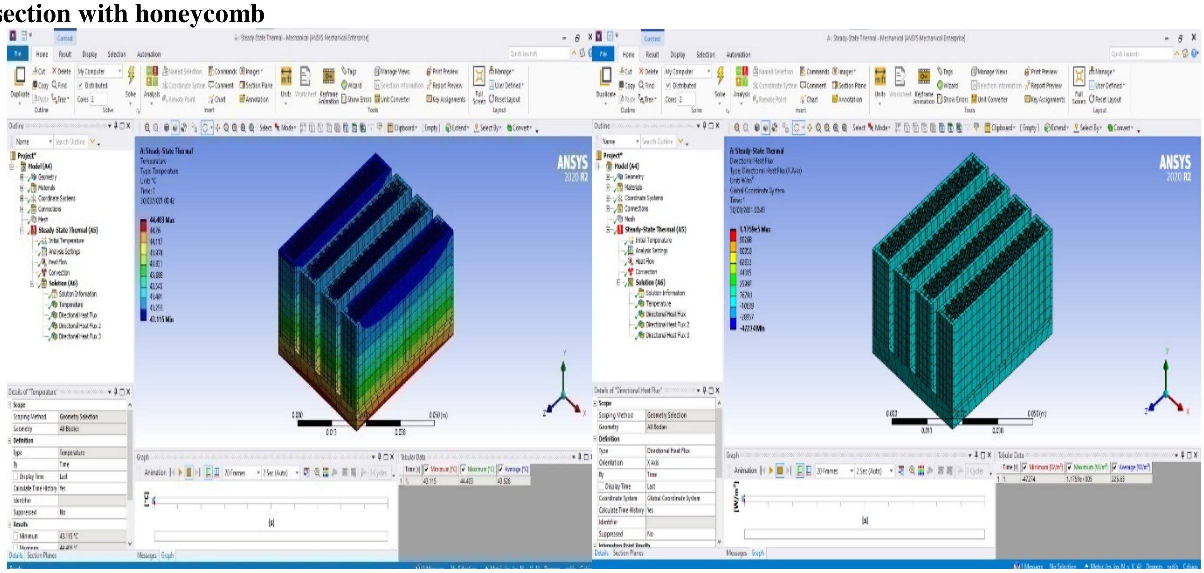


v) vi)

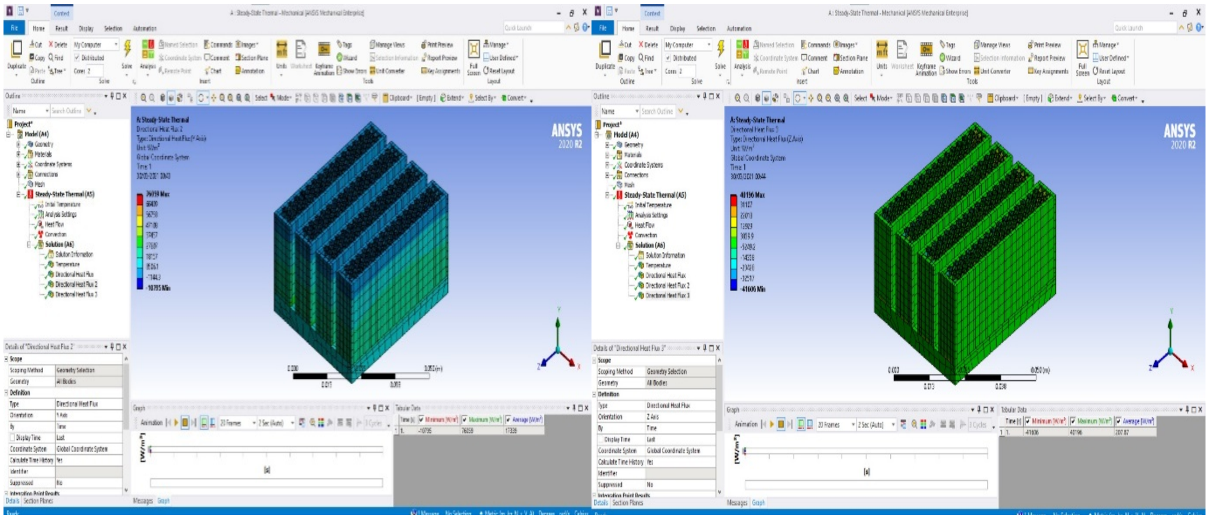
vii) viii)

Figure 9:- Steady state thermal analysis of rectangular non- perforated fin using Ansys®.

I section with honeycomb



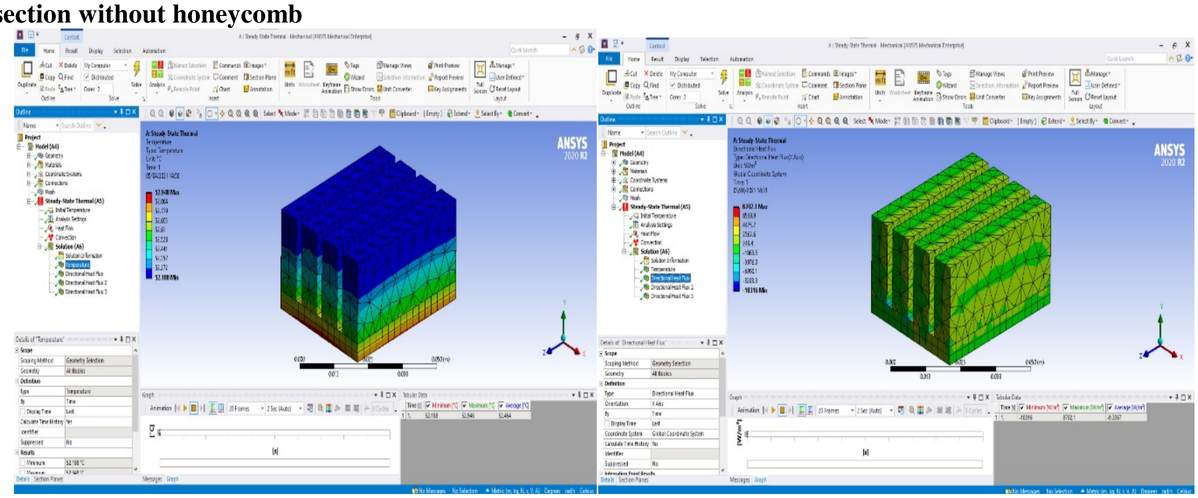
ix) x)



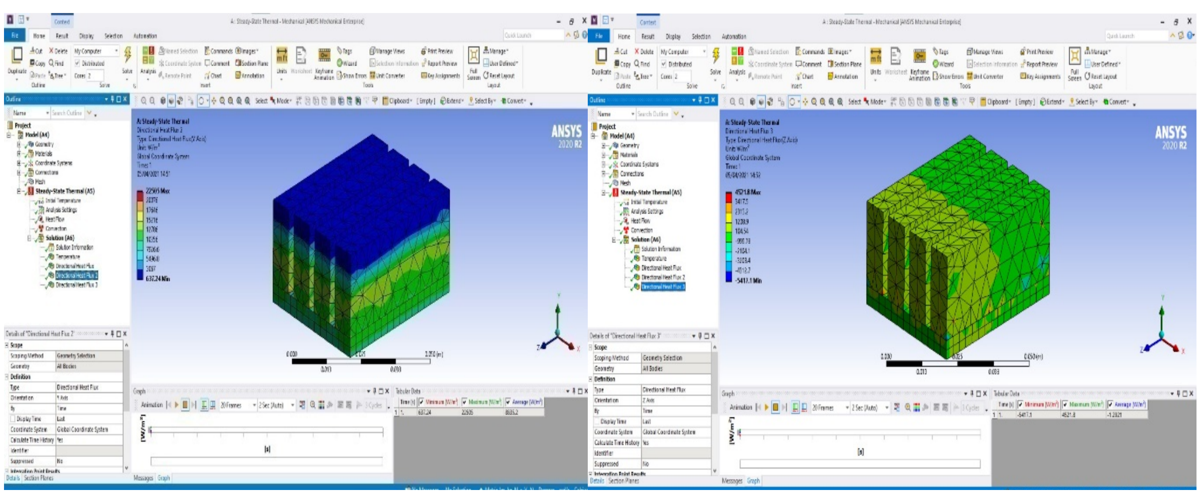
xi) xii)

Figure 10:- Steady state thermal analysis of I-sectional perforated fin using Ansys®.

I section without honeycomb



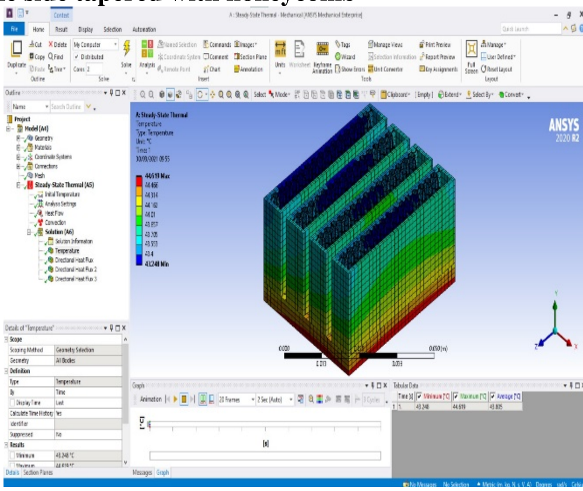
xiii) xiv)



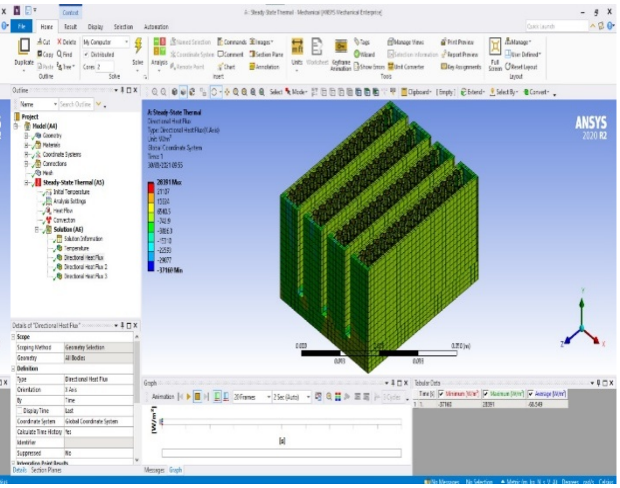
xv) xvi)

Figure 11:- Steady state thermal analysis of I-sectional non-perforated fin using Ansys®.

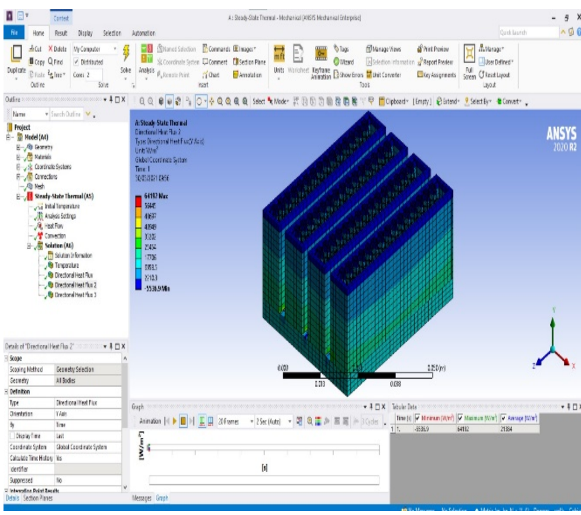
One side tapered with honeycomb



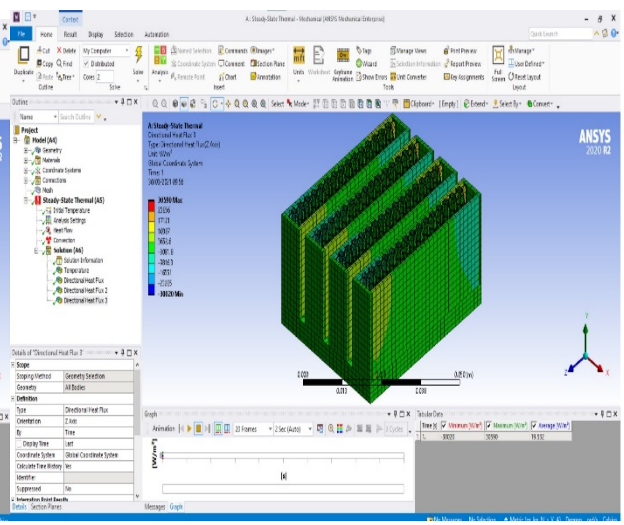
xvii)



xviii)



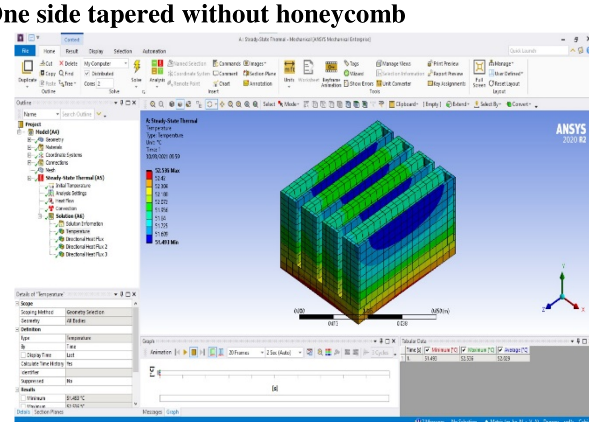
xix)



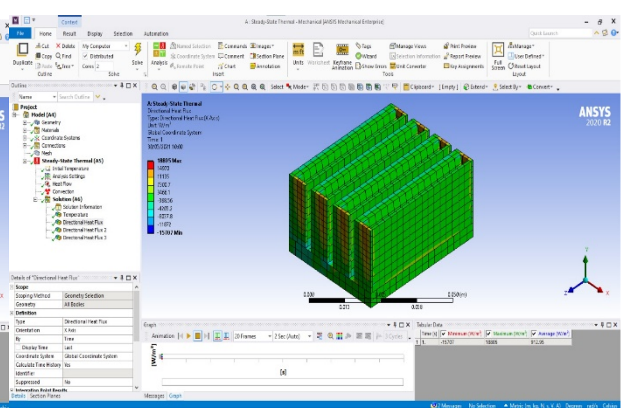
xx)

Figure 12:- Steady state thermal analysis of one side tapered perforated fin using Ansys®.

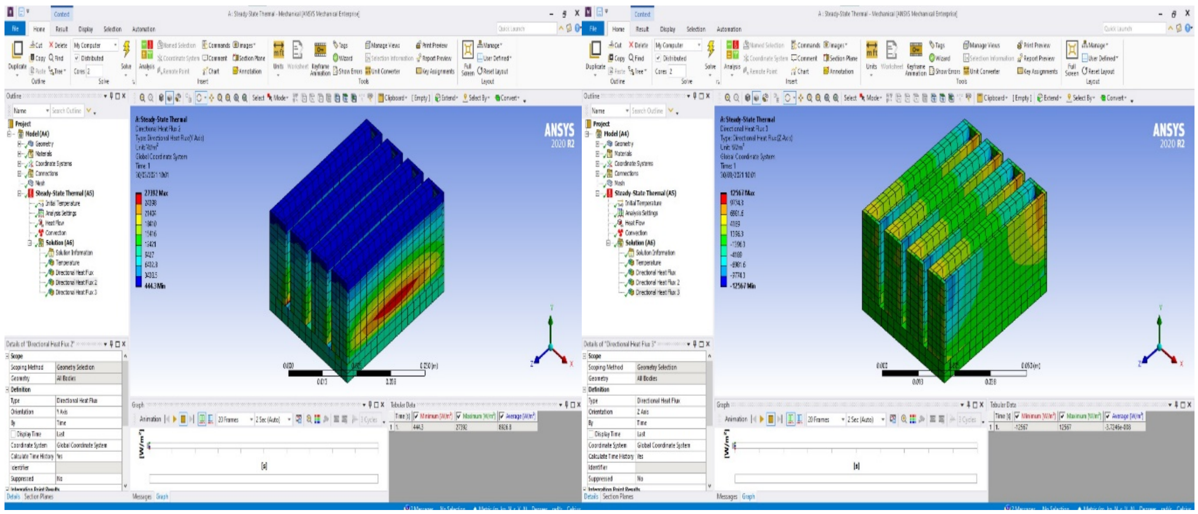
One side tapered without honeycomb



xxi)



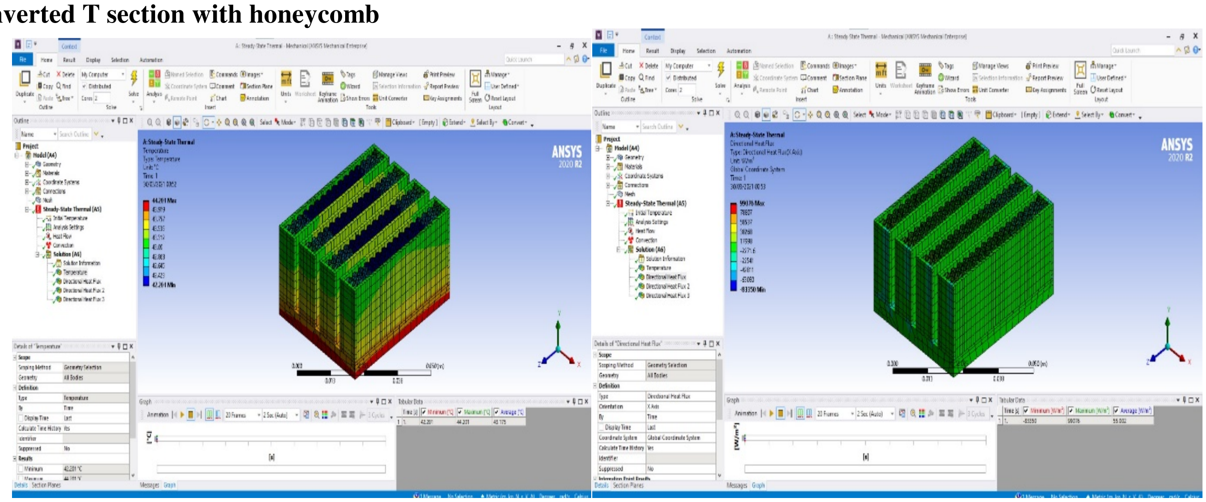
xxii)



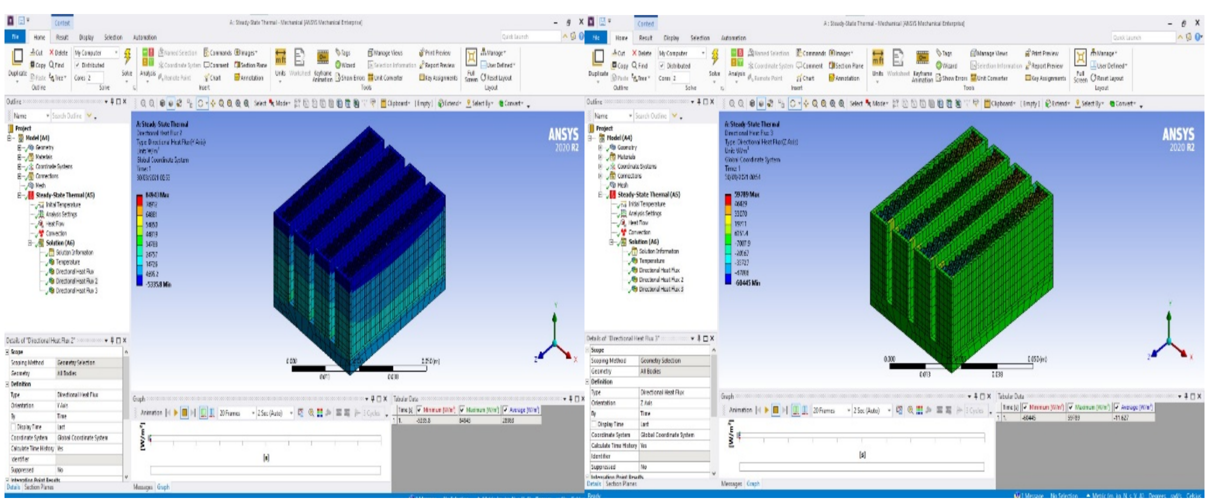
xxiii) xxiv)

Figure 13:- Steady state thermal analysis of one side tapered non-perforated fin using Ansys®.

Inverted T section with honeycomb



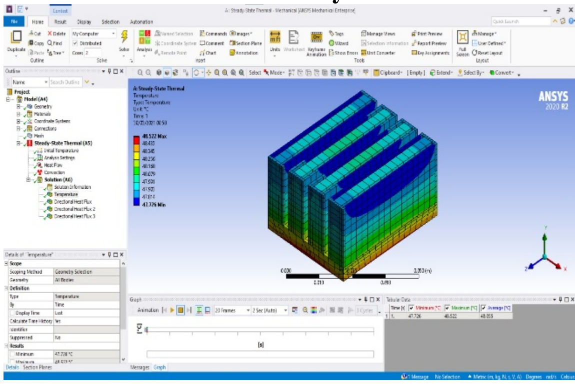
xxv) xxvi)

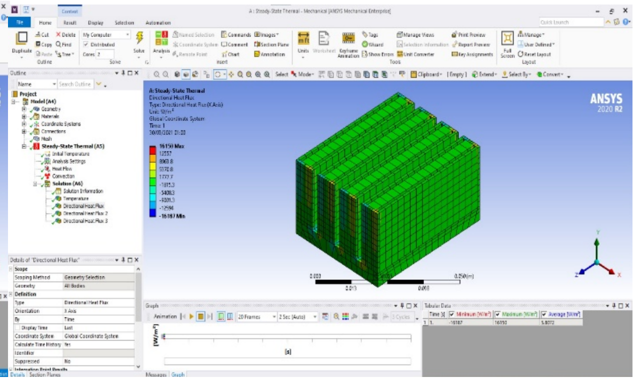


xxvii) xxviii)

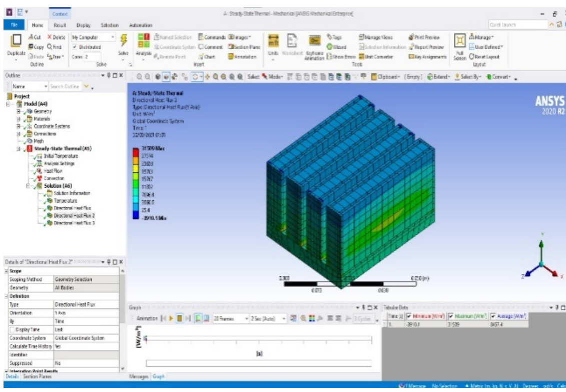
Figure 14 Steady state thermal analysis of inverted-T perforated fin using Ansys®.

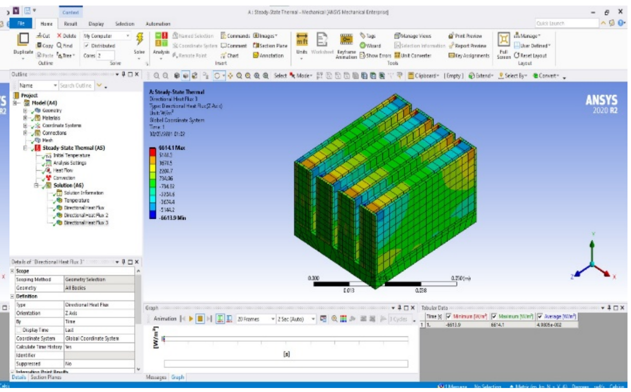
Inverted T section without honeycomb





xxix) xxx)





xxxi) xxxii)

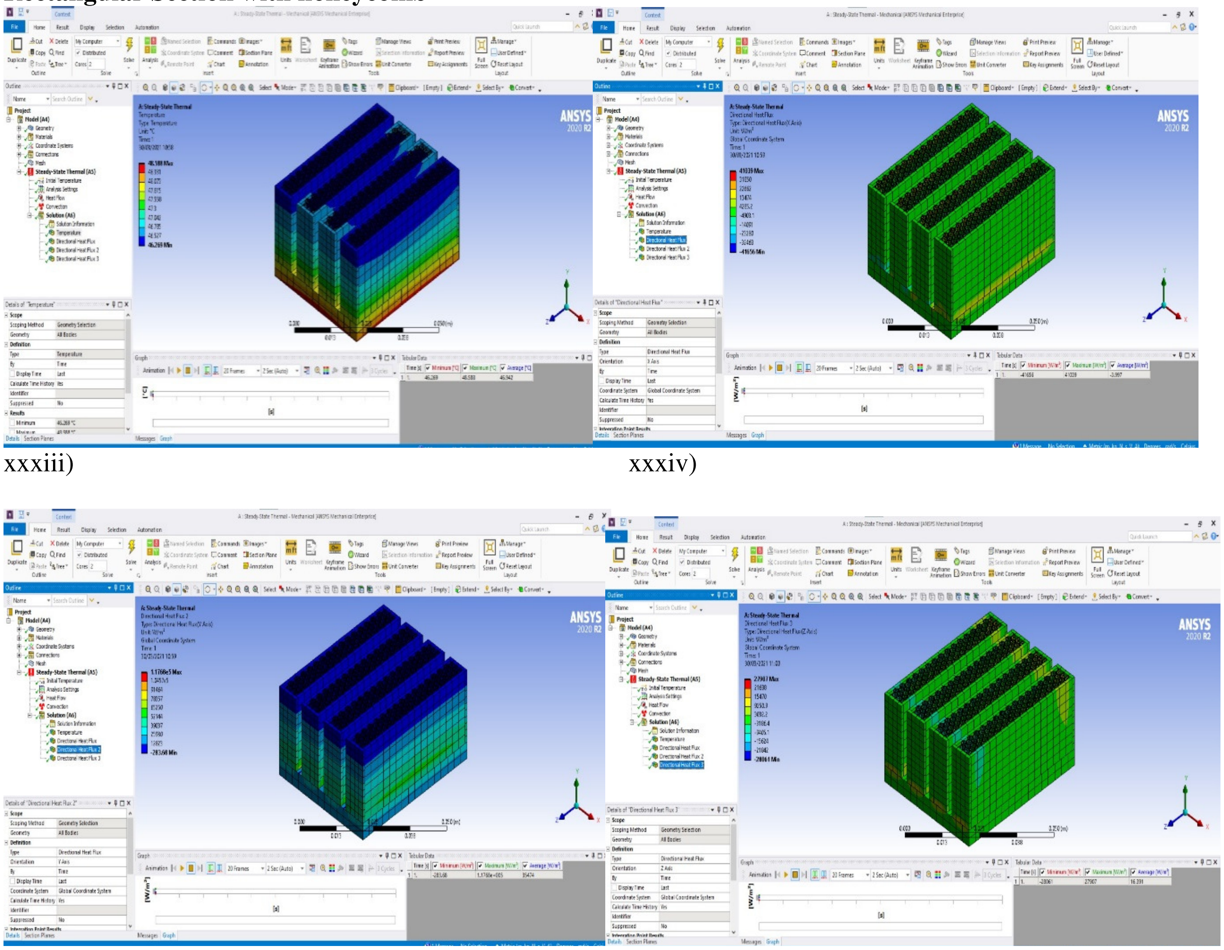
Figure 15:-Steady state thermal analysis of inverted-T non-perforated fin using Ansys®.

Table 2:-

Steady state thermal analysis using Ansys®								
Fig no.	Heat flow (W)	Numbers of figures		Type of fin	Output variables			
		With honeycomb	Without honeycomb		Temperature °C	Directional heat flux W/mm^2		
						X direction	Y direction	Z direction
8 - 9	20	i), ii), iii), iv)	v), vi), vii), viii)	Rectangular	i), v)	ii), vi)	iii), vii)	iv), viii)
10-11	20	ix), x), xi), xii)	xiii), xiv), xv), xvi)	I-sectional	ix), xiii)	x), xiv)	xi), xv)	xii), xvi)
12 -13	20	xvii), xviii), xix), xx)	xxi), xxii), xxiii), xxiv)	One side tapered	xvii), xxi)	xviii), xxii)	xix), xxiii)	xx), xiv)
14 -15	20	xxv), xxvi), xxvii), xxviii)	xxix), xxx), xxxi), xxxii)	Inverted T sectional	xxv), xxix)	xxvi), xxx)	xxvii), xxxi)	xxviii), xxxii)

Case 2: Heat Flow with 40W

i. Rectangular Section with honeycomb

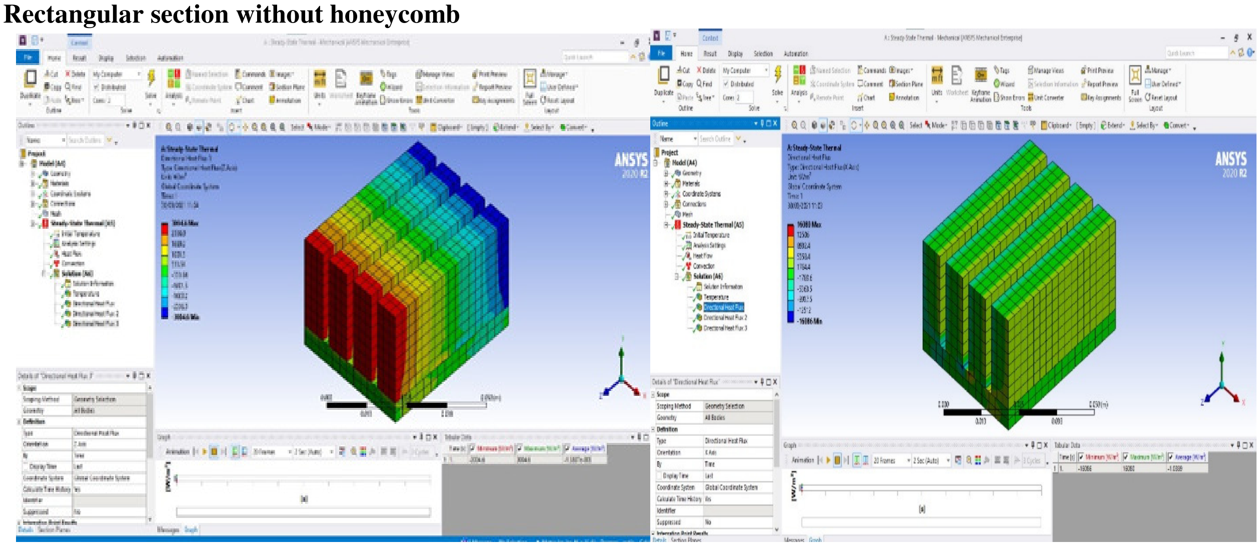


xxxiii) xxxiv)

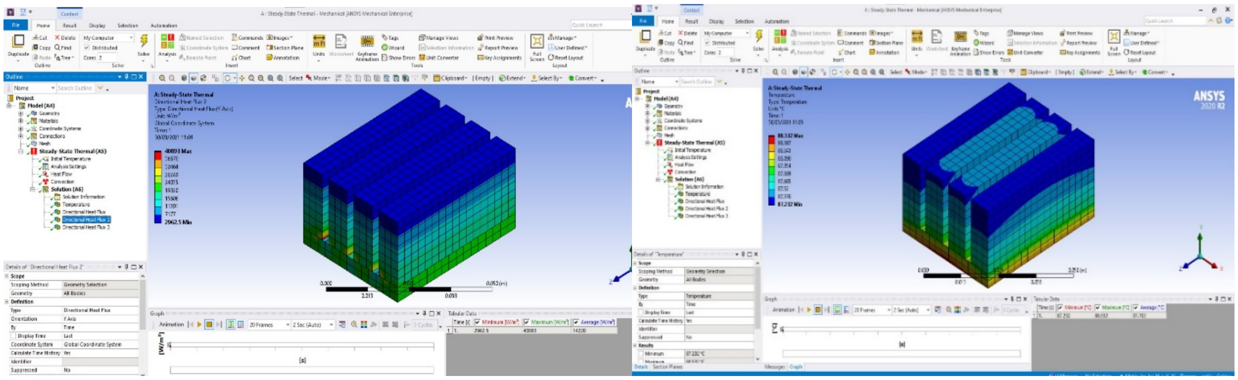
xxxv) xxxvi)

Figure 16:-Steady state thermal analysis of rectangular perforated fin using Ansys workbench®.

Rectangular section without honeycomb



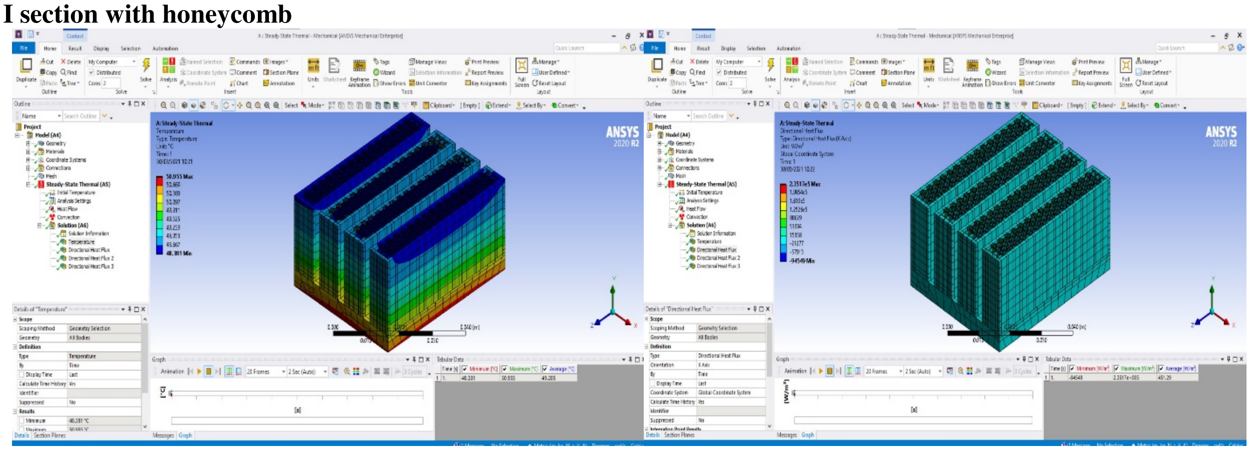
xxxvii) xxxviii)



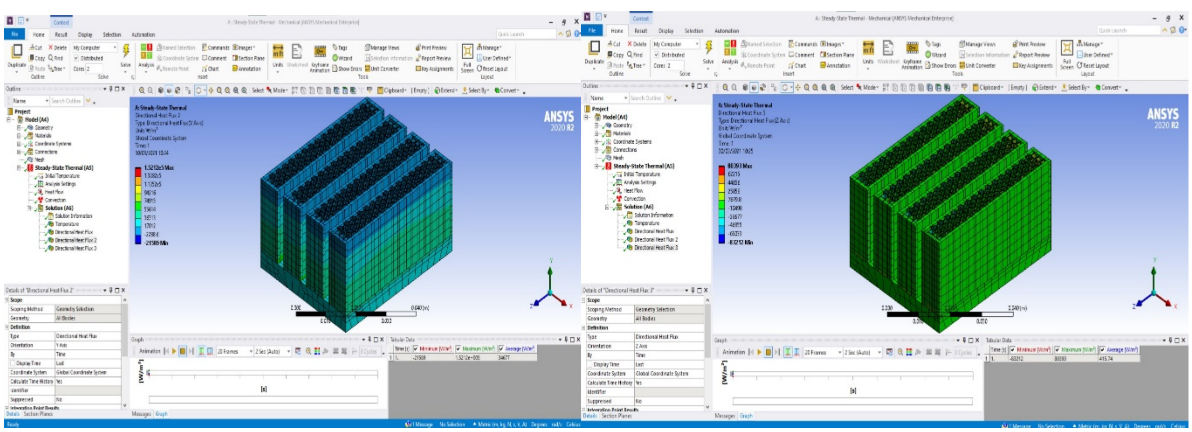
xxxix) xxxx)

Figure 17:- Steady state thermal analysis of rectangular non-perforated fin using Ansys®.

I section with honeycomb



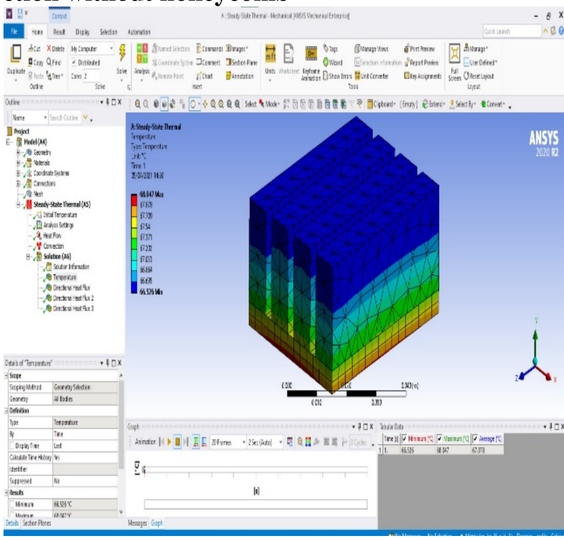
xxxxi) xxxxii)



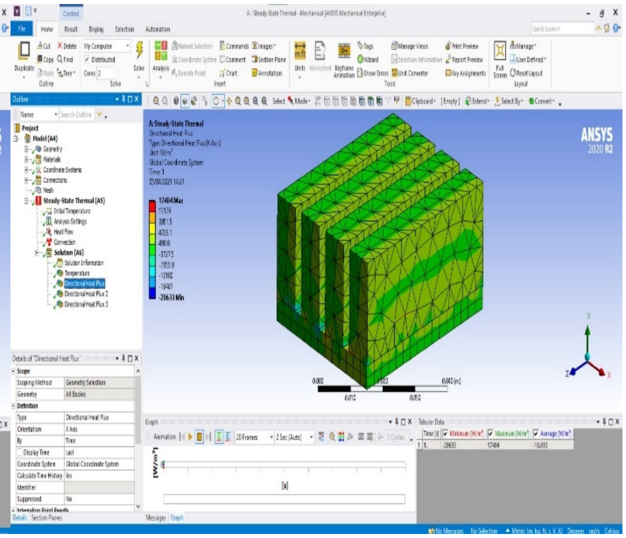
xxxxiii) xxxxiv)

Figure 18:- Steady state thermal analysis of I-sectional perforated fin using Ansys®.

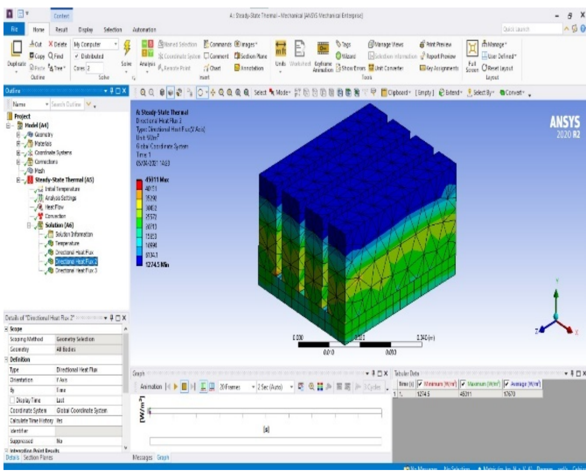
I section without honeycomb



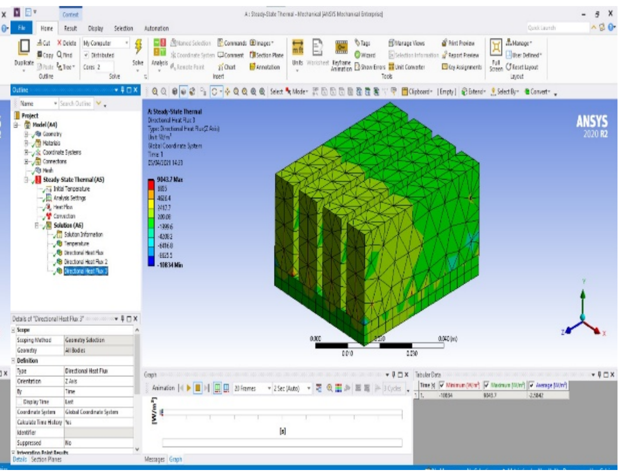
xxxxv)



xxxxvi)



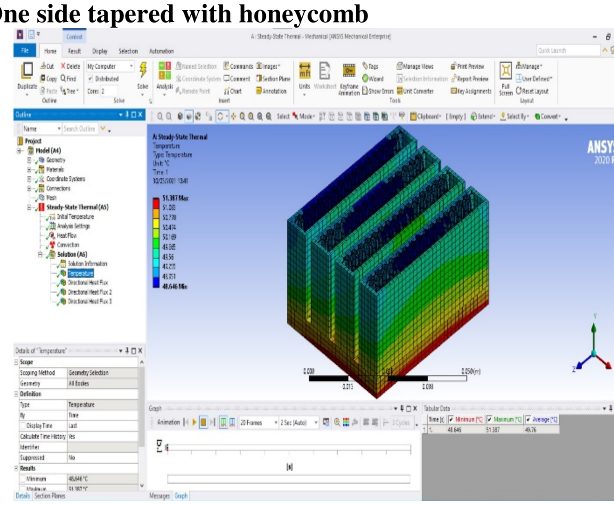
xxxxvii)



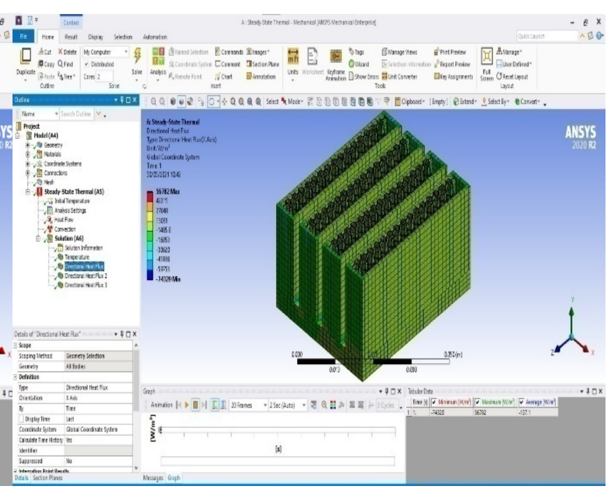
xxxxviii)

Figure19 Steady state thermal analysis of I-sectional non-perforated fin using Ansys®

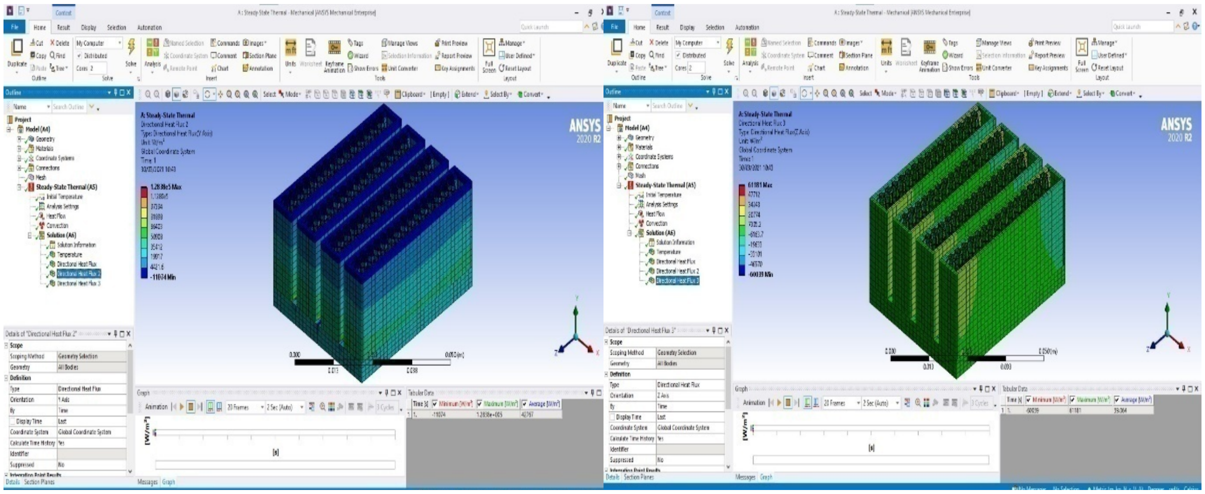
One side tapered with honeycomb



xxxxix)



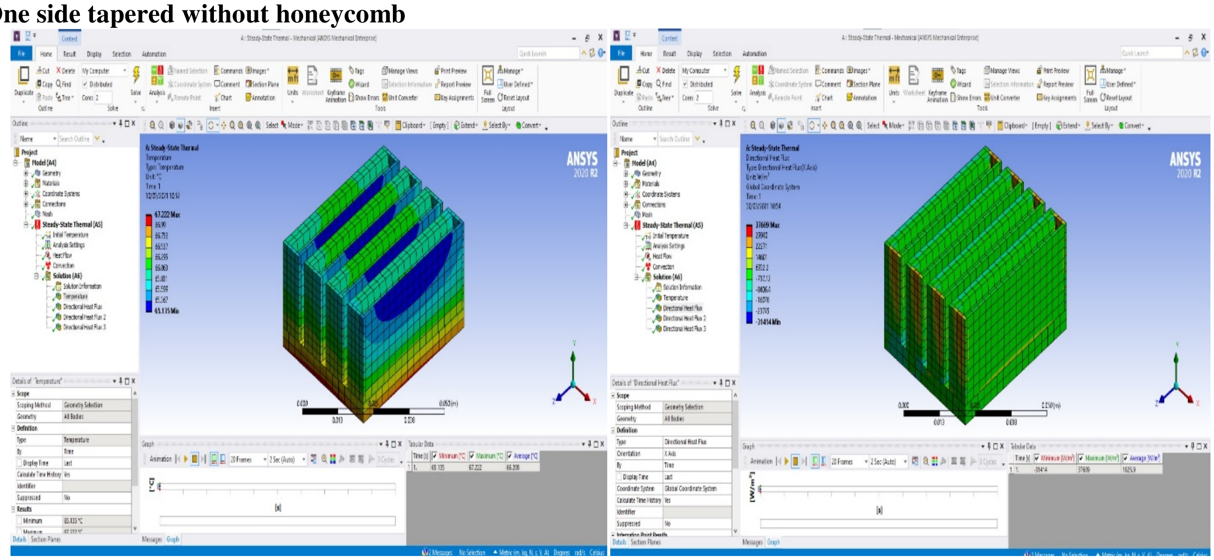
xxxxx)



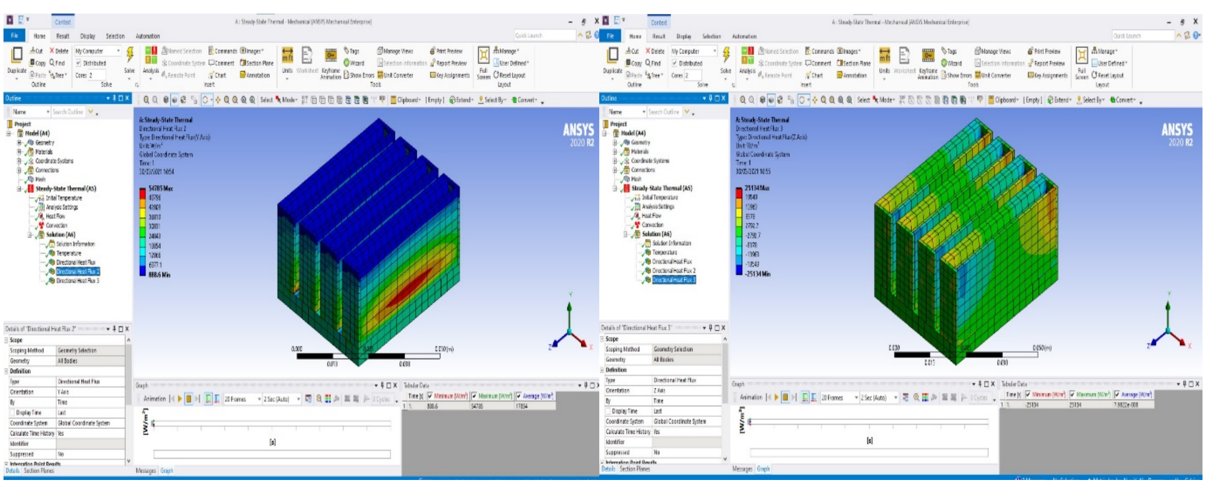
xxxxxi) xxxxxii)

Figure 20:- Steady state thermal analysis of one side tapered perforated fin using Ansys®.

One side tapered without honeycomb



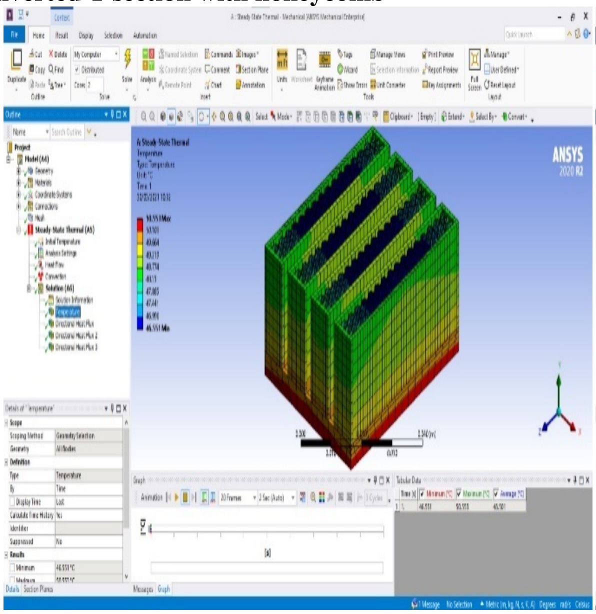
xxxxxiii) xxxxxiv)

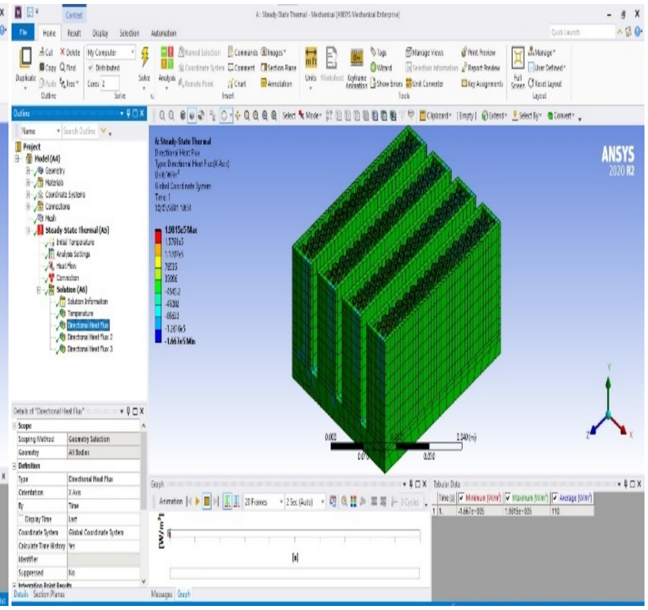


xxxxxv) xxxxxvi)

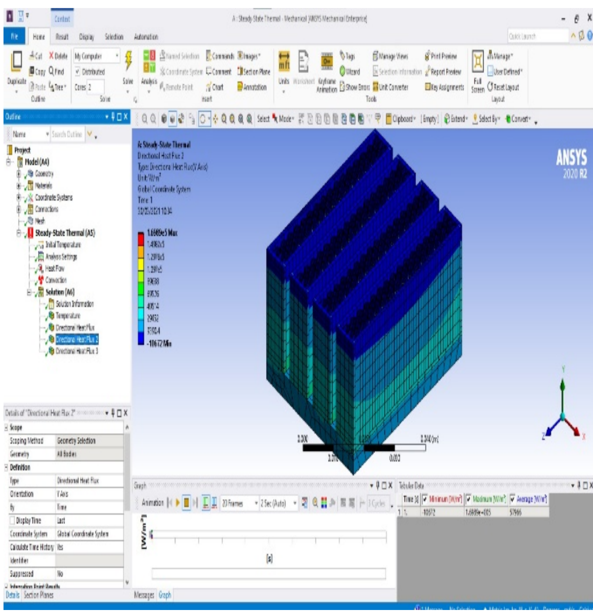
Figure 21:- Steady state thermal analysis of one side tapered non-perforated fin using Ansys®.

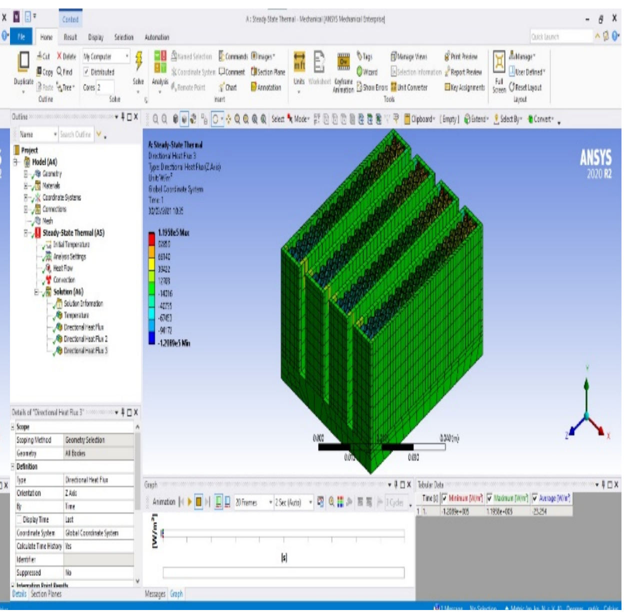
Inverted T section with honeycomb





xxxxxvii) xxxxxviii)

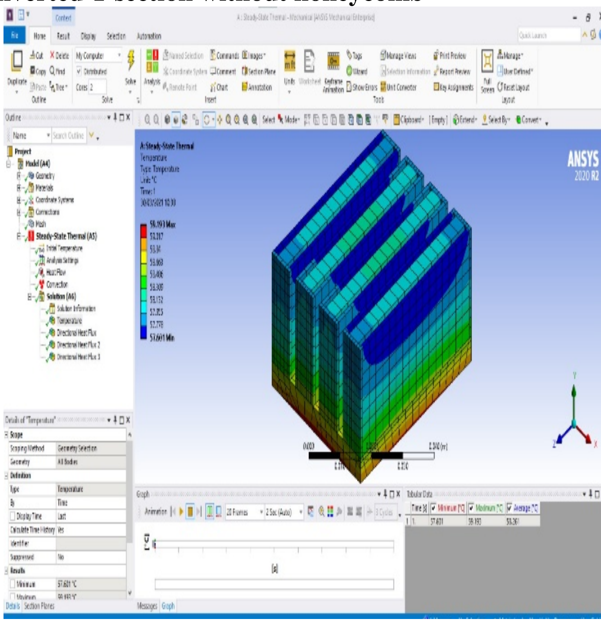




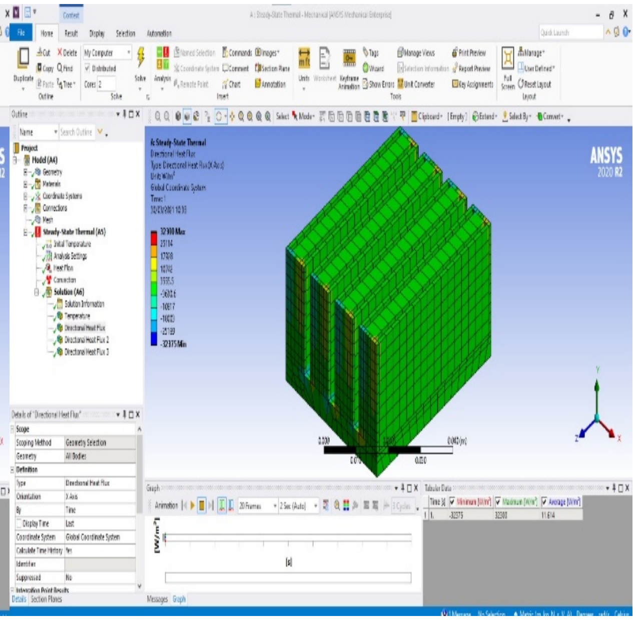
xxxxxix) xxxxxx)

Figure 22:- Steady state thermal analysis of inverted T perforated fin using Ansys®.

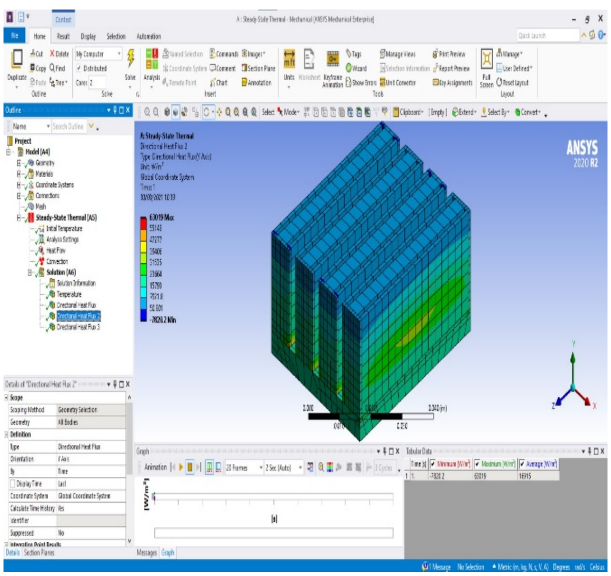
Inverted T section without honeycomb



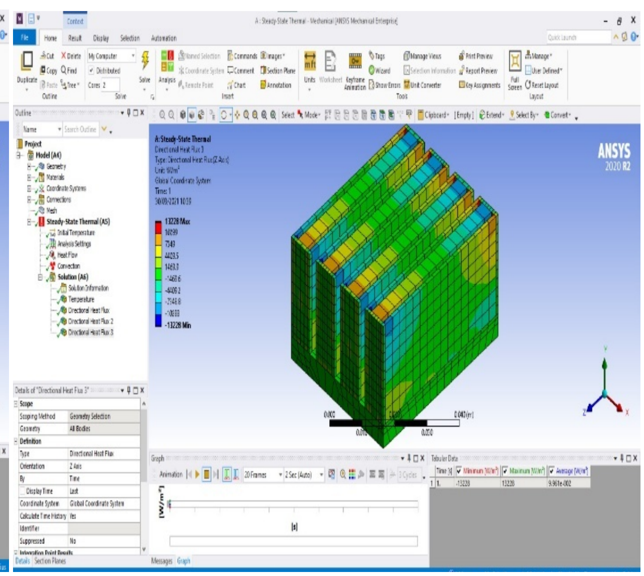
xxxxxxi)



xxxxxxii)



xxxxxxiii)



xxxxxxiv)

Figure 23:- Steady state thermal analysis of inverted-T non-perforated fin using Ansys®.

Table 3:-

Steady state thermal analysis using Ansys®								
Fig no	Heat flow (W)	Number of figures		Type of fin	Output variables			
		With honeycomb	Without honeycomb		Temperature °C	Directional heat fluxW/mm^2		
						X direction	Y direction	Z direction
16 -17	40	xxxiii), xxxiv), xxxv), xxxvi)	xxxvii), xxxviii), xxxxix), xxxx)	Rectangular	xxxiii), xxxvii)	xxxiv), xxxviii)	xxxv), xxxix)	xxxvi), xxxx)
18 -19	40	xxxxi), xxxxii), xxxxiii), xxxxiv)	xxxxv), xxxxvi), xxxxvii), xxxxviii)	I sectional	xxxxi), xxxxv)	xxxxii), xxxxvi)	xxxxiii), xxxxv)	xxxxiv), xxxxviii)
20 -21	40	xxxxix), xxxxx), xxxxxi), xxxxxii)	xxxxxiii), xxxxxiv), xxxxxv), xxxxxvi)	One side tapered	xxxxix), xxxxxiii)	xxxxx), xxxxxiv)	xxxxxi), xxxxxv)	xxxxxii), xxxxxvi)
22 -23	40	xxxxxvii), xxxxxviii), xxxxxix), xxxxxx)	xxxxxxi),xxxxxxii), xxxxxxiii),xxxxxxiv)	Inverted T sectional	xxxxxvii), xxxxxxi)	xxxxxviii), xxxxxxii)	xxxxxix), xxxxxxiii)	xxxxxx), xxxxxxiv)

Temperature variations of fin models without honeycomb perforations

Fin profile	Heat flow (watt)	Maximum (^{0}C)	Minimum (^{0}C)	Temperature drop (^{0}C)
I-Section	20	52.948	52.188	0.760
	40	68.047	66.526	1.521
Inverted T-Section	20	48.522	47.726	0.796
	40	59.193	57.601	1.592
One side tapered	20	52.536	51.493	1.043
	40	67.222	65.135	2.087
Rectangular	20	63.191	62.541	0.65
	40	88.532	87.232	1.3

Heat flux around all conditions of all fins in fin models without honeycomb perforations

Geometric condition/Fin profile	Heat flow (watt)	Directional Heat flux (W/mm^2)					
		Along X direction		Along Y direction		Along Z direction	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
I-Section	20	0.008702	-0.01032	0.022505	0.000637	0.004522	-0.00542
	40	0.017404	-0.02063	0.045011	0.001275	0.009044	-0.01083
Inverted T-Section	20	0.01615	-0.01619	0.031509	-0.00391	0.006614	-0.00661
	40	0.0323	-0.03238	0.063019	-0.00782	0.013228	-0.01323
One side tapered	20	0.018805	-0.01571	0.027392	0.000444	0.012567	-0.01257
	40	0.037609	-0.03141	0.054785	0.000889	0.025134	-0.02513
Rectangular	20	0.00804	-0.00804	0.020446	0.001481	0.001502	-0.0015
	40	0.01608	-0.01609	0.040893	0.002966	0.003005	-0.003

Temperature variations of fin models with honeycomb perforations.

Fin profile/geometric condition	Heat flow (Watt)	Maximum temperature (^{0}C)	Minimum temperature (^{0}C)	Temperature drop (^{0}C)
I-Section	20	44.403	43.115	1.288
	40	50.955	48.381	2.574
Inverted T-Section	20	44.201	42.201	2
	40	50.553	46.551	4.002
One side tapered	20	44.619	43.248	1.371
	40	51.387	48.646	2.741
Rectangular	20	43.219	42.06	1.159
	40	48.588	46.269	2.319

Heat flux around all conditions of all fins in fin models with honeycomb perforations

Geometric condition/Fin profile	Heat flow (watt)	Directional Heat flux					
		Along X direction		Along Y direction		Along Z direction	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
I-Section	20	0.11759	-0.04727	0.076059	-0.0108	0.040196	-0.04161
	40	0.23517	-0.09455	0.15212	-0.02159	0.080393	-0.08321

Inverted T-Section	20	0.099076	-0.08335	0.084943	-0.00534	0.059789	-0.06045
	40	0.19815	-0.1667	0.16983	-0.01067	0.11958	-0.12089
One side tapered	20	0.028391	-0.03716	0.064192	-0.00554	0.03059	-0.03002
	40	0.056782	-0.07432	0.12838	-0.01107	0.061181	-0.06004
Rectangular	20	0.020519	-0.02083	0.058839	-0.00014	0.013953	-0.01403
	40	0.041039	-0.04166	0.11768	-0.00028	0.027907	-0.02806

The Finite element thermal analysis (FEA) is conducted on base plate and fin of heat sink to determine temperature and heat flux. The above analysis was done to favor thermal management for miniaturized electronic component of desktop CPU. The research was done to establish an increment in heat dissipation under natural and steady state convection. for the operational reliability in terms of dissipation an optimal design for fin selection is of paramount importance. In this research analysis four different plate fins heat sinks rectangular. I-sectional, one side tapered, and inverted-T fins were selected. Based on the selection format the modelling was done in solid works® and then IGES files were imported to Ansys 20.0 for thermal analysis. The research was done for non-perforated and perforated fins in two different cases at two different heat flows 20W and 40W separately with constant $\bar{h}_c$. For the first case all nonperforated fins were thermally analyzed in Ansys software for temperature and directional heat fluxes along X, Y and Z directions. Similar procedure was done for perforated plate fin heat sinks. A total of 16 specimen were designed and analyzed under natural and steady sate convection. the base material selected was Aluminium alloy 6063-T6 and the material selected for fins was Natural Graphite for anisotropy point of view. The selection of materials was done on the basis of low weight to high strength ratio. Both the materials are good heat dissipators and in terms of cost and reliable availability. From the above analysis among the non-perforated fins inverted T sectional fin has shown a better heat transfer enhancement in terms of temperature drop and directional heat flux along Y axis between the fin base and the fin top as compared to other fins taken for thermal analysis. In order to further enhance the concept of heat transfer augmentation same analysis is done for honeycomb perforations and the simulation results obtained for the non-perforated T sectional fin validated with the simulation results for the inverted T sectional honeycomb fin at constant heat flow of 20W and 40W separately. The results obtained from the comparative analysis showed that the perforated fins showed better heat transfer performance under same dimensions and constant heat flows of 20W and 40W.

A lot of literature on the perforations using advanced materials and its comparison with the non-perforated has always shown the perforated fins have better heat transfer performance as compared with the non-perforated fins (75-78). Kumar et al (80) conducted the analytical simulation on four different fins using Ansys® under steady state and natural convection. The results of output variables temperature and total heat-flux were also obtained through experimentation and were found in good agreement with those obtained by Finite element software Ansys®. They found that the conical draft pin fin has maximum temperature of 81.68°C with maximum drop of 5.6°C but the total heat flux was found highest in 'circular pin- fin'. The validation results revealed that the conical draft pin fin showed better heat transfer properties as compared to other fins considered in the study. Deshmukh and Nagendra (81) conducted thermal analysis of a non-perforated plate fin heat sink using Ansys software to determine the effects of temperature and directional heat fluxes on heat transfer enhancement under steady state and natural convection. They found that the directional heat flux along Z has a greater influence on the heat transfer enhancement. Similar study was conducted by Sivakumar and Krishna (82) who analytically performed the numerical simulation on different perforated pin fins (rectangular, circular, triangular and interrupted) under steady state convection. they found the maximum temperature drop in interrupted rectangular pin fin in addition to total heat flux maximum heat dissipation was also found in interrupted rectangular pin fin and thus outperformed other pin fins. From the above literature it can be concluded that to improve the heat transfer properties analysis of the fins is very important to increase the heat dissipation rate. So, the principle implemented in this research work is to improve the heat dissipation through the fins under steady state and natural convection. From the above result analysis, it was evaluated that at 20W and at 40W for non-perforated fins temperature drop is more for one side tapered fin followed by inverted T fin, I sectional fin and then the rectangular fin. Same analysis was done for perforated fins at same flow rate and it was evaluated that the inverted T sectional fin had maximum temperature drop followed by one side tapered fin, I-sectional fin and the rectangular fin.

From the above analysis for directional heat fluxes along x, y, and z axis in perforated fins it was concluded that the maximum heat transfer is seen along y direction, it can be attributed due to heat flow along Y direction. Maximum heat transfer can be seen in inverted T section with 0.19815W/mm^2 as maximum along x direction whereas minimum is seen in rectangular plate fin heat sink. Thus, from above analysis it can be included that heat transfer properties increase with increase in heat flow validate the literature of previous researches that maximum heat dissipation can be obtained by increasing the heat flow (83-85). Similar analysis was also done for the non-perforated plate fins and it was seen that the maximum directional flux was seen in as along direction whereas minimum value was seen in as along direction. Thus, from above analysis inverted T sectional fin was seen as optimal plate fin heat sink both in temperature drop and in maximum dissipation of heat and overall better heat transfer enhancement. For the non-perforated fins, the overall maximum directional heat flux was observed in inverted T section at 40W along Y direction and minimum value of -0.003 was also seen in inverted T section. Thus, the required objective of the present research work was successfully obtained in terms of material usage, weight reduction i.e., high strength to low weight ratio was maintained, maximum heat dissipation, cost effectiveness by usage of advanced materials in terms of anisotropy and density benefits for compactness was achieved in the present research work which ultimately led to the selection of the optimal plate fin heat sink design.

Conclusion:-

In this study, the enhancement of natural convection heat transfer from a vertical perforated fin embedded with and without honeycomb perforations has been examined using finite element technique of ANSYS software. From the comparative analysis of rectangular, inverted T-section, I-section and one side tapered fins study the conclusion is that the directional heat flux of inverted T sectional fin is found maximum as compared to other fins. At 20W and 40W the maximum directional heat flux for non-perforated inverted T-sectional fin is found along Y-axis as 0.031509W/mm^{2}and 0.063019 W/mm^2 respectively. Similar analysis for perforated inverted T sectional fin calculated the maximum directional heat flux along Y axis as 0.084943 W/mm^2 and 0.16983 W/mm^2 respectively. The results obtained the favorable temperature drop for the inverted T sectional fin both for perforated and non-perforated fins. Where as in terms of material cost one side tapered outperformed the other fins both for the non-perforated and the perforated fins. The maximum temperature drop was also favorable for inverted T sectional fin for both the cases of heat flow and constant heat transfer coefficient. So, it can be concluded that the inverted T sectional fin has shown better heat transfer properties in this analysis. From the above study the following conclusions are made:

1. The thermal analysis of inverted T fin has shown overall better heat transfer enhancement as compared to other fins taken in this research work both for perforated and non-perforated case.
2. The one side tapered can be a better choice in terms of material cost and savings and has shown better thermal performance after inverted T section followed by I sectional fin and the rectangular fin.

Future scope

For future perspective same analysis can be done at different load conditions using various profiles of different types of heat sinks. Further improvement in performance can be done at different aspect ratios, orientations, number of fins can be incremented using different input parameters and at different output parameters. Metal 3D printing of heat sinks can also prove a new innovation in the present research work for further optimization where even complex profiles can be manufactured.

Following are the more researches which are recommended in this research work

1. Same study can be experimentally validated with the analytical work done.
2. For natural convection a 45degree angle of air flow between the fin with base can lead to more enhancement in the heat transfer.
3. Transient thermal analysis is also a new study in the same analytical form of work.
4. Same comparison can be done for plate and pin fin heat sinks of varying fin geometries.
5. Approach can be established for more advanced materials.

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