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Modelling, Thermal Design and Thermal Analysis of an Electronic Unit

Basavarajappa S[*1], Mallikarjuna V Bidari[2], Mujeebulla Khan Guttal[3]
[*1]Asst Professor, Mechanical Engineering Department, P.E.S.I.T.M, Karnataka, India
[2,3]Lecturer, Mechanical Engineering Department, S.T.J.I.T, Karnataka, India
basuinbetur@gmail.com

Abstract

Thermal design and thermal analysis is necessary to effectively and efficiently cool electronic components for its best operation. This cooling method is known as electronic cooling. In the present work a finned heat sink for Electronic Unit (EU) of an Air Vehicle (AV) has been designed using the empirical heat transfer correlations by considering natural convection and radiation heat transfer modes[1] for given heat load. The preliminary calculations indicated that plane surface was inadequate to dissipate the heat, thus necessitating fins to augment the heat dissipation to the ambient. Heat transfer calculations were carried out for finned surface to find out the maximum surface temperature, which is well below the maximum permissible temperature. Further fin space optimization has been made by using empirical correlation for given heat sink to bring down the surface temperature further below.3D modelling of all the parts of the electronic unit was carried out in SolidWorks2012.Using Flo.EFD software the CFD model is generated and analysed to obtain the total thermal mapping of the above electronic unit. The CFD results were used to validate the results obtained through the excel spread sheet program developed based on the empirical correlations available in the literature.

Keywords: Electronic Unit (EU), Natural Convection, Radiation, Empirical Correlations, Fin space optimization

Introduction

Electronic equipments are pervading almost every phase of modern living, from sewing machines to aviation industries. Whenever electrical current flows through a resistive elements heat is generated. The heat continues to be generated as long as the current continues to flow. As the heat builds up, temperature of the resistive element starts to rise. The electronic element fails when the temperature exceeds its permissible limit. So it is necessary to control the temperature of electronic unit within its permissible temperature limit for its reliable operation. The control of temperature in electronic unit especially which are used in air vehicles is very important because at high altitude the proper functioning of the electronic unit is very essential for the air vehicle to achieve its mission objectives. . Thermal analysis helps to design the electronic equipment for its better operation and we can apply effective method for control the temperature in air vehicle.

Problem Description

The unmanned air vehicle has two electronic units (EUs), one for controlling gimbal and other for controlling the unmanned air vehicle. The new Electronic Unit shown in Fig.1 has been developed by combining both the units and the earlier working features are improved for accurate performance. The working temperature limits of EU are -40^{o}C to +55^{o}C. The Electronic Unit is designed for a number of compacting actuators as per the mission requirements. The overall size of the mechanical chassis is 220mmx150mmx 180mm. It has to dissipate the heat load of 100 watts and heat load distribution is given below Table.1

TABLE.1 Heat load distribution for EU

Board/Module	Dimension	Total Power(W) (W)
PCB-1	120x144x1.6	5
PCB-2	120x144x1.6	5
PCB-3	120x144x1.6	5
PCB-4	120x144x1.6	5
300W Amplifier-1	84x58x16	40
300W Amplifier-1	84x58x16	40
Total Heat Load		100

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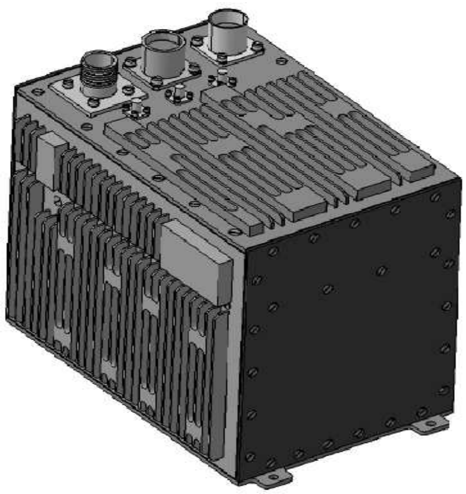

Fig.1 Three dimensional model of EU

Heat Transfer Correlations for Natural Convection & Radiation Cooling

Fluid tends to expand when heated, which results in a reduced density. In gravity field due to buoyancy effect the lighter fluid rises creating a movement in the fluid field called convection. Radiation is the transfer of energy by electromagnetic waves that are produced by bodies because of their temperature. A hot body radiates energy in all the directions. When this energy strikes another body, the part is absorbed & transformed into heat.

Natural convection and radiation heat transfer method is most easy to use than all other heat transfer methods, because no auxiliary equipment is required. The hot surfaces radiate energy directly to the cooler surroundings. At the same time, the air near hot surfaces is heated and rises and is replaced by cooled air. This convective air current provides additional heat transfer. The correlations used in natural convection and radiation are given below.

Natural Convection Correlations

The basic relation for free convection is given by:

$$N_u = C\,(G_r)^a\,(P_r)^b \quad (1)$$

Where:

$$N_u = \frac{hl}{k} \;:\; G_r = \frac{l^3 \rho^2 g\beta\Delta T}{\mu^2} \;:\; P_r = \frac{\mu\, Cp}{k}$$

Through rigorous experimental tests [2] of free convection with various fluids (both liquids and gases) flowing past horizontal cylinders and vertical plates, McAdams [2] for various geometries, orientation and flow conditions indicated by the magnitude of product ($G_r.P_r$). The simplified values are given in below Table.2

TABLE.2 Simplified Free Convection Equations for Air

Surface and orientation	Laminar flow $10^4 < G_r.\,P_r > 10^9$	Turbulent flow $G_r.\,P_r > 10^9$
(i) Vertical planes or cylinders	$1.42 * \left[\frac{\Delta T}{l}\right]^{0.25}$	$1.31[\Delta T]^{0.33}$
(ii) Horizontal cylinders	$1.32 * \left[\frac{\Delta T}{l}\right]^{0.25}$	$1.24[\Delta T]^{0.33}$
(iii)Horizontal plates: Heated plates facing upward or cooled plates facing downward	$1.32 * \left[\frac{\Delta T}{l}\right]^{0.25}$	$1.52\ [\Delta T]^{0.33}$
(iv) Horizontal plates: Heated plates facing downward or cooled plates facing upward	$0.59 * \left[\frac{\Delta T}{l}\right]^{0.25}$	....

Radiation Heat Transfer

The heat transferred by radiation is given by:

$$Q_r = \sigma\, f\, \varepsilon\, A\, (T_1^4 - T_2^4) \quad (2)$$

Equation (2) can also be written as:

$$Q_r = h_r\, A\, (T_{ss} - T_a) \quad (3)$$

Equating (2) and (3) we get that:

$$h_r\, A\, (T_{ss} - T_a) = \sigma\, F\, \varepsilon\, A(T_{ss}^4 - T_a^4) \quad (4)$$

$$\therefore h_r = \sigma\, F\, \varepsilon(T_1^2 + T_2^2)\,(T_1 + T_2) \quad (5)$$

The amount of heat transferred by available surface area considering Natural Convection and Radiation is given by.

$$Q = (\,h_c + h_r)\, A(T_{ss} - T_a) \quad (6)$$

Theoretical Calculations

Theoretical calculations are based on the assumption that the total heat generated inside the SEU i.e., 100watts is transferred to the chassis walls uniformly. From the chassis walls this heat is dissipated to the ambient by means of natural convection and radiation heat transfer modes. To augment the heat transfer through radiation, all the outside surfaces of the EU will be essentially coated black and the surfaces with black anodizing will have emissivity value of 0.9[3].

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Theoretical Results

By using equation (6) theoretical calculations has been carried out for different cases are as follows.

The steady state temperature of the chassis for an ambient temperature of 55 ^{0}C without any extended surfaces = 92.8 ^{0}C.

Fin Space Optimization

A question that often arises in the selection of a heat sink is whether to select one with closely packed fins or widely spaced fins for a given base area. A heat sink with closely packed fins will have greater surface area for heat transfer but a smaller heat transfer coefficient because of the extra resistance the additional fins introduce to fluid flow through the inter fin passages. A heat sink with widely spaced fins, on the other hand, will have a higher heat transfer coefficient but a smaller surface area. Therefore, there must be an optimum spacing that maximizes the natural convection heat transfer from the heat sink for a given base area WL, where W and L are the width and height of the base of the heat sink, respectively, as shown in Figure.2 When the fins are essentially isothermal and the fin thickness t is small relative to the fin spacing S, the optimum fin spacing for a vertical heat sink is given below [3].

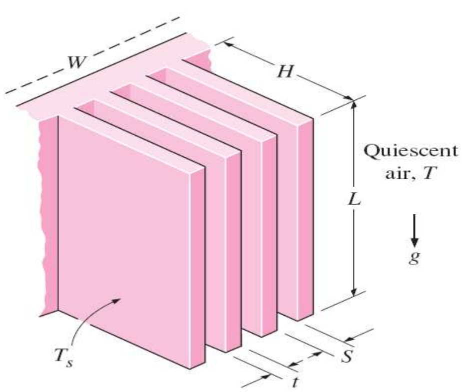

Fig.2 various dimensions of a finned surface

$$Nu = Gr*Pr \qquad (7)$$

$$Nu = Ra \qquad (8)$$

$$Sopt = 2.714\left(\frac{S^3L}{Ra}\right)^{0.25} \qquad (9)$$

$$Sopt = 2.714\left(\frac{L}{Ra^{0.25}}\right) \qquad (10)$$

The rate of heat transfer by natural convection from the fins can be determined from

$$Q = h\,(2nLH)(T_s - T_a) \qquad (11)$$

Where,

$$n = \frac{W}{(S+t)} \qquad (12)$$

Theoretical Calculations And Theoretical Results

Theoretical calculations are based on the assumption that the maximum permissible surface temperature of SEU is 80^{o}C at ambient temperature 55^{o}C. The steady state temperature

For the steady state temperature of 80^{o}C at an ambient temperature of 55^{0}C. By using the equations (10) and (11) optimized space value is 8.88 mm and total heat dissipated is 93.78 W. From the manufacturing point of view the S_{opt} is considered as 8.5 mm.

Effect of Fin Space Optimization

Effect of fin space optimization can be observed by using empirical correlations [3]

$$Ras = \frac{g\beta(Tss - Ta)S^3}{\nu^2}Pr \qquad (13)$$

$$Nu = \frac{hS}{k} = \left[\frac{576}{\left(Ras\frac{S}{L}\right)^2} + \frac{2.873}{\left(Ras\frac{S}{L}\right)^{0.5}}\right]^{-0.5} \qquad (14)$$

By using correlations available in literature here we calculated heat transfer co-efficient for s=6mm and optimum space S_{opt}=8.5

T_{ss}=80^{0}C=353K T_a=55^{0}C=328K S=6mm=0.006m

L=147mm=0.147m (length of fin)

$\beta = \frac{1}{Tf}$ $T_f = \frac{(353+328)}{2} = 340.5K$, $\therefore \beta$ =2.93685x10^{-03}

g=9.81 m/s^2 T_{ss}-T_a=25K S_{opt}=8.5mm

k=180w/m k

From heat transfer data hand book for Average temperature= $\frac{(80+55)}{2}$, T_{avg}=67.5^{0}C, properties of air has been selected.

ν=19.75x10^{-06} m^2/s Pr=0.6945

For S=6mm heat transfer coefficient has been calculated by using Eq.(13)and (14)

$\therefore$ h=12.957kw/m^2

Similarly For S_{opt}=8.5mm the heat transfer coefficient has been calculated by using Eq.(13)and(14)

$\therefore$ h=25.248kw/m^2

Governing Equations

The conjugate heat transfer and fluid flow on external surface were numerically modelled. Steady state continuity, momentum (Navier-Stokes) equations and energy equations are solved [4].

Continuity equation

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z} = 0$$

(15)

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Momentum equation

$$\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=v\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)-\frac{1}{\rho}\frac{\partial p}{\partial x} \quad (16)$$

Energy equation

$$\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{1}{\alpha}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad (17)$$

Boundary Conditions

The ambient temperature is specified as 55°C. Respective heat load values are applied to the components and PCBs. All modes of Heat transfer are considered (conduction, convection and radiation.) Outer surfaces of the chassis will be black anodized and hence for these surfaces the emissivity value is specified as 0.9, Surface View factor is considered as 1[2]. Air is taken medium for heat transfer. Gravity is ON along –y direction.

CFD Analysis

Theoretical calculations carried out are based on the assumption of uniform heat load. However, in practical situations the geometry, power dissipation, boundary conditions are non-uniform. So CFD analysis has been used to obtain the more accurate results. Heat load distributed has given in Table.1 for Electronic Unit and for Fin Space Optimized electronic Unit as shown in Fig.5 heat load distribution is given below Table.3

TABLE.3 Heat load distribution for space optimized EU

Board/Module	Dimension	Total Power(W) (W)
PCB-1	120x144x1.6	5
PCB-2	120x144x1.6	5
PCB-3	120x144x1.6	5
PCB-4	120x144x1.6	5
300W Amplifier 1	84x58x16	36.89
300W Amplifier 1	84x58x16	36.89
Total Heat Load		93.7801

Solution Procedure

Thermal analysis in SEU is obtained using Flo.EFD software. Following procedure has been carried out to analyse the SEU.

- 3D model of the SEU is imported from solid works 3D modelling software.
- The activities like removal of fillets, chamfers and holes are carried out to reduce the cell size and subsequently the computational time
- Heat loads were applied on the surface of the components.
- All modes of Heat transfer are considered (conduction, convection and radiation.)
- Solving the problem using Flo.EFD software.
- Post processing of the results
- Materials used are Aluminium Alloy 6061-0(SS) for housing.FR-4for all PCBs.
- Gravity is ON along -y direction.
- Outer surfaces of the chassis will be black anodized and hence for these surfaces the emissivity value is specified as 0.9.

Material Detail:

- Aluminium Alloy 6061-0(SS) for housing.

Thermal conductivity is 180 W/m K.

- FR4 for all PCBs .Equivalent thermal conductivity for PCBs is 0.25 W/m K.

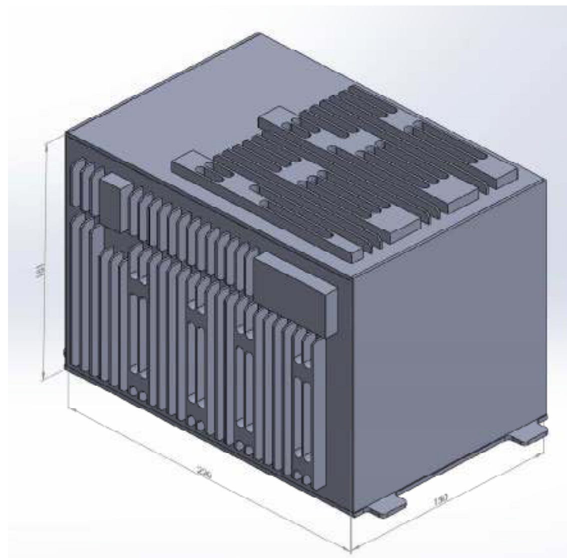

Fig.3 Three dimensional model of EU for Flo.EFD analysis

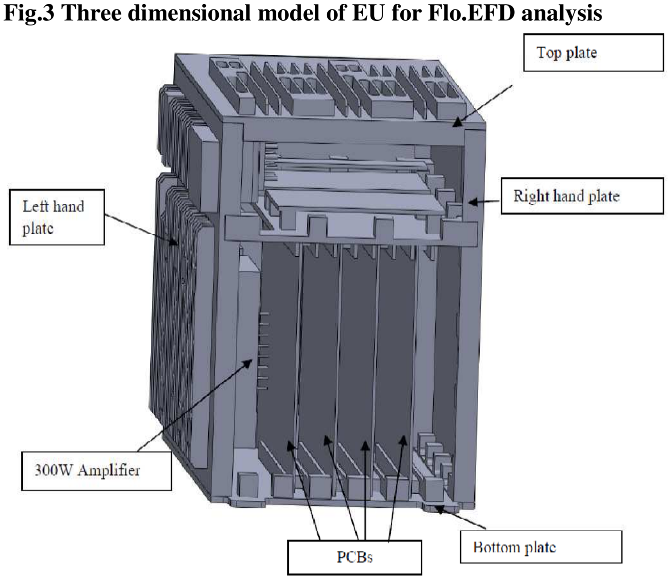

Fig.4 Construction details of Electronic Unit

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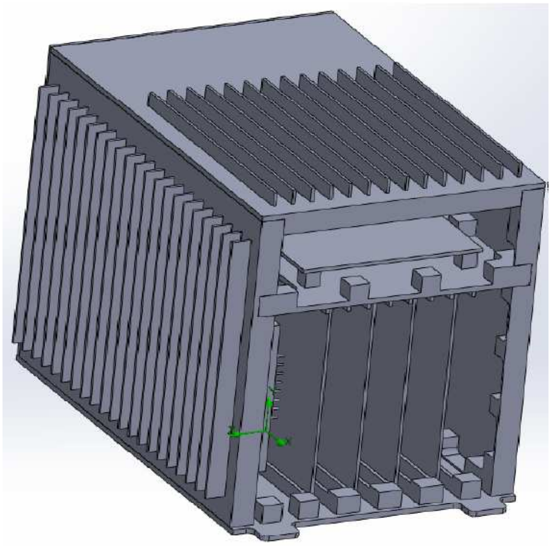

Fig.5 Space optimized EU (S_{opt}=8.5mm)

Computational Derails

Convergence is the property of numerical method to produce a solution which approaches the exact solution as the grid spacing, control volume size or element size is reduce to zero. The computation requires nearly 205 iterations for Servo Electronic unit and Fin Space Optimized SEU requires nearly 164 iterations for convergence (Velocity, Continuity and energy equations) for a 3D model of SEU. The iterations continuity the all residual error is smaller than 1e-08 Figure.6 and 7 shows meshed model of electronic unit and space optimized electronic unit respectively

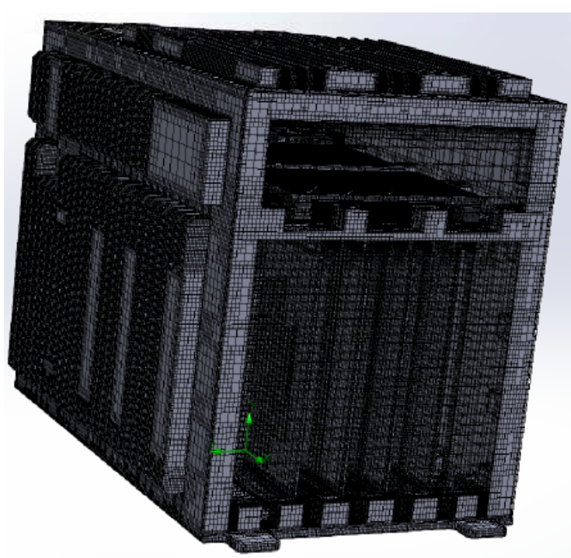

Fig.6 Three dimensional meshed model of EU

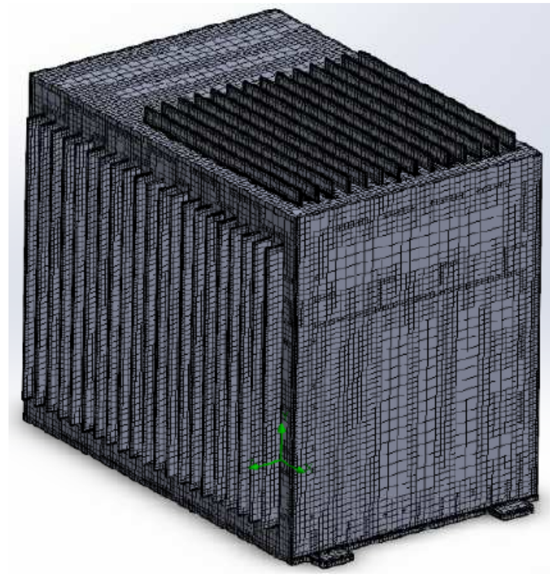

Fig.7 Three dimensional meshed model of space optimized EU

Results and Discussion

The fin optimization normally involves following geometry parameters:
Fin height H
Fin spacing S
Fin thickness T

The above are with respect to a base plate of given thickness, length and width [5].

In the current design and analysis, the internal space required and the external space constraints/space restriction by the user have pretty much dictated the fin height and length. For best heat transfer the base plate and the fin thicknesses have been kept at a reasonable minimum of 2mm based on manufacturability and strength considerations.

Thermal Contour for EU

Numerical simulation of SEU of size 20mmx150mmx180mm was carried out using Flo.EFD software. The temperature contour plots shown in figure 8, the maximum temperature for the SEU walls is observed at 300W amplifier mounting location for total heat load of 100W. The maximum surface temperature is 82.42^{0}C observed on left hand vertical plate of the EU. Inside the EU, the maximum temperature was observed on PCBs surface, which are mounted on bottom plate of SEU. Fig.9 shows that the PCBs mounted on the Bottom plate of SEU will reach a steady state temperature of 95.52^{0}C for an ambient temperature of 55^{0}C.

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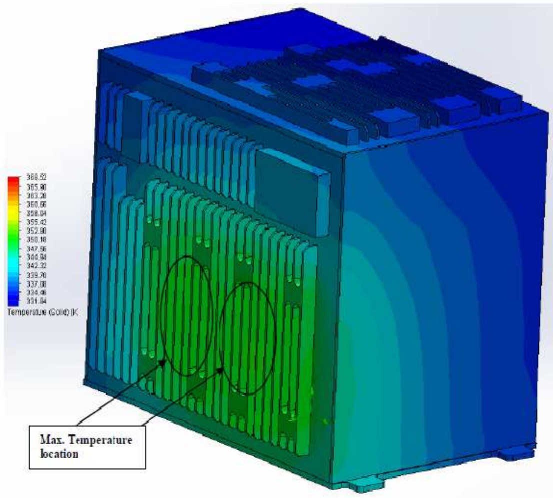

Fig.8 Temperature distributions on surface of EU

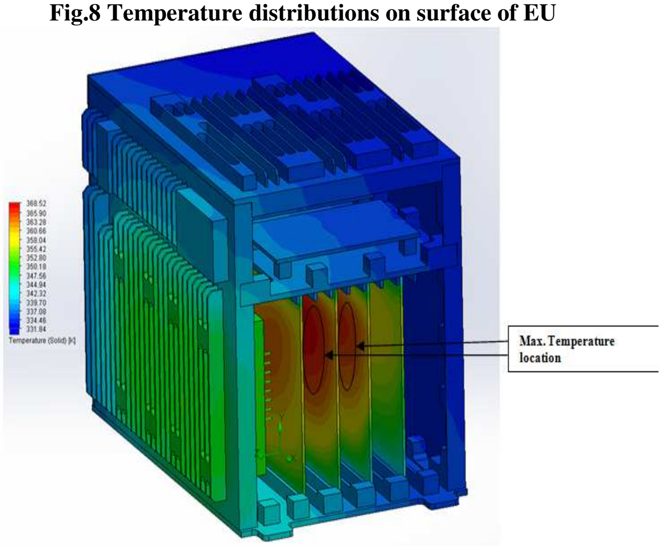

Fig.9 Maximum temperature on PCBs of EU (Maximum temperature = 95.52ºC)

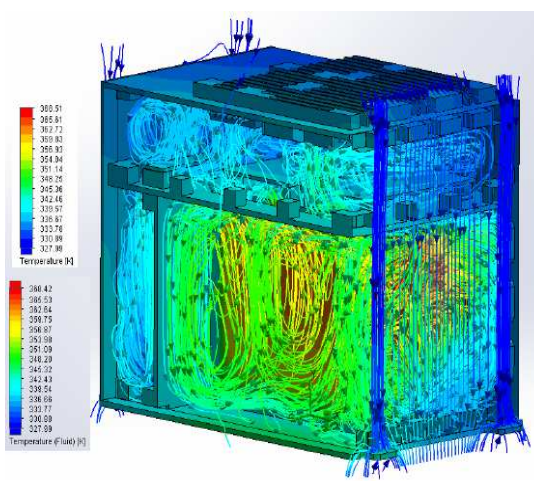

Fig.10 Flow distribution around the boards of EU

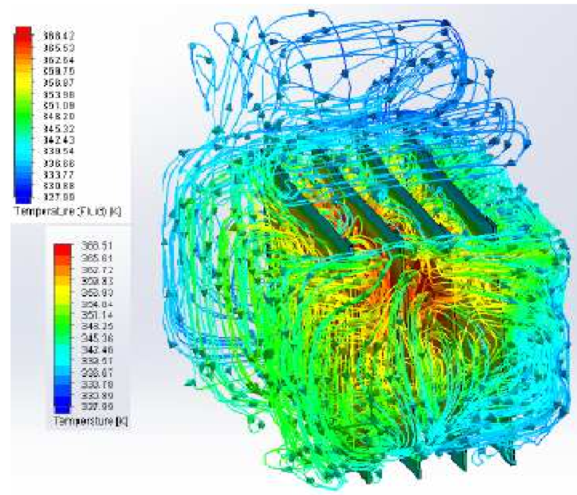

Fig.11 Flow distribution around the PCBs of EU

Figures.10 and 11 shows flow distribution around the boards of SEU and around the PCBs of SEU. Natural convection heat transfer takes place in air due to buoyancy effect.

Thermal Contour For Optimized Fin Spacing (S_{OPT}) EU

Numerical simulation was carried out for space optimized SEU of size 220mmx150mmx180mm. Space optimization was made for surface temperature T_S 80^0C for EU. And analysis was carried out using Flo.EFD software. The temperature contour plots shown in figure 12, the maximum temperature for the EU walls is observed at 300W amplifier mounting location for total heat load of 94.8W. The maximum surface temperature is 79.5^0C observed on left hand vertical plate of the EU. Inside the EU, the maximum temperature was observed on PCBs surface, which are mounted on bottom plate of EU. Fig.13 shows that the PCBs mounted on the Bottom plate of EU will reach a steady state temperature of 94.2 ^{0}C for an ambient temperature of 55^0C

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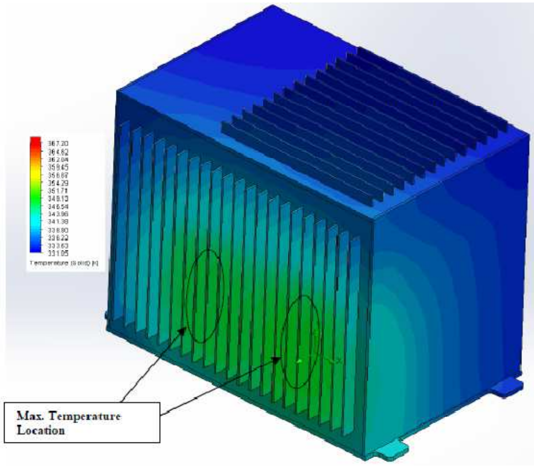

Fig.12 Temperature distribution on surface of space optimized EU

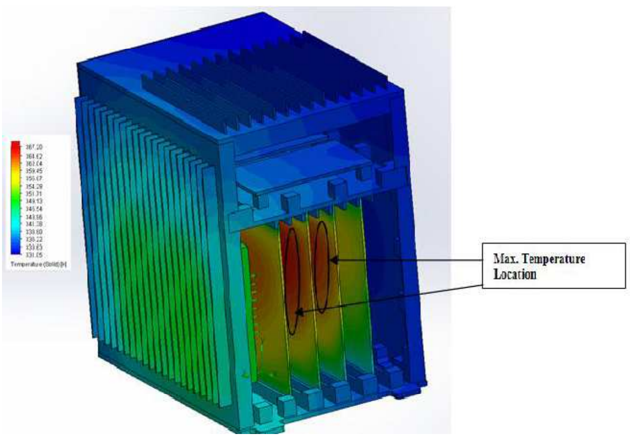

Fig.13 Maximum temperature on PCBs of S_{opt} EU (Maximum Temperature = 94.2ºC)

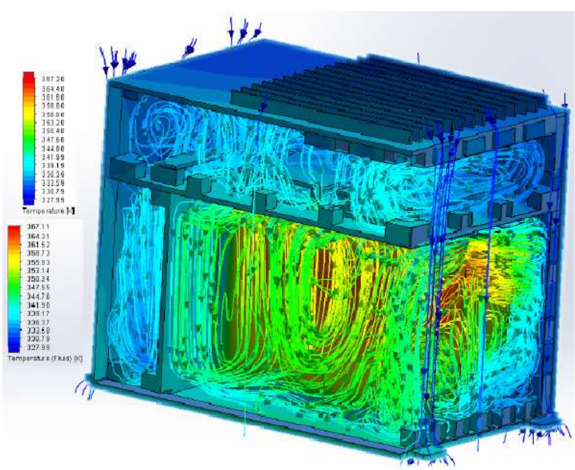

Fig.14 Flow distribution around the boards of S_{opt} EU

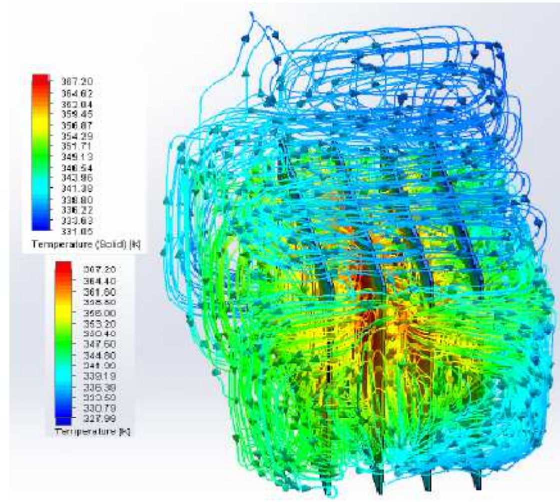

Fig.15 Flow distribution around the PCBs of S_{opt} EU

Figure.14 and 15 shows flow distribution around the boards of S_{opt} EU and around the PCBs of S_{opt} EU. Natural convection heat transfer takes place in air due to buoyancy effect.

Results Comparison

Comparison was carried out between the theoretical results and Flo.EFD results for finned and space optimized EU. It is found that the Flo.EFD results show close matching with theoretical results which are calculated by using the empirical correlations. Table 3 below gives the comparison between Flo.EFD results and theoretical results

TABLE.3 comparisons between Flo.EFD results and theoretical results

Type of SEU surface	Ambient temperature ^{0}C	Steady state temperature (Theory)^{0}C	Steady state temperature (Flo.EFD)^0 C
Finned surface	55	80.73	82.42
Space optimized surface	55	80	79.5

By comparing the results calculations obtained from empirical equations (13) and (14), the heat transfer coefficient obtained for S=6mm is 12.057kw/m^2 whereas the heat transfer coefficient obtained for fin space optimized value S_{opt}=8.5mm is 25.248kw/m^2. From this comparison it may be surmised that the fin optimized surface will dissipate heat adequately so as to maintain

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the temperatures to within the design goals. Similarly CFD comparison has been made from table 3 for S=6mm and S_{opt}=8.5mm, the fin space optimized surface temperature is less than the surface without space optimization for nearly same steady state temperature (Theoretical).

Conclusions

In the present work a finned heat sink for Electronic Unit (EU) of an unmanned air vehicle has been designed using the empirical heat transfer correlations by considering natural convection and radiation heat transfer modes for a heat load of 100W. The ambient temperature is taken as 55°C. Heat transfer calculations were carried out for plain surface and then finned surface. Hand calculations have been carried out for plain surface and surface temperature was found as 92.8°C, which is not acceptable. Hence it is clear that the surface is to be further cooled to limit the surface temperature below 85ºC.

This job is done by adding rectangular fins to the surface. Heat transfer calculations were carried out for finned surface and the maximum surface temperature is estimated as 80.73ºC, which is well below the maximum permissible temperature. Further space optimization has been made for given heat sink to bring down the surface temperature further below. As part of the project the 3D modelling of all the parts of the electronic unit was carried out in SolidWorks2012 software. Using Flo.EFD software the CFD model is generated and analysed to obtain the total thermal mapping of the above electronic unit. The CFD results were used to validate the results obtained through the excel spread sheet program developed based on the empirical correlations available in the literature. And it is found that the Flo.EFD results almost matching with theoretical results which are calculated by using the empirical correlations.

NOMENCLATURE

h : Heat transfer coefficient, (W/m^2-K)
l: Characteristic length, (m)
k: Thermal conductivity, (W/m-K)
ρ: Density, (kg/m^3)
g: Acceleration due to gravity, (m/s^2)
β: Coefficient of volume expansion. (-K)
μ: Dynamic viscosity, (kg/m-s)
C_p: Specific heat. (kJ/ kg K)
Q: Heat transfer rate in watts (W)
h_c :Convection heat transfer coefficient, (W/m^2-K)
h_r :Radiation heat transfer coefficient, (W/m^2-K)
A: Available surface area for heat transfer (m^2)
$T_{ss: Steady}$ state temperature (K)
T_a: Ambient temperature (K)
n: number of fins on the heat sink.
Ts: surface temperature of the fins.
t: fin thickness.(m)
W: width of heat sink.(m)
L: length of heat sink.(m)
S: fin spacing.(m)
S_{opt}: optimum fin spacing.(m)
N_u: Nusselt number
G_r: Grashoff number
P_r: Prandtl number
Q_r: Radiation heat transfer rate (W)
Ra: Rayleigh number
σ: Stefan-Boltzmann constant($5.67x10^{-8}$ W/m^2-k^4)
F: Shape factor
ε: Emissivity
ΔT: Temperature difference (T_{ss}-T_a)
Ras: Rayleigh number for space

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