Design And Computational Analysis Of Heat Sinks Next Generation Avionic Systems

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Abstract—The multiple challenges in the avionic industry which needs the thermal management system for its electronic components. To enhance the performance reliability and safety of its systems an innovative technology required to reduce the temperature developed at the source. Extended surfaces are called as heat sinks with different array systems under design considerations to enhance the performance an electronic component as if the internal temperature of the component arises by its application the module may fail to work efficiently which persistently challenge the engineers towards the heat dispersion process. The perforated heat sink designs are reviewed with conventional designs with the available innovations. Numerical and computational designs are to be tested to enrich the optimal heat sink design for the electronic component. Challenges for IC’s which operate at extreme temperatures as high of 125C to -55C in hot-to-cold weather condition of the aircraft industry, every 2C rise in temperature decreases by 10% of a silicon chip in terms of reliability. To reduce this problem multiple design parameters are to be tested for different geometrical extended surfaces, correlation between numerical models with simulation analysis can reduce both the time and money in evaluating heat transfer coefficient

Index Terms—Fin geometries, Heatsinks, Computational Fluid Dynamics, heat transfer, thermal management, fin array, extended surfaces, conduction, convection.

1 INTRODUCTION

Transforming of thermal energy from a heated body to a colder body with the help of an object or fluid, transfer of thermal energy is also known as heat transfer. Exchange of heat occurs till body and the surroundings reach the same temperature. According to the second law of thermodynamics, where there is a temperature difference between objects in heat transfer between them can never be stopped. The study of heat transfer deals with the rate at which such energy is transferred. Heat is thus the energy in transit between systems which occurs by virtue of their temperature difference when they communicate. Obviously, conditions of temperature disparity and communication must be fulfilled simultaneously for heat interaction between systems to occur. The finite temperature difference existing between the systems makes the process of heat exchange irreversible, i.e. flow of heat cannot be reversed. Heat transfer generally recognizes three different modes of heat transmission conduction, convection and radiation. These three modes are similar in that a temperature differential must exist and the heat exchange is in the direction of decreasing temperature. Each method has its, different physical picture and different controlling laws. Thermal convection is a process of energy, transport affected by the circulation or mixing of a fluid medium (gas, liquid or a powdery substance).

Convection is possible only in a fluid medium and is directly linked to the transport of medium itself. With respect to the origin, two types of convection are distinguished; forced and natural convection. The convective heat transfer between a surface and an adjacent fluid is prescribed by Newton’s law of cooling. In the heat transfer study, the surface that extends from an object is known as a fin. This is used to increase the rate of heat dissipation from or to the environment by increasing the rate of convection. The total of convection, conduction, or radiation of an object decides the amount of heat it dissipates. It increases with the difference in temperature between the environment and the object, also increasing the convection coefficient of heat transfer, or increasing the surface area. But, an increase in the area also causes increased resistance to the heat flow. Hence, the coefficient of heat transfer based on the total area (the base and fin surface area) comes out to be less than that of the base without fins, at the same temperature difference. If there is an increase in the area of surface proportionately more than the decrease in the heat transfer coefficient, the total heat dissipation rate increases.

2.1 Design

The heat exchanger domain consists of three connected channels: Entrance section, pin-fin section and exit section. The pin-fin section consists nine rows and seven columns of in-line and staggered pin-fins with axes perpendicular to the flow. Three different pin shapes are considered: cylindrical, rectangular and drop-shaped. These different pin shapes are arranged on a base plate 57.656 X 36.04 X 2 mm² with a distance of S1 = 7.207mm and S_T = 3.604mm. In inline arrangement there are 63 pin fins and staggered 32 pin fins. Mean radius of different fins is taken as 1.15mm and the length of fin as 23mm.

2.2 Computational analysis

By using CATIA software the fin models are designed and the data were passed to the ANSYS-FLUENT software for various analysis. No special boundary conditions were applied to pins. Since the each pin has a fluid contact on all sides of the solid region. An enclosure is created around it with a proportionate ratio from hydraulic diameter. When a grid containing this type of wall zone is imported into FLUENT, a “shadow” zone will automatically be created so that each side of the wall is a distinct wall zone. The governing equations solved by Fluent are the Navier-Stokes equations combined with the continuity equation, the thermal equation and the standard k-€ turbulence model with standard wall function was set for each mod-
el. Once the analyses are completed, the resulting data can be easily evaluated by the Fluent postprocessor.

2.3 Numerical analysis
The appropriate equation for the convective heat transfer between a surface and an adjacent fluid is prescribed by Newton’s law of cooling:

\[ Q = h A (T_s - T_f) \]

The convective heat transfer coefficient \( (h) \) can be defined as the rate of heat transfer between a solid surface and a fluid per unit surface area per unit time difference.

\[ h = \frac{q}{\Delta t} \]

Reynold’s number = \( \rho v^2 L / \mu \)

Nusselt number = \( Nu = \frac{hD_h}{k} \)

2.4 Fin Efficiency
Fin efficiency is defined as the actual heat transferred by the fin, divided by the heat transfer where the fin is to be isothermal. The performance of the fins can be determined in three different ways. They are

1. Effectiveness: The ratio of the heat transfer rate through the fin to the heat transfer rate of the object if it had no fin. The formula for this is

\[ \eta_f = \frac{\dot{Q}_f}{h A_f \theta_b} \]

2. Fin Efficiency: The Fin Efficiency must be always less than unity and its formula is

\[ \eta_f = \frac{\dot{Q}_f}{h A_f \theta_b} \]

3. Overall Surface Efficiency: The Overall surface Efficiency determines the performance of heat sinks

\[ \eta_o = \frac{\dot{Q}_b}{h A_f \theta_b} \]

Where:

- \( h \) = convection heat transfer coefficient, W/m²-K
- \( A_s \) = heat transfer surface area, m²
- \( T_s \) = temperature of the surface, °C
- \( T_f \) = temperature of the fluid sufficiently from the surface, °C
- \( q \) = amount of heat transferred (heat flux), W/m²
- \( h \) = heat transfer coefficient, W/(m²·K)
- \( \Delta T \) = difference in temperature between the solid surface and surrounding fluid area, K.

\( v \) is the max velocity of the object relative to the fluid (m/s)

\( L \) is a characteristic linear dimension,

\( D \) hydraulic diameter when dealing with river systems) (m)

\( \mu \) is the dynamic viscosity of the fluid (kg/(m·s))

\( \nu \) (nu) is the kinematic viscosity (\( \nu = \mu / \rho \)) (m²/s)

\( \rho \) is the density of the fluid (kg/m³).

\( K \) is the thermal conductivity of the material

\( L_f \) is the fin height (m)

\( T_f \) is the fin thickness (m)

\( W \) is the width of the fin

\( T_b \) is the thickness of the base

\( T_i \) is the thickness of the fin

\( A_b \) is the area of the base

3 Sections
The present work deals with heat transfer through natural convection. The circulation of the fluid medium is caused by buoyancy effects i.e., by the difference in densities of the cold and the heated particles due to the heat energy the hotter particles become less denser than the colder particles so they move upwards providing the natural convection process. The high conductive heat sink is arranged over a heat source to decrease ejection of heat from the source usually the heat sink is prefered more larger than the heat source or the electrical component that dissipates heat energy. The design section of the perforated fins is as shown in the below figures.
performance is found by the analysis softwares

4 RESULTS AND DISCUSSION

From the above analysis the theoretical and computational calculations shows that more heat transfer rate is observed in circular perforated pin fins compared among the models. The contours of heat transfer show that transfer rate is maximum for perforated circular fins, presence of perforations increase the surface area which results in high heat transfer rate. Figures.

**Fig: Temp contours circular pin fin arrangements (Inline and Staggered)**

**Fig: Temp contours drop shape pin fin arrangements (Inline and Staggered)**

**Fig: Temp contours circular pin fin arrangements (perforated Inline and Staggered)**

**Fig: Temp contours drop shape pin fin arrangements (perforated Inline and Staggered)**

**Graph: Variation of Heat Transfer with Velocity of Different Solid Pin Fins in Staggered Arrangement**

**Graph: Variation of Heat Transfer with Velocity of Different Perforated Pin Fins in Inline Arrangement**

**Graph: Variation of Heat Transfer with Velocity of Different Perforated Pin Fins in Staggered Arrangement**

**Graph: Variation of Heat Transfer with Velocity of Different Solid Pin Fins in Inline Arrangement**

**Graph: Variation of Heat Transfer with Velocity of Different Solid Pin Fins in Staggered Arrangement**
Different with and without Perforated Pin Fins in Different Arrangements

5 CONCLUSION
Thermal analysis on Circular and drop is carried out with and without perforated Pin fins. Heat transfer rate is increased by decreasing the Reynolds number by comparing inline and staggered ratios and interfin distance ratio the friction factor is varying.

6 REFERENCES

7 Theorems and Proofs

ENERGY BALANCE:

CONSIDER A SMALL LENGTH OF ΔX IN BETWEEN THE FIN. BEFORE ΔX THE HEAT TRANSFER IS \( Q_x \) AND AFTER THE ΔX THE HEAT TRANSFER IS \( Q_{x+\Delta X} \) AS SHOWN IN THE BELOW FIGURE.

Fig 4.1: Heat Transfer Through a Fin

THE TOTAL HEAT TRANSFER THROUGH FIN IS \( Q_x = Q_{x+\Delta X} + Q_{\text{CONV}} \)

\[
\frac{dQ_x}{dx} + \frac{\Delta Q}{\Delta x} = \frac{\partial Q_x}{\partial x}
\]

(FIN EQUATION)

\[
M^2 = \frac{hP}{KA} \quad (\text{FIN PARAMETER})
\]

BOUNDARY CONDITIONS

\( X=0; \quad T=T_B \quad \Theta=\Theta_B \)

\( X=L; \quad -KA\frac{\partial \Theta}{\partial x} = h\Theta \)

GENERAL SOLUTION TO EQUATION NO (7) IS

\( \Theta = A \cosh mx + B \sinh mx \)

DIFFERENTIATING THE EQUATION GIVES THE HEAT TRANSFER THROUGH FIN

\[
\dot{Q}_\text{FIN} = \frac{KAM\Theta_d \cosh (L-x) + \frac{h}{K} \sinh (L-x)}{\cosh L + \frac{h}{K} \sinh L}
\]

AT \( X=0 \)

\[
\dot{Q}_\text{FIN} = \frac{M \cosh L + \frac{h}{K} \sinh L}{\cosh L + \frac{h}{K} \sinh L}
\]

\( \dot{Q}_\text{TOTAL} = \dot{Q}_\text{FIN} + \dot{Q}_\text{BASE} \)

\( \dot{Q}_\text{TOTAL} = \dot{Q}_\text{FIN} + H(\text{AREA OF BASE PLATE} - \text{AREA OF FINS}) \Theta_B \)

\( \dot{Q}_\text{TOTAL} = \dot{Q}_\text{FIN} + H(A_B - A_f) \Theta_B \)

CALCULATION OF HEAT TRANSFER

REYNOLDS NUMBER:

\[
Re = \frac{\text{Inertial forces}}{\text{Viscous forces}} = \frac{\rho VL}{\mu} = \frac{V \times D}{\Theta}
\]
The flow is laminar, hence the Nusselt number formula is

\[ \text{Nu} = 0.3 + \frac{0.62 \text{Re}^{1/4} \text{Pr}^{1/3}}{1 + \left( \frac{\text{Re}}{230000/\text{Pr}^{2/3}} \right)^{1/5}} \]

Prandtl number at 50\(^0\) C is 0.698

By substituting all the values, we get Nusselt number as \( \text{Nu} = 8.335 \)

With the help of Nusselt number, the heat transfer coefficient can be calculated as

\[ h = \frac{\text{Nu} \times K}{D} = 10.41 \text{ W/m}^2\text{K} \]

Hence, the heat transfer through fins is calculated as

\[ Q_{\text{FIN}} = \frac{M c \text{osh}L + \frac{h}{\sin h L}}{\cosh L + \frac{h}{\sin h L}} \]

Where \( M = (H \times P \times K \times A)^{0.5} \); \( M = \frac{KP}{V^2} \)

The values of \( M \) and \( m \) are tabulated for different fins.

**Table 4.1: \( M \) and \( m \) values for different fins**

<table>
<thead>
<tr>
<th></th>
<th>CIRCULAR</th>
<th>SQUARE</th>
<th>DRAPE</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOLID</td>
<td>PERFORATED</td>
<td>SOLID</td>
<td>PERFORATED</td>
</tr>
<tr>
<td>M</td>
<td>0.062</td>
<td>0.0681</td>
<td>0.020</td>
</tr>
<tr>
<td>M</td>
<td>1775</td>
<td>1063</td>
<td>2510</td>
</tr>
</tbody>
</table>

By substituting all the values in the formula, the heat transfer through the fin can be determined. To calculate the total heat transfer find out the heat transfer through base plate and add the heat transfer.