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Finite Difference Simulation of Fin for Engine Cooling

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ABSTRACT

Finite difference simulation of horizontal fin with an aspect ratio of 3 has been conducted in the present study for cooling of electronic equipment. Two-dimensional heat conduction equation governs the physical problem. Three different materials Aluminium, Copper and Nickel will be studied to see the effect of thermophysical property on the cooling rate of fin at particular given temperature range or heat flux range. Appropriate boundary conditions have been adopted which represent the actual physical scenario of a fin. MATLAB used to simulate the governing equation and boundary conditions.

Keywords: Fin length, Heat conduction equation, Temperature distribution, Thermal properties, Taylor's series

INTRODUCTION

Fins are used in large number of applications to increase the heat transfer from hot surfaces. Typically, the fin material should have high thermal conductivity. The fin is exposed to a flowing fluid that cools it, with the high thermal conductivity allowing increased heat being conducted from the wall through the fin. Fins surfaces are broadly utilized as a part of industrial applications, for example, refrigeration, automobile, chemical process equipment, electronic equipment and electrical chips.

We know that just in case of internal combustion engines and electronic equipment, combustion of air and fuel takes place inside the engine cylinder and hot gases are produced. The temperatures of gases are going to be around 2300-2500°C. This is often a really hot temperature and will result into burning of oil film between the moving components and will result it seizing. So, this temperature should be reduced to regarding 150-200°C at that the engine can work most expeditiously. An excessive amount of cooling is additionally not efficiently since it reduces the thermal efficiency. So, the item of cooling system is to stay the engine running at its best in operation temperature. To avoid warming, and therefore the resulting unwell effects, the heat transferred to an engine component should be removed as quickly as attainable and be sent to the atmosphere. It'll be correct to mention the cooling system as a temperature regulation system. It ought to be remembered that abstraction of heat from the operating medium by means of cooling the engine elements may be a direct thermodynamic loss.

Extensions on the finned surfaces is used to increases the surface area of the fin. When the surface area increase, then more fluid contact will increase the rate of heat transfers from the base surface as compare to fin without the extensions provided to it. For ordinary fins, the thermal conductivity is assumed to be constant, but when temperature difference between the tip and base of the fin is large, the effect of temperature on thermal conductivity must be considered. Also it is very realistic that we consider the heat generation in the fin (due to electric current or etc.) as a function of temperature.

LITERATURE REVIEW

In spite of the very fact that the heat transfer from fins has been the subjected to various experimental and theoretical investigations, few systematic constant studies may be found within the literature. An extensive review and discussion of work done on the convective heat transfer in electronic equipment cooling was bestowed by Incropera [1], summarizing numerous convection cooling choices. A good variety of analytical and experimental work has been administered on this drawback since Elenbaas [2] initial introduced the matter of natural convection between vertical parallel plates. One of the earliest studies of the heat transfer from fin arrays is that of Starner and McManus [3] who presented heat transfer coefficients for four distinguish dimensioned fin arrays with the base vertical, forty five degrees and horizontal. They showed that incorrect application of fins to a surface really might reduce the entire heat transfer to a value below that of the base alone. The same experimental study was conducted by Welling and Woolbridge [4] on rectangular vertical fins. They examined optimum values of the ratio of fin height to spacing. In the previous two studies the fin length was constant. Average heat transfer coefficients and flow field observations were bestowed by Harahap and McManus [5] for two totally different fin lengths. They also investigated correlations representing all their experimental information by creating use of non -dimensional numbers and every one the relevant dimensions of fins. From flow pattern observations they conclude that the one chimney flow pattern yields higher rates of heat transfer.

Jones and Smith [6] studied the variations of the local heat transfer constant for equal vertical stabiliser arrays on a horizontal base over a large vary of fin spacing. A simplified correlation, an optimum arrangement for max heat transfer and a preliminary design technique were suggested. For a large vary of temperatures, Rammohan and Venkateshan [7] created an interferometric study of warmth transfer by free convection and radiation from a horizontal fin array. Correlations helpful for thermal design were presented. The authors stressed the importance of the mutual interaction between free convection and radiation. Recently, associate experimental investigation was meted out by Yüncü and Anbar [8] on natural convection heat transfer from rectangular fin arrays on a horizontal base for effects attributable to fin spacing, fin height and temperature distinction between fin base and surroundings. Optimum fin spacing values and a correlation were reportable. However, results are only for a set fin length and fin thickness. One among the primary numerical studies on this subject was meted out by Sane and Sukhatme [9]. Governing equations were resolved numerically employing a finite distinction technique. They obtained smart agreement with experimental information. Additionally, flow visualization studies were meted out so as to depict the zone during which the one chimney pattern happens [10]. Chaitanya et al [11] has analysed the thermal properties by varying geometry, material and thickness of cylinder fins by using ansys work bench. Transient thermal analysis determines temperatures and other thermal quantities that vary over time. Aluminium Alloy A204 is used and has thermal conductivity of 110-150W/mk. and also using Aluminium alloy 6061 which have higher thermal conductivities.

The present work investigates the results of a wide range of geometrical parameters to the heat transfer from horizontal fin arrays. Effects attributable to changes in fin spacing, fin height, fin length and temperature difference between fin and surroundings area unit investigated along, hence, preventing mistakes possible attributable to incorrect assumptions and usage of a constant value for one of the geometrical parameters.

NUMERICAL ANALYSIS

Physical Domain

As we know that heat lose by fin is due to forced convection, to reach to the reality constant heat flux and convective boundary conditions have considered on the walls of the fin shown in fig 1. An aspect ratio L/H=3.0 will be considered for the fin size. From the fig it can be observed that at left wall, heat flux boundary condition will be considered which represents that fin is attached to the surface of the automobile engine from left wall, which can also be seen from the arrow direction, while on other three walls convective boundary condition will be considered which represents that heat losses to the surroundings by convection, the same can be understand by arrow direction which are outward in the fin

Governing Equation

Two modes of heat transfer, convection and radiation will be neglected inside the fin, because present study is focusing on designing of a solid-fin and it is well known fact that inside the solid body heat transfer by conduction dominates over convection and radiation. So the two-dimensional heat conduction equation will govern the physical problem,

$$\rho C_p \frac{\partial T}{\partial t} = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{1}$$

A simplified form of the above governing equation can also be written as

$$\frac{\partial T}{\partial t} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{2}$$

Where: $\alpha = k/\rho C_p$ is thermal diffusivity in m^2/s, k is thermal conductivity in W/m-K, ρ is density in Kg/m^3, C_p is specific heat J/Kg-K, T is temperature in Kelvin (K), t is time in seconds.

Boundary Conditions

At x = 0 $\quad q = -k\frac{\partial T}{\partial x}$

At y = 0 $\quad q = k\frac{\partial T}{\partial y} = h(T - T_\infty)$

At x = L $\quad q = k\frac{\partial T}{\partial x} = -h(T - T_\infty)$

At y = H $\quad q = k\frac{\partial T}{\partial y} = -h(T - T_\infty)$

Where q = quantity of heat transfer (W/m^2), h = heat transfer coefficient (W/m^2-K), T_∞ = Ambient temperature (K)

Problem Formulation

The above equation (2) is in the form of partial differential equation, to obtain a solution it must be expressed in the form of approximate solution so that a digital computer which can perform only arithmetic and logical operations can be employed to obtain a solution. Taylor series expansion will be considered to convert these partial derivative terms in algebraic form, which can be discretized either by explicit scheme or by implicit scheme,

$$\frac{T_{i,j}^{n+1}-T_{i,j}^{n}}{\Delta t}=\alpha\left(\frac{T_{i+1,j}^{n}+T_{i-1,j}^{n}-2T_{i,j}^{n}}{(\Delta x)^2}+\frac{T_{i,j+1}^{n}+T_{i,j-1}^{n}-2T_{i,j}^{n}}{(\Delta y)^2}\right) \tag{3}$$

$$\frac{T_{i,j}^{n+1}-T_{i,j}^{n}}{\Delta t}=\alpha\left(\frac{T_{i+1,j}^{n+1}+T_{i-1,j}^{n+1}-2T_{i,j}^{n+1}}{(\Delta x)^2}+\frac{T_{i,j+1}^{n+1}+T_{i,j-1}^{n+1}-2T_{i,j}^{n+1}}{(\Delta y)^2}\right) \tag{4}$$

Equation 3 and 4 represents the explicit and implicit discretization of equation 2. In explicit scheme only one term is at the next time level (n+1) which is an unknown quantity while all other terms are at same time level (n) which are known, while in implicit scheme spatial derivative terms are also at the next time level (n+1) which makes it difficult to handle in computer code. Explicit scheme is simple to code but requires a condition to satisfy which is known as stability condition and leaves a limit on the time step for a particular grid selection while implicit scheme is free of this stability condition represented in eq.

$$\alpha\times dt\left(\frac{1}{(dx)^2}+\frac{1}{(dy)^2}\right)\le\frac{1}{2} \tag{5}$$

In the present study explicit scheme will be considered for simplicity of the scheme, As we know that term at time 'n+1' is unknown while terms at time 'n' are known. So from above equation it can be observed that all the terms on right hand side are at time 'n' (known) while only one term at time 'n+1' is there in left hand side which can be easily calculated as,

$$T_{i,j}^{n+1}=T_{i,j}^{n}+\alpha\Delta t\left(\frac{T_{i+1,j}^{n}+T_{i-1,j}^{n}-2T_{i,j}^{n}}{(\Delta x)^2}+\frac{T_{i,j+1}^{n}+T_{i,j-1}^{n}-2T_{i,j}^{n}}{(\Delta y)^2}\right) \tag{6}$$

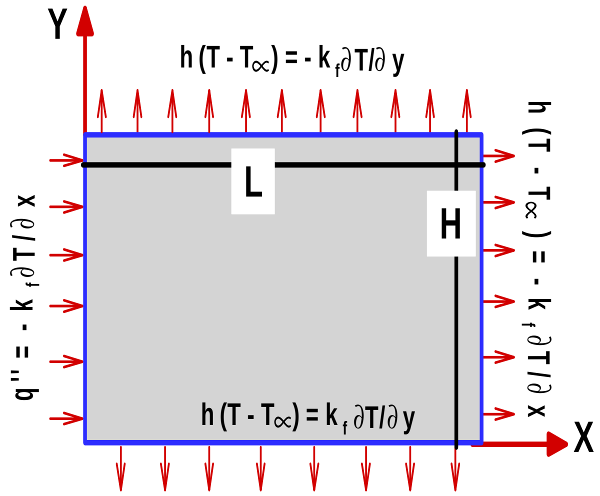


Fig.1. Schematic of horizontal fin

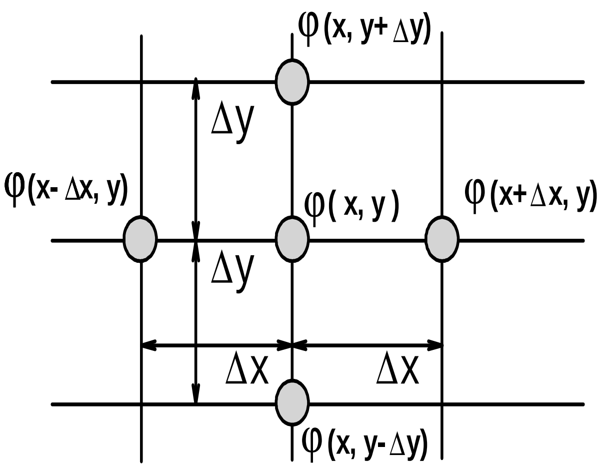


Fig.2 Grid element layout

Mathematical Technique

Finite difference scheme will be used to discretize the two-dimensional heat conduction equation. Collocated grid element in which all the properties are stored at the same location shown in fig. 2 will be used in the present problem. Graphical representation of the above equation is shown in fig. 2. Element shown in figure, represents a collocated grid where φ can be any dependent variable. Element in fig. is a five-node grid element, φ (x, y) represents the focal node while all other nodes have been represented corresponding to this.

Table -1 Thermophysical Properties

Material	Specific heat (KJ/Kg-K)	Density (Kg/m^3)	Thermal conductivity (W/m-K)
Aluminium	0.8963	2700	237
Copper	0.383	8920	401
Nickel	0.445	8908	91
Heat transfer coefficient 'h'=10.0 W /m^2-K			

RESULT AND DISCUSSION

Grid-Validation Test

A grid validation test on Aluminium has been conducted to check that at which grid-size results will be accurate. Increment in grid size or reaching to a finer grid means reduction of step size. From the fig it can be observed that with increment in grid-size cooling rate is changing. A grid-size of 61×61 has been found sufficient to capture temperature distribution inside fin.

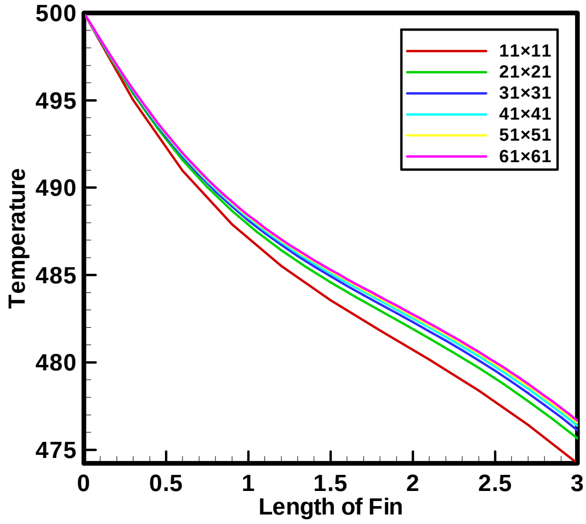


Fig.3 Effect of Grid-size

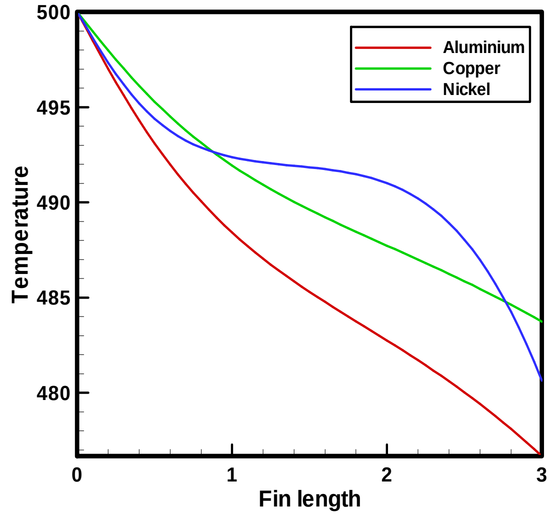
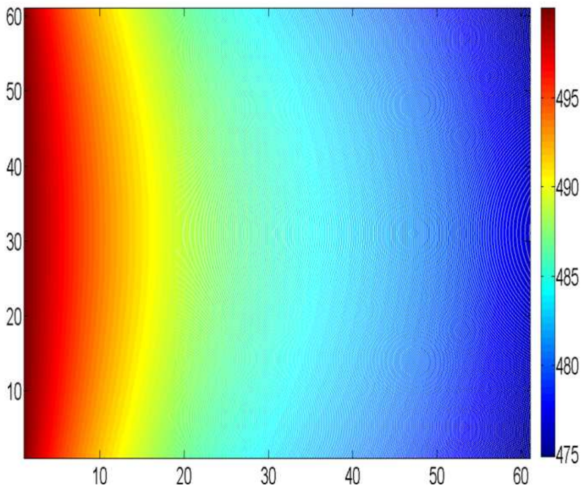
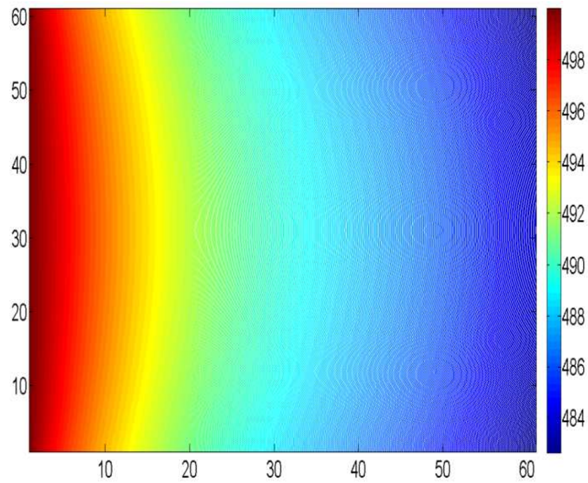


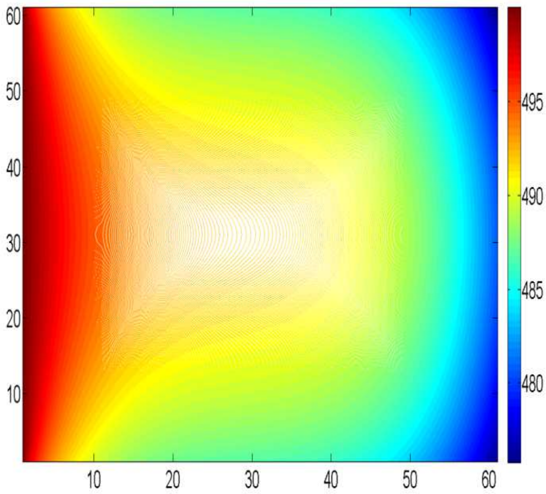
Fig.4. Effect of Fin material



(a) Aluminium



(b) Copper



(c) Nickel

Figure 5 (a-c) Temperature contours for different materials

Effect of Fin materials

Figure 4 represents the cooling rate for three different materials considered. This figure has been plotted on a grid-size of 61×61 which has been selected by grid-validation test. From the figure one can notice that highest cooling rate is for aluminium and lowest cooling rate for copper.

Figure 5 (a-c) represents the temperature contours inside the fin for different materials considered. Three different materials Aluminium, Copper and Nickel have been considered, these materials possess different thermal properties like thermal conductivity, specific heat and density. These results have been plotted at particular interval of time. From fig. 5 one can notice that highest cooling rate is for Aluminium which can be observed from the contour levels in the right side of the figures same observed in fig. 4.

CONCLUSION

In this paper formulation of two-dimensional heat conduction equation has been done two solve a rectangular fin problem. Two-dimensional heat conduction equation has been converted into algebraic form using Taylor's series expansion. A collocated grid element has been used. Effect of three different fin materials will be studied.

- A grid-size of 61×61 has been found appropriate to study the temperature distribution inside the rectangular fin.
- Highest cooling rate has been found for aluminium.
- Lowest cooling rate is for Nickel for same period of time

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