PART- I

INTRODUCTION AND LITERATURE REVIEW

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CHAPTER 1

INTRODUCTION AND BASIC OF HEAT TRANSFER

1.1 Introduction

Heat dissipation is a drastic issue to tackle due to continued integration, miniaturization, compacting and lightening of equipment. Heat dissipaters are not only chosen for their thermal performance; but also for other design parameters that includes weight, cost and reliability, depending on application. For example, weight and reliability are important for space applications. The thermal systems must be designed and sized to generate, dissipate the appropriate amount of unwanted heat with the required demand.

The successful working of thermal equipments depends on various factors, majorly cooling or heating of its certain parts. Fins are usually analyzed by assuming uniform heat transfer coefficient model on its surface. However, studies by various investigators revealed that it is not constant, but varies along the fin length. It is mainly because of non uniform resistance experienced by the fluid flow in the inter fin region. Increasing the heat transfer area heat dissipation rate improves, but increase of resistance to fluid flow causing reduction in heat transfer. In order to dissipate the heat of very high heat flux densities, the required heat sink must often be larger than device. Consequently, the heat sink performance is reduced. The inter fin resistance may be reduced by adding the notches or by adding the perforation to the fins. Adding a cross-fin in the center helps to increase the heat dissipation area but it forms the stagnant layer of hot air at the fin bottom. The fluid flow motion at the underside of the fin array can be improved by adding perforation to the fins.

Hence, it was decided to choose the topic “Thermal Performance Analysis by Natural Convection of Perforated Rectangular Fins Arrays with Cross Fins at Center” which has remained untouched by researchers.

In this research, experimental analysis and CFD simulation of heat transfer by natural convection from perforated and non perforated fin arrays along with fin arrays with and without cross fin at the center are presented. In the present research work, actual experiments and analysis of characteristics of heat transfer of perforated rectangular plate
fin arrays having cross fin at the center and without having cross fin by natural convection are carried, by varying the size of the perforation. Results are validated by CFD simulation (Software ANSYS Fluid Flow CFX).

The term perforation means holes of different geometries over the lateral surface of the fin. The solid fin arrays are having external dimensions same as that of perforated fins. In the experimentation, fin arrays were mounted on the rectangular base with constant spacing, the base was insulated to avoid the heat losses. Experiments were conducted for different heater inputs in controlled same environmental conditions. The steady state readings were noted. Calculation of ha, Nu, Ra from obtained readings is carried. Based on results various performance characteristics are plotted. From obtained plots, comparison of heat transfer of perforated fin arrays having cross fin and without cross fin with its external dimensionally equivalent non perforated fin arrays are carried out.

Another part, the simulation using software: it consists of three main steps. The CFD modeling, simulation and post processing are carried in ANSYS 12.0, Workbench environment with ANSYS system of fluid flow (CFX). ANSYS CFX has the capability of solving the convective transport of energy by fluid flow along with conjugate heat transfer (CHT) capability to solve the thermal conduction in solids. It also incorporates a wealth of models to capture all types of radiative heat transfer in and between fluids and solids, whether these are fully or semitransparent to radiation or opaque.

After putting on the proper boundary conditions, solvers are started to get results. When residuals go below the set level, it is said to be that the solution is converged. The results were examined in the post processing.

1.2 Heat Transfer

In the thermodynamics, heat transfer is the transfer of thermal energy from a heated body to a colder body. When an object or fluid, is at a different temperature than its surroundings or another body, transfer of thermal energy is also known as heat transfer. Exchange of heat occurs till body and the surroundings reach at the same temperature. According to the second law of thermodynamics, ‘Where there is a temperature difference between objects in proximity, heat transfer between them can
never be stopped’; it can only be slowed down. Energy flow due to temperature difference is called heat; and 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.

1.3 Modes of Heat transfer

Heat transfer generally recognizes three distinct 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.

1.3.1 Conduction

Thermal conduction is a mechanism of heat propagation from a region of higher temperature to a region of low temperature with a medium (solid, liquid, or gases) or between different mediums in direct physical contact. Conduction does not involve any movement of macroscopic portions of matter relative to one another. The thermal energy may be transferred by means of electrons which are free to move through the lattice structure of the material. In addition, or alternately, it may be transferred as vibrational energy in the lattice structure. Irrespective of the exact mechanism, the observable effect of conduction is an equalization of temperature.

Consider the flow of heat along the metal rod, Fig 1.1, one end of which is placed adjacent to a flame. The elementary particles (molecules, atoms, electrons) composing the rod, and which are in immediate vicinity of the flame, get heated. Because of the resulting temperature growth, their kinetic energy increases and this puts them in a violent state of agitation, and they start vibrating about their mean position. Consequently, these more active particles collide with less active molecules laying next to them. During a collision, the less active particles collide with excited, i.e., thermal
energy is imparted them. The process is repeated for layers after layer of molecules until the other end of the rod is reached.

![Fig. 1.1 Experiment explaining heat transfer by conduction](image)

Each layer of molecules is at a slightly higher temperature than the previous one, i.e. temperature gradient exists along the length of the rod. The rate of heat flow between the two ends depends upon the length of the rod, temperature difference between the two ends and the physical and chemical composition of the bar material.

Since the conduction is essentially due to random molecular motion, the concept is termed as microform of heat transfer and is usually referred to as diffusion of energy. The rate equation for one-dimensional steady state flow of heat by conduction is prescribed by the Fourier law:

$$Q = -Akdx$$  \hspace{2cm} (1.1)

Where $Q$ is the heat transfer rate, $A$ is the area of heat transfer surface, $dt$ is the temperature difference for a short perpendicular distance $dx$, and the thermal conductivity $k$ is a characteristic of the surface material. Since temperature gradient is negative in the positive $x$-direction, minus sign in the equation gives positive heat flow.

If $\delta$ is the path length in the direction of heat flow and $(T_1-T_2)$ is the temperature difference, then

$$= \hspace{2cm} (1.2)$$
The heat flux $q$ is the heat conducted per unit time per unit area and is given by

$$\text{Heat transfer in metal rods, in heat treatment of forgings and through the wall of heat exchange equipments are some practical examples of heat conduction.}$$

1.3.2 Convection

The thermal convection is a process of energy transport affected by the circulation or mixing of a fluid medium (gas, liquid or a powderery substance). Convection is possible only in a fluid medium and is directly linked with the transport of medium itself. Macroscopic particles of a fluid moving in space cause the heat exchange, and thus convection constitutes the macroform of the heat transfer. The effectiveness of heat transfer by convection depends largely upon the mixing motion of the fluid.

With respect to origin, two types of convection are distinguished; forced and natural convection.

1.3.2.1 Natural Convection

In the natural convection, the circulation of the fluid medium is caused by buoyancy effects, i.e., by the difference in the densities of the cold and heated particles. Consider heat flow from a hot plate Fig 1.2 (a) to atmosphere. The stagnant layer of air in the immediate vicinity of the plate gets thermal energy by conduction. The energy thus transferred serves to increase the temperature and internal energy of the air particles. Because of temperature rise these particles become less dense (and therefore lighter) than the surrounding air. The lighter air particles move upwards to a region of low temperature where they mix with and transfer a part of their energy to the cold particles. Simultaneously the cold air particles descend downwards to fill the space vacated by the hot air particles. The circulation pattern, upward movement of the warm air and the downward movement of the cold air, is called the convection currents.
A similar effect can also be demonstrated by water heating Fig. 1.2 (b) because of temperature rise, particles at the middle portion of pot becomes less dense hence lighter than surrounding fluid particles. The lighter fluid particles move upwards to a region of low temperature where they mix with and transfer a part of their energy to the cold particles. Hence the circulation pattern, upward movement of the warm fluid and downward movement of cool fluid.

Hence, natural convection concerns with the Designer of furnaces, house heating system, architectural projects, roads and concrete structures.

Some other examples of free convection are:

- Chilling effect of a cold wind on a warm body.
• Heat flow from a hot pavement to surrounding atmosphere and heating of air in a room by a stove,
• Cooling of billets in the atmosphere,
• Heat exchange on the outside of cold and warm pipes.
• Cooling of Transformers
• Transmission lines
• Rectifiers

1.3.2.2 Forced Convection

In forced convection, the flow of fluid is caused by a pump, fan as shown in Fig 1.3 or by atmospheric winds. These mechanical devices provide a definite circuit for the circulating currents and that speeds up the heat transfer rate.

![Fig 1.3 Forced convection](image)

The convection heat is affected to an appreciable extent by the nature of fluid flow. In the realms of fluid mechanics, essentially to types of flow are characterized, laminar flow and Turbulent flow.

**Laminar Flow:** The fluid particles move in flat or curved un-mixing layers or streams and follow a smooth continuous path. The paths of fluid movement are well-defined and the fluid particles retain their relative positions at successive cross sections of the flow passage. There is no transverse displacement of fluid particles; the particles remain in orderly sequence in each layer. Soldiers on a parade provide a somewhat crude analogy
to laminar flow. Reynolds number is used as a measure of turbulence in fluid flow operations, if \( \text{Re} < 2100 \), the fluid flow is laminar flow. In the heat transfer operation if \( \text{Re} > 10,000 \), the flow is considered as turbulent.

**Turbulent Flow**: The motion of a fluid particle is irregular, and it proceeds along erratic and unpredictable paths. The stream lines are intervened and they change in position from instant to instant. Fluctuating transverse velocity components are superimposed on the main flow and the velocity of the individual fluid elements fluctuate both along the direction of flow and perpendicular to it obviously a turbulent flow is eddying and sinuous rather than rectilinear in character. The turbulent flow resembles a crowd of commuters in a railroad station during the rush hour.

Examples of forced convection are:

- Flow of water in condenser tubes,
- Fluid passing through the tubes of a heat exchanger,
- Cooling of internal combustion engine,
- Air conditioning installation and nuclear reactors.

Regardless of the particular nature, the appropriate rate equation for the convective heat transfer between a surface and an adjacent fluid is prescribed by Newton’s law of cooling.

\[
Q = h A (T_s - T_f) \quad (1.4)
\]

Where \( Q \) is the convective heat flow rate, \( A \) is the area exposed to heat transfer, \( T_s \) and \( T_f \) are the surface and fluid temperatures respectively. The heat transfer co-efficient \( h \) depends upon the thermodynamic and transport properties (e.g. density, viscosity, specific heat and thermal conductivity of the fluid), the geometry of the surface, the nature of fluid flow, and the prevailing thermal conditions.

**1.3.3 Radiation**

Thermal radiation is the transmission of heat in the form of radiant energy or wave motion from one body to another across Fig 1.4 an intervening space. Unlike heat by conduction and convection, transport of thermal radiation does not necessarily affect the material medium between the heat source and the receiver? An intervening medium is not even necessary and the radiation can be affected through vacuum or a space devoid of
any matter. Radiation exchange, fact, occurs most effectively in a vacuum. A material present between the heat source and the receiver would either reduce or eliminate entirely the propagation of radiation energy.

Fig 1.4 Heat transfer by radiation

The mechanism of the heat flow by radiation consists of three distinct phases:

i) Conversion of thermal energy of the hot source into electromagnetic waves: all bodies above absolute zero temperature are capable of emitting radiant energy. Energy released by a radiating surface are continuous but is in form of successive and separate (discrete) packets or quanta of energy called photons. The photons are propagated through the space as rays; the movement of swarm photons is described as the electromagnetic waves.

ii) Passage of wave motion through intervening space: the photons, as carrier of energy, travel with unchanged frequency in straight paths with speed equal to that of light.

iii) Transformation wave into heat; when the photons approach the cold receiving surface, there occurs a reconversion of wave motion into thermal energy which is partly absorbed, reflected or transmitted through the receiving surface.

Thermal radiation is limited to range of wavelength between 0.1 and 100\(\mu\) of the electromagnetic spectrum. Thermal radiation thus includes the entire visible and infrared, and a part of ultra violet spectrum. It is to recognized that thermal radiation is the transfer of energy by disorganized photon propagation. In contrast, organized photon energy such as radio transmission can be macroscopically can be identified and is not considered heat. Further, emission of thermal radiation is
associated with thermally excited conditions which depend upon the temperature and the nature of the surface.

The most common and well-known evidence of radiation heat transfer is that represented by solar energy which passes through inter-stellar space (conditions close to that for perfect vacuum) on its way to the earth surface. Solar radiation plays important part in the design of heating and ventilating systems. Heat transfer by radiation is encountered in boiler furnaces, billet reheating furnaces and other types of heat exchange apparatus. The design and construction of engines, gas turbines, nuclear reactors and solar collectors is also significantly influenced by the radiation heat transfer.

The basic rate equations for radiation heat transfer are based on Stefan- Boltzmann law:

\[ E_b = \sigma_b A T^4 \]  

(1.5)

Where \( E_b \) is the energy radiated per unit time. \( T \) is the absolute temperature of the surface, and \( \sigma_b \) is the Stefan Boltzmann constant.

\[ \sigma_b = 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4 \]

\[ = 4.86 \times 10^{-8} \text{ kcal/m}^2\text{hr K}^4 \]

Above equation is essentially valid for an ideal radiator or a black body-suffix b designates a black surface. The radiant energy emitted by a real surface is less than that for an ideal emitter and is given by

\[ E = \sigma_b A T^4 \]

Where, \( \varepsilon \) is a radiative property of the surface and is called emissivity; its value depends upon surface characteristics and temperature.

1.4 Factors Influencing Heat Transfer Coefficient (ha):

As above discussed, the appropriate equation (1.4) 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 constant of proportionality \( h \) relates the heat transfer per unit time and unit area to the overall temperature difference. The unit of \( h \) is W/m\(^2\)-deg, and it is referred to as convective heat transfer coefficient, the surface conductance or the film coefficient.
The mostly average heat transfer coefficient (ha) is assumed to be a factor for comparing the heat transfer rate of various means. Some factors influencing the ha are listed below:
1. Thermal conductivity of the fluid
2. Density
3. Dynamic Viscosity
4. Specific heat
5. Velocity of flow
6. Surface roughness
7. Geometry
8. Orientation
9. Characteristic length
10. Buoyancy force
11. Flow condition (Laminar, Turbulent)

**1.4.1 Order of Magnitude of ha**

Fig1.5 gives the idea how the average coefficient of heat transfer (ha) varies with different systems and means.
1.5 Nusselt Number

Consider a heated wall surface at temperature $T_s$ over which a fluid is flowing with undisturbed velocity $V_\infty$ temperature $T_\infty$. The particle of fluid in intimate contact with the plate tend to adhere to it, and region of variable velocity builds up between the plate surface and the free fluid stream as indicated in Fig 1.6.

![Fig 1.6 Velocity and temperature profile in convective heat transfer](image)

The velocity decreases as it approaches the solid surface, reaching to zero (no slip conduction) in the fluid layer immediately next to the surface. This layer of stagnated fluid has been called the hydrodynamic boundary layer. The quantity of heat transferred in highly depends upon the fluid motion within the boundary layer, being determined chiefly by the thickness of the layer. The boundary layer thickness $\delta$ arbitrarily defined as the distance $y$ from the plate surface at which the velocity approaches 99% of free stream velocity.

Likewise a region of fluid motion near the plate in which temperature gradients exists is the thermal boundary layer and its $\delta_t$ is defined as the value transverse distance $y$ from the plate surface at which

$$= 0.99$$

At the plate surface, there is no fluid motion and the energy transport can occur only by conduction. From energy balance, this heat transport must equal the heat transferred by convection in to rest of the fluid. Thus

$$= - = h A (T_s - T_\infty)$$

(1.7)
Heat flow rate is thus dependent upon temperature gradient at the wall, and the temperature gradient is influenced by the fluid velocity; high temperature gradients are associated with the higher velocities.

If temperature field of the fluid varies only in the direction of the coordinate normal to the plate surface, then

\[
- \frac{\partial T}{\partial n} = h
\]  

(1.8)

Thus, the convective coefficient \( h \) can be evaluated from knowledge of fluid temperature distribution in the neighborhood of surface.

Introducing the characteristic dimension \( l \), the equation can be re set as

\[
\text{Nu} \frac{l}{k} = \frac{\frac{\partial T}{\partial n}}{\frac{\partial T}{\partial y}}
\]  

(1.9)

\[
= \frac{\frac{\partial T}{\partial n}}{\frac{\partial T}{\partial y}}
\]  

(1.10)

The dimensionless parameter \( hl/k \) is called Nusselt number. Apparently the Nusselt number may be interpreted as the ratio of temperature gradient at the surface to an overall or reference temperature gradient.

The Nusselt number is a condiment measure of the convective heat transfer coefficient. For a given value of Nusselt number, the convective surface coefficient \( h \) is proportional to thermal conductivity \( k \) of the fluid, and inversely proportional to the significant length \( l \).

1.6 Extended Surface (Fins)

In the heat transfer study, the surface that extends from an object is known as a fin. Which 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 of temperature between the environment and the object, also increasing the convection coefficient of heat transfer, or increasing the surface area. But, increase of the area also causes increase resistance to the heat flow. Hence, 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. Since the coefficient of heat transfer powerfully depends on the system of fluid flow, by understanding of flow patterns from the fins is much useful to the designer. Sometimes, it is not economical or it is not practicable to modify the first two options. Adding a fin to a physical object, however, increases the surface area and can sometimes be an economical solution to heat transfer problems.

1.6.1 Types of Extended Surfaces

When the fins are cylindrical or conical, they are called spines. A straight fin is an extended surface of the plane wall. The cross section area of the fin may be uniform or vary with the distance from the wall. Annular fins are attached circumstantially to a cylindrical surface. The area of the straight fins can be calculated as a product of the fin thickness and width. Fig 1.7 and 1.8 shows the different types of the fins and fin arrays.

a) Rectangular profile of longitudinal fin.
b) Trapezoidal profile of longitudinal fins.
c) Fin of rectangular profile equipped on radial cylindrical tube.
d) Fin of the truncated conical profile.

Fig 1.7 Types of extended surfaces (Basic profiles) [53]

Following are the different types of fin arrays; we may come across in this study:
a) Elliptical pin fin array  

b) Rectangular pin fin array  

c) Round pin fins  

d) Straight pin fins  

e) Plate fin array  

f) Sintered copper plate fins  

g) Square pin fins  

h) Inverted V notched fin arrays
i) Rectangular plate fin array  

j) Rectangular plate fin array with cross fin at the center

k) Perforated Rect. fin array (Circular perforation)  
l) Perforated rectangular fin array with cross fin at center (Circular perforation)

Fig. 1.8 Different types of fin arrays [Fig c-g, Source: 2009.Cisco Systems]

1.7 The Heat Dissipation through Fin

Fig 1.9 a) and b) shows the straight fin or longitudinal profile and circular profile respectively. One end of the fin is enclosed in a heating chamber. The other end is exposed to air.
Fig 1.9 a) Rectangular plate fin and b) Circular profile fin

Heat transfer across the rectangular fin and circular rod as shown in figures occurs by conduction. From the surface of the heat transferred to air film by convection.

At steady state conditions.

Heat conduction in to element = heat conducted out of the element + heat convected to the surrounding air from the element surface.

\[ v_1 = v_2 + V + P \frac{d(T - T_o)}{dx} \]  

(1.11)

\[ T_o \] A_c – Cross sectional area of fin.

P- Perimeter of the fin.

h – Convective heat transfer coefficient at the fin surface.

Equation (1.11) simplifies to,

\[ \frac{d^2}{dx^2} + \frac{1}{P} \left( \frac{d}{dx} \right) = -h \frac{d}{dx} \left( T - T_o \right) \]  

(1.12)

\[ \int_1^0 \left( T - T_o \right) = 0 \]  

(1.13)

Substitute \[ \int_1^0 = m^2 = \text{Constant} \]

\[ m^2 (T - T_o) = 0 \]  

(1.14)

Equation (1.14) is a second order linear differential equation. It is a general form of the thermal energy equation for one dimensional heat dissipation from the extended surface.

1.8 Applications of Heat Sinks (Fins)
Natural Convection is the fluid flow induced by buoyant forces, which arise from different densities, caused by temperature variations in the fluid. Natural convection from heat sinks has long been used for thermal management of low-power-density devices. This cooling technique has many advantages such as the absence of moving parts, of power consumption, and of maintenance necessity. In addition, it offers quiet operation, high reliability, and low cost. For these reasons, natural convection heat transfer plays an important role in many types of cooling systems including electronic industry which has attracted constant researches for decades. Fig.1.9 shows the photographs of the applications of the heat sinks commonly used. Some applications can be listed:

- Economizers for steam power plant
- Electrical transformers and motors
- Convectors for hot water and steam heating system
- Air cooled cylinders of aircraft engines, I.C. engines and air compressors
- Cooling coils and condenser coils in refrigerators and conditioners
- Electronic equipments

![Application of fin for cooling the CPU](image)
b) Transformer

c) Air-cooled engines
d) Electric Motor

Fig. 1.10 Application of fins [Source: http://Wikipedia.org]

1.9 Assumptions while Analyzing the Heat Transfer by Natural Convection

While carrying analysis of fins some assumptions are made that are listed as:

a) The heat flow and temperature distribution throughout the fin is independent of time. It means that heat flow is steady.

b) The fin material is isotropic and homogenous. Therefore thermal conductivity of material is uniform.

c) There is no heat source or heat generation inside the fin material.

d) The heat transfer coefficient is uniform and same over the entire surface.

e) The heat conduction is at steady state. It means steady state heat dissipation prevails.
f) Thickness of fin is very small as compared to its height or length. It means temperature gradient across the wall of the fin can be neglected.

g) Radiation heat exchange is neglected, and if any exists, it is included in the heat transfer coefficient.

h) Contact thermal resistance is negligible.

i) Heat conduction is one-dimensional. It means temperature at any cross section of the fin is uniform.

j) Temperature of the fluid surrounding the fin is uniform.

k) The heat transferred through the outer most edge of the fin is negligible as compared to its sides.

l) Temperature at the base of the fin is uniform.

m) The joint between the fin and the tube or bare surface of equipment offers no bond resistance.

n) The flow is laminar.

1.10 Closure

In this chapter overall introduction about the thesis is briefed, giving an overall brief outline about experimental procedures and CFD simulations are implemented. The basic modes of heat transfer like conduction, convection and radiation, Types of fins, application of the fins and assumptions of analyzing the fins are also briefed.