TEMPERATURE DISTRIBUTION AND HEAT FLOW IN CYLINDRICAL FIN WITH HEAT SOURCE

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ABSTRACT

Fin is used to increase heat transfer from heated surface to air. Industrial experience has shown that for same surface area, the fin array can transfer considerably more energy than single fin. Analysis of single fin is well known. However, when fins are placed in an arrangement, the convective patterns become consistent and the resulting heat transfer coefficient has not been predicted. The purpose of this project is to study the effectiveness of staggered cylindrical fin array with and without additional fin. Experimental data from the study is used to obtain the resulting heat transfer coefficient. The experimental study has been carried out through experiment by natural convection using test bench. The additional fin made from aluminum and fabricated in plate shape first before attach with staggered cylindrical array. From the measurement and experiment result, it shows the value of heat transfer coefficient for cylindrical array with additional fin is higher more than fifty percent compare to without additional fin. The relation of Nusselt number with heat transfer has been shown through the experimental result. The experimental result indicates that a staggered cylindrical fin array with additional fin performs better in heat transfer than staggered cylindrical fin array without additional fin. It showed that the additional fin affect the heat transfer much.

ABSTRAK

Sirip digunakan untuk meningkatkan pemindahan haba dari permukaan panas ke udara. Pengalaman industri telah menunjukkan bahawa bagi kawasan permukaan yang sama, pelbagai sirip boleh memindahkan tenaga yang lebih daripada sirip tunggal. Analisis sirip tunggal sudah diketahui. Walau bagaimanapun, apabila sirip diletakkan di dalam aturan, corak perolakan menjadi konsisten dan pekali pemindahan haba yang terhasil tidak diramalkan. Tujuan projek ini adalah untuk mengkaji keberkesanan pelbagai sirip silinder berperingkat dengan dan tanpa sirip tambahan. Data eksperimen dari kajian digunakan untuk mendapatkan pekali pemindahan haba yang terhasil. Kajian eksperimental telah dijalankan melalui eksperimen oleh olakan bebas menggunakan bangku ujian. Sirip tambahan yang dibuat daripada aluminium dan dibina dalam bentuk plat terlebih dahulu sebelum disertakan dengan pelbagai silinder berperingkat. Daripada hasil pengukuran dan eksperimen, kajian menunjukkan nilai pekali pemindahan haba untuk pelbagai silinder dengan sirip tambahan yang lebih tinggi lebih daripada lima puluh peratus berbanding dengan tanpa sirip tambahan. Hubungan nombor Nusselt dengan pemindahan haba telah ditunjukkan melalui keputusan eksperimen. Keputusan eksperimen menunjukkan bahawa pelbagai sirip berperingkat silinder dengan sirip tambahan melakukan yang lebih baik dalam pemindahan haba daripada pelbagai sirip silinder berperingkat tanpa sirip tambahan. Ini menunjukkan bahawa kesan sirip tambahan pemindahan haba adalah banyak.

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LIST OF SYMBOLS

Q	Amount of heat transfer
h	Convection heat transfer coefficient
As	Heat fin surface area
$T\infty$	Temperature of the surroundings
Pr	Prandtl number
Gr	Grashof number
Ra	Rayleigh number
A_c	Cross-sectional area of the heat fin
σ	Stefan-Boltzmann's constant
e	Heat fin's emissivity
tb	Thickness of baseplate
S_L	Longitudinal pitch
S_T	Transverse pitch
W	Width
L	Length
Ts	Surface temperature
Ti	Initial temperature
Nu	Nusselt number
D	Diameter

LIST OF ABBREVIATIONS

FYP	Final year project
UMP	Universiti Malaysia Pahang
ASME	American Society of Mechanical Engineers
IEEE	Institute of Electrical and Electronics Engineers

CHAPTER 1

INTRODUCTION

1.1 GENERAL BACKGROUND

Equipment that generates heat usually incorporates with fins. Finned surfaces are often employed to improve heat exchanging performance. On the other hand, for many years of efforts, the reduction of the size and cost of fins are the main targets of fin industries. Some engineering applications also require lighter fin with higher rate of heat transfer where they use high thermal conductivity metals in applications such as airplane and motorcycle applications.

The continuing increase of power densities in microelectronics and simultaneously to reduce the size and weight of electronic product have led to the increased importance of thermal management issues in this industry. The temperature at the junction of an electronic package (chip temperature) has become the limiting factor determining the lifetime of the package.

The most common method for cooling packages is the use of aluminum pin-fin heat sinks. These heat sinks provide a large surface area for the dissipation of heat and effectively reduce the thermal resistance of the package. They often take less space and contribute less to weight and cost of production. For these reasons, they are widely used in applications where heat loads are substantial and/or space is limited.

However, cost of high thermal conductivity metals is also high. Thus, the enhancement of heat transfer can be achieved by increasing the heat transfer rate and decreasing the size and cost of fin. The major heat transfer from surface to surrounding fluid takes place by convection process.

Increasing the heat transfer mainly depend on heat transfer coefficient (h), surface area available and the temperature difference between surface and surrounding fluid. In this case of study, the optimal configuration for cylindrical fin array which is staggered cylindrical array with and without additional fin will be compare to determine the best performance of heat transfer through that type of fin.

1.2 PROBLEM STATEMENT

Since fins are used in many industrial fields, it is important to predict the temperature distribution within the fin in order to choose the configuration that provides maximum effectiveness. Otherwise, we have to provide a large surface area for the dissipation of heat and effectively reduce the thermal resistance of the package.

Fins are measure based on its effectiveness to dissipate heat. Better heat transfer rate through fins will cause better performance of fins that led to its best effectiveness. The efficiency of fins related closely with many variables such as heat transfer coefficient and fin configuration. Through this study, cylindrical array with and without additional fin will be compared to find the effectiveness between both condition thus solve the problem of cylindrical fin effectiveness.

1.3 OBJECTIVES

1. Study the effectiveness of staggered cylindrical array with and without additional fin based on temperature distribution and heat flow.
2. Perform and collect data experiment data by free convection.
3. The result then compared with published result if available.

1.4 SCOPES OF PROJECT

This project which based on experimental used test bench that available in Thermodynamic laboratory, University Malaysia Pahang as the main equipment. The test bench will be used to collect the temperature distribution in fins. The cylindrical fin and additional fin (plate) is made from aluminum which fabricated first before running the experiment. The experiment data then will be used to gain the heat transfer coefficient which based on cylindrical natural heat convection with two conditions which are cylindrical fin array with and without additional fin. The experimental result that obtained from this project will be compared with the published result on literature review if available.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter will discusses about the previous related study and researches on fin. The sources of the review are extracted from journals, articles, reference books and internet. The purpose of this section is to provide additional information and relevant facts based on past researches which related to this project. This chapter will cover the corresponding terms such as the fundamental of heat transfer, fins and enhancement of heat transfer by fins which had been proved experimentally.

2.2 HEAT TRANSFER

2.2.1 Introduction

Heat can be defined as the process by which energy transport takes place. When a physical body (object or fluid) is at a different temperature than it surroundings, transfer of thermal energy is known as heat transfer. This occurs in such a way that the body and the surroundings reach thermal equilibrium. Thus, heat always flows spontaneously from a hot material to a cold one (2nd Law of Thermodynamics). Transfer of energy occurs mainly through convection, conduction and radiation.

2.2.2 Concept of Boundary Layer

Boundary layer is a layer of fluid in the immediate area of a bounding surface where effects of viscosity of the fluid are considered in detail. Laminar boundary layers come in various forms and can be classified according to its structure and the conditions under which they are created. The thickness of the velocity boundary layer is normally defined as the distance from the solid body at which the flow velocity is 99% of the free stream velocity.

The boundary velocity is the velocity that is calculated at the surface of the body in an inviscid flow solution. The boundary layer represents a deficit in mass flow compared to an inviscid case with slip at the wall. It is the distance by which the wall would have to be displaced in the inviscid case to give the same total mass flow as the viscous case.

The no-slip condition requires the flow velocity at the surface of a solid object be zero and the fluid temperature be equal to the temperature of the surface. The flow velocity will then increase rapidly inside the boundary layer, governed by the boundary layer equations. The thermal boundary layer thickness is similarly the distance from the body at which the temperature is 99% of the temperature found from an inviscid solution.

The ratio of the two thicknesses is governed by the Prandtl number. If the Prandtl number is 1, the two boundary layers are the same thickness. If the Prandtl number is greater than 1, the thermal boundary layer is thinner than the velocity boundary layer. If the Prandtl number is less than 1, which is the case for air at standard conditions, the thermal boundary layer is thicker than the velocity boundary layer.

At high Reynolds numbers, it is desirable to have a laminar boundary layer. This results in a lower skin friction due to the characteristic velocity profile of laminar flow. However, the boundary layer certainly thickens and becomes less stable as the flow develops along the body, and finally becomes turbulent, which is known as boundary layer transition. At lower Reynolds numbers, it is relatively easy to maintain laminar flow.

2.3 FUNDAMENTAL OF CONVECTIVE HEAT TRANSFER

2.3.1 Introduction

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion. Convection deals with the movement of a mass away from a heat source into an area of lower temperature or pressure. As the mass leaves the area, it carries energy with it which will dissipate to the cooler surroundings with a lower pressure.

The equation that governs the rate of heat transfer by convection in a heat fin is known as Newton's law of cooling and is expressed as

$$Q = hA\,(T - T_{\infty}) \tag{2.1}$$

Where;
Q= amount of heat transfer,
h =convection heat transfer coefficient,
As=heat fin surface area,
T =temperature of the heat fin at a specific location
$T\infty$ =temperature of the surroundings.

The heat transfer coefficient, h is not a property of fluid. It is an experimentally determined parameter whose value depends on all variables influencing convection such as surface geometry, the nature of fluid motion, the properties of fluid and the bulk fluid velocity.

Table 2.1: Typical values of convection heat transfer coefficient

Type of convection	***h*, W/m^2. °C**
Free convection of gases	2-25
Free convection of liquids	10-1000
Forced convection of gases	25-250
Forced convection of liquids	50-20000
Boiling and condensation	2500-100000

2.3.2 Force Convection

The important components of forced convection heat transfer analysis are given by Newton's Law of Cooling

$$Q = hA\,(T_w - T_\infty) = hA \cdot \Delta T \tag{2.2}$$

The rate of heat Q transferred to the surrounding fluid is proportional to the object's exposed area A, and the difference between the object temperature T_w and the fluid free-stream temperature T_∞.

h comes from term of the convection heat-transfer coefficient. Other terms describing h include film coefficient and film conductance.

Two-dimensional flow analysis over a flat plate serves well to illustrate several key concepts in forced convection heat transfer.

The viscosity of the fluid requires that the fluid have zero velocity at the plate's surface. As a result a boundary layer exists where the fluid velocity changes from u_∞ in the free stream (far from the plate) to zero at the plate. Within this boundary layer, the flow is initially laminar but can proceed to turbulence once the Reynolds Number *Re* of the flow is sufficiently high. The transition from laminar to turbulent for flow over a flat plate occurs in the range,

$$3 \text{ x } 10^5 < Re_x < 3 \text{ x } 10^6 ,$$

$$Re_x = \frac{\rho u \infty x}{\mu} \quad (2.3)$$

2.3.3 Natural Convection

Natural convection or free convection is caused by buoyancy forces due to density differences caused by temperature differences in the fluid. At heating the density change in the boundary layer will cause the fluid to rise and be replaced by cooler fluid that also will heat and rise. This continues phenomena are called free or natural convection. Boiling or condensing processes are also referred as a convective heat transfer processes.

The heat transfer per unit surface through convection was first described by Newton and the relation is known as the Newton's Law of Cooling. The equation for convection can be expressed as:

$$q = k\, A\, dT \quad (2.4)$$

where;

q = heat transferred per unit time (W)

A = heat transfer area of the surface (m^o)

k = convective heat transfer coefficient of the process (W/m^2K or $W/m^{2o}C$)

dT = temperature difference between the surface and the bulk fluid (K or ^oC)

In natural convection, determining a series of dimensionless numbers helps to give optimum fin configuration in a given surface area. These dimensionless entities are:

- Grashof number — the ratio of heated air buoyancy to viscous forces resisting air movement.

- Prandtl number — the ratio of air momentum to thermal diffusivity. This tells the engineer the amount of internal stresses inside an airflow stream. Prandtl is the reciprocal of the Reynolds number used in forced-convection analysis.
- Rayleigh number — the product of the Grashof and Prandtl number. This dimensionless number determines the type of airflow (laminar, transition, or turbulent along a heated fin surface).

Table 2.2: Dimensionless numbers in natural convection

Parameter	**Formula**	**Interpretation**
Grashof Number:	$Gr=\frac{g\beta \Delta T L^3}{v^2}$	Ratio of fluid buoyancy stress to viscous stress.
Rayleigh Number:	$Ra=Gr{\cdot}Pr$	

2.4 CONDUCTION HEAT TRANSFER

Conduction is a means of heat transfer where a material's molecules will get excited by high temperatures and as a result, transfer energy throughout the material. As the molecules start to collide with each other, the material will begin to give off heat. This process occurs while the material is completely static. The equation that governs the rate of heat transfer by conduction in a heat fin is known as Law of Thermal Conduction and expressed as

$$Q=kA_c\,\frac{dT}{dx} \qquad (2.5)$$

where;

Q = rate of heat transfer by conduction,

k= thermal conductivity of material,

A_c= cross-sectional area of the heat fin,

$\frac{dT}{dx}$= variation of temperature with respect to position.

2.5 RADIATION OF HEAT TRANSFER

Radiation is the energy emitted by matter in the form of electromagnetic waves or photon as a result of the changes in the electronic configurations of the atoms or molecules. It involves the movement of energy from a material by emanating thermal electromagnetic waves. These waves will carry energy with them away from the material in order to lower its temperature.

The equation that governs rate of heat transfer by radiation is known as Stefan Boltzmann's law and is expressed as

$$Q = \sigma\, e A_s\, (T^4 - T^4_{\infty}) \qquad (2.6)$$

where;

Q= amount of heat transfer,

σ = Stefan-Boltzmann's constant,

e = heat fin's emissivity,

A_s= heat fin surface area,

T= temperature of the heat fin at a particular location,

$T\infty$= temperature of the heat fin's surroundings.

2.6 EXPERIMENTAL STUDIES

2.6.1 Single Cylinders

Heat transfer from circular cylinder in countless flow has been explored by many researchers. A summary of experimental correlations of heat transfer from single cylinder is given below.

Table 2.3: Experimental correlation of single cylinder, Waqar Ahmed Khan (2004)

Configuration	Authors	Correlations / Models	Re_D Range
Single Cylinder 	Refai Ahmed and Yovanovich (1997)	$Nu_D = 0.76 + 0.73 Re_D^{1/2} F(Pr, \gamma_D)$ $F(Pr, \gamma_D) = \dfrac{Pr^{1/3}}{\left[(2\gamma_D + 1)^3 + \dfrac{1}{Pr}\right]^{1/6}}$ $\gamma_D = \dfrac{1}{[1 + (Re_D^{0.75}/300)^5]^{1/5}}$	$10^4 - 10^5$
	Quarmby and Fakhri (1980)	$Nu_D = 0.123 Re_D^{0.651} + 0.00416 \left(\dfrac{D}{H}\right)^{0.85} Re_D^{0.792}$	$1 - 2 \times 10^5$
	Churchill and Bernstein (1977)	$Nu_D = 0.3 + \dfrac{0.62 Re_D^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \cdot \left[1 + \left(\dfrac{Re_D}{282000}\right)^{5/8}\right]^{4/5}$	$10^2 - 10^7$

Other experimental studies and their findings related to heat transfer from a single cylinder in an infinite flow are described below.

According to Eckert and Soehngren (1952), they investigated experimentally on the distribution of heat transfer coefficients around circular cylinder in cross flow at Reynolds number from 20 to 500. They found that the thermal boundary layers were quite thick, especially for the lower Reynolds numbers, with separation occurring further downstream than at high Reynolds numbers

Moreover, the contribution of the stagnant region at the downstream side of the cylinder to the over-all heat transfer was low (≈ 15%), but the heat flow into the upstream side was much higher than into the downstream side.

Cimtalay and Fulton (1994) used multiple trades off methods to study the parameter design of heat sink. They had developed a mathematical model to optimize and to evaluate a heat sink on chip.

2.7 FINNED SURFACES

Finned surfaces are frequently used as an efficient method of rejecting waste heat from electronic equipment. These finned surfaces, commonly known as heat sinks, are economical and highly reliable when cooling is by natural convection and radiation. Several authors have developed thermal relationships for closed channels and parallel plates, but there were only one general analytical model for fin arrays, that described by Fritsch (1970) . Fritsch neglects temperature variation in the base plate and use parallel flat plate relations for the convective coefficient.

However, many practical heat sink designs consist of a series of relative short fins which attached to a heated base plate and cannot be accurately approximate by parallel flat plates. The base plate creates additional surface area and a corner geometry have a undesirable effect on heat transfer rates.

Additional experimental investigations have been conducted by lzume and Nakamura (1965) who have also developed a mathematical relationship describing heat transfer from fin arrays. However, the relationship does not hold in the limiting cases of very large or very small fin length to fin spacing ratios (L/S). Donovan and Rohrer (1971) theoretically investigated the radiative and convective heat transfer characteristics of heat conducting fins on a plane wall, but they were mostly concerned with the effectiveness of the extended surfaces and the single contributions of radiation and convection for a film coefficient which was not a function of the fin geometry.

2.7.1 Configuration of fin

a) Jubran *et al.*(1993) performed an experimental investigation on the effects of inter fin spacing, shroud clearance, and missing pins on the heat transfer from cylindrical pin fins arranged in staggered and in-line arrays. They found that the optimum inter fin spacing in both span wise and stream wise directions is 2.5D regardless of the type of array and shroud clearance used. They also found the effect of missing fins to be negligible for the in-line array but more significant for the staggered arrays.

b) Later, Kai Shing Yang *et al.*(2007) performed experiment for the staggered arrangement, and find the heat transfer coefficient increases with the rise of fin density for pin fin heat sinks. For a staggered arrangement where deflection flow pattern vanishes, the elliptic pin fin yields slightly better performance than circular (cylinder) pin fin surface.

c) Marster (1975) studied the heat transfer properties of a single vertical row of heated cylinders under natural convection and presented the result for variety of combinations of spacing and number of cylinder.

d) Sparrow and Vemuri (1985) found that the rate of heat transfer from the fin baseplate assembly increased with increasing fin surface area

2.7.2 Selection of Heat sink

There are two type of heat sink which are closely packed fins and widely spaced. Both types of heat sink provide their own characteristics which are: