NUMERICAL ANALYSIS ON THERMAL PERFORMANACE OF STAGGERED PIN-FIN ASSEMBLY

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To my beloved wife and children

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ABSTRACT

Steady state heat transfer and friction characteristics and performance analysis for flow through a rectangular channel with heat sinks of circular cross-section pin fins in turbulence flow was studied in the present study. The heat sink was immersed in the channel with fluid flowing at constant inlet temperature and pressure of uniform approach velocity approaching the pin fin at normal to pins. This study had investigate the performance of heat sinks numerically by modelling the model using mesh generator, GAMIT and analyzing the model using commercially available Computational Fluid Dynamic software, FLUENT. The study concentrated on pins with staggered arrangements with variations in the clearance above pins (C) to pin height (H), clearance ratio (C/H = 0.35, 1, 3, 7 and 15) and ratio of distance between pins to the pin diameter (Sx/d = 1, 1.5, 2, 2.5 and 3) along the streamwise direction while pin spacing along the spanwise direction was maintain fixed. Ascertainment of the performance of the model were by using the ratio of duct average Nusselt number for pinned model base on projected area (Nu) to the duct average Nusselt number of a smooth surface (Nu_s). The turbulence flow of Reynolds number 10000 – 50000 were modelled and simulated using the k-ε modelling option of FLUENT. The comparatively lower Reynolds numbers of 10000 – 30000, lower interfin distance of Sx/d = 2 and clearance ratio should be a preferred staggered arrangement option.

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LIST OF SYMBOLS

A	Heat-transfer area
C_p	Specific heat of air at constant pressure
f	Friction factor
F_α	Value of F table
h	Heat-transfer coefficient
h_k	Height of the fins
H	Height of the duct
k	Conductivity of air
L	Length of base plate
$\dot{m}$	Air-mass flow rate
Nu	Nusselt Number (hD/k)
Q	Heat rate
Re	Reynolds number
T	Steady state temperature
U	Mean velocity of the air
W	Width of the base plate and the duct.
ΔP	Pressure difference
υ	Kinematic air viscosity of
μ	Overall mean performance value
ρ	Density of air
η_{th}	Overall thermal efficiency
ε	Heat transfer effectiveness

Subscripts

a	air
av	average
conv	convection
cond	conduction
in	inlet
out	outlet
rad	radiation
s	surface

CHAPTER 1

INTRODUCTION

1.1 Background

The continuing increase of power densities in microelectronics and the simultaneous drive to reduce the size and weight of electronic products have led to the increased importance of thermal management issues in this industry. The most common method for cooling microelectronics packages is by the use of aluminium 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 the weight and cost of the product.

For these reasons, they are widely used in applications where heat loads are substantial and/or space is limited. They are also found to be useful in situations where the direction of the approaching flow is unknown or may change. They offer a low cost, convenient method for lowering the thermal resistance and in turn maintaining equipment temperature at a safe level for long term, reliable operation.

The overall performance of a pin-fin heat sink depends on a number of parameters including the dimensions of the base plate and pin-fins material, thermal joint resistance, location and concentration of heat sources. These parameters make the optimal design of a heat sink very difficult. Traditionally, the performance of heat sinks is measured experimentally or numerically and the results are made available in the form of design graphs in heat sink catalogues. Analytical and empirical models for the fluid friction and heat transfer coefficients are used to determine optimal heat sink design.

1.2 Classification of heat sink

Pin-fin heat sinks consist of a base and an array of integrally attached pins. They can be classified in many ways, either by the density of the pins or the arrangement of the pins. Figure 1.1 shows the (i) low or high density and Figure 1.2 and Figure 1.3 show the arrangement, in-line and staggered respectively.

The effective cooling scheme for pin-fin heat sinks is forced convection where forced air creates a significant amount of air in between the pins and enhancing the heat sink's efficiency

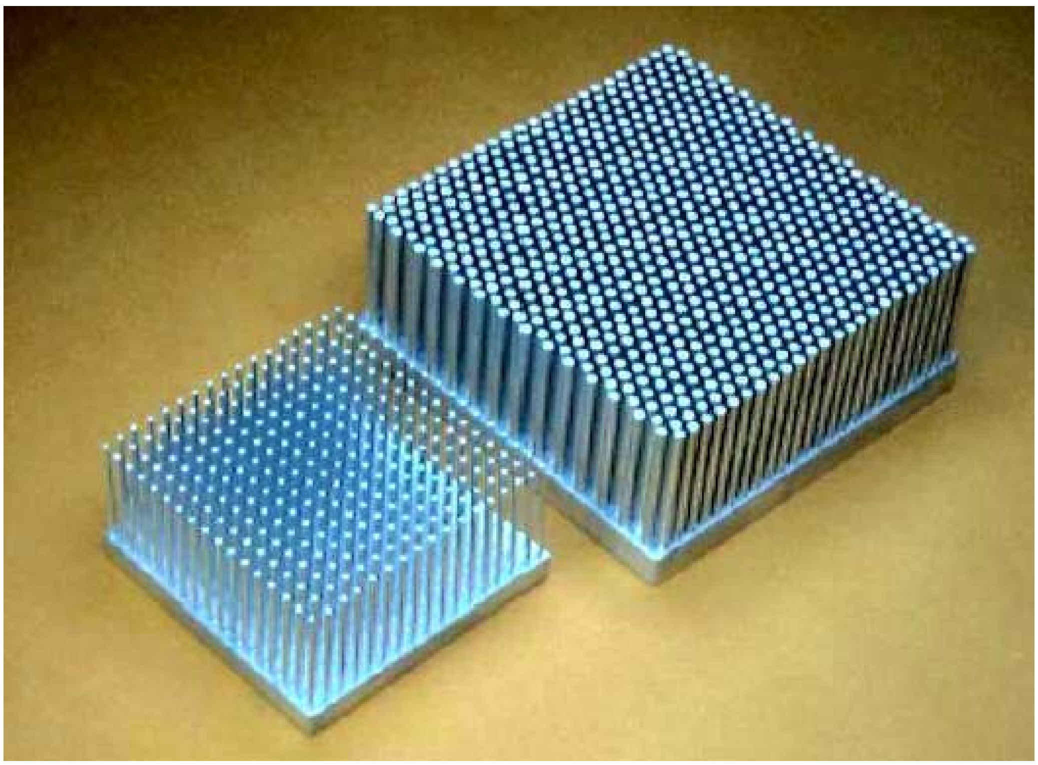

Figure 1.1: Sample of Low and High Density heat sink (Alpha, Japan)

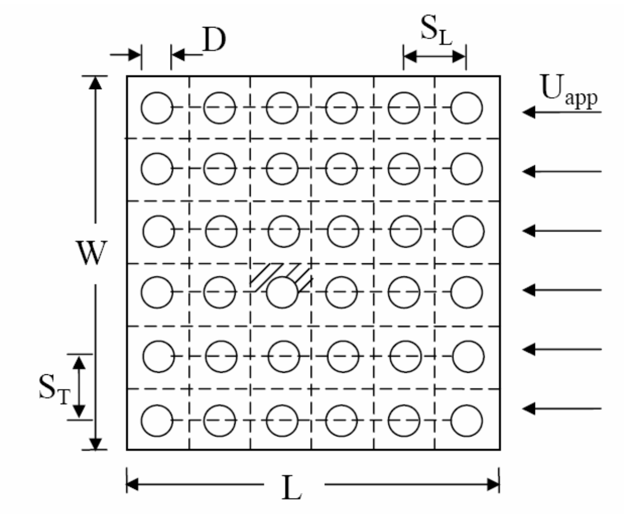

Figure 1.2: Typical foot print of inline arrangement heat sink [15]

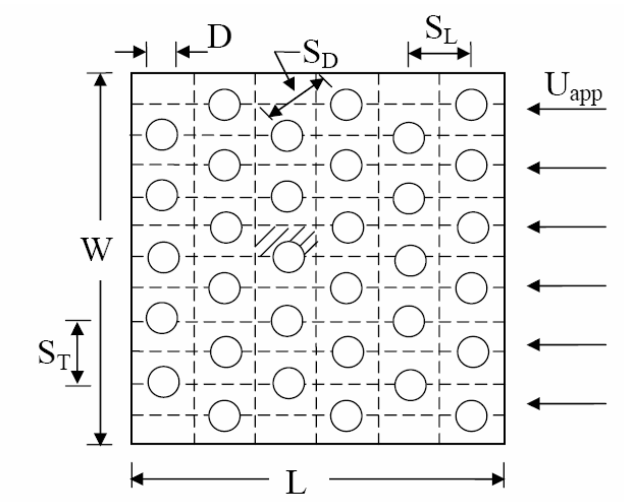

Figure 1.3: Typical foot print of staggered arrangement of heat sink. [15]

1.3 Flow and Heat Transfer Behaviour across Pin-Fin Heat Sinks

Various experimental works for tube banks provides some guidance in understanding the flow behaviour in pin-fin heat sink. Those experiments conducted show that the flow over tubes is controlled by the pressure gradient, the fluid viscosity, and the Reynolds number. This flow is more complex than that over a single tube, due to no uniformity of the velocity field, high turbulence, and other factors including longitudinal and transverse pitches.

Flow over tubes within a bank involves significant blockage of the flow passage; the pressure gradient at the tube surfaces is affected by the degree of flow constriction. The tube rows of a bank are either arranged staggered or inline in the direction of fluid velocity. The configuration is characterised by the pin diameter and by the spanwise or transverse pin pitch S_T and the streamwise or longitudinal pitch S_L measured between the pin centres.

Flow conditions within the bank are dominated by boundary layer separation effects and by wake interaction, which in turn influence convection heat transfer.

The applicability of the previous experimental work regarding heat transfer from cross flow of tube banks has definite limitations when it comes to the analysis of pin fin heat sinks. There are differences between the pin fins and tube banks. One difference is that in tube banks, the tube array consists of very long hollow cylinders that generally carry some fluid internally. As a result, the surface temperature of the cylinder surface can be assumed isothermal along its length. Whereas, in a pin fin heat sink, the so-called "fin effect" results in a very definite temperature gradient along the height of the pins.

The heat transfer coefficient associated with a pin is determined by its position in the bank. The coefficient for a pin in the first row is approximately equal to that of a single pin in cross flow, whereas larger heat transfer coefficient are associated with pins of the inner rows. The pins of the first row act as turbulence grids, which increases the heat transfer coefficient for the pins in the following rows. In most configuration, however, heat transfer condition stabilised, such that little change in the coefficient for pin beyond the fourth of fifth row.

Generally, the average heat transfer coefficient for entire tube bundle of 10 or more rows are correlated by Grimison, of the form [14]

$$Nu_D = C_1 \operatorname{Re}_{D.\max}^m \qquad (1.1)$$

Where C_1 and m are constant and

$$\mathrm{Re}_{D,\max} \equiv \frac{\rho V_{\max} D}{\mu} \tag{1.2}$$

.

The Reynolds number $\mathrm{Re}_{D,\max}$ for the foregoing correlations is based on the maximum fluid velocity occurring within the pin bank.

For staggered configuration, the maximum velocity may occur either at the transverse plane A_1 or the diagonal plane A_2 of Figure 1.4 above. It will occur at A_2 if the rows are spaced such that $2(S_D - D) < (S_T - D)$, in which case

$$V_{\max} = \frac{S_T}{2(S_D - D)} V \tag{1.3}$$

.

If $V_{\max}$ occurs at A_1,

$$V_{\max} = \frac{S_T}{S_T - D} V \tag{1.4}$$

.

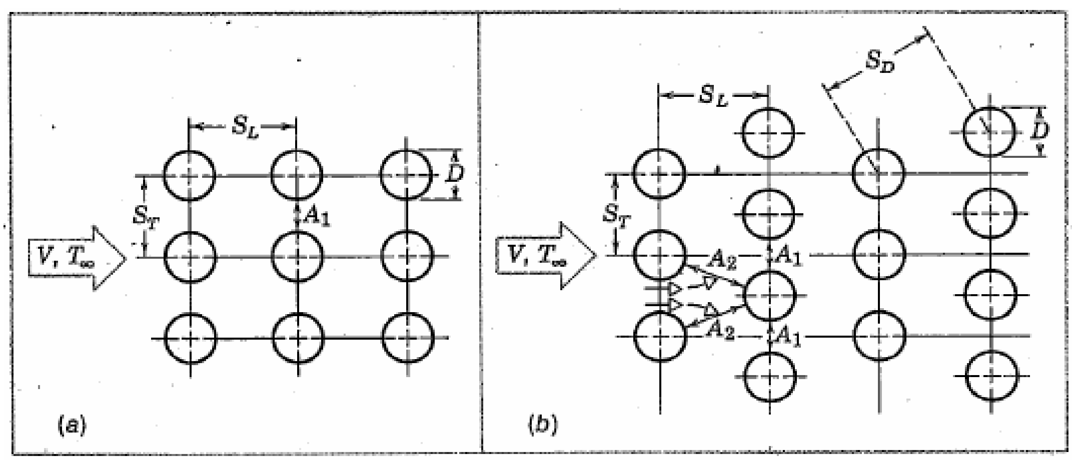

Figure 1.4: Pin fin arrangement (a) Inline (b) Staggered [14]

A more generalised correlation is the correlation proposed by Zhukhaukas, of the form [14]

$$Nu_D = C_1 \mathrm{Re}_{D.\max}^{m} \mathrm{Pr}^{0.36} \left(\frac{\mathrm{Pr}_\infty}{\mathrm{Pr}_S} \right)^{1/4} \tag{1.5}$$

Where all properties except $\Pr_S$ are evaluated at T_∞ and the constant C and m are as listed in table below.

Table 1.1: Constant for equation (2) for tube bank in cross flow [14]

Configuration	$\mathrm{Re}_{D,\max}$	**C**	**m**
Staggered			
$\frac{S_T}{S_L} < 2$	$10^3 - 2 \times 10^5$	$0.35\left(\frac{S_T}{S_L}\right)^{1/5}$	0.6
$\frac{S_T}{S_L} > 2$	$10^3 - 2 \times 10^5$	0.4	0.6

One should recognise that there is generally as much interest in the pressure drop associated with the flow across a bank of pin-fin as in the overall heat transfer rate. The power required to move the fluid across the bank is often a major operating expense and is directly proportional to the pressure drop. The most reliable pressure drop relation is given by Zhukaukas, and was also reported in most other researchers. Tahat et al. [5, 6] and O. N. Sara [7, 12] also use the same relations in their experimental analysis. The relation is in the form

$$\Delta p = N_L \chi \left(\frac{\rho V_{\max}^2}{2} \right) f \qquad (1.6)$$

The friction factor f in terms of Euler number Eu and the correction factor χ are as per plots below.

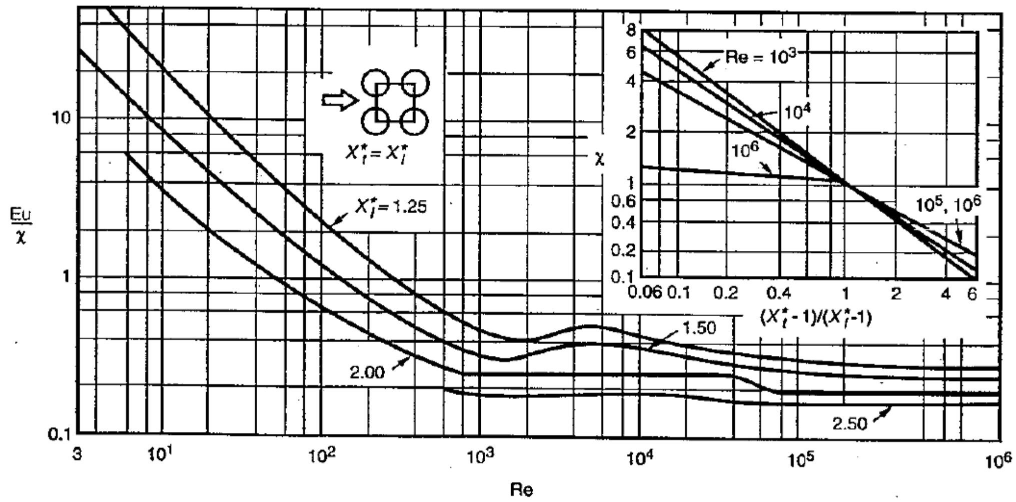

Figure 1.5 (a): Friction factor Eu and correction factor χ for inline tube bundle arrangement [17]

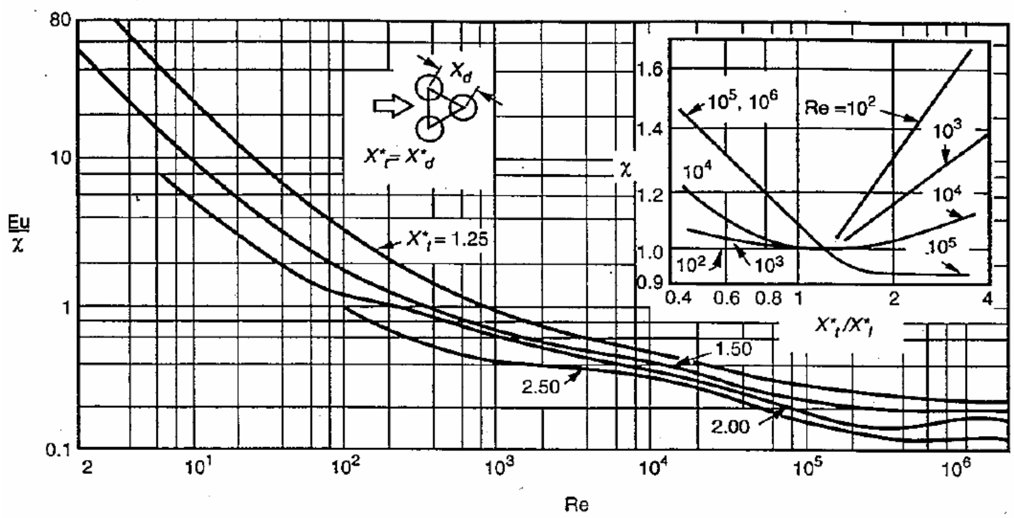

Figure 1.5 (b): Friction factor Eu and correction factor χ for staggered tube bundle arrangement [17]

1.4 Scope of Study.

The objective of this study is to obtain an analysis on the heat transfer and friction factor on a staggered arrangement of pin fins of smaller diameter size for untoward application as heat sink. The scope will analyse the influence of pin shape, inter fin spacing and the clearance gap on heat transfer and friction characteristics.

This study will be carried out using commercially available Computational Fluid Dynamic (CFD) software. The software chosen is FLUENT.