CHAPTER 1

INTRODUCTION

1.1 Rationale

In electronic systems and circuits, there is heat generated and it causes a reduction on the performances and also the reliability of the electronic circuits. The heat could be the internal heat generation or come from external sources.

At present, electronic circuit is designed to have a smaller size but its performance is higher.Many circuits have been integrated to be a main circuit called integrated circuit(IC). Figure 1.1 shows a prediction of the number of transistors contained in IC of a central processing unit (CPU). With the recent technology, the micro-electronics is going quickly and it could be estimated that the number is double in every 18 months (Azar, 2000).

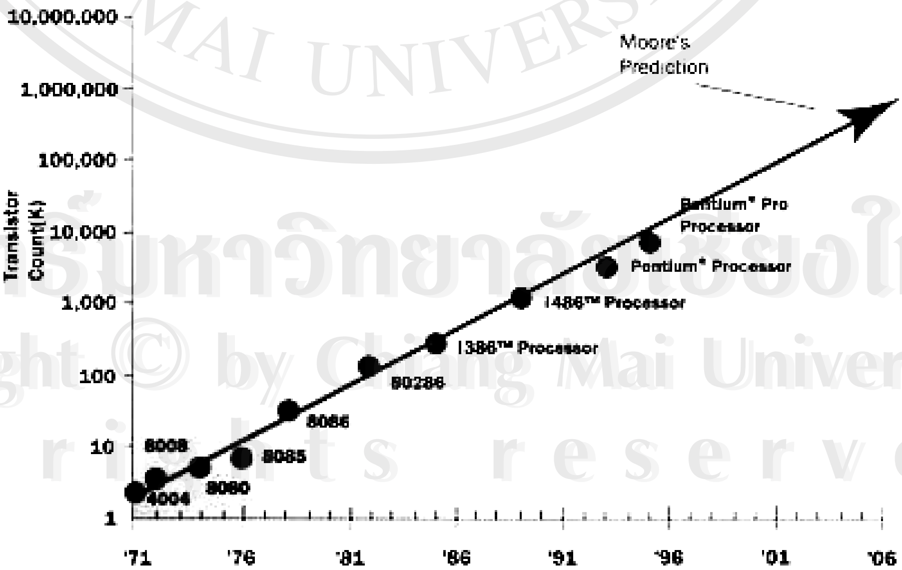

Figure 1.1 Prediction of the number of transistors contained in IC of Intel CPU (Azar, 2000).

With high density of the electronic circuits, more internal heat could be generated in the IC. Figure 1.2 shows an example of heat dissipation in a CPU.

When the electronic parts are contained in any printed circuit board (PCB), it is necessary to have a good thermal management to control the temperature inside not to exceed the designed limit thus the circuits could be operated for a long period with high reliability.

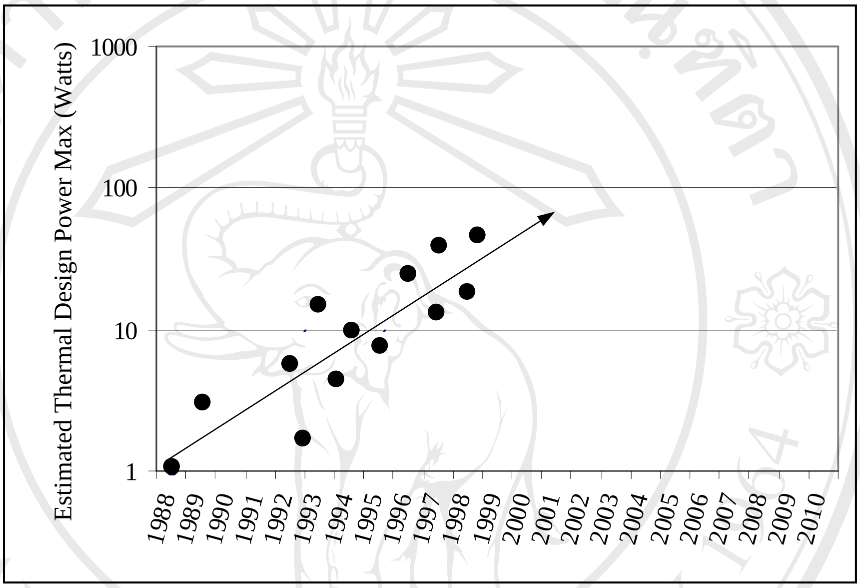

Figure 1.2 Power dissipation of a CPU (Azar, 2000).

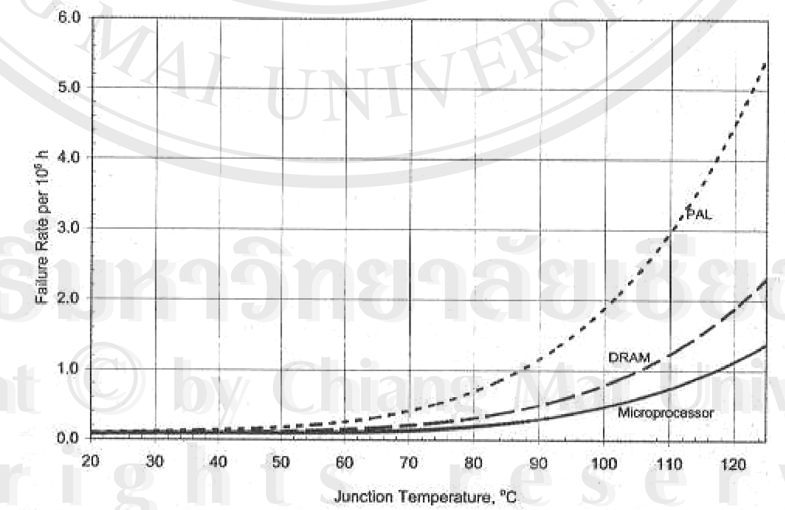

Figure 1.3 Rate of failure in electronic equipment with various junction temperature (Ralph, 2001).

With poor thermal management, the temperature of the electronic equipment will increase and some damages in the electronic part could happen when its temperature is over the recommended value from the manufacturer. The rate of change of the chemical properties in the material is double when the tem perature increases every 10^{o}C (Ralph, 2001) which increases the failure of the system operation. Figure 1.3 shows the failure rate of three electronic parts which are: Programmable Array Logic (PAL), 256K Dynamic Random Access Memory (DRAM) and a microprocessor at different values of junction temperature. The slope increases drastically when the temperature is over 90^{o}C.

Recently, there are many cooling methods in thermal management to replace air cooling, for example, heat rejection with water cooling, micro-refrigerator cooling, heat-pipe cooling, and thermoelectric cooling. However the air cooling method still has an advantage due to its low maintenance and simple operation so this technique is still widely used for cooling the electronic circuits.

1.2 Literature Review

1.2.1 Geometric Optimization of Cooling Techniques

Morega and Bejan (1994) evaluated optimal spacing between a PCB having artificial heat sources and an adiabatic surface. The calculation was done numerically with two cases: when the heat sources were embedded in the PCB and allocated on the PCB.

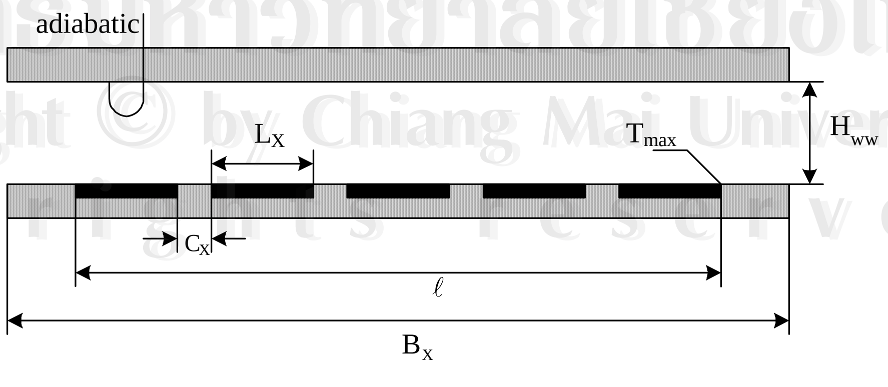

Figure 1.4 Study on the positions of the heat sources when the heat sources are embedded in the surface of PCB (Morega and Bejan, 1994).

Figure 1.4 shows the PCB having a set of 5 heat sources embedded at the upper surface. There was an adiabatic plate over the PCB with a spacing of H_{ww}. The PCB total length was B_X and the length of each heat source was L_X . The spacing between the heat sources was C_X and the total length from the first heat source to the last one was *l*.

Each heat source generated the same value of heat rate. The maximum temperature (T_{max}) occurred at the outer edge of the last heat source. The maximum overall thermal conductance of the stack (M_{max}) was found to be related with $H_{ww(opt)}$ as shown in Figure 1.5 and could be calculated by

$$H_{\max} = \frac{\dot{Q}_{\max}/[H_{ww}(T_{\max} - T_\infty)]}{B_X C_p (\rho \Delta P / \mathrm{Pr})^{\frac{1}{2}}} \tag{1.1}$$

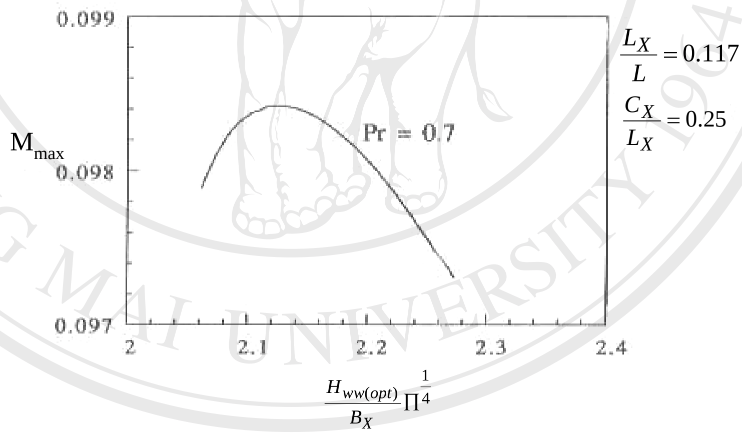

Figure 1.5 Effect of $H_{ww(opt)}$ on the maximum heat (Morega and Bejan, 1994).

Another allocation of the heat sources is shown in Figure 1.6 The heat sources now were placed on the PCB and the thickness of each heat source was B.

The optimum spacing could be calculated by

$$\frac{H_{ww(opt)}}{\ell} \cong 2.7 \Pi_\ell^{-\frac{1}{4}} . \tag{1.2}$$

The overall thermal conductance of the stack is correlated adequately by $M_{max} \approx 0.08$, which in physical terms means that

$$\frac{\dot{Q}_{max}/H_{ww}}{T_{max}-T_{\infty}} \approx 0.08k\,\frac{B_y}{B_X}\,\Pi_L^{\frac{1}{2}}. \qquad (1.3)$$

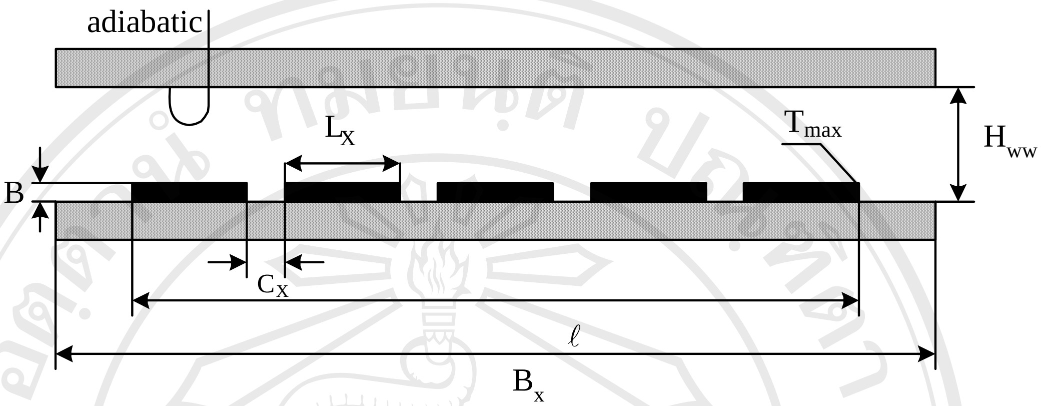

Figure 1.6 Allocations of the heat sources on the PCB (Morega and Bejan, 1994).

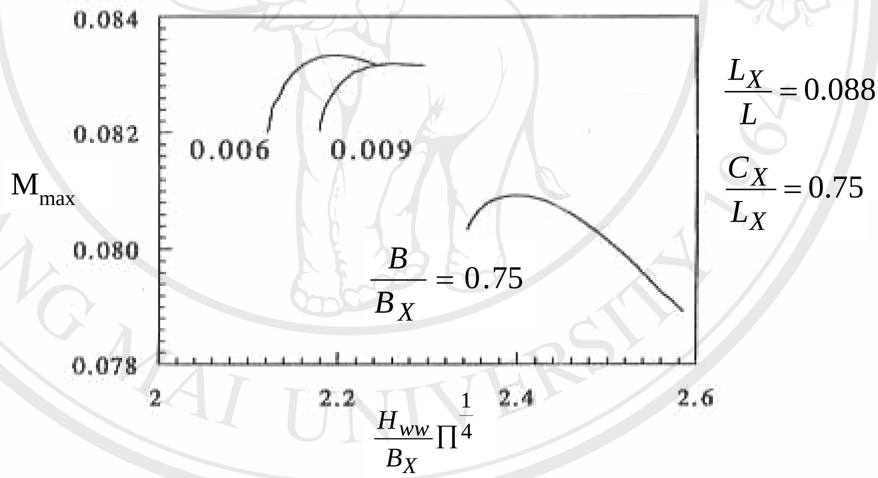

Figure 1.7 Effect of the heat source thickness on the value of M_{max} (Morega and Bejan, 1994).

1.2.2 Heat Transfer from Chips with In-line Arrangement

Table 1.1 shows geometric specifications of the tested chips. The chips were allocated in in-line arrangement and were tested in an air channel as shown in Figure 1.8. Each chip was a square with the dimension L_X which was in a range of 25.4-69.8 mm. The ratio of the total length with chip thickness, L/B was in a range of

2.67-8.75. The chip density D, was in a range of 0.25-0.69 and the ratio between the channel height to the chip thickness, H_{ww}/B was in a range of 1.5-10.

Table 1.1 Geometric specifications of the tested models (Lee and Kim, 1996).

Investigator	L_X [mm]	L_X/B	D	H_{ww}/B
Wirtz and Dykshoorn (1984)	25.4	4.0	0.25	1.5-4.6
Sparrow et al. (1982)	26.7	2.67	0.64	2.7
Anderson and Moffat. (1992)	37.5	3.95	0.59	1.5-4.6
Wirtz et al. (1994)	56.0	8.75	0.49	1.5-10
Wirtz and Mathur.(1994)	69.8	6.00	0.45	2.0
Wirtz and Colbon. (1995)	69.8	6.00	0.45-0.69	2.0

The value of the average heat transfer coefficient of the models in Table 1.1 varied with average air velocity as shown in Figure 1.9 The results were as follows:

1. The deviation of all the data from the average values was less than ± 25 %.
2. The heat transfer coefficient h decreased when the value of L increased.
3. The value of h increased with increase of wind velocity and it was found that $h \sim V^n$ where n was in arrange of 0.6-0.8 .

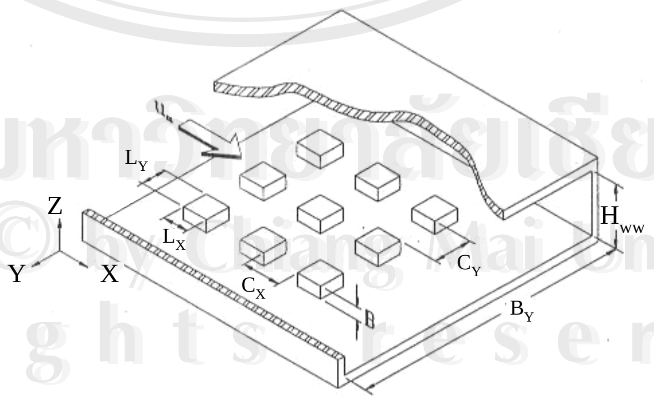

Figure 1.8 Allocations of the chips on the PCB tested in the channel
(Wirtz and colbon, 1995)

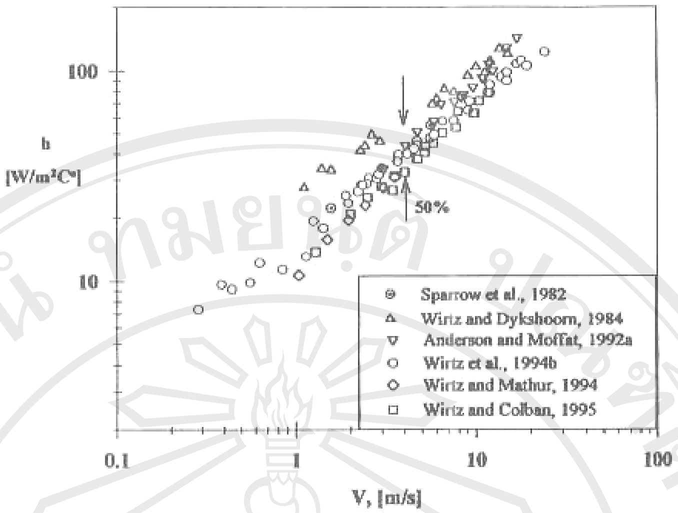

Figure 1.9 The heat transfer coefficient h of the models in Table 1.1.
(Lee and Kim, 1996).

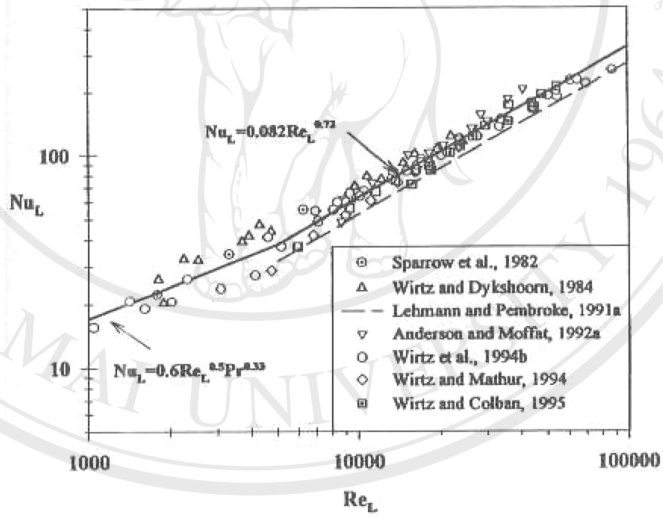

Figure 1.10 Empirical correlation of Nusselt number with Reynolds number
(Lee and Kim, 1996).

Figure 1.10 showed an empirical correlation of Nusselt number, $Nu_L = hL_X/k$, with Reynolds number, $Re_L = \rho V L_X/\mu$. The deviation of the results from the average value was ± 10 %. Moreover, it could be found that there was a change in the slope which is the change from laminar to turbulent regions at $Re_L \approx 5000$. The correlations could be written as

$$Nu_L = 0.6\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{0.33}, \mathrm{Re}_L \le 5000 \qquad (1.4)$$

and

$$Nu_L = 0.082\,\mathrm{Re}_L^{0.72}, \mathrm{Re}_L > 5000\ . \qquad (1.5)$$

Pr was Prandtl of the fluid. Eqns (1.4) and (1.5) were valid in the laminar and turbulent regimes, respectively.

Lehmann and Wirtz(1985) also considered convection from the chip rows when the flow in the channel was fully developed. The Reynolds number was controlled to be about 1000, 2000 and 3000.

Moffat et al (1985), Moffat and Anderson (1988), Anderson and Moffat(1990) and Anderson and Moffat(1991) studied the row arrangements of the electronic chips. The techniques and some data had been performed to predict the operating temperature and the heat transfer coefficient. There are some researchers were working in the same work [Sridar et al(1990), Faghri et al(1995), Wirtz and Weiming(1991), and Kang(1992)].

1.3. Enhancement Techniques

A large number of choices exist for the enhancement of forced-convection cooling of electronic equipment. The following strategies may be identified for the design and implementation of any enhancement technique:

- Improvement of convective heat transfer coefficient: Improvement can be accomplished by the choice of fluid with superior thermo physical properties or by increasing flow velocities.
- Increase of surface area available for heat transfer: This has traditionally been accomplished by the use of enhanced surface (fins), and a large body of literature exists on their design and implementation.
- Flow modulation: This strategy aims to increase mixing in the flow using device such as vortex generators, tabulators, and swirl flow devices. Increases

due to flow modulation in heat transfer coefficient of as much as 100% have been reported for turbulent flows.

These three strategies may often not be entirely distinct; for instance, flow modulation is essentially a means for increasing heat transfer coefficients by selectively increasing the local velocity or shears in the flow near the surface to be cooled. Similarly, arrangements such as louvered fins increase both the heat transfer coefficient as well as the surface area (Webb, 1987)

1.3.1 Inherent Enhancement at Electronic Component Surface

It is important to recognize that electronic components have naturally enhanced heat transfer (due to enhanced hA products) when compared to predictions based on thermally ordered conditions (smooth surface, vibration-free analyses). In addition to the beneficial flow-field disruption caused by virtue of their protrusion, this natural enhancement results from the surface roughness of the circuit boards, vibrations in electrical equipment, and electric fields in power equipment (Kraus and Bar-Cohen, 1983). The surface roughness of a printed circuit board has been shown to lead to an increase in the heat transfer coefficient relative to a smooth surface by up to a factor of 10 (Wenthen, 1977).

1.3.2 Enhancement by Controlling Geometric Layout

Significant enhancements in heat transfer can be realized by controlling the geometric layout of the chips and the assembly of the boards. Staggering the chips on a board has been shown in several studies to increase the heat transfer coefficient relative to an inline configuration for a given inlet velocity and channel height. In air-cooling experiments, Hollworth and Fuller (1987) obtained as much as a 50% increase in heat transfer; in water, a 10 to 40% enhancement was seen to be accompanied by a 20 to 110% increase in pressure drop (Garimella and Eibeck, 1991c). Figure 1.11. shows the heat transfer coefficients for inline and staggered arrangements of an array of chips with water as coolant (Garimella, 1991). The flow visualizations in Figure 1.12 (water with hydrogen-bubble tracers and laser-sheet illumination) demonstrate the superior mixing obtained by staggering the chips in an array. With fan power held constant, Ashiwake et al. (1983) obtained a 70% drop in cooling-air temperature by staggering the chips of an inline array.

Even in inline arrays, the heat transfer coefficient can be increased by increasing either the stream wise spacing between chips (35-40%) or the span wise spacing (15%) as shown in Garimella and Eibeck (1990,1991b). General recommendations about component orientation and the use of mixed component sizes were drawn by Azar and Russell (1991) based on flow patterns visualized in water, There are also indications that chips in the horizontal orientation have lower thermal resistances than those in the vertical orientation, even in forced convection due to buoyancy effects; the vertical orientation also results in greater asymmetry in component temperature distribution (Azar et al., 1989). This effects would, however, be swamped at sufficiently large Reynolds numbers.

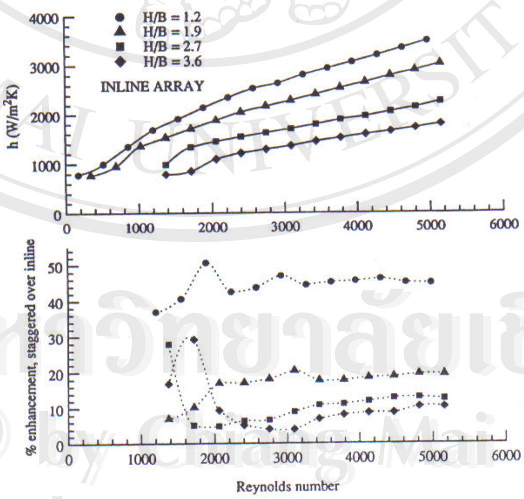

Figure 1.11 Heat transfer coefficients in water for an inline array of chips, and the percent enhancement obtained by fully staggering the chips (by on chip width); the Reynolds number (Re_H) is based on the channel height, and the chip spacing is equal to the chip width and length.

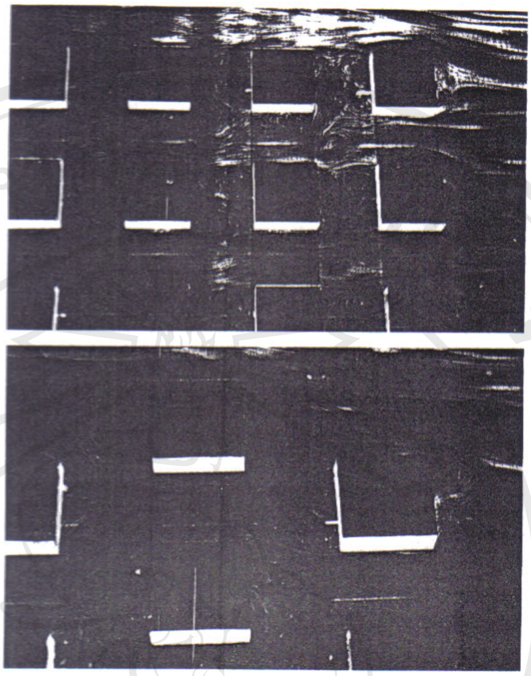

Figure 1.12 Flow patterns visualized with hydrogen bubbles in water and staggered array of chip; flow is from right to left (H/B=3.6, Re_H=3450)

1.3.3 Enhanced Surface

A vast amount of literature exists on the use of extended surfaces for heat transfer enhancement. Reported studies have ranged from analytical solutions for the simpler fin configurations to detailed conduction and conjugate heat transfer computations and to experimental databases for a large variety of extended surfaces. Detailed discussions are widely available (Bergles et al., 1983,1991) and (Webb, 1987,1994).

1.3.4 Roughness Elements

Roughness elements, either random in nature or in a repeated-rib configuration, have been extensively studied as a technique for heat transfer enhancement in channel flows. The roughness elements considered are typically much smaller than the ribs and obstructions of the preceding section-at least and order of magnitude less than the boundary layer thickness, and in tubes, one or two orders of magnitude less than the tube diameter. Nakayama (1982) reviewed the influence of roughness Reynolds number, Prandtl number, and geometrical parameters on the momentum and heat transfer roughness functions, for both granular, three-dimensional surface roughness and for repeated-rib roughness. It was concluded from comparing a large number of studies in the literature that the relative merits of roughening a surface are large when the roughness Reynolds number is small; granular (3-D) roughness provided more favorable results than rib (2-D) roughness. The heat transfer enhancement is also larger for higher Prandtl-number fluids. It should be pointed out that in implementing roughness as a heat transfer enhancement technique variable physical properties have a more pronounced effect on heat transfer in rough passages than in smooth passages (Wassel and Mills, 1979). Since fairly large temperature variations are experienced in electronic applications, care should be taken to account for variable-property effects.

1.3.5 Vortex Generators and Barriers

Significant work has been done for decades on the use of vortex generators to enhance heat transfer and to control flows in different applications. Figure 1.13 illustrates the principle of operation of a half-delta-wing vortex generator and the longitudinal vortex formed; a pair of counter-rotation vortex generators are also shown.

Garimella and Eibeck (1991 a) reported a study in which half-delta wings were placed upstream from, and on the same wall as, each stream wise column of chips; heat transfer enhancement with water as coolant was studied as a function of steam wise position and Reynolds number. Vortex generators of two heights (one and two times the chip height) were studied. As with other kinds of barriers, the greatest enhancement was observed at the second row of chips downstream and at a Reynolds number in the transitional flow regime, as shown in Figure 1.14 .

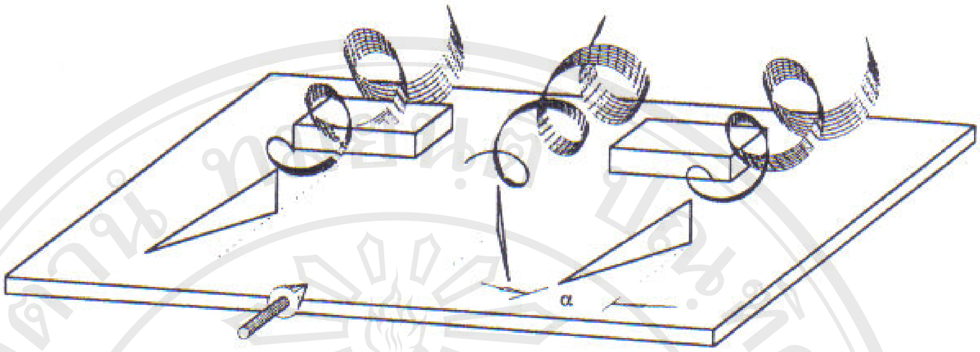

Figure 1.13 Schematic of the single and counter-rotating pair of longitudinal vortices produced by half-delta-wing vortex generators.

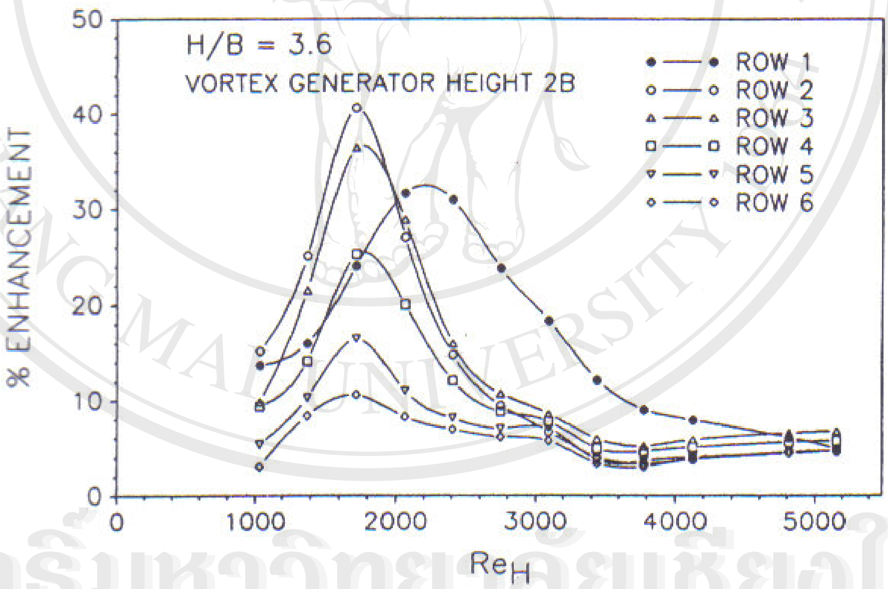

Figure 1.14 Heat transfer enhancement with vortex generators twice the chip in height placed upstream from Row 1, H/B = 3.6 (Garimella and Eibeck, 1991a).

Anderson and Moffat (1991) suggested the introduction of scoops (Figure 1.15a) in the low-velocity recirculation region downstream of each chip to enhance thermal mixing and thus reduce overall temperature rise. A decrease in the component temperature rise of up to 19% was observed as a result of the scoops in the row just downstream (see Figure 1.15b); the associated increase in pressure drop over the eight stream wise rows of chips with scoops introduced behind one row was found to be 11%.

Heat transfer effects due to horseshoe vortices (see Figure 1.16) along the wall downstream from a wall-mounted cylinder and streamlined cylinder were studied by Fisher and Eibeck (1990), who observed local increases in heat transfer of 20 to 30%.

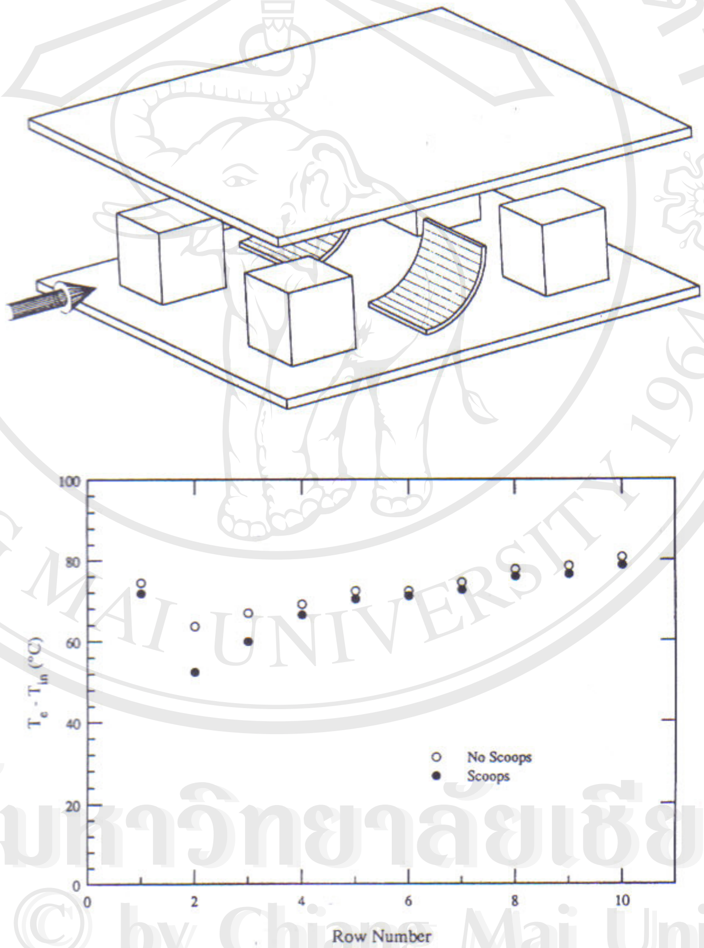

Figure 1.15 (a) Schematic of the scoops installed between two rows of chips;
(b) Comparison of the component temperature rise with and without scoops; the scoops are placed between rows 1 and 2 (Anderson and Moffat, 1991).

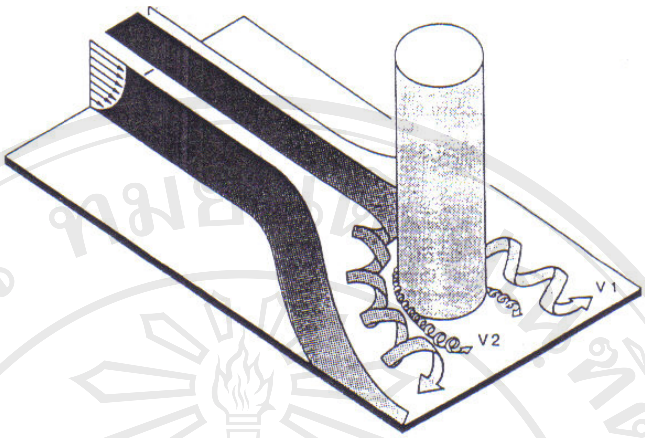

Figure 1.16 Simplified schematic of the horseshoe vortex formed at a wall-mounted cylinder (Goldstein and Karni, 1984).

1.3.6 Impinging Jets

Impinging jets may be classified as free-surface of submerged, and the spent flow can be in a confined or unconfined state. Also, the nozzles producing the jets could be round orifices of slots, singly or in arrays, based on the application (Figure 1.17). Free-surface jets are those where the fluid issuing from the jet is different from the ambient fluid, resulting in a distinct free surface separating the two fluids; an example is a jet of water issuing into surrounding air. In submerged jet impingement, on the other hand, the ambient fluid is the same as the impinging jet; this is the case for air-in-air or water-in-water jets.

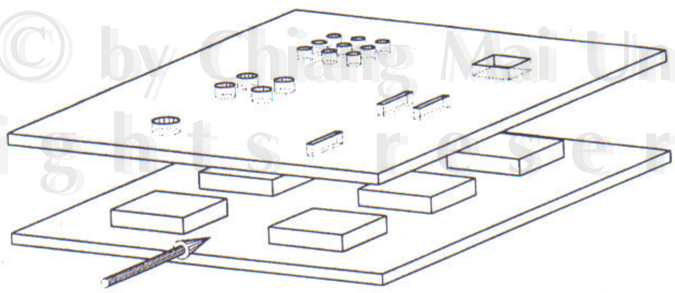

Figure 1.17 Various configurations of round and slot jets, singly and in Arrays.

For round jets, the local heat transfer coefficient on the target surface has a bell-shaped distribution with respect to radial distance from the stagnation point. The maximum value occurs at the stagnation point and decreases symmetrically with radial distance. Under some conditions, secondary maxima are observed in these curves corresponding to the location where the wall jet becomes turbulent [den Ouden and Hoogendoorn, 1974; Martin, 1977] and where the recirculation flow reattaches (Garimella and Rice, 1994). The local heat transfer distribution for a confined and submerged round jet of FC-77 illustrating the secondary peaks for different nozzle to target spacing is shown in Figure 1.18; similar results have been obtained in air (for instance, Huber and Viskanta (1994).

1.3.7 Compact Heat Sinks

One of the most compact and high-performance heat dissipation devices is the micro channel heat sink, first proposed by Tuckerman and Pease (1981,1982 as a technique to lower the convection resistance between the substrate and coolant.

Another example of compact heat sinks is the heat pipe. The advantages of the high heat transfer rates obtained through phase-change heat transfer can be coupled with forced convection using a heat pipe, which dissipates heat by a cyclic process involving a coolant that undergoes evaporation and condensation in a contained unit (for instance, see Dunn and Reay , 1994).

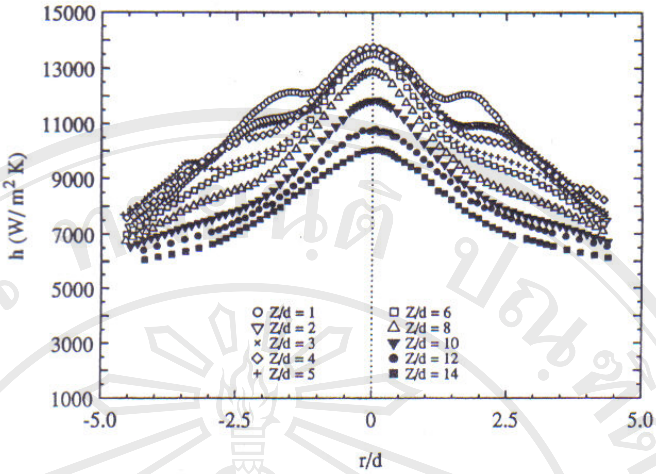

Figure 1.18 Local heat transfer coefficient distribution for a confined, round jet of FC-77 with a diameter of 1.59 mm, at different nozzle to target spacing and a Reynolds number of 13,000 (Garimella and Rice, 1994).

At the normal operating temperatures of chips, typically less than 100 °C, Babin and Peterson (1990) achieved heat dissipation rates of over 125 W/cm^2 with a maximum total power of 65 W. Unlike other enhancement techniques where the heat flux can be used to calculate the power dissipated, the maximum power that can be carried away by a heat pipe in governed by the capabilities of the phase change process. A heat-pipe design for cooling high flux/high power chips was recently explored by North and Avedisian (1993). The design involved a series of holes drilled into a manifold base plate lined with sintered copper powder that served as the wick. With an air-cooled condenser section, a maximum heat flux of 47 W/cm^2 and a total power of 900 W were achieved with surface temperatures under 100 °C

1.3.8 Improved substrate conduction

On approach, which does no fall into the traditional classification of enhancement techniques but is of great interest due to its passivity, is improved substrate conduction. Since the choice of board materials, die-attach and bonding techniques are not primarily governed by thermal management considerations, the approach may not always be practicable. However, if thermal concerns were to be addressed at an early design stage, the improvement of substrate conduction could be considered an enhancement parameter. Antonetti (1990) reviewed the progress made in the development of predictive theory, recent experimental studies, and examples of thermal interface management in current electronic equipment. The theory and the applications of constriction and spreading resistance concepts as applied to microelectronic thermal management were reviewed by Yovanovich (1978). Through their models Culham and Yovanovich (1978) and Negus and Yovanovich (1986) illustrated that the thermal conductivity is the single most important design parameter for reducing board temperatures. A marginal increase in the relative percentage of copper in the board was shown to result in an increased effective thermal conductivity (for board conductivity in the range of 1 to 10 W/mK) and significantly lower board temperatures. The effect on board temperature of the small alterations in circuit-board thickness was shown to be not sufficient to warrant changing the design of the board thickness (which is often not feasible). Thus increasing the chip spacing to a distance greater than twice the chip width would produce little benefit in terms of decreased thermal resistance.

1.3.9 Hybrid Techniques

Some of the enhancement techniques as described before may be used in combination with each other. Air cooling is often supplemented by liquid cooling as the ultimate heat sink. The substrate carrying the integrated circuit chips is mounted within a sealed module-cooling assembly containing a fluorocarbon liquid coolant. Heat from the chips is transferred to the fluorocarbon using internal fins and then to water flowing through an externally attached cold plate.

It could be seen that there are many methods to control the module temperatures and for the heat transfer enhancement techniques, very few research works are reported. This research study is to consider the air cooling parameters affecting the temperature of the electronic modules mounted in printed circuit board. Only the heat convection which is the main heat transfer is concentrated. Moreover, different methods on heat transfer enhancement such as a use of vortex generator and a use of PCB coated with high thermal conductivity. The heat transfer coefficients recommended from the literatures are used to predict the module temperatures.

1.4 Objectives of the Present Study

1.4.1 To find out the parameters affecting the air cooling characteristics in electronic modules mounted on PCB. The parameters considered are the module position, the air velocity and the module temperature.

1.4.2 To develop techniques of heat transfer enhancement for air cooling of electronic module arrays.

1.4.3 To develop a tool for evaluating the positions of chip modules to control the module temperatures.

1.5 Scope of the Study

1.5.1 Each chip module is artificial rectangular aluminum block having a foil-heater inside. The dimensions of each are 1.8 cm x 5.8 cm x 0.6 cm.

1.5.2 The modules are arranged in in-line and staggered arrays.

1.5.3 In case of vortex generator, only the delta-winglet type is considered.