Study of pool boiling enhancement structures and development of two phase cooling solution for electronics package

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**STUDY OF POOL BOILING ENHANCEMENT
STRUCTURES AND DEVELOPMENT OF TWO PHASE
COOLING SOLUTION FOR ELECTRONICS PACKAGE**

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**SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING**

2006

STUDY OF POOL BOILING ENHANCEMENT STRUCTURES AND DEVELOPMENT
OF TWO PHASE COOLING SOLUTION FOR ELECTRONICS PACKAGE

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Study of Pool Boiling Enhancement Structures and Development of Two Phase Cooling Solution for Electronics Package

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A thesis submitted to the Nanyang Technological University
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LIST OF SYMBOLS

C	Specific heat [J/kg – K]
C_p	Channel pitch[m]
C_w	Channel width[m]
CHF	Critical Heat Flux [W/m ²]
D_c	Cavity mouth diameter[m]
d_b	Bubble departure diameter
g	Gravitational acceleration [m/s ²]
h	Convective heat transfer coefficient [W/m ² K]
h_{fg}	Latent heat of vaporization [J/kg – K]
k	Thermal Conductivity [W/m – K]
Nu	Nusselt number
Na	Nucleation site density
P	Pressure [Pa = N/m ²]
P_d	Pore diameter[m]
P_p	Pore pitch[m]
Pr	Prandtl number
q	Heat flux [W/m ²]
Q_{tun}	Latent heat evaporation in the tunnel during bubble cycle[J]
r	Cavity mouth radius[m]
R	Bubble radius[m]
Re	Reynolds number
T	Temperature [°C or K]
t_g	Bubble growth time[Sec]
t_w	Bubble waiting time [Sec]
$?T$	Wall superheat [°C or K]
$?$	Liquid contact angle
d	microlayer thickness[m]
$?$	Condenser fin efficiency
μ	Dynamic viscosity [N.s/m ²]
$?$	Density [Kg/m ³]
s	Surface Tension [N/m]
$?$	Kinematic viscosity ? Specific volume [m ³ /kg]
$?_{fg}$	Difference in specific volumes of saturated liquid and vapor

Subscripts

amb	ambient
$crit$	critical
l	fluid
h	homogenous
sat	saturation
sub	sub cooling
v	vapor

ABSTRACT

Advances in microelectronics design and manufacturing made possible higher transistor density in a single device. IC chips are now being stacked vertically for better electrical performance and compact system design, which leads to high heat flux in the order of $50\text{W}/\text{cm}^2$. Air cooling has poor cooling efficiency for high power application, therefore liquid cooling /boiling heat transfer is preferred due its high heat removal capabilities. Pool boiling heat transfer is investigated for electronic cooling in this research with the target to achieve $50\text{W}/\text{cm}^2$.

The limiting heat flux in pool boiling is called critical heat flux (CHF) and it is $\sim 15\text{W}/\text{cm}^2$ using Fluorocarbon liquids (FC) from a plain silicon surface. A novel method of boiling heat transfer enhancement was developed and characterized. The boiling heat transfer from the silicon surface was enhanced by combination of surface modification and micro structure named “Micro-machined porous channels” (MMPC). Boiling heat transfers from plain and enhanced surfaces were characterized using a pool boiling experimental setup. The porous channels are effective in providing heat from all the sides of the bubble formed on the base surface. The impact of micro structure features like pore diameter, pore pitch, channel width, channel pitch on heat transfer enhancement was investigated. The micro structure with pore diameter equal to the channel width was found to have maximum enhancement among the structures tested. Maximum heat flux of $45.6\text{ W}/\text{cm}^2$ was achieved by the technique developed in the research, which is 300% heat transfer enhancement.

An empirical model is developed to predict the heat transfer over wide range of temperature based on the experimental data and the model predicts with error of 20%.

Design guidelines and boiling enhancement using MMPC structure are derived.

A compact two phase cooling solution for a 40mm BGA flip-Chip package was designed and developed. The micro structure was integrated with the cooling solution and cooling capacity was estimated with and without the micro structure. Maximum heat flux of 41.8 W/cm² is demonstrated by the passive cooling solution with micro structure.

CHAPTER 1: INTRODUCTION

1.1 Motivation

Technological advances in microelectronics design & manufacturing have led to higher transistor density and enhanced functionality from a single Integrated circuit (IC). With higher integration, there is an associated increase in heat flux for various applications. Integration of many transistors for multifunctional system on a chip or in a package is becoming increasingly dependent on the safe limits of heat dissipation from the chip surface. Mudawar (2000) has made a good assessment of high heat flux thermal management schemes. Thermal issues have emerged as an obstacle to electronic packaging at the chip, module and system levels. One reason for this trend is the alarming increase in component concentration in the chip itself through Ultra Large Scale Integration (ULSI) technologies. System in a Package (SiP) technologies is considered to be the fourth of wave of packaging innovation according to the International Technology Roadmap for Semiconductors in 2003 (ITRS). It has the advantage of integrating different IC technologies and functions in the same package for specific electrical, thermal and mechanical requirements. Vertically stacked package is the preferred technology for many SiP applications because of the advantages of smaller size, shorter signal routing, and reduced wiring density. According to the ITRS, cost per pin paradigm doesn't apply to SiP, because of many die to die interconnections within the package. Instead a new metric to measure the area effectiveness is silicon efficiency. Vertically stacked packages have over 250% Silicon efficiency, compared to 100% of silicon efficiency in wafer level package. Vertical stacking is achieved by different techniques viz. die level stacking, package level stacking, and wafer level stacking. Currently, SiP application is limited to die level & package level stacking for lower power applications such as memory devices,

base band, logic devices etc. The industry is looking for such technology to meet the growing demands from cost performance computing and communication devices. However, high module thermal resistance of the vertically stacked package is currently a limiting factor. Power dissipation for cost performance application is about 50 W/cm^2 for the near term according to the ITRS. Since multiple chips are stacked vertically, it difficult to remove the heat from the intermediate devices.

1.2 Present cooling methods

Air cooling is the preferred technique due to its cost advantages and ease of implementation, but it has severe limitations in removing heat from the closely stacked structure of the package. The poor thermal properties of air require forced circulation at speeds, which often approach the upper accepted limits for noise and vibration. Current systems use mounted fans to force circulate air for cooling of the chips as shown Fig. 1.1.

Figure 1.1: An air-cooled heat sink

But even with the finned surfaces and the increased convection speed, air is not likely to handle the high heat fluxes projected for the next generation of devices in the range of $\sim 50 \text{ W/cm}^2$.

Liquid cooling is another attractive option. Water and Fluorocarbons (FC) are used for liquid cooling solutions. Liquid has good thermal transport properties; certainly far better than air. Water has poor dielectric properties, which limits its application to indirect cooling across an electrically isolating, low thermal resistance bridge. The

indirect cooling configurations are currently available for dissipating high fluxes from the device, but factors such as thermal contact resistance between the device and cooling liquid water preclude the dissipating of high fluxes.

Figure 1.2: A water-cooled thermal solution

Failure to meet the harsh thermal requirements of electronic packaging with indirect water-cooling has spurred interest in cooling by direct immersion in dielectric fluids. Direct cooling completely eliminates the thermal bridge separating the device from the coolant. But despite the complete elimination of the thermal resistance separating the device from the coolant, direct immersion is not without shortcomings. Intimate contact of the liquid with the device places stringent chemical and electrical compatibility constraints, limiting the choice of coolant to a few fluids, many of which are chlorofluorocarbons. However the direct immersion cooling is preferred over indirect water-cooling for the low thermal contact resistance, which is a major part of the total thermal resistance in a high heat flux devices. Fig. 1.3 shows the heat transfer attainable with different cooling techniques. Forced convection greatly improves the cooling performance compared to natural convection for all the fluid. Liquid cooling can be applied with or without boiling. But boiling can reduce the device temperature significantly compared to single phase liquid cooling.

Figure 1.3: Cooling technique and its heat transfer

1.3 Proposed cooling solution

Boiling heat transfer has the highest heat removal capacity than other cooling modes, so it requires minimum resources like heat removal space, working fluid etc. Also the direct liquid cooling by boiling produces low thermal interface resistance, if the working fluid is highly wetting and fills through all the narrow spaces in the 3D stacked module. Hence pool boiling technique is proposed for the thermal management of 3D stacked module. A copper casing is designed to contain the 3D stacked module and the dielectric fluid is partially filled in the casing. The dielectric fluid FC-72 is chosen because it has a saturation temperature of 56 °C at atmospheric pressure, which is much lower than the recommended working temperature of the IC devices. When the device temperature goes above the working fluid saturation temperature, the working fluid evaporates by absorbing the heat. The copper casing functions as condenser for the two-phase cooling system. The casing area is typically 10 –20 times larger than the chip area. Therefore area of the inner walls of the casing will be sufficient for the vapor condensation. A suitable air cooled heat sink is

attached on the copper casing to remove the heat. Fig. 1.4 shows the arrangement of the proposed cooling solution for the 3D stacked module.

Figure 1.4: Schematic of proposed two-phase cooling solution for an electronics package

1.4 Objective

The current study aims to investigate the nucleated boiling heat transfer enhancement for electronics cooling and design a novel boiling heat transfer enhancement technique to achieve 50 W/cm^2 . A two-phase passive cooling solution for an high power electronics package will also be developed by integrating the enhanced boiling heat transfer technique with chip.

1.5 Scope of work

The scope of the work comprises of two parts

- a. Development of pool boiling enhancement structure and investigation of boiling enhancement, which includes:
 - Analysis of the factors influencing the nucleate pool boiling heat transfer and designing of boiling structure with heat transfer enhancement at low wall superheat as well as at high wall heat fluxes.

- Design and development of experimental rig for nucleate boiling heat transfer characterization
 - Development of empirical model and design guide lines for the boiling heat transfer enhancement
- b. Development of two-phase cooling solution for an electronics package which includes,
- Design of two-phase compact cooling solution by integrating boiling enhancement structure with an electronics package
 - Estimation of the cooling limits from this solution

CHAPTER 2: LITERATURE REVIEW

2.1 Review of nucleate boiling transfer mechanism

Scientific research into boiling and the industrial use of boiling spans several decades. Extensive data have been accumulated from experimental studies dealing with a diverse array of conditions. Nucleation boiling has been utilized extensively in industry because it is one of the most efficient modes of heat transfer. The reason for this is that nucleate boiling is capable of transferring amounts of energy many times greater than those transferred by convection or conduction. This undoubtedly comes from phase changes on the heated surface. Pool boiling represents the traditional starting point for discussion of boiling heat transfer. The mechanism of pool boiling is relatively understood, however it is not possible to predict heat transfer for this boiling system with the same precision associated with single-phase systems. Fig. 2.1 shows a typical pool-boiling curve from a smooth surface. As the heat input to the surface is increased, sub-cooled boiling occurs. In sub-cooled boiling, natural convection is the main mode of heat transfer. At the certain value of wall superheat, vapor bubble appears on the heater surface, which is called boiling incipience point. The bubble formed on cavities and scratches which preexist with gas/vapor nuclei on the surface. Sudden inception of a large number of cavities at certain wall superheat causes reduction in the heater temperature at the constant heat flux. But this behavior is not observed, when the boiling curve is obtained by reducing the heat flux. The heat flux on the surface during the sub-cooled boiling varies with the wall superheat roughly as given in Eq. (2.1).

$$q \propto \Delta T^{5/4} \dots (2.1)$$

After the boiling inception, the boiling curve slope is increased drastically. In partial nucleate boiling; discrete bubbles are released randomly located at active sites on the heater surface. The density of the nucleation sites and the frequency of the bubble release increase with the increase in wall superheat. The heat flux on polished surfaces varies with the wall superheat roughly as

$$q \propto \Delta T^{2 \sim 5} \dots\dots\dots(2.2)$$

Figure 2.1: Typical nucleate pool boiling curve

The maximum or critical heat flux (CHF) sets the upper limit of the fully developed nucleate boiling for the safe operation of the system. After maximum heat flux is reached, the heater surface is rapidly covered with vapor. The surface is nearly insulated and the surface temperature rises very rapidly. When the rate of heat input is controlled, the heater surface passes quickly through the transition boiling and reaches film boiling. But for electronics cooling the nucleate boiling is most preferred, because of the limitation of the chip safe operating temperature.

2.1.1 Bubbles formation during nucleate boiling

Bubbles are created during nucleate boiling. The liquid evaporation at the bubble–liquid interface is one of the modes that have enhanced the heat transfer to the bulk liquid. The classical nucleation theory is used to explain the fundamental and the hydrodynamic effect during bubble formation. Bubbles exist in the bulk liquid due to attractive forces between molecules at the liquid – vapor interface and the surface tension σ . Bubbles growth are generated by buoyancy force, which cause them to rise up the bulk liquid and hence transfer heat to the surrounding liquid. It has been observed experimentally that in order for bubbles to form at a heater surface, the temperature of the heater surface T_w must exceed the liquid saturation temperature T_{sat} sufficiently; a phenomenon known as heterogeneous boiling.

Nucleation theories can be further classified as either homogeneous or heterogeneous. Several theories based on statistical mechanism have been proposed for homogeneous nucleation in a pure liquid. One approach, presumes that numerous molecules have the activation energy required for existence in the vapor phase. These energetic molecules could combine through collisions to form a cluster, which is formed a vapor bubble. Theories of this type yield high liquid superheat for nucleation in a pure liquid, for example 50 °C for water at 1 atm. Such high superheat is contrary to experimental observations. In real system, the liquid contains foreign particles and dissolved gas that could act as nuclei.

Theoretically, embryos with very small radii exist in the bulk liquid and can serve as nuclei for bubble formation. However for a static bubble to exist in the liquid, the basic requirement would be the thermodynamic balance on the bubble. The forces resulting from the difference between the pressure inside the bubble p_v and the pressure in the liquid p_l acting on its projected area πr^2 must be equate with the

surface tension σ acting on its circumference $2\pi r$, where r is the radius of the bubble. Eq. (2.3) shows the reduced form of the thermodynamic equation for determining bubble radius. Hence a high pressure difference force $\pi r^2 (p_v - p_l)$ would be required to overcome the surface tension force

$$p_v - p_l = \frac{2\pi r \sigma}{\pi r^2} = \frac{2\sigma}{r} \dots\dots\dots(2.3)$$

Using thermodynamic tables, the relationship between liquid temperature difference and surface tension is derived as shown in Eq. (2.4). The pressure in the vapor is approximated as the saturated pressure where $m = \frac{dp}{dT}$. Likewise, the liquid within the vicinity of the vapor nucleus should be highly superheated in order to initiate the bubble upward motion.

$$T_v - T_{sat} = \frac{2\sigma}{mr} \dots\dots\dots(2.4)$$

As the bubbles rise up the bulk fluid, heat flux within the bubble will be transported outwardly to the liquid. When bulk liquid temperature drops below the saturation temperature, the vapor will condense causing the bubble to collapse. However, when the liquid temperature T_l increase to a value higher than the vapor temperature T_v in the bubble, heat will flow into the bubble at the liquid/vapor interface will evaporate and the bubble will grow or eventually depart from the nucleated sites. The Clausius–Clapeyron equation $\frac{dp}{dT} = \rho h_{fg} / T_{sat}$ may be substituted into Eqs. (2.3) and (2.4) to determine the critical radius required for bubble growth. Hence the two equations becomes,

$$\rho_v - \rho_l = \frac{\rho_v h_{fg} (T_v - T_{sat})}{T_{sat}} = \frac{2\sigma}{r_c} \dots\dots\dots(2.5)$$

$$T_v - T_{sat} = \frac{2\sigma T_{sat}}{\rho_v h_{fg} r_c} = \frac{A}{r_c} \dots\dots\dots(2.6)$$

Where the nucleation parameter A is given by $A = 2\sigma T_{sat} / \rho h_{fg}$. This analysis demonstrates that a nucleus having a critical radius r_c can grow into a bubble in a uniformly superheated liquid. Classical nucleation theory is based upon the assumption that a hemispherical nucleus having the same radius r_c as the nucleation cavity at which it is located appears after the departure of the previous bubble and awaits the conditions required to nucleate a bubble. As explained before, this will occur when the temperature of the liquid surrounding the nucleus is sufficiently greater than the temperature of the vapor in the nucleus such that there is a net flow of heat inward, thereby enabling the nucleus to grow.

Consider an idealized conical cavity with a pre-existing gas phase as shown in Fig. 2.2. For a segment of a spherical bubble, this exists in thermal equilibrium with a large volume of liquid. The pressure inside and outside the bubble is P_v and P_l respectively. The net pressure force,

$$P_v - P_l = \frac{2\sigma}{R} \dots\dots\dots(2.7)$$

In order to arrive at a general expression for the superheat required for nucleation, it is necessary to relate T and P along the saturation line.

$$T_w > T_{sat} + \frac{2\sigma T_{sat}}{r h_{fg} \rho_g} \dots\dots\dots(2.8)$$

Figure 2.2: Vapor bubble in a conical cavity

As the wall superheat is raised, the radius of curvature decreases from (1) to (2) as shown in Fig.2.2, the minimum radius corresponds to hemispherical state (3) with the radius equal to the cavity radius. With further increase in wall superheat, the bubble is no longer stable. The bubble nucleates and grows at a rate dependent upon the rate of heat transfer from the surrounding liquid. The wall superheat required for incipient boiling is dependent on the cavity mouth radius. But when the heater is submerged in a pool of liquid, a temperature gradient exists in the liquid adjacent to the heater surface. Nucleation cannot take place from a cavity unless the environment surrounding the bubble is sufficiently high in temperature.

Several approaches have been proposed for determining the incipient wall superheat for boiling from preexisting nuclei. One of the approaches proposed by Hsu (1962) suggests that an embryo will become a bubble if the temperature of the liquid at the tip of the embryo is at least equal to the saturation temperature corresponding to vapor pressure in the bubble (Refer APPENDIX G).

$$\Delta T = \left(\frac{4\sigma T_{sat} \nu_{fg} h_l}{K_l h_{fg}} \right) \left[1 + \left(1 + \frac{K_l h_{fg} \Delta t_{sub}}{2\sigma T_{sat} \nu_{fg} h_l} \right)^{0.5} \right] \dots \dots \dots (2.9)$$

Another approach is corresponding to a critical point of instability of the vapor-liquid interface. The interface is considered to be stable or quasi-stable if the curvature of the interface increases with an increase in vapor volume. Wang & Dhir (1993) studied the instability of the vapor-liquid interface in a spherical cavity and showed that nucleation occurs when non-dimensional curvature of the interface attains its maximum value. They obtained the following relation between wall superheat and diameter D_c of the nucleation cavity

$$\Delta T = \frac{4\sigma T_{\text{sat}}}{\rho_v h_{\text{fg}} D_c} K_{\text{max}} \dots\dots\dots (2.10)$$

where

$$K_{\text{max}} = 1 \text{ for } \phi \leq 90^\circ$$

$$K_{\text{max}} = \sin \phi \text{ for } \phi > 90^\circ$$

2.1.2 Nucleation site density:

The number of sites that become active increases as wall heat flux or superheat increases. Because addition of new nucleation sites influences the rate of heat transfer from the surface, knowledge of nucleation site density as a function of wall superheat is needed for prediction of nucleate-boiling heat flux. Several other parameters also affect the site density, including the procedure used in preparing the heater surface, surface finish, surface wettability, heater material thermo-physical properties, and heater thickness. Little attention had been given to the effect of these parameters on the density of active sites. Kocamustafaogullari & Ishii (1983) correlated the cumulative nucleation site density reported by various investigators for water boiling on a variety of surfaces at pressures of 1–198 atm.

$$N_a \sim \Delta T^{m1} \dots\dots\dots (2.11)$$

Sultan & Judd (1983) studied the bubble growth pattern at neighboring sites during nucleate pool boiling of water on a copper surface. They found that elapsed time between the start of bubble growth at two neighboring active sites increased as the distance separating the two sites increased. They proposed that thermal diffusion in the substrate in the immediate vicinity of the boiling surface might be responsible for this behavior. Their work suggests that some relation may exist between distribution of active nucleation sites and bubble nucleation phenomenon. Judd & Chopra (1993) reported results of interactions between neighboring sites that lead to activation of inactive sites and deactivation of active sites. They noted that for separation distances between nucleation sites less than one bubble diameter at departure, the formation of a bubble at the initiating site promotes the formation of bubbles at the adjacent sites (site seeding). For separation distances between one and three bubble diameters at departure, formation of a bubble at the initiating site inhibits the formation of bubbles at the adjacent site (deactivation of sites). For distances greater than three bubble diameters at departure, nucleation at one location is not influenced by activation at another site. However, the significance of these processes with respect to heat transfer during well-established nucleate boiling on thick heaters is expected to be small.

2.1.3 Bubble dynamics

Bubble growth: Vapor bubble occurs as a result of evaporation all around the bubble interface. The energy for evaporation is supplied from the superheated liquid layer that surrounds the bubble after its inception. The bubble growth process on a heater surface, however, is more complex because the bubble shape changes continuously during the growth process, and superheated liquid is confined to a thin region around the bubble. Mikic et al (1970) obtained an analytical solution for the bubble growth rate by using a geometric factor to relate the shape of a bubble growing on the heater

surface to a perfect sphere and accounting for the thermal energy stored in the superheated liquid layer prior to bubble inception. Since the initial energy content of the superheated liquid layer surrounding the bubble depends on the waiting time, the model shows the dependence of bubble growth rate on waiting time. Another view of the bubble growth is due to evaporation occurring at the base of the bubble in that the microlayer between the vapor-liquid interface and the heater surface plays an important role. Cooper & Lloyd (1969) not only confirmed the existence of a microlayer underneath isolated bubbles formed on glass or ceramic surfaces but also deduced the thickness of the microlayer from the observed response of the heater surface thermocouple. They noted that an expression for local thickness as

$$\delta \sim \sqrt{\nu_l t_g} \dots\dots\dots(2.12)$$

Bubble departure: The diameter to which a bubble grows before departing is dictated by the balance of forces that act on the bubble. These forces are associated with the inertia of the liquid and vapor, the liquid drag on the bubble, buoyancy, and the surface tension. An approximate solution to this equation yielded the bubble diameter at departure

$$D_b = 0.0148\beta \left[\frac{2\sigma}{g(\rho_l - \rho_v)} \right]^{0.5} \dots\dots\dots(2.13)$$

Where β is in degree. The contact angle presents a problem in applying this equation, because a dynamic contact angle seems to be required rather than the static contact angle. Although significant deviations of the bubble diameter at departure with respect to Fritz's expression have been reported in the literature, especially at high system pressures, his correlation did provide a correct length scale for the boiling process. Some investigators disagree about the role of surface tension. Generally,

surface tension tends to push the bubble against the wall and thus inhibits bubble departure. However, Cooper et al (1978) found that in some cases surface tension assisted bubble departure by making the bubble spherical.

Bubble release frequency: Initial growth rate of vapor bubbles is independent of subcooling of the liquid up to a certain degree, after which there is a decrease in bubble growth time as the liquid subcooling increases further. The bubble waiting time increases as the liquid subcooling increases initially and then decreases with further increase in subcooling. This observation suggests that a change in heat transfer mechanism has occurred, as the reversal of bubble waiting time with increasing subcooling occurs at the same degree of subcooling. Also both bubble waiting time and growth time decreases as the level of heat flux increases as reported by Ibrahim & Judd (1985) and Judd (1999)

2.1.4 Nucleate boiling heat transfer

In partial nucleate boiling, or in the isolated bubble regime, transient conduction into liquid adjacent to the wall is an important mechanism for heat transfer from an upward-facing horizontal surface. After bubble inception, the superheated liquid layer is pushed outward and mixes with the bulk liquid. The bubble acts like a pump in removing hot liquid from the surface and replacing it with cold liquid. Combining the contribution of transient conduction on and around nucleation sites, microlayer evaporation underneath the bubbles, and natural convection on inactive areas of the heater, one can write an expression for partial nucleate-boiling heat flux as

$$Nu = f(R_e, P_r) \dots \dots \dots (2.14)$$

It is difficult to obtain any general theoretical method of calculating heat transfer coefficient in nucleate boiling. The boiling heat transfer depends on the physical condition and preparation of the surfaces, how well the liquid wets the surface and how efficiently the liquid displaces air from the cavities. The details of the heat transfer mechanisms may be altered as heater geometry or the angular position of the surface with respect to the direction of gravitational acceleration is varied. For example, on a downward-facing surface, the bubbles may slide along the heater surface for some distance after leaving the nucleation site but before moving away from the heater surface. During the movement of the bubbles along the heater surface, cyclic disruption and reformation of the thermal layer will occur and, in turn, will result in a higher heat transfer rate. In partial nucleate boiling, the downward-facing surfaces accommodate heat fluxes that are higher than those on an upward-facing horizontal surface or a vertical surface. However, at high heat fluxes or in fully developed nucleate boiling, the data for all of the surfaces fall on a single line. This observation indicates that when evaporation is the dominant mode of heat transfer, the orientation of the plate has little effect on dependence of heat flux on wall superheat.

Fully developed nucleate boiling correlations: Numerous attempts have been made to correlate the portion of the boiling curve that is log-linear. The most successful models are semi-analytical and involve a form of dimensional analysis. Rohsenow et al (1956) developed the most widely used correlation for nucleate boiling. Adopting the argument that the major portion of the heat is transferred directly from the surface to the liquid, a conventional combination of dimensionless group is sought. The latent heat transfer of the bubble is represented as

$$\frac{c(T_w - T_{sat})}{h_{fg}} = C_{sf} \left[\frac{q}{h_{fg} \mu_l} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^n \left[\frac{c\mu}{k} \right]^m \quad (2.15)$$

It was possible to correlate data for a wide range of fluids at different pressures utilizing $n = 0.33$ and $m = 1.7$, where C_{sf} has different value for each fluid-surface combination. At this point, it is apparent that prediction of nucleate boiling is difficult without direct information on the fluid and surface of interest.

Tien (1962) proposed a hydrodynamic model of stagnation flow is proposed for saturated nucleate boiling over a flat surface. Through the established analytical results in axisymmetrical stagnation flow, a relation between the heat-transfer coefficient and the thermal boundary-layer thickness induced by rising bubbles is obtained, and a good agreement with measured results in the low heat-flux region is indicated. The predicted heat-transfer result is given as Eq. (2.16). The predicted relation is found to be in agreement with boiling data of most liquids.

$$q = 61.3 P_r^{0.33} k_l n_s^{0.5} \Delta T \quad (2.16)$$

2.2 Peak nucleate boiling theory

The vapor-liquid exchange process that is responsible for much of the cooling effectiveness realized with phase change requires uninterrupted liquid access to the device surface. Unfortunately, higher heat fluxes are dissipated by the production of more vapor bubbles per unit surface area. This bubble crowding leads to significant vapor coalescence that eventually begins to restrict the liquid access to the device surface. Once the vapor-liquid exchange process is interrupted, the power dissipated in the device itself is no longer rejected and the device temperature begins to escalate

uncontrollably. This is the upper-most point and is termed as CHF, which constitutes the upper heat flux design for any phase-change cooling system. There are two important theories for the occurrence of CHF.

Based on Bubble packing: The number of nucleation sites, becomes so numerous that neighboring bubbles or vapor columns coalesce, causing a vapor blanketing of the surface. Rohsenow and Griffith (1956) correlation is given by

$$q_{crit} = 143 h_{fg} \rho_v \left[\frac{\rho_l - \rho_v}{\rho_v} \right]^{0.6} \text{ Btu / hr ft}^2 \dots\dots\dots(2.17)$$

where h_{fg} in Btu/lbm and ρ_l & ρ_v in lbm/ft³

Based on Hydrodynamic instability: At high heat flux, the number of sites is large and the vapor generation rate is very high that, the area between the bubbles columns for liquid flow to the surface is reduced. The relative velocity is so large that the liquid-vapor interface becomes unstable; this starves the surface of liquid and forms a vapor blanket. Zuber (1958) proposed critical heat flux Eq. (2.18) and the hydrodynamic theory breaks down at small values of the dimensionless size R' .

$$q_{crit} = 0.131 h_{fg} \rho_v \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{0.25} \dots\dots\dots(2.18)$$

$$R' = R \left[\frac{g (\rho_l - \rho_v)}{\sigma} \right]^{0.5} < 0.5 \dots\dots\dots(2.19)$$

Macrolayer dryout model: In the macrolayer model, the lateral coalescence of vapor bubbles or columns results in the formation of a vapor slug or vapor mushroom connected to the heated surface by numerous small vapor stems. A thin layer of liquid, called the macrolayer, is then trapped between the vapor mushroom and the heater surface. When evaporation, resulting from heat addition, depletes the liquid layer, the surface dries out and, critical heat flux occurs. Fluid properties such as density, latent heat of vaporization, temperature, specific heat, and heater properties such thickness, density, specific heat, thermal conductivity and surface quality affect critical heat flux. The CHF analysis by the macrolayer dryout model leads to the same result as the hydrodynamic instability model as proposed by Katto (1992), Haramura and Katto (1983)

$$\frac{q}{\rho_v h_{fg}} = Const \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{0.25} \dots\dots\dots (2.20)$$

Model based on Contact angle: The effect of contact angle on CHF is not included in the above models. Experimental investigation shows that CHF reduces significantly as the contact angle increases, which leads to believe the CHF is linked to the liquid –vapor-solid interaction at the wall. Kandlikar (2002) proposed a new model including the contact angle. The critical heat flux condition is reached when the recoil force overcomes the combined surface tension and hydrostatic forces along the heater surface.

$$q_{crit} = h_{fg} \rho_v^{1/2} \left[\frac{1 + \cos \theta}{16} \right] \left(\frac{2}{\Pi} + \frac{\Pi}{4} (1 + \cos \theta) \cos \phi \right)^{1/2} [\sigma g (\rho_l - \rho_v)]^{1/4} \dots\dots\dots (2.21)$$

where : Contact angle θ ; orientation angle of heater with horizontal ϕ

2.3 Parameters affecting the CHF

Effect of Pressure: System pressure distribution has effect on the CHF of the test surface as a higher system pressure increases the vapor density in the fluid. This increase in vapor density is capable of sustaining higher CHF. Anderson and Mudawar (1989) verified the Zuber's correlation through experimentation with a heated silicon surface immersed in FC-72 and varying system pressure between 1 to 3 atm. The CHF for FC-72 at atmospheric pressure was found to be 15.01 W/cm^2 and a 20% improvement of CHF was noted with increased system pressure. Also Bar-Cohen and McNeil (1992) measured about 25% increase for 1-to 3 atm changes.

Effect of dissolved gas: Dissolved gas in the working fluid enhances nucleate boiling heat transfer. You et al (1995) had shown experimental results that incipient temperature is lower with high level of gas-saturated liquid having 0.0056 mole/mole. A hysteresis is observed for a moderate gas-saturated liquid. The CHF increases with higher level of dissolved gas at a constant total pressure. With stirring, boiling incipient temperature is a few degrees above the saturation temperature and the hysteresis is eliminated.

Effect of liquid subcooling: Elkassabgi & Lienhard (1988) investigated maximum heat fluxes during subcooled pool boiling on horizontal cylinders. They identified three subcooling regimes. For low subcooling, the maximum heat flux varies linearly with subcooling in a manner. At moderate subcooling, bubbles surrounded the heater without detaching. For these subcooling levels, the maximum heat flux varied slightly nonlinearly with liquid subcooling and was determined by natural convection from the outer edge of the bubble boundary layer. At high subcooling, the maximum heat flux was independent of liquid subcooling and was limited by the evaporation rate at the heater surface and not by the rate at which energy could be removed by natural

convection from the outer edge of the bubble boundary layer. Experimental investigation with increasing degree of liquid subcooling on CHF have been conducted by Ivey and Morris (1966) and they proposed a correlation between CHF at liquid subcooling and the CHF at saturation. The correlation show that the CHF at different degree of subcooled liquid depends on the liquid characteristics and the Jakob number (Ja).

$$CHF_{sub} = CHF_{sat} \left[1 + C \left(\frac{\rho_g}{\rho_f} \right)^{0.25} Ja \right] \text{-----} (2.22)$$

Where C is a constant value and is part of the fluid characteristic and Jakob number is the ratio of sensible to latent energy absorbed during phase change. Hwang and Moran (1981) determined the C value of 0.061 for a 5mm silicon heater immersed in FC-72, while recent studies with silicon heater and FC-72, the value for C was found to lies within 0.05 to 0.065 for FC-72 at liquid subcooled less than 25 °C.

Liquid wettability: Chowdhury and Winterton (1985) investigated the liquid surface tension effects and liquid wettability in boiling heat transfer by using simple quenching techniques with metal cylinders immersed in saturated water or methanol. Emphasis was placed on the surface energy of the liquid by measured by the contact angle. It is shown that contact angle has a very strong influence on nucleated boiling, improved wetting increased heat flux and CHF at a given superheat. Some researchers have proposed using nonwetting coating which establishes a larger liquid – vapor interface radius within the cavity, thus requiring less superheat for its existence.

2.4 Review of boiling heat transfer enhancement

The heater temperature has to be higher than the boiling point (T_{sat}) of the working fluid for boiling, which is termed as wall superheat. It is inversely proportional to the nucleation cavity radius. Thus the larger the radius, the smaller is the nucleation

superheat. But Hsu (1962) has shown that only a range of radii could be effective from a plain surface in nucleation boiling. Large nucleation radii extended beyond the thermal boundary layer may not be active. Hence trapping the vapour larger than the theoretical limits is done by surface modification. Generally boiling enhancement by surface modification can be broadly categorized into two methods: (1.) Surface promoters by modifying the surface using sand blasting/ machining /acid etching or Sintering and spraying particles; (2.) Surface modification by micro-machining. The advantages of various boiling enhancement techniques are to reduce the boiling incipience temperature and increase the CHF. This review is done to understand the factors influencing the nucleate boiling and techniques to achieve better heat transfer performance by nucleate boiling.

2.4.1 Boiling enhancement by surface modification

A surface can be modified for boiling enhancement by changing the surface topography and surface chemistry. Surface roughness is the most important factor in boiling heat transfer as it closely relates to the nucleation site for bubble generation. However, it is difficult to relate the effects of surface roughness on nucleation because nucleation does not merely relate to the geometric size of the roughness but also to aging, surface chemistry, surface-liquid combination and wettability. The boiling heat flux from the surface can be enhanced by various surface treatments like sand blasting, acid etching, coating of fine particles etc. Benjamin & Balakrishnan (1997) showed that as micro roughness (R_a) increases, the nucleation site density decreases and then increases for a given heat flux. Aged heater requires less wall superheat than fresh heater. Dhir (1992) compared boiling performances on silicon chip with different roughness. The surface roughness fabricated by chemical vapour deposition of SiO_2 is R_a 1.1, 18.7 and 309.3 nm. Comparison of experimental results showed

that the surface roughness was effective in enhancing nucleate boiling, but the heat transfer performance was not directly related to the surface roughness. Maximum heat flux of 22.2 W/cm^2 is achieved for the maximum chip temperature of 85°C .

The hydrodynamic theory predicted maximum heat flux for well wetted surfaces. But large deviation of data is observed with the different surface wettability. Liquid static contact angle is used as a measure of wettability. Active nucleation site density increases with increase in contact angle. The contact angle is changed by the degree of oxidation of the surface. Webb (1994) shown that a thin coating of nonwetting material may improve the nucleating characteristics of water on a boiling surface. A nonwetting coating on the interior wall of the nucleating cavity will enhance its stability as a nucleating site. Thus a lower temperature will be needed to activate the cavity. The nonwetting coating establishes a larger liquid –vapor interface radius within the cavity, which requires less superheat for its existence. Experiments are done on single conical cavity 0.08mm in diameter, formed by pressing a needle into the boiling surface. It is easier to maintain boiling and get reproducible results from the paraffin cavity than from a clear surface. Gaertner (1965) performed further work with artificial nucleation sites, covering the inside surface of the cavities with a nonwetting material. Water boiled on a variety of surfaces having closely spaced nucleation sites formed by needle sharp punches and parallel scratches. The lower surface energy material is coated over the surface containing artificial sites and then removed from the flat face by abrasion, leaving a thin film of the material deposited in the cavity. The coated cavity surface boiled at a lower superheat and remained active for a much longer time.

Spraying and painting are other methods of surface treatment for boiling enhancement tested by O'Connor et al (1995). The enhanced surfaces are fabricated by baking the

particles onto the surface structures. Porous metallic coating is considered to be the most viable enhancement technique. Particle size, particle shape, coating thickness and porosity are parameters, which influence the boiling heat transfer. Preferred coating thickness is in the range of 3-4 particle diameters. Optimum cavity geometry and high nucleation site density are important for the boiling heat transfer enhancement. Microstructures using sprayed alumina 0.3-0.5 microns, painted diamond 1-3 microns and 8-12 microns have been tested on a silicon chip. Boiling incipience temperature is up to 80% lower than a plain silicon surface. The CHF increases with increase in particle size. A maximum of 103% increase in CHF is possible from the surface painted with diamond particles. A combination of liquid subcooling and diamond coated surface produced around 224% increase in CHF.

2.4.2 Boiling enhancement by microstructures

Anderson & Mudawar (1989) used a regular pattern of seventeen circular holes of 0.36 mm diameter on 0.72 mm centre was drilled in horizontally, vertically and inclined arrays using FC-72 for boiling enhancement. The boiling began at the edge of all the three surfaces, which indicates no moderate vapour embryo, existed within the hole. The magnitude of CHF and the associated superheat for the horizontal and vertical cavities were almost the same as a plain vapour blasted surface, showing no effect on the particular pattern. But the surface with inclined (30°) holes showed 25% increase in heat flux. Similarly 10 microns holes on 1 mm, 2 mm and 4 mm pitch drilled in an inline and triangular pattern was tested by Surapong Chatpun et al (2004). The smaller the pitch, the higher the rate of bubble generation was reported. For the same cavity spacing, the inline pattern was found to enhance the heat transfer better than triangular pattern.

Rainey et al (2000 & 2003) tested horizontally oriented pin fin structures using FC 72 at atmospheric pressure. The dimensions of the fins used in the study are 1 mm x 1mm and height of 0 to 8 mm. The heat transfer performance is enhanced with the square pins up to a fin length of 4 mm. Above 4 mm, the surface temperature at the fin tip is too low for boiling. The nucleate boiling and CHF performance of the 0° orientation (horizontal base / vertical fins) is found to be slightly better than the 90° orientation (vertical base/horizontal fins) due to a major portion of the finned surface area in 90° orientation facing downwards. The fins are also coated with micro-porous coating using 1-20 microns aluminium particles. Micro-porous coated fins of 4 mm length have CHF of 74.1 W/cm² and the surface plain fins of the same length has CHF of 71.3 W/cm². The slope of the heat flux curve of micro porous surface fins change at about 50 W/cm² for the tested fin lengths.

Guglielmini et al (1996) tested copper pin-fin structures in horizontal and vertical orientation using dielectric fluid. The fin size tested were 0.4 mm and 0.8 mm and length of 3mm. Nucleate boiling heat flux increases as the pressure increased from 0.5atm to 2atm. The boiling hysteresis seems to increase with the rise in test pressure. Heat transfer co-efficient for vertically oriented surface is better than horizontal surface. Heat flux at wall superheat of 15 K is 2.5-3 times higher for extended surfaces than plain surface. At low wall superheat (9 °C) the heat flux from the extended surface is 25-30 times higher than the plain surface.

Figure 2.3: Micro-fins for boiling enhancement

Honda et al (2002) tested micro pin-fin structures with submicron roughness in FC72. The fin size is 50 x 50 microns and 60 microns height as shown in Fig. 2.4. The RMS roughness of the fins is 30 nm. The fins with sub-micron scale roughness showed a higher heat transfer than plain fins. Photographic observation revealed that a small amount of vapour is left within the gap between the fins. The critical heat flux is 2.3 times larger than a plain smooth surface. Critical heat flux almost increased linearly with liquid sub-cooling. Gas dissolved FC 72 showed lower boiling incipient temperature. In the non-boiling region, micro-pin-fin structures follow almost the same heat vs. wall superheat curve as plain surface. This indicates that the height of the fin is smaller than the thermal boundary layer.

Figure 2.4: Micro Pin-fin structure for boiling enhancement

Miller et al (1990) reported micro hexagonal dimples on a silicon surface for heat transfer enhancement in nucleate boiling in FC-72. The hexagonal dimples of 9.5 microns diameter and 3.3 microns deep on a 12.7 mm x 12.7 mm silicon surfaces are

used for the experiments. Boiling incipience temperature of around 4 °C has been observed for the immersion time of 2 days and greater. But wall superheat of 15 °C is required for a mirror polished surface over 64 hours as observed.

Figure 2.5: Micro-dimple for boiling enhancement

Kubo et al (1999) reported boiling enhancement with re-entrant cavities with 1.6 microns and 3.1 microns mouth diameter at a pitch of 0.1mm and 1mm. All the variations showed almost the same heat transfer performance in the non-boiling region using FC-72. Boiling incipience temperature is about 10 K for both the chips at liquid subcooling of 3 K. The heat transfer performance of the enhanced surfaces is about 16 W/cm². The heat transfer performance of the gas dissolved FC 72 is higher than of degassed FC-72.

Figure 2.6: Re-entrant cavity for boiling enhancement

Ramaswamy et al (2003) have tested multi-layers of micro fabricated structures for boiling enhancement in FC-72. The structure had channels cut on top and bottom side of a silicon or copper block in 90° orientations as shown in Fig. 2.7. The depth of the channels is more than half of the block thickness, so it formed square pores at the intersection. Pore size (channel width) was varied from 90 – 320 microns and tested. As the pore size increased, the heat transfer performance improved at low heat flux of 4-20 W/cm². But, at high heat flux the performance was similar for all the pore sizes. Effect of pore pitch was studied by varying the pitch from 0.5 - 2.1 mm (pore size 150 microns). The smaller the pores pitch, the better the heat dissipation. But at high heat flux (>30 W/cm²) percentage improvement with the smaller pore pitch was minimum. A multi-layer arrangement of the enhancement structures were also tested, but the heat dissipation was not better with multi-layers. Around 10 °C, wall superheat, the heat dissipation from a three-layer stack was almost the same as single-layer performance.

Figure 2.7: Micro-fabricated structure for boiling enhancement

CHAPTER 3: DEVELOPMENT OF THE BOILING ENHANCEMENT TECHNIQUE

3.1 Analysis of boiling enhancement techniques

Heat flux from a 3D-stacked module is in the range of 50 W/cm^2 for a cost performance application. This research is aimed at designing a boiling enhancement structure for cooling 3D stacked module. Also it should be easy to manufacture and have better heat transfer performance. It is seen earlier that boiling from plain surface is mainly due to heterogeneous or cavity nucleation theory. A certain amount of air or other gases are dissolved in the liquid, which leads to nucleation earlier than the homogeneous nucleation theory. But the amount of gas content in a continuous boiling system is not the predominant mode for the nucleation. Only the surface irregularities / cavity play a major role in nucleation from any plain surfaces. Effect of surface condition has been studied for the nucleate boiling and reported by different researchers. Statistical parameters of the surface like R_a or R_q did not correlate well with the boiling performances from plain metal surfaces. The boiling incipience and critical heat flux is different for different preparation method of the surfaces like machining, acid etching etc. The surface cavity shape and distributions are important for the nucleation rather than the mean value of the surface peak and valleys. Machined surfaces have regular peak and valleys by the cutter movement. It is not very advantageous to the nucleate boiling. But random orientation of the surface micro-cavities as in the acid etched surface is found to enhance the nucleate boiling better. Fig. 3.1 shows a photographic view of the machined and acid etched stainless steel surface tested by Klemen (2003)

Figure 3.1: Surface modification for boiling enhancement

Die attach epoxy is commonly used in electronics packaging industry to attach silicon to the PCB. It is a polymer adhesive, which is an organic adhesive that consist of long chains of thermosetting polymers and are widely used as die attach material. The polymer adhesives also contain a large percentage of silver fillers for good thermal conductivity. The silver particles are dispersed quite uniformly in the epoxy; which could modify the surface effectively for nucleate boiling. An SEM image of the die attach epoxy coated silicon surface prepared for this research is given in Fig. 3.2. The modified surface contains randomly distributed artificial pores, which are suitable for nucleate boiling heat transfer enhancement.

Figure 3.2: SEM image of die attach epoxy coated silicon surface

One of the difficulties associated with the use of dielectric liquids like FC-72 is its high wettability, which requires high wall superheat for boiling incipience. The

micro-porous surface helps to initiate the boiling at lower wall superheat temperature. The coated surface increases the probability of nucleation sites, which remain active for repeated cycles of heating and cooling. Furthermore the surface contains cavities with larger effective mouth radii than natural surfaces, thereby initiating boiling at lower wall superheat. Literature review showed that boiling from a silver particle coated aluminum surface was much earlier than the plain surface. Boiling incipience temperature was up to 80% lower for the coated surfaces than the plain surfaces. The adhesive contains microns size particles filled up to 80 % by weight and the particle size varies from 1 micron to 5 microns. It can be easily dispensed on the chip surface using conventional packaging processes. The adhesive is also useful for the attachment of any additional boiling enhancement structures.

Boiling in an electronics system is a cyclical process, the heating and cooling cycles are repeated many times due to switching of the ON or OFF state of the system. Hsu (1962) modeled the boiling process for a cavity in a smooth surface with the vapor situated in the thermal boundary layer. Only a range of radii of vapor nuclei could activate under these conditions. Those with a large radii extending beyond the thermal boundary layer will not activate. Re-entrant cavities, either natural or artificial are good for the nucleate boiling enhancement. Because of the high thermal conductivity of the micro structures relative to the liquid, reentrant cavities trap nearly uniformly the superheated liquid. Consequently their nuclei with large radii are not limited by the Hsu's (1962) model, which enhance the boiling heat transfer significantly. Literature review showed that micro-holes and reentrant cavities with micron size mouth opening produced limited boiling enhancement. Therefore a porous channels with large mouth opening is proposed in this research.

The heat transfer phenomena and mechanisms responsible for boiling enhancement are peculiar to the geometry. Heat can leave the surface either as latent heat or sensible heat in the superheated liquid. Latent heat can be transferred both inside the channels and outside the surface. The liquid passing in and out of the channels can transport sensible heat. Arshad and Thome (1983) carried out boiling visualization using water in rectangular and triangular covered channels. They observed the process of nucleation as follows: 1. Formation of thin liquid film, 2. Nucleate boiling, and 3. Film dry out. They observed that nucleation always took place in the corner of rectangular or triangular channels, but never at the pores in the cover plates over the channels. But in this research the modified bottom surface should also participate in nucleate boiling in addition to the channel corners of the micro structure.

Nakayama et al (1982) studied the two-phase boiling process inside the porous channels and proposed three modes of boiling in the similar structures as shown in Fig. 3.3. The modes are:

- Dried up mode (D-M): the tunnel space is filled with vapour and the heat transfer mechanism is similar to boiling from a plain surface with artificial pores.
- Suction- Evaporation mode (S-E): Gives highest heat transfer due to the evaporation of liquid from the tunnel walls and the tunnel is continuously fed through the inactive pores.
- Flooded mode (F-M): Most of the tunnel space filled with liquid and the active pores operate as isolated nucleation site. Hence pore size and arrangements are important to have Suction-Evaporation mode of boiling in such enhancement structures.

Figure 3.3: Boiling mechanism in a porous surface

Some researchers have tested plate fins, pin-fin structures, “V” grooves, etc for boiling enhancement. However the top side of the open channels doesn’t supply sufficient heat for the bubble growth. Moreover it is difficult to modify the bottom surface of the open structures. Some researches have fabricated short pin-fin structures on a silicon chip with micro-roughness walls. The wall micro-roughness did not enhance the boiling performance. Therefore a closed section with porous surface in combination with modified bottom surface is proposed in this research.

3.2 Design of novel boiling enhancement structure

A novel boiling enhancement structure called “Micro Machined Porous Channels” (MMPC) has been designed. The structure has rectangular micro-channels etched on the bottom side of the silicon wafer. A regular pattern of micro-holes is etched from

the topside of the wafer, which open the channels and forming a porous structure. Fig. 3.4 show the enhancement structure designed for this research

Figure 3.4: Schematic of boiling enhancement structure: Micro Machined Porous Channels

The designed porous channel feature such as channel width, channel pitch, pore size and pore pitch influence the boiling performances. The channel width has to be small enough to restrict the bubble size. Channel width of 0.3 mm and 0.5 mm are chosen for this research. The channel pitch is not that critical for closely spaced structures. But the pore size and pitch are critical for bubble growth and release. Therefore many variations of the pore size and pitch are incorporated in the design. The boiling enhancement structure is fabricated by dry etch techniques like Reactive Ion Etching (RIE). Only two masks are required to fabricate the structures, one mask is for the micro-channel etching and the second is for etching the holes. As RIE is a batch process, cost of fabrication is less compared to machined microstructures.

Figure 3.5: Design parameters of boiling enhancement structure

Ten types of boiling enhancement structures are fabricated. The effect of pore diameter, pore pitch, channel size and channel pitch were investigated and guidelines were generated. The structures are named as MMPC 1, 2 etc for an easy identification. Table 3.1 shows the structure dimensions and identification numbers.

Table 3.1 Boiling enhancement structure details

Structure identification	Channel height (m)	Channel width (m)	Pore size (m)	Channel pitch (m)	Pore pitch (m)
MMPC 1	0.00037	0.0003	0.0002	0.0005	0.0008
MMPC 2	0.00037	0.0003	0.0002	0.0005	0.001
MMPC 3	0.00037	0.0003	0.3/0.1	0.0005	0.0015
MMPC 4	0.00037	0.0003	0.0003	0.0005	0.0008
MMPC 5	0.00037	0.0003	0.0003	0.0005	0.001
MMPC 6	0.00037	0.0003	0.0003	0.0005	0.0015
MMPC 7	0.00037	0.0005	0.0003	0.001	0.0008
MMPC 8	0.00037	0.0005	0.0003	0.0007	0.0008
MMPC 9	0.00037	0.0005	0.0004	0.0007	0.0008
MMPC 10	0.00037	0.0005	0.0005	0.0007	0.0008

The structure is attached on a boiling surface using a thin layer of die attach epoxy (Sumitomo Bakelite: CRM-1295SA) is coated on the die surface. The epoxy is spread uniformly on the die surface using a screen and squeeze. Then the structure is places on the coated surface using a pick and place machine (SRT). Then it is cured in an oven for 1 hr at 150 °C as recommended in the material data sheet. The cross section of the attached structure is shown in Fig.3.6

Figure 3.6 Cross section of boiling enhancement structure

3.2.1 Thermal interface resistance across boiling enhancement structure and chip

Thermal interface resistance across the boiling structure and chip should be small, so that temperature differences between the chip and boiling enhancement structure is small. The structure is attached using high thermal conductive adhesive, which has thermal conductivity of 10 W/mK. A thin layer of epoxy is applied (25 microns) for minimum thermal resistance across the interface. Fig. 3.7 shows thermal resistance network across the interface. Combined thermal interface resistances across the interface ($R_1 + R_2 + \text{etc.}$) are calculated based on one dimensional heat conduction.

Interface thermal resistance = Number of channel wall * Interface thickness /
(Thermal conductivity of interface material * Area of interface)

Figure 3.7 Heat flow path across boiling enhancement structure and chip

The heat generated in the chip is partly used for boiling the liquid on the chip surface and the remaining heat is conducted through the enhancement structure. It is assumed that 50% of the heat generated in the chip is conducted through the structure walls. Thermal interface resistance between the chip and the structure is 0.04 °C/W, which is only 4% of the total thermal resistance. The temperature drop across the interface for 30 W power is less than 1.5 °C.

CHAPTER 4: EXPERIMENTAL STUDIES

4.1 Design of Pool Boiling Experimental Setup

An experimental rig is designed to test the boiling enhancement by the micro-machined porous channels and compare its boiling performance with a plain silicon surface. Fig. 4.1 shows the schematic view of the experimental setup designed for this study. A rectangular aluminum tank with dimension 150mm X 150mm X 200mm and 5mm thick is used for the main body of the experimental setup. Aluminum has high thermal conductivity; which helps to maintain the uniform bulk fluid temperature. The tank is sufficiently large as compared to the test surface. Cylindrical tank would provide a more symmetrical design, but it is difficult to attach a viewing window on a curved surface. In view of this, a rectangular tank with a square base has been designed to reduce the fabrication difficulties. A 10mm thick aluminum plate is welded to the bottom side of the tube. The bottom plate has 70mm diameter opening to mount the boiling surface. Two 70 mm diameter opening are provided on the side walls of the tank and transparent polycarbonate windows are fixed to observe the boiling process. The pool is heated to the test condition by a “U” shaped AC cartridge heater of 200 watts capacity. The heater is supplied with heating power by an AC source through a PID controlled switch. The heater is mounted 5 cm above the boiling surface, so that the liquid movement around the heater does not disrupt the fluid near the boiling surface. The exterior of the tank is shielded with insulating foam to prevent heat loss and to have minimum fluctuation of the bulk temperature.

Figure 4.1: Schematic of experimental set-up for nucleate boiling characterization

A water-cooled cold plate is mounted on the topside of the tank. The cold plate is used to prevent pressure built-up within the rig by condensing the vapor generated from the test surface. It is made of 12.5mm thick copper block with “U” shaped built-in water channels. It acts like a heat exchanger, removing heat from the boiling tank through the use of circulating water within the channels. Heat is removed from the interior of the tank by condensation of the vapor contacting the inner surface of the condenser. This design of the condenser is based on the rate of condensation required with the following parameters; water flow rate and inlet temperature. The detailed design of the condenser is shown in Fig. 4.2.

Figure 4.2 Schematic of the condenser

The efficiency of the condenser is assessed through a theoretical calculation on the rise in temperature of the coolant. The heat input from the auxiliary heaters is used to maintain the temperature of the bulk fluid and does not create any vapor during its operation. Hence the heat input to the auxiliary heater balances with the amount of heat loss from external surfaces of the rig. By the conservation of energy, the heat supplied to the test surface is equal to heat removed by the condenser. Maximum heat input to the test surface is 50 W/cm^2 , if the flow rate of the cooling water of 1.5 liters/min is used during the experiments, then the rise in water temperature is only $\sim 0.5^\circ\text{C}$. The small temperature rise in the coolant indicated the effective use of the condenser in the test facility.

Two “K” type sealed thermocouples 24 AWG are inserted into the tank. One thermocouple is attached very near the boiling surface, which is connected to the PID switch to control the pool temperature. Another thermocouple is placed 10mm above the boiling surface to monitor the pool temperature. A pressure gauge is mounted on the cold plate to monitor the test pressure during the experiments. A $\frac{1}{4}$ ” stainless steel venting port is provided in the cold plate is to release the trapped air in the tank.

The cold plate is fixed to the tank by threaded screws and a 5 mm thick gasket is provided to isolate it thermally from the hot tank surface.

Figure 4.3: Photograph of experimental rig

An experiment was conducted to assess the temperature isolation from the condenser to the tank. The pool was maintained at 53 °C and the experimental set-up external temperature profile was measured using Infrared camera (AVIO 610). The effectiveness of the condenser and temperature isolation by the thick rubber gasket is clear from the temperature plot. The use of a thick rubber gasket between the boiling tank and condenser is aimed at reducing the heat transfer from the heated aluminum tank to the condenser. The condenser has a high rate of heat removal, and heat will be lost if there is strong interaction between the condenser and the boiling tank. The condenser is mainly to cool the vapor inside the tank. The IR image as shown in Fig.4.4 was captured during the experiment. The temperature profile between the condenser and the boiling setup when surface's insulation is removed. The red color

indicates temperature ranging between 50 °C to 55 °C while a light blue color code indicates temperature between 35 °C to 30 °C. The large temperature variation of 20 °C across the condenser indicates that thermal isolation exists between the tank and the condenser.

Figure 4.4: IR image of the experimental rig

4.2 Boiling surface preparation

The main criterion for the selection of heaters for the test surface is to enable maximum heat transfer during the experiment. In addition, the test surface should have temperature sensors to record surface temperature. Some researchers used heaters and thermocouples that are attached in the test surface. Direct attachment of thermocouples on the boiling surface results some amount of heat conducted by the sensors before the measurement which affects the overall accuracy of the readings. Therefore a special test die (Thermal die) is used as boiling surface in this study. The thermal die is similar in construction and the surface condition is similar to the electronic chip. The thermal die dimensions are 12 mm x 12 mm x 0.7mm. The die contains planar poly-silicon resistor; which can generate heat flux in the range between 1 W/cm² to 100 W/cm² using DC power. The thermal die also contains PN

diodes for temperature sensing. These diodes have temperature sensitive parameters that varies linearly with the temperature; allowing temperature measurement within the solid-state devices without any mechanical contact. Based on work by Bar-Cohen and McNeil (1992) a heater thickness of less than 1mm is capable of providing CHF data free of thermal conductance and capacitance effects.

The test die is mounted on a low conductive substrate of 40 mm x 40 mm x 0.8 mm thickness using tiny solder bumps (100 bumps). The substrate is attached to a FR4 printed circuit board (PCB) to bring all the electrical connections outside. Fig. 4.5 shows the boiling surface assembly. Silicon being a good thermal conductor, the heat generated at bump side of the die is quickly spread to the top die surface. PN junction diodes implanted in the die surface represents the boiling surface temperature, as the die thickness is only 0.7 mm. One temperature sensor is placed at the center of the die and another temperature sensor is placed at 3mm away from the die center. Two sensors are given mainly to verify the accuracy of the solid-state sensor. However for all the heat transfer calculations are done based on the sensor located at die center.

Most of the heat dissipated in the test die flow to the boiling surface as the heat transfer co-efficient is high and the die thermal conductivity is 150 W/mK. However a small percentage of heat is conducted through the substrate and PCB. Single-phase numerical analysis is conducted to estimate the heat loss through the PCB and it is found to be less than 2-4% for different heat flux. The details of the analysis and numerical models are explained in APPENDIX C.

Figure 4.5: Photograph of boiling surface

Surface topography impacts the nucleated boiling and affects the result of boiling incipient temperature and CHF. The surface roughness is measured by “Veeco” laser surface profile meter and contact angle of the boiling surface is measured by droplet test. The surface topography of the silicon thermal die is investigated using Scanning Electron Microscope (SEM). The surface roughness of four thermal dice is measured and the roughness R_a is consistent with the surfaces measured. The root mean square and average value of the surface roughness is $\sim 100 \times 10^{-6} \text{ m}$

Figure 4.6 : (a) Surface roughness by Laser profilometer (b) SEM photo of plain silicon

The wetting behavior of FC-72 on the silicon surface is investigated by measuring the contact angle. A drop of the liquid (FC-72) is deposited onto the test surface. A focus lens with built-in grating is used to capture the droplet contact angle when comes into contact with the silicon die. A detailed procedure and instrument set-up is given APPENDIX J. The contact angle of FC-72 on silicon surface was found to be 16° , which is very close to the supplier 3M data.

4.3 Experimental setup calibration

4.3.1 Assembly of experimental setup

The detailed assembly of the boiling setup comprises the boiling tank, condenser, PCB with thermal die, auxiliary heater, pressure gauge, and thermocouples as illustrated in Fig. 4.5. Thick rubber gaskets were placed between components to prevent leakage of fluid/vapor during the experiment. This experiment requires two sets of electrical circuit (i) Circuit to preheat the bulk fluid to a specified temperature and to maintain the temperature of bulk fluid. This circuit requires a closed loop temperature controller to control the fluctuation of pool temperature, (ii) Circuit to supply heat to the boiling surface (Thermal chip) for the boiling characterization.

The circuit 1 is broken down into 2 sub-circuits; the first set which include a thermocouple, PID controller and a power source; a second set that comprises a pair of auxiliary heater, a 120VAC power source, an ammeter and a thyristor. These two circuits are interfaced with a control signal from the PID controller to the thyristor. These two sub-circuits are required to maintain the pool temperature at a constant value. The first subset circuit is used to measure the temperature of the bulk fluid and the temperature controller updates the thyristor at an interval of 30sec. If there is a discrepancy between the preset temperature value and the measured temperature

recorded by the thermocouple, the PID controller sends a signal to the thyristor to regulate the current supplied to the auxiliary heater.

Figure 4.7: Electrical connection schematic to the experimental setup

The PID temperature controller is powered by a 240 VAC source while the K-type thermocouple is attached to the input point of the controller. The PID controller compares both the measured value and the set temperature readings and output a signal to the thyristor, which controls the amount of current supplied to the parallel auxiliary heaters. The circuit 2 comprises of a DC power supply. The DC power source supplies a heating source to the thermal die. Also a source meter is connected to measure the voltage of the diode while biasing a small current (1mA) to measure the temperature.

4.3.2 Preheating of the pool

The tank is filled with FC-72 until the auxiliary heater is submerged. Next, the required pool temperature is set in the PID controller with default control parameters and 240VAC power is supplied to the auxiliary heater to raise the bulk temperature close to the specified value. This step happens in less than two minutes. Then the PID controller is set to auto tuning mode to control the supply to the heater. The default settings for the PID temperature controller are Proportional gain **K**: 5, Integrative term **I** : 60 , Derivative term **D** : 15

4.3.3 Experimental procedure

Firstly all the temperature sensors are calibrated by comparing with K type thermocouple which is calibrated and traceable to national standard. The boiling surface is cleaned using IPA solution and all the electrical connections are soldered to the PCB. The surface roughness and profile has been measured using SEM & Optical surface profiler.

Two “U” cartridge heaters and viewing windows are assembled to the tank and leak tight joint is achieved using suitable gaskets. The boiling surface is assembled at the bottom of the tank with thick gasket using M6 screws. FC – 72 is the cooling fluid used for the experiments and ~1.5 liters of the liquid is filled in the tank. Finally the tank is sealed using the water-cooled cold plate. The cold plate is fitted with a pressure gauge and two sealed K type thermo-couples.

A 100V-AC source is connected to the cartridge heater through a PID controller. A K type thermocouple inserted into the bulk fluid is connected to the controller as a feed back sensor. The bulk fluid is heated to a desired temperature and it is maintained at the temperature by the PID controller.

A high capacity DC source (Xantrex: XHR150-7 with Maximum of 7Amps & 150V) is connected to the heater in the boiling surface. All the diodes (temperature sensors) are connected manually to a source meter (Keithely 2400) to measure the forward diode voltage at 1mA source current. The diode voltage is converted into temperature based on the calibration data reported in APPENDIX B.

The experiment readings are taken only when the pool reaches steady state. The pool temperature is continuously monitored and if settles within ± 0.5 °C of the set value, it is considered as steady state. Also pool temperature is maintained within the limit throughout the experiments.

The DC source is switched on and heat is supplied to the boiling surface. A sufficient time is allowed before measuring boiling surface temperatures and heating power. Then the heat to the boiling surface is increased and the measurements are continued till the critical heat flux.

The wall heat flux is defined as amount of heat supplied per unit chip surface area. The wall superheat is the difference between chip temperature and FC 72 boiling point (56 °C). Boling heat transfer co-efficient is wall heat flux divided by the wall superheat.

4.3.4 Calibration of waiting time

The waiting time between each heat input stage affects the temperature stability and accuracy of the experimental results. Therefore experiments are first conducted to determine the waiting time required for repeatable reading. Three experiments are conducted using the same boiling surface and bulk fluid temperature. But the waiting time for the experiments is 10 minutes, 15 minutes and 20 minutes respectively. The wall temperatures using 15 minutes and 20 minutes waiting time are quite similar and

within 0.8° C temperature variation compared to the boiling curve with 10 minutes. Therefore all the experiments will be conducted with a waiting time of 15 minutes.

4.3.5 Testing of repeated boiling on the surface

Experiments were conducted to evaluate the effect of surface conditions after repeated boiling. A new test surface with an average surface roughness of 101nm was used in this experiment and the surface was cleaned with acetone to remove any oxides and impurities left on the surface before commencing the experiments. The boiling curves obtained from repeated runs (seven runs) on the surface. The results of 1st, 2nd, 3rd and 7th run were plotted, the graph showed decreased in boiling incipient temperature during subsequent runs. The incipient temperature recorded for the 1st run was found to be 20.2 °C while in the 2nd run the incipient decreases to 15.8 °C and eventually stabilized after the 3rd run. The CHF for the thermal test die remains unchanged even after repeated run. Therefore boiling curves of 3rd run is taken as the useful data.

Figure 4.8 Calibration of test surface for repeated boiling

4.3.6 Measurement uncertainty analysis

The measurement uncertainties for the heat flux and temperature measurement in the diodes were calculated using the Zeroth Order Uncertainty rule. The Zeroth Order Uncertainty uses the Root-Sum-Square analysis to investigate the fixed or random

errors within the instruments. As the uncertainty for heat flux and surface temperature are based on fixed and random errors introduced by the instrumentation alone and have no contribution taken from the process; the Zeroth Order rule can be applied. Fixed errors are value that can be obtained on repeated observations at the same test condition with the same instruments and the same procedure. The DC power source has fixed instrumentation errors of $\pm 0.08\%$ and $\pm 0.07\%$ for voltage and current respectively. The equation for surface heat flux is a function of applied current, voltage and the factor of heat transferred to thermal die, given by $Q\{V, I, C_1\} = V \cdot I \cdot C_1$

Where C_1 is the factor of heat used for pool boiling, V and I are the voltage and current from the DC power source respectively. The formulate for Root-sum-square analysis is given by Eq. (4.1)

$$u_Q = \left[(u_{qv})^2 + (u_{qi})^2 + (u_{qC_1})^2 \right]^{1/2} \dots\dots\dots (4.1)$$

Where $u_{qv} = Q\{(V + \delta V), I, C_1\} - Q_{nom} = \pm 0.08\%$

$$u_{qi} = Q\{V, (I + \delta I), C_1\} - Q_{nom} = \pm 0.07\%$$

$$u_{qC_1} = Q\{V, I, (C_1 + \delta C_1)\} - Q_{nom} = \pm 2\%$$

Q_{nom} is nominal value of heat flux (evaluated without consideration of errors) the predicated uncertainty for calculating surface heat flux in the boiling test experiment was estimated to be $\pm 2.0\%$.

The uncertainty for measuring diode temperature was determined using the same approach as the method highlighted above. The estimated uncertainty for the temperature offset value is $\pm 0.5\text{ }^\circ\text{C}$ while the instrumental errors for voltage reading using source meter is $\pm 0.002\%$. The total uncertainty in measuring diode's

temperature u_T can be determined using Eq. (4.2). The total uncertainty in surface temperature is found to be $\pm 0.5\%$

$$u_T = \left[(u_{t1})^2 + (u_{t2})^2 \right]^{1/2} \dots\dots\dots(4.2)$$

Where $U_{t1} = T_j \{ (V + \delta V), T_o \} - Tj_{nam} = \pm 0.002\%$

$$U_{t2} = T_j \{ V, (T_o + \delta t_0) \} - Tj_{nam} = \pm 0.5\%$$

4.3.7 Nucleate boiling characterization on silicon surface using FC-72

An experiment was conducted to investigate the accuracy of the test setup designed for this research. The bulk liquid was set to saturation temperature (56 °C) using closed looped feedback controlled by a thermocouple placed in the same liquid level with boiling surface. The experiments were repeated three times with a 30 minutes interval between each experiment. The boiling curves were recorded for the third run and reported. Fig. 4.9 shows the boiling curve conducted at saturation temperature for a plain silicon surface. The CHF of plain surface was found to be 14.8 W/cm², which is in good agreement with the hydrodynamic model proposed by Zuber (1958) as per Eq. (4.3). Based on Zuber's correlation, the CHF using FC 72 properties @ 56° C is 15 W/cm². Also Honda et al (2002) reported CHF of 15.5 W/cm² from silicon surface with FC 72.

$$q_{crit} = 0.131 h_{fg} \rho_v \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{0.25} \dots\dots\dots(4.3)$$

Figure 4.9: Boiling curve - Saturated nucleate boiling of FC 72 on silicon chip

In summary an experimental setup for pool boiling heat transfer characterization has been developed. Experimental design details and experimental procedure are explained. A good calibration is required for accurate data collection; therefore repeatability and reproducibility of the data are tested and verified. Finally the critical parameter of the boiling characterization is CHF, which is extracted using the designed experimental setup and verified with data reported in literature.

CHAPTER 5: CHARACTERIZATION OF BOILING

ENHANCEMENT

Boiling performances of the enhanced surfaces are characterized using the experimental set-up and procedure as explained in chapter 4. A separate boiling surface is prepared for each enhancement structure. The boiling test is conducted three times for each structure by supplying heat to the surface in small increments until the critical heat flux is reached. Heat flux to the test surface is supplied in increments of $\sim 1 \text{ W/cm}^2$ during boiling incipience and CHF regions and $\sim 2 \text{ W/cm}^2$ during fully developed nucleate boiling regions. Sufficient time is allowed before measuring surface temperatures and heating power as reported in the earlier chapter. Three runs are conducted for each surface with a time gap of 30 minutes for the pool to cool to the room temperature. Boiling heat flux and surface temperatures are recorded manually only for the third run and reported. The pool temperature is preheated to 53°C , which is very close to the saturation point. During the experiments, the pool temperature varied by no more than $\pm 0.5^\circ\text{C}$, with no boiling occurring in the bulk of the liquid. It is important for the experimental accuracy, because boiling of the bulk fluid will cause liquid movement in the experimental set up and affects the bubble departures from the boiling surface.

5.1 Boiling heat transfer enhancement by coating silver particles

Boiling enhancement due to the surface modification by the silver filled epoxy coating is characterized and used as a base line. A thin layer of die attach epoxy (Sumitomo Bakelite: CRM-1295SA) is coated on the die surface. The epoxy is spread uniformly on the die surface using a screen and squeeze. The thickness is measured using optical Z scope and found to be ~ 50 microns thick. Then the coated test surface is cured in an oven for 1 hr at 150°C as recommended in the material data

sheet. The epoxy contains 88% of silver flakes by weight. Pool boiling experiments are conducted on the die attach coated surface. A maximum heat flux of 18.1 W/cm^2 is achieved and it is 1.2 times higher heat flux from the plain surface. The boiling curve of the coated surface is shown in Fig. 5.1.

Figure 5.1 Boiling curve of silver particle coated silicon surface

5.2 Boiling heat transfer enhancement MMPC structure

The boiling enhancement structure is attached on the boiling surface using silver filled epoxy. The micro-channels are facing the backside of the boiling surface and help to hold the bubble during its growth. The regular pattern of holes on the top side of the boiling enhancement structures are required to release the bubble, as well as feed the micro-channels with the working fluid. The pore size and pore pitch determines the amount of liquid flow into the channels during the bubble cycle. A dimensionless ratio of pore size and pore pitch (PR) and Percentage area of pore on the channel (%PO) representation of the liquid feeding rate. The channel size, diameter of the hole and the pitch are critical for the boiling enhancement. Ten types of boiling enhancement structures are characterized and the structure details, PR and %PO and CHF are listed in table 5.1.

The CHF achieved by each structure is based on the pore diameter, pore pitch and percentage pore opening. Pore size in the structure predominantly influences the CHF. Smaller the pore size, lesser the heat transfer enhancement. MMPC 1 and 2 have smallest pore diameter of all the structures tested. Therefore the CHF achieved is less than 30.0 W/cm^2 . When the pore size is increase to 0.3 mm, the CHF has also increased significantly. However, suitable pore pitch is required for higher heat transfer enhancement. Smaller the pore pitch, more number of pores present per unit area, which leads to flooding of the channels. The flooded channels are equivalent to a planar surfaces submerged in a liquid and the enhancement structure is not effective.

Table 5.1 Boiling heat transfer enhancement

Structure name	Channel Height (m)	Channel width (m)	Pore size (m)	Channel pitch (m)	Pore pitch (m)	PR	% PO	CHF (W/cm^2)
MMPC 1	0.00037	0.0003	0.0002	0.0005	0.0008	0.25	13%	29.3
MMPC 2	0.00037	0.0003	0.0002	0.0005	0.001	0.2	9.6%	29.8
MMPC 3	0.00037	0.0003	0.3/0.1	0.0005	0.0015	---	15.7%	32.9
MMPC 4	0.00037	0.0003	0.0003	0.0005	0.0008	0.38	17%	26.6
MMPC 5	0.00037	0.0003	0.0003	0.0005	0.001	0.3	21.6%	38.3
MMPC 6	0.00037	0.0003	0.0003	0.0005	0.0015	0.2	15.6%	31.5
MMPC 7	0.00037	0.0005	0.0003	0.001	0.0008	0.38	17.7%	30.4
MMPC 8	0.00037	0.0005	0.0003	0.0007	0.0008	0.38	17.7%	33.8
MMPC 9	0.00037	0.0005	0.0004	0.0007	0.0008	0.5	31.4%	39.3
MMPC 10	0.00037	0.0005	0.0005	0.0007	0.0008	0.63	49%	45.6

At the same time, larger the pore pitch give less number of pores per unit area, which leads to early dry-out of the channels. When the channels are dried out, the boiling

occurs only on the top face of the structure and hence the CHF achieved is small. These trends are evident in MMPC 4, 6, 7 and 8. Largest pore size of 0.5mm showed highest heat transfer enhancement in this research.

The boiling curves of all the structures are given below in Figs. 5.2-5.11 and the measurement data is given in APPENDIX E

Figure 5.2 Boiling curve with MMPC 1

Figure 5.3: Boiling curve with MMPC 2

Figure 5.4: Boiling curve with MMPC 3

Figure 5.5: Boiling curve with MMPC 4

Figure 5.6: Boiling curve with MMPC 5

Figure 5.7: Boiling curve with MMPC 6

Figure 5.8 Boiling curve with MMPC 7 structure

Figure 5.9: Boiling curve with MMPC 8

Figure 5.10: Boiling curve with MMPC 9

Figure 5.11: Boiling curve with MMPC 10

5.3 Analysis of heat transfer enhancement by silver particles coating

The boiling incipience occurred at 4 °C wall superheat for the silver particle coated surface compared to 17 °C for the plain silicon surface. The early boiling incipience is due to the large diameter of the article cavities formed by the silver particles on the surface. The gas/air trapped in the cavity helps to initiate and sustain the boiling at low wall superheat. The boiling curve rises sharply and it is almost vertical on a log-log scale for the silver particle coated surface. It shows the boiling enhancement is very high with the coating. The boiling heat transfer of a plain and die attached coated surface are compared in Fig. 5.12. The die attach coated surface has 6 times larger heat flux at 10 °C wall superheat and 22% higher CHF than the plain chip surface. The silver particle coating increases the potential nucleation sites and creates a porous surface, thus enhancing the boiling heat transfer.

Figure 5.12: Comparison of boiling curves: Plain and coated surface

At low wall superheat, the heat transfer is mainly by single-phase convection of the bulk liquid and less due to boiling heat transfer. A higher heat transfer coefficient is observed with the plain surface compared to the silver particle coated surface at low wall heat flux. The silver filled epoxy thermal conductivity is only 10 W/mK. A thin layer of the epoxy coating introduces additional thermal resistance between the

boiling surface and the cooling liquid. Once the boiling heat transfer mode is predominant, the heat transfer co-efficient of the silver particle coated surface is higher than the plain surface due to enhanced boiling effects.

5.4 Analysis of heat transfer enhancement by MMPC structures

The porous channels help to confine the bubble in the channels while it is growing and provide heat from all the sides. The channel width is the limiting bubble size generated in the channel. Also the channel walls avoid lateral coalescence of the bubble along the boiling surface. The smaller the channel width, the large the number of channels formed per unit length of the boiling surface. However very small channel width restricts the bubble growth and may not produce the desired enhancement. In this study channel widths of 0.3 mm and 0.5 mm are tested. In the following sections the boiling heat transfer by MMPC structures are analyzed.

5.4.1 Effect of Pore Size: MMPC structure with 0.3 mm channel width

MMPC structures with circular holes of diameter 0.2 mm and 0.3 mm at a pitch of 0.8mm are investigated. The pore diameter on the enhanced structure influences the volume of liquid intake into channels during the boiling process for the same pore pitch. The larger the pore diameter, more liquid is fed into the channels. The boiling performance of MMPC 1 and MMPC 4 is presented in Fig. 5.13. Structures with small pore diameter produced better enhancement at low heat fluxes ($< 10 \text{ W/cm}^2$). However at high wall heat fluxes MMPC 4 with $P_R > 0.3$ produced better boiling enhancement. The boiling curve of PMC 4 is almost vertical in a log-log scale around 10°C wall superheat, because pores are $\sim 30\%$ of the channel area and enhancement by silver particle coating is predominant at low wall superheat.

Figure 5.13: Effect of pore size on the boiling enhancement (Pore pitch 0.8mm)

MMPC 1 produced higher heat transfer co-efficient at low wall heat fluxes $< 6 \text{ W/cm}^2$. Nakayama et al (1982) studied the nucleate boiling process inside the channels and proposed three modes of boiling viz. Dried up mode (D-M), Suction-Evaporation mode (S-E), Flooded mode (F-M) as explained in Chapter 3.

Boiling of the liquid and bubble departure from the channels created a pressure difference and fresh liquid is supplied through other pores. Therefore the pore area per unit channel length influences the boiling mode inside the porous channels. MMPC 1 has smaller pore diameter, hence the boiling process is mainly S-E at low wall heat fluxes and partially D-M at high wall heat fluxes. But for MMPC 4, the boiling process is F-M at lower wall heat fluxes due to larger pore diameter and S-E at high wall heat fluxes. At a low wall heat fluxes, MMPC 1 has higher heat transfer than MMPC 4. But at wall heat fluxes higher than 10 W/cm^2 , MMPC 4 has better heat transfer than MMPC 1. The boiling performances of MMPC 3 and MMPC 6 are compared in Fig. 5.14. Both the structures have 0.3mm pore size at 1.5mm pitch, but MMPC 3 has 160 additional feeding holes of 0.1 mm size. Heat transfer by MMPC 3

is slightly better than MMPC 6, therefore the feeding holes of 0.1 mm are not very effective in boiling enhancement.

Figure 5.14: Effect of pore size on the boiling enhancement (Pore pitch 1.5mm)

5.4.2 Effect of Pore Size: MMPC structure with 0.5 mm channel width

MMPC structures with circular holes of diameter 0.3 mm, 0.4 mm and 0.5 mm at a pitch of 0.8 mm are investigated. The boiling performances of MMPC 8, MMPC 9 and MMPC 10 are presented in Fig. 5.15.

Figure 5.15: Comparison of heat transfer co-efficient (Pore pitch 0.8mm)

It is obvious that, structures with large pore opening area showed a sharp rise of the boiling curve about 10 °C wall superheat. This is mainly due to the boiling

enhancement by silver particle coating of the base surface. The three structures have similar heat transfer at low wall heat fluxes, but MMPC 10 has highest heat transfer co-efficient at high wall heat fluxes. MMPC 8 and MMPC 9 produced almost similar heat transfer co-efficient at high wall heat fluxes. It is prepared to have a flat curve of heat transfer co-efficient for a wide range of wall heat fluxes. Comparing Fig. 5.13 and Fig. 5.15, a combination of 0.5 mm pore and 0.2 mm pore with channel size of 0.5 mm may produce better heat transfer co-efficient for low and high wall heat fluxes.

5.4.3 Effect of pore pitch: MMPC structure with 0.3mm channel width

The pore pitch also influences the volume of liquid intake into channels during the boiling process. The wider the pore pitch, the lesser is the liquid intake for the same pore diameter. The boiling heat transfer results of MMPC 1 and MMPC 2 are shown in Fig. 5.16. Up to 10 °C wall superheat, surface treatment and microstructure contributes to the boiling enhancement. Beyond 10 °C wall superheat, the boiling heat enhancement is mainly by the microstructure, the wall heat fluxes varies linearly with wall superheat. The larger the pore opening area the better is the boiling enhancement at low wall heat fluxes. MMPC 1 produced heat transfer enhancement of 52%, 30%, 23%, 24% & 20% at a wall heat flux of 10 W/cm², 14 W/cm², 20 W/cm², 25 W/cm² and 29 W/cm² respectively

Figure 5.16: Effect of pore pitch on the boiling enhancement (Pore size 0.2mm)

Effect of pore pitch on boiling heat transfer results with 0.3 mm pore diameter is shown in Fig. 5.17. As the pore diameter is equal to the width of the channel, the heat transfer enhancement at low wall superheat (around 10 °C) is predominantly by the surface treatment. But beyond 10 °C, the pore dimensions changes the mode of boiling either as S-E or D-M. The heat transfer enhancement by MMPC 4 and MMPC 5 is higher as their P_R increases beyond 0.3. Boiling curves of MMPC 4 & MMPC 5 almost merged at higher heat fluxes beyond (1×10^5 W/m²).

Figure 5.17: Effect of pore pitch on the boiling enhancement (Pore size 0.3 mm)

Boiling heat transfers of all the micro machined structures with 0.8 mm pore pitch are compared in Fig. 5.18. The channels size and pore sizes are different. The boiling performances of MMPC 4 and MMPC 10 are higher at wall heat fluxes $> 1 \times 10^5$ W/cm², because its pore size is equal to channel width. Therefore the bubble released from the channels is the largest for its channel size.

Figure 5.18: Comparison of heat transfer co-efficient (0.2mm and 0.3mm pore size)

5.4.4 Effect of channel width: 0.3 mm pore size and 0.8 mm pore pitch

Channel width determines the number of channels per unit length of the boiling surface. MMPC 4 has 23 channels of 0.3mm width and MMPC 8 has 17 channels of 0.5mm. The microstructure with more channels having similar pore size and pore pitch has resulted better in enhancement. The channel walls break the boiling surface into a number of smaller sections and avoid lateral coalescence of the bubble. Boiling performances of MMPC 4 and MMPC are compared in Fig. 5.19

Figure 5.19: Effect of channel width on the boiling enhancement

In summary, nucleate pool boiling heat transfer enhancement was demonstrated by surface modification and micro-machined structures. Ten variations of micro machined structures were characterized and maximum of 45.6 W/cm², which 310% higher than the plain surface. The porous channels are effective in providing heat from all the sides of the bubble formed on the base surface. The impact of micro structure features like pore diameter, pore pitch, channel width, channel pitch on heat transfer enhancement was investigated and design guide lines were generated.

CHAPTER 6: EMPIRICAL MODEL FOR HEAT TRANSFER IN ENHANCED SURFACES

6.1 Physical model

The growth of bubbles and heat transfer on structured surfaces like MMPC is quite different from that on a plane surfaces. In porous structures, the bubbles are generated inside the channels, which are dominated by thin film evaporation mechanism. Activation and bubble growth are depending on the supply liquid into the channels. Nakayama et al (1980) were the first to develop a semi-empirical model. Using the experimental data, the semi-empirical model was developed with eight empirical constants. Chien and Webb (1998) used a similar approach and modified the model and reduced empirical constants. Recently Ramaswamy et al (2003) used the similar approach to predict for a wide range of wall superheats. The analytical model proposed by the above researchers is evaluated for the structured surface designed in this research. The following section shows the approach used to predict the heat transfer in the channels.

The total heat flux from structured surfaces is the sum of

1. Tunnel heat flux (q''_{tun}) due to the thin-film evaporation inside the tunnels of the enhanced surface.
2. Sensible heat flux (q''_{ex}) due to the external convection induced by bubble agitation.

$$q'' = q''_{\text{tun}} + q''_{\text{ex}} \dots \dots \dots (6.1)$$

Their dynamic model is based on the sequence of events during one complete cycle of bubble growth and departure. The temporal variation of the liquid meniscus thickness during a bubble cycle was ignored in their model. In the Nakayama et al model, the external convection heat flux does not include the bubble frequency and bubble

departure diameter. The bubble frequency and departure diameter are important factors for external heat flux. Haider and Webb (1997) have formulated an external heat flux model based on transient micro-convection, which accounts for the bubble diameter and frequency.

The heat transfer rate in the tunnel is governed by evaporation of liquid menisci. The external heat flux is contributed by transient conduction and convection caused by the departing bubbles. A bubble cycle includes three periods: Waiting period (Δt_w) Bubble growth period (Δt_g) and liquid intake period (Δt_e)

Figure 6.1: Bubble cycle in porous structures

1. Waiting period(Δt_w) : In this period, liquid is evaporated in the tunnel. However the vapor is constrained inside the tunnel by the surface tension on the pores. Bubble embryos protrude from the pores when the vapor pressure is greater than the breakthrough pressure of the pore ($\Delta t_g = 4\sigma P_d$). During this period the tunnel is filled with vapor except for the liquid menisci in the corners.

2. Bubble growth period (Δt_g) : In this period, vapor passes through surface pores and increases the bubble radius. Bubble growth is controlled by the inertia force of the liquid surrounding the bubble and the superheated vapor generated in the tunnel and the assumption is discussed in Chien and Webb (1998). At the beginning of this period embryos having radius $R = P_d/2$, protrude from surface pores. The liquid in the tunnel evaporates and the radius of the meniscus changes during each bubble cycle. Note that the effective evaporation temperature increases as the meniscus radius decreases, because the surface tension on the menisci increases the saturation pressure of the liquid menisci. No evaporation will occur on the meniscus if the elevation of the saturation temperature due to capillary pressure (σ/R) of the meniscus equals the wall superheat. Therefore, the meniscus radius at the end of this period, should be no less than non-evaporation radius, R_{ne} , defined by

$$R_{ne} = \frac{\sigma}{\Delta T_{ws}} \left(\frac{dT}{dP} \right) \dots \dots \dots (6.2)$$

3. Liquid intake period (Δt_e) : after the bubble departs, liquid flows into the tunnel, and is retained in the corners. In a very short period after the bubble departs, the pressure in the tunnel is lower than that of the liquid pool. Therefore, liquid flows in to the tunnel during this liquid intake period. The surface pore and tunnel dimensions control the amount of the liquid in the tunnel and the liquid meniscus radius. As observed by Nakayama et al., and Chien and Webb, this period is much shorter than the other two periods, therefore, this period is neglected.

6.2 Mathematical model

Based on the above physical model, the tunnel heat flux is the product of the latent heat transferred at the menisci during a bubble cycle and the bubble frequency, as given by

$$q''_{tun} A_{tun} = Qm'' f \dots\dots\dots(6.3)$$

Where (Q''_m) is the latent heat from menisci in one bubble cycle.

Assumptions for tunnel heat transfer in the analytical model:

1. The tunnels are vapor filled, except for liquid menisci in the corners.
2. The distribution of the liquid menisci is uniform along the tunnel length.
3. The latent heat of the vapor bubbles is by evaporation inside the tunnels.
4. Evaporation of the micro layer above the subsurface tunnel is neglected.
5. The evaporation rate is predicted by one-dimensional conduction heat transfer from the liquid meniscus region.
6. For a rectangular tunnel cross-section, liquid menisci exist in each of the four corners.
7. For high thermal conductivity fins (e.g., copper, silicon), the wall temperature is constant over the fin height.

Assuming one-dimensional heat condition in the liquid, the heat flux is given by

$$q'' = \frac{kl}{\delta} (T_w - T_s) \dots\dots\dots(6.4)$$

Figure 6.2: Liquid evaporation in the channels

When the liquid meniscus becomes very thin, the saturation temperature elevates because of the disjoining pressure and capillary pressure (σ/R). Disjoining pressure is the intermolecular forces responsible for equilibrium of contact angle of the film with the contacting surface. The elevation of saturation temperature due to the capillary pressure is significant when the meniscus radius (r_m) is very small. DasGupta et al (1993) assumed a power –law dependency of the disjoining pressure on (δ). Experimentally, they found the disjoining pressure = (C_H/δ^3), where C_H is Hamaker constant. Accounting for the effect of capillary pressure and disjoining pressure on the liquid menisci, the equivalent saturation temperature (T_s), in Eq. (6.4) is given by

$$T_s = T_l \left(1 + \frac{\sigma/R + CH / \delta^3}{h_{fg} \rho_l} \right) \dots \dots \dots (6.5)$$

The latent heat transfer rate at the menisci is given by

$$\frac{dQ_{tun}}{dt} = A \Delta T_{ws} \int \frac{k_l}{\delta(t)} \dots \dots \dots (6.6)$$

Where $\delta(t)$ is the local thickness of the liquid meniscus

The total latent heat in the tunnel during one bubble cycle is given by

$$Q_{tun} = \int \frac{dQ}{dt} dt \dots \dots \dots (6.7)$$

Alternatively the latent heat transfer in the tunnel is given as single dimension heat conduction through the liquid in the corners

$$Q_{tun} = L N_m N_c \int_0^{cycletime} \int_0^{\pi/4} \frac{k_l \Delta_{sup} r_m(t)}{(r_m(t) + \delta_{ne}) \sec(\phi) - rm(t)} d\phi dt \dots \dots \dots (6.8)$$

Latent heat transfer in each time step is given by

$$Q_{tun} = h_{fg} \rho_l \Delta V \dots \dots \dots (6.9)$$

New meniscus radius after each time step is calculated based on change in volume

ΔV

$$r_{m, new} = \left[r_{m, old}^2 - \frac{\Delta V}{LN_m N_c (1 - \Pi / 4)} \right]^{1/2} \dots\dots\dots (6.10)$$

6.2.1 Modeling bubble frequency

a. Waiting period:

At the initiation of the waiting period, the vapor in the tunnel is assumed to be in saturated state corresponding to the system pressure. The vapor behaves as an ideal gas and the density and volume of vapor inside the tunnel vary linearly with time.

Based on the first assumption, the initial conditions are $(P_{v0} = P_{sat})$ and $(T_{v0} = T_{sat})$

. At the end of waiting period, the pressure inside the tunnels is equal to the breakthrough pressure at the pore and is given by

$$P_{v1} = P_{v0} + \frac{4\sigma}{d_b} \dots\dots\dots (6.11)$$

The Clausius-Clayperon equation is expressed as

$$\frac{dT_v}{dP_v} = \frac{T_{v0}}{\rho_{v0} h_{fg}} \dots\dots\dots (6.12)$$

And the equation of state for an ideal gas is

$$P_v = \rho_v RT \dots\dots\dots (6.13)$$

Combining the above equation the temperature and density at the end of waiting period can be evaluated as

$$T_{v1} = T_{v0} + \left(\frac{4\sigma}{P_d} \right)_v \frac{T_{v0}}{\rho_{v0} h_{fg}} \dots\dots\dots (6.14)$$

A heat balance equation can be written for heat transfer in the tunnel as

$$\rho_{v1} = \frac{\rho_{v0}RT_{v0} + 4\sigma/P_d}{R(T_{v0} + \frac{4\sigma T_{v0}}{P_d \rho_{v0} h_{fg}})} \dots\dots\dots(6.15)$$

$$dQ_t / dt = h_{fg} \frac{d_{mv}}{d_t} \dots\dots\dots(6.16)$$

Where (d_{mv}/d_t) is rate of vapor mass added in the tunnel. This can be expressed as

$$\frac{d_{mv}}{dt} = V_{vm} \frac{d\rho_v}{dt} + \rho_{vm} \frac{dV_v}{dt} \dots\dots\dots(6.17)$$

A simpler approach by assuming that the density and volume vary linearly with time is given below. Hence for the waiting period (Δt_w) the following expressions hold

$$\frac{d\rho_v}{dt} = \frac{(\rho_{v1} - \rho_{v0})}{\Delta t_w} \dots\dots\dots(6.18)$$

$$\frac{dV_v}{dt} = \frac{(V_{v1} - V_{v0})}{\Delta t_w} \dots\dots\dots(6.19)$$

The final expression for latent heat transfer inside the tunnel is

$$\frac{dQ_t}{dt} = h_{fg} \left\{ V_{vm} \left(\frac{\rho_{v1} - \rho_{v0}}{\Delta t_w} \right) + \rho_{vm} \left(\frac{V_{v1} - V_{v0}}{\Delta t_w} \right) \right\} \dots\dots\dots(6.20)$$

The right – hand side of Eq. (6.20) depends only on structure dimensions. The left hand side of the Eq. (6.20) is evaluated similar to Eq. (6.8) and solving the equation provides the bubble frequency. The variable on the right hand side of Eq. (6.20) are expressed as

$$V_{v0} = V_t \dots\dots\dots(6.21)$$

$$V_{v1} = V_{v0} + N_p (P_d^2 / 4 + P_d^3 / 12) \dots\dots\dots(6.22)$$

Where V_t is tunnel volume. The term V_{vm} and ρ_{vm} can be expressed as

$$V_{vm} = (V_{v0} + V_{v1})/2 \dots\dots\dots(6.23)$$

$$\rho V_m = (\rho_{V1} + \rho_{Vo})/2 \dots \dots \dots (6.24)$$

b. Bubble growth period.

The bubble growth period is derived based on the pressure drop across liquid – vapor interface by Chien and Webb as follows

$$\Delta t_g = 0.0296 \left(\frac{7 \rho_l T_s}{\Pi h_{fg} \rho_v \Delta T_{ws}} \frac{d_b - P_d}{d_b + P_d} \right)^{1/2} \left(\frac{d_b - P_d}{2} \right) \dots \dots \dots (6.25)$$

6.2.2 Modeling bubble departure diameter

From a force balance, Nakayama et al. developed an empirical correlation, given by

$$d_b = C_b \left(\frac{2\sigma}{(\rho_l - \rho_v)g} \right)^{1/2} \dots \dots \dots (6.26)$$

To predict the bubble departure diameter on structured surfaces it is assumed that the buoyancy force equals the surface tension force in an opposite direction, and neglected the inertia force of vapor bubble. In Eq. (6.26), the empirical constant C_b was established from the Nakayama et al. experimental data. For R-11, they chose $C_b = 0.42$, while for water $C_b = 0.22$. Their force balance analysis is supported by the analysis on plain surface by Kolev (1994), who showed that only static mechanical equilibrium governs the bubble departure for plain surfaces.

According to the Chien and Webb observations using a high-speed video camera on the boiling tests of methanol, R-11 or R – 123, the typical bubble shape is spherical on the structured surfaces. They assumed that the bubble shape above an active pore is spherical, and that the bubble attaches to the surface pore during the bubble growth. By a geometrical analysis, the angle α can be evaluated and it has been assumed that angle α equals angle θ for a spherical bubble developing in a pore by Chien and Webb

Figure 6.3 Bubble contact angle

The buoyancy force on the bubble is given by $F_B = (\rho_l - \rho_v)V_b / g$ where V_b is the volume of this bubble, given by

$$V_b = \Pi \frac{d_b^3}{12} \left(1 + \cos \theta + \frac{\cos \theta \sin^2 \theta}{2} \right) \dots \dots \dots (6.27)$$

$$\text{where } \theta = \sin^{-1} \left(\frac{P_d}{d_b} \right)$$

The surface tension force on the pore is given by $F_s = \sigma \Pi d_b \sin(\theta)$.

The balance between buoyancy and surface tension forces gives

$$\sigma p_d = (\rho_l - \rho_v)g \frac{d_b^3}{12} \left(1 + \cos \theta + \frac{\cos \theta \sin^2 \theta}{2} \right) \dots \dots \dots (6.28)$$

The explicit form Eq. (6.28) as given by Chien and Webb

$$d_b = \left[\frac{Bo + \sqrt{Bo^2 + 2304(96/Bo - 3)}}{192 - 6Bo} \right]^{1/2} p_d \dots \dots \dots (6.29)$$

$$Bo = [P_d^2 (\rho_l - \rho_v)g] / \sigma$$

Bubble departure diameter based on the empirical correlation by Chen et al

$$d_b = \left(\frac{6\sigma P_d}{(\rho_l - \rho_v)g} - \frac{P_d^3}{2} \right)^{1/3} \dots \dots \dots (6.30)$$

Ramaswamy et al (2003) predicted the bubble diameter by balancing Buoyancy force, Unsteady growth force, Surface tension, Lift force, Bubble inertia, and Liquid inertia forces during the bubble growth in iterative method.

All the models were developed primarily for isolated bubble regime. The assumption was that the individual bubbles departing the surface do not coalesce either laterally or vertically. If the bubbles coalesce, the prediction of the bubble departure diameter is no more valid. Also a coalesced bubble may hover around for a longer time or get pulled out by a departing bubble from an adjacent site. Hence the point of coalescence either vertical or lateral is a meaningful criterion to define the upper limit of this model.

6.3 Semi-analytical model for MMPC structures

The bubble departure diameter is the critical parameter for the prediction of the heat transfer in nucleate boiling. The semi-analytical formula for the bubble departure diameter given above all based on the physical dimensions of the channel, pore diameter, fluid properties. Therefore an Excel spread sheet is created to calculate the bubble diameter for the for the MMPC structure. Table 6.1 and Table 6.2 show boiling enhancement structure details and bubble departure diameter based on Nakayama et. al, Chien and Webb and Ramaswamy et al

Table 6.1: Enhancement structure details

Pore dia (m)	0.0003	Tsat (K)	329
Pore pitch(m)	0.0008	Wall sup (K)	25
Channel width (m)	0.0003	Channel ht	0.00037
Channel length (m)	0.012	Tunnel volume	3.0636E-08
Channel pitch (m)	0.0005	No. Pore / channel	15
Number of channels	23	Pore density	2395833

Table 6.2: Bubble departure diameter

Bubble departure diameter	
d_b (Chien)	0.00081
d_b (Nakayama)	0.00801
d_b (Chen)	0.00112

Bubble diameter by Ramaswamy et.al	
Assumed value of d_b	0.000810
Buoyancy force (F_B)	0.000006
Lift force (F_L)	0.000192
Surface tension force (F_{st})	0.000004
Unsteady growth force (F_{duty})	0.000198
Bubble inertia force (F_{bi})	0.000000
$F_{duty} + F_{st} + F_{bi}$	0.000202
$F_B + F_L$	0.000197

Bubble departure diameter for the pore diameter of 0.0003m by Chien et al , Ramaswamy et al and Chen et al are 0.0008 m, 0.0008 m and 0.00112 m respectively. But the bubble departure diameter by Nakayama model is one order larger. However all the predicted bubble departure diameter is larger than the channel pitch of the MMPC structures. The larger bubble departure diameter may coalesce with the neighboring bubble. But it is not observed during the experimentation. The bubbles for the pores are isolated from the neighboring bubbles are clearly noticed for the all the structures tested in the research at low and medium wall superheat. Therefore the above semi-analytical models may not be suitable for the very close channels as in this research. Moreover the tunnel heat transfers are calculated based on the change in meniscus radius in the channel corners. It is difficult to measure the liquid meniscus radius change during the bubble cycle. Chen et al and Ramaswamy et al have estimated the meniscus radius by assuming a radius to predict the heat transfer and compared it with the experimental data. This method is an iterative process; the

assumed radius is changed till the prediction equals the experimental data. Therefore an empirical modeling is developed in this research based on the experimental data. The model will provide a guide line in designing the structure.

6.4 Empirical model for MMPC structure

The boiling heat transfer in the channels is directly proportional to the volume of liquid flow into the channels /sec. At low wall superheat upto 20 °C the heat transfer enhancement is dominated by the surface modification by the silver particles coating. Therefore the empirical model is developed for the heat transfer prediction beyond the wall superheat of 20 °C. The boiling curves of all the structures are almost linear beyond wall superheat of 20 °C. The volume of liquid flow into the channels is depending on the pore diameter (P_d) and Pore pitch (P_p). Also the volume of liquid in the whole structure depending on the channel width (C_w) and Channel pitch (C_p).

The volume of liquid flow in to the channels per second per unit area is proportional to the channel width and pore diameter and inversely proportional to pore pitch and channel pitch. An empirical equation of Volume flow rate ($m^3/m^2/Sec$) of liquid into the channel is proportional to the channel width, pore size, pore pitch and channel pitch which given as

$$V_l \propto C_w^{a1} P_d^{a2} P_p^{-a3} C_p^{-a4} \dots\dots\dots(6.31)$$

The index a_1 , a_2 , a_3 and a_4 are derived by comparing with the experimental data at wall superheat of 25°C. The indexes are calculated by assuming power as 1 for all the variables initially and varying one at a time. The indexes are calculated based on minimum error with the experimental data. The following section shows the index calculation

When $a_1=0.09$, the % error is found to be minimum with the experimental data

Table 6.3: Error reduction for index a1

MMPC identification #	Heat flux @25°C	Normalized heat flux	C _w	P _d	C _p	P _p	Volume @ a1=1	Normalized	% error
6	13.4	1.00	0.0003	0.0003	0.0005	0.0015	0.1200	1.00	0.33
2	13.7	1.02	0.0003	0.0002	0.0005	0.001	0.1200	1.00	2.04
8	14.8	1.11	0.0005	0.0003	0.0007	0.0008	0.2679	2.23	-101.60
7	16.6	1.24	0.0005	0.0003	0.001	0.0008	0.1875	1.56	-26.30
1	17.7	1.32	0.0003	0.0002	0.0005	0.0008	0.1500	1.25	5.38
9	19.3	1.44	0.0005	0.0004	0.0007	0.0008	0.3571	2.98	-106.77
4	19.8	1.48	0.0003	0.0003	0.0005	0.0008	0.2250	1.88	-26.89
5	20.1	1.50	0.0003	0.0003	0.0005	0.001	0.1800	1.50	-0.21
10	25.0	1.87	0.0005	0.0005	0.0007	0.0008	0.4464	3.72	-99.32

Volume @ a1=0.1	Normalized	% error	Volume @ a1=0.09	Normalized	% error	Volume @ a1=0.08	Normalized	% error
177.7	1.0	0.3	192.8	1.0	0.3	209.0	1.0	0.3
177.7	1.0	2.0	192.8	1.0	2.0	209.0	1.1	-6.3
250.5	1.4	-27.3	270.3	1.4	-26.7	291.6	1.5	-36.7
175.4	1.0	20.2	189.2	1.0	20.6	204.1	1.1	14.4
222.2	1.3	5.4	240.9	1.3	5.4	261.3	1.4	-2.6
334.0	1.9	-30.6	360.4	1.9	-29.9	388.9	2.0	-40.2
333.3	1.9	-26.9	361.4	1.9	-26.9	392.0	2.0	-37.6
266.6	1.5	-0.2	289.1	1.5	-0.2	313.6	1.6	-8.7
417.5	2.3	-25.9	450.5	2.3	-25.3	486.1	2.5	-35.1

Index “a2” is calculated with a1=0.09 in Eq. (6.31). When a1=0.55, the % error is found to be minimum with the experimental data

Table 6.4: Error reduction for index a2

MMPC identification #	Heat flux @25°C	Normalized heat flux	C _w	P _d	C _p	P _p	Volume @ a2=1	Normalized	% error
6	13.4	1.00	0.0003	0.0003	0.0005	0.0015	192.8	1.0	0.4
2	13.7	1.02	0.0003	0.0002	0.0005	0.001	192.8	1.0	2.1
8	14.8	1.11	0.0005	0.0003	0.0007	0.0008	270.3	1.4	-26.6
7	16.6	1.24	0.0005	0.0003	0.001	0.0008	189.2	1.0	25.7
1	17.7	1.32	0.0003	0.0002	0.0005	0.0008	240.9	1.2	13.2
9	19.3	1.44	0.0005	0.0004	0.0007	0.0008	360.4	1.9	-26.5
4	19.8	1.48	0.0003	0.0003	0.0005	0.0008	361.4	1.9	-25.2
5	20.1	1.50	0.0003	0.0003	0.0005	0.001	289.1	1.5	19.7
10	25.0	1.87	0.0005	0.0005	0.0007	0.0008	450.5	2.3	-25.2

Volume @ a2=0.9	Normalized	% error	Volume @ a3=0.55	Normalized	% error	Volume @ a3=0.52	Normalized	% error
433.8	1.0	0.3	7418.2	1.0	0.3	11128.6	1.0	0.3
451.7	1.0	-2.0	8903.0	1.2	-17.6	13629.7	1.2	-20.0
608.3	1.4	-26.6	10402.5	1.4	-26.6	15605.6	1.4	-26.6
425.8	1.0	25.7	7281.7	1.0	20.7	10923.9	1.0	20.7
564.7	1.3	9.6	11128.8	1.5	-13.6	17037.1	1.5	-15.9
788.1	1.8	-22.9	12185.8	1.6	-14.1	18019.8	1.6	-12.5
813.4	1.9	-25.3	13909.1	1.9	-26.9	20866.1	1.9	-26.9
650.7	1.5	19.6	11127.2	1.5	-0.2	16692.9	1.5	-0.2
963.4	2.2	-19.0	13776.9	1.9	0.5	20146.7	1.8	3.0

Index “a3” is calculated with a1=0.09, a2=0.55 in Eq. (6.31). When a3=-0.92, the % error is found to be minimum with the experimental data

Table 6.5: Error reduction for index a3

MMPC identification #	Heat flux @25°C	Normalized heat flux	Cw	Pd	Cp	Pp	Volume @ a3=1	Normalized	% error
6	13.4	1.00	0.0003	0.0003	0.0005	0.0015	7418.2	1.0	0.3
2	13.7	1.02	0.0003	0.0002	0.0005	0.001	8903.0	1.2	-17.6
8	14.8	1.11	0.0005	0.0003	0.0007	0.0008	10402.5	1.4	-26.6
7	16.6	1.24	0.0005	0.0003	0.001	0.0008	7281.7	1.0	25.7
1	17.7	1.32	0.0003	0.0002	0.0005	0.0008	11128.8	1.5	-4.2
9	19.3	1.44	0.0005	0.0004	0.0007	0.0008	12185.8	1.6	-24.3
4	19.8	1.48	0.0003	0.0003	0.0005	0.0008	13909.1	1.9	-26.9
5	20.1	1.50	0.0003	0.0003	0.0005	0.001	11127.2	1.5	-0.2
10	25.0	1.87	0.0005	0.0005	0.0007	0.0008	13776.9	1.9	0.5

Volume @ a3=-1.1	Normalized	% error	Volume @ a3=-0.92	Normalized	% error	Volume @ a3=0.95	Normalized	% error
14213.0	1.0	0.3	4409.4	1.0	0.3	5359.2	1.0	0.3
17763.9	1.2	-22.4	5123.2	1.2	-13.8	6302.9	1.2	-15.2
21224.0	1.5	-34.9	5880.1	1.3	-20.4	7282.7	1.4	-22.7
14856.8	1.0	20.9	4116.1	0.9	24.5	5097.9	1.0	23.1
22705.9	1.6	-11.0	6290.6	1.4	-8.0	7791.2	1.5	-10.0
24862.4	1.7	-32.4	6888.1	1.6	-8.5	8531.1	1.6	-10.6
28378.4	2.0	-35.1	7862.2	1.8	-20.7	9737.6	1.8	-23.0
22201.8	1.6	-4.4	6403.1	1.5	3.0	7877.5	1.5	1.8
28108.9	2.0	-6.0	7787.5	1.8	5.4	9645.1	1.8	3.6

Index “a4” is calculated with a1=0.09, a2=0.55, a3=-0.92 in Eq. (6.31). When a4=-0.95, the % error is found to be minimum with the experimental data

Table 6.6: Error prediction for index a4

MMPC identificati on #	Heat flux @25°C	Normalize d heat flux	C _w	P _d	C _p	P _p	Volume @ a4=-1	Normaliz ed	% error
6	13.4	1.00	0.0003	0.0003	0.0005	0.0015	4409.44	1.00	0.33
2	13.7	1.02	0.0003	0.0002	0.0005	0.001	5123.15	1.16	-13.82
8	14.8	1.11	0.0005	0.0003	0.0007	0.0008	5880.08	1.33	-20.44
7	16.6	1.24	0.0005	0.0003	0.001	0.0008	4116.06	0.93	24.54
1	17.7	1.32	0.0003	0.0002	0.0005	0.0008	6290.64	1.43	-7.99
9	19.3	1.44	0.0005	0.0004	0.0007	0.0008	6888.10	1.56	-8.53
4	19.8	1.48	0.0003	0.0003	0.0005	0.0008	7862.21	1.78	-20.66
5	20.1	1.50	0.0003	0.0003	0.0005	0.001	6403.06	1.45	2.99
10	25.0	1.87	0.0005	0.0005	0.0007	0.0008	7787.53	1.77	5.37

Volume @ a4=-1.1	Normaliz ed	% error	Volume @ a4=-.95	Normaliz ed	% error	Volume @ a4=-.92	Normali zed	% error
9429.46	1.00	0.33	3015.31	1.00	0.33	2400.50	1.00	0.33
10955.71	1.16	-13.82	3503.37	1.16	-13.82	2789.04	1.16	-13.82
12158.31	1.29	-16.45	4089.20	1.36	-22.48	3288.45	1.37	-23.72
8212.61	0.87	29.60	2913.94	0.97	21.88	2368.54	0.99	20.24
13452.33	1.43	-7.99	4301.73	1.43	-7.99	3424.62	1.43	-7.99
14242.61	1.51	-4.94	4790.21	1.59	-10.37	3852.19	1.60	-11.49
16813.10	1.78	-20.66	5376.42	1.78	-20.66	4280.18	1.78	-20.66
13692.75	1.45	2.99	4378.61	1.45	2.99	3485.82	1.45	2.99
16102.38	1.71	8.51	5415.70	1.80	3.77	4355.20	1.81	2.79

Based on the minimum error, Eq. (6.31) can be written as

$$V_l \propto C_w^{0.09} P_d^{0.55} P_p^{-0.92} C_p^{-0.95} \dots\dots\dots(6.32)$$

Next the volume of liquid flow into the channels are depends on the wall superheat.

Heat transfer rate is higher at high wall superheat. By the law of conservation of energy, more liquid flow into the channels at higer wall superheat. Fig. 6.4 shows the boiling curve and linear fit for all the MMPC structure beyond wall superheat of 20 °C . An average of all the strcuture slope (0.65) is used for the prediction of volume of liquid flow into the channels at different wall superheat as shown in Eq. (6.33)

$$V_l \propto (0.65\Delta T) C_w^{0.09} P_d^{0.55} P_p^{-0.92} C_p^{-0.95} \dots\dots\dots(6.33)$$

Figure 6.4. Slopes of boiling curves with MMPC structure

The proportionality constant in Eq. (6.33) is calculated by trial and error comparing with the experimental heat flux of the all the structures. Eq. (6.34) shows the empirical equation of volume of the liquid flow into the channels

$$V_l = 1.4 \times 10^{-8} (0.65 \Delta T) C_w^{0.09} P_d^{0.55} P_p^{-0.92} C_p^{-0.95} \dots\dots\dots(6.34)$$

The heat flux in the channels is calculated by multiplying the volume of liquid, liquid density and latent heat of the liquid

$$q''_{\text{tun}} = V_l \rho_l h_{fg} \dots\dots\dots(6.35)$$

Total heat transfer from the enhancement structure is the sum of the heat transfer in the channels and heat transfer on the external surface. A hydrodynamic model is proposed by Tien (1962) for saturated nucleate boiling over a flat surface as given in Eq. (6.36). The relation was reported to agree with boiling data of most liquids, therefore it is adopted for the external heat transfer calculation in this research.

$$q''_{\text{ex}} = 61.3 K_l n_s^{1/2} \text{Pr}^{1/3} \Delta T_{\text{sup}} \dots\dots\dots(6.36)$$

Where n_s is nucleation density, which is the pore density of the MMPC structure.

Based this empirical model the flux for all the structures are compared with experimental data and found good agreement. The empirical model predicts well for

all the 10 structures designed in this research. A comparison of experimental data and the model predictions are given in APPENDIX F.

In summary, a review of boiling mechanism in structured surface was made and semi-analytical model proposed by other researchers were evaluated. The semi-analytical models were found be not suitable for the porous channels designed in this research. Therefore an empirical model was developed based on the volume of liquid flow into the channels per second and it was found to match well with the experimental results.

CHAPTER 7: DEVELOPMENT OF TWO-PHASE COOLING

SOLUTION FOR ELECTRONICS PACKAGE

One of the objectives of this study is to develop a two-phase cooling solution integrating the designed boiling enhancement structure into the electronics package. Advanced electronics packaging is required for high power single chip / multi chip devices. Most of the times the devices will have localized hot spot, which is detrimental to the device performances. The devices are also stacked vertically for shorter electrical connections between different functions and the overall heat dissipation is increased drastically. Therefore a two-phase cooling is most suited for such high power application. Some of the potential applications of two – phase passive cooling with the boiling enhancements are schematically shown in Fig. 7.1. The boiling heat transfer enhancement structure will be attached on the backside of the device and it removes the heat and transport effectively to the package casing.

Figure 7.1: Schematic of enhancement structure application for electronics cooling

7.1 Design of two-phase cooling solution

A flip-chip package is taken as the test vehicle for the two-phase cooling solution. The package overall dimensions are 40x 40mm with 356 I/O. A flip chip of 12mm x12mm with full array solder bumps is attached on a 4 layers BT substrate. The chip is a thermal die containing heater and diode for heat dissipation and temperature measurement respectively. Fig.7.2 shows the package used for the two-phase cooling.

Figure 7.2: Photograph Single chip BGA package

A copper housing having overall dimensions 44mm x 44mm x 14mm is designed as the casing. Fig. 7.3 shows the details of the casing, the casing is filled with FC-72 and attached over the package using M2.5x 8mm screws. The joints are made air tight using an “O” ring. Condensation heat transfer co-efficient for a horizontal surface is calculated as per Eq. (7.1) and eighteen fins of 2 mm x 1 mm are provided on the inner face of the casing to meet ~100watts of heat load.

$$h_{condensation} = 0.943 \left(\frac{g \rho_l (\rho_l - \rho_v) h_{fg} K_l^3}{\mu_l (T_v - T_w) L} \right)^{1/4} \dots\dots\dots (7.1)$$

Figure 7.3 Photograph of two-Phase cooling solution

7.2 Testing of two-phase cooling solution

Two packages are assembled for the testing, one package is attached with the MMPC 10 and the second package is without any boiling enhancement structure. A water-cooled cold plate is placed on the package casing to remove the heat. Fig. 7.4 shows a schematic of the test arrangement.

Figure 7.4: Schematic of cooling solution test setup

The casing is filled with de-gassed FC-72 and attached over the package tightly. A cold plate is placed over the casing and 10 PSI pressure is applied using a mechanical press. A 32 AWG “K” type thermocouple is fixed to the cold plate to measure the sink temperature. A DC source is connected to the heater in the chip. The PN diode

located at the center of the chip connected to a source meter (Keithely 2400) to measure the forward diode voltage.

Figure 7.5: Photograph of two-phase cooling solution test set-up

The diode voltage is converted into temperature based on the calibration data. The test setup is shown in Fig. 7.5. DC power is supplied to the chip in small increments till the chip temperature rises sharply showing the CHF condition. The experiments are repeated for three times and chip temperature and heat flux in the 3rd run are reported. The boiling curve of plain surface tested in a package (two-phase cooling solution) and pool boiling experimental setup is compared in Fig. 7.6

Figure 7.6: Test results of passive cooling solution: Plain chip

Critical heat flux of the plain chip in the package testing is higher than the pool-boiling testing results. In package testing, the liquid very close to the chip is only at

saturated temperature and the liquid surrounding the chip is lower than the saturation point. The package case temperature is lower than the liquid saturation temperature because it is in good contact with the cold plate. Because of the sub-cooled liquid in the casing, the boiling curve is different from the pool boiling test results.

The second package with the MMPC 10 structure is tested similar to the procedure explained above. 14cc of FC-72 is filled in the casing and three runs of the experiments are conducted. Critical heat flux of the MMPC 10 structure is 8% lower than the pool-boiling testing results. Chip temperature versus the die heat flux is shown in Fig. 7.7. It is observed that, the chip temperature is higher for the corresponding heat flux compared to the pool boiling test data. It is because the casing temperature is much lower than the die temperature; which cools the liquid below the saturation point. Maximum heat flux of 41.8 W/cm^2 is achieved with the two phase cooling solution; which is 2.1 times higher than the plain chip in similar arrangement. Reason for the lower CHF in package testing is that, the boiling is vigorous at high chip temperature and the instability of the bubble column is caused by mixing of the liquid in the constrained space of the copper casing.

Figure 7.7: Test results of cooling solution: With boiling enhancement structure

CHAPTER 8: SUMMARY OF THE STUDIES

Liquid cooling has many advantages over air-cooling for high power application. Mainly it is preferred due its high heat transfer capabilities within a small cooling space; which is suitable for electronics cooling application. *Direct liquid cooling* has additional advantages of least thermal interface resistance between the chip and cooling medium. Pool boiling heat transfer is selected for the study, as it is suitable for a passive cooling solution design. Nucleate pool boiling heat transfer is a complex, non-linear process. The mechanism of boiling was well researched, however it is not possible to predict heat transfer by the boiling with the same precision associated with single-phase cooling. The boiling involves many processes and sub processes. This includes three aspects: the liquid, the heated wall, and the liquid–wall interface. Most of the processes are non-linear in nature, interacting in a complex manner.

Boiling mechanism and driving forces involved in the pool boiling heat transfer is discussed in **Chapter 2**. The boiling incipience temperature is the parameter that was investigated for effective cooling solution design. The temperature overshoot in boiling incipience has to be small, so that it will not affect the chip performance and cause failure. The vapor-liquid exchange process is responsible for much of the cooling realized with pool boiling heat transfer. It requires uninterrupted liquid access to the boiling surface over the time. Unfortunately, higher heat fluxes are dissipated by the production of more vapor bubbles per unit surface area. This bubble crowding leads to significant vapor coalescence that eventually begins to restrict the liquid access to the boiling surface. Once the vapor-liquid exchange process is interrupted, the power dissipated in the device itself is no longer rejected and the device temperature begins to escalate uncontrollably; which is called the limiting heat flux.

Various hydrodynamic, mechanistic models are reviewed and semi-empirical models are reported in chapter 2.

In direct liquid cooling, the cooling liquid should have good chemical compatibility with the chip and packaging materials, more than its thermo-physical properties. The operating pressure has effect on the nucleate boiling performance, but package construction will be expensive for high-pressure operations. Heater thermal properties and surface conditions are almost identical in most of the conventional package construction. Hence an efficient *boiling enhancement technique* is required, which could be *easily integrated with the electronics packaging processes*. The heat transfer from the heater surface by pool boiling can be enhanced using various techniques viz. Surface modification, application of electric field and mechanical force, secondary nucleation, addition of surfactant to the working fluid, mixing of two liquid etc.

Surface modification is most suitable technique for heat transfer enhancement for electronics cooling application. In addition to the enhanced heat transfer from the surface, the boiling incipient temperature has to be small for the efficient cooling of electronics. Review of recent works on the boiling enhancement by surface modification for electronics cooling application is reported in the chapter 2. The heater surface conditions and dissolved gas content of the liquid are most important parameters for the boiling incipience. Conventional surface finish of the silicon wafer used for the IC fabrication is effective for early boiling incipience. The micro cavities present on the silicon wafer don't have large mouth radius. Also the nucleation site density may not be high as the surface is smooth. Therefore an additional coating of fine particles is required for boiling heat transfer enhancement from a plain silicon surface. Conventional die attach adhesive, which has fine silver particles can be used

for boiling enhancement. Boiling incipience temperature was up to 80% lower for the silver coated surfaces as reported in literature. Various method of surface treatment and its benefits are discussed in chapter 2. Surface promoters as explained above produced limited heat transfer enhancement; it is not sufficient for the target heat flux in the range of 50 W/cm^2 for the research. Critical heat flux in nucleate boiling using Fluorocarbon (FC) liquid is around 15 W/cm^2 from an untreated silicon surface. Heat transfer enhancement of more than 300% must therefore be targeted for this study.

An additional *micro machined structure* is therefore investigated for the pool boiling heat transfer enhancement. The bubble cannot grow from a cavity unless the environment surrounding the bubble is sufficiently high in temperature. Hsu (1962) postulated that the boiling process from a cavity is controlled by a thermal boundary layer thickness. As the boundary layer thickness increases, the bubble diameter grows larger. But its thickness is limited by the transient conduction from the boiling surface. Only a range of radii of vapor nuclei could activate under these conditions and large radii extending beyond the thermal boundary layer will not activate. Micro-machined structure like pin-fin array, channels, and porous surfaces/structures will produce large cavity and provide heat around the bubble. The fin size and channel width are the parameters influencing the heat transfer co-efficient. Optimum design of such micro machined structure is crucial to meet a range of heat flux from the chip. The chip surfaces coated with micro-particles are found to enhance the nucleate boiling at a lower heat flux range, while micro machined structures, like micro-studs enhances the boiling heat transfer at higher heat fluxes.

A *novel boiling enhancement technique* combining the surface modification and micro-machined boiling enhancement structure is designed for this study. The technique is found to have good heat transfer with low wall superheat, and high wall

heat fluxes. The chip surface is modified by coating with silver filled polymer adhesive (die attach epoxy) and attaching a micro machined structure named “Micro-machined Porous Channels” (MMPC). The micro machined structure is fabricated using <100> silicon. The structure has rectangular micro-channels etched on one side of the wafer. A regular pattern of holes is etched on the other side of the wafer, which open the channels and form a porous structure. The porous channels provide heat to the growing bubbles from all the sides. Suitable structure features like channel width, pore size, pore pitch and channel pitch are selected for the experimental investigation. The boiling mechanism in porous channels is reviewed and range of pore sizes and pore pitch are designed for the study. Surface modification with silver particles filled epoxy produces heat transfer enhancement at low wall superheat and the micro machined structure provides enhancement at high wall superheat. The MMPC structures are micro-machined on 8-in wafer by DRIE process using two masks. Conventional back end of line electronics packaging processes have been used for the development of the boiling enhancement technique. The design and manufacturing details of the micro machined structure are reported in **Chapter 3**.

Experimental design for the pool boiling characterization is critical for the accuracy of the results. An experimental rig has been designed and calibrated in this study. A special boiling surface has been prepared with embedded heater and solid state temperature sensors. The boiling surface is suitable for the small increment of heat flux. Multiple temperature sensors in the boiling surface are designed to measure temperature uniformity over the entire surface. Repeatability of the experiments are tested and found that the boiling curves are consistent after 3rd consecutive boiling from a surface. Effect of waiting time between the heat flux increments are tested and found that 15 minutes is sufficient for the system to reach steady state. The maximum

boiling heat flux on a plain silicon surface using FC-72 is found to be 14.85 W/cm^2 , which is matching well with the hydrodynamic model proposed by Zuber (1958). The experimental result is also compared the data reported by Honda et al (2002) and found to be consistent. The experimental design details, experimental procedure and calibration results are reported in **Chapter 4**.

Boiling enhancement with the designed micro structures (MMPC) have been ***characterized*** using the pool boiling experimental rig. The developed enhancement technique by combining surface modification and microstructures have produced heat transfer enhancement at low and high wall heat fluxes. The boiling incipience temperature of silver particle coated surface is only 4°C above the saturation temperature compared to 17°C for the plain silicon surface. Ten micro structures are developed and characterized. Maximum of 310% of heat transfer enhancement is achieved with MMPC 10 structure. The limiting heat flux with MMPC 10 structure is 45.6 W/cm^2 ; which is very close to the research objective. The pore diameter and the pore pitch are found to have most effect on the boiling heat transfer. Micro structures with pore diameter equal to the channel width produced higher heat transfer at low and high wall superheat temperature. Micro structure with large pore opening area showed sharp rise of the boiling curve about 10°C wall superheat. It is mainly by exposing large area of the silver particle coated base surface to the pool of liquid. Number of pore area per unit channel length influences the boiling mode inside the porous channels. Higher pore density floods the channel at low wall superheat and produces less heat transfer enhancement, but it is suitable for high wall heat flux. A balance of the pore pitch and pore size is required for the effective heat transfer enhancement. Boiling heat transfer results and analysis of the micro structure features on the enhancement are discussed in **Chapter 5**.

The nucleate boiling heat transfer in the porous structures as designed in the study is very different from that a plain surface. The heat transfer in the channels is mainly by evaporation of liquid at the corners of the channels. This was proposed first by Nakayama et al (1980) and many other researchers have verified and adopted the model. Some of the *semi-analytical models* reported in the literature are reviewed and analytical equations for heat transfer in porous structures are listed in the chapter. The bubble departure diameter is the main parameter used for the calculation of the bubble frequency and heat transfer in the channels. The bubble departure diameter based by the semi-analytical model is depending on the pore diameter and liquid physical properties. The bubble diameter predicted for the MMPC structure is found to be bigger than the pore pitch of the designed structures. Therefore semi-analytical model reported may not be suitable for this research. Therefore an *empirical model* is developed to predict the heat transfer by the MMPC structure. The empirical model is found to predict the heat transfer for the 10 types of micro structure design with in an error level of $\sim 20\%$. The details of the force balance in bubble cycle and semi-analytical, empirical model details are explained in **Chapter 6**.

One of the objectives of this study is to develop a two-phase cooling solution using the designed boiling enhancement structure. A *compact passive cooling solution* is designed *for a Flip-Chip BGA package*. Micro structure MMPC 10 is chosen for the study, because the structure showed highest heat transfer enhancement in pool boiling characterization. A copper housing having overall dimensions of 44mm x 44mm x 14mm is attached over the package. Two packages are assembled, one with micro structure and another one without the micro structure. The heat flux versus chip temperature is recorded until the maximum heat flux for both the tests. Maximum heat flux of $20\text{W}/\text{cm}^2$ is demonstrated by the passive cooling solution with out the

micro structure. The critical heat flux of a plain silicon surface in pool boiling test is 14.80 W/cm^2 , which is 26% lower than the package level test result. In the passive cooling design, the liquid temperature is much colder near the casing and there is a large variation in the pool temperature and it is considered as sub-cooled boiling. The passive cooling solution with the micro structure showed good heat removal capability, however the maximum heat flux recorded is 8% lower than the pool boiling test. At high wall heat fluxes, vigorous boiling happens at chip surface and bubble cycle frequency is very high. In the passive cooling design the bubble movement generates turbulence in the pool due to the confined space. The pool turbulence makes the bubble column unstable which led to the vapor blanketing the chip surface. However a maximum heat flux of 41.8 W/cm^2 could still be achieved. The development of this two-phase cooling solution for an FC-BGA package is presented in **Chapter 7**.

CHAPTER 9: CONCLUSIONS AND RECOMMENDATIONS

9.1 Concluding Remarks

Pool boiling heat transfer was investigated for electronics cooling in this study with the target to achieve 50 W/cm^2 . The critical heat flux (CHF) was found to be $\sim 15 \text{ W/cm}^2$ using Fluorocarbon liquids (FC) from a plain silicon surface. A novel method of boiling heat transfer enhancement was developed and characterized. The boiling heat transfer from the silicon surface was enhanced by a combination of surface modification and micro machined structure named “Micro-machined porous channels” (MMPC).

An experimental setup was designed for the nucleate boiling heat transfer characterization. Boiling heat transfer from plain and enhanced surfaces were characterized. A maximum heat flux of 45.6 W/cm^2 was achieved by the technique developed in the study, which is an improvement of 300% over the plain silicon surface. The porous channels of the micro machined structure was effective in providing heat from all the sides of the bubble formed on the base surface.

The impact of micro structure features like pore diameter, pore pitch, channel width, channel pitch on heat transfer enhancement was investigated. The micro structure with pore diameter equal to the channel width was found to have maximum enhancement among the structures tested. An empirical model was developed to predict the heat transfer from the micro machined structures and the model was found to predict with a maximum error of 20% error over a wide range of wall temperatures.

A compact two-phase cooling solution for a 40 mm BGA flip-Chip package was designed and developed. The micro structure was integrated with the electronics package and cooling capacity was estimated with and without the micro structure.

Maximum heat flux of 41.8 W/cm^2 was demonstrated by the passive cooling solution with micro machined structure.

9.2 Recommendations for future work

Micro machined porous channels structure combined with surface treatment is found to be effective for the boiling heat transfer enhancement. The pore size and density significantly influenced the mode of boiling in the channels. In this study, pore sizes of 0.1 mm, 0.2 mm, 0.3 mm, 0.4 mm and 0.5 mm were investigated and maximum heat transfer enhancement was found with 0.5 mm pore diameter. There may be a need to investigate if pore diameters larger than 0.5 mm would continue to produce an improvement in the enhancement. Unfortunately, producing a new set of MMPC structures is a costly and time-consuming affair, and it is left for future investigators to extend the scope of the present studies.

Nucleate boiling heat transfer mechanism could be different for horizontal and vertical orientations. In this study, the micro machined structures are designed and tested for horizontal orientation only. When the porous channels are oriented vertically, the channels will be open sideways along the gravity direction. These could provide other effects such as the interference or mixing of the vapor columns. It is suggested that further studies be done to investigate the heat transfer enhancement in other orientations, since electronic equipment design or their use in portable equipment may sometimes necessitate the packages to be operated in these orientations

Bubble departure diameter predicted by the semi analytical models are larger than the channel pitch of the structures designed in this research. Theoretically, when the bubble diameter is larger than the channel pitch, it would coalesce with the

neighboring bubble, if they occur simultaneously. Although, this was not observed during the experimentation in this research, it is worthwhile making a further effort to understand more about the mechanism of bubble growth in structured surfaces like the MMPC structure.

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APPENDIX A

Experimental setup drawing

APPENDIX B

Temperature sensor calibration procedure

The calibration process involves determination of the diode forward bias voltage at different environment temperatures. Calibration of diodes has been done using a temperature oven (ESPEC PH-101). The test die mounted on a test board is placed in the oven and the oven temperature is maintained at a contact temperature. The die temperature is measured by a K type thermocouple mounted near the die center. Measured temperature is assumed to be the Junction temperature as the system is in thermal equilibrium. Figure below shows the schematic diagram of calibration setup.

The diodes are connected to a source meter (Keithely 2400) and its forward biased voltage at 1mA source current is recorded. The test chip and the reference thermocouple are placed in the oven and measurements are taken after thermal equilibrium has been reached. The temperature of the oven is initially set to 50°C and increased in step of 10°C until maximum temperature of 120°C. It is verified and proved the TSP has a linear relationship with temperature. As the diodes have a temperature dependent nature, it is used as temperature sensors to measure the test surface temperatures during the pool boiling. The temperature of the reference thermocouple is plotted against that of the voltage reading in each diode and the results are shown below

Calibration of diodes for temperature sensing

The equation for computing junction temperature from the measured diode voltage is determined by linear fitting the points. The equation for this straight line is given in equation.

$$T_j = mV_f + T_o$$

where T_j is the junction temperature, m is the slope, V_f is the diode voltage and T_o is the offset temperature.

The voltage across the diodes varied linearly with temperature; a proportionality factor of 754.31 (°C per mV) and an offset temperature that varies from one diode to another. As the diodes are fabricated in batch process, their temperature sensitivity was identical within the same wafer. Generally, only the offset temperature varied between the diodes and it can be eliminated by adjusting the offset temperature on device-by-device basis using the reference thermocouple.

APPENDIX C

Estimation of heat flow from the boiling surface towards the PCB

Single phase numerical analysis is conducted using commercial CFD tool Flotherm (4.1). One quarter of the experimental setup is modeled. The condenser is modeled as a constant temperature source. The test chip, substrate and PCB are modeled with full details. Mixing of two liquid is not possible using Flotherm; therefore a pool of FC-72 liquid is modeled in good contact with the substrate and PCB in a closed chamber. This will allow the heat conduction from the test die to the pool of liquid through the substrate & PCB.

Flotherm simulation model of the experimental setup

Boiling from the test surface is modeled using surface heat exchange process by assigning suitable heat transfer co-efficient on the chip surface with a reference temperature of 56° C (FC-72 saturation temperature). Numerical analysis was conducted with 5 W/cm²; 10 W/cm²; 20 W/cm² and different heat transfer co-efficient. Heat flow from the test chip to the substrate & PCB calculated by Flotherm are listed in the table.

Heat flow analysis using Flotherm

Heat dissipated in the test die	Heat transfer co-efficient applied on the test die surface to represent boiling	% of the heat dissipated in the die conducted to the substrate & PCB
5 W/cm ²	3000 W/m ² K	4.6%
10 W/cm ²	5000 W/m ² K	2.8 %
10 W/cm ²	10000 W/m ² K	1.8 %
20 W/cm ²	5000 W/m ² K	2.6 %
20 W/cm ²	10000 W/m ² K	1.5 %

The heat lost through the PCB is very small, therefore it is ignore in boiling heat transfer calculation.

APPENDIX D

Micro-machined porous channels fabrication steps

Process	Process description	Tool
Wafer start: P-type wafer 8" silicon substrate: <100>	---	---
Lithography, PR coating	Spin on 1.0um PR coating (No EBR)	Nikon (Lithography tool)
Fiducial Mask patterning	Fiducial mask (<i>Mask-1</i>)	Nikon
Microscope Inspection	Inspect alignment pattern	Scope
Silicon etching	Silicon etching for fiducial.	P-5000
PR strip	PR removal	P-5000
Piranha Clean	Final clean before oxide deposition	
Backgrind the wafer and polish	Thickness required: 500-525um	
Channel trench mask (Mask 1)	Resist thickness : 10um. (No EBR)	EVG
Channel trench etch (Silicon etch)	Trench depth required: 350um	STS (DRIE tool)
Resist strip	Do resist strip in PRS3000 for 15 minutes.	Wet bench
Deposit PECVD oxide	Deposit 2um oxide on the etched surface	Plasmatherm
Backside via trench mask (Mask 2)	Resist thickness : 10um. (No EBR)	EVG
Via etch (Silicon etch)	Trench depth required: 200um	STS (DRIE tool)
Resist strip	Do resist strip in PRS3000 for 15 minutes.	Wet bench
Strip 2um oxide on the channel surface	BOE etch till dewet.	Wet bench

APPENDIX E

Experimental data of nucleate boiling characterization

1. Boiling test data of plain silicon surface

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
51.4	51.2	0.2	0.2	-4.2	-4.6
55.6	55.7	0.7	0.5	-0.4	-0.3
59.1	58.4	1.3	0.9	3.1	2.4
65.8	63.7	2.2	1.5	9.8	7.7
68.4	67.8	2.8	1.9	12.4	11.8
72.9	72.0	3.4	2.4	16.9	16.0
69.2	69.5	4.2	2.9	13.2	13.5
74.1	74.2	5.0	3.5	18.1	18.2
75.7	75.2	5.5	3.9	19.7	19.2
76.3	75.5	6.0	4.2	20.3	19.5
76.5	75.7	7.1	5.0	20.5	19.7
76.6	76.3	8.7	6.0	20.6	20.3
77.0	76.7	10.0	7.0	21.0	20.7
77.8	77.2	11.3	7.8	21.8	21.2
77.5	77.0	11.8	8.2	21.5	21.0
78.2	77.6	14.2	9.8	22.2	21.6
78.5	77.9	15.5	10.8	22.5	21.9
78.7	78.1	17.1	11.8	22.7	22.1
78.9	78.4	18.1	12.5	22.9	22.4
79.0	78.4	18.7	13.0	23.0	22.4
79.2	78.7	19.6	13.6	23.2	22.7
79.2	78.8	20.6	14.3	23.2	22.8
79.5	79.0	21.3	14.8	23.5	23.0

2. Boiling test data of silver particle coated silicon surface

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.4	51.8	0.0	0.0	-3.6	-4.2
60.4	60.7	1.4	1.0	4.4	4.7
64.5	64.5	2.9	2.0	8.5	8.5
65.7	65.7	4.0	2.8	9.7	9.7
65.7	65.8	4.6	3.2	9.7	9.8
65.7	65.9	5.5	3.9	9.7	9.9
65.7	65.9	6.3	4.4	9.7	9.9
65.6	65.9	7.0	4.9	9.6	9.9
65.8	66.0	8.1	5.6	9.8	10.0
65.8	66.1	9.9	6.8	9.8	10.1
65.9	66.2	11.7	8.1	9.9	10.2
66.1	66.3	13.3	9.2	10.1	10.3
66.3	66.6	14.9	10.3	10.3	10.6
66.5	66.8	16.1	11.2	10.5	10.8
66.5	66.9	17.2	12.0	10.5	10.9
66.6	67.0	18.4	12.8	10.6	11.0
67.2	67.3	19.7	13.7	11.2	11.3
67.4	67.6	20.9	14.5	11.4	11.6
67.5	67.6	21.6	15.0	11.5	11.6
67.5	67.8	22.4	15.5	11.5	11.8
67.7	67.9	23.1	16.1	11.7	11.9
67.9	68.2	23.8	16.5	11.9	12.2
68.6	68.8	24.8	17.2	12.6	12.8
69.0	69.3	26.1	18.1	13.0	13.3

3 : Boiling test data with MMPC 1 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.4	52.6	0.0	0.0	-3.6	-3.4
64.6	65.7	2.7	1.9	8.6	9.7
62.6	61.5	4.1	2.8	6.6	5.5
59.4	60.6	5.8	4.0	3.4	4.6
61.7	62.3	9.1	6.3	5.7	6.3
65.3	66.8	11.7	8.1	9.3	10.8
68.5	70.4	14.5	10.1	12.5	14.4
71.3	73.8	17.6	12.2	15.3	17.8
76.7	78.0	20.9	14.5	20.7	22.0
78.8	80.0	23.4	16.3	22.8	24.0
81.1	81.8	25.4	17.7	25.1	25.8
84.0	85.2	28.6	19.9	28.0	29.2
87.1	87.8	31.7	22.0	31.1	31.8
89.6	90.7	34.0	23.6	33.6	34.7
91.2	92.4	36.0	25.0	35.2	36.4
93.5	94.5	38.4	26.7	37.5	38.5
95.1	96.2	39.7	27.6	39.1	40.2
96.8	98.0	41.1	28.5	40.8	42.0
98.1	99.4	42.2	29.3	42.1	43.4

4 : Boiling test data with MMPC 2 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.7	52.5	0.0	0.0	-3.3	-3.5
57.1	56.7	0.7	0.5	1.1	0.7
70.3	69.8	2.8	2.0	14.3	13.8
71.2	70.8	4.2	2.9	15.2	14.8
70.3	68.4	5.7	3.9	14.3	12.4
66.4	65.9	8.3	5.7	10.4	9.9
70.8	70.3	11.3	7.8	14.8	14.3
74.7	73.9	14.3	9.9	18.7	17.9
78.2	77.2	17.3	12.0	22.2	21.2
82.0	80.9	20.2	14.0	26.0	24.9
84.4	83.7	22.9	15.9	28.4	27.7
87.8	86.9	25.9	18.0	31.8	30.9
91.0	90.5	29.0	20.1	35.0	34.5
94.8	94.2	31.7	22.0	38.8	38.2
97.8	96.8	34.9	24.3	41.8	40.8
100.3	99.4	36.5	25.3	44.3	43.4
101.9	101.2	37.8	26.3	45.9	45.2
102.6	101.9	38.9	27.0	46.6	45.9
103.9	102.8	40.1	27.8	47.9	46.8
105.4	104.4	41.5	28.8	49.4	48.4
107.3	106.8	42.9	29.8	51.3	50.8

5: Boiling test data with MMPC 3 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
53.0	53.0	0.0	0.0	-3.0	-3.0
61.4	61.3	1.4	0.9	5.3	5.4
65.2	64.4	2.8	1.9	8.4	9.2
65.6	64.3	3.8	2.6	8.3	9.6
65.9	65.2	4.2	2.9	9.2	9.9
65.2	64.0	5.8	4.0	8.0	9.2
64.4	63.4	7.7	5.3	7.4	8.4
64.6	63.7	8.8	6.1	7.7	8.6
65.2	64.7	11.8	8.2	8.7	9.2
68.1	67.1	14.7	10.2	11.1	12.1
71.5	70.4	17.6	12.2	14.4	15.5
76.3	75.2	20.4	14.2	19.2	20.3
79.2	79.5	23.0	16.0	23.5	23.2
83.4	83.7	26.1	18.1	27.7	27.4
86.7	86.3	28.9	20.1	30.3	30.7
90.0	89.5	32.1	22.3	33.5	34.0
95.8	95.3	34.7	24.1	39.3	39.8
97.3	98.0	37.1	25.8	42.0	41.3
98.1	98.1	38.4	26.7	42.1	42.1
100.0	100.1	40.4	28.1	44.1	44.0
100.9	101.6	41.8	29.0	45.6	44.9
102.3	102.7	43.1	29.9	46.7	46.3
103.7	104.2	44.5	30.9	48.2	47.7
105.3	106.0	45.9	31.9	50.0	49.3
107.2	107.7	47.3	32.9	51.7	51.2

6: Boiling test data with MMPC 4 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.2	52.1	0.0	0.0	-3.8	-3.9
67.0	66.1	2.5	1.7	11.0	10.6
71.1	70.0	3.3	2.3	15.1	14.1
68.3	67.3	4.1	2.8	12.3	11.8
67.9	67.5	5.4	3.7	11.9	11.8
67.3	67.5	8.5	5.9	11.3	11.4
66.7	65.6	11.2	7.8	10.7	9.6
66.5	65.3	14.2	9.9	10.5	9.8
68.1	66.3	17.6	12.2	12.1	11.0
72.1	71.3	20.6	14.3	16.1	15.7
76.5	75.1	23.1	16.1	20.5	20.4
80.7	79.6	28.2	19.6	24.7	23.5
83.1	82.0	31.1	21.6	27.1	26.7
84.5	83.6	33.0	23.0	28.5	28.2
86.2	84.3	34.2	23.8	30.2	29.3
87.0	85.0	35.0	24.3	31.0	29.8
87.8	85.6	36.1	25.0	31.8	31.0
88.2	86.0	37.2	25.9	32.2	32.2
89.4	87.4	38.3	26.6	33.4	33.4

7: Boiling test data with MMPC 5 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.5	52.6	0.0	0.0	-3.5	-3.4
57.0	57.0	1.0	0.7	1.0	1.0
65.6	65.9	2.6	1.8	9.6	9.9
65.5	64.9	3.3	2.3	9.5	8.9
64.1	63.3	4.2	2.9	8.1	7.3
63.6	63.6	5.5	3.8	7.6	7.6
63.8	64.3	7.1	4.9	7.8	8.3
63.5	63.6	8.7	6.0	7.5	7.6
63.8	63.6	10.0	7.0	7.8	7.6
63.9	64.5	11.5	8.0	7.9	8.5
63.9	64.4	12.8	8.9	7.9	8.4
66.0	66.2	15.9	11.1	10.0	10.2
67.2	67.5	17.5	12.1	11.2	11.5
71.0	71.9	20.2	14.0	15.0	15.9
75.7	75.7	23.2	16.1	19.7	19.7
78.8	78.7	26.2	18.2	22.8	22.7
81.3	81.9	28.8	20.0	25.3	25.9
84.5	85.0	31.9	22.2	28.5	29.0
86.8	87.3	34.8	24.1	30.8	31.3
89.4	89.5	37.5	26.0	33.4	33.5
92.0	92.0	40.3	28.0	36.0	36.0
94.8	95.3	43.1	29.9	38.8	39.3
99.6	100.0	47.3	32.9	43.6	44.0
101.2	101.3	48.6	33.8	45.2	45.3
104.4	104.2	51.8	35.9	48.4	48.2
106.6	106.8	53.5	37.1	50.6	50.8
108.5	108.3	55.1	38.3	52.5	52.3

8: Boiling test data with MMPC 6 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.0	52.7	0.0	0.0	-4.0	-3.3
62.6	61.7	1.3	0.9	6.6	5.7
68.9	67.8	2.4	1.7	12.9	11.8
65.1	66.5	3.8	2.7	9.1	10.5
65.3	67.0	4.5	3.1	9.3	11.0
65.7	65.6	5.6	3.9	9.7	9.6
65.6	65.0	7.1	4.9	9.6	9.0
65.9	65.4	8.6	5.9	9.9	9.4
67.2	66.0	10.3	7.2	11.2	10.0
68.2	66.8	11.7	8.1	12.2	10.8
71.4	69.7	14.5	10.1	15.4	13.7
76.7	74.8	17.2	12.0	20.7	18.8
82.7	80.9	20.5	14.2	26.7	24.9
87.0	84.7	22.9	15.9	31.0	28.7
89.7	87.5	25.9	18.0	33.7	31.5
92.3	90.0	28.8	20.0	36.3	34.0
95.5	93.1	31.4	21.8	39.5	37.1
98.8	96.5	34.6	24.1	42.8	40.5
102.4	100.6	37.3	25.9	46.4	44.6
104.1	102.2	38.7	26.9	48.1	46.2
105.5	103.7	39.8	27.7	49.5	47.7
107.0	104.9	41.2	28.6	51.0	48.9
109.0	106.5	42.6	29.6	53.0	50.5
110.2	108.3	44.0	30.6	54.2	52.3
112.0	110.1	45.4	31.5	56.0	54.1

9: Boiling test data with MMPC 7 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.6	52.0	0.0	0.0	-3.4	-4.0
58.7	58.5	1.0	0.7	2.7	2.5
61.3	61.9	2.3	1.6	5.3	5.9
60.9	61.1	3.3	2.3	4.9	5.1
61.5	61.7	4.1	2.8	5.5	5.7
61.6	61.5	5.0	3.5	5.6	5.5
62.3	62.0	6.0	4.1	6.3	6.0
63.1	62.6	7.5	5.2	7.1	6.6
64.6	64.2	9.4	6.5	8.6	8.2
69.9	69.0	12.1	8.4	13.9	13.0
75.1	74.1	14.4	10.0	19.1	18.1
77.7	77.0	16.8	11.7	21.7	21.0
81.5	80.0	18.6	13.0	25.5	24.0
84.6	84.3	20.6	14.3	28.6	28.3
86.8	86.4	22.6	15.7	30.8	30.4
90.7	90.8	26.0	18.1	34.7	34.8
93.2	92.6	27.2	18.9	37.2	36.6
93.9	94.0	28.2	19.6	37.9	38.0
95.9	95.5	29.3	20.4	39.9	39.5
97.6	96.7	30.6	21.3	41.6	40.7
99.1	98.6	31.9	22.2	43.1	42.6
101.9	101.1	33.1	23.0	45.9	45.1

10: Boiling test data with MMPC 8 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.6	52.9	0.0	0.0	-3.4	-3.1
58.2	58.4	0.8	0.5	2.2	2.4
68.1	68.5	2.6	1.8	12.1	12.5
70.9	71.2	3.3	2.3	14.9	15.2
68.1	67.5	4.5	3.1	12.1	11.5
64.1	65.3	5.6	3.9	8.1	9.3
65.3	66.3	8.7	6.0	9.3	10.3
66.4	67.6	11.1	7.7	10.4	11.6
70.2	69.7	14.4	10.0	14.2	13.7
75.0	75.5	17.3	12.0	19.0	19.5
79.8	79.7	20.7	14.4	23.8	23.7
82.0	82.0	22.6	15.7	26.0	26.0
85.4	85.5	25.5	17.7	29.4	29.5
88.4	88.0	28.2	19.6	32.4	32.0
91.3	91.1	31.4	21.8	35.3	35.1
94.5	94.5	34.8	24.2	38.5	38.5
96.4	96.6	36.9	25.6	40.4	40.6
97.8	98.1	38.3	26.6	41.8	42.1
99.0	98.9	39.5	27.5	43.0	42.9
99.6	99.7	40.7	28.2	43.6	43.7
101.0	101.1	41.9	29.1	45.0	45.1
102.4	102.4	43.2	30.0	46.4	46.4
103.8	103.9	44.5	30.9	47.8	47.9
105.0	105.0	45.9	31.9	49.0	49.0
107.0	107.0	47.3	32.9	51.0	51.0
108.2	108.1	48.7	33.8	52.2	52.1

11: Boiling test data with MMPC 9 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.5	52.4	0.0	0.0	-3.5	-3.6
63.1	63.3	1.8	1.2	7.1	7.3
63.8	63.7	2.8	1.9	7.8	7.7
64.4	64.4	4.3	3.0	8.4	8.4
65.0	64.7	5.8	4.0	9.0	8.7
66.0	65.8	7.5	5.2	10.0	9.8
66.6	66.2	8.6	6.0	10.6	10.2
68.1	67.3	11.9	8.3	12.1	11.3
69.5	68.4	14.9	10.4	13.5	12.4
70.7	69.4	17.7	12.3	14.7	13.4
72.7	71.2	20.4	14.2	16.7	15.2
75.0	73.5	23.3	16.2	19.0	17.5
79.3	78.1	26.7	18.5	23.3	22.1
82.4	82.1	29.0	20.2	26.4	26.1
86.5	86.0	32.4	22.5	30.5	30.0
89.5	88.6	35.0	24.3	33.5	32.6
92.3	91.2	36.8	25.5	36.3	35.2
94.5	93.1	38.2	26.5	38.5	37.1
97.1	97.0	40.7	28.3	41.1	41.0
100.5	99.3	43.4	30.1	44.5	43.3
103.8	103.3	46.5	32.3	47.8	47.3
107.0	105.8	49.4	34.3	51.0	49.8
109.6	109.5	51.4	35.7	53.6	53.5
111.0	110.1	52.7	36.6	55.0	54.1
112.9	112.5	54.6	37.9	56.9	56.5
116.7	114.8	56.6	39.3	60.7	58.8

12: Boiling test data with MMPC 10 structure

Temperature sensor 1 (°C)	Temperature sensor 2 (°C)	Heating power (W)	Heat flux (W/cm ²)	Wall superheat (T1)	Wall superheat (T2)
52.2	52.8	0.0	0.0	-3.8	-3.2
60.9	61.2	1.4	0.9	4.9	5.2
64.7	64.8	2.7	1.9	8.7	8.8
64.9	65.1	3.7	2.6	8.9	9.1
64.5	65.0	4.7	3.2	8.5	9.0
64.4	64.7	6.0	4.1	8.4	8.7
64.7	64.7	7.1	5.0	8.7	8.7
64.7	64.8	8.6	6.0	8.7	8.8
65.6	65.4	10.4	7.2	9.6	9.4
66.0	65.6	11.3	7.9	10.0	9.6
67.0	66.5	14.5	10.1	11.0	10.5
67.9	67.2	17.2	12.0	11.9	11.2
68.9	67.9	20.5	14.2	12.9	11.9
71.4	70.1	23.3	16.2	15.4	14.1
72.9	72.1	25.7	17.8	16.9	16.1
73.6	73.6	28.6	19.9	17.6	17.6
76.8	76.6	31.9	22.1	20.8	20.6
79.8	79.2	34.5	23.9	23.8	23.2
83.3	82.4	38.0	26.4	27.3	26.4
85.6	84.8	40.6	28.2	29.6	28.8
91.2	90.4	45.2	31.4	35.2	34.4
95.0	94.3	49.1	34.1	39.0	38.3
101.0	99.8	53.8	37.4	45.0	43.8
106.2	105.4	58.7	40.7	50.2	49.4
109.2	107.9	60.7	42.1	53.2	51.9
111.3	110.3	62.5	43.4	55.3	54.3
113.9	112.5	64.3	44.7	57.9	56.5
114.0	113.6	65.6	45.6	58.0	57.6

13: Boiling curve of all the 10 structures

14. Two-phase cooling solution testing data of plain chip

Chip temperature (°C)	Power (W)	Heat flux (W/cm ²)	Sink temperature (°C)
28.4	0.0	0.0	29.4
35.9	0.9	0.6	29.5
54.9	3.7	2.5	29.6
79.8	8.6	6.0	29.8
83.4	11.8	8.2	30.1
85.9	15.5	10.8	30.4
88.5	19.8	13.8	30.5
90.7	24.6	17.0	31.0
92.3	26.6	18.5	31.3
92.7	27.8	19.3	31.8
93.2	28.8	20.0	32.0

14. Two-phase cooling solution testing data of package with MMPC 10 structure

Chip temperature (°C)	Power (W)	Heat flux(W/cm ²)	Sink temperature (°C)
29.2	0.0	0.0	29.3
40.7	1.1	0.8	29.4
57.8	4.9	3.4	29.5
70.0	11.4	7.9	30.1
74.8	15.7	10.9	30.5
81.9	20.7	14.4	30.8
85.9	26.5	18.4	31.0
91.5	33.1	23.0	31.9
96.1	39.1	27.2	32.0
98.8	42.1	29.2	32.3
101.7	45.6	31.7	32.6
105.6	50.7	35.2	34.0
107.3	52.5	36.4	36.6
109.5	56.3	39.1	37.8
110.1	58.2	40.4	38.4
111.0	60.1	41.8	39.1

APPENDIX F

Empirical model prediction and comparison with experimental data

MMPC 1				MMPC 2			
q" exp	T sup	q" model	% deviation	q" exp	T sup	q" model	% deviation
14.5	20.7	15.6	-7.4	12.0	22.2	12.6	-5.1
16.3	22.8	15.7	3.4	15.9	28.4	16.1	-1.3
17.7	25.1	17.3	2.1	18.0	31.8	18.0	-0.2
19.9	28.0	19.3	2.9	20.1	35.0	19.8	1.7
22.0	31.1	21.4	2.7	22.0	38.8	21.9	0.5
23.6	33.6	23.1	2.3	25.3	44.3	25.1	0.9
25.0	35.2	24.2	3.3	26.3	45.9	26.0	1.0
26.7	37.5	25.8	3.3	27.0	46.6	26.4	2.4
27.6	39.1	26.9	2.5	27.8	47.9	27.1	2.6
28.5	40.8	28.1	1.5	28.8	49.4	27.9	3.1
29.3	42.1	29.0	0.9	29.8	51.3	29.0	2.7

MMPC 5				MMPC 6			
q" exp	T sup	q" model	% deviation	q" exp	T sup	q" model	% deviation
18.2	22.8	15.4	15.4	12.0	20.7	10.1	15.5
20.0	25.3	17.2	13.9	15.9	31.0	15.1	5.2
24.1	30.8	20.9	13.4	18.0	33.7	16.4	8.8
26.0	33.4	22.7	12.8	20.0	36.3	17.6	11.9
28.0	36.0	24.4	12.7	21.8	39.5	19.9	8.8
29.9	38.8	26.3	12.1	25.9	46.4	22.5	13.2
32.9	43.6	29.6	9.9	26.9	48.1	23.4	12.9
33.8	45.2	30.7	9.1	28.6	51.0	24.8	13.2
35.9	48.4	32.9	8.5	29.6	53.0	25.8	12.7
37.1	50.6	34.4	7.4	30.6	54.2	26.3	14.0
38.3	52.5	35.6	7.0	31.5	56.0	27.2	13.6

MMPC 4				MMPV 7			
q" exp	T sup	q" model	% deviation	q" exp	T sup	q" model	% deviation
16.1	20.5	17.0	-5.8	11.7	21.7	10.2	12.6
19.6	24.7	20.4	-4.1	13.0	25.5	12.0	7.3
21.6	27.1	22.4	-3.8	15.7	30.8	14.6	6.9
23.0	28.5	23.6	-2.8	18.1	34.7	16.4	9.2
23.8	30.2	25.0	-5.2	18.9	37.2	17.5	7.2
24.3	31.0	25.6	-5.3	20.4	39.9	18.8	7.7
25.0	31.8	26.3	-5.0	21.3	41.6	19.6	7.9
25.9	32.2	26.6	-2.9	22.2	43.1	20.3	8.4
26.6	33.4	27.6	-3.9	23.0	45.9	21.6	5.9

MMPC 8				MMPC 9			
q" exp	T sup	q" model	% deviation	q" exp	T sup	q" model	% deviation
14.4	23.8	15.3	-6.3	18.5	23.3	17.1	7.6
15.7	26.0	16.7	-6.4	20.2	26.4	19.3	4.2
17.7	29.4	18.9	-6.8	22.5	30.5	22.3	0.9
21.8	35.3	22.6	-3.5	25.5	36.3	26.6	-4.2
24.2	38.5	24.7	-2.1	26.5	38.5	28.2	-6.4
25.6	40.4	25.9	-1.0	28.3	41.1	30.1	-6.5
26.6	41.8	26.8	-0.6	30.1	44.5	32.6	-8.2
27.5	43.0	27.6	-0.5	32.3	47.8	35.0	-8.5
29.1	45.0	28.9	0.6	34.3	51.0	37.4	-8.9
30.0	46.4	29.8	0.6	36.6	55.0	40.3	-10.1
30.9	47.8	30.7	0.7	37.9	56.9	41.7	-10.0
31.9	49.0	31.4	1.4	39.3	60.7	44.5	-13.3

Structure 10			
q" exp	T sup	q" model	% deviation
22.1	20.8	16.9	23.6
23.9	23.8	19.4	18.9
26.4	27.3	22.2	15.9
28.2	29.6	24.1	14.5
31.4	35.2	28.6	8.9
34.1	39.0	31.7	7.1
37.4	45.0	36.6	2.0
40.7	50.2	40.9	-0.4
43.4	55.3	45.0	-3.7
44.7	57.9	47.1	-5.5
45.6	58.0	47.2	-3.6

APPENDIX G

Boiling incipience and size range of nucleation by Hsu (1962)

Engineering surface is characterized by a distribution of cavities with various size and geometries. Experimental observations have indicated that for a given superheat, there exists a size range of active cavities. At higher heat fluxes or superheat, the range of active cavities is extended in both directions to both larger and smaller cavities.

Griffith and Wallis experiment: For 25 μ m diameter of artificial cavities, the wall superheat of boiling incipience is 11k rather than 1.3k predicted by

$$T - T_{sat} = \left(\frac{2\sigma}{r_c} \right) \left(\frac{dp}{dT} \right)_{T_{sat}} = \frac{T_{sat} v_g}{h_{fg}} \left(\frac{2\sigma}{r_c} \right)$$

As shown in the figure, assume there is a bubble nucleus sitting at the mouth cavity. The nucleus can be assumed as residual vapor from the preceding bubble. At the beginning of the bubble cycle, relatively cool bulk liquid which has replaced the void left by a departed bubble. As the time goes on, the liquid is warmed up through the transient –conduction process and the thermal layer grows. Because of the strong turbulence in the bulk, the thermal layer cannot grow indefinitely. Assume that the limiting thermal boundary layer is δ , while for $y \geq \delta$ the temperature remains at T_∞ .

Mathematical model for transient-conduction phenomenon

$$\frac{\partial^2 T_l}{\partial y^2} = \frac{1}{\alpha_l} \frac{\partial T_l}{\partial t}$$

With the following boundary condition:

$$t = 0; T_l = T_\infty$$

$$y = 0; T_l = T_w$$

$$y = \delta; T = T_\infty$$

The temperature profile for the differential equation is given by Carslaw and Jaeger

$$\xi = \frac{T_l - T_\infty}{T_w - T_\infty} = \frac{\delta - y}{\delta} + \frac{2}{\pi} \sum_{n=1}^{\infty} \frac{\cos n\pi}{\pi} \sin \left[(n\pi) \frac{\delta - y}{\delta} \right] \exp \left(- \frac{n^2 \pi^2 \alpha_l t}{\delta^2} \right)$$

$$t \rightarrow \infty, \quad \xi = \frac{\delta - y}{\delta}$$

Bubble interior temperature

$$T_g - T_{sat} = \frac{2\sigma T_{sat} v_g}{r h_{fg}}$$

Criterion for the bubble embryo to grow is the condition that the liquid temperature at the bubble cap ($y=y_b$) is equal to or greater than the bubble interior temperature

$$T_l(y = y_b) \geq T_{sat} + \frac{2\sigma}{r_b} \frac{T_{sat} v_g}{h_{fg}}$$

Size range of nucleation

$$r_{c,\min} < r < r_{c,\max}$$

$$r_{c,\max} = \frac{\delta}{2} \left[1 - \frac{\theta_{sat}}{\theta_w} + \sqrt{\left(1 - \frac{\theta_{sat}}{\theta_w} \right)^2 - \frac{4A}{\delta\theta_w}} \right]$$

$$r_{c,\min} = \frac{\delta}{2} \left[1 - \frac{\theta_{sat}}{\theta_w} + \sqrt{\left(1 - \frac{\theta_{sat}}{\theta_w} \right)^2 - \frac{4A}{\delta\theta_w}} \right]$$

$$A = \frac{2\sigma T_{sat} v_g}{h_{fg}}$$

$$where \quad \theta \equiv T - T_{\infty} \Rightarrow \theta_{sat} = T_{sat} - T_{\infty}; \quad \theta_w = T_w - T_{\infty}$$

No cavity will be effective if the $r_{c,\min}$ and $r_{c,\max}$ equation is negative

$$\left(1 - \frac{\theta_{sat}}{\theta_w} \right)^2 - \frac{4A}{\delta\theta_w} < 0$$

Superheat for boiling incipience

$$T_w - T_{sat} = \frac{2A}{\delta} + \sqrt{\left(2\theta_{sat} + \frac{2A}{\delta} \right) \left(\frac{2A}{\delta} \right)}$$

APPENDIX H

Silver particle filled epoxy: Material data sheet

<i>Electrical Conductive Die Attach Paste</i> <i>"SUMIRESIN EXCEL"CRM-1295SA</i>				
<u>Characteristics</u> High Thermal Conductivity Low modulus at high temp. and Higher MSL performance Better void performance				
<u>Properties</u>				
Item	unit	CRM-1295SA	Test method	
Appearance	-	Silver color		
Specific gravity	-	5.0		
Viscosity *1	Pa.s (PS)	23.0 (230)	E-type viscometer	
Thixotropic index*2	-	3.2		
Tack free time	Min.	30	SB method	
Filler content	Wt %	88		
Ionics*3	Na	2	Ion chromatography	
	Cl	10		
Volume resistivity	Ohm-cm	5E-5	SB method	
Die shear strength	RT	19.6c (2000)	Tension Gauge	
	@250C	6.9 (700)		
Elastic modulus	RT	11000	DMA	
	@250C	320		
Tg	deg C	70	TMA	
CTE	Alpha-1	30		
	Alpha-2	90		
Moisture absorption*4	%	0.1	SB method	
Thermal conductivity	W/mK	10	Laser-flash method	

*1:Viscosity @2.5rpm
 *2:Viscosity ratio of 0.5rpm/2.5rpm
 *3 PCT 125deg C,20hrs
 *4 85deg C,85%RH, 72hrs
 (Typical value)

Typical Application

LFCSP,E-BGA,EP package

Dispensing Method

Dispense, Line-draw

Typical Cure Condition

Ramp up to 150C by 60min.

+60min/150C in Oven

Ramp up to 175C by 60min

+30min/175C in Oven

Storage Condition

- Shelf Life : 3 Months

(Store at -15C)

6 Months

(Store at -40C)

- Thawing Condition : 1hr/RT

(5 or 10cc Syringe)

- Pot Life@25C : 24hrs

Use Up within 24hrs

After Thawing

SUMITOMO BAKELITE CO.,LTD.

prepared in Jun. 2003

APPENDIX I

Contact angle measurement

The contact angle of FC-72 on a silicon surface is measured using NRL Contact Angle Goniometer, (Model 100-00-230). The procedure for the contact angle is given below.

A silicon chip of 12mm x 12mm is placed on the stage and necessary leveling is done. The microscope base plate is adjusting such that the optical axis is aligned with the silicon chip. The syringe tip is brought above the silicon surface and a drop of FC-72 is placed using a screw in the syringe. The microscope is moved to locate the edge for the drop. A cross-line in the microscope is aligned above the base of the drop to create a wedge of light bounded by the two cross-lines. The contact angle is measured directly by noting the marking on the glass attached with the platform.

APPENDIX J

Comparison of boiling curve with reported literature

In pool boiling, the heat source is submersed in a pool of liquid. As the heat dissipation increases, the heat transfer from the surface is primarily natural convection of the liquid. When the wall temperature increases sufficiently above the liquid saturation temperature, boiling incipience occurs. Once boiling of the liquid begins, heat removal rate is increased due to phase change heat transfer. This causes a sudden drop in wall temperature. This results in a “S” shaped curve at the boiling incipience point. The boiling curve obtained in this work showed similar shape as shown in the figure below.

The boiling curve obtained by other research also showed similar shape. For better comparison, a boiling curve obtained by Chang and You (1996) is given below. The profile compares well with our measurements.

Publication

1. **Navas Khan**, D.Pinjala, K.C.Toth ,” Pool Boiling Heat transfer Enhancement by Surface modification/Micro structures for Electronics cooling: A Review”,EPTC,Singapore,2004, pp272-280
2. **Navas Khan**, D.Pinjala, K.C.Toth, “ Passive Two-Phase cooling solution for 3D stacked module with novel heat transfer enhancement” EPTC, Singapore, 2005, pp 677-682
3. **Navas Khan**, K.C.Toth, D.Pinjala,C.C.Wong “ Boiling heat transfer enhancement using micro-machined porous channels for electronics cooling” ISHMT/ASME Heat and Mass Transfer conference, India,2006