CHAPTER 1

INTRODUCTION

1.1 Motive of the Present Study

With the recent quick development of the IC (Integrated Circuits) technology, the electronic equipments are designed to be light, thin, short, and small to satisfy the needs of the industry and daily life. As the size of the electronic equipments decreases, the density of the power dissipation in them increases significantly. It is well known that the IC junction temperature must be kept under 85^{o}C to avoid being damaged [1]. The heat removal methods based on gas cooling are normally not sufficiently effective for the high-emission heat components. The direct liquid cooling is better. Moreover, the two-phase flow boiling is one of the most effective methods because the high latent heat involves in the process. To use the liquid or phase-change cooling, coolants must be chemically stable, inert, venomousless, and dielectric. The coolant FC-72, a fluorocarbon liquid manufactured by the 3M company, satisfies the above requirements and its boiling point (56^{o}C) is appropriate for the electronic cooling (Table 1.1). Up to the present, the bubble characteristics associated with the cooling of electronic components by flow boiling of dielectrics remain poorly understood. The situation is even worse for flow boiling on some enhanced surfaces. Hence the relation between the bubble behaviors and heat transfer performance can not be delineated. The use of surface micro-structures to enhance the heat transfer performance is expected to be beneficial for electronic cooling. Thus the understanding of the flow boiling processes of dielectric liquid on micro-structure enhanced surfaces and the associated heat transfer becomes increasingly important.

1.2 Literature Review – Heat Transfer Performance, Bubble Characteristics and Enhanced Surface Micro-structures

In what follows the literature relevant to the present study is reviewed. Specifically, the literature on the use of dielectric liquids for cooling of electronic equipments including the single-phase convection and boiling heat transfer on enhanced surfaces will be briefly examined.

1.2.1 Single-phase Forced Convective Heat Transfer

Incropera et al. [2] experimentally investigated single-phase convective heat transfer in a horizontal rectangular-channel with single and four-row arrays of 12 discrete flush-mounted heat sources. The working fluids they used were water and FC-77 for the channel Reynolds number ranging from 1,000 to 14,000. They developed a model to predict the relation between the Reynolds number and Nusselt number for the turbulent flow regime with $5{,}000 < Re_D < 14{,}000$. Unfortunately, the measured data were significantly under-predicted in the laminar flow regime. Slightly later, Incropera [3] conducted a critical review on convection heat transfer associated with electronic equipment cooling and showed the enhanced heat transfer with pin and longitudinal fins. In a continuing study they [4] considered an 1×10 array of discrete heat sources, flush mounted to protruding substrates located on the bottom of a horizontal flow channel. They found that when the Reynolds number varied from a very low value to a very high value, the resulting flow regimes included laminar mixed convection, transition from mixed convection to laminar forced convection, laminar forced convection, transition from laminar forced convection to turbulent forced convection, and turbulent forced convection.

Similar experiments were carried out by Maddox and Mudawar [5, 6] for a single heat source flush mounted on one wall of a vertical rectangular channel. Besides, the enhancing surfaces including the microstuds, microgrooves, and micro pin-fins were also examined. They observed that the single-phase convection heat transfer coefficient increased with increasing flow velocity and the micro-structure-enhanced surfaces effectively promoted the heat transfer performance. Later, Gersey and Mudawar [7] tested forced convection of FC-72 over a series of nine in-line microelectronic chips. Furthermore, they investigated the effects of orientation on heat transfer performance and proposed an empirically generalized equation for their experimental data. On the other hand, Samant and Simon [8] investigated heat transfer from a small heated region to R-113 and FC-72. They combined the experimental data for R-113 and FC-72 to develop an empirical correlation. In addition, a numerical modeling of turbulent heat transfer from discrete heat sources to water and FC-72 in a liquid-cooled channel was carried out by Xu et al. [9]. They observed that the wall temperatures became sharply peaked at the leading edges of the heat sources and increased with increasing distance to the trailing edge. Garimella and Eibeck [10] analyzed the heat transfer characteristics of an array of protruding elements in single-phase forced convection of water for the channel Reynolds number ranging from 150 to 5,150. They noted that the heat transfer coefficient decreased with the decreasing channel Reynolds number and the Nusselt number decreased with increasing ratio of the channel height to element height.

The single-phase heat transfer correlations proposed in the above studies are listed in Table 1.2.

1.2.2 Two-phase Heat Transfer Performance

Mudawar and his colleagues [7, 11] experimentally examined flow boiling of FC-72 over flush-mounted heat sources. The multi-heat sources in the vertical flow channel were arranged in an 1 × 9 array for the flow velocity ranging from 13 cm/s to 400 cm/s and for the liquid subcooling from 3 ^{o}C to 36 ^{o}C with the system pressure at 1.36 bar. They found that the heat transfer coefficient in the single phase region varied linearly with the flow velocity and the boiling incipience was delayed for a higher flow velocity. Similar experiments were conducted by Heindel et al. [12] also for FC-72. They defined the wall temperature overshoot at the boiling inception as the temperature difference between the maximum temperature recorded under the single-phase convection condition and the corresponding theoretical temperature under the boiling conditions. According to their experimental results, at increasing velocity the heat flux increased but the temperature overshoot decreased. In the fully developed boiling, the velocity exhibited little effect on the boiling curves, which was also supported by McGills et al. [13]. On the other hand, at increasing liquid subcooling the temperature overshoot decreased but the heat flux increased.

In investigating the critical heat flux of the flush-mounted and protruded chips with the dielectric coolant FC-72 in vertical channel flow boiling experiments, Tso et al. [14] found that for both cases the temperature of the chip surface decreased with the increases in the flow velocity and liquid subcooling in the partial boiling region and the result was opposite to that of Willingham and Mudawar [11]. While in the fully-developed boiling region both the flow velocity and subcooling temperature had little effects on the temperature of the chip surface. Besides, the critical heat flux also increased with the flow velocity and liquid subcooling. The temperature of the chip surface for the flush-mounted chips was lower than the protruded chips in the partial boiling region, but they were nearly equal in the fully developed boiling region. Moreover, the experimental results revealed that the critical heat flux of flush-mounted chips was higher than the protruded chips. Similar experiment was carried out by McGills et al. [13] with refrigerant R-113 and they reached the same conclusion. Besides, they found that there were three different regions in their experiments for flush-mounted and protruded chips. They defined the convective

CHF region when $We > 100$ and the pool boiling region when $We < 10$. The transition region was defined for $10 \leq We \leq 100$, here We denotes the Weber number.

Samant and Simon [8] analyzed the heat transfer from a small region to refrigerants R-113 and FC-72 and noted that as the flow velocity and subcooling temperature increased, the temperature excursion and boiling hysteresis appeared to decrease. An increase in the subcooling temperature caused some delay in the departure from nucleate boiling (DNB), which was considered to result from an increased rate of bubble collapse in the subcooling flow. An experimental study of the subcooled flow boiling of FC-72 on a flat surface and a micro-porous coated surface from Rainey et al. [15] showed that the enhancement of CHF from the micro-porous coating increased with increasing liquid subcooling and decreased with increasing flow velocity.

1.2.3 Bubble Characteristics

Ma and Chung [16] investigated bubble dynamics in reduced gravity forced-convection boiling with FC-72 flow over a thin gold film semi-transparent heater specifically designed to generate single bubble for experimental observation. The bubble departure size was found to be bigger in the micro-gravity than in the normal gravity. At increasing flow rate, the bubble departure time and departure size decreased. This was also observed by Situ et al. [17] subsequently. Besides, they also indicated that the bubble growth rate dropped sharply after lift-off. In a few milliseconds after the bubble lift-off, the liquid velocity was higher than the averaged bubble speed, and the bubbles were accelerated by the liquid. In addition, Yin et al. [18] examined the subcooled flow boiling of R-134a in an annular duct and noted that both the bubble departure size and bubble departure frequency reduced at increasing liquid subcooling.

Experiments conducted by Bang et al. [19] and Chang et al. [20] for R-134a and water focused on the behavior of near-wall bubbles in subcooled flow boiling. They identified four different two-phase flow patterns including the discrete attached bubbles, sliding bubbles, small coalesced bubbles and large coalesced bubbles or vapor clots for the heat flux increasing from a low value. With the increase in the mass flux of the flow, the size of coalesced bubbles decreased. In a recent experiment Bang et al. [21] further noticed the presence of the R-134a vapor remants below the discrete bubbles and coalesced bubbles, and the presence of an interleaved liquid layer between the vapor remants and bubbles. Besides, the bubble layer could be divided into two types, a near wall-bubble layer dominated by small bubbles and a following bubble layer prevailed by large coalesced bubbles.

By using digital imaging and analyzing techniques, Maurus et al. [22] examined the bubble characteristics and local void fraction in subcooling flow boiling. The bubble population was noted to increase with the heat flux and the bubble density reduced drastically at increasing mass flux. Besides, the bubble size increased with an increase in the heat flux but reduced with an increase of the mass flux. In a continuing study [23] they further showed that the effects of the heat flux and mass flux on the bubble size distribution were weak for small bubbles but became more pronounced for bigger bubbles. The total bubble life time, the remaining lifetime after the detachment process and the waiting time between two bubble cycles decreased significantly as the mass flux increased. Moreover, they pointed out that the bubble behavior was dominated by the temperature of the thermal boundary layer and the turbulence intensity due to the heat transport in the liquid near the interface. On the other hand, an experimental analysis was completed by Thorncroft et al. [24] to investigate the bubble growth and detachment in vertical upflow and down flow boiling. They found that the bubble growth rate and bubble departure diameter increased with the Jacob number (ΔT_{sat}) and decreased with increasing mass flux.

1.2.4 Enhanced Surface Micro-structures

Maddox and Mudawar [5, 6] studied flow boiling of FC-72 over a single 12.7 mm × 12.7 mm copper heat source with microstud, microgroove, and cylindrical micro pin-fin-enhanced surface flush mounted on a wall of a vertical rectangular-channel whose cross section area was 38.1 mm in width and 12.7 mm in height. The enhanced surface micro-structures were found to significantly enhance heat transfer performance and could reduce boiling hysteresis. In particular, the enhanced surface could augment turbulent mixing and develop multiple thermal entry regions at the top surfaces of individual surface micro-structures resulted from a serpentine motion of liquid between the adjacent rows on the surface micro-structures.

Heat transfer in pool boiling of FC-72 on silicon chips with the surface micro-structures of micro-pin-fins was investigated by Honda et al. [25, 26]. The sizes of the pin-fins ranged from 10 μm to 50 μm in thickness, 60 μm to 270 μm in height and the fin pitch was twice the fin thickness. They found that both the nucleate boiling heat transfer and the critical heat flux were effectively enhanced by the micro-pin-fins. At increasing wall superheat the boiling curve for the micro-pin-finned chip was characterized by a sharp increase in the heat flux. Moreover, the boiling phenomena observed in their study revealed that a small amount of vapor was left within the gap between the pin fins when a growing bubble had left the surface. For a fixed pin-fin thickness, the critical heat flux rose with the increase in the height of pin-fins. On the other hand, Honda and Wei [27] conducted a critical review on the boiling heat transfer enhanced by the surface-structures. They indicated that all the surface micro-structures including the micro-roughness, micro-reentrant, and micro-porous structures were helpful in reducing the boiling incipience superheat. Surface cavities were effective in increasing critical heat flux and in enhancing nucleate boiling. Generally, the micro pin-fins were most effective in increasing CHF and micro-porous structures were most effective in enhancing nucleate boiling heat transfer. Besides, the time interval of the bubble growth with surface micro-structures was longer than with smooth surface.

In investigating flow boiling heat transfer with micro-finned and flat surfaces Mizunuma et al. [28] observed the cyclic blocking of the channel by overgrown vapor bubbles, which was similar to slugging or necking, preceding the occurrence of CHF. The frequency and amplitude of the temperature and pressure variations were the main differences between the slugging and necking. Besides, at a larger total flow rate the larger resistance in the grooves was noted and it was more sensitive to the developing boiling. Recently, Ramaswamy et al. [29] examined the effects of varying geometrical parameters on boiling from microfabricated enhanced structures. They indicated that increasing pore size caused a higher heat dissipation and the pore pitch had more significant effect on the heat transfer performance.

The effects of pressure, subcooling, and dissolved gas on the FC-72 pool boiling heat transfer from microporous, square pin-finned surface were examined by Rainey et al. [30]. They showed the improved nucleate boiling performance with the pressure and the formation of the temporal dry-out near the base of the fins. Besides, the shift of the entire upper portion of the boiling curve upward was delayed at increasing subcooling. Honda et al. [25, 26] noted the increase in the boiling performance with the fin length (area). Finally, heat transfer from the microporous finned surface was found to have better heat transfer performance than the plain finned surface [30].

Qu and Mudawar [31] investigated the incipience boiling heat flux in micro-channel heat sinks and noted that the enhanced surface structures led to much higher heat transfer coefficients and better temperature uniformity. Similar effects were obtained by raising the inlet flow velocity and lowering the inlet temperature. A numerical solution of bubble cavitation in a microchannel carried out by Lin and Cheng [32] considered the corners in the microchannel as pseudo microcavities, which lowered the incipience boiling temperature for the heterogeneous nucleation. Besides, a larger contact angle is also helpful in reducing the nucleation temperature. On the other hand, the boiling flow in a silicon micro-channel was examined by Ferret et al. [33] to elucidate the heat transfer performance for laminar flow. They found that the heat transfer coefficient was significantly influenced by the wall temperature in every flow condition and the higher wall temperature resulted in a lower heat transfer coefficient. Recently, bubble dynamics in single microchannel and two parallel microchannels were explored by Lee et al. [34] and Li et al. [35]. Lee et al. [34] indicated that the size of the departure bubble from the microchannel wall was mainly governed by the surface condition and drag of the bulk flow, and their data well agreed with the modified Levy correlation. Besides, they found that the bubble departure frequency in the microchannel is comparable to that in an ordinary sized channel. Then, Li et al. [35] noted that bubbles in the microchannels grew linearly with time and the bubble growth rate increased with increasing heat flux.

1.3 Review of Correlation Equations for Two-phase Flow Boiling

A generalized empirical correlation for subcooled flow boiling was developed by Shah [36] and it was applicable to the water, refrigerants, and organic fluids at high flow rates (Re > 10,000). The two-phase flow boiling heat transfer coefficient was considered to be the single liquid-phase convective heat transfer coefficient multiplied by a factor. Besides, the saturated flow boiling heat transfer coefficient was correlated by Chen [37]. He divided the boiling heat transfer coefficient into two parts: a microconvective (nucleate boiling) contribution which could be estimated by the pool boiling correlations and a macroconvective (non-boiling forced convection) contribution which could be estimated by the single-phase correlation such as the Dittus-Boelter equation [38]. In addition, Rainey et al. [15] proposed a flow boiling heat transfer correlation for plain discrete heat sources in a rectangular channel.

There exist some experimental studies of bubble characteristics in flow boiling, such as bubble departure diameter, growth rate, and active nucleation site density. A vertical upflow subcooled flow boiling study was carried out by Levy [39] and he took the buoyancy force, drag force, and surface tension force into consideration to develop a correlation for the bubble departure diameter. Koumoutsos et al. [40] investigated forced convection boiling at an artificial nucleation site and found that the vapor bubbles slided along the heated wall for a distance before they lifted off. From the measured data, they proposed an empirical relation between the lift-off bubble diameter and departure bubble diameter for pool boiling. An analytical model of vapor bubble departure in forced convection boiling was developed by Klausner et al. [41] to predict the time rate of change of the radius for a spherical growing vapor bubble at a wall. They found that the bubble departure diameter was significantly influenced by the mean liquid velocity and wall superheat. Later, they [42] proposed an improved model to predict the departure and lift-off diameters in forced convection boiling. The major improvement is that the inclination angle can be determined on a dynamic basis and the surface tension force is small compared to other forces acting on a vapor bubble at the points of departure and lift-off. Furthermore, Klausner and Zeng [43] pointed out that the vapor bubble growth rate strongly affected the departure and lift-off processes and it depended significantly on the system pressure. In addition, Zeitoun and Shoukri [44] investigated the bubble behavior and mean bubble diameter in subcooled upward flow boiling in a vertical annular channel. A dimensionless correlation for the mean bubble diameter as a function of the mass flux, local subcooling and heat flux was proposed. Recently, the incipience point of net vapor generation (IPNVG) in low-flow subcooled boiling was predicted by Sun et al. [45] and a correlation of the detached bubble diameter at IPNVG was also obtained for water and for R-12 in natural and forced boiling conditions.

On the other hand, correlations for onset nucleate boiling heat flux and active nucleation site density were examined by Basu et al. [46]. The active nucleation site density was noted to depend only on the static contact angle and wall superheat. The velocity and local liquid subcooling did not affect the nucleation site density independently. Instead their effect was implicit in the relation between the wall superheat and local heat flux. Recently, a newly developed model from Hibiki and Ishii [47] gave good predictions of active nucleation sites over a wide range of flow condition. The correlations proposed in the above studies are listed in Tables 1.3 - 1.5.

1.4 Objective of this Study

The above literature review clearly indicates that the pool boiling heat transfer from micro-structured surface and the associated bubble dynamics have received some attention. But heat transfer and bubble characteristics in flow boiling of dielectric coolants on micro-structured surfaces are less explored. In this study an experiment is carried out to investigate the FC-72 flow boiling heat transfer and associated bubble characteristics on a single silicon chip with micro-pin-fins on its surface. The chip is flush mounted on the bottom of a horizontal rectangular channel. Effects of the coolant mass flux and inlet subcooling, imposed heat flux, and fin arrangement and geometry on the flow boiling heat transfer performance and bubble behavior will be investigated in detail.

Table 1.1 Thermodynamic properties for FC-72.

Properties	FC-72
Appearance	Clear, colorless
Average Molecular Weight	338
Boiling Point (1 atm)	56^{o}C
Pour Point	-90^{o}C
Estimated Critical Temperature	449K
Estimated Critical Pressure	1.83 $\times$ 10^{6} pascals
Vapor Pressure	30.9 $\times$ 10^{3} pascals
Latent Heat of Vaporization (at normal boiling point)	88 J/g
Liquid Density	1680 kg/m^{3}
Kinematic Viscosity	0.38 centistokes
Absolute Viscosity	0.64 centipoise
Liquid Specific Heat	1100 J kg^{-1} C^{-1}
Liquid Thermal Conductivity	0.057 W m^{-1} ^{o}C^{-1}
Coefficient of Expansion	0.00156 ^{o}C^{-1}
Surface Tension	10 dynes/cm
Refractive Index	1.251
Water Solubility	10 ppmw
Solubility in Water	<5 ppmw
Ozone Depletion Potential	0

Table 1.2 Some single-phase convection heat transfer correlations for electronic cooling.

Reference	Working Fluid	Heat Transfer Correlation	Conditions
Incropera et al. [4]	Water & FC-77	$Nu_L = 0.13Re_D^{0.64}Pr^{0.38}(\mu_o / \mu_h)^{0.25}$	Heat sources size: 12.7mm ╳12.7mm Arrays: 4 row ╳ 3 line Inlet temperature: 14 &30 ^oC $5000 < Re_D < 14000$
Gersey and Mudawar [7]	FC-72	$Nu_L = 0.362Re_L^{0.614}Pr^{1/3}$	Heat sources size: 10mm ╳10mm Arrays: 9 row ╳ 1 line Flow velocity: 13 ~400 cm/s
Samant and Simon [8]	R-113 & FC-72	$Nu_H = 0.47Re_H^{0.58}Pr^{0.5}$	Test patch size: 0.25mm ╳ 2.0mm Bulk velocity: 2.05 ~ 16.86 m/s Pressure at the patch: 118.8 ~ 338.1 kPa
Garimella and Eibeck [10]	Water	$Nu = 1.31Re_a^{0.48}(LS / B)^{0.15}$	Heat sources size: 1.9 cm ╳ 1.9 cm $150 < Re_H < 5150$ Arrays: 5 row ╳ 6 line

Table 1.3 Some flow boiling two-phase heat transfer correlations for electronic cooling.

Reference	Working Fluid	Heat Transfer Correlation	Conditions
Shah [36]	Water, refrigerant, organic fluid	$h_{2\phi} = \begin{bmatrix} \psi_0 \cdot h_\ell, \text{ if } \dfrac{T_{sat} - T_\ell}{T_w - T_{sat}} < \min \left\langle 2, 6.3\times10^4 Bo^{1.25} \right\rangle \\ (\psi_0 + \dfrac{T_{sat} - T_l}{T_w - T_{sat}}) \cdot h_\ell, \text{ if } \dfrac{T_{sat} - T_\ell}{T_w - T_{sat}} > \min \left\langle 2, 6.3\times10^4 Bo^{1.25} \right\rangle \end{bmatrix}$ where $\psi_0 = 230 \cdot Bo^{0.5}$, if $Bo > 0.3 \times 10^{-4}$ $= 1 + 46 \cdot Bo^{0.5}$, if $Bo < 0.3 \times 10^{-4}$ $Bo = \text{Boiling number} = \dfrac{q''}{G \cdot i_{\ell v}}$	Re > 10000 Subcooled boiling
Chen [37]	-	$h_{tp} = h_{conv} + h_{nb} \Leftrightarrow h_{tp} = Fh_l + Sh_{pool}$ $h_{pool} = 0.00122 \cdot ((k_l^{0.79} c_{p,l}^{0.45} \rho_l^{0.49}) /$ $(\sigma^{0.5} \mu_l^{0.29} i_{fg}^{0.24} \rho_g^{0.24})) \cdot \Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75}$; $F = 2.35(1/X_{tt} + 0.213)^{0.736}$, $for X_{tt} > 0.1$ $S = (1 + 0.12 \mathrm{Re}_{tp}^{1.14})^{-1}$, $for \mathrm{Re}_{tp} < 32.5$ $\mathrm{Re}_{tp} = G(1-x)D / \mu_l (F^{1.25})(10^{-4})$	-

Table 1.4 Correlations for bubble departure diameter in flow boiling

Reference	Working Fluid	Bubble Departure Diameter Correlation	Conditions
Levy [39]	-	$d_d = C_o\sqrt{\frac{\sigma D}{\tau_w}}$, where Co = 0.015 is an emperical const. D is the hydraulic pipe diameter τ_w is the wall shear stress	Analytical model
Koumoutsos et al. [40]	Water	$\frac{r}{r_o} = [1-\varepsilon\cdot\frac{\rho_g U\upsilon}{g_o\sigma}]^{n/2}$ where $\varepsilon\cdot\frac{\rho_g U\upsilon}{g_o\sigma} = 0.015$ sec/cm, n=4 r is the bubble lift-off diameter & r_o is the bubble departure diameter for pool boiling	Horizontal flow boiling Flow velocity: 0 ~ 25 cm/s
Klausner et al. [41]	R-113	$r(t) = \frac{2}{3}\frac{B^2}{A}[(t^+ + 1)^{3/2} - (t^+)^{3/2} - 1]$ where $t^+ = \frac{A^2 t}{B^2}$, $A = [\frac{\pi}{7}\frac{h_{fg}\rho_v\Delta T_{sat}}{\rho_l T_{sat}}]^{1/2}$, $B = [\frac{12}{\pi}\eta_l]^{1/2}\frac{\Delta T_{sat} Cp_l \rho_l}{h_{fg}\rho_v}$	Horizontal flow boiling Mass flux: 112 ~ 287 kg/m^{2}s Heat flux: 11 ~ 26 kW/m^2

Table 1.4 Continued

Reference	Working Fluid	Bubble Departure Diameter Correlation	Conditions
Zeng et al. [42]	R-113	$r(t) = \frac{2b}{\sqrt{\pi}} Ja \sqrt{\eta t}$ where $Ja = \frac{\rho_l Cp_l \Delta T_{sat}}{\rho_v h_{fg}}$ η is the thermal diffusivity	Horizontal flow boiling Wall superheat: 5.5 ~ 12 °C Flow velocity: 0.35 ~ 1 m/s
Zeitoun and Shoukri [44]	Water	$\frac{Ds}{\sqrt{\sigma/g\Delta\rho}} = \frac{0.0683(\rho_l/\rho_g)^{1.326}}{Re_l^{0.324}(Ja + \frac{149.2(\rho_l/\rho_g)^{1.326}}{Bo^{0.487} Re_l^{1.6}})}$ where $Ja = \frac{\rho_l Cp_l \Delta T_{sat}}{\rho_g h_{fg}}$, $Bo = \frac{q}{Gh_{fg}}$, $Re_l = \frac{GD}{\mu_l}$	Vertical upward flow boiling Mass flux: 151.4 ~ 411.7 kg/m^{2}s Heat flux: 286.5 ~ 705.5 Kw/m^2
Sun et al. [45]	Water & Freon-12	$D_B = C[\frac{\sigma}{(\rho_f - \rho_g)g + (3/4)(G^2/D_e \rho_f)}]$ where C is a empirical constant	-

Table 1.5 Correlations for heat transfer & active nucleation site density in flow boiling

Reference	**Working Fluid**	**Heat Transfer and Active Nucleation Site Density Correlations**	**Conditions**
Rainy et al. [15]	FC-72	$q'' = 5.39 \cdot \Delta T_{sat}^{3.63}$	Two 1-cm^2 copper plain surface $18700 < Re < 174500$ Subcooling temperature: 4 ~ 20 K
Basu et al. [46]	Water	$Na = 0.34[1 - \cos(\phi_s)] \cdot \Delta T_w^{2.0}$, $\Delta T_{w,ONB} < \Delta T_w < 15\,^{o}C$ $Na = 0.34 \times 10^{-5}[1 - \cos(\phi_s)] \cdot \Delta T_w^{5.3}$, $\Delta T_w \geq 15\,^{o}C$	ϕ_s : 1 deg to 85 deg System pressure: 1.0 bar to 137.5 bar Local liquid subcooling: 1.7 ^{o}C to 80 ^{o}C Velocity: pool to 17 m/s
Hibiki and Ishii [47]	-	$Nn = \bar{N}n\left\{1 - \exp\left(-\frac{\theta^2}{8\mu^2}\right)\right\}\left[\exp\left\{f(\rho^+)\frac{\lambda'}{Re}\right\} - 1\right]$ where $\bar{N}n = 4.72 \times 10^5$ sites/m^2, $\mu = 0.722$ rad, $\lambda' = 2.50 \times 10^{-6}$ m $Re = \frac{2\sigma\{1+(\rho_g/\rho_f)\}/P_f}{\exp\{i_{fg}(T_g - T_{sat})/(R \cdot T_g \cdot T_{sat})\} - 1}$ $f(\rho^+) = -0.01064 + 0.4826\,\rho^+ - 0.22712\,{\rho^+}^2 + 0.05468\,\rho^+$ $\rho^+ = \log(\rho^*)$, $\rho^* = \Delta\rho / \rho_g$	0 kg/m^{2}s $\leq$ mass velocity $\leq$ 886 kg/m^{2}s 0.101 MPa $\leq$ pressure $\leq$ 19.8 MPa 5^{o} $\leq$ contact angle $\leq$ 90^{o} 1.00 $\times$ 10^4 sites/m^2 $\leq$ active nucleate site density $\leq$ 1.51 $\times$ 10^{10} sites/m^2