Chapter 1

INTRODUCTION

1.1 Motivation and Objective of the Present Study

As the VLSI technology has made progress in the past several decades, the trend of the electronic industry is aimed to decreasing the chip size and increasing the circuit concentration and numbers of transistor in a silicon chip. Those advances in chip design have also produced larger power dissipation in a small area. Today's electronic components draw high levels of power and run extremely hot. For example, most computing modules made by CMOS transistors can generate significant heat in operations. Also, the performance of CPUs is strongly affected by the operating temperature [1]. Hence, there is a strong need to improve current capacities for the thermal management of electronic components. The challenge for thermal engineers is to develop effective cooling schemes which are capable to remove the large amount of heat being generated from chips.

The traditional microprocessor cooling method is mainly uses the devices to reduce the operating voltage of component and to use high-performance heat sink or multiple air fans to produce the forced convection condition. However, the large thermal resistance due to the poor thermal properties of air between electronic component and ambient which lead electronic component to extreme high temperature when dissipating high heat flux that as shown in Figure 1.1 [2]. As the limit of component temperature has been reached and any further escalation in heat flux will result in unacceptable high component temperatures. And also the reduction in the operating voltage of circuits has not been able to reverse the trend toward higher power electronic component caused by increased complexity and operating frequency. Therefore, as the traditional cooling method is not able to provide the desire heat dissipation, the direct immersion liquid cooling with phase change heat transfer by using dielectric liquids as the promising cooling schemes for high-powered electronic components has brought about more attention gradually.

Due to the imperious demands of micro electronic cooling technology, the development of the phase change heat transfer engineering is expected to reach the heat dissipating capacity of maximum heat flux to 100 Wcm^{-2} and also to keep the lower operating temperature of the power dissipation of component. Through the high efficiency heat transfer performance of phase change process of the nucleate and flow boiling, the application of electronic cooling by using dielectric fluid become an alternate choice.

The primary candidate direct cooling liquids are the perfluorocarbons (FC series) made by 3M Inc. These highly wetting liquids are non toxic, chemically inert to packaging materials and posses high dielectric strengths. They also haverelatively low critical pressures, thermal conductivity and specific heats and vary large air solubility (40-50% by volume at atmospheric conditions in comparison with 2% by volume for water).

Most early record using the direct liquid cooling can be traced back to the military electronics system in the end of 1940 ages, and then this technology is considered to solve the high temperature problem of the electronic components seriously. The successful application are the supercomputer Cray-2 (single phase forced convection with dielectric fluid FC-72) and SS-1 (single phase impinging flow with dielectric fluid FC-77) in the end of 1980 ages. Direct liquid cooling can reduce the packaging thermal resistance and improve the complexity of component efficiently [3].

Although the heat transfer performance of boiling can offer two order magnitude of heat flux value than that of single phase [4][5], however, both pool boiling and flow boiling are restricted in the nucleate boiling region for applications by two factors: boiling hysteresis and critical heat flux which are shown in Figure 1.2. Boiling hysteresis is a behavior of delay in the incipience of nucleate boiling. The heating surface is continually cooled by natural convection and surface temperature is raised with power input until the sufficient superheat is reached for initial boiling. High boiling incipience superheat will result in high thermal stress of material and will reduce the operation life of component. As the surface heat flux of component is increased to approach the Critical Heat Flux (CHF), the temperature of component will be raised rapidly to reach the melding point and result the burnout of component.

As mentioned above, pool boiling is a high efficiency heat transfer technology but the hysteresis and CHF restrict its utilization. Therefore, the techniques to reduce the boiling incipience superheat and to increase the CHF value will extend the limitation of boiling heat transfer in electronic cooling applications.

1.2 Literature Review

Literature Review-Enhance Pool Boiling

A direct immersion cooling in dielectric liquid with phase change is a good method for removing the large amount of heat generated from high power density devices. There are several parameters which can be used to enhance the pool boiling heat transfer performance such as the operation pressure, sub-cooling of working fluid, dissolved gas content, surface orientation and structure.

1.2.1 Operation Pressure

The effects of pressure on the nucleate boiling performance of flat surfaces have been well studied in the literature. Increased pressure has been found to improve the nucleate boiling performance. Nishikawa et al. [6] attributed this behavior to an increased range of cavity radius that might be activated at a given wall superheat with increased pressure (increased active nucleation site density). As shown by Cichelli and Bonilla [7] and others, the CHF value could be raised up to about one-third of the critical pressure and then be reduced with increasing pressure. The effect of pressure on the nucleate boiling and CHF performance of finned heaters appeared to be consistent with flat heater observations [8-10]. In addition, Abuaf et al. [8] observed that the CHF for their flat and finned surfaces did not follow Zuber's [11] CHF correlation at very low pressures but leveled off with decreasing pressure instead.

1.2.2 Sub-cooling of Working Fluid

Sub-cooling has been found to have little or no effect on the fully developed nucleate boiling performance [12], however, CHF can be raised with increased sub-cooling with a strong dependence [13-14]. Mudawar and Anderson [3] studied the effects of sub-cooling on various cylindrical fin arrays in FC-72 at 1 atmosphere. Except near the neighborhood of CHF, their results showed insensitivity of the nucleate boiling curve to sub-cooling and could dissipate up to 160 Wcm^{-2} with a sub-cooling of 35 K. To the author's knowledge, no studies have addressed the effects of dissolved gas on finned surfaces, however, Rainey [15] observed a dramatic change in the slope of nucleate boiling curve of finned surfaces at high heat fluxes which indicated that sub-cooling and/or dissolved gas might significantly affect the boiling performance in the high heat flux region.

1.2.3 Dissolved Gas

Since the highly wetting fluids used in electronics cooling research can typically absorb large amounts of non-condensable gases, the effects of dissolved gases on the boiling performance are also critical to understand. McAdams et al. [16] reported a strong enhancement of dissolved gas on the boiling curve at low heat fluxes (partially developed nucleate boiling) but only a weak influence at high heat fluxes (fully developed nucleate boiling). Watwe and Bar-Cohen [17] observed no effect of dissolved gas on CHF.

1.2.4 Surface Orientation

In some practical applications, the electronic device may be placed in vertical orientation because of the limitation of system working space. Therefore, numerous experimental studies have been carried concerning the orientation effect on boiling heat transfer. Marcus and Dropkin [18], Githinji and Sabersky [19], and Nishikawa et al. [20] observed that boiling heat transfer coefficients could be increased with the increasing in inclination angle at low heat flux region. Bonjour and Lallemand [21-22] classified the boiling flow patterns on vertical heated surface into three regimes: the regime of isolate bubble or partial nucleate boiling $q''/q''_{crit} > 0.2$, fully developed nucleate boiling for $0.2 < q''/q''_{crit} < 0.7$ and the transition to critical heat flux for $q''/q''_{crit} > 0.7$. Fujii et al. [23] explained the enhancement of boiling heat performance on vertical surface as the results from the more intense agitation of thermal boundary layer by departing bubbles rising from the surface.

Kumagai et al. [24] found that the longitudinal fin orientated on vertical base surface provided better, more stable heat transfer performance than the vertical fin orientated on horizontal base surface. Guglielmini et al. [9] observed that square pin fin array surfaces on the horizontal base orientation provided slightly better heat transfer performance than the vertical base orientation. Rainey et al. [25] also found the same trend on their micro-porous, square pin fin array boiling in FC-72.

Bar-Cohen and Schweitzer [26] investigated vertical isoflux channel boiling in water with various plate spacing. The authors found that wall superheat decreased as the channel spacing narrowed. Rampisela et al. [27] experimentally studied the influence of orientation and gap size of the channel filled with water for different peripheral condition of the sides of channel. An improvement of the heat transfer coefficient of about 100% was found on the test channel with a lowest gap size (1mm) compared with that obtained in the unconfined situation. The increase of thermal performance was resulted in sluggish flow observed during the experiments. Hwang and Moran [28] analyzed the effects of orientation and gap size on boiling heat transfer performance. The result showed no heat transfer change until the gap size was reduced to 0.51mm. For this value of the gap size, an improvement of heat transfer was detected at low heat flux, and a reduction of 70% in the critical heat flux was found compared with that of unconfined condition. Xia et al. [29] used R-113 as working fluid and studied the influence of gap size, plate height, and heat flux on natural convective boiling with a narrow vertical channel. The range of gape size investigated was from 5 mm to 0.8 mm. The authors observed that the narrowest gap size could reach as much as a fivefold enhancement of heat transfer in nucleate boiling region but, in the case of high value of heat flux, the same channel width produced a drastic reduction of CHF. Kim et al. [30] investigated the pool boiling on one-dimensional inclined rectangular channels by changing the orientation. It was observed that the CHF decreased as the surface inclination angle increased and as the gap size decreased. Bergles and Misale [31] studied two dielectric fluids boiling in vertical narrow channel with various gap sizes. The results showed that by reducing the channel width would decrease the heat transfer coefficient in natural convection, but would increase the heat transfer coefficient in nucleate boiling. Monde [32-33]experimentally and theoretically studied heat transfer enhancement due to the bubbles passing through a vertical narrow channel with asymmetric heating. Howard and Mudawar [34-35] experimentally and theoretically analyzed orientation effects on pool boiling critical heat flux for nearly vertical surfaces. They found that surface orientations could be divided into three regions: upward-facing (0° to 60°), near-vertical (60° to 165°) and downward-facing ($>165^\circ$). Each region was associated with a unique CHF trigger mechanism.

1.2.5 Surface Structure

Some surface area enhanced technologies by using extend surfaces like fin array boiling in dielectric fluid have been studied broadly. Bondurant and Westwater [36] and Klein and Westwater [37] experimentally analyzed the transverse fins and cylindrical spins surfaces boiling in R-113 and water. The test results showed that the fin spacing could be as close as to 1/16 inch without boiling bubbles interference and closer spacing would cause reduction in the heat duty. Haley and Westwater [38] observed a long single horizontal fin boiling in isopropyl alcohol and R-113. The results showed that various heat transfer modes including the stable film, transition, and nucleate boiling along with free convection could occur simultaneously along a fin indicating the extremely complex nature on boiling fins. Mudawar and Anderson [3] investigated low-profile microstructure cylindrical surfaces boiling in FC-72 and FC-87. They proposed a numerical analysis method of a single fin model to predict the boiling heat transfer performance of their multiple fin surfaces with reasonable accuracy. However, Siman-Tov [39] found that the analysis of a single fin might not suitable for the fin array. Guglielmini et al. [9, 40] studied the effects of orientation, geometrical configuration for extended surface boiling in HT-55 and FC-72 and found straight fin surfaces could provide better boiling performance compared with plane surfaces. Their experiment also showed that, for the same surface superheat, the vertical orientation finned surfaces allowed higher heat duty than the horizontal arrangement and uniformly spaced finned surfaces showed slightly better boiling performance. Moreover, You et al. [15, 41] studied the plain and micro-porous, square pin fin array in saturated FC-72 and showed that pin fins produced resistance to vapor/bubble departure and longer bubble residence time. This would increase the flow resistance to the approaching re-wetting liquid and caused localized dry-out situation near the base of the fins. Hirono [42] proposed that the interference between adjacent rectangular fins might reduce the boiling heat transfer performance in the high heat flux heat region.

1.2.6 Micro Structure

Another method which has been studied extensively for the enhancement of pool boiling heat transfer is the small-scale surface enhancement using a variety of techniques: surface roughening, porous metallic coatings, microporous coatings and MEMS fabrication as well as others. MEMS devices use integrated circuit fabrication techniques to create extremely small devices, from 1 μm to 1 mm in length. Zhang and Shoji [43] studied the physical mechanisms of nucleate site interaction in pooling boiling and found that there are three crucial effect factors: hydrodynamic interaction between bubbles, thermal interaction between nucleate sites, horizontal and declining bubble coalescences. Besides, Shoji and Takagi [44] also constructed a single artificial cavity to analyze the nonlinear behavior and low dimensional chaos of the bubbles. Recently, Honda [45] observed silicon chip with the micro-pin-fin boiling in FC-72 and found micro-pin-fin chip showed a considerable heat transfer enhancement. Phadke et al. [46] investigated a silicon re-entrant cavity surface boiling in saturated and subcooled R-113. The authors observed a significant reduction in boiling incipience superheat and temperature excursion.

1.2.7 Aging

Czikk et al. [51] and Ogata and Nakayama[52] observed that a thin oxide layer could be formed on the heating surface if the heating surface was placed in the air after experimental run. Then particles in the air might be accumulated and trapped in the nucleate cavities on the test surfaces in the result of reduction of the nucleate sites. These oxidization and particle trap mechanism ultimately reduce heat transfer coefficient (the boiling curve shift to right).

Literature Review-Investigations of Boiling Characteristics

1.2.8 Flow Pattern Visualization

The pool boiling mechanism is very complex in consideration with several parameters including the latent heat transfer, natural convection and micro-convection. Several investigators used visualization as a tool to quantify the relative contribution of the above mentioned parameters. Nakayama et al. [47-49] were among the first to carry out visualization of the boiling process from a structured surface. The surface consisted of a rectangular channel covered with a thin sheet with pores at a regular pitch and the entire structure was immersed in R-11. Arshad and Thome [50] conducted visualization from similar surfaces to investigate the mechanism of boiling inside the channels. Xia et al. [29] observed that the boiling hysteresis attenuated as the channel narrowed and the first local bubble site was active only on the upper part of the narrow channel.

1.2.9 Correlation Equations for Pool Boiling on FC-72

Base on the available experimental data and theoretically analysis from the open literature, some correlating equations for pool boiling were proposed.

The complete process of liquid heating, nucleation, bubble growth, and departure, collectively referred to as the ebullition cycle, is the major mechanism of heat transfer from a superheated wall during the nucleate boiling. Two factors of this process that affect the rate of heat transfer during the ebullition cycle are the bubble diameter at departure, as well as the frequency at which bubbles are generated and released. For liquid which wets the heating surface, the size of bubbles at departure from the heating surface has been studied by a number of researchers. Cole and Rohsenow [53] correlated the departure diameter for various fluids. Base on an analogy between the bubble release process and natural convection, Zuber et al. [54] proposed a correlation for bubble departure frequency. Rohsenow [55] postulated that heat flows from the surface first to the adjacent liquid, as in any single-phase convection process, and that the high heat transfer coefficient associated with nucleate boiling is a result of local agitation due to liquid flowing behind the wake of departure bubble. Stephan and Abdelsalam [56] proposed nucleate boiling correlations for several classes of fluids (water, organics, refrigerants and cryogens) using statistical regression techniques. Cooper [57] developed a nucleate boiling heat transfer correlation base on reduced pressure. A correlation in terms of the wall superheat, the Prandtl number, a surface-liquid interaction parameter (the ratio of the thermal conductivity, density, and specific heat of solid to the liquid), and a dimensionless surface roughness parameter was developed by Benjamin and Balakrishnan [58]. Some empirical correlations proposed in the literature for pool boiling heat transfer are summarized in Table 1.1. As dielectric fluid is expected to be the operating liquid in electric cooling application, hence, many of studies have been develop correlations for dielectric fluid by using the power-law curve-fit (a simplified form the Rohsenow's correlation [55] [3] [15] [25] [40]

[59]). The detailed correlations are list in Table 1.2.

The CHF represents the transition from nucleate boiling to film boiling and is associated with the formation of a film of vapor which partially blankets the heater surface. Under constant heat flux, as often encountered in electronic cooling situations, CHF can produce a catastrophic increase in temperature of the surface and component. Zuber [11] derived an analytical relation for CHF on an infinite, upward-facing horizontal surface by assuming it was hydrodynamically controlled and that vapor departed the boiling surface in cylindrical jets. This analysis yielded a relation for CHF. Rainey et al. [25] observed that four parameters to affect CHF in the experiment are: pressure, subcooling, fin length, and surface microstructure (plain or microporous). The authors correlated these parameters with experimental data in order to establish correlations for predicting the CHF value on finned surfaces. Later, Chang and You [60] also developed an empirical CHF correlation relevant to orientation. The relevant correlations on CHF value prediction/estimation are listed in Table 1.3. Finally, Table 1.4 lists a summary of the studies for dielectric fluid boiling heat transfer considering the effects of pressure, sub-cooling, dissolve-gas, and fin geometry on finned surfaces.

1.3 Objectives and Scope of Present Study

The above literature review clearly reveals that although the pool boiling heat transfer on boiling enhanced surface including mini-finned surface, micro-finned surface and cavity surface have been extensively studied, however, the pool boiling heat transfer mechanism and flow patterns on boiling enhanced surfaces are still poorly understood. It is relatively important in fundamental research to explore the characteristics of heat transfer and flow patterns with various geometry parameters and orientations.

In this present study, a series of pool boiling experiments are established and employed in order to investigate the boiling heat transfer characteristics including the boiling incipience superheat, heat transfer coefficient and critical heat flux on mini-finned surfaces, micro-finned surfaces and micro artificial-cavity surfaces immersed in the dielectric fluid with various geometry parameters and orientations. Moreover, the direct visual observation is also used to investigate the flow pattern of the tested surfaces for understanding the boiling heat transfer mechanism of the enhanced structure.

Table 1.1 Heat transfer correlation for pool boiling.

Reference	Heat Transfer Correlations
Cole and Rohsenow [53]	$D_d = \left(\dfrac{Bo\sigma}{g(\rho_l - \rho_v)}\right)^{0.5}$ $Bo = (0.04Ja)^2$ $Ja = \left(\dfrac{\rho_l Cp_l \Delta T_{sat}}{\rho_v i_{lv}}\right)$
Zuber [54]	$fD_b = 0.59\left[\dfrac{\sigma g(\rho_l - \rho_v)}{\rho_l^{\ 2}}\right]^{0.25}$
Rohsenow [55]	$\dfrac{\dot{Q}}{A} = h\Delta T = \dfrac{A_{mc}}{A} q''_{mc} + \dfrac{A_{nc}}{A} q''_{nc}$ $q''_{mc} = \dfrac{2}{\sqrt{\pi}}\sqrt{k_l \rho_l Cp_l}\sqrt{f}\,\Delta T = h_{mc}\Delta T$ $h_{mc} = \dfrac{2}{\sqrt{\pi}}\sqrt{k_l \rho_l Cp_l}\sqrt{f}$ $q''_{nc} = h_{nc}(T_w - T_{sat})$ $\left[\dfrac{Cp_l \Delta T_{sat}}{i_{iv}}\right] = C_{sf}\left[\dfrac{q''}{\mu_l i_{lv}}\left(\dfrac{\sigma}{g(\rho_l - \rho_v)}\right)^{1/2}\right]^n \left[\dfrac{Cp\mu}{k}\right]_l^{m+1}$
Stephan and Abdelsalam [56]	$h = 207 \cdot \dfrac{k_l}{D_b}\left(\dfrac{q'' D_b}{k_l T_{sat}}\right)^{0.745}\left(\dfrac{\rho_v}{\rho_l}\right)^{0.581}\left(\dfrac{\nu_l}{\alpha_l}\right)^{0.533}$ $D_b = 0.0146\beta\left[\dfrac{2\sigma}{g(\rho_l - \rho_v)}\right]$
Cooper [57]	$h = 55(q'')^{0.67} M^{-0.5}\,\mathrm{Pr}^m(-\log_{10}(\mathrm{Pr}^{-0.55}))$ $m = 0.12 - 0.2\log_{10} R_a$
Benjamin and Balakrishnan [58]	$n = 218.8(\vartheta)^{-0.4}(\mathrm{Pr})^{1.63}\left(\dfrac{1}{\gamma}\right)(\Delta T)^3$ $\vartheta = 14.4 - 4.5\left(\dfrac{R_a P}{\sigma}\right) + 0.4\left(\dfrac{R_a P}{\sigma}\right)^2$

Table 1.2 Pool boiling heat transfer correlation for dielectric fluid.

Reference	Heat Transfer Correlations	Heating Surface Mode
Mudawar [3]	$q''_b = a \cdot \Delta T_b^n$ a and n varied with ΔT_b and structure	Plain and cylinder finned surfaces
Rainey [15]	$q''_b = 1600 \cdot \Delta T_b^{2.4}$ for $q''_b < 5 \times 10^5 W/m^2$ $q''_b = 1.08 \times 10^5 \cdot \Delta T_b^{0.62}$ for $q''_b > 5 \times 10^5 W/m^2$	Microporous coated, finned surfaces, saturated state
Rainey [25]	$q''_b = 5.49 \times 10^4 \Delta T_b - 8.68 \times 10^6 P_{sys}{}^{-0.814}$ for low heat flux region $q''_b = (284 P_{sys} + 1670 \Delta T_{sub} + 84200) \cdot \Delta T_b^{0.6}$ for high heat flux region	Microporous coated, finned surfaces, saturated and gas saturated state
Guglielmini [40]	$h_{ext} = 5.55 q''_{ext}{}^{2/3}$ $q''_{ext} = 170 \Delta T_{sat}{}^3$	Uniform and non-uniform configuration finned surfaces
Rainey [59]	$q''_b = 1100 \cdot \Delta T_b^{2.4}$	Microporous coated, plain surfaces

Table 1.3 Critical heat flux correlations for pool boiling.

Reference	Heat Transfer Correlations
Zuber [11]	$q''_{CHF,Z} = (\pi / 24) h_{lv} \rho_v^{1/2} [g\sigma(\rho_l - \rho_v)]^{1/4}$
Rainey [25]	$q''_{t,CHF,sat} / q''_{CHF,Z} = 1.23 - 51L$ $q''_{t,CHF,sat} = q''_{b,CHF,sat} / (1000L + 1)$ $q''_{CHF,sub} = q''_{CHF,sat} [1 + 0.84(\rho / \rho)^{1/4} Ja / Pe^{1/4}]$
Chang and You [60]	$q''_{CHF} / q''_{CHF,0^\circ} = 1.0 - 0.0012 \cdot \theta \cdot \tan(0.414\theta) - 0.122 \cdot \sin(0.318 \cdot \theta)$

Table 1.4 Pool boiling heat transfer studies for finned surfaces.

Reference	Fluid	Experimental Parameters				Flow pattern visualization	CHF
		Pressure	Sub-cooling	dissolved-gas	Geometry		
Mudawar [3]	FC-72 FC-86	-	0-35℃	-	Microstud-enhanced stud surfaces Stud length = 40mm Stud diameter = 12.7mm	-	$159 W/cm^2$
Rainey [15]	FC-72	-	-	-	Microporous coated pin fin array surfaces $S = 1mm$, $L = 1-8mm$	-	$129.4 W/cm^2$
Rainey [25]	FC-72	30-150kPa	0-50℃		Microporous coated pin fin array surfaces $S = 1mm$, $L = 1-8mm$	-	$140 W/cm^2$
Guglielmini [9]	HT-55	50-200kPa	-	-	Square pin fin array surfaces $S = 0.4-0.8mm$, $L = 3mm$	-	$100 W/cm^2$
Guglielmini [40]	FC-72	50-200kPa	-	-	Uniform and non-uniform configuration square pin fin array surfaces $W = 0.4-0.8mm$, $L = 3-6mm$	-	$75 W/cm^2$

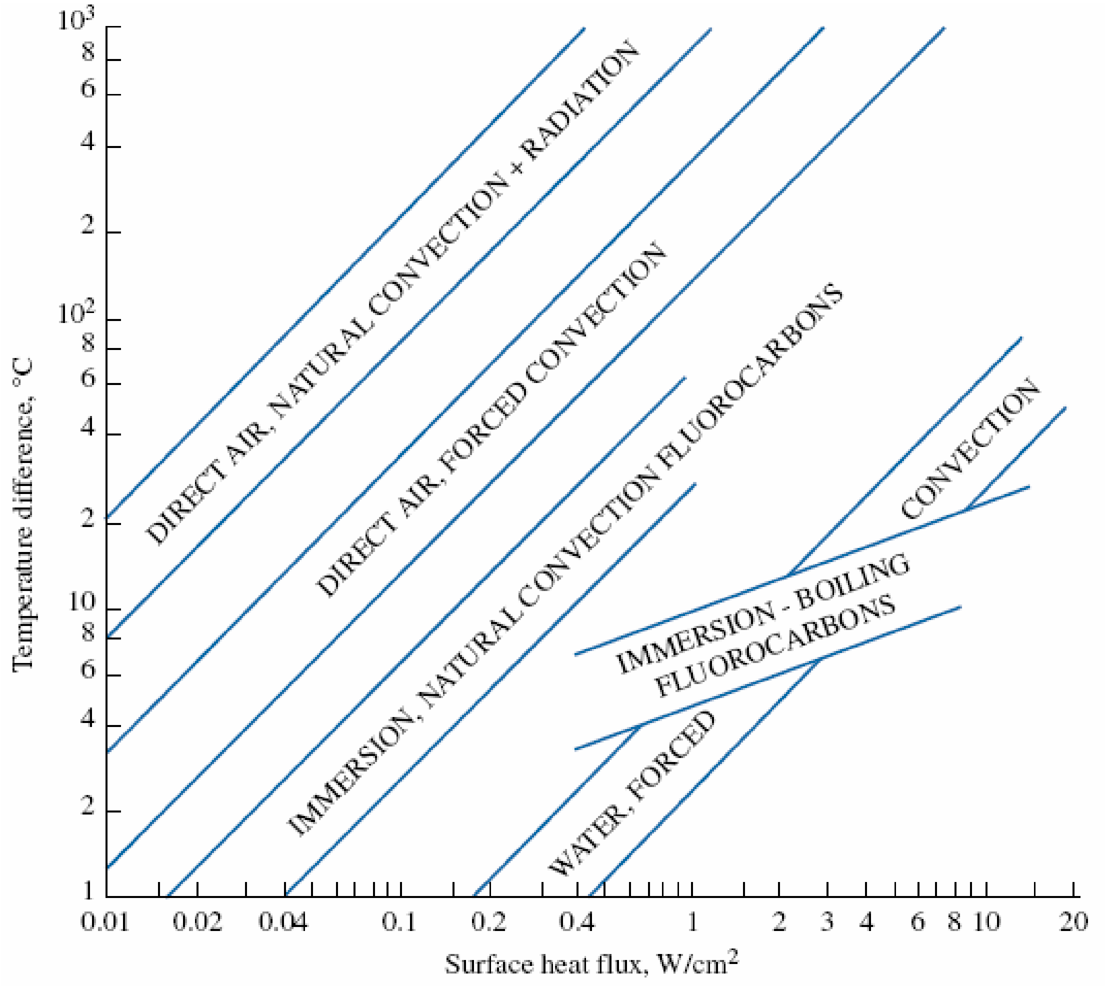

Figure 1.1 Temperature differences attainable as a function of heat flux for various heat transfer mode and various coolant fluids [2].

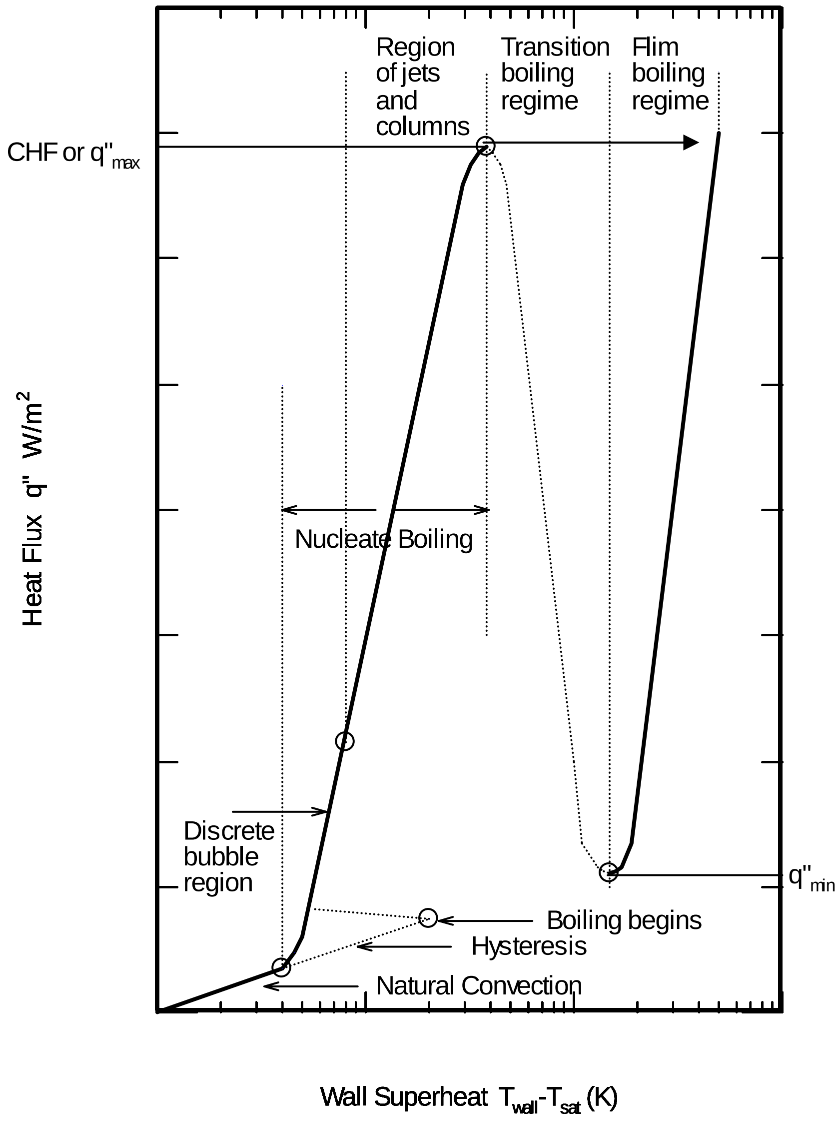

Figure 1.2 The typical boiling curve.