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**NANOSTRUCTURED SURFACE ENHANCED BOILING FOR ADVANCED
COOLING**

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SCHOOL OF MECHANICAL AND AEROSPACE ENGINEERING
NANYANG TECHNOLOGICAL UNIVERSITY
2014

NANOSTRUCTURED SURFACE ENHANCED BOILING FOR ADVANCED COOLING

SUBMITTED

BY

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ABSTRACT

With continuous increase in transistor packing density along with component miniaturisation, highly efficient cooling schemes are required to better cope with the exponential rise in chip level heat flux. In this aspect, nucleate pool boiling with the coating of carbon nanotube (CNT) onto heat transfer surfaces using chemical vapour deposition (CVD) technique has been shown to be an effective cooling method which could be applied in various cooling schemes. However, the thermal transport mechanisms from these surfaces and their nucleation boiling heat transfer characteristics under different parameters such as coating areas and surface orientation are still not well understood.

In this thesis, detailed investigations on saturated pool boiling heat transfer of CNT coated surfaces with FC-72 as the coolant fluid are presented. In total, two batches of CNT coated surfaces of different silicon substrate surface roughness and CNT coating configuration were produced. In the first batch, 30%, 60%, 90% and interlaced 60%-CNT surfaces were coated on bare silicon of 450 nm surface roughness whereas in the second batch, 100% and interlaced 70%-CNT surfaces were fabricated on bare silicon of 15 nm surface roughness. In addition to saturated pool boiling conducted at 0° (upright) surface orientation, a range of surface orientations from 0° to 180° for the second batch of CNT surfaces were also investigated.

The investigations revealed that CNT coating resulted in lower heat fluxes and lower wall superheats for boiling incipience as compared to the base substrate (bare silicon). In addition, critical heat flux (CHF) and nucleate boiling heat transfer were also

enhanced by the CNT coating where the magnitude of enhancement were found to increase with increasing CNT coating area and perimeter. On the other hand, increasing surface orientations exhibited adverse effect on the heat transfer performance of CNT coated surfaces where substantial reduction in the average heat transfer coefficient with surface orientations of 60° and above were observed. By performing high speed imaging, surface microstructure and surface wettability analyses, mechanisms for boiling incipience, nucleate boiling, CHF and the effect of surface orientations of the CNT coated surfaces on boiling heat transfer performance are elucidated.

With the experimental data obtained, correlations are proposed for the average heat transfer coefficient (h_{ave}) and CHF for the second batch of CNT surfaces with the effect of surface orientations. Finally, based on the Rohsenow model [24] for nucleate pool boiling, a general correlation for characterising the saturated pool boiling curves of CNT coated surfaces, including the effects of CNT coating area and perimeter, substrate surface roughness and surface orientations, was developed. The correlation shows good agreement with the experimental results.

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NOMENCLATURE

A	Area [m^2]
q''	Heat flux [W/cm^2]
h	Heat transfer coefficient [$\text{W}/\text{cm}^2 \cdot \text{K}$]
D_b	Vapour bubble departure diameter [m]
C_d	Constant coefficient from experimental data [-]
g	Gravitational constant [m/s^2]
U	Velocity [m/s]
h_{fg}	latent heat of evaporation [kJ/kg]
C_{pl}	Specific heat of working liquid [$\text{J}/\text{kg} \cdot \text{K}$]
C_{sf}	Coefficient in the Rohsenow correlation [-]
R	Surface roughness [nm]
T	Temperature [K]
ΔT	Wall superheat [K]
α	Thermal conductivity [$\text{W}/\text{m} \cdot \text{K}$]
f	Frequency [s^{-1}]
r_b	Radius of vapour embryo [m]
r	Cavity size [m]

Greek letters

ϕ	Fluid-surface contact angle [$^\circ$]
β	Cavity cone angle [$^\circ$]
μ_l	Liquid dynamic viscosity [$\text{Pa} \cdot \text{s}$]
ρ_l	Density of working liquid [kg/m^3]
ρ_g	Saturated vapour density [kg/m^3]
σ	Surface tension [N/m]
η	Kinematic viscosity [m^2/s]
θ	Surface orientation [$^\circ$]
δ	Thermal boundary layer [m]
ε	Error [%]

λ_H	Helmholtz critical wavelength [m]
λ_C	Taylor critical wavelength [m]

Dimensionless Numbers

Bo	Bond Number
Ja	Jakob Number
Nu	Nusselt Number
Pr	Prandtl Number
Ra	Rayleigh Number
Re	Reynolds Number

Subscripts

ave	Average
b	Bubble
c	Cavity
min	Minimum
max	Maximum
NC	Natural convection
sat	Saturation
w	Wall
∞	Bulk fluid

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PUBLICATIONS ARISING FROM THIS THESIS

Journal Paper

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CHAPTER 1

INTRODUCTION

1.1 Background

1.1.1 Thermal management challenges in modern electronic devices

Since 1970, integrated circuit (IC) chips saw a two-fold increase in transistor count every two years as projected by Moore [1]. Today, the transistor count has continued to increase beyond several billions as compared to hundreds of thousands in the early 1980s due to the advancement in “very large-scale integration” (VLSI) and BiCOMS technologies. This continuous increase in transistor packing density along with component miniaturisation resulted in the exponential rise in chip level heat flux, reaching 50 to 100 W/cm² [2], and hence, efficient cooling schemes are critical to maintain the chip within benign operating temperature. However, due to the poor thermal transport properties of air, conventional force convection cooling techniques with this medium were restricted to relatively low heat flux (between 0.001 and 0.01 W/m²·K) and were unable to cope with the increasing heat dissipation requirements of modern chips. The immersion cooling technique with significantly greater heat removal capability shed light into alleviating the thermal management challenges when it was successfully developed and implemented in modern supercomputers such as the Cray-2 and ETA 10. As the name suggests, this technique involves direct immersion of chip packages in the coolant fluid and allowed heat removal through a passive phase change process of the coolant, otherwise known as nucleate pool boiling heat transfer. Dielectric fluids emerged as viable coolants for immersion cooling application owing to their electrical compatibility and low boiling points, and have

been widely studied. Mudawar [3] provided a comparison of various cooling schemes and showed that heat transfer with nucleate pool boiling with dielectric fluid performed almost 10 times better than single phase forced convection. Bar-Cohen [4] advocated the use of dielectric fluids for their low boiling points which reduced electronic components temperature excursions and failure rates, and evaluated the potential applications of nucleate pool boiling. Despite the various advantages highlighted above, the inferior thermophysical properties of dielectric fluids which resulted in poorer nucleate boiling heat transfer coefficient and lower critical heat flux (CHF) as compared to water, remain the main drawback of using dielectric fluids in immersion cooling.

To overcome this drawback, much attention in the past three decades were focused on surface modification techniques to further augment nucleate boiling heat transfer performance with relative success. For instance, Anderson and Mudawar [5] achieved remarkable enhancement in CHF (51.1 W/cm^2) by modifying a boiling surface with microstud structures and O'Connor et al. [6] reported up to 80% heat transfer enhancement with a surface painted with diamond microstructure. With the advancement in nanotechnology and lithography techniques, several novel surface modification techniques such as nanoparticle coating [7], nanowires synthesis [8] and porous media heat sink [9], as shown in Fig. 1.1, have also demonstrated remarkable potential in enhancing nucleate boiling performance. Along with the advent in these surface modification techniques, it was also noticed that conventional theories of nucleate boiling such as the microconvection heat transfer model [10] and the hydrodynamic instability model for CHF [11] were unable to provide reasonable predictions of their heat transfer characteristics. Hence, a detailed parametric study

enhancement in CHF (1.28 MW/m^2) as compared to a plain copper surface (1.14 MW/m^2). Similarly, Pialago et al. [11] coated surfaces with MWCNT-Cu composite powder using the cold spraying technique which also resulted in random arrangement of the CNT structures. The coated surfaces produced have CNT concentrations between 5 and 15 vol% and the coating thicknesses were measured to be between $15.5 \text{ }\mu\text{m}$ and $178.3 \text{ }\mu\text{m}$. Pool boiling of these surfaces was studied with saturated R134a and the results showed enhancement in heat transfer for all CNT-Cu coated surfaces as compared to plain Cu surface. As high as 74% enhancement was recorded for the surface coated with the highest CNT concentration.

The second class of CNT coated surfaces was initially investigated by Ujereh et al. [12] who conduct pool boiling experiments on MWCNT synthesised with plasma-enhanced chemical vapour deposition (PECVD) as shown in Fig. 1.2(a). The diameter of the CNT was approximately 50 nm and the lengths varied between 20 and 30 μm . Pool boiling experiments were conducted with FC-72 and their results revealed that the CNTs enhanced CHF by 45%. Furthermore, the fully coated CNT surface also displayed as high as 450% improvement in heat transfer coefficient as compared to bare silicon. Subsequently, Ahn et al. [13] investigated the pool boiling heat transfer performance of vertically aligned CNT arrays produced with CVD technique as shown in Fig. 1.2(b). The CNTs were synthesised on an atomically smooth silicon surface and the coating thickness was between 9 and 25 μm . Both saturated and subcooled boiling experiments were performed with PF-5060 and the results showed moderate heat transfer enhancement for CNT coated surfaces as compared to bare silicon in the nucleate boiling regime. For instance, at wall superheat of 10°C under saturated

boiling, 64.6% and 33% higher heat fluxes were observed for surfaces coated with 9 and 25 μm thick CNTs, respectively.

Fig. 1.2 SEM images of (a) MWCNT from Ujereh et al. [12] (b) MWCNT from Ahn et al. [13]

Despite the remarkable pool boiling heat transfer performance of CNT coated surfaces demonstrated by different investigators above, published pool boiling data with surfaces coated using CVD technique remain scarce. In addition, significant discrepancies in heat transfer enhancement with CNT coated surfaces are evident among the published data and the key thermal transport mechanisms involved are still unclear. Moreover, the effects of variation in experimental parameters such as CNT coating area and surface orientation have also not been determined. All these factors render further investigation with CNT coated surfaces necessary in order to provide a complete understanding of the nucleate pool boiling characteristics.

1.2 Objectives and Scope

This thesis seeks to provide a more comprehensive understanding on the pool boiling heat transfer characteristics of CNT coated surfaces with dielectric fluid. The CNT coated surfaces in this study are prepared using CVD technique. Surfaces with

different CNT coating coverages and patterns are prepared and tested under various boiling conditions and surface orientations. Through the aid of high speed visualisation and the experimental data obtained, this research also aims to identify the key mechanisms involved in pool boiling on CNT coated surfaces and to develop correlations for the predicting their pool boiling performance. The fundamental goals are to engineer CNT coated surfaces with optimal heat dissipation capability and to identify conditions favourable for practical application of CNT coating in immersion cooling.

The scope of this research includes the following:

1. Parametric study on the effects of CNT coated surfaces of different coating patterns and area coverages on pool boiling heat transfer.
2. Investigation on the effects of surface orientation on pool boiling heat transfer performance of CNT coated surfaces.
3. High speed visualisation of pool boiling phenomenon to gain knowledge on the bubble dynamics for different surfaces and their effects on pool boiling heat transfer.
4. Formulation of correlation to predict the pool boiling performance of bare silicon and CNT coated surfaces.
5. Numerical thermal analysis to evaluate the accuracy of the experimental setup and experimental results.

1.3 Outline of Thesis

There are seven chapters in this thesis. Chapter 1 introduces the existing challenges in thermal management of electronic devices. This is followed by a brief overview on the

current research trends and setbacks in overcoming the thermal crisis where the motivation, scope and objectives for the present research are established. Chapter 2 is a detailed literature review on the fundamentals of nucleate pool boiling heat transfer that were developed over the last six decades and the recent developments in surface modification techniques for enhancing boiling heat transfer. Chapter 3 depicts the fabrication techniques and configurations of CNT coated surfaces produced for present research. In addition, details are also provided on the experimental setups employed to investigate the pool boiling heat transfer performance of CNT coated surfaces. In Chapter 4, investigation on the effects of CNT coating area coverages and perimeters on the saturated pool boiling heat transfer performance are presented and analysed. In addition, the possible thermal transport mechanisms involved are also identified. In Chapter 5, the combined effects of CNT coating and surface orientations on the saturated pool boiling heat transfer performances is presented and analysed, and attempts are made to determine the average heat transfer coefficient and CHF for the surfaces tested. A general correlation is also proposed for the prediction of the saturated pool boiling curves of CNT coated surfaces incorporating the effects of CNT coating area, CNT coating perimeter and surface orientation in Chapter 6. Finally the key results of the present research and the direction for future works are summarised in Chapter 7.

CHAPTER 2

LITERATURE REVIEW

2.1 Overview

The pool boiling curve which described the heat flux dissipated from a heated surface in relation to its wall superheats was obtained experimentally by Nukiyama in 1934 [14]. Figure 2.1 resembles the pool boiling curve from Nukiyama's experiments and is typically observed in most pool boiling processes where four distinct stages (natural convection, nucleate boiling, transition boiling and film boiling) could be demarcated based on the differences in their physical phenomenon. Due to the superior heat transfer performance in the nucleate boiling stage as compared to the three other stages, attempts to comprehend and quantify this complex phenomenon are of particular interest to many researchers over the past 80 years. The occurrences of boiling incipience and the critical heat flux (CHF) which respectively marked the beginning of the nucleate boiling stage and transition into film boiling have also been intensively studied. In addition, impelled by the advancements in microelectromechanical systems (MEMS) and nanotechnology as well as the need for more effective cooling methods to better cope with the increasing heat flux of electronic devices, the possibility of augmenting pool boiling heat transfer with micro/nanoscale modified surfaces has also been extensively studied in the last three decades.

This literature review serves to summarise and evaluate published works on the fundamentals of nucleate pool boiling and the recent developments in pool boiling of micro/nanostructured modified surfaces. It is comprised of two parts; in the first part, the various mechanistic models and empirical correlations proposed for the boiling

incipience, nucleate pool boiling heat transfer and CHF are presented. In the second part, recent developments in pool boiling heat transfer with the various types of micro/nanostructured surfaces and attempts by researchers to explain the phenomenon involved in this heat transfer process are highlighted.

Fig. 2.1 Nukiyama's pool boiling curve [14]

2.2 Pool boiling models

Dominated by different transport mechanisms, the nucleate boiling stage and CHF incipience were often treated as disjoint processes. For instance, mechanisms such as single-phase forced convection, transient conduction and microlayer evaporation contributed to the heat transfer rate in the nucleate boiling stage whereas the occurrence of CHF were mainly driven by the fluid-vapour instability and the macrolayer evaporation near the solid-fluid-vapour interface. On the other hand, incipience of nucleate boiling and bubble growth were seen to depend largely on surface cavities, wall superheat and thermal boundary thickness. Hence, existing pool boiling models could be further categorised into (1) models for bubble growth and

boiling incipience, (2) models for nucleate pool boiling heat transfer and (3) CHF models.

2.2.1 Models for bubble growth and boiling incipience

In heterogeneous boiling, bubbles are known to grow from the pre-existing vapour embryos that are trapped in the cavities of the heated surfaces. Based on an idealised conical shaped cavity, Bankoff [16] and Lorenz [17] proposed the effective cone angle for vapour entrapment shown in Eq. (2.1), where ϕ is the fluid-surface contact angle and β is the cavity cone angle. Subsequently, Yang and Kim [18] set the criteria for maximum cavity size for vapour entrapment by resolving the forces acting on the fluid-vapour interface and obtained Eq. (2.2).

$$\beta < \phi/2 \quad (2.1)$$

$$r_{c,max} = \sqrt{\frac{4\sigma\cos^2(\phi-\beta)}{[(\rho_l-\rho_g)g(1-\sin(\phi-\beta)) \times [1 + \frac{(\sin(\phi-\beta)-1)^2}{3\cos^2(\phi-\beta)}]}} \quad (2.2)$$

The minimum cavity size for bubble growth in relation to the wall superheat was first established by Griffith and Wallis [19]. It was assumed that the entrapped vapour embryo attained the size equal to the cavity size in order for the bubble growth to initiate. In addition, by assuming that thermal and static equilibriums of the vapour embryo at the cavity mouth, the Young-Laplace equation, Eq. (2.3) and the Clausius-Clapeyron equation [Eq. (2.4)] were employed and equated to obtain Eq. (2.5) which shows that, apart from the high dependence on the thermophysical properties of the fluid, the minimum cavity size for boiling incipience also reduces with increasing surface temperature. This also meant that the nucleation sites that were activated at

lower heat fluxes were generally larger than those activated at higher heat fluxes. Hence, the increase in heat flux also resulted in more nucleation sites being activated. However, the main drawback of Griffith and Wallis's model lies in its inability to predict the upper limit of the cavity size. As seen from Eq. (2.5), with increasing cavity size, the wall superheat required for boiling incipience approaches zero. This does not happen as it has been ascertained from experiments that a certain amount of wall superheat is required to initiate boiling.

$$p_v - p_l = \frac{2\sigma}{r_c} \quad (2.3)$$

$$\frac{dp}{dT} = \frac{h_{fg}}{T(v_v - v_l)} \quad (2.4)$$

$$r_{c,min} = \frac{2\sigma T_{sat} v_{fg}}{h_{fg}(T_w - T_{sat})} \quad (2.5)$$

Two separate models for predicting the minimum and maximum cavity sizes for boiling incipience were proposed by Hsu [20] and Han and Griffith [21]. In both models, the formation of a transient thermal boundary layer above the vapour embryo was seen as the driving mechanism for bubble growth. This thermal boundary layer which was warmer as compared to the vapour embryo resulted in net heat flux into the vapour embryo to provide heat of vaporisation. The thermal boundary layer thickness and temperature profile within the thermal boundary layer varied spatially and changed with time. As the bubble departs, the thermal boundary layer surrounding the bubble would also be carried away from the surface and the surrounding cooler bulk fluid quickly replenished the vacated site for the next growth cycle to begin. Based on the

above concept, the one-dimensional transient heat conduction equation was employed to obtain the temperature profile for the thermal boundary layer. Hsu [20] stipulated that the temperature of the vapour embryo required to initiate growth under thermostatic equilibrium is given by Eq. (2.6) where r_b is the radius of the vapour embryo and T_b represents the temperature of the vapour embryo. The main difference between Eqs. (2.5) and (2.6) is that the T_w term in Eq. (2.5) is replaced with T_b in Eq. (2.6). This difference arises as the vapour embryo model proposed by Hsu [20] was assumed to protrude out of the cavity mouth prior to growth and the height of the vapour embryo, r_h , was dependent on its contact angle and the cavity mouth angle as shown in Fig. 2.2 and Eq. (2.7). T_b was taken from the top of the vapour embryo. On the other hand, Griffith and Wallis [19] assumed that the vapour embryo radius was the same as the cavity radius and the bubble contact angle was fixed at 90° .

Fig. 2.2 Relationship between vapour embryo, contact angle and cavity mouth angle [20]

By equating Eq. (2.6), Eq. (2.7) and the transient thermal boundary layer equation, the range of active nucleation cavities related to the thermal boundary layer thickness, vapour embryo contact angle and wall superheats were determined by Eq. (2.8). Hsu's model allowed flexible application in saturated and subcooled conditions although the

measurement of the vapour embryo contact angle was not an easy task. For simplicity, Hsu [20] further assumed that r_h was twice that of r_c and determined C_1 and C_2 to be 2 and 1.6, respectively. In addition, by considering pool boiling under saturated condition, Eq. (2.8) could be further reduced to Eq. (2.11).

$$r_b = \frac{2\sigma T_{sat} v_{fg}}{h_{fg}(T_b - T_{sat})} \quad (2.6)$$

$$r_h = (1 + \cos\phi)r_b \quad (2.7)$$

$$r_{c \text{ (min,max)}} = \frac{\delta}{2C_1} \left\{ 1 - \frac{T_{sat} - T_\infty}{T_w - T_\infty} \pm \sqrt{\left[1 - \frac{T_{sat} - T_\infty}{T_w - T_\infty} \right]^2 - \frac{8T_{sat}\sigma C_2}{(T_w - T_{sat})\delta\rho_v h_{fg}}} \right\} \quad (2.8)$$

$$C_1 = \frac{(1 + \cos\phi)}{\sin\phi} \quad (2.9)$$

$$C_2 = 1 + \cos\phi \quad (2.10)$$

$$r_{c \text{ (min,max)}} = \frac{\delta}{4} \left[1 \pm \sqrt{1 - \frac{12.8T_{sat}\sigma}{(T_w - T_{sat})\delta\rho_g h_{fg}}} \right] \quad (2.11)$$

As opposed to Hsu [20], Han and Griffith [21] employed Eq. (2.5) and developed the active nucleation cavities model as shown in Eq. (2.12). It is observed that Eq. (2.12) takes the same form as Eq. (2.11) and is dependent on the wall superheat and thermal boundary layer but does not account for the variation in contact angle as ϕ was originally fixed at 90° .

$$r_{c \text{ (min,max)}} = \frac{\delta(T_w - T_{sat})}{3(T_w - T_{\infty})} \left[1 \pm \sqrt{1 - \frac{12(T_w - T_{\infty})T_{sat}\sigma}{(T_w - T_{sat})^2 \delta \rho_g h_{fg}}} \right] \quad (2.12)$$

In summary, Hsu's model could be viewed as the general equation which takes into consideration the dynamic condition of the thermal boundary layer developed over the vapour embryo which affects the range of cavity sizes that could be activated.

2.2.2 Nucleate pool boiling heat transfer models

Due to the complex nature of the processes involved, it is difficult for a single model to provide accurate prediction of the nucleate heat transfer characteristics for the wide variation of boiling conditions. Numerous models for nucleate pool boiling heat transfer have been developed over the last 60 years, with each model attempting to overcome the shortfall of its former. In general, these models can be classified as empirical-based correlations and mechanistic-based correlations. Empirical-based correlations mainly relate the heat transfer rate with fluid properties and surface characteristics whereas mechanistic-based correlations often require the knowledge of bubble dynamics and active nucleation site densities. Comprehensive reviews of existing nucleate pool boiling heat transfer models were provided by Dhira [22] and Kim [23] and the fundamental concepts of these models are briefly summarised in this section.

2.2.2.1 Empirical based models

One of the earliest models was proposed by Rohsenow [24] who envisaged the nucleate pool boiling process as a single phase force convection problem whereby heat was removed from the surface by the induced fluid motion as a result of the departing bubbles. The model could thus be developed based on the existing convection heat transfer relationship between Nusselt number, Reynolds number and Prandtl number as shown in Eq. (2.13). The characteristic length, L , of Reynolds number was assumed to be the bubble departure diameter where Rohsenow obtained from the experimental correlation proposed by Fritz [25], as shown in Eq. (2.14). Taking into consideration heat transfer by the departed bubbles as shown in Eq. (2.15) and the relation between bubble departure frequency and bubble departure diameter suggested by Jakob [26] represented in Eq. (2.16), the expression for the velocity component, U , of the Reynolds number was derived in Eq. (2.17). Substituting Eqs. (2.18) and (2.21) into Eq. (2.17), the final expression for the Rohsenow correlation containing the pool boiling heat transfer coefficient and wall superheat was obtained as Eq. (2.18).

$$\text{Nu} = \frac{1}{C_{sf}} \text{Re}^{1-r} \text{Pr}^{1-s} \quad (2.13)$$

$$L = D_b = C_d \phi \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \quad (2.14)$$

$$q'' = \frac{\pi}{6} D_b^3 N \rho_v h_{fg} f \quad (2.15)$$

$$f D_b = \text{constant} \quad (2.16)$$

$$U = \frac{q''}{\rho_l \cdot h_{fg}} \quad (2.17)$$

$$q'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_g)}{\sigma} \right]^{\frac{1}{2}} \left[\frac{C_{pl}(T_w - T_{sat})}{C_{sf} h_{fg} \text{Pr}_l^s} \right]^{\frac{1}{r}} \quad (2.18)$$

The terms C_{sf} , r and s are constants to be correlated with experimental data. From pool boiling experiments with most conventional fluids and surfaces, q'' was found to be proportional to $(T_w - T_{sat})^3$ and hence, r was chosen to be 1/3. C_{sf} and s , on the other hand, were influenced by the fluid and surface interaction and varied with different fluid-surface combinations. For instance, Piro [27] determined that pool boiling of water on copper surface yielded C_{sf} and s values of 0.015 and 0.81, respectively, whereas pool boiling of water on aluminium surface resulted in the C_{sf} and s values of 0.11 and 1.26, respectively. In addition, Liaw and Dhir [28] also verified that C_{sf} decreases linearly with increasing fluid-surface contact angle.

Subsequent to Rohsenow, a comprehensive study was performed by Stephan and Abdelsalam [29] where the method of regression analysis was applied to approximately 5000 pool boiling data. The analysis was carried out by equating Nusselt number to a set of dimensionless constant terms and a general correlation was developed as shown in Eq. (2.19). The correlation was applicable to different fluids and heater geometries with a mean absolute error of 22.3%. In addition, Eq. (2.19) could also be modified into four sub-correlations with lower mean absolute errors for four specific groups of fluids namely water, hydrocarbons, cryogenic fluids and refrigerants.

$$\frac{q'' D_b}{\Delta T k_l} = 0.23 \left(\frac{q'' D_b}{k_l T_{sat}} \right)^{0.674} \left(\frac{\rho_g}{\rho_l} \right)^{0.297} \left(\frac{h_{fg} D_b^2}{\alpha_l^2} \right)^{0.371} \left(\frac{\rho_l - \rho_g}{\rho_l} \right)^{-1.73} \left(\frac{\alpha_l^2 \rho_l}{\sigma D_b} \right)^{0.35} \quad (2.19)$$

For both the Rohsenow and Stephan and Abdelsalam models, it was observed that heat flux was proportional to $(T_w - T_{sat})^3$. However, Rohsenow's model suggested that heat

flux was related to the square root of gravitational acceleration, g , whereas Stephan and Abdelsalam's correlation showed that heat flux was independent of g . In addition, the exponential constants and coefficient, C_{sf} , of Rohsenow's model were required to be varied under different boiling conditions in order to achieve the required accuracy. In comparison, for Stephan and Abdelsalam's correlation, the exponential constants and coefficients terms were fixed. This method of correlating pool boiling data where Nusselt number was proportional to dimensionless parameters raised to fixed exponential constants was similarly proposed by Kutateladze [30], Labuntsov [31] and Kruzhilin [32]. The main advantage of this method is that the correlation is applicable to a wide range of fluids and boiling conditions without having to alter the constants although the accuracy of prediction would be compromised.

2.2.2.2 Mechanistic-based models

A mechanistic-based model for partial nucleate pool boiling (or isolated bubble boiling regime) was proposed by Mikic and Rohsenow [33]. In the model, the total heat flux of the heated surface was suggested to be dissipated by transient conduction around the active nucleation sites and natural convection around the inactive surface as shown in Eq. (2.20). The basic mechanism of transient conduction was originally proposed by Forster and Greif [34] and subsequent mentioned by Han and Griffith [21] where heat was assumed to be dissipated from the surface to the adjacent transient thermal boundary layer through conduction. As the bubbles grew in size and departed from the nucleation sites, this thermal boundary layer which surrounded the bubbles was also removed from the surface. In addition, the removal of the thermal boundary layer from each nucleation site does not overlap with the neighbouring sites and there was also no interaction between the vapour bubbles. Mikic and Rohsenow [33] assumed the

thermal boundary layer, which was removed by the departing vapour bubble to cover the surface with an area twice the bubble departure diameter and developed a relation for the heat dissipated by transient conduction as shown in Eq. (2.21). On the other hand, the expression for heat dissipated through natural convection was correlated by Han and Griffith [21] to be a function of Rayleigh number as shown in Eqs. (2.22) and (2.23). By combining Eqs. (2.21) and (2.22) or (2.23) into Eq. (2.20), the overall heat flux in the partial nucleate boiling could be obtained.

$$\frac{q_{total}}{A_{total}} = \left(\frac{q}{A}\right)_b + \frac{A_{nc}}{A_{total}} \left(\frac{q}{A}\right)_{nc} \quad (2.20)$$

$$\left(\frac{q}{A}\right)_b = \frac{K^2}{2} \sqrt{\pi k_l \rho_l C_{pl} f D_b^2 N (T_w - T_\infty)} \quad (2.21)$$

For laminar range, $10^5 < Ra < 2 \times 10^7$

$$\left(\frac{q}{A}\right)_{nc} = 0.54 \rho_l C_{pl} \left[\frac{\gamma g (T_w - T_\infty)^5 \alpha^3}{\sqrt{A_{total} \nu}} \right]^{1/4} \quad (2.22)$$

For turbulent range, $2 \times 10^5 < Ra < 3 \times 10^{10}$

$$\left(\frac{q}{A}\right)_{nc} = 0.14 \rho_l C_{pl} \left[\frac{\gamma g (T_w - T_\infty)^4 \alpha^2}{\nu} \right]^{1/3} \quad (2.23)$$

It should be noted that in Mikic and Rohsenow's model, no heat was assumed to be dissipated from the evaporation of fluid during the bubble growth process and the bubbles mainly helped in the removal of the thermal boundary layer to facilitate the next cycle of conduction heat transfer process. This concept is contrary to the report by Cooper and Lloyd [35] where significant heat transfer from the surface was observed during the period when evaporation of the thin fluid layer (also known as

microlayer) took place between the base of the bubble and the surface. By assuming that the microlayer of the fluid was much thinner as compared to the radius of the bubble such that lubrication flow could be applied, Cooper and Lloyd [35] simplified the momentum equation in the horizontal direction to a one-dimensional transient problem and determined that the initial thickness of the microlayer, δ , was related to the growth time of the vapour bubble, t , as shown in Eq. (2.24). Subsequently, Judd and Hwang [36] included the contribution of heat transfer effect of microlayer evaporation into Mikic and Rohsenow's model by adding an additional third term to the right hand side of Eq. (2.20). This term is shown in Eq. (2.25) where subscript e denotes the mode of heat transfer related to microlayer evaporation and V represents the volume of microlayer associated with each bubble growth.

$$\delta = 0.8\sqrt{\nu t} \quad (2.24)$$

$$\left(\frac{q''}{A}\right)_e = VN\rho_l h_{fg} f \quad (2.25)$$

Reviewing Eqs. (2.20) to (2.25), it can be seen that apart from the data for fluid thermophysical properties and wall superheat, three key parameters were required to be quantified for the mechanistic model pool boiling model to be complete. These parameters are the bubble departure diameter, D_b , the bubble departure frequency, f , and the active nucleation site density, N .

Bubble departure diameter

A mechanistic correlation for bubble departure diameter was proposed by Fritz [26] as shown in Eq. (2.14). This model was developed by balancing the buoyancy force required to lift the vapour bubble from the surface and the opposing surface tension

force which prevented the vapour bubble from departing and the coefficient, C_d , was approximated to be 0.0208. Subsequently, Staniszewski [37] and Cole [38] modified Eq. (2.14) by including an additional term which took into account the dynamic effect of bubble growth. More recently, Cole and Rohsenow [39] fitted the range of available data for a variety of fluids at lower pressure and saturated boiling conditions and obtained the correlation as shown in Eq. (2.26) where Ja^* is the modified Jakob number given by Eq. (2.27). The coefficient C_d was determined to be 1.5×10^{-4} for water and 4.65×10^{-4} for other fluids. The difference between Fritz's model and Cole and Rohsenow's model is that the former is largely applicable for atmospheric pressure and at lower heat fluxes whereas the latter accounts for the fluid thermophysical properties and hence is valid for different fluids.

$$D_b = C_d \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} (Ja^*)^{5/4} \quad (2.26)$$

$$Ja^* = \frac{\rho_l C_{pl} T_{sat}}{\rho_g h_{fg}} \quad (2.27)$$

Bubble departure frequency

Conceptually, the bubble departure frequency is inversely proportional to the total waiting and growth time of the bubble as shown in Eq. (2.28). Jakob [40] first observed this constant periodic cycle and diameter of vapour bubbles and developed an expression for the vapour flow velocity as shown in Eq. (2.29). Using dimensional analysis, McFadden and Grassmann [41] further extended a relationship between bubble departure frequency, f , and bubble departure diameter, D_b , to include the thermophysical properties of the fluid. Cole [42] then modified McFadden and Grassmann's relationship and included the effect of Jakob's number as shown in Eq.

(2.30) where m and r are constants to be determined. Using approximately 400 data points of different fluids and pressures, Cole [42] performed regression analysis and determined that the best fit line was obtained with r and m values of -2 and -3/4, respectively, hence reducing the expression to the final form as shown in Eq. (2.31).

$$f = \frac{1}{T_w + T_g} \quad (2.28)$$

$$f D_b = \text{constant} \quad (2.29)$$

$$\frac{f D_b^2 \rho_l^{1/2}}{(\sigma D_b)^{1/2}} = \left[\frac{g(\rho_l - \rho_g) D_b^2}{\sigma} \right]^m \left[\frac{\rho_l C_{pl}(T_w - T_{sat})}{\rho_v h_{fg}} \right]^r \quad (2.30)$$

$$f D_b = 0.59 \left[\frac{g(\rho_l - \rho_g) \sigma}{\rho_l^2} \right]^{1/4} \quad (2.31)$$

It is observed that the left hand side of Eq. (2.31) is essentially comprised of constant terms which are only dependent on the thermophysical properties of the bulk fluid and gravitational acceleration and is invariant with heat flux, wall superheat and pressure. Therefore, if information on D_b is known, f can similarly be determined. Over the past four decades, Eq. (2.31) has been employed in numerous pool boiling investigations with relative success. However, as this model did not account for other mechanisms such as microlayer evaporation, bubble interaction and cavity size, it was mainly limited to low heat flux application or within the isolated bubble regime.

Active nucleation site densities

Models of active nucleation site densities, N , are mainly empirical-based correlations. Gaertner and Westwater [43], for instance, found that N was directly proportional to q'' as shown in Eq. (2.32). Cornwell and Brown [44], on the other hand, investigated

copper surface in water and determined N to be a function of wall superheat as shown in Eq. (2.33).

$$N \sim (q'')^{2.1} \quad (2.32)$$

$$N \sim (T_w - T_{sat})^{4.5} \quad (2.33)$$

Mikic and Rohsenow [33] first proposed an expression which relates N with cavity sizes present on commercial available surfaces as shown in Eq. (2.34). D_s represents the size of the largest cavity present on the surface and D_c is the minimum cavity size for bubble growth derived by Griffith and Wallis [19] as shown in Eq. (2.5). In addition, from the experimental data obtained, m was determined to be 6.5.

$$N \sim \left(\frac{D_s}{D_c} \right)^m \quad (2.34)$$

Subsequently, Wang and Dhir [45] investigated the effect of surface wettability on N with water on copper surface. It was determined that for fluid-surface contact angles between 18° and 90° , N could be expressed as shown in Eq. (2.35) for which D_c is required to be smaller than $5.8 \mu\text{m}$ for the expression to be valid.

$$N_A = 5.0 \times 10^5 (1 - \cos\phi)(D_c) \quad (2.35)$$

2.2.3 Critical heat flux models

Hydrodynamic CHF model

Zuber [46] developed a model capable of directly predicting the CHF of pool boiling without the knowledge of nucleate boiling conditions which preceded it. This model assumed that CHF was achieved when the vertical vapour streams escaping the boiling

surface and the horizontal vapour-liquid layer hovering over the boiling surface became hydrodynamically unstable. In addition, in their hydrodynamically unstable state, the vertical vapour streams were assumed to conform to the Taylor critical wavelength, λ_C , whereas the horizontal vapour-liquid layer was assumed to obey the Helmholtz critical wavelength, λ_H . As the magnitude of Helmholtz wavelength was not defined, the critical Rayleigh unstable wavelength for gas jet in liquid, which gives $\lambda_H = \pi\lambda_C/2$, was adopted. Finally, through geometrical considerations, the ratio of area of the vapour stream to the total area of the heated surface was derived to be $\pi/16$ and the total heat transfer from the escaping vapour stream during CHF for an infinite flat plate was expressed as Eq. (2.38).

$$\lambda_C = \frac{2\pi}{\sqrt{\frac{g(\rho_l - \rho_g)}{\sigma}}} \quad (2.36)$$

$$U_g = \sqrt{\frac{2\pi\sigma}{\rho_l\lambda_H}} \quad (2.37)$$

$$q''_{CHF} = \frac{\pi}{16} \rho_v h_{fg} U_v \quad (2.38)$$

Substituting Eqs. (2.36) and (2.37) into Eq. (2.38), the relationship for CHF was obtained as shown in Eq. (2.39). It was found that the constant, C_1 , took the value of 1.197 under critical wavelength condition and had the value of 0.9094 during dangerous wavelength condition. Hence, C_1 was often assumed to be 1.

$$q''_{CHF} = C_1 \frac{\pi}{24} \rho_v h_{fg} [\sigma g(\rho_l - \rho_v)]^{1/4} \quad (2.39)$$

Subsequently, Sun and Leinhard [47] extended the hydrodynamic theory to include the geometrical effects of the boiling surface making the theory applicable to horizontal cylinders. In addition, Dhira and Leinhard [48] also examined the influence of fluid viscosity on CHF and showed that the fluid must be very viscous in order to affect the Taylor process.

Similar to the nucleate pool boiling models presented in the previous section, the hydrodynamic instability model for CHF is directly influenced by the thermophysical properties of the bulk fluid. On the contrary, the CHF model was developed solely on the basis of the vapour-fluid hydrodynamics and has no clear dependence on surface effect. The model has been successfully applied in numerous pool boiling conditions in the past five decades. However, the drawback of not considering the effect of heater surface characteristics was also pointed out by several investigators. Costello and Frea [49], for instance, conducted a series of experiments with different fluids and determined that the model was inconsistent in predicting the CHF with fluids of different wettabilities. Subsequently, Liaw and Dhira [28] also verified that CHF improved with increasing surface wettability.

Macrolayer dryout CHF model

Katto and Yokoya [50] proposed the occurrence of CHF as a result of the consumption of liquid macrolayer which existed immediately adjacent to the heated surface. As shown in Figure 2.3, under high heat fluxes, vertical streams of vapour leaving the heated surface were formed due to the continuous and vigorous departure of bubbles from the active nucleation sites under high heat fluxes. These vapour stems would then contribute to the growth of the vapour mushroom directly above them. Subsequently, the resupply of fluid to these nucleation sites was assumed to be from the streams of fluid adjacent to these vapour stems, also known as macrolayer. However, it was postulated that the resupply of fluid would only be possible when the vapour mushroom was detached from the macrolayer. Hence, if the hovering time of the vapour mushroom (time taken from vapour mushroom inception to departure) was longer than the time taken to evaporate the entire macrolayer, CHF would occur.

Fig. 2.3 Conceptual model of macrolayer dryout theory [50]

Haramura and Katto [51] applied mass conservation between the vapour steam and fluid resupply in the macrolayer Eq. (2.40) and energy balance at the heated surface Eq. (2.41), to obtain the velocity components of the vapour stem and fluid resupply.

Using Eqs. (2.40) and (2.41), the critical wavelength for Helmholtz instability in relation to the ratio of vapour area and heated surface area, could be defined as shown in Eq. (2.42). In addition, for a flat heated surface, it was assumed that the macrolayer thickness was equal to a quarter of the Helmholtz wavelength, resulting in the final expression for CHF as shown in Eq. (2.43).

$$u_v = \left[\frac{q}{\rho_g H_{fg}} \right] / \left[\frac{A_v}{A_w} \right] \quad (2.40)$$

$$\frac{u_l}{u_g} = \left(\frac{\rho_l}{\rho_g} \right) \left(\frac{A_v}{A_w} \right) \left[1 - \frac{A_v}{A_w} \right] \quad (2.41)$$

$$\lambda_H = 2\pi\sigma \left(\frac{\rho_l + \rho_g}{\rho_l \rho_g} \right) \left(\frac{A_v}{A_w} \right)^2 \left(\frac{\rho_g h_{fg}}{q} \right)^2 \quad (2.42)$$

$$q''_{CHF} = 0.72 \left(\frac{A_v}{A_w} \right)^{\frac{5}{8}} \left(1 - \frac{A_v}{A_w} \right)^{\frac{5}{16}} \left[\frac{\left(\frac{\rho_l}{\rho_g} + 1 \right)}{\left(\frac{11\rho_l}{16\rho_g} + 1 \right)^{3/5}} \right]^{\frac{5}{16}} \left\{ \rho_g h_{fg} [\sigma(\rho_l - \rho_g)]^{1/4} \right\} \quad (2.43)$$

As shown through the derivation of this model, CHF is directly related to the thickness of macrolayer, which is in turn related to the vapour to heater surface area ratio. As the A_v is the area of vapour stream where boiling takes place, the magnitude of A_v also indirectly indicates the number of nucleation sites. For instance, when $A_v \ll A_w$, the larger the ratio, the thicker will be the macrolayer and hence a greater CHF could be achieved.

Hydrodynamic and surface effect CHF model

Kandlikar [52] proposed that CHF was caused by the uncontrolled lateral expansion of the bubble which eventually blanketed the heater surface. Hence, CHF was assumed to take place when the force due to momentum change pulling the bubble interface into the fluid along the lateral direction overcomes the total opposite forces holding it. These opposing forces include the surface tension force on the vapour-fluid interface at the top of the bubble, the surface tension force on the vapour-fluid-solid interface at the bubble base and the force due to gravity acting on the bubble in the lateral direction parallel to the heater surface. In addition, by considering that the bubble diameter has half the wavelength of the Taylor instability of the vapour film as developed by Zuber [46], the following expression for CHF was obtained:

$$q''_{CHF} = h_{fg} \rho_v^{1/2} \left(\frac{1 + \cos\theta}{16} \right) \left(\frac{2}{\pi} + \frac{\pi}{4} (1 + \cos\theta) \cos\phi \right)^{1/2} [\sigma(\rho_l - \rho_v)]^{1/4} \quad (2.44)$$

As shown in Eq. (2.44), this model includes the hydrodynamic aspect of CHF and also the non-hydrodynamic properties such as heater surface orientation, θ , and the dynamic receding contact angle of the fluid, ϕ . For the surface orientation between 0° and 90° and fluids (water, refrigerants and cryogenic liquids) of different contact angles (14° to 107°), it was validated that the model provided an accurate prediction where the mean error ranged from 24.3% to 0.5%. More recently, Ahn et al. [53] experimented on micro/nanostructures coated surfaces and observed that CHF was significantly enhanced when the contact angle was measured to be near 0° . From the direct measurement of the fluid absorption rate, it was determined that the micro/nanostructures coating induced the capillary wicking effect which promoted in the resupply of fluid to the dryout region and delaying CHF. Taking into account the

capillary wicking effect, Kandlikar's model was subsequently modified by adding an additional term as shown in Eq. (2.45) to the right hand side of Eq. (2.31) where q''_{gain} is the critical heat flux gained due to liquid spreading, A_{heater} is the area of the heating surface, ε and δ_c is the porosity and the thickness of the coating respectively, A_{wetted} is the area wetted by capillary wicking and t indicates the duration of wetting by the capillary wicking.

$$q''_{gain} = \frac{\varepsilon \delta_c \rho_l h_g K}{A_{heater}} \frac{dA_{wetted}}{dt} \quad (2.45)$$

2.2.4 Summary on pool boiling models

In summary, the developments in pool boiling theories have achieved significant success to a large extent. Taking the modelling of CHF for instance, since the pioneering work by Zuber [46] which laid the foundation to approach this boiling crisis from the hydrodynamic perspective, subsequent researchers such as Sun and Leinhard [47], Dhir and Lienhard [48], Katto and Yokoya [49] and others had extended the concept to include the effects of macrolayer thickness, nucleation site density, wettability, heated surface geometries and other parameters which allowed the model to be applicable to different boiling conditions. In the isolated bubble and developed boiling regime, heat transfer models with different signatory mechanisms essential to predict different boiling conditions were also conceptualised over the last few decades. Models such as those that rely on the active nucleation site densities and microlayer evaporation developed by Wang and Dhir [45] and Cooper and Lloyd [35], respectively were accurate in predicting heat transfer in the isolated bubble boiling regime. The models were later improved by Zhao et al. [54] and Das et al. [55] to

include the developed bubble regime. All these models and others that have not been mentioned in this review provided the tools which can be adequately utilised to perform analyses under a wide variety of pool boiling conditions.

2.3 Advances in micro/nanostructure surfaces for pool boiling enhancement

Artificial modification to the heat transfer surfaces with the aim of augmenting pool boiling heat transfer have been extensively studied in the last three decades. Between mid-1980s and late-1990s, investigations were mainly focused on enhancing pool boiling heat transfer with microstructure surfaces produced from etching, laser-drilling and spraying techniques. The types of microstructures produced from these manufacturing techniques include microcavities [56], microfins [57], and microporous [58] surfaces. From the early-2000s, propelled by the advancement in MEMS techniques and nanotechnology at the turn of the millennium, modified surface employed in pool boiling studies saw a significant increase in nanoscale structure surfaces. These nanostructures include nanowires [8], nanoparticle deposition [7], nanoporous [9] and carbon nanotube [12] coated surfaces. This section attempt to classify and provide a brief review of the pool boiling performance on the heaps of micro/nanostructure surfaces investigations reported and special attention is given to the various proposed transport mechanisms for these micro/nanostructures. In addition, as the present study involves the investigation of surface orientation effects with CNT coated surface, recent studies and models developed for micro/nanostructure surfaces with different surface orientations are also reviewed and evaluated.

2.3.1 Classification of micro/nanostructures and their pool boiling performances

Microcavities

Several techniques were used to create artificial micro sized cavities with the aim of increasing heated surface's roughness and simulating nucleation sites for enhancing pool boiling heat transfer. Hwang and Moran [59], for instance, produced laser-drilled cavities of between 3 and 15 μm on silicon surfaces and tested these in dielectric fluid. Mudawar and Anderson [60], alternatively, created structured cavity arrays of 0.356 mm diameter with mechanical drilling and also produced random distribution of cavities of pore diameter 15 μm with sand and vapour blasting. Among the numerous techniques, adopting microelectronic photolithography processes in fabricating microcavities have also shown to significantly enhance pool boiling heat transfer. These fabrication processes often involved etching, chemical deposition and UV exposure to create array of highly structured micro-reentrant cavities on silicon chip as the heat transfer surface. Comprehensive studies with these micro-reentrant cavities enhanced surfaces in FC-72 dielectric fluid were reported by numerous investigators such as Nowell et al. [61], Kubo et al. [56], Baldwin et al. [62], Yu et al. [63] and Nimkar et al. [64] where the effects of parameters such as cavity mouth diameter, cavity depth, cavity density and cavity separation on promoting nucleate boiling were critically examined.

Kubo et al. [56] fabricated microcavity arrays on the boiling surfaces and the cavity mouth diameters varied between 3.1 and 1.6 μm and cavity density ranging from 980 and 81 cm^{-2} . It was determined that surface with the largest cavity size and density displayed the highest heat transfer enhancement. Yu et al. [63], on the other hand,

investigated on surfaces with microcavities of cavity mouth diameter ranging from 50 to 200 μm and cavity depths between 110 and 200 μm . For the cavity sizes of this range, it was validated that heat transfer performance was the highest with the smallest cavity mouth diameter of 50 μm and shallowest cavity depth of 110 μm . It was explained that the larger cavity sizes led to increased bubble diameter which caused early coalescence of adjacent bubbles whereas the deeper cavities were seen reduced bubble departure frequency and resisted the resupply of fluid which impeded the heat transfer performance. The effect on microcavity separation was studied by Nimkar et al. [64] on surfaces with pyramidal micro-reentrant cavity array of similar cavity mouth diameter but with different cavity separation tested in FC-72. Using the dimensionless separation distance, S/D_b , where S is the cavity separation and D_b is the bubble departure diameter, it was determined that the surface with the highest heat transfer performance has the S/D_b values between 1.70 and 2.50. Zhang and Shoji [65] identified the mechanisms of nucleation site interactions in relation to S/D_b . The hydrodynamic interaction between bubbles and, the horizontal and declining bubble coalescences were seen to promote bubble growth and bubble departure frequency whereas strong effect of thermal interaction between nucleation sites would cause nucleation sites to be inhibited and lower bubble departure frequency. These mechanisms coexisted in the nucleation site interactions however the dominance of individual mechanism was shown to depend on the values of S/D_b .

Fig. 2.4 Microcavities - (a) Kubo et al. [56] (b) Yu et al. [63]

Microfins

Studies on pool boiling heat transfer with finned surfaces began as early as the 1970s. However, as MEMS technique and nanotechnology were still premature during the early years, the finned structures fabricated were generally in the millimetre range. Investigators such as Klein and Westwater [66] and Guglielmini et al. [67] conducted pool boiling experiments with millimetre fin arrays by varying the fin lengths and fin widths where noticeable heat transfer enhancement over plain surface was observed. In addition, Rainey and You [68] also examined the heat transfer performance of millimetre fin array by varying the fin height at 1 mm, 2 mm, 4 mm and 8 mm for each array. The pool boiling results showed that the array with the greatest fin height showed the most pronounced enhancement in heat transfer in the natural convection regime and the heat transfer also increased with increasing fin height in the nucleate boiling regime. Using one-dimensional steady state conduction analysis and the empirically developed pool boiling heat transfer correlations for fins, Yeh [69] developed a model to obtain an optimal fin height and fin separation for pool boiling heat transfer as a function of the Biot number Bi , fin base and fin tip temperature. From

the model, it was determined that increasing the fin aspect ratio would cause a decrease in heat dissipated by the fins but increase heat transfer from the interfin spacing.

Microscale fins were first experimental investigated by Honda et al. [70] where the fabricated fins were in order of 10 μm . Dry etching technique was used and the fins (50 $\mu\text{m} \times 50 \mu\text{m} \times 60 \mu\text{m}$) produced were uniformly distributed with separation distance of 100 μm . In addition, a separate surface was prepared with the similar microfin array but with an additional layer of 3 μm SiO_2 deposited over the microfin array by sputtering technique to increase the porosity of the surface. These two surfaces were tested in FC-72 and results showed that, unlike the millimetre fins reported by Rainey and You [68], the microfins have similar heat transfer characteristics as plain surface in natural convection regime. However, the surface with both microfin array and SiO_2 coating demonstrated significant enhancement in heat transfer and CHF was augmented by up to 2.3 times as compared to bare silicon. In addition, microfin array also displayed a distinctively steep boiling curve where a very small change in wall superheat was observed with every increment in heat flux. Subsequently, Wei and Honda [71] explained that the steep boiling curve was due to the highly structured and uniformly distributed microfins which resulted in identical surface roughness. Hence, the condition for the activation of the nucleation sites would occur at almost the same wall superheat. This explanation agreed well with experimental observation as large numbers of bubbles were observed over a small range of wall superheat.

Fig 2.5 Microfins – (a) Honda et al. [70] (b) Wei et al. [71]

Micro/Nanoporous surfaces

Fabrication of micro and nanoporous surfaces often involved the use of electrolysis or electrodeposition techniques where metallic molecules or nanoparticles are deposited onto the substrate surface with the use of electrical current. Due to chemical reactions in the deposition process, hydrogen gas bubbles are generated as by-products and departed from the coating surface which resulted in the formation of porous structures. Depending on the magnitude of the applied electrical current, selection of electrolyte solution and duration for the electrolysis process, the cavity sizes and coating thickness of the porous structure could be varied. Li et al. [72] and Furberg and Palm [73], for instance allowed the formation of quasi-static hydrogen bubbles by controlling the electrolysis process fabricated porous surface with larger interconnected macro-pores of 105 μm and smaller dendritic structures of 0.1 to 1 μm . On the other hand, El-Genk and Ali [74, 75], produced microporous coatings of variable thickness, ranging from 95.1 to 230.3 μm , by altering the deposition duration between 15 and 44 s. Recently, aluminium surfaces with nanosize structured pores of diameter between 30 to 70 nm were fabricated by Lee et al. [76, 77]. Such small pore sizes were achieved with electropolishing and anodising processes where mixture of electrolyte such as

perchloric acid, ethanol and sulphuric acid were used. Zhang et al. [78] developed a technique to alter the nanoporous surface wettability by immersing the specimen into 1-dodecanethiol ethanol solution.

With FC-72 fluid, Furberg and Palm [73] found that the porous surface enhanced boiling heat transfer by up to 5 times as compared to plain surface. El-Genk and Ali [75], on the other hand, verified that over the range of microporous thickness (from 95.1 to 230.3 μm) tested, the coating thickness of approximately 171.1 μm exhibited the most optimal heat transfer performance. Subsequently, Zhang et al. [78] reported heat transfer augmentation of as much as 300% for hydrophobic nanoporous and 200% for hydrophilic nanoporous surface as compared to plain surface. The onset of nucleate boiling for hydrophobic and hydrophilic nanoporous surfaces were also recorded at the wall superheats of 2 K and 3.5 K, respectively, a significant reduction of wall superheat as compared to the plain surface where the boiling incipience was recorded at 7.5 K.

From high speed visualisation, Furberg and Palm [73] measured the bubble diameter and found it to be 10 to 30% smaller with greater bubble departure frequency which led to the heat transfer enhancement and it was postulated that this was the result of early bubble coalescence caused by the high nucleation site density of the porous structure. A smaller bubble departure diameter as compared to plain surface was similarly observed by Zhang et al. [78] for the hydrophilic nanoporous surface. However, for the hydrophobic surface the reverse was observed. It was hence suggested that the bubble departure size was also affected by the surface tension force which altered the magnitude of bubble spreading and coalescing of the adjacent

bubbles. In addition, the porous structure was also seen to increase the total surface area, which allowed evaporation to take place within the pores whereas the interconnected network within the structure served as effective channels for the vapour escape hence augmenting the overall heat transfer performance of the surface.

Despite the remarkable heat transfer capability of micro/nanoporous structures, the main issue when employing these surfaces was the instability that pool boiling performances yield. El-Genk and Ali [75], for instance, reported the largest heat transfer coefficient with 197.4 and 230.3 μm thick microporous surface at the first boiling test, although subsequent tests showed substantial degradation in heat transfer coefficient. In addition, repeatability in the pool boiling results was only obtained at the third and fourth tests. Lee et al. [77], in contrast, observed continuous degradation in heat transfer performance of the microporous surface with repeating boiling tests and the boiling results were not repeatable even after five runs of the experiments. With SEM, it was revealed that the cavities of the porous layer were filled and smoothened as a result of chemical reaction between the surface and the bulk fluid during the boiling process.

Fig. 2.6 Microporous – (a) Furberg and Palm [73] (b) El-Genk and Ali [75]

In the NTU Thermal and Fluids Laboratory, investigations have also been made in enhancing nucleate boiling heat transfer with porous graphite structures. Leong et al. [101], for instance, explored the heat transfer performance of graphite foams for high heat flux applications with a uniquely designed compact thermosyphon. Graphite foams (Kfoam and Pocofoam) of different porosities (75-78%) and pore diameter (0.31 to 0.5 mm) were tested in FC-72 and it was determined that the all graphite foams significantly reduced the boiling thermal resistance by about 2 times as compared to copper. In addition, high speed imaging also revealed that the graphite foam with larger pore diameter promoted bubble generation and detachment hence resulted in higher heat transfer performance. Subsequently, Pranoto et al. [102] extended the study to include both block and finned graphite foams where the surfaces were tested in both FC-72 and HFE-7000 dielectric fluids. For the same fin-to-block surface area ratio, the block graphite foams consistently outperformed the finned structures in the nucleation boiling heat transfer as the block graphite contained more nucleation sites which resulted in larger bubble densities. In addition, for the same surface, the study also showed the FC-72 was a better heat transfer coolant as compared to HFE-7000.

Nanowires

To date, two types of nanowires were studied in pool boiling experiments; Si nanowires and Cu nanowires. The Si nanowires were synthesised using wafer scale electroless etching technique where the Si wafer was immersed in a solution of HF acid with AgNO_3 . As a result of redox reaction, areas of the surface not covered by Ag particles were oxidised and were subsequently etched by the HF acid leading to the formation of Si nanowire arrays ranging between 20 and 300nm in diameter and 40 to

50 μm in height. The sizes and densities of the nanowires largely depend on the concentration of the solution and etching time. For Cu nanowires, membranes with aluminium nanopores were used as template for the deposition of Cu particles. The membrane was initially bonded to the silicon substrate and electrochemical deposition technique was used to deposit the Cu into the nanopores. Subsequently, the aluminium template was removed by etching in NaOH solution resulting in array of Cu nanowires of approximately 200 nm in diameter and 5 to 122 μm in height.

The nucleate pool boiling performance of these nanowires was separately investigated by Chen et al. [79], Yao et al. [80] and Lu et al. [81]. Chen et al. [79] experimented with deionised water at saturated temperature and atmospheric pressure conditions and it was determined that the heat transfer coefficient and CHF of both Si and Cu nanowires were more than twice that of plain Si surface. It was revealed through SEM that bundles of nanowires were formed during the fabrication process which resulted in microcavities (1 to 6 μm) between the bundles. These microcavities sizes were suggested to be deal sites for bubble formation. In addition, static contact angle measurement with water also showed that the nanowires coating produced superhydrophilic surface with near 0° contact angle and suggested that this was due to the small characteristic diameter of the nanowire which provided large capillary pumping force to wet the surface. During boiling, this capillary pumping mechanism assisted in the resupply of liquid to the dryout region and delayed the occurrence of CHF. From the experimental results, Yao et al. [80] showed that for nanowires of heights between 2 μm and 35 μm , the heat transfer coefficient increased with increasing nanowires height and Lu et al. [81] made the same observation for nanowire

heights between 16 and 122 μm . Hence, varying the height of the nanowires had insignificant effect on the enhancement in CHF.

Fig. 2.7 Silicon nanowires – (a) Lu et al. [81] (b) Chen et al. [79]

CNT array

Motivated by the exceptional heat transfer properties, mechanical properties and porous nature of CNT, the possibility of augmentation nucleate boiling heat transfer with CNT coated surfaces have received significant attention in the recent years. The outline of studies reported on CNT coated surfaces is presented in Section 1.1.2 and the detailed boiling results with various type of CNT array coated surfaces and the proposed heat transfer mechanisms are elaborated in this section.

To-date, two groups of researchers had investigated the pool boiling heat transfer effect of CNT coated surfaces that were fabricated using CVD techniques. Ujereh et al. [12] coated the surface with MWCNTs with PECVD technique whereby the MWCNTs had approximately 50 nm diameter and 20 to 30 μm height. MWCNT arrays of different CNT densities and coating patterns as tabulated in Table 2.1 were produced. The CNT densities were controlled by varying the duration of submerging the substrate into dendrimer catalyst solution before undergoing PECVD for CNT

growth. In addition, the patterned CNTs were prepared by covering the substrate surfaces where CNT coating were not required with a layer of tape before exposing the substrate into the catalyst. After undergoing the PECVD process, only surfaces that were exposed to the catalyst would be coated with CNT.

Table 2.1 Types of CNT surfaces fabricated by Ujereh et al. [12]

Sample	Description	CNT Diameter (nm)	CNT Height (μm)	CNT Density (CNTs/ μm^2)
Light array	Surface fully coated with CNT	50	20-30	30
Dense array	Surface fully coated with CNT	50	20-30	60
Grid pattern	Illustrated in Figure 2.14(a). The spacing between CNT grid were derived based on Taylor instability wavelength and had approximately 60% of the total surface area covered with CNTs	50	20-30	20
Island pattern	Illustrated in Figure 2.14(b). The surface was sparsely coated with CNT clusters and approximately 5% of the total surface covered with CNTs.	50	20-30	20

Figure 2.8 (a) Grid patterned CNT coated surface (b) Island patterned CNT coated surface [12]

The surfaces were tested in FC-72 under saturated temperature and atmospheric pressure conditions. As compared to the light array, the dense array showed better heat transfer performance whereby the wall superheat for boiling incipience were observed to be 2°C lower and heat transfer coefficient was relatively higher. It was suggested that the increased enhancement was due to the higher nucleation site density in the dense array. However, the CHF for dense array was lower than the light array by approximately 22% and it was explained that the less densely packed light array allowed better accessibility by the fluid to the dryout areas and which delayed CHF. Patterned CNT surfaces have also displayed significant heat transfer enhancement over non-coated surface. For instance, the grid patterned surface demonstrated as high as 285% heat transfer enhancement as compared to bare silicon surface despite having only 60% CNT area coverage and island patterned showed approximately 38% heat transfer improvement with only 5% CNT coating. It was also suggested that the enhancement was due to the formation of cavities as a result of the highly meshed CNT structure and the voids that were present between CNTs. These cavities were ideal nucleation sites for low contact angle fluids such as FC-72. In addition, it was also noted that despite having only 60% CNT area coverage, the heat transfer performance of grid patterned surface was similar to the fully coated light array with slightly reduced CHF. No explanation were provided to account for this observation, however, it could be deduced that there exists an optimal coating area for which heat transfer could be optimised.

Ahn et al. [13, 82] was the second group of researchers who investigated with CNT coated surfaces where the effects of the variation in CNT heights and degree of

subcooling were explored. In their studies, vertically aligned CNT arrays with heights of 9 μm and 25 μm were synthesised with CVD technique without adopting the plasma enhancement subprocess. The experiments were conducted between 0° and 30° subcooled temperatures. Due to the atomically smoothed substrate surface used, much lower heat fluxes were reported. For instance, the maximum CHF values of 4.35 W/cm^2 and 4.46 W/cm^2 were observed for 9 μm and 25 μm CNT, respectively. Variation in CNT heights were shown to have insignificant effect on CHF and heat transfer coefficient and the maximum enhancement in CHF and heat transfer coefficient of 28% and 64.6%, respectively, as compared to bare silicon surface was recorded. For the improved CHF, it was postulated that the hair-like structure of CNTs protruded into the superheated liquid layer underneath the vapour mushroom which enhanced the formation of these vapour stems and delayed CHF.

Comparing the experimental results reported by Ujereh et al. [12] and Ahn et al. [13, 82], it was apparent that significant differences in the magnitude of heat transfer enhancement existed even though the same method was used to produce the CNT array. For this observation, it was explained that the discrepancy was due to the dendrimer-based catalyst used by Ujereh et al. [12] which introduced an additional degree of surface roughness.

2.3.2 Effects of surface orientation

Studies on surface orientation with plain surfaces

Nucleate pool boiling at different surface orientation with dielectric fluids (HFE-7100) was investigated by El-Genk and Bostanci [83] on smooth copper surface. At surface orientations between 0° and 90° , the nucleate boiling heat flux increased with

increased inclination angle when the wall superheat was lower than 20 K and decreased with increased inclination angle when the wall superheat was higher than 20 K. This was because under the former condition, increasing surface orientation increased boundary layer mixing by the departing bubbles and promoted heat transfer whereas under the latter condition, the large amount of bubbles generated formed a vapour layer which increased in accumulation and bubble residence time with increasing surface orientation and impeded heat transfer. For surface orientation larger than 90° , the vapour layer accumulation effect initiated at the wall superheat of 13 K which resulted reduction in heat transfer performance with increasing surface orientation. Priarone [84] subsequently conducted a comprehensive comparison on saturated nucleate pool boiling between FC-72 and HFE-7100 from 0° to 175° on copper surface. It was concluded that for plain copper surface, the nucleate heat transfer coefficient and CHF of HFE-7100 were higher than those of FC-72 by 25% and 40%, respectively under the same operating condition and this was primarily due to the superior thermal properties (such as latent heat, specific heat and thermal conductivity) of HFE-7100 as compared to FC-72. However, due to the higher boiling point of HFE-7100 (61°C) as compared to FC-72 (56°C), it was also observed that the wall superheat was relatively higher at CHF.

Studies on surface orientation with coated surfaces

Jung et al. [85] first investigated the combined effect of surface coating and surface orientation on nucleate and film boiling with R-11. The coated surfaces were produced using a spray coating technique to deposit metallic particles on plain mild steel plate. As a result of increased surface roughness, the coated surfaces showed 2 to 3 times

enhancement in heat transfer performance at constant heat flux as compared to plain copper surface. In addition, for both coated and uncoated surfaces, the wall superheats reduced with increasing surface orientation at low to moderate heat fluxes (10-40 kW/m²). However at higher heat fluxes, wall superheats were insensitive to surface orientation. It was proposed that two mechanisms were associated with the nucleate boiling for inclined surfaces: evaporative mechanism and bubble agitation mechanism. Evaporative mechanism refers to the conventional nucleate boiling mechanisms where heat is transferred from the surface through the formation and departure of bubbles such as the transient conduction model presented in Section 2.2.2 and is independent of surface orientation. Bubble agitation mechanism, on the other hand, is a strong function of surface orientation and is caused by the sweeping of the inclined surface by the departing bubbles which generated turbulence in the surrounding fluid, resulting in enhanced convection. From the experimental results, it was suggested that the bubble agitation mechanism dominated at low to moderate heat fluxes whereas evaporative mechanism dominated at high heat fluxes.

Chang and You [86] studied the effect of orientation on microporous surfaces produced by coating copper and aluminium particles of sizes between 1 and 50 μm with epoxy binder. Their experimental results with FC-72 similarly showed the reduction in CHF with increasing surface orientation. However, the nucleate boiling curves for microporous surface were almost insensitive to surface orientation. Rainey and You [87] subsequently coated diamond particles of sizes between 8 and 12 μm on 4 cm² and 25 cm² surfaces to determine the combined effect of heater size, orientation and surface enhancement on nucleate boiling. It was also demonstrated that the coated surfaces were unaffected by surface orientation. In addition, due to the large number

of active nucleation sites, the coated surfaces were also found to be insensitive to the variation in heater size.

Parker and El-Genk [88] carried out a detailed and systematic investigation of porous graphite at different orientations with FC-72 and HFE-7100 under saturated and 10 K, 20 K and 30 K subcooled boiling conditions. As opposed to Priarone [84], their experiments indicated that the nucleate boiling heat transfer coefficient of FC-72 on porous graphite was higher than those obtained from HFE-7100 and the values increased with increasing liquid subcooling. It was suggested that the higher heat transfer coefficient of FC-72 was due to the higher vapour pressure and lower surface tension of the fluid as compared to HFE-7100 which decrease the size of the bubble diameter and increased the departure frequency. This increase in bubble departure frequency in turn induced boundary layer mixing and enhanced heat transfer. However, at high heat fluxes, this mechanism was negated by the increased lateral coalescence of the bubbles. Lastly, for every boiling curve it was observed that there exists a point of maximum heat transfer coefficient. At wall superheats below the point of maximum heat transfer coefficient, the heat transfer performance would increase with increasing surface orientation whereas at wall superheats above this point, increasing surface orientation would result in a reduction in heat transfer performance.

CHF models with surface orientation effect

A gradual reduction in CHF as surface orientation increased from 0° to 90° followed by steeper decline in CHF as the orientation angles increased from 90° to 180° is a phenomenon observed in most pool boiling investigations reported. In general, two approaches were adopted in developing the CHF models incorporating the effects of

surface orientation: (1) mechanistic approach and (2) direct correlating experimental data with the hydrodynamic CHF model as shown in Eq. (2.39).

CHF models developed using the mechanistic approach includes that proposed by Kandlikar [52] as shown in Eq. (2.44) which took into consideration of the effect of momentum flux at the bubble base. This CHF model was validated to be accurate for surface orientation from 0° to 90° and fluid-surface contact angle between 14° and 107° . However, for pool boiling with dielectric fluids such as FC-72 which has near 0° contact angle with most surfaces and at surface orientation more than 90° , the square root on the right hand side of Eq. (2.44) would lead to an imaginary CHF value and hence, limit the application of this model. Howard and Mudawar [89], on the other hand, developed a mechanistic model capable of providing accurate CHF prediction for boiling surfaces between 60° to 165° inclinations. The model suggested that CHF was initiated when the momentum of the generated vapour overcame the interface pressure force that preserved the contact between the boiling surface and the wetting front. An iterative scheme was required to obtain a prediction of CHF at a particular heater orientation where the CHF value was first estimated and the various parameters such as the local wave velocity, vapour layer thickness and critical wavelength were required for the iteration process.

A more commonly adopted approach in developing the CHF model with surface orientation effect was to directly correlate the hydrodynamic CHF model [Eq. (2.39)] with experimental data. With the use of linear or non-linear regression method, a variable term as a function of surface orientation was often introduced to constant term of Eq. (2.39). Vishnev [90] and Chang and You [86], for instance, incorporated the

orientation effect by correlating Eq. (2.39) with the experimental data obtained for helium and water at different surface orientations and obtained the relationship as shown in Eqs. (2.46) and (2.47), respectively. Priarone [84] recently also developed a correlation for the CHF of FC-72 and HFE-7100 on copper surface as shown in Eq. (2.48). El-Genk and Bostanci [83, 91] then further developed the CHF correlation to include the combined effect of subcooling and orientation for HFE-7100 on copper surface as shown in Eq. (2.49).

$$\frac{q_{CHF,\theta}}{q_{CHF,0^\circ}} = \frac{(190 - \theta)^{0.5}}{190^{0.5}} \quad (2.46)$$

$$\frac{q_{CHF,\theta}}{q_{CHF,0^\circ}} = 1.0 - 0.000120\theta \tan(0.414\theta) - 0.122\sin(0.318\theta) \quad (2.47)$$

$$\frac{q_{CHF,\theta}}{q_{CHF,0^\circ}} = 1 - 0.001117\theta + (7.79401 \times 10^{-6})\theta^2 - (1.37678 \times 10^{-7})\theta^3 \quad (2.48)$$

$$\frac{q_{CHF,\theta}}{q_{CHF,0^\circ}} = C(\theta) \times R(\theta, \Delta T_{sub}) \quad (2.49a)$$

$$C(\theta) = [(0.229 - 7.79401 \times 10^{-6}\theta)^{-6} + (0.577 - 2.98 \times 10^{-3}\theta)^{-6}]^{(-1/6)} \quad (2.49b)$$

$$R(\theta, \Delta T_{sub}) = (1 + (0.016 + 1.05 \times 10^{-4}\theta + 3.986 \times 10^{-7}\theta^2)) \Delta T_{sub} \quad (2.49c)$$

2.3.3 Summary

Structured modified surfaces employed in pool boiling research were geared towards nanoscale from microscale in the recent years. This paradigm shift was partially driven

by the advancement in nanotechnology and MEMS technique but more importantly, investigations with nanostructured surfaces such as nanowires by Chen et al. [79] and CNT array by Ujereh et al. [12] had also demonstrated unprecedented heat transfer performance in pool boiling. At the same time, however, several inconsistent results were reported by different researchers. On the effect of height of nanostructured surfaces, for instance, Yao et al. [80] who investigated on pool boiling with nanowire heights ranging from 2 μm to 35 μm observed increasing enhancement with increasing nanowire heights and attributed this effect to the increase in cavity with increasing nanowire height. On the other hand, Chen et al. [79] who performed pool boiling with nanowire heights from 16 μm to 122 μm reported insignificant effect on heat transfer. In addition, Ahn et al. [13] who conducted pool boiling with CNT heights within the similar range (9 μm and 25 μm) as Yao et al.'s [80] nanowire also showed that the effect of CNT heights were negligible on heat transfer performance. Another inconsistency was pertaining to the discrepancy between heat transfer enhancements reported by Ujereh et al. [12] and those reported by Ahn et al. [13]. Even though Ahn et al. [13] explained that the high enhancement reported by Ujereh et al. [12] was due to the substrate dendrimer catalyst, the difference in the heat transfer enhancement between Ujereh et al. [12] and Ahn et al. [13] was as high as 375% and if this significant difference in heat transfer enhancement was solely due to the substrate surface characteristics and preparation process, it would be imperative to determine their influence on pool boiling with CNT coated surfaces. In addition, CNT heights that had been investigated were of the order of 10 μm and those within the order of 100 μm had not been explored and apart from Ujereh et al. [12] who conducted experiments with patterned CNT coatings of 60% and 5% CNT area coverage,

systematic investigation on the effect of CNT coating area have also not been performed.

On the investigation of surface orientations, increasing attention has been focused on nucleate boiling with dielectric fluids and enhanced surfaces in the recent years. However, inconsistency on the pool boiling results was also observed. Chang and You [86] and Rainey and You [87], for instance demonstrated that diamond particle coated surfaces were insensitive to surface orientation whereas Parker and El-Genk [88] who conducted pool boiling on porous graphite reported a reduction in heat transfer coefficient with increasing surface orientation at high heat fluxes. In addition, the investigation of surface orientation effects with CNT coated surfaces has not been performed and correlations with these surfaces have also not been reported.

Due to high packing density of electronic devices, circuit boards and heat sinks were often required to be positioned at different orientations in order to overcome space constraints. Hence, the application of CNT array coating for immersion cooling of electronic devices requires comprehensive understanding of their intrinsic transport mechanisms and the effects of surface orientation for the optimisation their heat transfer capacity. Amid the inconsistent and incomplete pool boiling data with CNT surfaces highlighted above, motivation for further research with these surfaces remains pertinent.

CHAPTER 3

EXPERIMENTAL SETUP AND PROCEDURES

3.1 Experimental facilities

In order to perform the studies outlined in Section 1.2, two water-cooled thermosyphons (Thermosyphon 1 and Thermosyphon 2) were designed and fabricated. The main difference between the two setups is that in Thermosyphon 1, the test specimens are adhered onto a fix heating block at the upright position whereas in Thermosyphon 2, a rotational assembly is employed to allow the specimens to be tested at different orientations. The experimental facilities used in the pool boiling investigation for Thermosyphon 1 and Thermosyphon 2 are shown schematically in Figs. 3.1 and 3.2, respectively. Both the setups are comprised of (1) a fluid chamber, (2) a water-cooled condenser system and (3) a heater section where the test specimens are attached. The fluid chamber is made of an aluminium body of external dimensions 125 (L) $\times$ 125 (W) $\times$ 120 (H) mm and a Teflon base as support. Two viewing panels made of acrylic are installed along the wall of the aluminium body to allow visual observation and recording of the boiling process. The water cooled condenser system is comprised of a Polyscience model 8206 circulating water bath and a copper cooling coil located at the top of the fluid chamber. The circulating water bath provides constant supply of cold water at approximately $22.5^{\circ}\text{C} \pm 2^{\circ}\text{C}$ to the cooling coil which in turn cools the FC-72 vapour and maintains the fluid level constant throughout the experiment. Two cartridge heaters are immersed into the FC-72 and are used to bring the fluid to its saturation temperature. The cartridge heaters are positioned at the back of the fluid chamber and at a distance from the test specimen to prevent any interaction with the fluid motion around the test specimen.

Thermosyphon 1 heater section

Details of the heater section of Thermosyphon 1 are shown in Fig. 3.3. A copper block consisting of $40 \times 40 \times 10$ mm square base and $10 \times 10 \times 30$ mm rectangular shaft as shown in Fig. 3.3a is used as the heat conductor. A blind hole is drilled at the centre of the copper block such that the Watlow E1E61-L12 rod heater (31.75 mm in length and 6.35 mm in diameter) is tightly fitted within. The rod heater is used as the heating element where the DC power supplied to the rod heater can be varied using the power controller and the level of power input can be monitored with the Carlo Gavazzi MW14-96 power meter. In order to minimise heat loss, the entire copper block and rod heater are embedded within the Teflon insulation with the exception of the top copper surface where heat is dissipated, as shown in Fig. 3.3b. To investigate with the various CNT coated surfaces and bare silicon, the specimen to be tested is bonded onto the top copper surface using a highly conductive thermal adhesive (Omegabond 202). This heater section assembly allows the heat generated from the rod heater to be conducted through the copper block and dissipated to the bulk fluid at the specimen surface. Lastly, three K-type thermocouples are positioned along the side of the copper block at 2 mm, 20 mm and 40 mm from the top of the copper block for Thermocouples 1, 2 and 3, respectively, as shown in Fig 3.3b.

Thermosyphon 2 heater section

Details of the heater section of Thermosyphon 2 are shown in Fig. 3.4. This assembly consists of a $10 \times 10 \times 30$ mm rectangular copper block which is tightly fitted within the Teflon insulation assembly ($50 \times 50 \times 64$ mm). The Teflon insulation is subdivided into three separate pieces: a $50 \times 50 \times 10$ mm top piece, a $50 \times 50 \times 25$ mm centre

piece and a $50 \times 50 \times 25$ mm bottom piece. These three Teflon pieces are assembled with 22 pieces of screws along the parameters and the 2 mm thick gaskets are positioned between the Teflon interfaces to prevent any seepage of fluid into the Teflon insulation. In addition, high temperature resistance sealant was also applied along the Teflon interfaces to prevent any seepage of fluid. A Watlow E1E61-L12 rod heater (31.75 mm in length and 6.35 mm in diameter) embedded within the copper block is used as the heating element where the DC power supplied to the rod heater is varied with the power controller and the level of power input is monitored with the Carlo Gavazzi MW14-96. The copper block and the rod heater are surrounded by the Teflon insulation at all sides except at the top copper surface. The specimen to be tested is bonded onto the copper top surface with Omegabond 202 such that the heat generated by the rod heater is conducted through the copper block and dissipated to the surrounding fluid by the test specimen. In order for the test specimen to be rotated to different orientations for the investigation of orientation effect on nucleate pool boiling, the assembly comprising of the rod heater, copper block, Teflon insulation and test specimen is tightly secured to an aluminium hollow shaft. A 20 mm-diameter through-hole is drilled along the wall of the fluid chamber for the aluminium shaft to be inserted and a lip seal to be seated firmly between the through-hole and shaft to prevent any fluid seepage while permitting the shaft to be rotated from the external of the fluid chamber. In order to determine the nucleate performance of the test specimens, three K-type thermocouples are positioned at the top copper surface (or just beneath the test specimen), 10 mm from the top copper surface (or at

Fig. 3.1 Thermosyphon 1 setup and facilities

Fig. 3.2 Thermosyphon 2 setup and facilities

Fig. 3.3 Heat section of thermosyphon 1 (a) copper block (b) thermocouple location

Fig. 3.4 Heat section of Thermosyphon 2 (a) copper block (b) thermocouple location

the interface between the top and centre Teflon pieces) and 27 mm from the top copper surface (or at the interface between the centre and bottom Teflon pieces) for Thermocouples 1, 2 and 3 respectively, as shown in Fig. 3.4b. The hollow section of the aluminium shaft allowed these thermocouples to be channelled out of the thermosyphon where the ends are connected to the Yokogawa MW100 data acquisition unit. All thermocouple signals are transmitted to the data acquisition unit which are in turn displayed and recorded on the central processing unit. Lastly, a high speed camera (FASTCAM 1024 PCI) is employed for visual analysis of the nucleate boiling process for different surfaces and different heat fluxes.

In summary, Thermosyphon 1 was implemented for the following experiments and the surfaces tested are schematically presented in Fig 3.5:

1. Saturated pool boiling with bare silicon, 90% CNT coated surface, 60% CNT coated surface, 30% CNT coated surface and interlaced 60%-CNT coated surface.

On the other hand, Thermosyphon 2 was used for the following experiments and the surfaces tested are shown schematically in Fig. 3.6:

1. Saturated pool boiling with bare silicon at surface orientations of 0° , 30° , 60° , 90° , 120° , 150° and 180°
2. Saturated pool boiling with 100% CNT coated surface at surface orientations of 0° , 30° , 60° , 90° , 120° , 150° and 180°
3. Saturated pool boiling with interlaced 70%-CNT coated surface at surface orientations of 0° , 60° and 90°

Fig. 3.5 Schematics of (a) 90% CNT coated surface (b) 60% CNT coated surface (c) 30% CNT coated surface (d) Interlaced 60%-CNT coated surface

Fig. 3.6 Schematics of (a) 100% CNT coated surface (b) Interlaced 70%-CNT coated surface

3.2 Experimental procedures and data reduction

An air pressure test was performed on the fluid chamber prior to the experiments to ensure that the chamber was tightly sealed. This was achieved by filling the chamber with compressed air up to 0.5 bar gauge pressure and the pressure was observed to maintain for 10 minutes. To perform the experiments, the chamber was first filled with FC-72. For Thermosyphon 1, the fluid level was maintained at 80 mm above the test specimen whereas for Thermosyphon 2 the rotation assembly was fully immersed in the fluid and the test specimen was at least 40 mm beneath the fluid level when the rotation assembly was in the upward facing position (0°). For both setups, the FC-72 was then heated to its saturation temperature by setting the power input to the cartridge heaters to approximately 180 W depending on the boiling conditions required. The fluid was maintained at the respective temperature for minimum 30 minutes to allow degassing and to ensure that the steady state condition was attained. The electrical power to the rod heater was then raised at increments of between 0.7 and 1.7 W and the system was allowed to stabilise for 10 minutes for every increment at lower heat flux and 15 to 20 minutes near CHF. The steady state condition was reached when the thermocouple reading was within $\pm 0.5^\circ\text{C}$ for more than 2 minutes. The occurrence of CHF was identified by the unstable increase in wall temperature after which the power supply to the rod heater was immediately switched off. For both setups, experiments were conducted at least twice under the same condition to verify the repeatability of the results. Throughout the experiments, the system was maintained at atmospheric pressure. All thermocouples were calibrated prior to the experiments and the

temperatures were acquired under quasi-steady state condition with the Yokogawa MW100 data acquisition unit at the sampling rate of 5 Hz averaged over 1 minute.

In both thermosyphon setups, the wall temperatures (T_w) were obtained from Thermocouple 1 (or T1 as shown in Figs. 3.3 and 3.4). The thickness of the bare silicon was measured to be 0.5 mm and the layer of epoxy adhesive between the top copper surface and bare silicon was estimated to be 0.05 mm thick. Therefore, the position of Thermocouple 1 in Thermosyphon 1 was determined to be 2.55 mm beneath the boiling surface whereas the position of Thermocouple 1 in Thermosyphon 2 was located at 0.55 mm beneath the boiling surface. The temperature of FC-72 (T_{sat}) was evaluated under atmospheric condition and the heat rate generated from the rod heater (q) was recorded directly from the power meter. Based on the measured q , T_w and T_{sat} , the nucleate boiling heat transfer coefficient (h) was calculated using Eq. (3.1) as follows:

$$h = \frac{q''}{(T_w - T_{sat})} \quad (3.1)$$

The uncertainty of the thermocouple measurements was $\pm 0.5^\circ\text{C}$ while the uncertainty of the current and voltage readings from the power meter were 0.01 A and 0.1 V, respectively. Using the method described by Moffat [92], the average uncertainties of the heat flux (q'') and the heat transfer coefficient (h) are calculated from Eqs. (3.2) and (3.3), respectively. In the single-phase convection regime, the uncertainty of h was calculated to be $\pm 14.3\%$ whereas in the nucleate boiling regime, it was evaluated to be $\pm 5.6\%$.

$$\delta q''/q'' = \sqrt{(\delta V/V)^2 + (\delta I/I)^2} \quad (3.2)$$

$$(\delta h/h)^2 = \sqrt{(\delta T_w/T_w)^2 + (\delta T_{sat}/T_{sat})^2 + (\delta q''/q'')^2} \quad (3.3)$$

3.3 Thermal analysis of heater section

It is not possible to determine the temperature of the top boiling surface (or wall temperature) by directly measuring it as the presence of the thermocouple would hinder and disturb the boiling process. Hence, the wall temperature was represented by the measurements taken from Thermocouple 1 for both setups. In order to verify the accuracy of employing Thermocouple 1 as the wall temperature, a one-dimensional (1-D) thermal resistance analysis was performed on the heater sections in this section. In addition, the heat rate dissipated from the test specimen was taken to be equal to the heat rate generated by the rod heater as the heat loss through the Teflon insulation was assumed to be negligible. This assumption similarly requires verification. In this section, a two-dimensional (2-D) axisymmetric thermal simulation on the heater sections was also conducted to estimate the amount of heat losses.

3.3.1 1-D thermal resistance analysis

A 1-D thermal resistance analysis was conducted vertically from the position of Thermocouple 1 to the top silicon surface to determine the temperature difference between the two positions. Using 1-D Fourier's law and assuming energy to be conserved, the temperature change in the vertical direction with respect to the vertical distance and thermal conductivity can be represented as shown in Eq. (4.4a), where subscripts *c*, *o* and *si* denote copper block, Omegabond 202 thermal adhesive and silicon respectively. In addition, by assuming temperature continuity [Eqs. (4.4b) and (4.4c)], the thermal resistance from the Thermocouple 1 position to the top silicon surface as shown in Eq. (4.4d) was derived.

$$-k_c \frac{T_{c,2} - T_{c,1}}{y_c} = -k_o \frac{T_{o,2} - T_{o,1}}{y_o} = -k_{si} \frac{T_{si,2} - T_{si,1}}{y_{si}} \quad (4.4a)$$

$$T_{c,1} = T_{o,2} \quad (4.4b)$$

$$T_{o,1} = T_{si,2} \quad (4.4c)$$

$$R = \frac{T_{c,2} - T_{si,1}}{q''} = \frac{y_c}{k_c} + \frac{y_o}{k_o} + \frac{y_{si}}{k_{si}} \quad (4.4d)$$

The thermal conductivity values of the copper block, Omegabond 202 thermal adhesive and silicon were taken to be 400 W/m·K, 16.61 W/m·K and 149 W/m·K respectively and the values of y_c , y_o and y_{si} are given in Section 3.1. Using Eq. (4.4d), the thermal resistance, R , was calculated to be $1.137 \times 10^{-5} \text{ m}^2\cdot\text{K}/\text{W}$ and $0.637 \times 10^{-5} \text{ m}^2\cdot\text{K}/\text{W}$ for Thermosyphons 1 and 2, respectively. At the peak heat flux of $20 \text{ W}/\text{cm}^2$, for instance, this corresponds to the maximum temperature difference of 2.3°C for Thermosyphon 1 and 1.3°C for Thermosyphon 2. At this heat flux, the measured temperature was observed to be approximately 100°C to 110°C and hence the temperature difference was between 1% and 2% of the measured temperature. Due to the small temperature difference, the temperature measured by Thermocouple 1 could be considered as the wall temperature.

3.3.2 2-D axisymmetric thermal simulation

A two-dimensional, steady-state, axisymmetric thermal analysis was performed on both the heater sections using the ‘COMSOL Multiphysics’ software to determine the heat loss of the setup. In the simulation, the physical geometry and thermophysical properties of the copper block and Teflon insulation were used. The heat generated

from the rod heater was assumed to be evenly distributed along the internal surface of the copper block and the nucleate heat transfer coefficient was taken to be uniform along the boiling surface. In addition, heat was also assumed to be dissipated from the boundary of the Teflon insulation to the surrounding fluid and ambient air through natural convection. The simulation was conducted at various heat fluxes with the FC-72 temperature of 56°C and ambient air temperature of 24°C. The energy equation of Eq. (4.5) was deployed and was further simplified with the assumptions, interface conditions and boundary conditions detailed in Appendix A for Thermosyphon 1 heater section and Appendix B for Thermosyphon 2 heater section.

$$0 = \alpha \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right] \quad (4.5)$$

The simulation result for Thermosyphon 1 heater section on bare silicon with the steady-state temperature distribution at heat flux of 15 W/cm² is shown in Fig. 3.7. Based on the temperature contour, it can be observed that for the copper block temperature variation is significant in the y -direction whereas there is negligible temperature dependence in the x -direction. Moreover, the temperature variation is mainly localised along the shaft of the copper block whereas the base of the copper block is generally at uniform temperature. As the copper block is insulated on all sides except at the top surface, the majority of the heat generated from the rod heat can only be dissipated on the top silicon surface to the surrounding FC-72 fluid. In addition, due to the larger surface area of the copper block, it is also observed that the majority of the heat loss is through the copper base.

The simulation result for Thermosyphon 2 heater section on bare silicon at the heat flux of 10 W/cm² is shown in Fig. 3.8. Similarly, temperature variation in the copper

block is observed in the y-direction where heat is dissipated from the silicon top surface. Unlike the heater section in Thermosyphon 1, the copper block in this setup does not have the square base which significantly reduced the amount of heat losses. Comparing Figs. 3.7 and 3.8, it is noticed that the temperature range along both copper blocks are similar even though the heat fluxes are substantially different. This is because the temperature profile of the copper block is largely dependent on the heat transfer performance of the top silicon surface. It can be noticed in the subsequent section that two different batches of silicon surfaces are used for Thermosyphons 1 and 2 and the heat transfer coefficients of the silicon for the two setups are different. This accounts for the similar temperature profiles observed and should not be mistaken as simulation error.

Lastly, based on the temperature distribution along the copper block and the Teflon insulation boundary, the maximum heat loss was estimated to be approximately 4.0% for Thermosyphon 1 heater section and 1.5% for Thermosyphon 2 heater section.

Fig 3.7 Simulated steady-state temperature distribution of Thermosyphon 1 heater section at heat flux of 15 W/cm^2 .

Fig 3.8 Simulated steady-state temperature distribution of Thermosyphon 2 heater section at heat flux of 10 W/cm^2 .

3.4 Preparation of CNT coated surfaces

All CNT coated surfaces were prepared using chemical vapour deposition (CVD) technique where CNTs were directly synthesised on the silicon surface resulting in vertically aligned CNT structures. The detailed surface preparation procedures are depicted by Sun et al. [93] and are briefly summarised in this section.

The single-sided polished Si wafer was deposited with a layer of Fe using electron-beam deposition technique. The Fe layer was 0.8 nm thick and served as catalyst for the CNT growth. Vertically aligned MWCNTs were then grown on the Si substrate in a CVD system which was carried out in a 12.4 mm quartz tube furnace using pure acetylene (C_2H_2) as the carbon source with argon (Ar) and hydrogen (H_2) as both the forming and carrier gas. The required CNT coating density, thickness and area coverages could be obtained by varying the total gas flow rate, source and carrier gas mixture, growth temperature and growth duration.

For the purpose of the present investigation, two batches of CNT coated surfaces were separately prepared. The first batch was produced by synthesising CNTs on 8×8 mm Si substrate of 0.5 mm thickness. The root mean square roughness of the silicon substrate surface was determined to be 450 nm with the use of confocal image profiler as shown in Fig. 3.9a. The maximum CNT coverage of approximately 90% (Fig. 3.9b) was achieved with an average CNT diameter of 10 nm. As shown in Fig. 3.9c, SEM imaging also revealed that the CNT structures produced were vertically aligned with one end rooted to the Si substrate. The thickness of the CNT for this batch was measured to vary between 200 μm and 230 μm and the average thickness was approximately 215 μm . In addition, due to the high aspect ratio of the CNTs,

significant entanglement and meshing between CNTs were also observed as depicted in Fig. 3.9d. From 90% uniformly coated CNT surfaces, a razor blade was subsequently used to remove excessive CNT coating to produced 60% and 30% uniformly coated CNT surfaces as well as the interlaced 60%-CNT coated surface as shown in Fig. 3.5. The pool boiling performances of all these surfaces were experimental investigated in Thermosyphon 1.

Fig. 3.9 (a) Confocal surface roughness profile of batch 1 silicon substrate surface (b) Photograph of 90% uniformly coated CNT surface (c) SEM side view of CNT structure at 200 μm scale (d) SEM top view of CNT structure at 1 μm scale.

The second batch was produced by synthesising CNTs on 10×10 mm Si substrate of 0.5 mm thickness. The root mean square roughness of the silicon substrate was determined to be approximately 15 nm with the use of confocal image profiler as shown in Fig. 3.10a. In comparison, the silicon surfaces used in this batch were much smoother than those in Batch 1 and this was primarily due to the difference in surface

polishing method adopted. For this batch, a 100% CNT coverage (Fig. 3.10b) over the silicon surface was achieved and the averaged CNT diameter was similarly around 10 nm. However, unlike the CNTs produced in Batch 1, insignificant variation in the CNT coating thickness was observed from this batch. CNT with near uniform thickness of approximately 215 μm as shown in Fig. 3.10c was measured where the same intertwining CNT structures (Fig. 3.10d) as those shown in Batch 1 was observed. Lastly, the razor blade was also used to remove excessive CNT coating from the 100% CNT coating to produce the interlaced 70%-CNT coated surface as shown in Fig. 3.6. The pool boiling performances of these surfaces were experimental investigated in Thermosyphon 2.

Fig. 3.10 (a) Confocal surface roughness profile of batch 2 silicon substrate surface (b) Photograph of 100% uniformly coated CNT surface (c) SEM side view of CNT structure at 200 μm scale (d) SEM top view of CNT structure at 100 nm scale.

CHAPTER 4

EFFECTS OF CNT COATING ON SATURATED POOL BOILING HEAT TRANSFER

This section presents and analytically evaluates the saturated pool boiling results of FC-72 on uniformly coated CNT surfaces with 30%, 60% and 90% of CNT area coverages and the interlaced 60%-CNT coated surface. The experimental setup with Thermosyphon 1 (as depicted in Section 3.1) was employed for this investigation and all surfaces tested were obtained from the first batch of bare silicon and fabricated CNT coated surfaces (as illustrated in Section 3.5). All experiments were conducted under atmospheric pressure.

4.1 Consistency and repeatability assessment

Bare silicon served as the control surface to assess the heat transfer performance of CNT coated surfaces. Repeated boiling tests were conducted for bare silicon and the boiling curves are presented in Fig 4.1. Both the boiling curves are almost identical and the differences in their wall superheats are within the limits of experimental uncertainty, indicating that the results with bare silicon were highly reproducible. The heat transfer characteristic of the bare silicon was evaluated by averaging the temperatures of the repeated tests and the curve is plotted in Fig. 4.2. The occurrence of boiling incipience was observed at the heat flux of 5 W/cm^2 and the wall superheat of 12.2°C . In the nucleate boiling regime, the gradient of the curve was relatively steeper than that in the natural convection regime which signified the higher heat transfer. This trend continued until CHF was achieved at 16 W/cm^2 with the wall superheat of 42.5°C .

Fig. 4.1 Saturated pool boiling curves of FC-72 on bare silicon.

Investigations on the saturated pool boiling performances of plain surfaces such as bare silicon and plain copper in FC-72 were also reported by Honda et al. [70], Ujereh et al. [12], Ahn et al. [13], Priarone [84] and Parker and El-Genk [94]. These results are plotted in Fig 4.2 which serve as comparisons against the present study. Significant differences in the pool boiling results among different researchers were observed even for similar surfaces. As heat transfer in boiling was largely influenced by the presence of surface cavities, the disparities observed could be due to the differences in the surface preparation techniques which resulted in different surface roughness. For instance, Honda et al.'s [70] bare silicon has a surface roughness of 2.3 nm whereas in the present study, Batch 1 bare silicon has a roughness of 450 nm. As shown in Fig 4.2, this increase in surface roughness of the bare silicon used in the present study could have resulted in better heat transfer performance as compared to Honda et al.'s [70] bare silicon between low to intermediate heat fluxes. On the other hand, comparing between bare silicon and plain copper surface, a common trend was

observed whereby bare silicon demonstrated much higher boiling incipience temperature, higher wall superheats and lower CHF. This phenomenon was similarly shown in the present study. Based on Eq. (2.5), the minimum cavity radius for boiling incipience is related to the fluid thermophysical properties and also the wall superheat. Due to the lower surface tension, σ , and latent heat of vaporisation, h_{fg} , of FC-72 as compared to water, the minimum cavity size required for incipience ($4.41 \times 10^{-6} \text{ m} \cdot \text{K}$ for FC-72 and $2.51 \times 10^{-8} \text{ m} \cdot \text{K}$ for water) is much larger. However, the bare silicon has a smoother surface than plain copper and there are limited sizeable cavities to initiate boiling at low temperatures. Hence, boiling incipience was observed only at higher superheats where smaller cavities could be activated [based on Eq. (2.5)]. In addition, the smoother bare silicon surface also contained fewer active nucleation cavities which resulted in poorer heat transfer performance and higher wall superheats as compared to plain copper.

Fig. 4.2 Comparison of saturated pool boiling curves of FC-72 on bare silicon and plain copper surfaces.

Similar to bare silicon, experiments with the 90% uniformly coated CNT surface were also repeated to verify the reproducibility of the boiling curves. Figure 4.3 presents the first four boiling experiments conducted with the newly prepared CNT surface. Results of the initial test showed lower CHF and higher wall temperature although, noticeable increment in CHF and reduction in wall temperature were observed with subsequent tests. After two boiling tests with the CNT surface immersed in FC-72 for more than 10 hours, results of successive tests, represented by Tests 3 and 4 in Fig. 4.3, were found to be consistent and within the limits of experimental uncertainty. These results demonstrate that there is some degree of CNT ageing effect in pool boiling application where the aged CNT surface possessed improved heat transfer performance as compared to new CNT surface. The effect of ageing with enhanced surfaces was also reported by Parker and El-Genk [94] and Lee et al. [77] who investigated porous graphite and porous aluminium, respectively. Parker and El-Genk [94] suggested that the observed ageing was caused by the deposition of the plasticizers from the epoxy used in the test assembly which leached into the bulk fluid, altering the fluid surface tension, and hence affected the boiling results. On the other hand, Lee et al. [77] mentioned that the ageing effect was caused by the chemical reaction which took place between the aluminium surface and the bulk fluid. The chemical reaction resulted in the smoothening of porous surface after repeated boiling tests. However, ageing effect with CNT surfaces was not reported by Ujereh et al. [12] and Ahn et al. [13] who similarly experimented with vertically aligned CNT surfaces prepared using the CVD coating technique but unlike the present study where CNT coating thickness was in 200 μm range, the CNT thicknesses investigated by Ujereh et al. [12] and Ahn et al. [13] were only between 9 and 45 μm . Due to the large CNT coating thickness

employed in the present study, it was possible that the CNTs were initially not well wetted by surrounding fluid which explained the higher superheat and earlier CHF as shown in Test 1. Following continuous fluid immersion and successive boiling tests, optimal fluid wetting paths were established which allowed stable and maximum wetting of the CNTs. As a result, repeatable boiling curves shown as Tests 3 and 4 were obtained and higher heat transfer performance was recorded. Lastly, it was also noted that throughout the experiments, there was no visual degradation of the CNT coating. For the rest of the chapter, only the results for the aged CNT surface would be presented. For convenience, the aged CNT surface would be referred simply as CNT surface.

Fig. 4.3 Saturated pool boiling curves of FC-72 on 90% uniformly coated CNT surface

As shown in Fig. 4.4, the pool boiling curve obtained in the present study for the 90% uniformly coated CNT surface is distinctly different from those reported by Ujereh et al. [12] and Ahn et al. [13]. An obvious explanation for the disparities observed is the

difference in the CNT thickness employed in the present study (215 μm) and those used by Ujereh et al. [12] and Ahn et al. [13] which were only between 9 μm and 30 μm . However, it should be noted that even though Ujereh et al. and Ahn et al.'s CNTs were of similar thickness, significant difference in their pool boiling curves was also observed. An alternative cause for the disparities in the boiling curves could be due to the differences in the silicon substrates' surface characteristics which resulted from the different preparation procedures used. For instance, Ujereh et al. [12] coated a layer of dendrimer catalyst on the substrate prior to the deposition of CNT. The dendrimer layer might have introduced additional surface roughness which further augmented nucleate boiling. On the other hand, Ahn et al.'s [13] silicon substrates were atomically smoothed which have limited cavities to start with. The influence of the substrate surface characteristics on the boiling curve of CNT coated surfaces are discussed in the subsequent chapter.

Fig. 4.4 Comparison of saturated pool boiling curves of FC-72 CNT coated surfaces and bare silicon

4.2 Effects of CNT coating area

Figure 4.5 shows the comparison of the saturated pool boiling results of the 30%, 60% and 90% uniformly coated CNT surface and bare silicon obtained in the present study.

(a)

(b)

Fig. 4.5 Comparison of (a) saturated pool boiling curves (b) heat transfer coefficient for different CNT area coverages.

4.2.1 Effects on CHF

As shown in Fig. 4.5(a), the CNT coated surfaces display relatively higher CHF compared to bare silicon and the magnitude of CHF similarly increases with increased area coverage. For the 90% uniformly coated CNT surface, CHF was registered at 24 W/cm^2 with the wall superheat of 51.1°C which corresponds to 50% enhancement in CHF as compared to bare silicon. On the other hand, the surface coated with 30% CNTs recorded a CHF of 21 W/cm^2 at a wall temperature of 108.4°C while the 60% CNT coated surface achieved a CHF of 22 W/cm^2 at a wall temperature of 108.1°C . In comparison with bare silicon, approximately 30 to 50% enhancement in CHF was observed for CNT coated surfaces and the highest enhancement of 50% was achieved with the 90% CNT coated surface.

Carbon nanotubes are known to be hydrophobic in nature. As shown in Fig. 4.6, contact angle measurements conducted on the 90% CNT surface using the sessile drop method with deionised water confirmed the hydrophobicity of the CNT surface. The contact angle was measured to be 156° . Subsequently, the sessile drop test was also conducted on the same CNT surface using FC-72. As shown in Fig. 4.7, the images captured during the drop test at 0.2-s intervals revealed that as the FC-72 droplet came into contact with the CNT, it rapidly spread and seeped into the CNT layer. This liquid spreading phenomenon could likely be due to the high wettability of FC-72 and the enhanced capillarity of CNT coating resulting from highly meshed CNT structures. As a result of this liquid spreading effect, the fluid would then be able to rapidly reach and replenish the dryout region during boiling and hence delay the occurrence of CHF for CNT surfaces as compared to bare silicon surface.

Fig. 4.6 Sessile drop test with deionized water.

Fig. 4.7 Sessile drop test with FC-72.

Ahn et al. [53] highlighted the liquid spreading effect on micro/nano structured surfaces as a result of capillary wicking action and developed a model to determine the enhancement in CHF due to this phenomenon. By performing the time integral to the model developed by Ahn et al. [53] and rearranging the terms, the following expression could be obtained:

$$\frac{A_{wetted}}{A_{heater}} = \left(\frac{q''_{gain}}{\varepsilon \delta \rho_l h_g} \right) t \quad (5.1)$$

In Eq. (5.1), q''_{gain} is the critical heat flux gained due to liquid spreading, A_{heater} is the area of the heating surface, ε and δ is the porosity and the thickness of the coating respectively, A_{wetted} is the area wetted by capillary wicking and t indicates the duration of wetting by the capillary wicking. It could be observed from Fig. 3.9 that even though the CNTs were grown vertically on the substrate, intertwining and meshing among individual CNTs resulted in the possible formation of capillary network, for which capillary wicking action could be induced. For the 30%, 60% and 90% CNT surfaces tested, the porosity and thickness of their CNT structures were similar. In

addition, with the similar bubble ebullition and departure cycle experienced by the CNT surfaces, the time allowed for capillary wicking to replenish the dryout region can also be assumed to be identical for the CNT surfaces. Employing Eq. (5.1) and with the assumptions stated above, the enhancement in CHF (or q''_{gain}) for the CNT surfaces is, hence, only influenced by A_{wetted} . From the experimental results obtained, CHF was found to increase from 21 to 24 W/cm², as the CNT coating area was increased from 30 to 90%. This is because the increase in CNT coating area increased the number of sites for capillary wetting to take place which increased A_{wetted} , and enhanced CHF.

4.2.2 Effects on boiling incipience

For CNT coated surfaces, boiling incipience was observed at relatively lower heat fluxes and lower wall superheats as compared to bare silicon. For instance, boiling incipience was recorded at the heat flux of 4 W/cm² for all CNT coated surface, approximately 1 W/cm² lower than bare silicon. In addition, the incipience wall superheats were registered at 8.5°C, 8.7°C and 8°C for the 30%, 60% and 90% uniformly coated surfaces, respectively, whereas bare silicon's incipience wall superheat was as high as 12.2°C. Despite reducing the incipience wall superheats with CNT coating, there was, however, no clear dependence of CNT coverage area on the incipience wall superheat.

A model for predicting the range of cavity sizes for boiling incipience at saturated bulk fluid temperature was proposed by Hsu [29] as shown in Eq. (2.11) and rewritten as follows for convenience:

$$r_{c (min,max)} = \frac{\delta}{4} \left[1 \pm \sqrt{1 - \frac{12.8T_{sat}\sigma}{(T_w - T_{sat})\delta\rho_g h_{fg}}} \right] \quad (2.11)$$

The cavity sizes for incipience are related to the fluid properties of FC-72, wall superheats and the thermal boundary layer thickness. The fluid properties are obtained from the product information for FC-72 Fluorinert™ by 3M and are summarised in Appendix C whereas the wall superheats are determined experimentally. In order to estimate the range of cavity sizes, the thermal boundary layer thickness, δ , during incipience is the remaining term required to be determined. Based on the transient thermal boundary layer concept as described in Section 2.2, the layer of superheated liquid is required to initiate bubble growth. In addition, the successful growth and departure of bubble is required to take place within the boundary layer because growing out of the boundary layer would result in condensation and eventual collapse of bubble. On the basis of this assumption, the minimum thickness of the thermal boundary layer can be estimated to be equal to the height of the bubble at departure which is approximately $1.6r_b$ [29]. The average bubble departure radii, r_b , for bare silicon and CNT surface during incipience were estimated to be 0.264 mm and 0.173 mm, respectively (detailed derivation is shown in Appendix D). Hence, δ was taken to be 0.264 mm which allowed bubble growth to take place on both surfaces.

The nucleation cavity range for FC-72, with the calculated δ , corresponding to the wall superheat is plotted in Fig. 4.8. At the incipience superheat of 8°C as obtained experimentally for CNT coated surface, the cavities that could be activated were ranged between 0.45 μm and 250 μm whereas with the incipience superheat of 12.2°C for bare silicon, the activated cavities were between 0.29 μm and 250 μm . Comparing

the range of activated cavities sizes between bare silicon and CNT at incipience, there is relatively insignificant difference in their cavity sizes at the upper limit as the approximate size of $250\ \mu\text{m}$ was calculated for both surfaces. However, at the lower limit, the higher incipience superheat of bare silicon resulted in significantly smaller minimum cavity size (55% smaller) as compared to the CNT coated surface. This analysis indicated that the higher incipience superheat of bare silicon was primarily due to the presence of smaller surface cavities which could only initiate bubble growth at higher wall temperature. This postulation also agreed well with the surface measurement results, as shown in Fig. 3.9, where bare silicon has a smoother surface while the intertwining CNT structures created sizeable cavities to initiate boiling at lower temperature.

Fig. 4.8 Range of active cavities for FC-72.

4.2.3 Effects on nucleate boiling heat transfer

In the nucleate boiling regime, CNT coated surfaces resulted in lower wall superheats as compared to bare silicon and this is indicated by the leftward shift of the pool boiling

curve as shown in Fig. 4.5(a) which also corresponds to the improvement in heat transfer coefficient as shown in Fig. 4.5(b). Furthermore, the experimental results also demonstrate that the magnitude of enhancement in nucleate boiling heat transfer increases with increasing CNT area coverage. For instance, at the heat flux of 12 W/cm^2 , the heat transfer coefficient for bare silicon was calculated to be $0.43 \text{ W/cm}^2\cdot\text{K}$, whereas the 30%, 60% and 90% CNT coated surfaces have heat transfer coefficients of 0.46, 0.48 and $0.54 \text{ W/cm}^2\cdot\text{K}$, respectively. These values translate to 6%, 11%, and 27% enhancements in the heat transfer coefficients for the 30%, 60% and 90% CNT coated surfaces, respectively as compared to bare silicon. The maximum enhancement in heat transfer coefficient of 50% as compared to bare silicon was recorded with the 90% CNT coated surface at the heat flux of 15 W/cm^2 , whereas the highest heat transfer coefficient of 0.56 W/cm^2 was obtained with the 90% CNT coated surface at the heat flux of 18 W/cm^2 .

Figure 4.8 shows that as the wall superheat is increased beyond the incipience temperature, the range of cavities that could be activated correspondingly increases. From the SEM micrograph (Fig. 3.9), most of the cavities created by the CNT structures were found to be between 1 and $0.1 \mu\text{m}$ as represented by the shaded region in Fig. 4.8. This indicates that the cavities that were activated on the CNT coated surfaces are mainly submicron in size with the wall superheat increased beyond the incipience temperature. However, as bare silicon has very small overall cavity quantity and submicron sized cavities, this resulted in fewer active nucleation sites and thus, poorer heat transfer performance. The larger quantity of submicron nucleation sites on the CNT coated surfaces were likely to be created by the intertwining and meshing among CNTs which resulted in the formation of micro-pores and cavities. In addition,

with the increase in CNT coating areas, more active nucleation sites could be created and this was directly reflected in the experimental results where the heat transfer performance increased with increasing CNT coating areas.

The relative nucleate boiling heat transfer coefficient of CNT coated surfaces with respect to bare silicon ($h_{CNT}/h_{silicon}$) from 5 W/cm² to 16 W/cm² is plotted in Fig. 4.9(a). It was observed that for heat fluxes above 6 W/cm², the CNT coated surfaces consistently outperformed bare silicon. For heat fluxes above 10 W/cm², the prominent trend of increasing heat transfer coefficient enhancement with increasing CNT coating areas was also observed. However, it was also noticed that even though the CNT area coverage for these surfaces varied linearly from 30% to 60% to 90%, the increment in heat transfer enhancement was non-linear. This is clearly illustrated in Fig 4.9(b), where $h_{CNT}/h_{silicon}$ is re-plotted against the CNT area coverages where each curve corresponds to a particular heat flux. It is observed that for all heat fluxes from 10 W/cm² to 16 W/cm², the heat transfer enhancements increase gradually with the increase in CNT coating area from 0% (bare silicon) to 30%, followed by steeper increases from 30% to 60% and eventually an exponential increase from 60% to 90%. If the enhancement in heat transfer was solely due to the additional nucleation sites as a result of the increase in CNT top surface area, the enhancement observed should be proportional to the CNT area coverage. In this regard, the results were indicative of additional parameters influencing the exponential increase in heat transfer enhancement with increasing CNT coating area. This would be discussed in detail in Section 4.3.

(a)

(b)

Fig. 4.9 Relative nucleate boiling heat transfer coefficient of CNT coated surfaces with respect to bare silicon (a) against heat flux (b) against CNT area coverage.

4.3 Effects of CNT coating perimeter

Visual observations of the boiling experiments show that most of the bubbles were departing from the edges of the CNT (or the interface between CNT coating and silicon) between low (5 W/m^2) and intermediate (16 W/m^2) heat fluxes. Due to the

limitation of Thermosyphon 1 setup, clear images of the bubble nucleation phenomenon were not captured. However, similar phenomenon was also observed in the experiments conducted with bare silicon and CNT coated surfaces using Thermosyphon 2 where clear images were taken. These images are reproduced in Figs. 4.10 and 4.11 for the ease of explanation. Apart from the noticeable increase in bubbles generated for CNT coated surface as compared to bare silicon which confirmed the presence of more nucleation sites, the photographs also show that, for the CNT coated surface, most bubbles were formed and departed along the perimeter of the CNT coating at the heat flux of 6 W/cm^2 . Due to the overcrowding bubble growth along the CNT perimeter, early coalescence of bubbles localised along the CNT perimeter were also observed. Only with the further increase in heat flux were more bubbles formed on the coating surface.

Fig 4.10 Front view photographs of saturated pool boiling from the bare silicon surface at 90° inclination angle at (a) 6.1 W/cm^2 (b) 9.8 W/cm^2 (c) 11.8 W/cm^2 .

Fig 4.11 Front view photographs of saturated pool boiling from the CNT coated surface at 90° inclination angle at (a) 6.1 W/cm^2 (b) 9.8 W/cm^2 (c) 16.3 W/cm^2 .

The meshing of CNTs along the edges could have similarly resulted in the formation nucleation sites. In addition, as the CNT coating used in the present study has the thickness of $215\text{ }\mu\text{m}$, the thick CNT structure increases the possibility of the formation of sizeable cavities along the edge of the CNT. These cavities allowed vapour bubbles to grow not only on the top of the CNT coating but also along its perimeter, as illustrated in Fig 4.12. With the calculated range of active nucleation sites plotted in Fig. 4.8, the incipience of boiling along the CNT perimeter at lower heat flux and hence lower superheat is indicative of the presence of larger micro-sized cavities.

Fig 4.12 Illustration of vapour bubble formation along CNT perimeter and top surface.

Based on the above analysis, both the CNT coating area and perimeter have significant effect on nucleate boiling and hence, explained the exponential increase in heat transfer enhancement shown in Fig. 4.9(b) as the increase in coating area not only increases the active nucleation sites on the CNT top coating but also along its perimeter. The interlaced 60%-CNT coated surface was tested and the results are compared against the 90% and 60% CNT coated surfaces as shown in Fig. 4.13. The interlaced surface has a horizontal separation, D , between the CNT layers where the width of each CNT layer was approximately twice the separation, $2D$. With this coating configuration, the perimeter of the CNT coating for interlaced surface was increased by approximately 240% as compared to the 90% CNT coated surface. However, the total CNT coated area was reduced to approximately 60%. The pool

boiling results with interlaced surface revealed that between the lower and intermediate heat flux regimes, the wall superheats were lower than the 90% CNT surface. This result indicates that within these heat fluxes, the interlaced surface has better cooling performance as compared to the 90% CNT surface. For instance, approximately 15% enhancement in heat transfer was observed for the interlaced surface as compared to 90% CNT surface at the heat flux of 13 W/m^2 . However, as the heat flux was increased beyond 18 W/cm^2 , the enhancement effect due to the edges introduced by the interlaced surface diminished and deterioration in heat transfer as compared to 90% CNT surface was observed. On the other hand, even with similar coating area, the heat transfer performance of interlaced surface surpassed that of 60% CNT coated surface for the entire nucleate boiling regime and this was primarily due to the increase in coating perimeter of the interlaced surface.

Fig. 4.13 Comparison of (a) saturated pool boiling curves of 60%, 90% and interlaced-60% CNT area coverages.

Lastly, for the interlaced surface, lateral bubble growth along the edges of the CNT surface was also observed. As illustrated in Fig 4.14, the occurrence of lateral bubble growth at neighbouring nucleation sites also led to the coalescence of bubbles at low heat fluxes. This in turn induced mixing of thermal boundary layer and promoted heat transfer.

Fig. 4.14. Schematic of lateral bubble growth, coalescence and departure of interlaced CNT coated surface.

CHAPTER 5

COMBINED EFFECTS OF CNT COATING AND SURFACE ORIENTATION ON SATURATED POOL BOILING HEAT TRANSFER

This section presents and analytically evaluates the saturated pool boiling results of FC-72 on the uniformly coated CNT surface with 100% CNT coverage, the interlaced CNT coated surface with 70% CNT coverage and the bare silicon at surface orientations ranging from 0° to 180° . Unless otherwise specified, the experimental setup with Thermosyphon 2 (as depicted in Section 3.2) was employed for this investigation and all surfaces tested were obtained from the second batch of bare silicon and fabricated CNT coated surfaces (as illustrated in Section 3.6). All experiments were conducted under atmospheric pressure.

5.1 Consistency and repeatability assessment

The bare silicon similarly serves as control surface for comparison against the heat transfer performance of CNT and interlaced surfaces. The initial boiling tests for all surfaces were conducted at the surface orientation of 90° and experiments were repeated at least once for each surface. Similar to the first batch of bare silicon, results of the repeated pool boiling tests conducted with the present batch of bare silicon (second batch), as shown in Fig. 5.1(a), demonstrated highly reproducible boiling curves and the difference in their wall superheats were within the limits of experimental uncertainty. For the 100% uniformly coated CNT surface, a total of four sets of experiments were conducted, as shown in Fig. 5.1(b). In the first test, relatively higher wall superheats and lower CHF was recorded. As the CNT was subject to repeated tests, further increment in wall superheats and CHF was observed.

Eventually, reproducible boiling curves were obtained with Test 3 and 4, where the differences in their wall superheats are within the limits of experimental uncertainty. The effect of CNT ageing where the aged CNT possessed improved heat transfer performance was observed for the 90% uniformly coated CNT surface and was illustrated in Section 4.1. In the present study with 100% uniformly coated CNT surface, similarly CNT ageing trend was demonstrated which reaffirmed the hypothesis. Finally, it should also be noted that throughout the experiments, no visual degradation of CNT coating was observed.

Fig. 5.1 Saturated pool boiling curves at 90° surface orientation of (a) bare silicon (b) 100% CNT surface.

Figure 5.2 compares the nucleate pool boiling curves, at 0° surface orientation, obtained experimentally for the second batch of bare silicon and 100% uniformly coat CNT and the pool boiling curves of bare silicon and 90% uniformly coated CNT from the first batch (Fig. 4.5). As the bare silicon surfaces of both batches have distinctly different surface roughness as shown in Figs. 3.9(a) and 3.10(a), the comparison made in Fig. 5.2 allows the effects of silicon substrate surface roughness on the heat transfer performance of CNT coated surfaces to be examined.

Firstly, the difference in surface roughness between the two bare silicon surfaces is reflected in the significantly dissimilar pool boiling curves as shown in Fig 5.2. The boiling incipience for the second batch of bare silicon was recorded at the heat flux of 3.1 W/cm^2 and at the superheat of 17.7°C . As compared to results obtained from first batch of bare silicon (incipience heat flux of 5 W/cm^2 and superheat of 12.2°C), the incipience of boiling with the second batch of bare silicon occurred at relatively high superheat but lower heat flux. The higher incipience superheat of the second batch bare silicon was suggestive of smaller nucleation cavities as compared to the first batch of bare silicon. This was highly possible as confocal measurement revealed that the surface of the second batch bare silicon (15 nm) was much smoother than the first batch bare silicon (450 nm) and hence, should possess smaller surface cavities. In addition, the smoother surface of the second batch bare silicon also drastically deteriorated the nucleate boiling heat transfer performance of the surface where its wall superheat at a given heat flux was much higher than that of the first batch bare silicon. For instance, at the heat flux of 7 W/cm^2 , the wall superheat for the first batch bare silicon was 15.7°C whereas the wall superheat for second batch bare silicon was as high as 29.2°C . This translates to approximately 50% reduction in heat transfer for

the second batch bare silicon. The calculated averaged heat transfer coefficients in the boiling regime for the first and second batch of silicon surface are 0.427 W/cm^2 and 0.244 W/cm^2 , respectively, with a difference of approximately 43%, whereas the maximum difference was approximately 54% at the heat flux of 6 W/cm^2 . Similarly, CHF was also lower for the second batch bare silicon and was recorded at 13 W/cm^2 as compared to 16 W/cm^2 for the first batch bare silicon.

Fig. 5.2 Comparison of saturated pool boiling curves of Batch 1 and Batch 2 bare silicon surfaces and CNT coated surfaces at 0° surface orientation.

On the other hand, the difference in surface roughness of bare silicon substrates seemed to have lesser effect on the boiling curve when the surfaces were coated with CNTs. As shown in Fig 5.2, even though the 90% CNT surface was produced from first batch bare silicon with greater surface roughness and the 100% CNT surface was produced from second batch bare silicon with smoother surface, the disparities between both boiling curves are less significant and both curves are almost identical up to 10 W/cm^2 . The average heat transfer coefficient in the nucleate boiling regime for the 90% CNT surface is 0.526 W/cm^2 whereas that of 100% CNT surface is

approximately $0.454/\text{cm}^2$ and the maximum difference between the two boiling curves is approximately 23% at the heat flux of 20 W/cm^2 . Both the 90% and 100% CNT coated surfaces were fabricated with similar techniques as detailed in Section 3.4. In addition, the results of 30%, 60% and 90% CNT coated surfaces shown in Fig. 4.5 also demonstrate the increase in heat transfer performance with increasing CNT area coverage. Hence, the discrepancy in results between the 90% and 100% CNT surfaces, whereby the 100% CNT surface showed lower heat transfer coefficient than the 90% CNT surface, would suggest that the surface roughness of bare silicon substrates, which the CNTs were coated on, has a direct influence on the heat transfer performance of the CNT coated surfaces. As the CNT coatings are porous, it was likely that when the surfaces were immersed in FC-72, the highly wetting fluid was able to penetrate the coating and wet the bare silicon surface. However, because the CNT structures were highly dense, it was unlikely that the bubble embryos that were formed at the silicon surface were able to overcome the resistive force of the CNT structures and successfully nucleate. Conversely, along locations where the CNTs were less dense, such as along the edge of the coating or along the coating perimeter, possible bubble nucleation from the silicon surface might occur. Hence, this may also explain the numerous bubble formations along the CNT coating perimeters.

In summary, it should be noted that the majority of the active nucleation sites for the 90% and 100% CNT coated surfaces were still produced from the porous structures formed by the CNTs as explained in detailed in Sections 4.2 and 4.3. This is evident as the 90% CNT coated surface demonstrated up to 50% enhancement in the heat transfer coefficient when compared to its base silicon whereas the 100% CNT coated surface possessed as high as 86% enhancement in the average heat transfer coefficient,

in the nucleate boiling regime, when compared to the second batch bare silicon. However, the bare silicon substrates served to further augment the heat transfer performance of the CNT surfaces by producing additional bubbles along the coating perimeters where the CNTs were less dense. This resulted in the 90% CNT coated surface outperforming the 100% CNT coated surface.

5.2 Combined effects of CNT coating and surface orientations

The effects on CNT coating and surface orientations were investigated. The bare silicon and 100% CNT coated surfaces were tested from 0° to 180° orientations whereas the interlaced 70%-CNT coated surface was tested from 0° to 90° orientations. Their results are presented and discussed in this section.

5.2.1 Effects on boiling incipience

Figure 5.3 compares the superheats at boiling incipience of bare silicon, 100% CNT and interlaced 70%-CNT surfaces for the range of orientations tested. The boiling incipience heat fluxes for the 100% CNT and the interlaced 70%-CNT surfaces were recorded at 2.4 W/cm² whereas the incipience for bare silicon was observed at the heat flux of 3.1 W/cm², suggesting that the heat flux required for initiating nucleate boiling was independent of surface inclination. In addition, the incipience superheats for the 100% CNT and interlaced 70%-CNT surfaces are also significantly lower as compared to bare silicon and this was primarily due to the effects of CNT coatings stated in Section 4.2.2. However, for all surfaces, there is no noticeable dependence of incipience superheat on orientation which is in agreement with the results of El-Genk and Bostanci [83] who tested dielectric fluid HFE-7100 on plain copper surface. In the present experiments, the boiling incipience superheat for bare silicon varied from

25.3°C to 17.3°C and boiling incipience superheats for the 100% CNT and interlaced 70%-CNT were recorded between 5.0°C and 9.0°C and between 7.3°C and 9.8°C, respectively.

Fig. 5.3 Comparison of the boiling incipience superheats of the bare silicon and the CNT surface at different surface orientations.

5.2.2 Effects on nucleate boiling heat transfer

The boiling curves of bare silicon and 100% CNT at the surface orientations of 0° (upward facing), 30°, 60°, 90°, 120°, 150° and 180° (downward facing) and the boiling curves of interlaced 70%-CNT at the surface orientation of 0°, 60° and 90° are presented in Fig. 5.4 where the end of each curve indicates the occurrence of CHF. In addition, the corresponding heat transfer coefficients of these surfaces are plotted in Fig. 5.5.

For bare silicon between 30° and 90° surface orientations, Figs. 5.4(a) and 5.5(a) show slight enhancements in heat transfer performance over 0° surface orientation at lower heat fluxes ($\leq 10.8 \text{ W/cm}^2$). For instance, at the same heat flux of 7 W/cm^2 , the superheats for 0°, 30°, 60° and 90° are 29.1°C, 27.9°C, 26.7°C and 27.6°C,

respectively. This reduction in superheats corresponds to 4.2%, 8.8% and 5.4% enhancements in heat transfer coefficient for 30°, 60° and 90°, respectively over 0° surface inclination. In all, for surface orientations smaller than 90° and at heat fluxes lower than 10.8 W/cm², increase in heat transfer enhancement with increasing surface orientation was observed up to 60° and this was followed by a reduction in enhancement as the surface orientation increased to 90°. However, as the heat flux increased beyond 10 W/cm², the enhancement in heat transfer for 30° to 90° surface orientations diminished which subsequently led to earlier occurrence of CHF as compared to 0° surface orientation. Comparatively, the boiling tests for bare silicon at 120° and 150° inclination angles show better heat transfer performance than 0° inclination angle for much shorter range of heat fluxes, i.e. when heat fluxes ≤ 4.9 W/cm². As the heat flux was increased beyond 4.9 W/cm², consistent heat transfer deterioration for 120° and 150° as compared to 0° prevailed and this deterioration increases as the orientation was increased from 120° to 150°. The above trends obtained for saturated pool boiling with bare silicon at different orientations were also reported by several investigators who experimented with plain surfaces [83, 84]. Increasing surface orientation enables heat transfer to be enhanced at lower heat fluxes as the surface gradient allow departing vapour bubbles to traverse across the boiling surface, thereby inducing thermal boundary layer mixing. This mixing of thermal boundary layer with surrounding cooler bulk fluid caused by the traversing bubbles in turn cools the boiling surface and improves sensible heat transfer efficiency. In addition, as the inclination angle increases, such as the case for 120° and 150°, the residence time of the traversing bubbles near the surface was also increased. This further promotes boundary layer mixing and hence, explains the heat transfer

enhancement observed for 120° and 150° at relatively low heat fluxes ($\leq 4.9 \text{ W/cm}^2$). Conversely, as more nucleation sites are activated with increasing heat flux, higher inclination angles and longer bubble residence time would result in premature accumulation and coalescence of vapour bubbles over the surface which impede heat transfer. This phenomenon agreed well with the experimental results where heat transfer deterioration as compared to 0° was observed beyond 4.9 W/cm^2 for large inclination angles ($> 90^\circ$) and at much higher heat flux (10.8 W/cm^2) for inclination angle $\leq 90^\circ$.

The boiling curves and heat transfer coefficient of the 100% CNT surface at different orientations are presented in Figs. 5.4(b) and 5.5(b), respectively. In comparison, the 100% CNT surface was capable of dissipating greater heat flux than bare silicon at all surface orientations for a given wall superheat and this was primarily due to the larger number of nucleation sites of the CNT surface as mentioned in Section 4.2.2. However, unlike bare silicon where heat transfer enhancement was seen as surface orientation increased (up to 90°), the 100% CNT surface demonstrated the highest heat transfer coefficient at 0° orientation and the heat transfer performance reduced with increasing surface inclination. As pointed out in Section 4.2.2, most of the nucleation sites were localised along the perimeter of the CNT coating at low heat fluxes which resulted in overcrowding bubble growth along the boundary. With increasing surface orientation, lateral coalescence of the overcrowding bubbles near the surface easily occurs, forming a vapour layer which hinders heat transfer and increases wall superheat. Based on high speed visualisation, between 0° and 30°, bubbles departed nearly perpendicular to the surface and minimal bubble interaction was observed. However, as the surface orientation was increased to 60°, significant accumulation and

coalescence of departing bubbles along the CNT perimeter took place. This phenomenon agreed well with the experimental result of Fig 5.5(b) which shows that the reduction in heat transfer coefficient was initially insignificant as surface orientation was increased from 0° to 30° but substantial when the inclination angle was to 60° and 90° . For instance, at the heat flux of 12.8 W/cm^2 , the heat transfer coefficients of CNT surface at 0° and 30° are approximately $0.48 \text{ W/cm}^2\cdot\text{K}$ and $0.45 \text{ W/cm}^2\cdot\text{K}$, respectively but as the surface orientation was increased to 60° and 90° , 14% and 21% reductions in heat transfer coefficients were separately recorded.

The pool boiling results with the interlaced 70%-CNT surface is plotted in Fig. 5.4(c) and the corresponding heat transfer coefficient is shown in Fig. 5.5(c). Similar to the 100% CNT surface, the interlaced 70%-CNT surface display higher heat transfer performance than the bare silicon surface at all orientations that were experimented (i.e. 0° , 60° and 90°). In addition, pool boiling curves of the interlaced 70%-CNT surface also show similar dependence on surface orientation as the 100% CNT surface where the most optimal heat transfer orientation was observed at 0° and the increase in surface orientation only resulted in the rightward shift of the boiling curves as shown in Fig. 5.4(c), indicating the reduction in heat transfer performance. The increased accumulation of the vapour bubbles along the CNT edges which further suppressed heat transfer as the surface orientation increased, similar to that seen in the 100% CNT surface, elucidated this trend.

Fig. 5.4 Comparison of the saturated pool boiling curves at different surface orientations with FC-72 on (a) bare silicon and (b) 100% CNT surface (c) interlaced 70%-CNT.

Fig. 5.5 Comparison of the saturated pool boiling heat transfer coefficient at different surface orientations with FC-72 on (a) bare silicon and (b) 100% CNT surface (c) interlaced 70%-CNT.

Figure 5.6 compares the heat transfer coefficient of the 100% CNT surface and the interlaced 70%-CNT surface at 0° and 90° inclination angles and it can be clearly seen that the heat transfer characteristics of the two surfaces are highly dependent on the surface orientation rather than effects of CNT coating area and perimeter. For instance, at the same surface orientation, surfaces with different CNT coating (100% and interlaced 70%) show smaller differences in heat transfer coefficient as compared to surfaces with the same CNT coating but at different surface orientations (0° and 90°). In addition, the interlaced 70%-CNT surface also show higher heat transfer coefficient than 100% CNT surface at 0°. However, this enhancement in heat transfer diminishes as the surface inclination is increased to 90°. For instance at 0°, up to 8% enhancement for interlaced 70%-CNT as compared to 100% CNT was recorded but when surface orientation was increased to 90° no significant enhancement was recorded. This is likely due to the additional CNT edges of the interlaced surface which introduced larger regions where bubbles accumulated and resulted in further deterioration in heat transfer as compared to 100% CNT when surface inclination increased to 90°.

In an effort to compare the overall boiling performance of bare silicon, 100% CNT and interlaced 70%-CNT in relation to surface orientation, the average heat transfer coefficients was derived by averaging the heat transfer coefficient over the range of nucleate boiling regime (i.e. from 3.1 W/cm² to CHF for 100% CNT and interlaced 70%-CNT surfaces and from 3.1 W/cm² to CHF for bare silicon). As plotted in Fig. 5.7, both 100%-CNT and interlaced 70%-CNT surfaces have the highest average heat transfer coefficient (0.45 W/cm²·K) at 0° and is 86% greater than bare silicon at the same orientation hence, indicating the most ideal condition for CNT coating

application. It should be noted that even though higher heat transfer coefficient was recorded for the interlaced 70%-CNT surface as compared to 100% CNT surface in the intermediate heat flux range, it displays poorer heat transfer coefficient in the lower and upper heat flux regimes which resulted in the average heat transfer coefficient to be similar to that of 100% CNT surface. In addition, the plot also clearly displays the gradual increment in average heat transfer coefficient of bare silicon from 0° to 90° before the average heat transfer coefficient reduces as the surface inclination is increased above 90°. Alternatively, for 100% CNT surface, the initial gradual decrease in average heat transfer coefficient with the increment of surface inclination from 0° to 30° was followed by steep reduction in average heat transfer coefficient from 30° to 90° before levelling off as surface inclination increases from 90° to 150°. The interlaced 70%-CNT surface, on the other hand, shows a similar trend as the 100% CNT surface.

Fig. 5.6 Comparison of heat transfer coefficient of 100% CNT and interlaced 70%-CNT surfaces at 0° and 90° surface orientations.

Fig. 5.7 Comparison of the average heat transfer coefficient of bare silicon, 100% CNT and interlaced 70%-CNT surfaces at different surface orientations.

The linear regression (least square) method was adopted to empirically correlate the average heat transfer coefficient as a polynomial function of surface orientation for the bare silicon and CNT coated surfaces (100% CNT and interlaced 70% CNT). The objective function is shown in Eq. (5.1), where h_{ave} is expressed as a function of θ and C_1 , C_2 , C_3 and C_4 are constants to be determined. In order to minimise the error of C_1 , C_2 , C_3 and C_4 , Eq. (5.1) is rewritten as shown in Eq. (5.2), where S is to be minimised by taking the partial derivative of each constant. The computation was carried in MATLAB in accordance to the scheme shown in Fig. 5.8 and the pseudocode is shown in Appendix E. The correlations developed from the linear regression method for both bare silicon and CNT surfaces are shown in Eqs. (5.3) and (5.4).

$$h_{ave}(\theta) = C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3 \quad (5.1)$$

$$S = \{h_{ave}(\theta) - [C_1 + C_2\theta + C_3\theta^2 + C_4\theta^3]\}^2 \quad (5.2)$$

Fig. 5.8 Computational scheme for h_{ave} correlation.

Bare silicon:

$$h_{ave} = 0.25 + (4.20 \times 10^{-4})\theta + (1.32 \times 10^{-5})\theta^2 + (8.13 \times 10^{-8})\theta^3 \quad (5.3)$$

CNT surfaces (100% CNT and interlaced 70%-CNT):

$$h_{ave} = 0.46 + (2.09 \times 10^{-3})\theta + (1.98 \times 10^{-4})\theta^2 + (7.82 \times 10^{-8})\theta^3 \quad (5.4)$$

5.2.3 Effects on critical heat flux

Figure 5.9 depicts the CHF values of bare silicon, 100% CNT and interlaced 70%-CNT surfaces recorded at different orientations and compares the results with data obtained by various researchers for FC-72 on different surfaces. In the present study, CHF for the three surfaces was reduced gradually as the orientation increased from 0°

to 150° followed by a steeper decline at 180° . These results agree well with the general trend reported by various researchers as shown in Fig. 5.9. As mentioned above, the increased surface orientation increased the residence time of the vapour bubbles which resulted in early bubble coalescence and advanced CHF. In addition, the CNT coated surfaces also exhibits higher CHF as compared to bare silicon for all orientations although there is no clear dependence of the magnitude of enhancement on orientation. At 0° orientation, CHF for the 100% CNT surface (20.1 W/cm^2) is 42% higher than bare silicon (14.1 W/cm^2) and for the entire range of orientations, CHF enhancement for the 100% CNT surface lies between 34% and 50%. On the other hand, the interlaced 70%-CNT was found to enhance CHF between 30% and 35% as compared to bare silicon for the range of orientations tested. As mentioned in Section 4.2.1, the enhancement in CHF was likely due to the capillary wicking induced by micro/nanostructured surfaces which resulted in the rapid resupply of fluid to the dry out region and delay the boiling crisis. The nanoscale CNT structures form microscale cavities and intertwining meshes which are analogous to micro/nanostructured surfaces which are capable of supplementing FC-72 to the dry out region by the aid of capillary action and increasing CHF.

The hydrodynamic model for CHF developed by Zuber [46] is shown in Eq. (2.39), and is rewritten as Eq. (5.5) for convenience. The constant term, C , was originally proposed by Zuber [46] to be $\pi/24$ (or 0.131), which was partially related to the ratio of the vapour column surface area during CHF to the total area of the heating surface. Recently, researchers [83, 84] who investigated in pool boiling with surface orientations have also suggested that C could further broken down, as shown in Eq. (5.6), to include the surface orientation effect, where $C_{sat,0^\circ}$, is evaluated under

saturated pool boiling at 0° orientation. Chang and You [86] and Pirarone [84] separately obtained 0.15 and 0.165 for $C_{sat,0^\circ}$ for plain copper surface whereas microporous and porous graphite enhanced surfaces yielded higher $C_{sat,0^\circ}$ values [86, 94]. For the present study, $C_{sat,0^\circ}$ for bare silicon, 100% CNT and interlaced 70%-CNT surfaces are 0.13, 0.19 and 0.18, respectively. $f(\theta)$ can be hence be determined with the use of linear regression analysis, where the objective function and the error term to be minimised is shown in Eqs. (5.7) and (5.8). Using similar computational scheme as shown Fig. 5.8, the function $f(\theta)$ which is applicable to predict the CHF with respect to the surface orientation for the bare silicon, 100% CNT and the interlaced CNT surfaces is developed as written in Eq. (5.9).

Fig. 5.9 CHF of bare silicon, 100% CNT and interlaced 70%-CNT surfaces at different surface orientations.

$$q'' = C \rho_g^{1/2} h_{fg} [g \sigma (\rho_l - \rho_g)]^{1/4} \quad (5.5)$$

$$C = C_{sat,0^\circ} \times f(\theta) \quad (5.6)$$

$$f(\theta) = C_1 + C_2 \theta + C_3 \theta^2 - C_4 \theta^3 \quad (5.7)$$

$$S = \{f(\theta) - [C_1 + C_2 \theta + C_3 \theta^2 - C_4 \theta^3]\}^2 \quad (5.8)$$

$$f(\theta) = 1 - (1.69 \times 10^{-3})\theta + (2.48 \times 10^{-5})\theta^2 - (1.82 \times 10^{-7})\theta^3 \quad (5.9)$$

CHAPTER 6

CORRELATIONS FOR SATURATED POOL BOILING CURVES WITH CNT COATED SURFACES

Saturated pool boiling results with CNT coated surfaces suggest that parameters such as CNT coating area, CNT coating perimeter, substrate surface roughness and surface orientation have direct influence on the overall heat transfer characteristics and the pool boiling curves of the surfaces. In this section, a unified correlation is proposed to characterise the saturated pool boiling curves of CNT coated surfaces, incorporating the effects mentioned above.

The single phase forced convection model for pool boiling proposed by Rohsenow [24], as shown in Eq. (2.18) and rewritten below, provides reasonably good predictions of the pool boiling characteristics for different types of surfaces and fluids. This correlation relates the different heat transfer regimes with the fluid thermophysical properties and wall superheat. In addition, the coefficient, C_{sf} , and the exponential term, r , which take into account the effects of surface characteristics, surface-fluid combination and surface orientation, also allow flexibility for their numerical values to be altered in accordance to the different boiling conditions. Jung et al. [85] proposed the range of C_{sf} and r which depended on surface orientation for R-11 and subsequently, a similar method was adopted by Priarone [84] for FC-72 and HFE-7100 at low heat fluxes. In both cases, the exponential term, s , which similarly depended on surface-fluid combination, was assumed to be constant for all orientations and it was determined that orientation had only modest effect on C_{sf} whereas variation of r was relatively more significant for plain copper surface. In addition, Chang and You [95] also employed Rohsenow's correlation to characterise the pool boiling curves of

FC-72 on diamond particle coated surfaces where r and C_{sf} were observed to vary with the mean particle diameter.

The Rohsenow model was widely used for correlating pool boiling data as this model allow flexibility for the constants to be altered to fit the different pool boiling conditions. In addition, this semi-empirical model also provides substantial relation to the boiling mechanisms such as bubble dynamics and thermophysical properties of the fluid. Hence, this model was selected to correlate the experimental data obtained for the bare silicon and CNT surface in the present study.

$$q'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_g)}{\sigma} \right]^{\frac{1}{2}} \left[\frac{C_{pl}(T_w - T_{sat})}{C_{sf} h_{fg} \text{Pr}_l^s} \right]^{\frac{1}{r}} \quad (2.18)$$

6.1 Computational scheme

The Gauss-Newton method was employed to compute C_{sf} , r and s for bare silicon at 0° orientation with the experimental data and the thermophysical properties of FC-72. Based on the computed value of s with bare silicon, this value was then retained for the computation of C_{sf} and r of the remaining CNT and bare silicon surfaces of different coating patterns and orientations. Assuming the objective function, Eq. (2.18) takes the form $y = f(x)$ and using the Taylor series expansion, the first order linearised model is shown in Eq. (6.1) where y_i is the heat flux obtained experimentally, $f(x_i)_j$ is the computed heat flux, a_k are constants to be correlated (i.e. C_{sf} , r and s), Δa_k is the difference in values between the current and previous steps, and e_i is the truncation error of higher order derivatives. From the linear least-square theory and assuming e_i to be negligible, Eq. (6.1) can be rewritten in matrix form as shown in Eq. (6.2).

$$y_i - f(x_i)_j = \sum_{k=0}^m \frac{\partial f(x_i)_j}{\partial a_k} \Delta a_k + e_i \quad (6.1)$$

$$[Z_j]^T [Z_j] \{\Delta A\} = [Z_j]^T \{D\} \quad (6.2a)$$

$$\{D\} = \begin{Bmatrix} y_1 - f(x_1) \\ y_2 - f(x_2) \\ \vdots \\ y_n - f(x_n) \end{Bmatrix} \quad (6.2b)$$

$$[Z_j] = \begin{Bmatrix} \frac{\partial f_1}{\partial a_0} & \frac{\partial f_1}{\partial a_1} & \cdots & \frac{\partial f_1}{\partial a_m} \\ \frac{\partial f_2}{\partial a_0} & \frac{\partial f_2}{\partial a_1} & \cdots & \frac{\partial f_2}{\partial a_m} \\ \vdots & \vdots & \cdots & \vdots \\ \frac{\partial f_n}{\partial a_0} & \frac{\partial f_n}{\partial a_1} & \cdots & \frac{\partial f_n}{\partial a_m} \end{Bmatrix} \quad (6.2c)$$

$$\{\Delta A\} = \begin{Bmatrix} \Delta a_0 \\ \Delta a_1 \\ \vdots \\ \Delta a_m \end{Bmatrix} \quad (6.2d)$$

The computation was carried out by using MATLAB in accordance to the iterative scheme shown in Fig. 6.1 and the pseudocode is shown in Appendix G. The thermophysical properties of FC-72 were first substituted into the objective function [Eq. (2.18)] and partial differentiation was performed on Eq. (2.18) with respect to the constants to be determined (C_{sf} , r). Initial guess values of C_{sf} and r together with the values of the wall superheats determined experimentally were then substituted into the partial derivatives to obtain $[Z_j]$, which started the iterative process. Similarly, the initial guess values of C_{sf} , r and the experimental wall superheats were also substituted into Eq. (2.18) to obtain the calculated q'' . The difference between the calculated and

experimental q'' was then computed to obtain $\{D\}$. Using the linear least-square equation [Eq. (6.2a)] and the computed values of $[Z_j]$, $\{D\}$ and $[Z_j]^T$, the new values of C_{sf} , and r were obtained. Finally, the program would then compute the difference between the new and the previous values of C_{sf} , and r where the stopping criterion is defined as Eq. (6.3) and η is chosen as 0.0001.

$$\{\Delta A_1\} - \{\Delta A_0\} < \eta \quad (6.3)$$

6.2 Computational results

The s term was initially calculated to be 3.5 with the bare silicon (Batch 2) data at 0° orientation and this value was retained for the remaining surfaces and boiling conditions. Table 6.1 shows the computational results of C_{sf} and r for bare silicon surfaces and CNT coated surfaces of different area coverages and perimeters at 0° surface orientation. For these surfaces, the C_{sf} values ranged between 4.58 and 14.6 and the r values are between 0.768 and 1.247. With the exception of Batch 2 bare silicon, the remaining surfaces are within ± 1.6 (26%) and ± 0.11 (10%) of the average C_{sf} and r values respectively. On the other hand, the Batch 2 bare silicon resulted in distinctly different C_{sf} (14.6) and r (0.768) values from the remaining surfaces and this was primarily due to the highly smooth surface which resulted in the reduction in nucleate boiling heat transfer and the drastic rightward shift of the boiling curves. In order to assess the accuracy of the correlations, the mean computational error, ε , for each correlated data was calculated using Eq. (6.4) where q''_{exp} denotes the experimental heat flux, q''_{com} is the calculated heat flux based on the correlation obtained for each type of surface and N represents the number of experimental points for individual surface boiling curve. The calculated ε values for individual surfaces

are shown in Table 6.1. It can be seen that reasonably accurate prediction was achieved with the Rohsenow model where the maximum ε is 22.7% for the interlaced 70% CNT surface and the minimum is 7.9% for the Batch 1 bare silicon. The pool boiling curves of the Batch 1 bare silicon and interlaced 60% CNT with their corresponding correlations are plotted in Fig. 6.2. It can be seen that in the natural convection regime, at the boiling incipience and as the boiling was approaching CHF, larger differences between the correlations and the experimental results were observed which contributed to large portion of ε . As the correlation, which based on Rohsenow model, was developed on the basis of convective heat transfer induced by the motion of the vapour bubbles during boiling, the mechanisms in natural convection and CHF regimes were not accounted for in this model and hence explained the larger error observed. However, in the boiling regime, which covered the major portion of the boiling curves, the correlations agreed exceedingly well with the experimental data.

$$\varepsilon = \sqrt{\frac{1}{N} \sum_{i=1}^N \left(\frac{q''_{exp} - q''_{com}}{q''_{exp}} \right)^2} \quad (6.4)$$

Table 6.1 C_{sf} and r values of bare silicon and CNT surfaces at 0° surface orientation.

	Bare silicon (Batch 1)	Bare silicon (Batch 2)	30% CNT (Batch 1)	60% CNT (Batch 1)	90% CNT (Batch 1)	100% CNT (Batch 2)	Interlaced 60% CNT (Batch 1)	Interlaced 70% CNT (Batch 2)
C_{sf} ($\times 10^{-5}$)	6.86	14.6	6.60	6.73	5.64	6.62	4.58	6.43
r	1.167	0.768	1.142	1.065	1.084	1.089	1.247	1.146
$\varepsilon(\%)$	7.9	9.3	8.1	9.3	10.2	10.9	15.6	22.7

Fig. 6.1 Gauss-Newton iterative scheme.

Fig. 6.2 Comparison of experimental results against correlation.

The computational results for the Batch 2 bare silicon, 100% CNT and interlaced 70%-CNT surfaces at surface orientations between 0° and 180° are shown in Table 6.2 where the respective C_{sf} and r values are similarly applicable for the entire range of heat fluxes. Between 0° and 150° , there is insignificant variation in C_{sf} for bare silicon but r increased systematically with increasing orientation. The difference in the r values at 0° and at 150° was approximately 26% and was further increased to 50% at 180° . The computed results for bare silicon agreed well with the trend observed by Priarone [84] and Jung et al. [85] for the plain surface. Unlike bare silicon, C_{sf} for 100% CNT and interlaced 70% CNT surfaces increased whereas r reduce with increasing orientation and the variation in C_{sf} is also greater than the variation in r . For instance, between 0° and 150° , C_{sf} values of 100% CNT surface vary by 42% while r values only differ by 14%. This unique trend for CNT surfaces is primarily due to the leftward shift of the boiling curve with increasing orientation. For a fixed surface

orientation, it can also be observed that there are only marginal differences in C_{sf} and r values between 100% CNT and interlaced 70% CNT surfaces. Lastly, based on the C_{sf} and r values computed for the surfaces between 0° and 180° surface orientation, the constant can be expressed as polynomial functions of θ for both bare silicon and CNT surfaces as shown in Eqs. (6.5) and (6.6).

Bare silicon:

$$C_{sf} = 1.44 \times 10^{-4} + (2.61 \times 10^{-7})\theta - (7.62 \times 10^{-9})\theta^2 + (4.32 \times 10^{-11})\theta^3 \quad (6.5a)$$

$$r = 0.80 - (7.23 \times 10^{-4})\theta + (1.38 \times 10^{-5})\theta^2 \quad (6.5b)$$

100% CNT/ Interlaced 70% CNT surfaces:

$$C_{sf} = 6.33 \times 10^{-5} + (6.07 \times 10^{-7})\theta - (5.91 \times 10^{-9})\theta^2 + (2.34 \times 10^{-11})\theta^3 \quad (6.6a)$$

$$r = 1.17 - (7.28 \times 10^{-3})\theta + (4.75 \times 10^{-5})\theta^2 \quad (6.6b)$$

Table 6.2 C_{sf} and r values of bare silicon (Batch 2), 100% CNT surface (Batch 2) and 70% Interlaced (batch 2) from 0° to 180° surface orientation.

	Surface	0°	30°	60°	90°	120°	150°	180°
C_{sf} ($\times 10^{-5}$)	Bare silicon	14.6	14.2	13.8	14.3	14.6	14.6	20.0
	100% CNT	6.62	6.99	8.43	9.36	9.12	9.51	12.0
	Interlaced 70% CNT	6.43	-	8.32	8.74	-	-	-
r	Bare silicon	0.768	0.820	0.861	0.820	0.897	0.967	1.151
	100% CNT	1.089	1.083	0.971	0.918	0.954	0.979	1.518
	Interlaced 70% CNT	1.146	-	1.024	1.029	-	-	-
ε (%)	Bare silicon	9.3	12.5	10.6	10.6	9.8	6.0	30.2
	100% CNT	10.9	10.7	8.4	16.0	17.7	13.2	11.4

	Interlaced 70% CNT	22.3	-	21.9	17.8	-	-	-
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6.3 A general correlation for saturated pool boiling of FC-72 with CNT coated surfaces

In order to include the effects of silicon substrate surface roughness, CNT coating area, CNT coating perimeter and surface orientation into the C_{sf} and r terms, Eq. (6.7) is proposed, where C_{sf} and r are functions of $R_{silicon}$ which denotes the root mean square (rms) surface roughness of the bare silicon surface, A_{CNT} which is the CNT coated area expressed in terms of bare silicon area and P_{CNT} the CNT coated perimeter expressed in terms of bare silicon perimeter. For instance, the 30% CNT coated surface has $R_{silicon}$ of 450 nm, A_{CNT} of $0.3L^2$ and P_{CNT} of $2.2L$ where L is the length of the bare silicon.

$$C_{sf} = C_{sf}(R_{silicon}, A_{CNT}, P_{CNT}, \theta) \quad (6.7a)$$

$$r = r(R_{silicon}, A_{CNT}, P_{CNT}, \theta) \quad (6.7b)$$

The effect of surface orientations has been considered and incorporated into C_{sf} and r as shown in Eqs. (6.5) and (6.6). For saturated pool boiling of surfaces at 0° orientation, Eq. (6.7) can be rewritten in dimensionless form as shown in Eq. (6.8).

$$C_{sf,0^\circ} = C_{sf,0^\circ} \left(\frac{R_{silicon}}{R_{silicon,ref}}, \frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right) \quad (6.8a)$$

$$r_{0^\circ} = r_{0^\circ} \left(\frac{R_{silicon}}{R_{silicon,ref}}, \frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right) \quad (6.8b)$$

Based on the computed C_{sf} and r values for Batch 1 and Batch 2 bare silicon surfaces in Section 6.2, the constants can be related to the dimensionless surface roughness as shown in Eq. (6.9) where the reference surface roughness, $R_{silicon,ref}$, is taken to be the root mean square surface roughness of Batch 1 bare silicon, 450 nm.

$$C_{sf,silicon,0^\circ} = 6.68 \times 10^{-5} \left(\frac{R_{silicon}}{450} \right)^{-0.222} \quad (6.9a)$$

$$r_{silicon,0^\circ} = 1.167 \left(\frac{R_{silicon}}{450} \right)^{-0.123} \quad (6.9b)$$

In order to include the effects of CNT coating area and perimeter, the dimensionless parameter $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ is employed. Figure 6.3 shows the variation of $\frac{C_{sf,0^\circ}}{C_{sf,silicon,0^\circ}}$ and against $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ where $C_{sf,silicon,0^\circ}$ is obtained from Eq. (6.9a) and $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ is calculated from the respective CNT coated surfaces. For instance, the 30% CNT coated surface with A_{CNT} and P_{CNT} values of $0.3L^2$ and $2.2L$, respectively has the calculated $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ value of 4.49. This approach is similarly adopted for the variation of $\frac{r_{0^\circ}}{r_{silicon,0^\circ}}$ against $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ as shown in Fig. 6.4. By using the least square curving fitting method, the exponential relationship between the constants and the dimensionless parameter can be expressed as follows:

$$\frac{C_{sf,0^\circ}}{C_{sf,silicon,0^\circ}} = 1.054 \left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)^{-0.0949} \left(\frac{R_{silicon}}{450} \right)^{0.17} \quad (6.10a)$$

$$\frac{r_{0^\circ}}{r_{silicon,0^\circ}} = 0.994 \left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)^{0.0183} \left(\frac{R_{silicon}}{450} \right)^{-0.11} \quad (6.10b)$$

Fig. 6.3 Variation of $\frac{C_{sf,0^\circ}}{C_{sf,silicon,0^\circ}}$ against $\frac{L^2+P_{CNT}^2}{L^2+A_{CNT}}$ for Batches 1 and 2 CNT-coated surfaces.

Fig. 6.4 Variation of $\frac{r_{0^\circ}}{r_{silicon,0^\circ}}$ against $\frac{L^2+P_{CNT}^2}{L^2+A_{CNT}}$ for Batches 1 and 2 CNT coated surfaces.

By substituting Eq. (6.9a) into Eq. (6.10a) and Eq. (6.9b) into Eq. (6.10b), a general correlation for saturated pool boiling with FC-72 on CNT coated surfaces with the effects of silicon substrate surface roughness, CNT coating area and perimeter is developed from the Rohsenow model where $C_{sf,0^\circ}$ and r_{0° can be written as

$$C_{sf,0^\circ} = 7.23 \times 10^{-5} \left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)^{-0.0949} \left(\frac{R_{silicon}}{450} \right)^{-0.052} \quad (6.11a)$$

$$r_{0^\circ} = 1.16 \left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)^{0.0183} \left(\frac{R_{silicon}}{450} \right)^{0.013} \quad (6.11b)$$

Finally, to take into account the effects of surface orientation, Eqs. (6.11a) and (6.11b) can replace the first right hand term of Eq. (6.6) developed for surface orientation to give the generalised expressions for C_{sf} and r as follows:

$$C_{sf} = C_{sf,0^\circ} + (6.07 \times 10^{-7})\theta - (5.91 \times 10^{-9})\theta^2 + (2.34 \times 10^{-11})\theta^3 \quad (6.12a)$$

$$r = r_{0^\circ} - (7.28 \times 10^{-3})\theta + (4.75 \times 10^{-5})\theta^2 \quad (6.12b)$$

It should also be noted that the above expressions are only applicable for CNT coated surface up to the minimum coating area of 30% whereas for the prediction of pool boiling heat transfer with bare silicon surfaces, Eq. (6.9) should be used.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

This section serves to summarise the major findings of this thesis and the relevant contributions to the development of pool boiling heat transfer. In addition, suggestions for further research are also provided to further advance the knowledge and application of the present work.

7.1 Conclusions

In this thesis, saturated pool boiling of surfaces coated with CNT using CVD technique was extensively studied and the following briefly summarises the major findings.

1. Ageing effect of CNT coated surfaces

The newly prepared CNT coated surfaces have demonstrated lower CHF and higher wall temperature in the initial test. Only after two boiling tests and more than 10 hours of immersion in FC-72, were consistent pool boiling results achieved where the aged CNT surface possessed improved heat transfer performance as compared to new CNT surface. This trend was observed for all the newly prepared CNT coated surfaces which confirmed the ageing of CNT. Due to the large CNT coating thickness employed in the present study, it was postulated that the CNTs were initially not well-wetted by surrounding fluid which explained the initial higher superheat and earlier CHF. Only with the continuous immersion of the surface and successive boiling tests, were the optimal fluid wetting paths established and hence allowed stable and maximum wetting of the CNTs.

2. Effect of CNT coating on boiling incipience

CNT coating resulted in lower heat fluxes and lower wall superheats for boiling incipience as compared to bare silicon. However there was no clear dependence of incipience wall superheat and heat flux on the CNT coating area, perimeter and surface orientations. Based on the transient thermal boundary layer developed by Hsu [29] and the bubble departure diameter captured from the experiment, the range of active nucleation sites for FC-72 was computed and it was determined that the lower incipience superheat of CNT surfaces was primarily due to the sizeable cavities created by the intertwining CNT structures.

3. Effect of CNT coating on nucleation boiling heat transfer

CNT coated surfaces have demonstrated superior nucleate boiling heat transfer performances as compared to the silicon substrates for all variants of coating area and perimeters tested in this research. Based on the analysis conducted with Hsu's model [29] and the SEM micrograph taken for the CNT surfaces, it was determined that the submicron sized cavities created by the intertwining CNT structures resulted in the formation of large bubble quantities that enhanced heat transfer. In addition, as the quantity of cavities increased with increased CNT coating area, enhancement in heat transfer was also found to increase with increasing CNT area coverage where the highest heat transfer coefficient of 0.56 W/cm^2 was obtained with the 90% CNT coated surface at the heat flux of 18 W/cm^2 . Furthermore, due to the large CNT coating thickness ($215 \text{ }\mu\text{m}$) employed in this study, it was determined that sizeable cavities were also formed along the sides of the CNT coating which allowed the formation of vapour bubbles. Hence, in addition to larger CNT coating area, enhancing boiling heat

transfer could also be accomplished by increasing the CNT coating perimeter which was evidently demonstrated with the interlaced 60% and 70% CNT surfaces. Finally, for the interlaced surfaces, the phenomenon of lateral bubble growth along the edges of the CNT surface was also observed leading to the early coalescence of bubbles at low heat fluxes which might induce mixing of thermal boundary layer and further promote heat transfer.

4. Effect of CNT coating on CHF

CNT coatings have shown to enhance CHF as compared to bare silicon and the CHF was seen to increase with increasing CNT coating area where the highest CHF was recorded for the 90% CNT surface at 24 W/cm^2 and approximately 50% enhancement was achieved. A sessile drop test on the CNT surface with FC-72 revealed that as the droplet came into contact with the CNT, it rapidly spread and seeped into the CNT layer. The liquid spreading effect on micro/nano structured surfaces was determined to be the result of capillary wicking action caused by the capillary network created by the meshing of CNT structures which assisted to replenish the dryout region during boiling and delay the occurrence of CHF.

5. Effects of silicon substrate roughness on nucleate boiling heat transfer of CNT coated surfaces

Silicon surfaces of different surface roughness have shown to exhibit significantly different pool boiling heat transfer performance. The average heat transfer coefficients in the boiling regime for the first batch of silicon surface was 0.427 W/cm^2 ($R_{\text{rms}} = 450 \text{ nm}$) whereas the second batch of silicon with smoother surface ($R_{\text{rms}} = 15 \text{ nm}$) was determined to be 0.244 W/cm^2 . Although less significant, the heat transfer

performance of the silicon substrates was also shown to have direct influence on the overall heat transfer performance of the CNT coated surfaces. This is evident when comparing the 90% CNT surface produced from first batch silicon substrate and 100% CNT surface produced from second batch silicon substrate where even with higher CNT coating area, the 100% CNT surface ($h_{ave} = 0.454 \text{ W/cm}^2$) has displayed lower average heat transfer coefficient as compared to 90% CNT surface ($h_{ave} = 0.526 \text{ W/cm}^2$). It was suggested that the bare silicon substrates served to further augment the heat transfer performance of the CNT surfaces by producing additional bubbles along the coating perimeters where the CNTs were less dense which resulted in the 90% CNT coated surface outperforming the 100% CNT coated surface.

6. Combined effects of CNT coating and surface orientations

Bare silicon, 100% CNT and interlaced 70% CNT surfaces were also tested between 0° and 180° surface orientations and it was found that surface orientation has no significant effect on heat transfer performance of 100% CNT surface with the initial increase in orientation from 0° to 30° . There was, however, substantial reduction in the average heat transfer coefficient, indicated by the rightward shift of the boiling curve, with increasing orientation 60° and above. The effect of orientation on CNT surface was contrary to those obtained from bare silicon where gradual increment in heat transfer between 0° and 90° preceded the decrease in heat transfer as orientation increased above 90° . A similar trend was observed for the interlaced 70% CNT surface. At 0° orientation, the highest enhancement in the average heat transfer coefficient of 86% was recorded for both interlaced 70% CNT and 100% CNT surfaces

(0.45 W/cm²·K) as compared to bare silicon (0.24 W/cm²·K), denoting that this is the most suitable orientation for CNT surface application.

7. Correlations for saturated pool boiling with CNT coated surfaces

With the use of linear regression (least square) method, correlations of h_{ave} as polynomial functions of surface orientations were proposed for bare silicon, 100% CNT and interlaced 70% CNT where h_{ave} was determined to decrease with the increase in surface orientation. In addition, on the basis of the hydrodynamic instability model, this thesis also attempted to correlate CHF of the CNT surfaces by incorporating the effects of surface orientations, rewritten in Eq. (7.1). The constant term, C , originally proposed by Zuber [46] was further broken down and correlated as a polynomial function of surface orientation. The correlation shows good agreement with the experimental results of all surfaces where the errors were found to be within the experimental uncertainty range. From this study, the CHFs for all surfaces were determined to reduce gradually as the orientation increased from 0° to 150° followed by a steeper decline at 180°.

$$q'' = C_{sat,0^\circ} \{ 1 - (1.69 \times 10^{-3})\theta + (2.48 \times 10^{-5})\theta^2 - (1.82 \times 10^{-7})\theta^3 \} \rho_g^{1/2} h_{fg} [g\sigma(\rho_l - \rho_g)]^{1/4} \quad (7.1)$$

The pool boiling curves for all surfaces were characterised with the Rohsenow model, rewritten in Eq. (7.2), where the exponential s term was fixed at 3.5 and the Gauss-Newton computational method was adopted to determine the constant C_{sf} and the exponential term r . Reasonably good correlations were obtained for all surfaces where

the highest mean computational errors were calculated to be 10.6%, 17.7% and 22.7% for bare silicon, uniformly coated CNT and interlaced CNT surfaces, respectively.

$$q'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_g)}{\sigma} \right]^{\frac{1}{2}} \left[\frac{C_{pl}(T_w - T_{sat})}{C_{sf} h_{fg} \text{Pr}_l^{3.5}} \right]^{\frac{1}{r}} \quad (7.2)$$

Finally, with the computed values of C_{sf} and r , a general correlation capable of characterising the saturated pool boiling curves of CNT coated surfaces was developed. By introducing dimensionless functions and parameters related to the silicon substrate surface roughness, CNT coating areas, CNT coating perimeters and surface orientations to the C_{sf} [Eqs. (7.3) and (7.4)] and r [Eqs. (7.5) and (7.6)], this correlation allowed the flexibility to predict the nucleate boiling performances of a wide variation of CNT coated surfaces.

$$C_{sf,0^\circ} = 7.23 \times 10^{-5} \left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)^{-0.0949} \left(\frac{R_{silicon}}{450} \right)^{-0.052} \quad (7.3)$$

$$C_{sf} = C_{sf,0^\circ} + (6.07 \times 10^{-7})\theta - (5.91 \times 10^{-9})\theta^2 + (2.34 \times 10^{-11})\theta^3 \quad (7.4)$$

$$r_{0^\circ} = 1.16 \left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)^{0.0183} \left(\frac{R_{silicon}}{450} \right)^{0.013} \quad (7.5)$$

$$r = r_{0^\circ} - (7.28 \times 10^{-3})\theta + (4.75 \times 10^{-5})\theta^2 \quad (7.6)$$

Arising from this thesis, the different mechanisms and parameters affecting pool boiling performances such as boiling incipience, heat transfer coefficient and CHF of CNT coated surfaces have been identified. First, it was determined that both CNT coating area and perimeter have significant effects on pool boiling heat transfer with FC-72 as coolant and the optimisation of heat transfer performance with CNT coated surface can be achieved by maximising $\left(\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}} \right)$. Second, even though less

significant, the surface roughness of the silicon substrate also contributed to the heat transfer performance of the CNT coated surfaces, where the substrates with higher surface roughness have shown to possess higher heat transfer coefficient even with marginally smaller CNT coating (i.e. 90% as compared to 100%). Third, due to reduced bubble residual time, experiments have also verified that the highest heat transfer coefficient and CHF were achieved with CNT coated surfaces at the 0° (upright) orientation. Finally, the correlations developed for averaged heat transfer coefficients, CHF and characterization of pool boiling curves, which provided reasonably accurate prediction when validated against present experimental data, can be adopted when employing CNT coated surfaces for immersion cooling applications.

7.2 Recommendations for future work

1. Further validation of the pool boiling correlations

In this thesis, general correlations for the characterisation of pool boiling curves with CNT coated surfaces as shown in Eqs. (7.2) - (7.6) were proposed. The correlations agreed reasonably well with all the surfaces that have been investigated. However, it can also be observed from Figs. 6.3 and 6.4 that the majority of the experimental data lies between 0 and 10 of $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$, representing the uniformly coated CNT surfaces, and this is followed by the interlaced CNT surfaces with $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ in the 50s range. Hence, further investigations of different CNT coating configurations with $\frac{L^2 + P_{CNT}^2}{L^2 + A_{CNT}}$ between 10 and 40 can be conducted to validate the accuracy of the correlation in this range.

2. Subcooled pool boiling and CNT coated surfaces

Pool boiling under subcooled conditions has been widely investigated as boiling performed in a highly subcooled fluid would exhibit high temperature gradient between the bulk fluid and boiling surface causing a negative temperature gradient of surface tension, i.e. $-\frac{\partial\sigma}{\partial T}$. This resulted in convective fluid flow from the warmer boiling surface to the cooler bulk fluid, also known as thermocapillary convection or Marangoni effect, which might further augment heat transfer. It had been shown by Petrovic et al. [96] and Marek and Straub [97] that depending on the Marangoni number (Ma) and the degree of subcooling, the Marangoni effect on the heat transfer process varies. Similarly, subcooled boiling investigation can also be extended to include CNT coated surfaces as the presence of the coating could have introduced additional complexity such as capillary induced heat transfer to the boiling process.

3. Numerical simulation of pool boiling with CNT coated surfaces

The correlations developed in this thesis can provide reasonably accurate predictions of saturated pool boiling with CNT surfaces and can be employed for different CNT coating areas and surface orientations. Future research can focus on the development of numerical models from the fundamental approach such as employing the Navier-Stokes and energy equations for macroscopic length-time scale modelling or the Lattice Boltzmann equation for mesoscopic length-time scale modelling to evaluate the pool boiling performance of CNT surfaces. In the simulation of single bubble growth using Lattice Boltzmann Method (LBM) for instance, Gong and Cheng [101] chose a 150×400 lattice structure for the simulation with the lattice spacing and time spacing of 1.0. Their results indicated good agreement with the analytical model on the bubble departure diameter proposed by Fritz [26]. Simulation of full scale

nucleation boiling, however, requires the computational scheme to be further refined such that the computational time and cost can be reduced for practical application.

4. Fabrication of boiling heat transfer media with additive manufacturing techniques

Advancements in additive manufacturing or 3D printing technology in the recent years have revolutionised the scientific and engineering field. This manufacturing technique is capable of fabricating complex models. Optimisation of surfaces for nucleate boiling heat transfer requires the surface nucleation sites to be maximised. Additive manufacturing provides vast opportunities for thermal engineers to develop optimal heat transfer surfaces. However, research on enhancing heat transfer surfaces with additive manufacturing technique is still very much in its infancy with few publications reported [98, 99, 100]. Further research in this field is necessary to provide a deeper understanding of its potential and applicability in addressing existing challenges in thermal sciences and engineering.

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APPENDIX A

THERMOSYPHON 1 HEATER SECTION SIMULATION FORMULATION

A.1 Copper block

The following assumptions are made in the development of the thermal simulation for the copper block:

1. Steady state heat conduction, $\frac{\partial}{\partial t_c} = 0$
2. Due to the square geometry and homogeneity of the heater material, the temperature distribution in the x - and z -axes are assumed to be similar hence reducing to a 2D heat conduction problem, with temperature depending on only in the x - and y -directions, $\frac{\partial}{\partial z_c} = 0$
3. Axisymmetric about the y -axis
4. No convection heat transfer involved, $v \cdot \nabla T = 0$
5. No heat source, $q = 0$ (heat generated by the rod heater is taken care of at the boundary conditions)
6. Homogenous material, constant material and thermal properties in all directions
7. Heat flux is conserved across the copper-Teflon interface

With the above assumptions, the energy equation is simplified as follows:

$$\frac{\partial^2 T_c}{\partial x^2} + \frac{\partial^2 T_c}{\partial y^2} = 0 \quad (A1)$$

Boundary conditions:

x -direction,

1. At $x = 0$, $\frac{\partial T_c}{\partial x} = 0$ (Axisymmetric)
2. At $x = x_1$, $q'' = -k_c \frac{\partial T_c}{\partial x}$ (Heat transfer by the rod heater at the rod heater-copper block interface)
3. At $x = x_2$, $-k_c \frac{\partial T_c}{\partial x} = k_T \frac{\partial T_T}{\partial x}$ (Conservation of heat flux across copper block-Teflon interfaces)
4. At $x = x_3$, $-k_c \frac{\partial T_c}{\partial x} = k_T \frac{\partial T_T}{\partial x}$ (Conservation of heat flux across copper block-Teflon interfaces)

y -direction,

1. At $y = y_1$, $-k_c \frac{\partial T_c}{\partial y} = k_T \frac{\partial T_T}{\partial y}$ (Conservation of heat flux across copper block-Teflon interfaces)
2. At $y = y_2$, $-k_c \frac{\partial T_c}{\partial y} = k_T \frac{\partial T_T}{\partial y}$ (Conservation of heat flux across copper block-Teflon interfaces)
3. At $y = y_4$, $q'' = -k_c \frac{\partial T_c}{\partial y}$ (Heat transfer by the rod heater at the rod heater-copper block interface)
4. At $y = y_5$, $q'' = h(T_y - T_{f,sat})$ (Heat transfer from the boiling surface to the surrounding bulk fluid)

A.2 Teflon insulation

The following assumptions are made in the development of the thermal simulation for the Teflon insulation:

1. Steady state heat conduction, $\frac{\partial}{\partial t_T} = 0$
2. Due to the square geometry and homogeneity of the heater material, the temperature distribution in the x - and z -axes are assumed to be similar hence reducing to a 2D heat conduction problem, with temperature depending on only in the x - and y -directions, $\frac{\partial}{\partial z_T} = 0$
3. Axisymmetric about the y -axis
4. No convection heat transfer involved, $v \cdot \nabla T = 0$
5. No heat source, $q = 0$
6. Homogenous material, constant material and thermal properties in all directions
7. Heat flux is conserved across the copper-Teflon interface

Governing equation:

$$\frac{\partial^2 T_T}{\partial x^2} + \frac{\partial^2 T_T}{\partial y^2} = 0 \quad (A2)$$

Boundary conditions:

x -direction,

1. At $x = 0$, $\frac{\partial T_T}{\partial x} = 0$ (Axisymmetric)

2. At $x = x_4$, $-k_T \frac{\partial T_T}{\partial x} = h_{a,nc}(T_x - T_a)$ (Heat transfer to ambient air by natural convection)
3. At $x = x_5$, $-k_T \frac{\partial T_T}{\partial x} = h_{a,nc}(T_x - T_a)$ (Heat transfer to ambient air by natural convection)

y-direction,

1. At $y = 0$, $-k_T \frac{\partial T_T}{\partial y} = h_{a,nc}(T_y - T_a)$ (Heat transfer to ambient air by natural convection)
2. At $y = y_4$, $-k_T \frac{\partial T_T}{\partial y} = h_{a,nc}(T_y - T_{a,sat})$ (Heat transfer to ambient air by natural convection)
3. At $y = y_5$, $-k_T \frac{\partial T_T}{\partial y} = h_{f,nc}(T_y - T_{f,sat})$ (Heat transfer to bulk fluid by natural convection)

Note: The heat transfer coefficient for natural convection is obtained from the following correlation:

$$h = \frac{k}{L} \left(0.68 + \frac{0.76 \text{Ra}_L^{\frac{1}{4}}}{\left(1 + \left(\frac{0.492k}{\mu C_p} \right)^{\frac{9}{16}} \right)^{\frac{4}{9}}} \right), \text{Ra}_L \leq 10^9$$

$$h = \frac{k}{L} \left(0.825 + \frac{0.387 \text{Ra}_L^{\frac{1}{6}}}{\left(1 + \left(\frac{0.492k}{\mu C_p} \right)^{\frac{9}{16}} \right)^{\frac{8}{27}}} \right), \text{Ra}_L > 10^9$$

APPENDIX B

THERMOSYPHON 2 HEATER SECTION SIMULATION FORMULATION

B.1 Copper block

The following assumptions are made in the development of the thermal simulation for the copper block:

1. Steady state heat conduction, $\frac{\partial}{\partial t_c} = 0$
2. Due to the square geometry and homogeneity of the heater material, the temperature distribution in the x - and z -axes are assumed to be similar hence reducing to a 2D heat conduction problem, with temperature depending on only in the x - and y -directions, $\frac{\partial}{\partial z_c} = 0$
3. Axisymmetric along the y -axis
4. No convection heat transfer involved, $v \cdot \nabla T = 0$
5. No heat source, $q = 0$ (heat generated by the rod heater is taken care of at the boundary conditions)
6. Homogenous material, constant material and thermal properties in all directions
7. Heat flux is conserved across the copper-Teflon interface

With the above assumptions, the energy equation is simplified as follows:

$$\frac{\partial^2 T_c}{\partial x^2} + \frac{\partial^2 T_c}{\partial y^2} = 0 \quad (B1)$$

Boundary conditions:

x -direction,

1. At $x = 0$, $\frac{\partial T_c}{\partial x} = 0$ (Axisymmetric)
2. At $x = x_1$, $q'' = -k_c \frac{\partial T_c}{\partial x}$ (Heat transfer by the rod heater at the rod heater-copper block interface)
3. At $x = x_2$, $-k_c \frac{\partial T_c}{\partial x} = k_T \frac{\partial T_T}{\partial x}$ (Conservation of heat flux across copper block-Teflon interfaces)

y -direction,

1. At $y = y_1$, $-k_c \frac{\partial T_c}{\partial y} = k_T \frac{\partial T_T}{\partial y}$ (Conservation of heat flux across copper block-Teflon interfaces)
2. At $y = y_2$, $q'' = -k_c \frac{\partial T_c}{\partial y}$ (Heat transfer by the rod heater at the rod heater-copper block interface)
3. At $y = y_3$, $q'' = h(T_y - T_{f,sat})$ (Heat transfer from the boiling surface to the surrounding bulk fluid)

B.2 Teflon insulation

The following assumptions are made in the development of the thermal simulation for the Teflon insulation:

1. Steady state heat conduction, $\frac{\partial}{\partial t_T} = 0$
2. Due to the square geometry and homogeneity of the heater material, the temperature distribution in the x - and z -axes are assumed to be similar hence reducing to a 2D heat conduction problem, with temperature depending on only in the x - and y -direction, $\frac{\partial}{\partial z_T} = 0$
3. Axisymmetric along the y -axis
4. No convection heat transfer involved, $v \cdot \nabla T = 0$
5. No heat source, $q = 0$
6. Homogenous material, constant material and thermal properties in all directions
7. Heat flux is conserved across the copper-Teflon interface

Governing equation:

$$\frac{\partial^2 T_T}{\partial x^2} + \frac{\partial^2 T_T}{\partial y^2} = 0 \quad (B2)$$

Boundary conditions:

x -direction,

1. At $x = 0$, $\frac{\partial T_T}{\partial x} = 0$ (Axisymmetric)

2. At $x = x_3$, $-k_T \frac{\partial T_T}{\partial x} = h_{f,nc}(T_x - T_{f,sat})$ (Heat transfer to bulk fluid by natural convection)

y-direction,

1. At $y = 0$, $-k_T \frac{\partial T_T}{\partial y} = h_{f,nc}(T_y - T_{f,sat})$ (Heat transfer to bulk fluid by natural convection)
2. At $y = y_3$, $-k_T \frac{\partial T_T}{\partial y} = h_{f,nc}(T_y - T_{f,sat})$ (Heat transfer to bulk fluid by natural convection)

Note: The heat transfer coefficient for natural convection is obtained from the following correlation:

$$h = \frac{k}{L} \left(0.68 + \frac{0.76 Ra_L^{\frac{1}{4}}}{\left(1 + \left(\frac{0.492k}{\mu C_p} \right)^{\frac{9}{16}} \right)^{\frac{4}{9}}} \right), Ra_L \leq 10^9$$

$$h = \frac{k}{L} \left(0.825 + \frac{0.387 Ra_L^{\frac{1}{5}}}{\left(1 + \left(\frac{0.492k}{\mu C_p} \right)^{\frac{9}{16}} \right)^{\frac{8}{27}}} \right), Ra_L > 10^9$$

APPENDIX C

PROPERTIES OF FLUORINERT™ FC-72 [C1]

Physical Properties

Not for specification purposes

All values determined at 25°C unless otherwise specified

Properties	FC-72
Appearance	Clear, colorless
Average Molecular Weight	338
Boiling Point (1 atm)	56°C
Pour Point	-90°C
Estimated Critical Temperature	449 K
Estimated Critical Pressure	1.83 x 10 ⁶ pascals
Vapor Pressure	30.9 x 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 °C ⁻¹
Coefficient of Expansion	0.00156 °C ⁻¹
Surface Tension	10 dynes/cm
Refractive Index	1.251
Water Solubility	10 ppmw
Solubility in Water	<5 ppmw
Ozone Depletion Potential	0

3M™ Fluorinert™ Electronic Liquid FC-72 Electrical Properties

Properties	FC-72
Dielectric Strength	38 kV, 0.1" gap
Dielectric Constant	1.75
Electrical Resistivity	1.0 x 10 ¹⁵ ohm cm

[C1] Fluorinert Electronic Liquid FC-72 product information, www.3m.com/fluids, 3M Corporation.

APPENDIX D

DERIVATION OF BUBBLE DEPARTURE RADII AT BOILING INCIPIENCE

The purpose of this section is to derive the bubble departure radii, r_b , at boiling incipience for the first batch of bare silicon and 90% uniformly coated CNT surface such that values can be used to determine the thermal boundary layer thickness as mentioned in Section 4.2.2. Experiments with these surface were conducted in Thermosyphon 1 but due to the limitation of this setup, clear images of the boiling phenomenon could not be captured and hence ,photographic measurement of the bubble sizes was not performed. Hence, existing correlations were used to compute r_b with the aid of collected experimental data.

The bubble departure radius, r_b , can be expressed in term of the Boiling number, Bo as shown in Eq. (D.1) where Bo is in turn related to Ja as shown in Eqs. (D.2) and (D.3). Combining these equations, the final expression of r_b in terms of the fluid thermophysical properties and wall superheats is shown in Eq. (D.4). By substituting the physical properties of FC-72 as shown in Appendix C, and the wall superheats of 12.2°C and 8°C for bare silicon and 90% CNT coated surface respectively, r_b was computed to be 0.264 mm and 0.173 mm, respectively for bare silicon and 90% CNT coated surface.

$$r_b = 0.5 \sqrt{\frac{Bo\sigma}{g(\rho_l - \rho_v)}} \quad (D.1)$$

$$Bo = (0.04Ja)^2 \quad (D.2)$$

$$Ja = \frac{\rho_l C_{pl} \Delta T_{sat}}{\rho_v h_{fg}} \quad (D.3)$$

$$r_b = \frac{0.02 \rho_l C_{pl} \Delta T_{sat}}{\rho_v h_{fg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \quad (D.4)$$

In order to assess the accuracy of Eq. (D.4) in predicting the bubble departure radius, Eq. (D.4) was similarly used to compute the bubble departure radius for the second batch of bare silicon and the 100% CNT coated surface that was experimented in Thermosyphon 2. Fortunately, this setup allowed clearly images of the bubble departure phenomenon to be captured and the computed values could be compared against the measured bubble size from the photographs. The incipience wall superheats for the bare silicon and 100% CNT coated surface was recorded at 17.7°C and 6.8°C, respectively. Based on these wall superheats, the computed bubble departure radii were 0.383 mm and 0.147 mm for bare silicon and 100% CNT, respectively. The photographs of the boiling phenomenon for bare silicon and 100% CNT are shown in Fig. D.1. Samples of 11 to 16 bubbles were measured using J-image (a commercially available software) for each surface and the average bubble radii of 0.315 mm and 0.165 mm for bare silicon and 100% CNT coated surface were determined from the measurement. Based on the calculated and measured values of the bubble sizes, the maximum discrepancy was approximately 18%.

From this assessment, Eq. (D.4) has shown to provide relatively accurate prediction of the bubble departure radius for both bare silicon and CNT coated surface and the computed bubble radii for the first batch of bare silicon and 90% coated CNT surface could be used.

(a)

(b)

Fig. D.1 Photographs of nucleate boiling phenomenon of (a) bare silicon (b) 100% CNT coating at boiling incipience.

APPENDIX E

AVERAGE HEAT TRANSFER CORRELATION – MATLAB PSEUDOCODE

```

% Averaged heat transfer coefficient – Example: Bare Silicon%
% Insert variable%
>> syms q a b c d x
% Define equation%
>> s = (q - (a + b*x + c*x^2 + d*x^3))^2;
% Differentiate with respect to variable%
>> A = diff(s,a);
>> B = diff(s,b);
>> C = diff(s,c);
>> D = diff(s,d);
% Substituting experimental data%
>> AA = subs(A, q, 0.243);
>> AAA = subs(AA, x, 0);
>> AA2 = subs(A, q, 0.250);
>> AAA2 = subs(AA2, x, 30);
>> AA3 = subs(A,q, 0.255);
>> AAA3 = subs(AA3,x,60);
>> AA4 = subs(A,q, 0.251);
>> AAA4 =subs(AA4,x,90);
>> AA5 = subs(A,q, 0.233);
>> AAA5 = subs(AA5,x,120);
>> AA6 = subs(A,q, 0.227);
>> AAA6 = subs(AA6,x,150);
>> AA7 = subs(A,q, 0.119);
>> AAA7 = subs(AA7,x,180);
>> AAAA = AAA + AAA2 + AAA3 + AAA4 + AAA5 + AAA6 + AAA7;
>> BB = subs(B, q, 0.243);
>> BBB = subs(BB, x, 0);
>> BB2 = subs(B, q, 0.250);
>> BBB2 = subs(BB2, x, 30);
>> BB3 = subs(B,q, 0.255);
>> BBB3 = subs(BB3,x,60);
>> BB4 = subs(B,q, 0.251);
>> BBB4 =subs(BB4,x,90);
>> BB5 = subs(B,q, 0.233);
>> BBB5 = subs(BB5,x,120);
>> BB6 = subs(B,q, 0.227);
>> BBB6 = subs(BB6,x,150);
>> BB7 = subs(B,q, 0.119);
>> BBB7 = subs(BB7,x,180);
>> BBBB = BBB+BBB2+BBB3+BBB4+BBB5+BBB6+BBB7;
>> CC = subs(C, q, 0.243);
>> CCC = subs(CC, x, 0);
>> CC2 = subs(C, q, 0.250);

```



```

>> CCC2 = subs(CC2, x, 30);
>> CC3 = subs(C,q, 0.255);
>> CCC3 = subs(CC3,x,60);
>> CC4 = subs(C,q, 0.251);
>> CCC4 =subs(CC4,x,90);
>> CC5 = subs(C,q, 0.233);
>> CCC5 = subs(CC5,x,120);
>> CC6 = subs(C,q, 0.227);
>> CCC6 = subs(CC6,x,150);
>> CC7 = subs(C,q, 0.119);
>> CCC7 = subs(CC7,x,180);
>> CCCC = CCC+CCC2+CCC3+CCC4+CCC5+CCC6+CCC7;
>> DD = subs(D, q, 0.243);
>> DDD = subs(DD, x, 0);
>> DD2 = subs(D, q, 0.250);
>> DDD2 = subs(DD2, x, 30);
>> DD3 = subs(D,q, 0.255);
>> DDD3 = subs(DD3,x,60);
>> DD4 = subs(D,q, 0.251);
>> DDD4 =subs(DD4,x,90);
>> DD5 = subs(D,q, 0.233);
>> DDD5 = subs(DD5,x,120);
>> DD6 = subs(D,q, 0.227);
>> DDD6 = subs(DD6,x,150);
>> DD7 = subs(D,q, 0.119);
>> DDD7 = subs(DD7,x,180);
>> DDDD = DDD+DDD2+DDD3+DDD4+DDD5+DDD6+DDD7;
% Converting equations to matrix%
>> [M, N] = equationsToMatrix([AAAA, BBBB, CCCC, DDDD], [a, b, c, d]);
% Obtaining value of a, b, c and d%
>> v = MN;
% Converting fraction to decimal%
>> v1 = vpa(v)

```

APPENDIX F

CRITICAL HEAT FLUX CORRELATION – MATLAB PSEUDOCODE

```

%CHF correlation – Example: Bare Silicon%
% Insert variable%
>> syms q a b c d e x
% Define equation%
>> s = (q - (a + b*x + c*x^2 + d*x^3)*e)^2;
% Differentiate with respect to variable%
>> A = diff(s,a);
>> B = diff(s,b);
>> C = diff(s,c);
>> D = diff(s,d);
% Substituting experimental data%
>> AA = subs(A, q, 14.1);
>> AAA = subs(AA, x, 0);
>> AA2 = subs(A, q, 14.1);
>> AAA2 = subs(AA2, x, 30);
>> AA3 = subs(A,q, 13.5);
>> AAA3 = subs(AA3,x,60);
>> AA4 = subs(A,q, 12.8);
>> AAA4 =subs(AA4,x,90);
>> AA5 = subs(A,q, 11.8);
>> AAA5 = subs(AA5,x,120);
>> AA6 = subs(A,q, 10.3);
>> AAA6 = subs(AA6,x,150);
>> AA7 = subs(A,q, 6.1);
>> AAA7 = subs(AA7,x,180);
>> AAAA = AAA + AAA2 + AAA3 + AAA4 + AAA5 + AAA6 + AAA7;
>> BB = subs(B, q, 14.1);
>> BBB = subs(BB, x, 0);
>> BB2 = subs(B, q, 14.1);
>> BBB2 = subs(BB2, x, 30);
>> BB3 = subs(B,q, 13.5);
>> BBB3 = subs(BB3,x,60);
>> BB4 = subs(B,q, 12.8);
>> BBB4 =subs(BB4,x,90);
>> BB5 = subs(B,q, 11.8);
>> BBB5 = subs(BB5,x,120);
>> BB6 = subs(B,q, 10.3);
>> BBB6 = subs(BB6,x,150);
>> BB7 = subs(B,q, 6.1);
>> BBB7 = subs(BB7,x,180);
>> BBBB = BBB+BBB2+BBB3+BBB4+BBB5+BBB6+BBB7;
>> CC = subs(C, q, 14.1);
>> CCC = subs(CC, x, 0);
>> CC2 = subs(C, q, 14.1);
>> CCC2 = subs(CC2, x, 30);
>> CC3 = subs(C,q, 13.5);
>> CCC3 = subs(CC3,x,60);
>> CC4 = subs(C,q, 12.8);
>> CCC4 =subs(CC4,x,90);
>> CC5 = subs(C,q, 11.8);
>> CCC5 = subs(CC5,x,120);

```

```

>> CC6 = subs(C,q, 10.3);
>> CCC6 = subs(CC6,x,150);
>> CC7 = subs(C,q, 6.1);
>> CCC7 = subs(CC7,x,180);
>> CCCC = CCC+CCC2+CCC3+CCC4+CCC5+CCC6+CCC7;
>> DD = subs(D, q, 14.1);
>> DDD = subs(DD, x, 0);
>> DD2 = subs(D, q, 14.1);
>> DDD2 = subs(DD2, x, 30);
>> DD3 = subs(D,q, 13.5);
>> DDD3 = subs(DD3,x,60);
>> DD4 = subs(D,q, 12.8);
>> DDD4 =subs(DD4,x,90);
>> DD5 = subs(D,q, 11.8);
>> DDD5 = subs(DD5,x,120);
>> DD6 = subs(D,q, 10.3);
>> DDD6 = subs(DD6,x,150);
>> DD7 = subs(D,q, 6.1);
>> DDD7 = subs(DD7,x,180);
>> DDDD = DDD+DDD2+DDD3+DDD4+DDD5+DDD6+DDD7;
% Substituting fluid properties %
>> AAAA1 = subs(AAAA, e, 14.1);
>> BBBB1 = subs(BBBB, e, 14.1);
>> CCCC1 = subs(CCCC, e, 14.1);
>> DDDD1 = subs(DDDD, e, 14.1);
% Converting equations to matrix%
>> [M, N] = equationsToMatrix([AAAA1, BBBB1, CCCC1, DDDD1], [a, b, c, d]);
% Obtaining value of a, b, c and d%
>> v = M\N;
% Converting fraction to decimal%
>> v1 = vpa(v)

```

APPENDIX G

SATURATED POOL BOILING CURVE CORRELATION – MATLAB PSEUDOCODE

```
%Bare silicon (batch 2) – 0degree surface orientation%
>> syms a T C b q r c
>> a = 1101/88000;
>> b = ((0.793*10^-2/(9.81*(1602.2-13.21)))^0.5)/(4.33*10^-4*88000);
>> c = (1101*4.33*10^-4/0.054)^3.5;
>> q = ((a/c*T/C)^(1/r))/(b*10000);
>> q1 = diff(q,C);
>> q2 = diff(q,r);
>> C0 = 0.0001;
>> r0 = 0.1;
>> n = 0;
>> A0 = [C0; r0];
>> for i=1:50
>> q11 = subs(q1, [T, C, r], [6.7, C0, r0]);
>> q12 = subs(q1, [T, C, r], [13.7, C0, r0]);
>> q13 = subs(q1, [T, C, r], [17.6, C0, r0]);
>> q14 = subs(q1, [T, C, r], [20.6, C0, r0]);
>> q15 = subs(q1, [T, C, r], [23.1, C0, r0]);
>> q16 = subs(q1, [T, C, r], [25.8, C0, r0]);
>> q17 = subs(q1, [T, C, r], [29.2, C0, r0]);
>> q18 = subs(q1, [T, C, r], [32.9, C0, r0]);
>> q19 = subs(q1, [T, C, r], [36.4, C0, r0]);
>> q110 = subs(q1, [T, C, r], [39.2, C0, r0]);
>> q111 = subs(q1, [T, C, r], [39.2, C0, r0]);
>> q112 = subs(q1, [T, C, r], [42.2, C0, r0]);
>> q113 = subs(q1, [T, C, r], [44.6, C0, r0]);
>> q114 = subs(q1, [T, C, r], [52.7, C0, r0]);
>> q21 = subs(q2, [T, C, r], [6.7, C0, r0]);
>> q22 = subs(q2, [T, C, r], [13.7, C0, r0]);
>> q23 = subs(q2, [T, C, r], [17.6, C0, r0]);
>> q24 = subs(q2, [T, C, r], [20.6, C0, r0]);
>> q25 = subs(q2, [T, C, r], [23.1, C0, r0]);
>> q26 = subs(q2, [T, C, r], [25.8, C0, r0]);
>> q27 = subs(q2, [T, C, r], [29.2, C0, r0]);
>> q28 = subs(q2, [T, C, r], [32.9, C0, r0]);
>> q29 = subs(q2, [T, C, r], [36.4, C0, r0]);
>> q210 = subs(q2, [T, C, r], [39.2, C0, r0]);
>> q211 = subs(q2, [T, C, r], [39.2, C0, r0]);
>> q212 = subs(q2, [T, C, r], [42.2, C0, r0]);
>> q213 = subs(q2, [T, C, r], [44.6, C0, r0]);
>> q214 = subs(q2, [T, C, r], [52.7, C0, r0]);
>> z = [q11, q21; q12, q22; q13, q23; q14, q24; q15, q25; q16, q26; q17, q27; q18, q28; q19, q29;
q110, q210; q111, q211; q112, q212; q113, q213; q114, q214];
>> qq1 = subs(q, [T, C, r], [6.7, C0, r0]);
>> qq2 = subs(q, [T, C, r], [13.7, C0, r0]);
>> qq3 = subs(q, [T, C, r], [17.6, C0, r0]);
>> qq4 = subs(q, [T, C, r], [20.6, C0, r0]);
>> qq5 = subs(q, [T, C, r], [23.1, C0, r0]);
>> qq6 = subs(q, [T, C, r], [25.8, C0, r0]);
>> qq7 = subs(q, [T, C, r], [29.2, C0, r0]);
>> qq8 = subs(q, [T, C, r], [32.9, C0, r0]);
```

```

>> qq9 = subs (q, [T, C, r], [36.4, C0, r0]);
>> qq10 = subs (q, [T, C, r], [39.2, C0, r0]);
>> qq11 = subs (q, [T, C, r], [39.2, C0, r0]);
>> qq12 = subs (q, [T, C, r], [42.2, C0, r0]);
>> qq13 = subs (q, [T, C, r], [44.6, C0, r0]);
>> qq14 = subs (q, [T, C, r], [52.7, C0, r0]);
>> D = [1.4-qq1; 2.4-qq2; 3.1-qq3; 4.2-qq4; 4.9-qq5; 6.1-qq6; 7-qq7; 7.9-qq8; 8.9-qq9; 9.8-qq10;
10.8-qq11; 11.8-qq12; 13-qq13; 14-qq14];
>> A = z\D;
>> A0 = A0 + A;
>> C0 = A0(1,1)
>> r0 = A0(2,1)
>> n = n+1
>> A0
>> end
sprintf('%0.6f', A0)

```