EBULLIENT COOLING OF ELECTRONICS: FUTURE TRENDS AND RECENT ADVANCES

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ABSTRACT

Semiconductor chip heat fluxes are again rising rapidly and threatening to overwhelm available thermal management techniques. The use of direct pool boiling heat transfer from the chips to dielectric liquids is a primary future alternative, but is limited by the relatively low pool boiling CHF of most candidate liquids. This paper summarizes the progress made during the past two decades towards a refined understanding and improved predictive accuracy of the baseline pool boiling critical heat flux of dielectric coolants and discusses, as well, opportunities for and progress towards passive enhancement of this key boiling metric.

NOMENCLATURE

a	Constant in Equation (12)	L	Length of the heater [m]
b	Constant in Equation (12)	L'	Non-dimensional characteristic length
B	Subcooling constant	P	Pressure [Pa]
c_p	Specific heat [$Jkg^{-1}K^{-1}$]	Q	Heat flux [Wm^{-2}]
CHF	Critical heat flux [Wm-^{2}]	r	Radial coordinate [m]
g	Gravitational acceleration [ms-^{2}]	R	Radius of the heater [m]
h	Heat transfer coefficient [Wm-^{2}K^{-1}]	S	Thermal activity parameter
h_{fv}	Latent heat of evaporation [Jkg^{-1}]	q"	Heat flux [Wm^{-2}]
Ja	Jacob number	t	Time [s]
k	Thermal conductivity [$Wm^{-1}K^{-1}$]	T	Temperature [°C]

z	axial coordinate [m]

Greek Symbols

α	Proportional
δ	Thickness [m]
λ_D	Taylor wavelength [m]
ρ	Density [kgm^{-3}]
σ	Surface tension [Nm^{-1}]
τ_D	Hovering period [s]
ΔT	Temperature difference [K]
μ	Dynamic viscosity [N s/m^2]

Subscripts

boil	Boiling
d	Dry spot
ex	Experimental
f	Liquid
gen	Generation
h	Heater
ini	Initial
m	Measured
pre	Predicted
rad	Radial
sat	Saturation
sub	Subcooling
sur	Surface
TME	Thermal management of electronics
v	Vapor

INTRODUCTION

The reliable operation of electronic equipment often requires that device temperatures be maintained below 95 °C and that the maximum temperature variation be small and predictable (Bar-Cohen, 2000). The temperature of the chip in which these devices are embedded is a function of two factors: the dissipated heat flux and the overall heat transfer coefficient afforded by the cooling mechanism. Despite the precipitous drop in transistor switching energy that has characterized the solid-state semiconductor revolution, the cooling requirements of microelectronic components have not diminished. As the 21st century begins, high performance chip power dissipation exceeds 100 W, some three orders-of-magnitude above the chips of the early 1960's, and informed opinion suggests that a 250 W chip could become a reality within the present decade. Thermal management is thus one of the key challenges in advanced electronic packaging and considerable improvement in thermal packaging will be needed to successfully exploit the continued "Moore's Law" acceleration in semiconductor technology (NEMI 2002).

In the late 1990's, under the influence of market forces, thermal management of nearly all categories of electronic equipment devolved to air cooling, relying on passive techniques - for PDA's and commercial electronics, active cooling - for notebook and desk-top computers, as well as telecommunication modules, and very aggressive use of high performance heat sinks – for the cooling of high end servers, routers, and supercomputer nodes. The packaging community consensus (NEMI, 2002) suggests that, early in the second decade of this century, power dissipation will rise to 150 W for chips powering desktop computers and close to 300W for chips used in the highest performance computers. While projected increases in chip size, to approximately 25mm on a side, are expected to keep the chip heat flux around 30W/cm^2 in the desk-top (or cost/performance) category, the fastest microprocessors, operating at some 3 GHz in the "high-performance" category, are anticipated to display average heat fluxes above 50W/cm^2 and peak values well in excess of 100W/cm^2.

Table 1: Thermophysical properties of dielectric liquids at atmospheric pressure (3M Specialty Fluids)

Property	FC-40	FC-72	HFE-7100	HFE-7200	Water
T_{sat} (°C)	156.0	56.0	61.0	76.0	100.0
ρ_f (kg/m^3)	1870.0	1623.0	1500.0	1430.0	957.8
ρ_v (kg/m^3)	25.0	12.7	9.6	9.26	0.5956
μ_f (mN m/s^2)	3.54	0.457	0.61	0.61	0.279
C_{pf} (J/kg K)	-	1097.8	1180.0	1210.0	4217.0
k_f (W/mK)	-	0.052	-	-	0.68
h_{fv} (kJ/kg)	711.6	84.97	125.6	122.6	2257.0
σ_f (N/m)	0.016	0.0084	0.014	0.014	0.0589
P_{cr} (kPa)	1176.0	1840.0	-	-	22100.0

Following Bergles and Bar-Cohen (1990), it may be argued that cost-effective thermal management of such large silicon chips with power dissipation between 150W and 300W and highly non-uniform heat fluxes, requires exploration of non-conventional thermal packaging techniques. Prime among these is the use of dielectric liquids, in direct contact with the chip surface, to remove the dissipated power, through convection, boiling, or evaporation. Such coolants must be non-toxic, chemically inert to the semiconductor and packaging materials, and possess high dielectric strengths as well as low dielectric constants. The perfluorocarbons (FC's) and hydro-fluoro-ethers (HFE's) made by 3M corporation provide this mix of properties, along with very low wetting angles on most engineering surfaces, and relatively low critical pressures, thermal conductivities, and specific heats, but air solubilities, approaching 50% by volume, some 25 times higher than in water. The fluorocarbon liquid FC-72 (T_{sat} = 56 °C at 101.3 kPa) has been successfully used for single-phase forced convection cooling of chips in the Cray-2 supercomputer (Danielson et al., 1987) and for evaporative spray cooling of the Cray SV-2 module (Pautsch and Bar-Cohen, 1999), while FC-77 (T_{sat} = 97 °C at 101.3 kPa) has been used for jet impingement cooling of chips dissipating nearly 90 W/cm^2 in the SS-1 supercomputer (Ing et al., 1993), and FC-72.

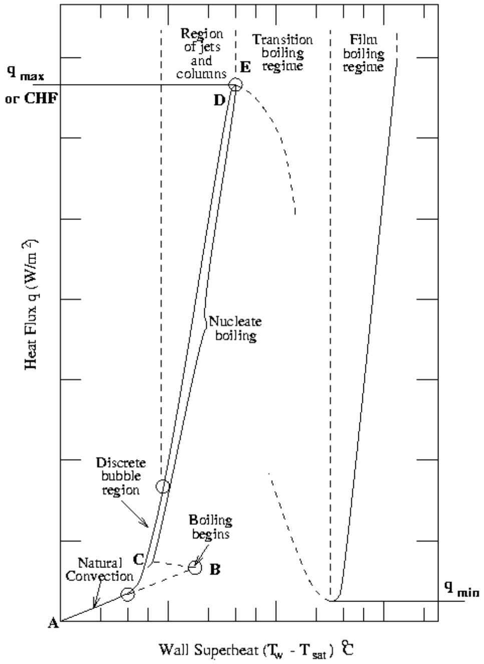

Figure 1: A typical pool boiling curve

Due to the relatively poor thermal properties of these fluorocarbons and hydroflouroethers, as compared to water (see Table 1), extremely large fluid flow velocities (≈ 5-10 m/s) are necessary to generate single-phase heat transfer coefficients sufficient to cool the high flux chips (Jiji and Dagan, 1987). Alternatively, “passive” immersion cooling, with nucleate pool boiling on the chip surface, can provide heat transfer coefficients of approximately 10^4 W/m^2 K at surface superheats between 10 and 20 °C, and eliminate the need for the expensive pumping power and hardware associated with single-phase cooling designs. However, as shown in Figure 1, the departure from nucleate boiling, or “Critical Heat Flux” (CHF), places an upper limit on this highly efficient heat transfer mechanism. The CHF condition is associated with the formation of an insulating film of vapor on the heated surface, leading to a large increase in the surface temperature. The CHF value for pool boiling of a fluorocarbon like FC-72, on a large, horizontal surface under saturation conditions, is approximately 17 W/cm^2 at 101.3 kPa (Watwe, 1996), in sharp contrast to the pool boiling CHF for Water, which is approximately 120 W/cm^2 at 101.3 kPa.

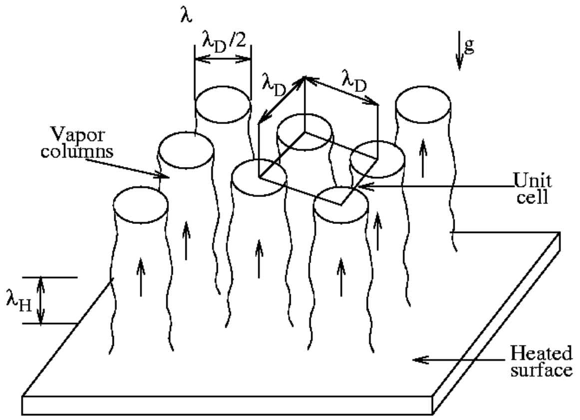

Figure 2: Vapor column configuration (Zuber, 1959)

Thus, despite the promise of direct liquid cooling with these dielectric fluids, a significant increase in the pool boiling CHF is necessary to accommodate the anticipated average chip dissipation heat flux of more than 50 W/cm^2 with peak values well above 100 W/cm^2. This paper summarizes the progress during the past two decades towards a refined understanding, improved predictive accuracy, and exploitation of opportunities for passive enhancement of the pool boiling critical heat flux of dielectric coolants.

CRITICAL HEAT FLUX MODELS

The earliest theoretical models for CHF are attributed to Kutateladze (1951) and Zuber (1959). Noting the similarity between "flooding" in distillation columns and the critical heat flux condition, Kutateladze (1951) obtained an expression for CHF based on a similitude analysis of the momentum and energy equations governing the two-phase flow near the heated surface. Subsequently, Zuber (1959) derived an analytical equation for CHF by assuming that it was hydrodynamically controlled. He postulated that vertical coalescence of vapor bubbles in the fully developed nucleate boiling regime resulted in the formation of cylindrical vapor jets or columns. The upward flowing vapor jets were separated by counter-flowing columns of liquid (see Figure 2). The Taylor instability mechanism observed in the film-boiling regime was assumed to apply at CHF, giving vapor column spacing equal to the most dangerous Taylor wavelength λ_D. The vapor jet diameter was arbitrarily chosen as $\lambda_D/2$. CHF was assumed to occur when the interface between the counter-flowing liquid and vapor streams became unstable and completely choked off the flow of liquid to the surface. Zuber (1959) used Rayleigh's stability analysis to calculate the "critical" vapor velocity at which the instability occurred. This analysis yielded the following relation for CHF:

$$q_{CHF} = \frac{\pi}{24} h_{fv} \sqrt{\rho_v} \left[g\, \sigma_f \left(\rho_f - \rho_v \right) \right]^{1/4} \tag{1}$$

Equation (1) was derived for a specific configuration, namely: saturated pool boiling of a liquid on an infinite, upward-facing horizontal plate. Despite these limitations, investigators have found it possible to use Equation (1) in a wide variety of situations, and have introduced correction factors to account for the effects of specific parameters, such as subcooling and heater geometry. Several investigators have questioned the vapor column configuration assumed by Zuber (1959), due to lack of photographic evidence on flat horizontal heaters.

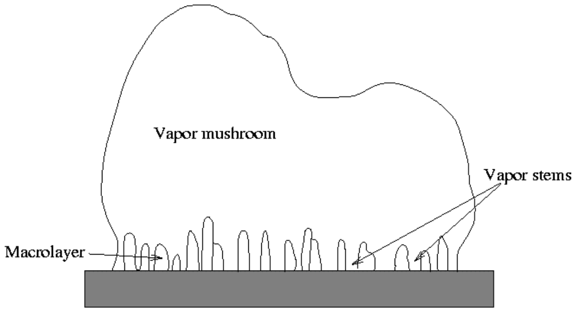

Figure 3: Vapor mushroom configuration (Haramura and Katto, 1983)

Photographic studies by Gaertner and Westwater (1960) and Gaertner (1965) failed to establish the predominance of vertical bubble coalescence and suggested the possibility that CHF may result when bubbles merge laterally, forming large slugs of vapor which blanket the surface. The periodic formation and departure of such vapor slugs is at the heart of the macrolayer model proposed by Haramura and Katto (1983). This model assumes that the lateral coalescence of vapor bubbles or columns results in the formation of a vapor slug or mushroom, connected to the heated surface by numerous small vapor stems (see Figure 3) and fed by the evaporation of the thin layer of liquid, called the macrolayer, then trapped between the vapor mushroom and the heater surface. Following a "hovering period," postulated by Haramura and Katto (1983) to end when the buoyancy forces and the momentum of the vapor exceeded the inertia of the surrounding liquid, the vapor mushroom departs and the heater surface is rewet by the surrounding liquid, renewing the mushroom formation and departure cycle.

Haramura and Katto (1983) assumed that the spacing between the vapor mushrooms was governed by Taylor instability and equal to λ_D and that the macrolayer thickness could be taken as $\lambda_H/4$, the unstable Helmholtz wavelength. They proposed that CHF is the heat flux at which the macrolayer is completely evaporated in one hovering period, but needed the vapor fraction at the heater surface to complete their model. The authors evaluated the vapor fraction by equating their expression for CHF with Equation (1) and consequently CHF values predicted using the Haramura and Katto model are identical to those obtained from Equation (1). The primary contribution of this model lies in the modified vapor-liquid configuration, and in the concept of periodic formation and removal of vapor mushrooms from the heated surface.

LIMITATIONS OF THE HYDRODYNAMIC MODELS

While the Kutateladze-Zuber model predicts CHF accurately for boiling on large well wetted heater surfaces, it completely ignores the effects of heater thermal properties and length scale. Early visual observations of pool boiling by Kirby and Westwater (1965), and van Ouwerkerk (1972) had shown that dry patches on the heater can be re-wet by the liquid after the departure of the vapor mushroom. Thus, macrolayer evaporation and local surface dryout as proposed by Haramura and Katto (1983) are not sufficient to cause CHF. Rather, following Ramilson and Lienhard (1987), for CHF to occur, it appears necessary for the temperature of the dry patch to reach the Leidenfrost point beyond which liquid-solid contact is no longer possible. The Unal et al. (1992) conduction-centered numerical investigation of saturated pool boiling supported this "hot spot" controlled critical heat flux mechanism.

More recently Theofanous et al. (2002) have conducted careful experiments in which the authors photographically recorded both heater surface temperature variations as well as the hydrodynamic behavior of the vapor and liquid above the heater. From their experiments, Theofanous et al. (2002) have shown the occurrence of both reversible and irreversible dry spots. The authors were able to correlate the occurrence of CHF with the appearance of irreversible dry spots. Additionally, they noted shrinkage of dry spots that was attributed to heater cool down due to radial conduction of heat away from the dry spot. These studies clearly point to the fact that the onset of CHF is not just a hydrodynamic instability phenomenon, but that it reflects, as well, the contribution of thermal diffusion in the heater wall. If conduction of heat from the dry spots is to play a role in determining the onset of CHF, heater thermal properties must also influence CHF. Such dependence of CHF on heater properties has been widely observed and is discussed in detail in the following section.

Heater Property Effects

Experimental studies by several researchers have shown that the heater thickness and thermal properties have a significant influence on CHF. Bernath (1960) was, perhaps, the first researcher to investigate the effects of thermal properties and thickness of the heater on the CHF by observing that thicker heaters produced higher CHF than thinner structures. His research on vertically oriented cylindrical heaters in water was later extended to both solid and hollow cylinders. Experimental results revealed that the solid cylinders had about 43% higher CHF than hollow structures. A study employing zirconium ribbon heaters immersed in toluene, was performed by Cole and Shulman (1966). Experimental results were similar to previously published studies, with 42% higher CHF for the thickest heater. Generally similar results were found by Carne and Charlesworth (1966), who completed a detailed set of experiments, with a wider heater thickness selection, using ribbon heaters in saturated N-propanol, Tachibana et al. (1967), who performed a series of experiments with a variety of materials and thickness in water, and Guglielmini and Nanei (1976), whose experimental study involved the use of different cylindrical heaters formed with an electroplating technique, Grigoriev et al. (1978), whose experimental study with circular copper heaters in helium showed that CHF increased asymptotically until a thickness of 350μm, and Bar-Cohen and McNeil (1992), who showed that the critical heat flux on thin, sputtered horizontal platinum heaters was inferior to relatively thick, doped silicon heaters boiling in dielectric liquids. Caravalho and Bergles (1992) performed an extensive study on the thermal property effect. Their experimental study showed that thick copper blocks experienced higher CHF. But the Zuber (1959) relationship was found to over predict CHF by as much as 50 percent.

A comprehensive experimental study to determine the effects of thermal conductance/capacitance of the heater on CHF was performed by Golobic and Bergles (1997). Using heaters fabricated of ten different materials in a wide range of thicknesses (from 2 μm to 1000 μm), they determined the saturated FC-72 pool boiling CHF on horizontally suspended, vertically oriented, ribbon heaters. Fifty data sets for 5 mm high by 50 mm wide ribbons were reported. The asymptotic CHF value for the heaters was found to equal 13.5 W/cm^2. The empirical CHF values were non-dimensionalized by the asymptotic CHF values and correlated. Chang and You (1997) performed experiments with heaters of various thicknesses, boiling in saturated FC-72 under atmospheric pressure. The thinnest heaters were achieved by sputtering thin platinum films on a glass substrate. Thick heater experiments were performed with copper heaters at various thicknesses. The ratio of the experimental results to the classical Kutateladze-Zuber prediction (Equation 1) displayed the asymptotic behavior of CHF.

The first documented attempt to correlate heater surface property effects on pool boiling CHF was reported by Carne and Charlesworth (1966). Based on their experimental results with N-propanol, they proposed that the effect of the heater could be best correlated with the product of thickness and thermal conductivity. The parameter was derived from the assertion that some regions of the heater had more nucleation sites resulting in higher local heat flux values. Tachibana et al (1967) related the surface property effect to transient conduction in areas of the heater surface shrouded by vapor, preventing fresh liquid from wetting the surface and leading to CHF. This effect, they believed, could be correlated with the product of the volumetric heat capacity and the thickness. Guglielmini and Nanei's (1976) observation that thin cylindrical heaters experienced lower CHF led them to a correlation of the thickness required to achieve 90% of the "asymptotic" CHF. This thickness, δ_{90}, was found to vary inversely with the thermal effusivity to the 3.26 power, as

$$\delta_{90} = C\left(\sqrt{\rho_h C_h k_h}\right)^{-3.26} \qquad (2)$$

A comprehensive review of the experimental results and proposed correlations of the heater property effects on CHF, led Saylor (1989) to propose a hybrid "thermal activity" parameter, S, equal to the product of the heater thickness and the thermal effusivity, as

$$S = \delta_h \sqrt{\rho_h C_h k_h} \qquad (3)$$

Bar-Cohen and McNeil (1992) found the effect of S on the critical heat flux data, for sputtered horizontal platinum heaters and doped silicon heaters in dielectric liquids (providing a range of S values from 0.1 to 25), could be correlated in the form:

$$\frac{q_{CHF}}{q_{CHF-asy}} \propto \frac{S}{S+0.8} \qquad (4)$$

where S is the thermal activity parameter defined in Equation (3). This relationship yields 90 percent of the asymptotic CHF value for S equal to 8, while the 1% approach (99% of the asymptotic CHF) is attained at S=85. A summary of experimental studies of the heater property effect in dielectric liquids is given in Table 2.

Table 2: Summary of CHF vs. heater property effects studies in dielectric fluids

Authors	**Fluid**	**S**
You (1990)	FC-72	0.26
Chang and You (1997)	FC-72	21.5 – 119
Arik (2001)	FC-72	1.6 – 15.7
Arik (2001)	HFE-7100	9.8
Arik (2001)	HFE-7200	9.8
Golobic and Bergles (1997)	FC-72	< 8
Watwe and Bar-Cohen (1997)	FC-72	4.7

A more extensive literature review, focusing on low S value heaters and supporting numerical studies on the transient temperature rise of relatively thin heaters with poor thermal properties, led Watwe and Bar-Cohen (1997) to embrace the product of the thermal effusivity with the heater thickness in the correlation of CHF. In detailed numerical simulations, the temperature rise of a point in the center of a region blanketed by vapor was found to be controlled by radial diffusion, and to be governed by a relation of the form:

$$T_{sur} - T_{ini} = f\left(\frac{q\ R_d}{\delta_h \sqrt{\rho_h C_h k_h}}, \frac{r}{R}, \sqrt{t}\right) \tag{5}$$

Following closely on the proposals of Ramilson and Lienhard (1987) and Unal et al (1992), the Watwe-Bar-Cohen (1997) model is predicated on the assumption of an ever present non-uniform nucleation site density on the heater surface, which results in the formation of local vapor mushrooms, via lateral coalescence of individual vapor bubbles or the collapse of vapor columns. The depletion of the liquid under these vapor mushrooms then leads to the creation of local dry spots, whose temperature rises steeply under the influence of the imposed heat flux. The temperature rise is moderated by the ability of the underlying structure to locally absorb and/or conduct heat to parts of the heater still experiencing nucleate boiling.

If the temperature of the dry patch, at the end of the residence (or "hovering") time of the vapor mushroom, exceeds the Leidenfrost temperature, the returning liquid is unable to quench the overheated surface, local dry out proceeds to global dry out and CHF is said to occur. If the heater is very thin and has poor thermal properties, the dry patch temperature may exceed the Leidenfrost critical temperature soon after the formation of the first vapor mushroom anywhere on the heater surface. CHF on such heaters will then be highly sensitive to the non-uniformities in the nucleation site density and dependent on the S value of the surface, yielding a diffusion-driven lower limit on CHF. At the other end of the spectrum, the heater thickness and thermal properties may be so large as to successfully absorb and conduct the heat away from the local dry patch, thus keeping the dry patch temperature from exceeding the critical re-wet temperature. For such a high-S situation, the heater may be able to sustain the sequential formation and departure of many vapor mushrooms, relying on the subsequent re-wetting of the surface to locally cool the surface back to the nucleate boiling regime. CHF on such heaters would, then, occur primarily due to a hydrodynamic instability resulting in an extensive interruption of liquid flow to the heater. The hydrodynamic instability models would, therefore, provide the upper or asymptotic limit for CHF.

Watwe and Bar-Cohen (1997) provided the following equation to account for the effect of heater thermal properties:

$$\frac{q_{CHF}}{q_{CHF-asy}} \propto \frac{S}{S+0.1} \tag{6}$$

The critical heat flux values calculated via equation (6) reach 90 percent of the asymptotic CHF value when S is equal to 1, while the 1% approach occurs at S=10. It should, however, be noted that the S range in their data was relatively modest, spanning values from 0.5 to 4.7. Golobic and Bergles (1997) offer an alternative correlation of the heater surface effects on CHF, based on their experimental study of saturated pool boiling critical heat flux in FC-72. They correlated the influence of the heater material on CHF in terms of the S parameter, defined in equation (3). However, their relation took the form:

$$\frac{q_{CHF}}{q_{CHF-asy}} = 1 - e^{-\left[\frac{S}{2.44}\right]^{0.8498} - \left[\frac{S}{2.44}\right]^{0.0581}} \qquad (7)$$

Arik (2001) recently conducted experiments to further investigate the impact of the S parameter on CHF. They measured the pool boiling CHF by immersing test chips, with silicon die thicknesses of 100, 300 and 625 μm, in FC-72 under a range of pressures and liquid subcoolings. This data, along with that of Golobic and Bergles (1997) and that of Chang and You (1997), as well as earlier data discussed above, are plotted in Figure 4.

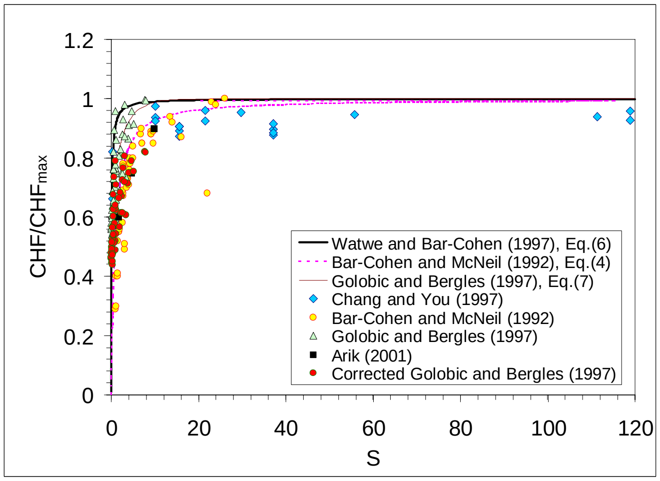

Figure 4: Variation of non-dimensional experimental CHF with the thermal activity parameter S

The three most recent correlations of the heater surface effects on CHF by Bar-Cohen and McNeil (1992), Golobic and Bergles (1997) and Watwe and Bar-Cohen (1997), all embody the dependence of CHF on the product of the heater thermal effusivity and thickness. These equations are plotted in Figure 5 which shows that equations (4), (6) and (7) yield nearly identical CHF values for S greater than 50, and all reach 90% of the asymptotic CHF value when S is higher than 8. However, significant differences do exist at intermediate values of S, where the locus of equation (7) is seen to lie between the loci of equations (4) and (6), while, at very low S values, the Golobic and Bergles relation (Equation (7)) yields higher fractions of the asymptotic CHF than the other correlations, e.g. when S equals 0.001, use of equation (7) results in q_{CHF}/q_{asy} of 0.521 vs 0.0123 for Equation (4).

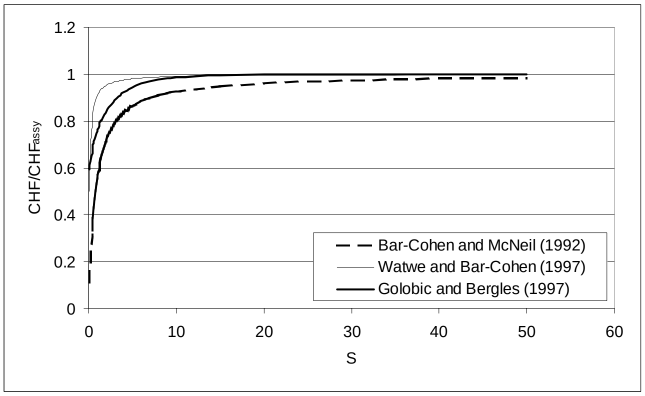

Figure 5: Variation of dimensionless CHF with the S parameter

A careful examination of Figure 5 reveals that, in the range of 0<S<1, equations (4) and (8) have relative errors of 52% and 13% respectively, while equation (9) is able to predict the experimental non-dimensional CHF values with a 9.2% relative error. When a range between 1 and 10 is considered, these errors become 16%, 21%, and 26% respectively. Finally, if the full range between 0 and 120 is taken into consideration, the Watwe and Bar-Cohen relation (1997) predicts the heater effusivity effect with a 16.6% error, while Eqs (4) and (8) were able to predict this behavior with 24% and 18% errors, respectively.

Table 3: Ratio of effusivities of heater materials compared to boiling fluids

Material	$\dfrac{\left(\sqrt{\rho C k}\right)_{\text{heater}}}{\left(\sqrt{\rho C k}\right)_{\text{FC}-72}}$	$\dfrac{\left(\sqrt{\rho C k}\right)_{\text{heater}}}{\left(\sqrt{\rho C k}\right)_{\text{Water}}}$
Silicon	51.5	9.5
Alumina	34.4	6.3
Copper	122.0	22.4
Steel	47.2	8.7

It is clear from the above discussion that there is compelling evidence for the effect of transient conduction in the heater on CHF, although the exact manner in which surface properties affect the onset of CHF remains unclear. It is nevertheless interesting to note that, as seen in Table 3, a comparison of the effusivities of various heater materials to that of a typical dielectric liquid, FC-72, reveals a far larger ratio than seen for water. , though for both fluids, the ratio is significantly greater than unity. This might suggest that thermal diffusion in the underlying heater plays a critical role in delivering heat to the liquid, freshly in contact with the hot surface following a local dryout episode. Given the relatively low effusivity of FC-72, compounded by the frequent use of thin heaters and foils, this might be of particular significance in the pool boiling of dielectric liquids from electronic surfaces. dielectric liquid, thus justifying the inclusion of this transient conduction parameter, $\delta_h\sqrt{(\rho_h C_{ph} k_h)}$, in the correlation for CHF in FC-72 and other dielectric liquids.

Length Scale Effects

In addition to the effect of heater thermal effusivity on CHF, length scale (length, width or diameter) has also been seen to have a significant effect on CHF. Several researchers including Lienhard and Watanabe (1966), Sun and Lienhard (1970), Lienhard and Dhir (1973) and Lienhard et al. (1973) have shown that the size of the heater influences the critical heat flux. The variation of CHF with heater size is typically expressed in terms of the parameter L’ given as:

$$L'(P) = L\sqrt{\frac{g(\rho_f - \rho_v)}{\sigma}} \qquad (8)$$

where, L is the characteristic heater dimension. Lienhard and Watanabe (1966) conducted experiments on horizontal wires immersed in distilled water and four reagent-grade organic liquids. They defined L’ using the wire radius and non-dimensionalized CHF with a parameter derived using the law of corresponding states. They found that CHF increased for decreasing L'. However, they also noted that there was a periodic fluctuation superimposed on this overall decreasing trend. The amplitude of this superimposed fluctuation increased as L’ decreased. Lienhard and Dhir (1973) measured CHF on horizontally-suspended, vertically oriented ribbon heaters immersed in several organic liquids. They defined L’ using the heater height. They found that for L’ < 2, there was a nearly monotonic increase in CHF with decreasing heater height. Moreover, the vertical heater data did not exhibit the fluctuations observed with wires. Lienhard et al. (1973) performed experiments using containers, which allowed an integral number of Taylor wavelengths between vertical walls. They proposed that an increase in the heater size allowed only a discrete number of additional vapor jets to form on the heater. The oscillations observed by Lienhard and Watanabe (1966) were attributed to changes in the vapor jet arrangement as the heater size is increased. Saylor et al. (1989) investigated the length scale effect on the critical heat flux on horizontal, simulated chip heaters immersed in FC-72. They proposed the following equation (L’ < 12.3) to account for this effect on thin film heaters:

$$q_{CHF} = 15.385 - 0.3639L' \qquad (9)$$

The critical heat flux was found to be invariant for L’ > 12.3 and had a value of 10.9 W/cm^2. McNeil (1992) obtained a better fit to the Saylor (1989) data by allowing the asymptotic CHF value to occur for L' > 20. The Saylor (1989) data were obtained on thin film heaters. Assuming that the Kutateladze-Zuber model (Equation 1) predicts the asymptotic value of critical heat flux, the Saylor (1989) data were multiplied by a constant factor. This factor was estimated such that the heat flux of 10.9 W/cm^2 for L’ > 20 was raised to the predictions from Equation (1). McNeil (1992) obtained the following correlation by curve fitting the corrected data:

$$\frac{q_{CHF}}{q_{CHF-asy}} = 1 + < 0.3014 - 0.01507L^{'} > \qquad (10)$$

CRITICAL HEAT FLUX ENHANCEMENT

Pressure Effects

The influence of pressure on CHF has been investigated by several researchers. In an early study, Morozov (1960) experimentally studied the effect of pressure on the saturated pool boiling CHF. He measured CHF on thin nichrome wire heaters immersed in methyl and isopropyl alcohol, for pressures ranging from atmospheric to 90% of the critical pressure, P_{cr}, of the respective liquids. He found that CHF increased with increasing pressure and attained a maximum value at approximately 35% of the critical pressure. A further increase in pressure resulted in a monotonic decrease in CHF. No correlation was proposed, but the author was able to collapse all the data to within ±20% of the mean curve by plotting the ratio CHF/P_{cr} against the reduced pressure P_r (P/P_{cr}) for each fluid.

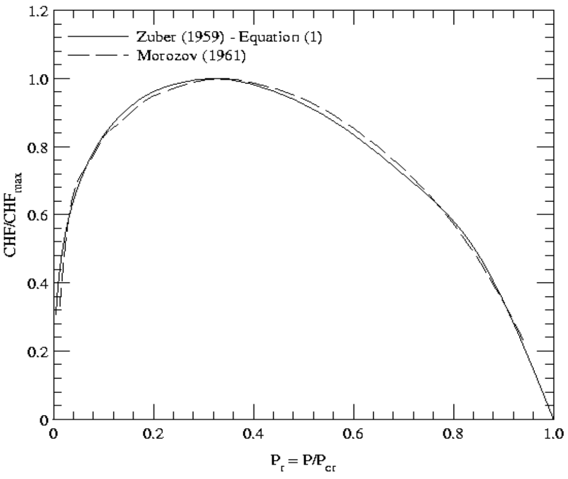

Figure 6: Variation of CHF with pressure

Lienhard and Schrock (1963) and Lienhard and Watanabe (1966) conducted experiments in a similar apparatus, with Nichrome wire heaters boiling saturated water, methyl alcohol, isopropyl alcohol, acetone, and benzene respectively. Pressure was varied from 0.02 to 101.3 kPa ($0.00009 < P_r < 0.0244$). In both studies, CHF was found to increase with pressure. While neither study provided an explicit relationship for the variation of CHF with pressure, Lienhard and Schrock (1960) noted that the Kutateladze-Zuber correlation (Equation 1) reflected their observed variation of CHF with pressure rather well. The effect of pressure on CHF is accounted for in the Kutateladze-Zuber and the Haramura-Katto models through the variation of physical properties with pressure. As shown in Figure 6 the pressure dependence of CHF predicted via Equation (1) shows excellent agreement with the faired mean curve representing the measurements of Morozov (1960).

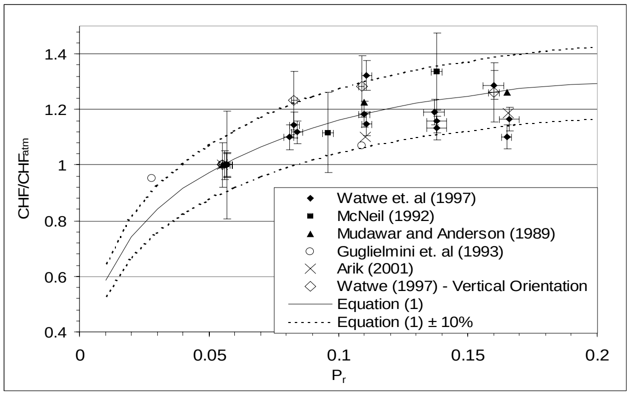

Figure 7: Empirically measured variation of CHF with pressure for FC-72

Mudawar and Anderson (1989) investigated the effect of pressure on saturated CHF for FC-72 boiling on square vertically oriented copper heaters. They varied the pressure between 101.3 and 303.9 kPa ($0.055 < P_r < 0.165$). An increase in pressure from 101.3 to 202.7 kPa resulted in an increase in CHF of nearly 23%. However, a further increase of pressure to 303.9 kPa increased CHF by only 3.7% as compared to 8.8% predicted by Equation (1). A similar weakening of the pressure dependence of CHF for pressures less than $0.35P_r$ was also reported by Guglielmini et al. (1993) and Watwe et al. (1997). However, McNeil (1992) and Arik (2001) have both reported that CHF increases continuously in the pressure range 101.3 to 303.9 kPA for FC-72. Figure 7 shows a composite graph comparing the experimental data on FC-72.

Subcooling Effects

In a pioneering study, Kutateladze (1951) suggested that CHF should increase with subcooling, and should depend on the Jacob Number, Ja. Subsequently, Kutateladze and Schneiderman (1953) performed experiments using graphite rods to boil water, ethanol, and iso-octane, and found that CHF increased linearly with subcooling. Zuber et al. (1963) proposed an extension to Equation (1) to account for the effect of subcooling on CHF. They reasoned that the subcooled liquid condensed part of the vapor, reducing the vapor-jet diameter and delaying the onset of hydrodynamic instability. They modeled the additional heat transfer to the bulk liquid as transient semi-infinite conduction at the liquid-vapor interface. The resulting expression predicted a linear increase in CHF with subcooling.

Ivey and Morris (1962) proposed one of the most widely known subcooling correlations. They performed an experimental study in which they boiled water on wire heaters. They postulated that the density ratio ρ_f/ρ_v and Ja determined the enhancement of CHF and proposed the following equation:

$$\frac{q_{CHF-sub}}{q_{CHF-sat}} = 1 + B\left(\frac{\rho_v}{\rho_f}\right)^{1/4} Ja \qquad (11)$$

Ivey and Morris (1962) found that Equation (11), with constant B equal to 0.1, was accurate to within 25% for the wire heaters tested.

In perhaps the most extensive study of subcooled CHF to date, Elkassabgi and Lienhard (1988) measured the critical heat flux on wires immersed in four different liquids (methanol, R-113, acetone, and isopropanol) with subcoolings ranging from 0 °C to 130 °C. The wire diameter was varied from

0.8 to 1.54 mm. Based on visual observations, the authors identified three subcooling regimes (referred to as ``low'", ``moderate", and ``high" subcooling) and proposed a different subcooling model and correlation for each regime. The influence of low subcooling on CHF was modeled in a manner similar to that proposed by Zuber et al. (1963). The correlation for this region was within 10% of Equation (11) and also had a similar form. In the region of moderate subcooling, the authors proposed that the heat transfer from the liquid closest to the heater, at its saturation temperature, was controlled by natural convection to the bulk fluid. They then used dimensional analysis to find an equation similar to standard natural convection correlations, which represented the data to within 7%. In the high subcooling regime, CHF attained its highest value and was found to be independent of subcooling. The authors postulated that the upper limit on CHF was imposed by the maximum rate at which molecules could leave a two-phase interface.

Hwang and Moran (1981) investigated the effect of subcooling on CHF for FC-86 (a high purity form of FC-72) boiling on 0.457 cm square, vertically oriented silicon chips. The subcooling was varied from 0 to 50 °C. They found that while CHF increased with subcooling, the rate of increase was only half that predicted by Equation (11). This reduction in the subcooling enhancement was attributed to heater geometry effects, because larger boundary layers were expected to form on flat vertical heaters as compared to wire heaters. Mudawar and Anderson (1989) investigated the subcooling enhancement of CHF for pool boiling of FC-72 on a 12.7 x 12.7 mm^2 vertical copper heater simulating an electronic chip. They correlated the linear variation of CHF with subcooling using Equation (11) with a constant C equal to 0.064. Carvalho and Bergles (1990) also investigated the influence of subcooling on the pool boiling CHF of FC-72 on thin, vertically oriented nichrome foil heaters. They found that CHF increased by approximately 3% per degree of subcooling in the subcooling range 0 to 30 °C. Some of the pool boiling experiments performed by Lee et al. (1992) included results for subcooling in FC-84 and FC-104 for both pure liquids and binary mixtures. Subcooling in the Lee et al. (1992) study was as high as 50 °C and a behavior similar to previous studies was reported. McNeil (1992) performed experiments with Silicon and thin film platinum sputtered on Pyrex heaters. He reported that CHF increased linearly with subcooling. A 66% increase in CHF was observed for a subcooling of 20 °C.

Watwe et al. (1997) presented experimental results in FC-72 on a PPGA chip package for a wide range of subcooling spanning 0 to 70 °C. Watwe et al. reported a value of 0.03 for the constant C in Equation (11) for horizontal surfaces (see Figure 8). However, Watwe et. al (1997) found that the subcooling enhancement was higher for vertically oriented heaters with the value of the constant C in

Equation (11) being 0.043 (see Figure 9). The CHF under saturation conditions was found to be nearly identical for both vertical and horizontal heaters.

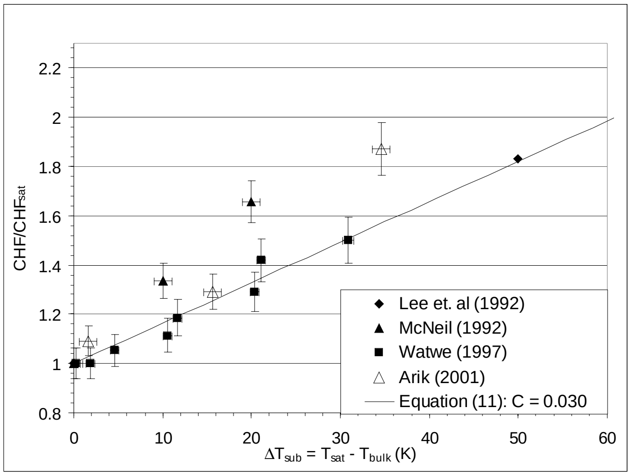

Figure 8: Impact of subcooling on CHF for FC-72 from horizontally oriented heaters

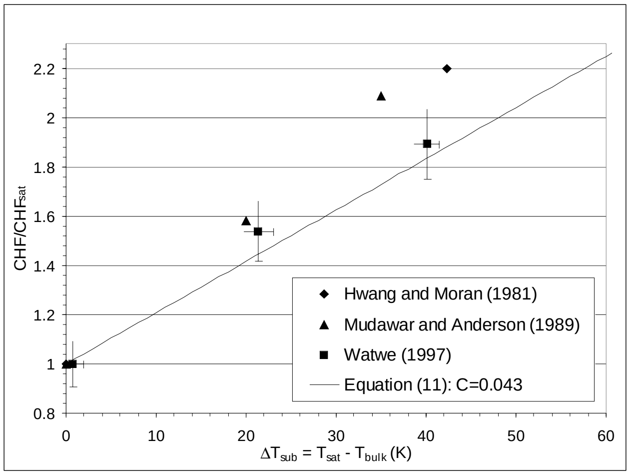

Figure 9: Impact of subcooling on CHF for FC-72 from vertically oriented heaters

Although Howard and Mudawar (1999) and Rainey and You (2001) reported that CHF decreases with increasing heater inclination from the horizontal, the data shown for vertically oriented heaters has a large spread with measurements which are larger than, equal to or smaller than the horizontal heater. Analytical models presented by Howard and Mudawar (1999) do show that lower CHF values are expected on vertical heaters. No data is presented on subcooling differences for the two orientations. Carefully controlled experiments are required to clearly demonstrate the true variation.

Dissolved Gas Effects

As discussed previously, the dielectric coolants have rather large air solubilities. The presence of dissolved air may have a significant influence on the boiling characteristics of these liquids. The influence of dissolved air on CHF has received relatively little attention. Behar et al. (1966) found that the presence of gas had no effect on the critical heat flux. The flow boiling studies of Yezhova et al. (1987) found that for small flow rates, CHF did not change with dissolved gas content because there was adequate time to degas the near-heater liquid. However, at high flow rates, rapid degassing near the heater led to a reduction in CHF. Fisenko et al. (1988) conducted pool boiling experiments using air saturated water and found that CHF increased by nearly 15 to 20% on vertically oriented heaters. However, they reported a 15 to 20% decrease in CHF on horizontally oriented wire heaters. The authors did not provide any explanation for the observed behavior. You (1990) found that for a constant total pressure of 101.3 kPa, CHF increased linearly with increasing dissolved gas content.

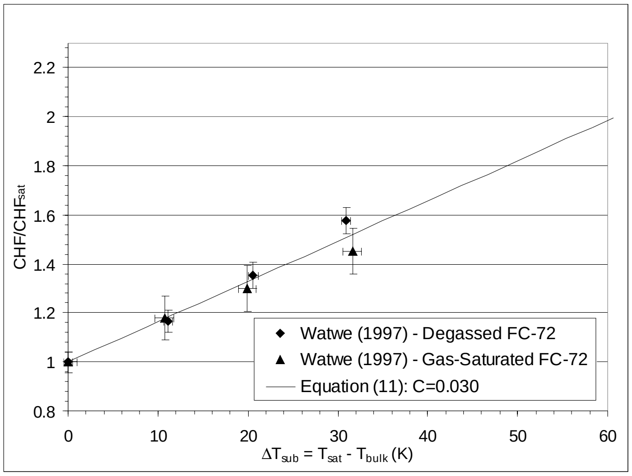

Figure 10: Variation of CHF for gas-saturated FC-72 with subcooling from the saturation temperature at total pressure.

You (1990) attributed the increase in CHF with dissolved air content to the apparent subcooling caused by localized degassing of the liquid close to the heater surface. However, Watwe and Bar-Cohen (1996) did not report any influence of dissolved air on saturation CHF measured on PPGA packages immersed in gas-saturated FC-72. Watwe (1996) was able to show that it was indeed possible for the near heater liquid to be fully degassed using a simple diffusion model. Additionally, as shown in Figure 10, Watwe (1996) found that the measured CHF data correlated well with the variation of CHF with subcooling if subcooling was computed based on the saturation temperature at the total pressure similar to the methodology proposed by You (1990).

Binary Mixtures

In addition to the parametric effects discussed above, researchers have investigated the possibility of enhancing pool boiling heat transfer rates via the use of binary mixtures. Dunskus and Westwater (1961) experimented with 11 different additives to Iso-propanol to characterize the additive properties which are required to enhance the boiling heat transfer rates. In particular they used two additives, with molecular weights of 286 and 1600 respectively, from the same homologous group. The additive boiling point was higher than that of Iso-propanol. The additive concentrations in their experiments never exceeded 0.5% by weight and there was no noticeable change in fluid properties. They concluded that when added in identical concentrations by weight, the mixture containing the additive with the higher molecular weight gave a higher critical heat flux value. Lee et al. (1992) conducted experiments with binary mixtures of the Ausimont perfluoropolyether liquids D80 (T_{sat} = 81 °C) and HT110 (T_{sat} = 110 °C), and the 3M fluorocarbon liquids FC-84 (T_{sat} = 80 °C) and FC-104 (T_{sat} = 101 °C). They found that CHF increased with increasing concentration of HT110, in mixtures of the perfluoropolyether liquids, but did not exceed the CHF of pure HT110. On the other hand, the experiments using mixtures of the 3M perfluorocarbon liquids showed no enhancement in CHF with increasing concentration of FC-104. Based on the findings of Dunskus and Westwater (1961), Lee at al. (1992) concluded that the increase in CHF in the Ausimont liquid mixtures was due to the higher molecular weight difference of 190 as compared to the lower value of 47 for the 3M liquids. Avedisian and Purdy (1993) reported higher CHF values in mixtures of n-pentane with n-propanol. They found that the increase could be correlated with increasing concentration of the more volatile liquid. Watwe and Bar-Cohen (1997) demonstrated significant CHF enhancement by adding to FC-72 low concentrations of FC-40 which had a higher saturation temperature, higher molecular weight, higher viscosity and higher surface tension. Figure 11 from Watwe (1996) shows the impact of mixture composition to boiling curves. Increasing concentrations of the higher boiling point liquid shift the boiling curve to the right and also lead to higher CHF. Watwe and Bar-Cohen (1997) used the associated shift in the wall superheat to relate the enhancement to localized depletion of the lower boiling point liquid in the near heater regions.

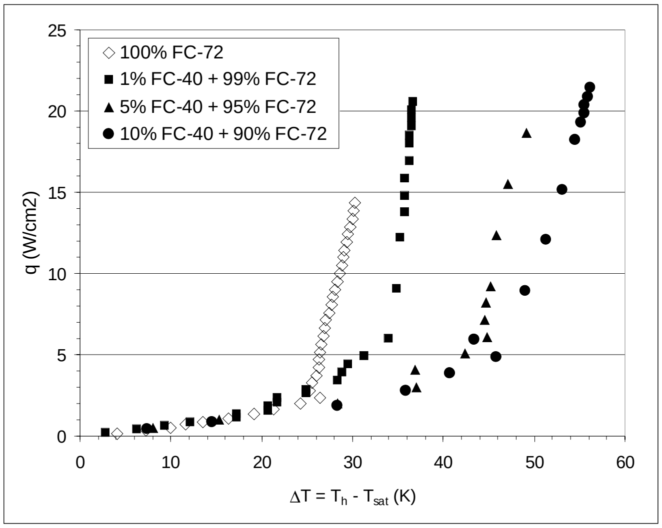

Figure 11: Comparison of boiling curves for FC-40/FC-72 mixtures with pure FC-72 at 101.3 kPa.

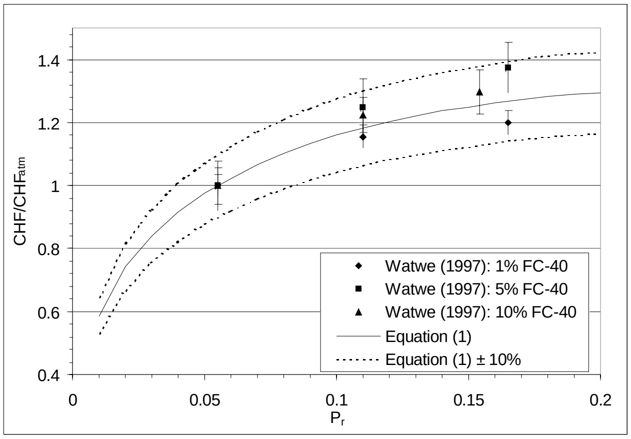

Figure 12: Variation of CHF with pressure for FC-72/FC-40 mixtures

The relative increase in CHF with pressure for the binary mixtures was still well predicted by the Kutateladze-Zuber correlation of Equation (1). Watwe and Bar-Cohen (1997) computed the critical pressure of the FC-72/FC-40 mixtures using the methods reported in Reid and Sherwood (1966). Figure 12 shows the variation of CHF with pressure measured by Watwe (1996). Figure 13 shows the variation of CHF with subcooling for FC-72/FC-40 mixtures as reported by Watwe and Bar-Cohen (1997). They postulated that localized depletion of the more volatile component in the mixture would occur in a manner similar to depletion of dissolved air from the near-heater regions. Consequently the near heater fluid would consist mainly of the higher boiling point fluid. Watwe and Bar-Cohen (1997) calculated subcooling for Figure 13 as the temperature difference between the saturation temperature of FC-40 and the bulk liquid temperature of the mixture.

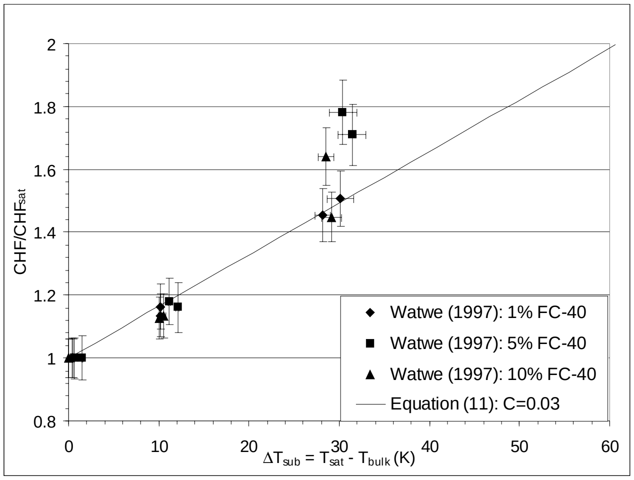

Figure 13: Variation of CHF with subcooling for FC-72/FC-40 mixtures

Surface Enhancement Effects

In addition to the salutary effect on CHF of heater thickness and high thermal effusivity, surface modifications can be used to raise the pool boiling CHF. The chemical composition and the physical character of the heat transfer surface are important to this effect. Berenson (1962) extensively investigated the effect of surface finish on nucleate boiling. Costello and Frea (1965) found that horizontally oriented, semi-circular stainless steel heaters coated with calcium carbonate yielded 50% higher CHF values than un-coated smooth heaters. Marto and Lepere (1982) investigated the heat transfer enhancement of three commercially available surfaces on a cylindrical tube geometry for FC-72 and R113. Although they reported as much as a 60% decrease in the wall superheat, they did not find any increase in the CHF. Chowdhury and Winterton (1985) reported higher CHF values on rougher surfaces as compared to smooth surfaces on copper and aluminum cylindrical surfaces immersed in methanol and water.

Mudawar and Anderson (1989) found that surface cavities 300 μm in diameter did not enhance CHF. Mudawar (1992) presented a comprehensive summary of his research on surface enhancements using techniques such as through the use of sandpaper, silica blast surface finishing and micro-fin attachments all of which resulted in substantial increases in CHF. Carvalho and Bergles (1990) attached small heat sinks to the heater surface to evaluate the impact of surface enhancement. The heat sinks were made of pure copper or with a porous coating matrix of 30 μm size particles. Nearly 5 to 6 times increase in CHF was observed using these enhancements.

In recent years, there have been several studies investigating the impact of heater surface enhancements through the use of micro-porous and porous coating techniques. You et al. (1992) presented a particle layering technique applied to a flat surface on which particles were sprayed. The sprayed surface showed a 109% increase in CHF for boiling in saturated FC-72. O'Connor et al. (1995) noted a 224% increase in CHF boiling on a heater covered with a paste of epoxy, diamond particles and isopropyl alcohol. Rainey et al. (2003) studied the impact of pressure, subcooling and dissolved gas on CHF from microporous heater surfaces. They found that although the effect of pressure and subcooling on the CHF on microporous surfaces was nearly identical to that on an un-enhanced surface the CHF magnitude on the microporous surfaces was much higher than that on the plain surface. It must be noted that the in addition to the increased nucleation site density on the porous surfaces, the addition of a coating serves to increase the heater effusivity or thermal activity parameter and would therefore be expected to cause some increase in the CHF. However, neither the increase in nucleation site density nor the increase in thermal activity parameter can explain the CHF values which are larger than the Kutateladze-Zuber (Equation 11) prediction. As seen from the variety of data sets discussed above, the Kutateladze-Zuber prediction appears to be an upper bound on the CHF attainable for a specific heater-liquid combination. Therefore there must be some other mechanism, which significantly enhances the CHF values on porous surfaces over and above the asymptotic limit. It may, thus, be argued that the development of two-phase flow in the porous structures alters the fundamental pool boiling behavior on the surface. Consequently, it is not the increased nucleation site density but rather two-phase flow in the microporous capillary structures which results in the significant increase in the upper bound on pool boiling behavior

Combined Effects on CHF

The previous discussion has illustrated that the pool boiling CHF can be significantly influenced by heater length scale, heater properties, pressure and subcooling. Watwe et al. (1997) presented the following composite equation to predict the combined effects of pressure, subcooling and heater properties:

$$q''_{CHF,TME} = \frac{\pi}{24} h_{fv} \sqrt{\rho_v} [\sigma_f g(\rho_f - \rho_v)]^{1/4} (\frac{S}{S+0.1})$$

$$[1 + < 0.3014 - 0.01507 L'(P) >](1 + 0.030[(\frac{\rho_f}{\rho_v})^{0.75} \frac{c_{p_f}}{h_{fv}}]\Delta T_{sub}) \qquad (12)$$

where $L'(P) = L\sqrt{\frac{g(\rho_f - \rho_v)}{\sigma}}$

The first term on the right hand side represents the classical Kutateladze-Zuber prediction that is the upper limit, saturation value of CHF on very large horizontal heaters. The second term is the effect of heater thickness and thermal properties. The third term accounts for the influence of the length scale on the CHF and is equal to unity or higher. If the expression between < > is a negative number it must be set to zero. The last term represents the influence of subcooling on CHF. Note that the constants a and b in Equation (12) have the values 0.3014 and 0.01507 respectively for horizontal heaters immersed in dielectric coolants. Similarly the subcooling constant in Equation (12) has the value 0.03 for horizontal heaters and 0.043 for vertical heaters immersed in dielectric coolants. The composite expression given by Equation (12) can be used to predict the critical heat flux in a variety of situations by using the appropriate constants a, b and B. A comparison of experimental results in dielectric fluids to the predictions from Equation (12) is presented in Figure 14.

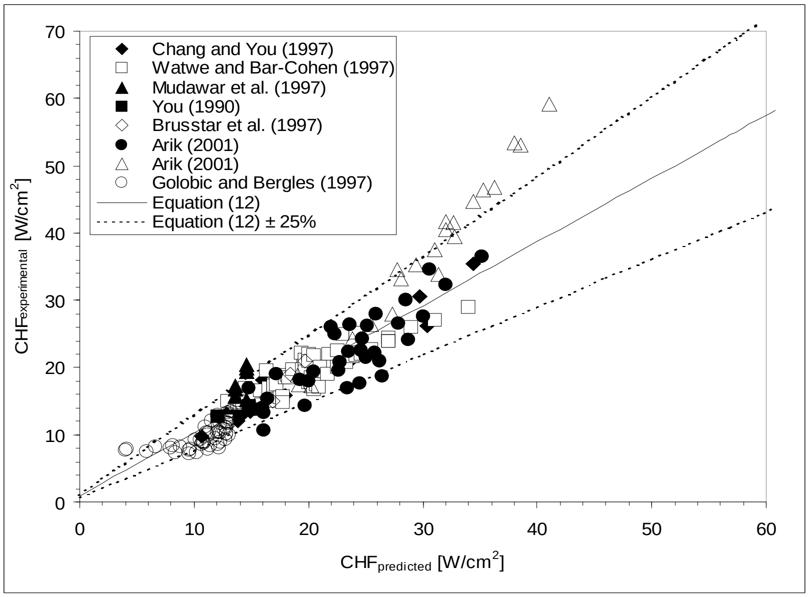

Figure 14: Comparison of experimental data with predictions from the composite CHF equation

SUMMARY

The foregoing has presented a comprehensive review of the available literature and recent experimental results for pool boiling Critical Heat Flux in dielectric liquids. Much of the discussion is devoted to elucidating the heater property effect on pool boiling CHF for these liquids. It was found that the CHF value rises asymptotically towards the Kutateladze-Zuber value, as the product of heater effusivity and thickness increases. It was also found possible to modify this thermal activity parameter to include the contribution of a conductive substrate. The pressure and subcooling enhancements of CHF were found to be substantial and predictable by the hydrodynamic dryout model and the Ivey and Morris relations, respectively. The beneficial effects of dissolved gas and binary mixtures on CHF were discussed and ascribed to the pseudo-subcooling effect resulting from the depletion of the more volatile species in the liquid adjacent to the heater. Additional significant enhancement of the CHF value was shown to result from the use of micro-porous coatings. A correlation, embodying the dependence of CHF on the product of the heater thermal effusivity and thickness, and including as well the effects of heater length and subcooling, is shown to correlate the pool boiling critical heat flux with a standard deviation of 12.5% for various heater materials and geometries, $0.2 < S < 120$, and in a large range of subcooling and pressure (0-75 K, 1-4.5 Bar) conditions within a 95 percent confidence level.

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