Sensible Heating and Boiling Incipience in Free-Falling Dielectric Liquid Films

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Controlling boiling incipience is of paramount importance for reliable operation of liquid-cooled microelectronic heat sources during power transients. This study focuses on heat transfer from a simulated multichip module to a falling film. Experiments have been performed to develop an understanding of the influence of surface tension and wetting characteristics on sensible heat transfer and boiling incipience in free-falling dielectric (FC-72) liquid films. The boiling results reveal that the vanishingly small contact angle of FC-72 precludes the application of correlations currently employed to predict incipience. Also, the temperature excursion commonly encountered upon boiling incipience in wetting fluids was nonexistent in all the experimental runs.

Introduction

Increasing power levels accompanied by decreasing chip areas are two trends projected for the VLSI circuits of the future. Effective cooling of these circuits must be provided to maintain chip surface temperatures below 85°C, which is the temperature limit set for reliable operation. Currently utilized cooling techniques include mostly direct cooling by air and indirect cooling by water. Large thermal resistances combined with noise and vibration limit air cooling to lower power density chips, while chemical and electrical incompatibility of water with electronic components requires indirect cooling schemes for that fluid. The IBM water-cooled Thermal Conduction Module (TCM) is currently capable of dissipating 30 W/cm^2 at the chip level (Bar-Cohen, 1985). Although this system is expected to meet the cooling demands of many future VLSI circuits, its planar packaging configuration and complex hardware precludes its use in future systems which may require packaging several multichip modules in a single 3-D cooling container. Thus, the need exists for new cooling technologies which provide mid-range cooling rates of 10-50 W/cm^2 at the chip level and high volumetric dissipation rates which satisfy constraints imposed by the 3-D system architecture.

An alternative approach to electronic cooling involves direct, liquid- immersion cooling, where a fluorocarbon fluid is utilized as a means of reducing the large thermal resistance associated with water-cooled modules and, consequently, increasing the volumetric heat dissipation capability of cooling hardware. Dielectric fluorocarbon liquids are appropriate for direct immersion cooling because they are chemically and electrically suitable for direct contact with electronic components; however, their poor thermal transport properties impose severe limitations on the performance of electronic cooling systems. Therefore, heat transfer enhancement by means of single-phase forced convection or boiling is needed to satisfy cooling requirements of high power electronic chips. Heat transfer enhancement by means of forced convection was demonstrated in studies by Baker (1972), Tuckerman and Pease (1981), Samant and Simon (1986), Ramadhyani and Incropera (1987) and Maddox and Mudawwar (1988). However, single-phase forced convection systems require large flow rates and substantial pumping requirements. The use of large pumps for coolant circulation may be necessary for microelectronic chips which dissipate in excess of 100 W/cm^2. However, for systems involving heat fluxes in the range of 10–50 W/cm^2, passive direct-immersion cooling may provide a viable alternative to forced convection.

The disadvantages described for forced convection cooling stimulated an interest in cooling techniques which utilize pool boiling (Bergles et al., 1968; Hwang and Moran, 1981; Marto and Lepere, 1982; Moran et al., 1982; Anderson and Mudawwar, 1988). Figure 1 shows a schematic representation of an encapsulated pool boiling thermosyphon cooling system which was originally described by Aakalu et al. (1973). In this type of system the vertically-mounted smooth chips act as the evaporator section of a two-phase, gravity-driven thermosyphon. The sealed container is partially filled with a dielectric fluorocarbon liquid such as 3M FC-72 and the vapor generated at the chip surfaces is condensed by a finned air or water-cooled water exchanger. This system eliminates pumping requirements and resistances associated with indirect cooling methods. However, pool boiling with fluorocarbon liquids is often characterized by a large surface temperature overshoot at the onset of nucleate boiling (ONB). This temperature excursion has been investigated extensively by Bergles and Chyu (1982), Marto and Lepere (1982), Moran et al. (1982), Bar-Cohen and Simon (1986) and Anderson and Mudawwar (1988). Bar-Cohen and Simon concluded that the vanishingly small wetting angles associated with fluorocarbon liquids may cause a total flooding and deactivation of possible nucleation sites. In any case, the inability to control or predict this wall

Contributed by the Electrical and Electronic Packaging Division for publication in the JOURNAL OF ELECTRONIC PACKAGING. Manuscript received at ASME Headquarters October 21, 1988.

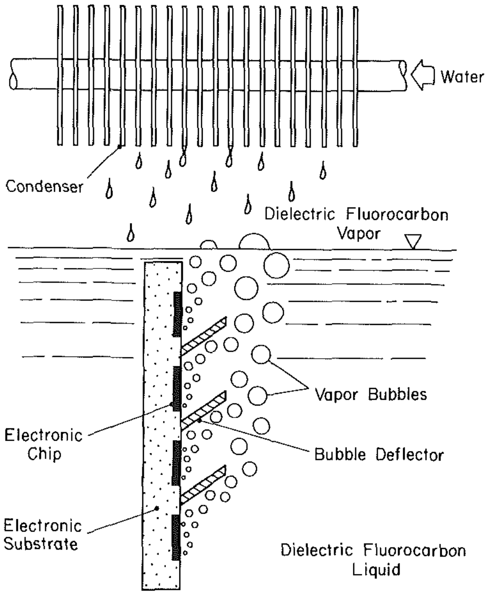

Fig. 1 Schematic representation of the encapsulated pool boiling thermosyphon cooling system

superheat excursion has restricted the development of direct-immersion cooling systems for more than two decades.

An alternative direct-immersion cooling technique which combines the advantages of minimal pumping requirements and low thermal resistance is the falling film concept shown in Fig. 2. This concept utilizes a gravity-driven two-phase thermosyphon feature similar to that shown in Fig. 1, where vapor generated by boiling at the chip surface or evaporation at the film interface is condensed by the water-cooled heat exchanger. Ideally, all of the liquid in the falling film should evaporate upon reaching the final chip, but reliability problems due to dryout require excess liquid flow in the system. Figure 3 shows a practical falling film concept proposed by Mudawwar et al. (1987) which addresses this reliability problem. The constant head reservoir which is fed by fluorocarbon condensate and a pump maintains reliable coolant flow. Although a pump is required for this system, its size would be much smaller than that required for forced convection systems since the proposed system requires a very small static head proportional to the elevation of the free liquid surface above the pump outlet. The pump flow rate is miniscule by comparison to closed channel cooling systems since the film thickness and mean velocity are typically less

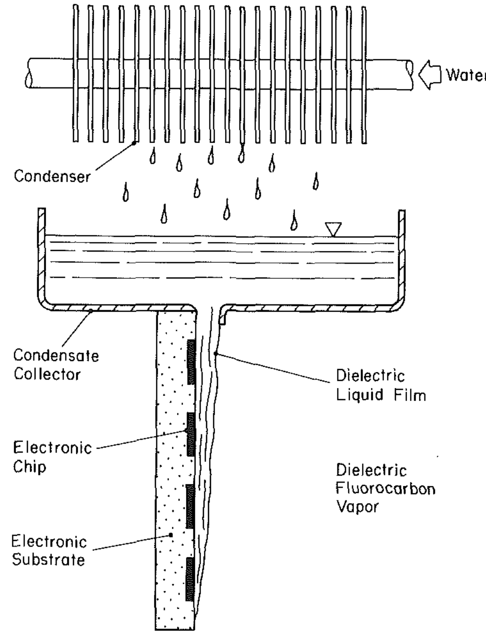

Fig. 2 Schematic representation of a passive falling-film thermosyphon cooling system

than 0.5 mm and 1.5 m/s, respectively. Another significant benefit to this falling film cooling technology is the absence of temperature overshoot at the onset of nucleate boiling as evident from the boiling results of Mudawwar et al. Furthermore, the proposed falling film concept facilitates packaging of a large number of multichip modules in a compact 3-D cooling container.

In order for the falling film concept shown in Fig. 3 to be utilized in electronic cooling, several design criteria must be thoroughly understood. The upper performance limit for such a device is the critical heat flux (CHF), which in a falling film is a function of fluid inlet velocity, film thickness and length of the heated section. Mudawwar et al. (1987) considered the parametric effects of inlet velocity and heater length on CHF and provided a wide data base for flush-mounted heat sources. In a more recent study by Mudawwar et al. (1988), the combined use of a shorter multichip module, high degree of fluid subcooling and surface microgrooves was recommended to boost CHF in thin FC-72 films to 40 W/cm^2. An example of the effect of heater length on CHF is shown in Fig. 4 for 25.4 and 127 mm vertically mounted heaters and iden-

Nomenclature

g = acceleration due to gravity
h_{fg} = latent heat of vaporization
h_H = heat transfer coefficient for sensible heating, $q/(T_w - T_m)$
h_H^* = dimensionless heat transfer coefficient, $h_H \nu_f^{2/3}/(k_f g^{1/3})$
k = thermal conductivity
L = length of heated section
P = absolute pressure
Pr = Prandtl number
q = local wall heat flux
r = bubble radius
r_{emb} = radius of initial vapor embryo
r_{max} = maximum cavity radius
r_{tan} = cavity radius based on the tangency criterion
R_g = gas constant
Re = film Reynolds number, $4\Gamma/\mu_f$
T = temperature
ΔT_{ex} = temperature overshoot at boiling incipience
ΔT_{sat} = $T_w - T_{sat}$
ΔT_{sub} = $T_{sat} - T_{in}$
U = mean film velocity
v = specific volume
v_{fg} = $v_g - v_f$
x = longitudinal distance from the upstream end of the heated section
y = distance perpendicular to the heated wall
Γ = mass flow rate per unit film width
θ = contact angle
μ = dynamic viscosity
ν = kinematic viscosity
ρ = density
σ = surface tension
ϕ = cavity cone angle

Subscripts

f = liquid
g = vapor
i = incipience
in = inlet
m = mean
sat = saturation
w = wall

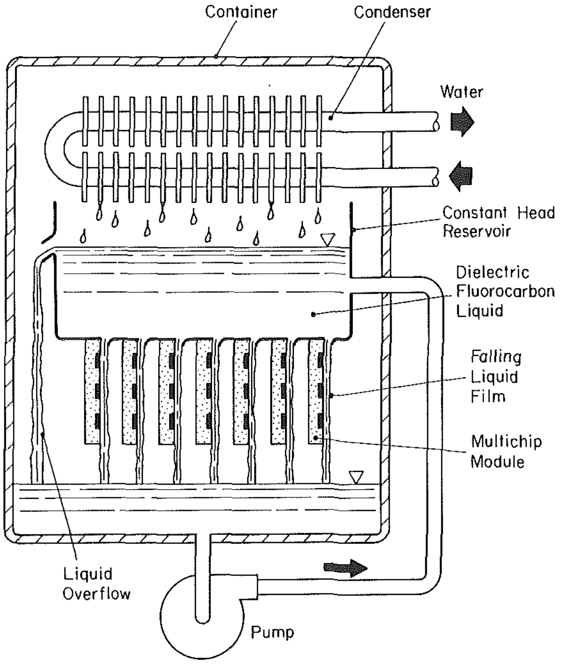

Fig. 3 Schematic of the semi-passive falling-film electronic cooling system

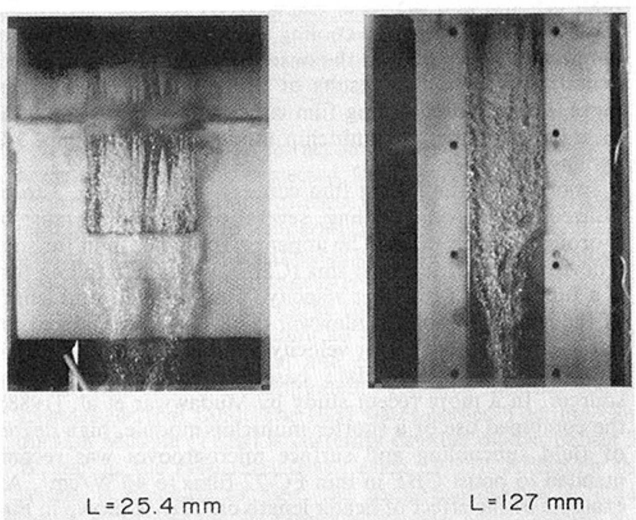

Fig. 4 Photographs of CHF in a falling film on a vertical rectangular heated surface at $U = 0.5$ m/s and $L = 25.4$ and 127 mm, respectively

tical inlet velocities of 0.5 m/s. The critical heat flux condition is more pronounced for the longer heater, where dryout occurred at the lower end of the test section following separation of the film away from the heated surface due to intense vapor generation.

Where CHF represents the upper performance limit for the design shown in Fig. 3, the onset of nucleate boiling (ONB) represents the lower limit for stable boiling in the falling film. Because the isothermal heat source used in the CHF study of Mudawwar et al. precluded the capability of measuring local incipience conditions, the present study has been performed to develop an understanding of sensible heat transfer and boiling incipience in free-falling dielectric (FC-72) liquid films.

Cerza and Sernas (1985) studied boiling nucleation criteria for a falling water film and determined that artificial cavities acting as bubble nucleation sites satisfied static pool boiling nucleation criteria. The authors are not aware of any other studies directly addressing boiling incipience in falling films. Nevertheless, incipience in forced convection systems has been

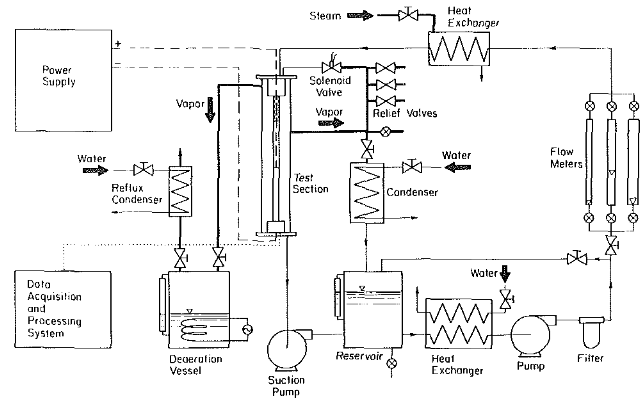

Fig. 5 Schematic diagram of the fluid delivery system

studied extensively. Early works by Sato and Matsumura (1963), Bergles and Rohsenow (1964) and Davis and Anderson (1966) predicted incipience based on the point of tangnecy between the liquid temperature profile in the vicinity of the heated surface and the superheat temperature profile required for mechanical equilibirum of a vapor bubble growing on a surface cavity. This tangency criterion is valid in most practical applications where a wide range of cavity sizes exists on the boiling surface. Another analysis used by Hino and Ueda (1985) and Sudo et al. (1985) predicts incipience based on the wall superheat required to activate the largest cavity available on the surface. Hino and Ueda used R-113, a fluorocarbon liquid whose properties and wetting characteristics closely approximate those of FC-72, and they reported large temperature overshoots at the onset of nucleate boiling.

Rather than simulating the actual multichip cooling system shown in Fig. 3, the present study involved fully developed free-falling film flow over a thin stainless steel tube to facilitate detailed local heat transfer measurements of the sensible heat transfer coefficient and wall superheat at incipience. The reduced CHF values in this study caused by increased heater length (L = 780 mm) are not characteristic of the CHF levels attainable with the much shorter multichip modules shown in Fig. 4.

Experimental Apparatus and Procedure

The experimental facility utilized in this study is the same as that used by Shmerler and Mudawwar (1987) except for minor changes which were made to accommodate the use of FC-72. A schematic of the test chamber, fluid delivery system and data acquistion system is shown in Fig. 5. The stainlesss steel heating surface used in these tests simulates a vertical array of flush-mounted computer chips. The fluid delivery system supplied high purity FC-72 to the test chamber at a controlled temperature and flow rate. The pressure in the test chamber was controlled by a valve upstream of the reflux condenser, with all tests being performed slightly above atmospheric pressure to avoid air leakage into the system. The entire system was constructed of stainless steel or compatible plastics to ensure working fluid purity. Experimental data were obtained for a Reynolds number range of 7000 to 24,000 and a Prandtl number range of 9.75 to 10.74, corresponding to inlet fluid subcooling from 21 to 8 K, respectively.

Figure 6 shows photographs of the test chamber and the sampling scoops used for mean fluid temperature measurements. In Shmerler and Mudawwar's tests, a uniform water film was created by injecting the liquid through a 300 mm long polyethelene tube with a mean porosity of 20 microns. In the present FC-72 tests, a PVC annulus packed with polyester fiber was fitted around the porous section to supply a uniform FC-72 film on the outside wall of the cylindrical surface. Without this annulus, the low surface tension

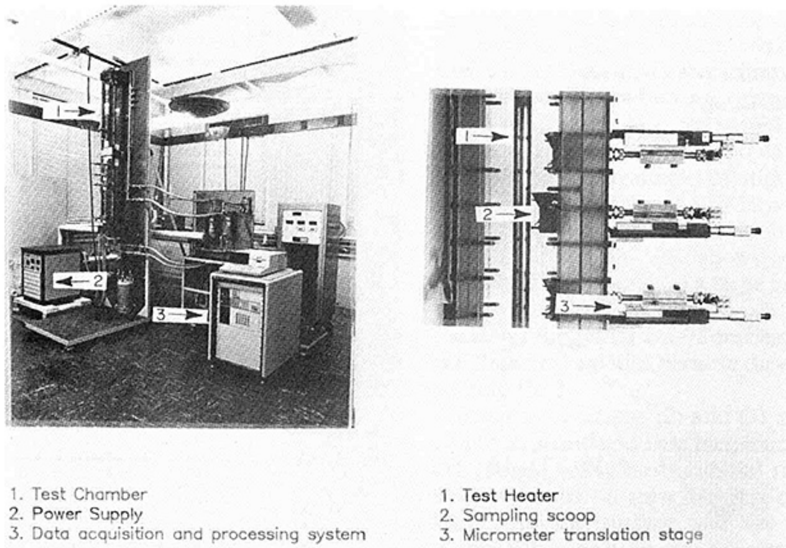

Fig. 6 Photographs of the test facility and sampling scoops

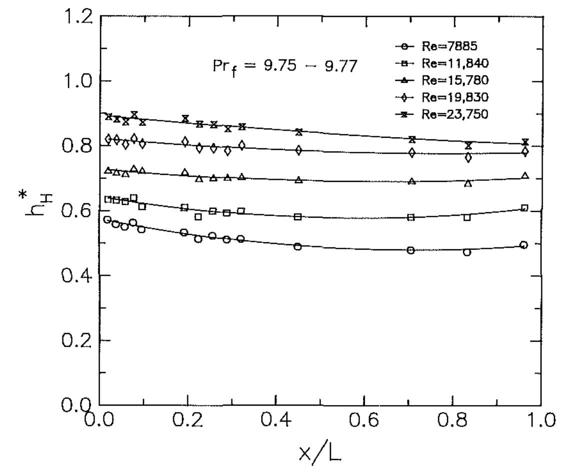

Fig. 7 Variation of the dimensionless heat transfer coefficient along the heated length for $Pr_f = 9.75$-9.77

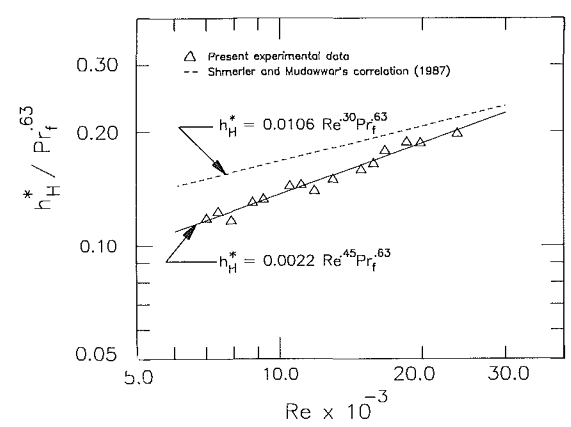

Fig. 8 Correlation of the dimensionless heat transfer coefficient

of FC-72 caused the film to spray outward away from the cylindrical surface. Following the injection annulus was a 757 mm long, 25.4 mm o.d., G-10 fiberglass plastic adiabatic section which allowed for hydrodynamic boundary layer development within the film. A 781 mm long, 25.4 mm o.d., stainless steel electrically heated test section with 0.41 mm wall thickness followed the adiabatic section. A low voltage, high DC current (up to 750 amps at 15 volts) was supplied through the stainless tube to generate constant heat flux along the length of the test section. The thin-walled tube allowed for local measurements of wall temperature by means of thermocouples attached to the inner wall of the tube. Both the G-10 adiabatic section and the stainless steel heater section were polished in vertical strokes with 600-grit paper and then cleaned with acetone. Microscopic photos of the stainless steel surface revealed that this method of polishing produced vertical grooves approximately 1 μm in width. The surface was also characterized by widely scattered nicks and scratches on the order of 40 μm or greater and a regular surface cavity distribution of approximately 0.5 to 10 μm.

Heat transfer coefficients for sensible heating were determined by measuring mean film and inside wall temperatures with copper-constantan thermocouples which were calibrated to an accuracy of 0.1°C. The mean film temperature was determined with the aid of the G-10 fiberglass sampling scoops shown in Fig. 6. The inside wall temperature was measured at 17 locations by thermocouple pairs oriented 180 degrees apart, which allowed for adjustment of the test section to ensure symmetrical film flow.

The test procedure consisted of simultaneously circulating and deaerating the fluid for 30 minutes. Once deaeration was complete the heater power was set to a point well below the incipient heat flux and the inlet temperature was set to the desired subcooling by means of the steam heat exchanger upstream of the test section. When steady state conditions were reached, the wall, film and system temperatures were recorded. To obtain boiling curves for each set of inlet conditions, the power was increased at eight minute intervals allowing sufficient time for steady state conditions. As the power was increased, boiling incipience was carefully observed and recorded for nucleation sites within circles centered at each thermocouple position. The test continued in this manner until critical heat flux conditions occurred at the downstream end of the tube, at which point the operation was terminated. In order to study boiling curve hysteresis, some tests were performed by increasing heat flux to a point of fully developed nucleate boiling and then decreasing heat flux to a point below boiling cessation. After a two hour waiting period, the heat flux was increased again to conditions of fully developed nucleate boiling.

Experimental Results

Sensible Heating. FC-72 sensible heating and boiling incipience data were obtained for Reynolds and Prandtl number

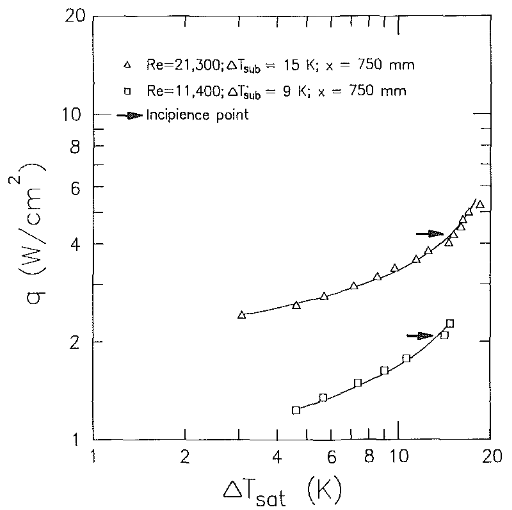

Fig. 9 Boiling curves corresponding to position $x = 750$ mm

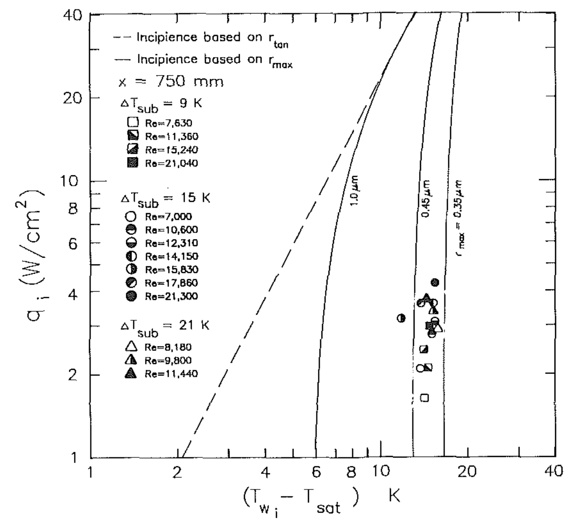

Fig. 11 Incipience conditions at position $x = 750$ mm

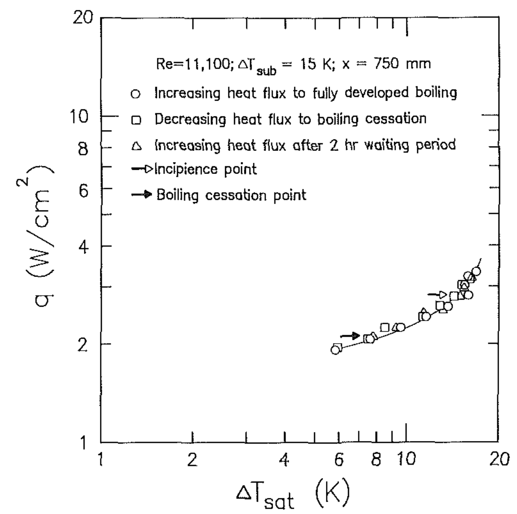

Fig. 10 Boiling curves corresponding to position $x = 750$ mm

ranges of 7000 to 24,000 and 9.74 to 10.74, respectively. The dimensionless heat transfer coefficient h_H^* for representative sensible heating data is plotted in Fig. 7 as a function of the dimensionless length x/L, where L is the length of the electrically heated test section. The fluid properties used to nondimensionalize the heat transfer coefficient are evaluated at the local measured mean film temperature, and the reference Reynolds and Prandtl numbers are evaluated at the fluid inlet temperature.

The dimensionless heat transfer coefficient exhibits only a gradual decrease at the upstream positions, indicating rapid development of the thermal boundary layer upstream of the first thermocouple position. The decrease in h_H^* continued for most tests until the middle of the heated length, at which point h_H^* gradually increased or reached a fairly constant asymptotic value. Shmerler and Mudawwar (1987) postulated that downstream heat transfer enhancement in falling films is a result of increased wave motion, and documented other studies where the same trend was detected. The downstream enhancement in h_H^* for FC-72 was very small compared to that of Shmerler and Mudawwar's water data. This phenomenon may be explained by the observation that a significant amount of FC-72 splashed away from the film, thereby significantly increasing film thickness and reducing the film flow rate. These effects outweighted the downstream enhancement of the heat transfer coefficient attributed to wave induced turbulent mixing. The splashing phenomenon encountered with FC-72 is attributed to its very low surface tension compared to water. This explanation is substantiated further by the observation that greater downstream splashing occurred at higher inlet temperatures due to decreased surface tension.

A space averaged h_H^* was determined for each run using temperature measurements obtained by the four thermocouples upstream of the last thermocouple position. Figure 8 shows a log-log regression curve fit of the present data along with Shmerler and Mudawwar's correlation for sensible heating of water films. Much of the difference between the two correlations can be attributed to surface tension effects since the greater amount of enhancement in Shmerler and Mudawwar's tests would increase the average of the downstream data points. The average heat transfer coefficient for FC-72 was correlated as

$$h_H^* = 0.0022\,\mathrm{Re}^{0.45}\mathrm{Pr}_f^{0.63} \tag{1}$$

This correlation has an average error of 1.6 percent with a maximum error and standard deviation of 4.4 percent and 0.165, respectively.

Boiling Incipience and Hysteresis. Figure 9 shows two boiling curves obtained by increasing heat flux starting from single phase forced convection conditions. Arrows in the figure indicate visually observed boiling incipience and are typically followed by an increase in the heat transfer coefficient. Figure 10 shows results of a test performed with increasing heat flux to fully developed nucleate boiling, followed by decreasing heat flux to boiling cessation. Following a two hour waiting period in the single phase mode, heat flux was again increased to a point where CHF conditions occurred at the downstream end of the heater. This test was performed to verify the negligible boiling curve hysteresis associated with falling films. It is evident from these data that, while the incipience and cessation point occur at distinct wall superheats, the boiling curve hysteresis is negligible due to the miniscule contribution of nucleate boiling between the two points to the overall heat transfer coefficient. This comes in sharp contrast with pool boiling results for dielectric fluids, where the onset of nucleation is accompanied by a large temperature overshoot due to the large difference between the nucleate boiling and natural convection heat transfer coefficients.

Incipience conditions for forced convection flow have been studied extensively and two theories are currently prominent. The first theory was proposed by Sato and Matsumura (1963) and Davis and Anderson (1966) for cases where a wide range of surface cavity sizes exists. The incipience equation based on their theory is

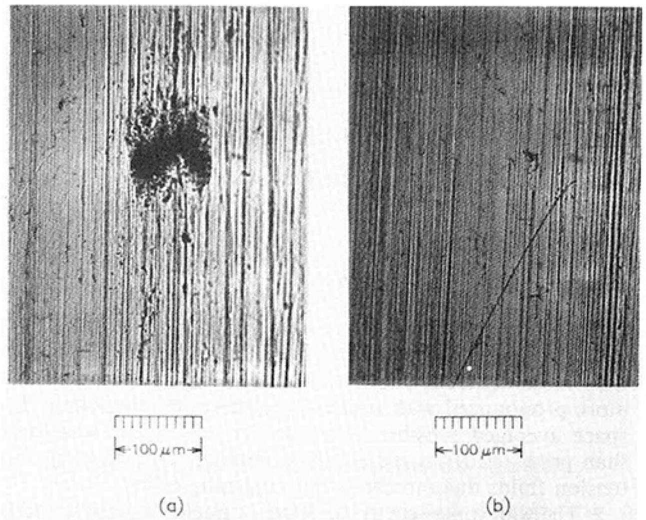

Fig. 12 Microscopic photographs of the heat transfer surface obtained at position $x = 750$ mm

$$q_i = \frac{k_f h_{fg}}{8\sigma T_{sat} v_{fg}} (T_{wi} - T_{sat})^2 \tag{2}$$

Equation (2) is based on the assumption that a bubble will grow beyond the mouth of a cavity when its entire surface is sufficiently superheated to maintain stable mechanical equilibrium at the vapor-liquid interface. The magnitude of the superheat is determined from the Clausius-Clapeyron equation and is inversely proportional to the cavity radius. The bubble interface is superheated by the surrounding liquid, the temperature of which is determined by the liquid thermal boundary layer profile near the heated wall. Since the tip of the bubble is the lowest temperature point on the bubble surfae, it was postulated that the bubble will not grow until the liquid temperature near the tip exceeds the required superheat. Thus equation (2) is based on the assumption that nucleation is initiated at a cavity whose radius is equivalent to the distance from the wall to the point of tangency, $y = r_{tan}$, between the liquid temperature profile and the Clausius-Clapeyron superheat equation.

The second frequently used incipience equation applies to conditions where the calculated radius for the first active cavity, r_{tan}, exceeds the maximum available cavity radius r_{max}. Incipience in this case was postulated to occur when the liquid temperature equals the required superheat at $y = r_{max}$. These assumptions result in the following equation:

$$q_i = \frac{k_f}{r_{max}} (T_{w_i} - T_{sat}) - \frac{2\sigma k_f v_{fg} T_{sat}}{h_{fg} r_{max}^2} \tag{3}$$

Equation (3) has been used by Hino and Ueda (1985) and Sudo et al. (1985) in their analyses of incipience data. This equation is most useful for smooth surfaces where the maximum cavity radius is smaller than the tangency radius.

Incipience conditions at thermocouple position $x = 750$ mm for three levels of subcooling and a range of Reynolds numbers are shown in Fig. 11. The incipience superheat appears to be independent of subcooling, flow rate, or wall flux. The data of Fig. 11 also indicate that r_{max} ranges from .35 to .45 μm. Hino and Ueda presented similar results for R-113 forced convection experiments with r_{max} ranging from 0.22 to 0.34 μm and with velocity and subcooling having no effect on incipience superheat. However these trends are different for those of less wetting fluids, such as water. Sudo et al. (1985), Bergles and Rohsenow (1964) and Sato and Matsumura (1963) all noted that higher water subcooling and velocity gave higher heat flux and wall superheat at the onset of nulceation.

An important consideration that has been overlooked in several incipience studies is the microscopic characterization of the heat transfer surface. Hino and Ueda concluded from their R-113 data that, according to equation (3), the largest available cavity radius r_{max} ranged from 0.22 to 0.34 μm. From the present data, we would expect the maximum cavity radius to range from 0.35 to 0.45 μm. However, microscopic analysis of the available cavities on the surface revealed a range of cavities from 0.5 to 10 μm, indicating large error between the actual surface condition and that predicted by equation (3). Figure 12 shows two microscopic surface photographs showing (a) a large pit that could not be removed by polishing and (b) a typical range of surface cavities. Apparently, the initial FC-72 nucleation sites did not appear at the "macro" cavities because they were flooded by the highly wetting fluid.

Because equations (2) and (3) are not valid for the present data it is postulated that incipience of highly wetting fluids is determined solely by the initial radius of the trapped vapor embryo, which in turn depends on the contact angle of the fluid with the surface and the depth and cone angle of available cavities. Figure 13 shows a schematic of bubble growth for a highly wetting fluid ($\theta \rightarrow 0°$). A small initial vapor embryo of size r_1 would require greater superheat than that corresponding to the cavity radius since the wall superheat required for stable mechanical equilibrium is inversely proportional to the smallest bubble radius. Lorenz et al. (1974) developed curves that gave ratios of r/r_{cav} for various fluid contact angles θ and cavity angles ϕ. They noted that for highly wetting fluids (small θ) the surface acts as though it had cavity sizes much smaller than the actual ones. However, with $\theta < 1°$ as is the case with FC-72, r/r_{cav} cannot be determined accurately from Lorenz's plots.

Because the minimum superheat required for incipience of highly wetting fluids is dependent upon the initial vapor embryo radius and not the cavity radius, the wall superheat can be immediately determined by integrating the Clausius-Clapeyron relation

$$\frac{dT}{dP} = \frac{T v_{fg}}{h_{fg}} \tag{4}$$

after assuming mechanical equilibrium at the vapor-liquid interface. That is,

$$P_g - P_f = \frac{2\sigma}{r} \tag{5}$$

Figure 14 shows embryo radius predictions based on several simplifying assumptions along with the exact value determined numerically from tabulated property values. For a wall superheat of 15 K, an embryo of 0.224 μm is expected. Figure 14 shows that the assumption $v_{fg} \cong R_g T/P$ gives satisfactory results over the range of superheats covered in the present study, while the other assumptions are not sufficiently accurate for superheats in excess of 3 K.

The boiling curves of Figs. 9 and 10 show that the temperature excursion commonly encountered in pool boiling of dielectric fluids is nonexistent in the present study. Similar results were reported by Mudawwar et al. (1987) for their FC-72 falling film study and by Toda and Uchida (1972) in their R-113 thin wall-jet study. Mudawwar et al. postulated that the lack of hysteresis could be attributed to vapor trapped within the cavities by film motion during initial wetting of the surface, or to microbubbles that may be deposited into cavities as they are swept by the film. Figure 10 provides another possible explanation for the absence of hysteresis in falling films. Though distinct, the boiling incipience and boiling cessation points fall on the same continuous boiling curve. This trend indicates that the large single-based forced convection heat transfer coefficient associated with thin films dominates the overall two-phase heat transfer coefficient for low bubble populations between the cessation and incipience

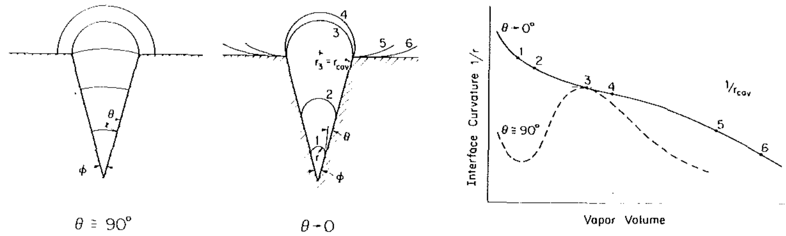

Fig. 13 Schematic representation of vapor embryo growth for highly wetting fluids

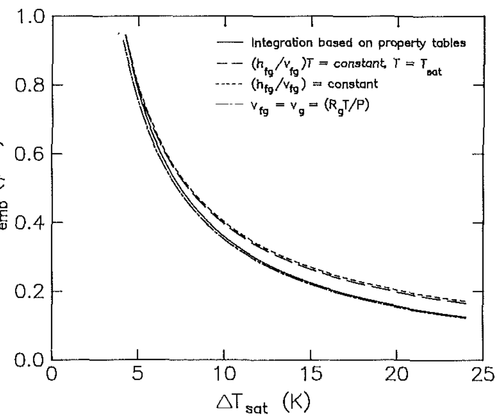

Fig. 14 Predicted vapor embryo radius as a function of wall superheat

points. In most pool boiling experiments, however, the boiling heat transfer coefficent outweighs the natural convection coefficent and boiling incipience is accompanied by a temperature drop because of the sharp increase in the overall heat transfer coefficient.

In the study by Mudawwar et al., the primary fluid delivery system was evacuated to less than 500 μm Hg prior to charging the degassed FC-72 into the system reservoir in order to reduce the volume of noncondensible gasses in the system. This was not possible in the present study because of pump cavitation at low pressures. Thus, the fluid was deaerated for 30 minutes as it was circulated through the system. In both studies, though, a sufficient number of vapor embryos were trapped in wall cavities allowing heterogeneous nucleation without any wall superheat excursion. Apparently, initial conditions in the test chamber had little or no effect on hysteresis at ONB for the two falling film studies. Another condition for comparing hysteresis in the present study to previous studies is the thermal mass of the heater and the location of thermocouples relative to the surface. The large thermal mass of the heater utilized by Mudawwar et al. may have dampened local hysteresis effects at ONB. The stainless steel heater utilized in the present study was 0.41 mm thick with thermocouples mounted on the inside wall. Hino and Ueda's stainless steel heater was 0.5 mm thick with thermocouples also mounted on the inside wall, and they reported hysteresis up to 20 K. In comparison to Hino and Ueda's heater, the instrumentation in this study would have been sensitive to local temperature overshoot had it indeed occurred. The absence of hysteresis at ONB makes the proposed falling film concept of Fig. 3 very suitable for electronic cooling applications.

Conclusions

This study has focused on the effects of surface tension and wetting characteristics of FC-72 on sensible heat transfer and boiling incipience in free-falling films. Key conclusions from the study are as follows:

1 The low surface tension of FC-72 results in splashing of the fluid away from the film interface. This splashing became more pronounced with decreasing degree of subcooling. The space averaged sensible heat transfer coefficient was lower than predicted from correlations obtained for higher surface tension fluids due mostly to this splashing effect.

2 The wall superheat at the onset of nucleate boiling (ONB) for FC-72 was found to be independent of flow rate, subcooling, or wall heat flux due to the small wetting angle of the fluid. Also, correlations based on r_{tan} or r_{max} were not physically valid because the radius of the initial vapor embryos was smaller than those of surface cavities.

3 The wall superheat excursion at the onset of nucleate boiling was nonexistent in the present falling-film study. This constrasts sharply with results of previous pool and flow boiling experiments where boiling incipience was characterized by severe wall temperature overshoots. The absence of hysteresis at ONB renders the falling film concept shown in Fig. 3 very suitable for cooling of electronic components.

Acknowledgment

The authors gratefully acknowledge the support of the Office of Basic Sciences of the U.S. Department of Energy (Grant No. DE-FG02-85ER13398). Also, generous donations of the liquid Fluorinert (FC-72) from the Industrial Chemical Products Division of the 3M Company are greatly appreciated.

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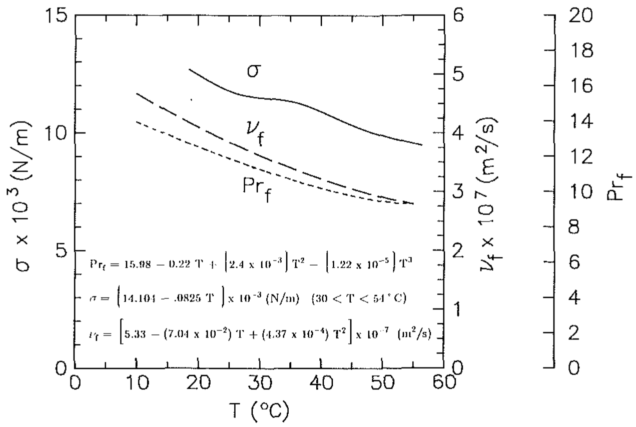

Fig. 15 Properties of FC-72 as a function of temperature

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APPENDIX

Figure 15 shows curve fits for FC-72 properties used in the present study. Most of the property values are based on tables supplied from the Industrial Chemical Products Division of 3M Company. The surface tension data were measured at Purdue's Boiling and Two-Phase Flow Laboratory using a precision Kruss K8 ring-type interfacial tensiometer equipped with a constant temperature bath. The kinematic viscosity and Prandtl number curve fits are based on recently updated viscosity tables obtained from 3M.