Comparison of Two-Phase Electronic Cooling Using Free Jets and Sprays

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The performances of free jets and sprays were compared experimentally in cooling a 12.7 × 12.7 mm² chip in order to ascertain the effects of key parameters on cooling performance and to develop correlations for critical heat flux (CHF) which are applicable to dielectric coolants. Increasing liquid flow rate and subcooling increased CHF for both cooling schemes. At high subcooling, comparable CHF values were attained with both for equal flow rates. However, spray cooling produced much greater CHF at low subcooling than did jet cooling. This phenomenon was found to be closely related to the hydrodynamic structure of the liquid film deposited upon the chip surface. In jet cooling, the film (wall jet), being anchored to the surface only at the impingement zone, was separated from the surface during vigorous boiling due to the momentum of vapor normal to the surface. The individual spray drops were more effective at securing liquid film contact with the surface at low subcooling, which delayed CHF relative to jet cooling with the same flow rate. This paper also discusses practical concerns associated with implementation of each cooling scheme including system reliability and the risk associated with premature CHF during chip power transients.

1 Introduction

Continued miniaturization of microelectronic components coupled with increasing chip clock speeds have caused the heat dissipation from today's computer chips to increase drastically compared to those developed only a decade ago. Despite many recent developments in pursuit of lower chip heat dissipation rates, many types of chips, especially those used in supercomputers, could exceed 100 W/cm² in the near future. Also, chips used in military electronics, especially those of high performance fixed wing aircraft, are constantly being repackaged into smaller and lighter modules; the ensuing heat dissipation rates are rapidly exceeding the capabilities of today's most advanced avionic thermal management schemes.

Both supercomputer and military electronics could greatly benefit from aggressive cooling schemes such as direct liquid immersion, especially those involving change of phase of the coolant. Unlike single-phase liquid-immersion cooling techniques, which produce chip temperature increases proportional to the increases in chip heat flux, liquid immersion with phase change capitalizes on the merits of nucleate boiling, allowing large increases in heat flux with only modest increases in chip temperature. Numerous studies have focused on direct immersion cooling with phase change to meet the demands for high flux, low temperature cooling (Mudawar, 1992). Many of these studies, as is the present, have been concerned with the prediction of critical heat flux (CHF), the upper limit for safe chip cooling in the nucleate boiling regime. The present study is concerned with two types of direct immersion cooling: unconfined jet-impingement cooling and spray cooling.

1.1 Unconfined Liquid Jet Cooling. The demand for high-heat-flux dissipation from surfaces in a large number of applications has stimulated much research in the area of jet impingement cooling which has spanned over fifty years. Early research focused mainly on single-phase cooling. Recently, however, researchers have turned their focus more toward two-phase cooling because of new technological advances in many industries. Jet impingement in being used to quench metallic alloys parts rapidly from relatively high temperatures corresponding to the film boiling regime in order to alter the metallurgical structure of the alloy and produce parts with superior mechanical properties. In other applications, jet impingement is being evaluated as a means of maintaining relatively low, steady temperatures in devices which dissipate enormous heat fluxes such as x-ray anodes, and lasers. Jet impingement liquid cooling can be implemented in three main configurations: free jet (jet formed in a vapor or gaseous ambient), submerged jet (jet formed in a liquid ambient), and confined jet (jet confined by a wall parallel to the heated surface). The present study is concerned only with free jets.

Ruch and Holman (1975) and Shibayama et al. (1979) examined jet boiling and recommended correlations to predict nucleate boiling heat transfer. However, most two-phase jet impingement studies during the past two decades have been concerned primarily with CHF. Katto and Kunihiro (1973) were among the first to study CHF of impinging jets and provide insight into the mechanisms involved. Using gravity-driven jets of liquid, they studied burnout on a circular copper heater. A considerable fraction of the mass of liquid available in the liquid film (wall jet) emanating from the impingement zone was observed to splash away from the heated surface during nucleate boiling due to the vigorous effusion of vapor at the surface. This resulted in the development of dry patches on the outer circumference of the heater, which propagated inward as CHF was approached. They concluded CHF occurs due to dryout of the fraction of liquid still capable of maintaining contact with the surface, and postulated the increase in CHF they measured with increasing jet velocity was the result of the jet's ability to better resist the splashing. Katto and Kunihiro demonstrated nozzle-to-heater spacing had little effect on CHF for the range of 0.63 to 42 nozzle diameters.

Four different CHF regimes have been proposed by Katto and Shimizu (1979) and Monde et al. (1987) in which CHF exhibits varying dependence on jet velocity, density ratio, ρ_f/ρ_g, and diameter ratio, D/d (for circular heaters). These

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regimes carry the designation L-, V-, I-, and HP-regime. The L-regime was observed only at very low flow rates and/or very large diameter ratios, where CHF occurs as a result of liquid deficiency as nearly all of the liquid impinging upon the heated surface is evaporated even without splashing. The V-regime, evident at elevated flow rates and encountered in most studies, was characterized by a strong dependence of CHF on jet velocity. The final two regimes, the mechanisms for which are not well documented in the literature, occurred only at high pressures and were identified primarily from deviation of their data from the V-regime correlation. Monde et al. (1987) developed a single correlation form to predict CHF in the V-, I-, and HP-regimes, assigning different empirical constants to the dimensionless groups in each regime.

$$\frac{q''_m}{\rho_g h_{fg} u_e} = f\left(\frac{\rho_f}{\rho_g}, \frac{\sigma}{\rho_f u_e^2 D}, \frac{d}{D}, \frac{c_{p,f}\Delta T_{\text{sub}}}{h_{fg}}\right). \quad (1)$$

Several recent studies have addressed the specific application of jet impingement to electronic cooling using dielectric liquids. Cho and Wu (1988) confirmed the splashing behavior proposed earlier by Monde, Katto, and their co-workers in describing CHF. Nonn et al. (1988, 1989) also studied jet impingement boiling of dielectric liquids and found a correlation for boiling in parallel channel flow proposed by Lee et al. (1987) correlated their own jet impingement CHF data quite well.

1.2 Spray Cooling. Both jets and sprays are produced by forcing liquid through a small diameter orifice. In the case of sprays, however, the liquid is purposely shattered into a dispersion of fine droplets prior to impact. A spray can be formed either by high pressure (plain orifice spray) or atomized with the aid of pressurized air (atomized spray). Key to achieving superior cooling performance with both types of sprays is reducing the surface area to volume ratio of the liquid by producing very small droplets. While atomized sprays are widely used in many industries because of their superior cooling, the existence of air greatly complicates deaeration and condensation of the coolant in electronic cooling applications. Plain orifice sprays are, therefore, the practical choice when considering spray cooling for these applications. The present study is concerned with only plain orifice sprays.

Most of the early research on spray cooling focused on the film boiling regime associated with the cooling of metals existing a blast furnace. Results from these studies are, unfortunately, of little value to electronic cooling system design. Of the studies concerned with the single-phase and nucleate boiling regimes, water has typically been the fluid of choice. Both Toda (1972) and Monde (1980) found the heat transfer coefficient increased in both regimes with increasing spray volumetric flux (coolant volume flow rate divided by surface area). Pais et al. (1989) suggested heat transfer could be ameliorated by minimizing drop size, maximizing drop concentration, and choosing drop velocities that would minimize drop rebound at the surface.

The most comprehensive studies on spray cooling conducted to date are those of Mudawar and Valentine (1989), which produced correlations for single-phase cooling, nucleate boiling, and CHF, and by Klinzing et al. (1992) which complemented the study by Mudawar and Valentine with correlations for the transition and film boiling regimes. These correlations have been very accurate in predicting the cooling history of metal surfaces cooled by water sprays (Mudawar and Deiters, 1994; Hall and Mudawar, 1994). Like the studies of Toda and Monde, these correlations clearly indicate spray volumetric flux is the dominant parameter influencing cooling rate. Increasing spray flux was found to increase the heat transfer coefficient in every regime except nucleate boiling, which was found to be insensitive to volumetric flux. The Mudawar and Valentine correlations also indicate smaller drops enhance CHF.

Only recently have the heat transfer mechanisms of dielectric coolants (e.g., FC-72 and R-113) been studied (Ghodbane and Holman, 1991; Holman and Kendall, 1993; Tilton et al., 1992). Unfortunately, no correlations have been proposed for CHF in spray cooling for these coolants.

1.3 Spray Cooling Versus Jet Impingement Cooling. Much more extensive work has been performed on jet impingement cooling than on spray cooling, resulting in a much better understanding of the mechanisms associated with each of the jet impingement boiling regimes. In a study whose objectives closely resemble those of the present, Cho and Wu (1988) conducted experiments comparing spray cooling to jet cooling using Freon-113. Nucleate boiling was observed to commence near the outer circumference of the heated surface for both. Near CHF, a large fraction of the surface was undergoing dryout with the jet, but not the spray. This partial dryout with the jets produced both large spatial temperature gradients and higher surface temperatures. They measured similar CHF values for sprays and jets, but suggested sprays are superior at inhibiting large temperature gradients along chip surfaces. They developed CHF correlations for both that allowed prediction of (dimensional) CHF in terms of Weber number based on liquid (jet or spray drop) velocity and heater diameter. Their correlations did not account for drop size or jet diameter as suggested by most earlier studies.

The objectives of the present study are to attain a better understanding of the heat transfer mechanisms of jets and spray cooling, and to develop CHF correlations for both especially

Nomenclature

c_p = specific heat at constant pressure
D = diameter of circular heaters
d = jet diameter
d_0 = orifice diameter of spray nozzle
d_{32} = Sauter mean diameter (SMD)
h_{fg} = latent heat of vaporization
l = twice the distance from jet center to chip corner = $\sqrt{2}L$
L = chip length (12.7 mm)
P = pressure
ΔP = pressure drop across nozzle
q'' = chip heat flux
q''_m = critical heat flux (CHF)
$q''_{m,p}$ = CHF at the edge of spray impact area
q''_* = dimensionless CHF
Q = volumetric flow rate
Q'' = volumetric spray flux
$\underline{Q}''$ = mean spray volumetric flux across impact area = $Q/(\pi L^2/4)$
Re_{d_0} = Reynolds number based on orifice parameters
T = temperature
ΔT_w = temperature difference between chip surface and fluid = $T_w - T_f$
ΔT_{sub} = liquid subcooling = $T_{\text{sat}} - T_f$
T_w = chip surface temperature
T_f = fluid temperature upstream of nozzle
u_e = jet velocity
We_{d_0} = Weber number based on orifice parameters
ϵ_{sub} = subcooling factor in jet CHF correlation
θ = spray cone angle
μ = absolute viscosity
ρ = density
σ = surface tension

Subscripts

f = liquid
g = vapor
sat = saturated
sub = subcooled
w = chip surface condition.

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suited for electronic cooling system design. Another objective of this study is to compare the cooling performances of both and address some of the practical issues associated with implementing each scheme.

2 Experimental Methods

Test Chamber. Figure 1(*a*) shows a sectional diagram of the test chamber which housed the chip module, jet or spray nozzle, and a water cooled condenser. As shown in Fig. 1(*b*), the simulated chip, which had a surface area of 12.7 × 12.7 mm^2, was machined from oxygen free copper. To the back of the chip was soldered a thick film resistor which supplied power to the copper surface. The copper and resistor were insulated on all sides save the impact surface with low conductivity fiberglass plastic. A thermocouple embedded directly beneath the center of the chip allowed extrapolation of the surface temperature, assuming one dimensional heat conduction. Since the primary focus of the present study was CHF, no attempt was made to determine the variations in chip surface temperature; hence, the use of only one thermocouple for the chip temperature measurement.

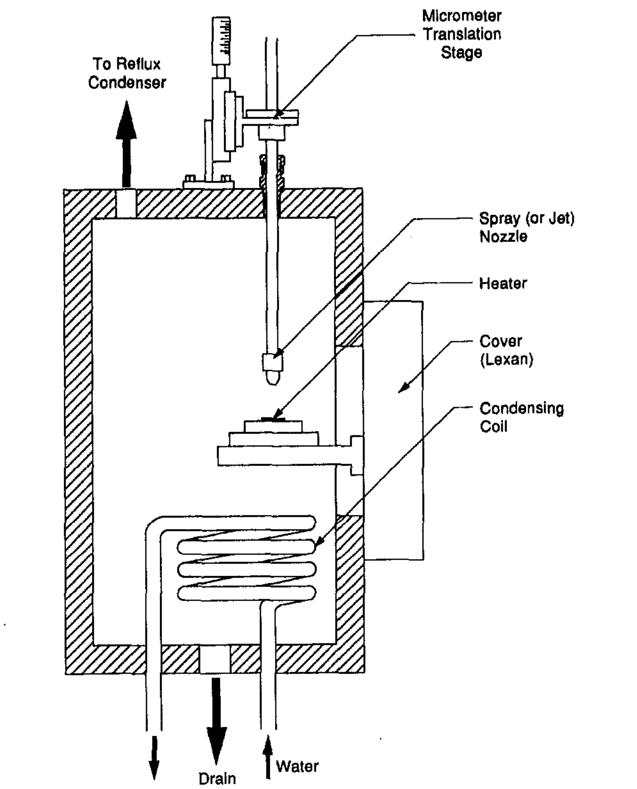

Fig. 1(*a*)

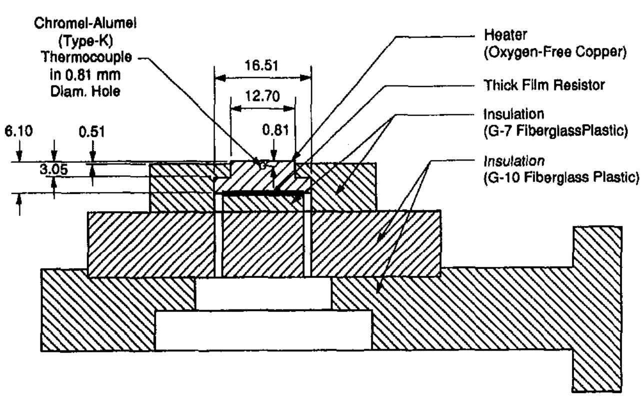

Fig. 1(*b*)

Fig. 1 (*a*) Schematic of test chamber. (*b*) construction of chip module.

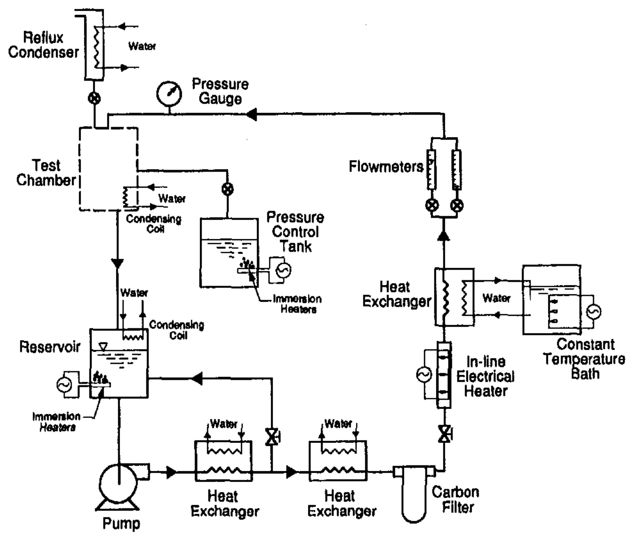

Fig. 2 Schematic of flow loop

The chip module was supported upon a transparent polycarbonate plastic (Lexan) window which was flanged to the side of the test chamber. Flow visualization was possible through the combined use of the window and a second, larger window which formed the front of the test chamber.

Different test nozzles could be fitted to the end of a vertical tube extending into the test chamber directly above the chip. The jet or spray nozzle could be raised or lowered relative to the chip surface with the aid of a digital micrometer translation stage mounted atop the test chamber.

Flow Loop. Figure 2 shows a schematic of the two-phase flow loop. Liquid (FC-72 or FC-87) was gravity fed from the loop reservoir into a magnetically coupled centrifugal pump situated directly beneath. The liquid was pumped into two branches, one leading to the test chamber, and the other back to the reservoir. Bypassing the flow in this manner enabled accurate control of both pressure and flow rate upstream of the jet or spray nozzle. The temperature of liquid advancing toward the test chamber was raised or lowered by two flat plate heat exchangers, and fine tuned with the aid of an in-line electrical heater and a third flat plate heat exchanger whose water temperature was controlled by a constant temperature bath. The jet or spray flow rate was measured by one of two flow meters with overlapping ranges, and inlet pressure was measured with a pressure gauge connected just upstream of the nozzle. The liquid impinged upon the chip surface inside the test chamber, draining directly into the reservoir.

Operating Procedure. The chip's surface was polished prior to each test, and any residue from the polish was removed with a cotton swab dipped in methanol. Following this procedure, the liquid was carefully deaerated by boiling in the reservoir, and allowing the vapor, mixed with noncondensible gasses, to exit through a reflux condenser situated above the test chamber. The vapor condensed and dripped back into the test chamber, while the noncondensible gasses were allowed to escape into the ambient. The reservoir liquid was boiled vigorously for about 15 minutes, then the pump was started. Deaeration continued for an additional 15 minutes with the fluid circulating through the loop in order to capture any noncondensible gasses in the loop itself.

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A valve connecting the test chamber to the reflux condenser was closed off following the deaeration to completely seal the loop from the ambient. Pressure at the inlet to the nozzle was then adjusted to 1.03 bar (15 psia) through the combined use of a pressure control tank connected to the test chamber and the condensing coil located in the chamber itself.

Steady state boiling data were recorded by a computer controlled Keithley 500 data acquisition system. Boiling curves were generated by raising the voltage across the chip's thick film resistor in small increments with the aid of an autotransformer and recording both the chip heat flux and chip surface temperature after the chip temperature reached steady state. Power increments were reduced to less than 1 W/cm^2 as CHF was approached in order to both preclude premature occurrence of CHF and ensure accurate CHF measurement.

Uncertainties in the reading of pressure, temperature, power, and flow rate all contributed to experimental error. Measurement uncertainties of the pressure gauge, flow meter, and watt transducer were ± 0.5, ± 1.6, and ± 1 percent, respectively. Heat loss to the heater surroundings was determined numerically to be less than 4.5 and 1 percent during the single-phase and boiling regimes, respectively. Thermocouple measurements were made with a ± 0.2°C accuracy.

3 Results and Discussion

3.1 Jet Cooling

Heat Transfer Trends. Three orifice plate type nozzles were employed in the jet cooling study. These nozzles were given the designation 1, 2, and 3, in the order of increasing orifice diameter; 0.66, 0.86, and 1.14 mm, respectively.

Figures $3(a)-3(c)$ show for jet nozzles 1, 2, and 3, respectively, boiling curves for liquid jets of FC-72 with $\Delta T_{\rm sub} = 33$°C striking the center of the chip. Each figure shows increasing jet flow rate increases the single phase heat transfer coefficient, delays incipience to a higher heat flux and increases CHF, features which are typical of most flow boiling systems.

Figures $4(a)-4(c)$ illustrate how CHF can also be greatly increased for the same flow rate by increasing the liquid subcooling. However, the benefits of higher subcooling in jet cooling extend well beyond increasing CHF. These benefits become readily apparent in comparing Figs. 5 and 6, which display photographs and schematics of the boiling on the chip surface for nozzle 1 using the same flow rate and $\Delta T_{\rm sub}$ of 13 and 33°C, respectively. Both figures show boiling commences near the edge of the chip and propagates inward towards the impingement zone as the heat flux is increased. The boiling takes the form of radial bubble streams which increase in number with increasing heat flux. At the lower subcooling ($\Delta T_{\rm sub} = 13$°C), the boiling front converged closer to the impingement zone even at relatively low heat fluxes, whereas the boiling front at 33°C remained clear of the impingement zone even as CHF was approached. Since the wall jet is "anchored" to the chip surface only at the impingement zone, the severe boiling and ensuing separation of liquid from the chip surface are clearly behind the relatively low CHF values at lower subcooling.

CHF Correlation. An extensive CHF data base was amassed in order to ascertain the effects of the key jet parameters. A quick inspection of these trends reveals CHF increases with increasing jet velocity, jet diameter, or subcooling. These trends are fairly consistent with the conclusions of most of the jet studies reviewed.

Monde and co-workers presented several correlations for CHF in saturated unconfined jets. In their most recent (Monde and Inoue, 1991), the heater diameter (most jet cooling studies were based on circular rather than square heaters) was replaced by a heater characteristic length, l, defined as twice the distance from the jet center to the most remote point on the heated surface regardless of surface geometry.

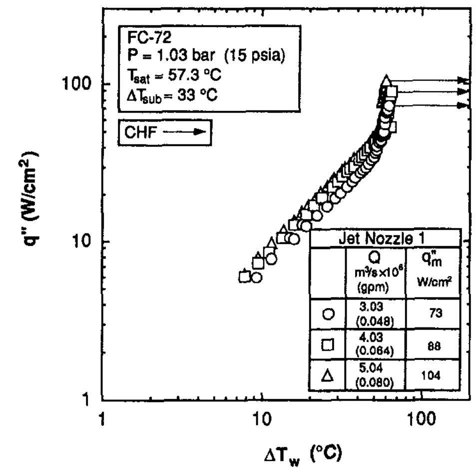

Fig. 3(*a*)

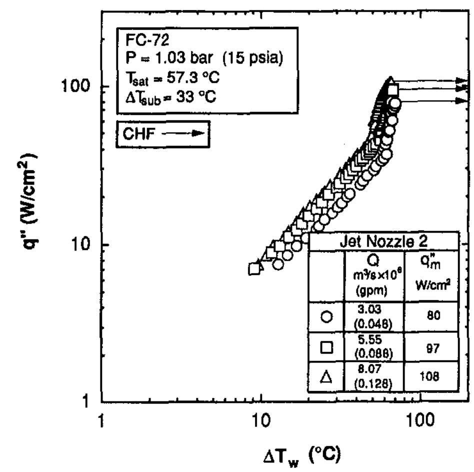

Fig. 3(*b*)

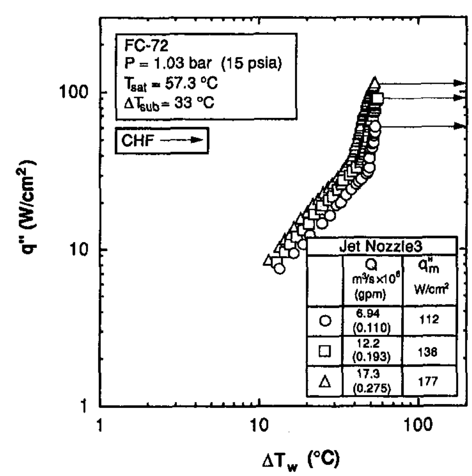

Fig. 3(*c*)

Fig. 3 Boiling curves for three flow rates at 33°C subcooling for (*a*) jet nozzle 1, (*b*) jet nozzle 2, and (*c*) jet nozzle 3

$$\frac{q''_m}{\rho_g u_e h_{fg}} = 0.221\left(\frac{2\sigma}{\rho_f u_e^2(l-d)}\right)^{0.343}\left(\frac{\rho_f}{\rho_g}\right)^{0.645}(1+l/d)^{-0.364}. \quad (2)$$

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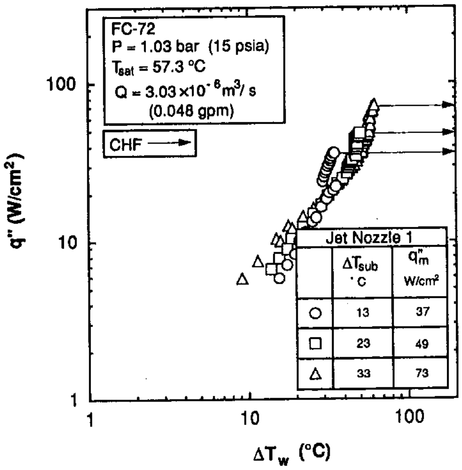

Fig. 4(*a*)

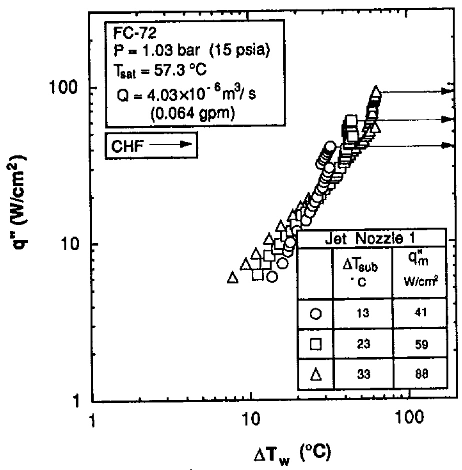

Fig. 4(*b*)

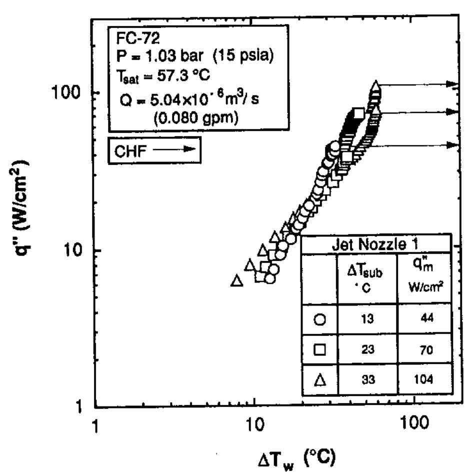

Fig. 4(*c*)

Fig. 4 Boiling curves for three subcoolings at (*a*) $Q = 3.03 \times 10^{-6}$, (*b*) $Q = 4.03 \times 10^{-6}$, and (*c*) $Q = 5.04 \times 10^{-6}$ m³/s for jet nozzle 1

One noticeable weakness in most of the reviewed jet studies was a tendency to neglect the effect of subcooling in CHF correlations. This effect was noticeable in a study by Monde and Katto (1978), but was later neglected even though some degree of subcooling must have been present to prevent cavitation within the jet orifice. Indeed, Fig. 7 shows equation (2) can accurately predict the FC-72 CHF data of the present study only when adjusted for the effect of subcooling.

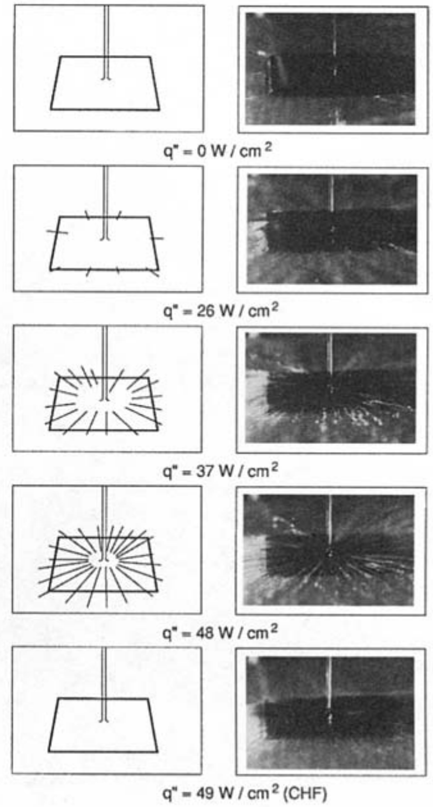

Fig. 5 Photographs and schematics of jet boiling on the chip surface using jet nozzle 1 with $Q = 4.04 \times 10^{-6}$ m³/s and $\Delta T_{sub} = 13$°C

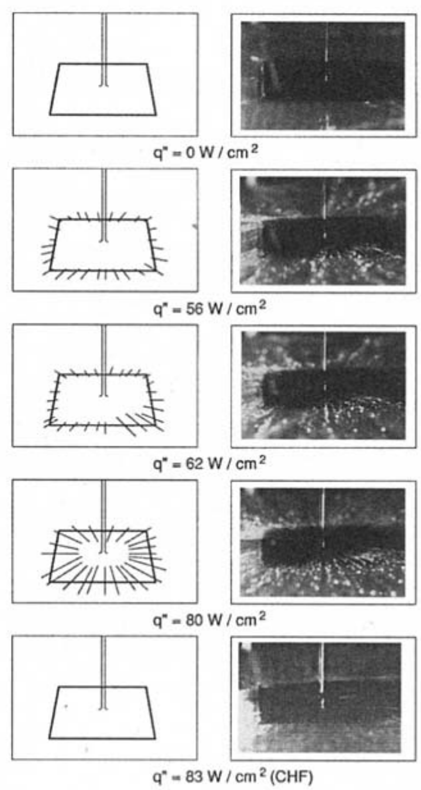

Fig. 6 Photographs and schematics of jet boiling on the chip surface using jet nozzle 1 with $Q = 4.04 \times 10^{-6}$ m³/s and $\Delta T_{sub} = 33$°C

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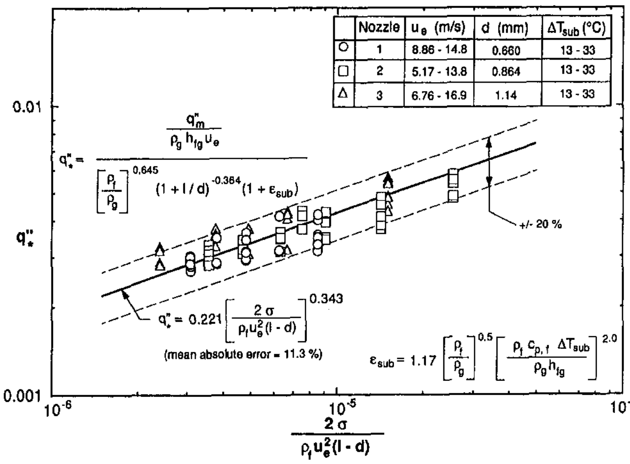

Fig. 7 Correlation of jet CHF data

$$\frac{q''_m}{\rho_g u_e h_{fg}} = 0.221 \left(\frac{2\sigma}{\rho_f u_e^2 (l - d)}\right)^{0.343} \left(\frac{\rho_f}{\rho_g}\right)^{0.645}$$
$$\times (1 + l/d)^{-0.364} \left[1 + 1.17 \left(\frac{\rho_f}{\rho_g}\right)^{0.5} \left(\frac{c_{p,f} \Delta T_{sub}}{h_{fg}}\right)^2\right] \quad (3)$$

where $l = \sqrt{2}L$. The small mean absolute error of this modified correlation, 11.3 percent, is proof of both its accuracy and effectiveness for electronic cooling system design. The subcooling factor in equation (3) is similar in form to the one recommended by Monde and Katto (1978),

$$\left[1 + 2.7 \left(\frac{\rho_f}{\rho_g}\right)^{0.5} \left(\frac{c_{p,f} \Delta T_{sub}}{h_{fg}}\right)^2\right],$$

but employs a smaller coefficient. In the subcooling factor, the exponent for ρ_f/ρ_g was preserved because Monde and Katto's study was conducted using more than one fluid, thereby producing large sensitivity to ρ_f/ρ_g, this density ratio was constant in the present study.

Practical Concerns. Perhaps the most alarming behavior observed with jet cooling was premature CHF which occurred during power transients. As indicated earlier, the CHF data used in developing the correlation presented above were obtained by increasing the chip power in very small increments. While this is the standard method for measuring CHF in boiling systems, it may not produce safe design limits for electronic cooling systems. An actual computer chip, whose thermal mass is in many cases smaller than the simulated chip used in the present study, often incurs large power fluctuations, and response of the jet to these fluctuations is of obvious interest. In the present study, it was observed that when the power was supplied to the chip in large increments, CHF occurred at heat fluxes as small as half the CHF measured by slowly incrementing the chip power. Visual observation revealed this reduction in CHF resulted from uncontrolled propagation of the boiling front toward the jet producing premature separation of the wall jet from the chip surface. Unanchored over most of the chip surface, the wall jet is free to separate from the surface due to the momentum of the vapor produced at the surface. Thus, any momentary increase in the vapor production due to a large power increment could bring about an unstable situation leading to complete wall jet separation and premature CHF.

3.2 Spray Cooling

Heat Transfer Results. The spray nozzles employed in the present study were of the Spraying Systems Unijet full cone series. This series was chosen for its range of flow rates, spray angle, and spray pattern. The nozzles selected were numbered 1 through 3 in order of increasing flow rate. The spray angles were relatively small, 46 to 63 degrees, but varied slightly with flow rate. The full cone spray pattern was produced by a vane inside the nozzle that induced controlled turbulence in the liquid prior to exiting the orifice.

For jets, the CHF data can be correlated relative to well defined characteristic velocity (jet velocity) and characteristic lengths (orifice diameter and heater size). Choosing appropriate scaling parameters for spray cooling is far more complex since the liquid is broken into a fine dispersion of droplets having different sizes, velocities, and trajectories prior to impacting the chip surface. Also, while jet cooling is fairly insensitive to nozzle distance from the chip surface, the spray coverage of the surface is very sensitive to both spray cone angle and the distance from the surface. These two issues contribute to great uncertainty in both predicting and implementing spray cooling.

The scaling parameters of sprays have been addressed in detail by Mudawar and Valentine (1989). They determined the appropriate length scale is the spray's Sauter mean diameter (SMD), d_{32}, which is defined as the diameter of the drop having a volume to surface area ratio that equals the volume to surface area ratio of the entire spray. In the present study, SMD was measured using an Aerometrics Phase Doppler Particle Analyzer (PDPA). Measurements made both along and away from the spray axis proved SMD was fairly uniform across the spray impact area.

Estes (1994) developed a correlation for SMD using the three full cone spray nozzles and three fluids, FC-72, FC-87, and water, according to a general form suggested by Lefebvre (1989). Liquid breakup was described as the net effect of turbulent forces evident in the liquid upstream of the nozzle orifice and aerodynamic forces downstream of the orifice. The correlation developed by Estes, Eq. (4), facilitates calculation of SMD using parameters which can be easily measured without the use of expensive particle sizing equipment.

$$\frac{d_{32}}{d_0} = 3.67 [\mathrm{We}_{d_0}^{1/2} \, \mathrm{Re}_{d_0}]^{-0.259}, \quad (4)$$

where d_0 is the diameter of the orifice and We_{d_0} and Re_{d_0} are, respectively, the Weber and Reynolds numbers based on orifice diameter and liquid velocity at the orifice prior to breakup.

$$\mathrm{We}_{d_0} = \frac{\rho_a (2\Delta P/\rho_f) d_0}{\sigma} \quad (5)$$

and

$$\mathrm{Re}_{d_0} = \frac{\rho_f (2\Delta P/\rho_f)^{1/2} d_0}{\mu_f}, \quad (6)$$

where ΔP is the pressure drop across the spray nozzle.

Mudawar and Valentine also determined the appropriate characteristic velocity for correlating spray CHF data is not the mean droplet velocity, but the volumetric flux which accounts for the cumulative effect of multiple droplet impact. The local spray volumetric flux, Q'', is defined as the volumetric flow rate for an infinitesimal target on the impact surface divided by the target area. The spatial average of Q'', $\overline{Q''}$, is simply the total volumetric flow rate of the spray divided by the portion of the surface directly impacted by the spray, the so called spray impact area.

For consistency between different spray tests, a criterion for optimum nozzle-to-surface distance was determined by collecting CHF values at identical flow rates and subcoolings for different nozzle-to-surface distances. Estes (1994) found CHF decreased significantly when the spray was concentrated at the center of the chip or when liquid was wasted by impinging outside of the chip surface. Optimum cooling (maximum CHF)

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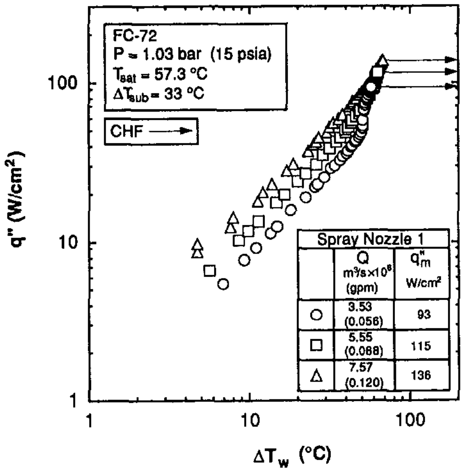

Fig. 8(*a*)

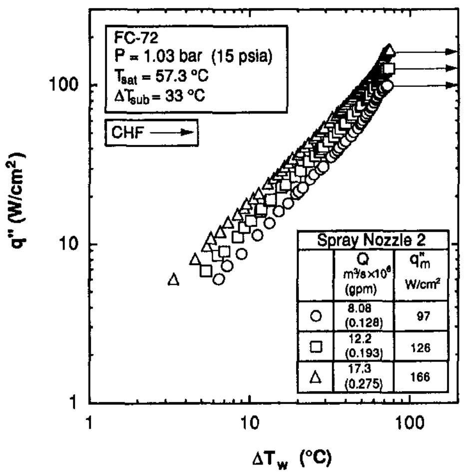

Fig. 8(*b*)

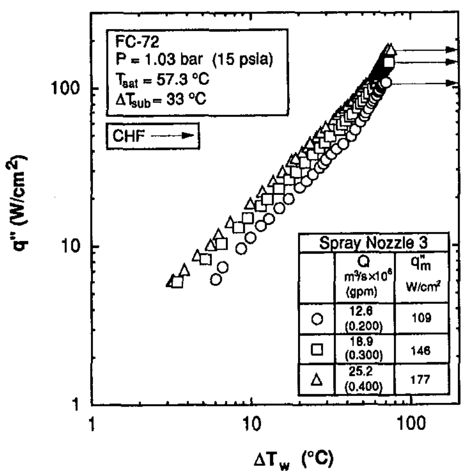

Fig. 8(*c*)

Fig. 8 Boiling curves for three flow rates at 33°C subcooling for (*a*) spray nozzle 1, (*b*) spray nozzle 2, and (*c*) spray nozzle 3

was attained when the spray just inscribed the chip surface. Therefore, all the subsequent heat transfer data were collected with the spray just inscribing the chip surface.

Figures $8(a)-8(c)$ show boiling curves for spray cooling using FC-72 at 33°C subcooling for spray nozzles 1, 2, and 3, respectively. These figures show both the single phase heat transfer and CHF increase with increasing flow rate. While a sharp increase in slope during nucleate boiling was evident at low flow rates, elevated flow rates did not exhibit this increase, a phenomenon rarely encountered in flow boiling situations. In fact, the boiling curves for the high flow rates show a very mild enhancement in cooling following incipience. This trend points to a cooling performance dominated by single phase heat transfer. This is particularly the case with the higher flow rates due to a large increase in the droplet number density.

Figures $9(a)-9(c)$ show, while increasing subcooling increases CHF for spray cooling, this increase is much smaller than the increase in CHF for jet cooling, Fig. $4(a)-4(c)$, over the same range of subcooling.

CHF Correlation. Mudawar and Valentine (1988) obtained, for water sprays, a data base for CHF from very small targets inside a large spray impact area. Critical heat flux ($q''_{m,p}$) and volumetric flux (Q'') were, therefore, both defined on a local (point) basis. The present data, on the other hand, consisted of sprays of FC-72 and FC-87, with impact areas totally inscribed within a square chip. Critical heat flux, q''_m, in the present study was measured as the chip power divided by the entire square surface area of the chip, while the average volumetric spray flux, $\overline{Q''}$, was measured as the total volumetric flow rate divided by the spray impact area ($\pi L^2/4$). The present data were correlated along with those of Mudawar and Valentine by defining critical heat flux and volumetric flux on a local basis assuming dryout at the edge of the impact area governs CHF for the entire chip surface. Critical heat flux at the edge, $q''_{m,p}$, was determined by assuming all of the heat was transferred through the spray impact area,

$$q''_{m,p} = \frac{q''_m L^2}{\frac{\pi}{4} L^2} = \frac{4}{\pi} q''_m. \tag{7}$$

The assumption of negligible cooling outside the spray impact area was verified experimentally by Mudawar and Valentine (1989) and Mudawar and Deiters (1994).

The volumetric spray flux at the outer edge of the spray impact area was determined from a spray flux distribution model detailed by Estes (1994).

$$\frac{Q''}{\overline{Q''}} = \tfrac{1}{2}(1 + \cos(\theta/2)) \cos(\theta/2), \tag{8}$$

where θ is the cone angle of the spray. Figure 10 shows the resulting correlation

$$\frac{q''_{m,p}}{\rho_g h_{fg} Q''} = 2.3 \left(\frac{\rho_f}{\rho_g}\right)^{0.3} \left[\frac{\rho_f Q''^2 d_{32}}{\sigma}\right]^{-0.35} \times \left[1 + 0.0019 \frac{\rho_f c_{p,f} \Delta T_{\text{sub}}}{\rho_g h_{fg}}\right] \tag{9}$$

fits the CHF data for water, FC-72, and FC-87 with a mean absolute error of 12.6 percent. Further details concerning the development of the spray CHF correlation can be found in another paper by the authors (Estes and Mudawar, 1995).

Practical Concerns. Perhaps the most serious drawback to spray cooling is its susceptibility to inconsistencies resulting from clogging or even minute manufacturing defects. Hall (1993) examined these problems by studying repeatability of spray characteristics (flow rate, pressure drop, SMD) over many weeks of testing. Even with a batch of new nozzles which carry the same part number, certain nozzles failed to reproduce the characteristics for that part number. He concluded sprays should always be checked for any inconsistency in flow pattern before use in an electronic cooling system. Hall also found brass nozzles showed virtually no repeatability in spray characteristics after only a few days of testing.

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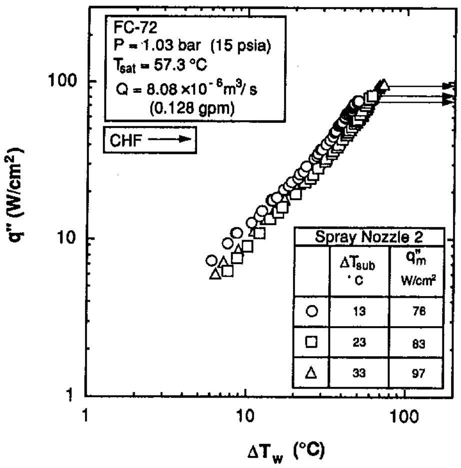

Fig. 9(*a*)

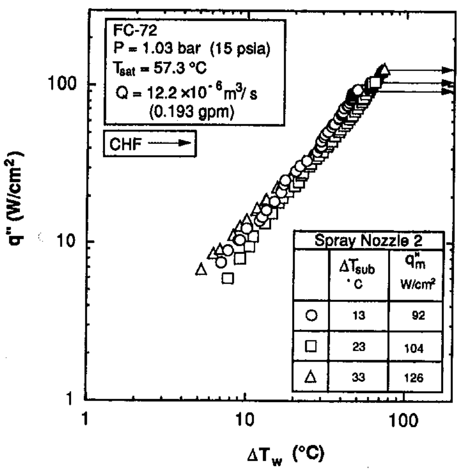

Fig. 9(*b*)

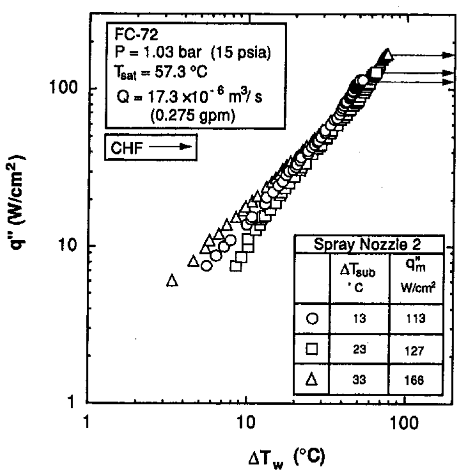

Fig. 9(*c*)

Fig. 9 Boiling curves for three subcoolings at (*a*) $Q = 8.08 \times 10^{-6}$, (*b*) $Q = 12.2 \times 10^{-6}$, and (*c*) $Q = 17.3 \times 10^{-6}$ m³/s for spray nozzle 2

	FC-72, P = 1.03 bar, T_{sat} = 57.3 °C			FC-87, P = 1.03 bar, T_{sat} = 30.7 °C	Water, P = 1.01 bar, T_{sat} = 100 °C
	Nozzle 1	Nozzle 2	Nozzle 3	Nozzle 1	Several full cone spray nozzles
	○	□	△	⊕	●
$Q'' \times 10^3$ m³s⁻¹/m²	16.6 - 52.4	55.6 - 145	86.3 - 216	16.6 - 49.7	0.600 - 0.216
$d_{32} \times 10^6$ m	140 - 196	181 - 225	182 - 214	110 - 195	405 - 1350
ΔT_{sub} °C	13 - 33	13 - 33	13 - 33	4 - 15	20 - 77

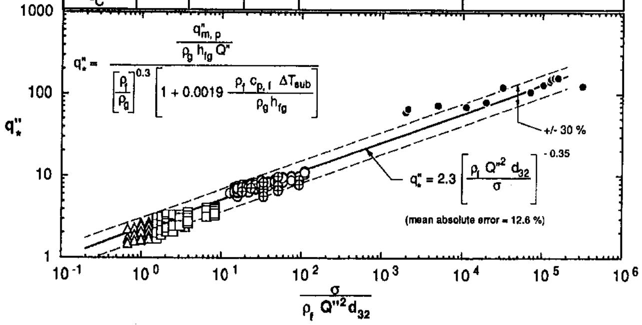

Fig. 10 Correlation of spray CHF data (Estes and Mudawar, 1995)

Jets concentrate much of the cooling in the impingement zone producing relatively large temperature gradients in the radial direction. The dispersed droplet impingement in spray cooling, on the other hand, diffuses cooling effectiveness over the entire spray impact area, resulting in a much more uniform surface temperature. A detailed discussion on the spatial temperature gradients in spray cooling can be found elsewhere (Mudawar and Deiters, 1994).

In general, single phase heat transfer was enhanced moderately with increasing volumetric flow rate for both jets and sprays. Critical heat flux was largely dependent on flow rate for both, however, CHF exhibited a larger dependence on subcooling with jets than with sprays. Figure 11 compares boiling curves for jet and spray cooling for 13°C subcooling and two flow rates. For equal flow rates, spray cooling provided better single phase heat transfer and significantly greater CHF values than jet cooling. Figure 12 compares CHF for spray and jet cooling for the high and low values of subcooling tested. At the larger subcooling, spray cooling provided only slightly larger CHF values than jets for a given flow rate. However, at

	Spray Nozzle 1	Spray Nozzle 2	Jet Nozzle 1	Jet Nozzle 2
	○	●	□	■
Q m³/s×10⁶ (gpm)	5.55 (0.088)	8.08 (0.128)	5.05 (0.080)	8.08 (0.128)
q''_m W/cm²	100	76	44	47

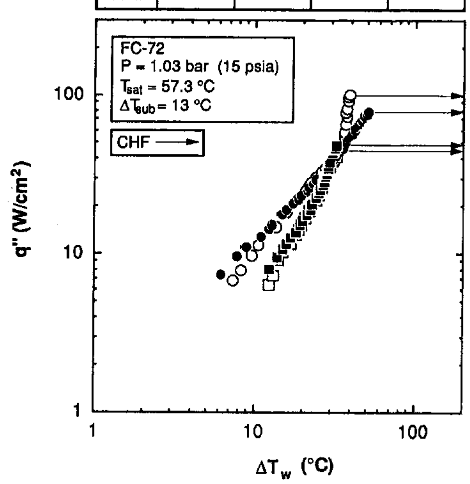

Fig. 11 Comparison of jet and spray boiling curves

It is therefore, recommended that only spray nozzles that are made from stainless steel or other corrosion and erosion resistant materials be used. It is also important that a good filtering system be employed to prevent clogging.

3.3 Comparison of Jet and Spray Cooling. One obvious advantage of spray over jet cooling which has not been addressed in the present study is surface temperature uniformity.

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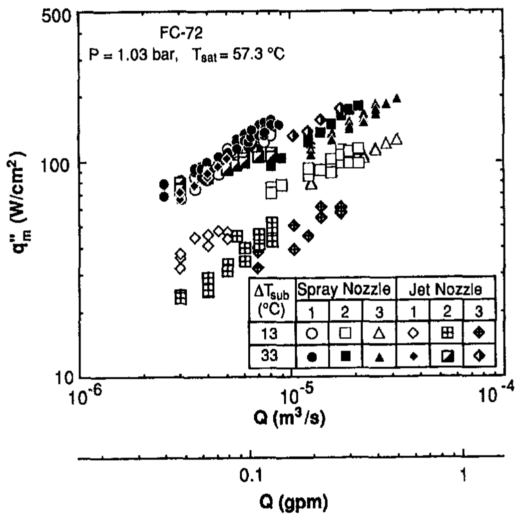

Fig. 12 Critical heat flux versus flow rate for jets and sprays

the lower subcooling, spray cooling yielded a much greater increase in CHF. One reason for the relatively poor CHF measured with jets at low subcooling is liquid film (wall jet) separation during vigorous boiling. Since the film is anchored to the chip surface only at the impingement zone, this separation can be triggered by any vapor momentum perpendicular to the surface. In contrast, sprays produce a dispersion of drops which anchor the liquid film over the entire spray impact area, thus resisting liquid separation even at low subcooling.

Weak attachment of the liquid film with the chip surface is also the reason for the premature CHF encountered in jet cooling during relatively large increases in chip power. CHF values in spray cooling were very reproducible, regardless of the size of power increment.

4 Conclusions

Experiments were performed with both jet and spray cooling in order to explore effects of the key parameters influencing thermal performance and to develop CHF correlations for each. The key conclusions from the study are as follows:

Jet Cooling. (1) Both single phase heat transfer and CHF are enhanced with increasing jet velocity and increasing jet diameter. Subcooling greatly increases CHF because a combination of weak attachment of the wall jet and large momentum of the bubbles could easily separate the liquid from the chip surface at low subcooling. It is, therefore, recommended that two-phase jet impingement cooling be confined using configurations such as those of Wadsworth and Mudawar (1992) and Johns and Mudawar (1995).

(2) Boiling always commences on the outer edges of the chip surface and propagates inward with increased heat flux. A larger fraction of the surface undergoes boiling prior to CHF at low as compared to high subcooling. CHF is accurately predicted by modifying a correlation developed by Monde and Inoue (1991) to account for the effect of subcooling.

(3) Dryout occurs on the chip surface prematurely when chip power is increased too quickly. The sensitivity of this premature dryout to the heater's thermal mass precludes any definitive assessment of CHF during transient heating in real electronic cooling systems employing jet impingement.

Spray Cooling. (1) Optimum cooling performance can be obtained when the spray impact area just inscribes the chip surface.

(2) Better single-phase cooling performance and higher CHF values can be attained with increasing spray flow rate or increasing subcooling, but the effect of subcooling on CHF is less pronounced than with jet cooling. The large contribution of single heat transfer in spray cooling results in an unusual boiling curve, especially for large flow rates, where the slope of the boiling curve shows little change between the single phase and nucleate boiling regimes. A new correlation was developed which predicts CHF accurately for wide ranges of spray flow rates and subcoolings and three different fluids.

(3) CHF is very repeatable with spray cooling even during chip power fluctuations. Therefore, system safety against burnout can be ascertained with much greater accuracy with spray cooling than with jet cooling. However, spray nozzles are more prone to inconsistent spray characteristics, erosion, and clogging, and should, therefore, be carefully examined prior to use in electronic cooling systems, and well maintained during the expected life of the cooling system.

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