

A Uniform Temperature, Ultra High Heat Flux Liquid Cooled, Power Semi-Conductor Package

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Abstract - A new type of heat exchange process is described for use in semiconductor heat sinks. This involves the use of subcooled, nucleate boiling at concave curved surfaces where radial acceleration, v^2/r , can be used to develop significant and beneficial buoyancy forces. This system provides a heat transfer surface with a uniform temperature, i.e., the boiling point of the fluid, and requires no vapor-liquid separation process since all vapor bubbles are immediately condensed in the subcooled liquid. Use of low boiling point dielectric fluids, e.g. FC 72 (BP 56°C), establishes a uniform heat sink temperature that is independent of location or environment and does not require elaborate coolant and/or environmental conditioning. The low junction to fluid specific thermal resistance, $\Theta_{jf} \sim 0.1^\circ \text{C/Wcm}^2$, and high heat flux dissipation capability, $\sim 6,000 \text{ W/cm}^2$, inherent in this design lends itself to a system design with high reliability and lower manufacturing costs. Failure rates are reduced and leakage currents are lowered by virtue of lower device operating temperatures, i.e. low boiling point coolants. Elimination of dry contacts and large mechanical forces, e.g. Disc Type Package, substantially reduces life shortening thermal stresses and lowers thermal resistance. The design lends itself to compact lightweight construction for high voltage series operation of devices and permits simplified parallel operation of high power devices. Experimental studies at Coriolis Corporation have demonstrated a heat flux of 6300 watts/cm² using FC77 with a pressure drop of only 2.70 pounds per square inch, over a one centimeter long heat transfer channel. In this paper we describe the design, construction, and operation of a liquid-cooled high power semi-conductor package and cooling technique for improved device reliability, simplified parallel operation of multiple devices, and efficient high voltage series construction.

I. INTRODUCTION

The performance of high power devices such as diodes, thyristors, GTO's, MOSFETS etc. is often limited by the heat removal capabilities of the thermal package. To achieve reasonable heat flux removal rates, device packages are often bulky and heavy, and this can complicate the packaging of these devices for series, parallel or series-parallel operation. For parallel operation, the need to maintain uniform junction temperatures is further complicated by those applications subject to surge-current overloads that can cause hot spots within a device.

A common method for power device cooling is the use of high mechanical forces (up to 10 tons) to press the Disc Type package against the external heat sink which employs massive and bulky clamping springs and insulators. The sliding contacts resulting from this method have relatively high thermal resistance so the performance of the device in both steady state and surge is limited, and this construction additionally suffers from degradation of thermal fatigue life (Glascocock and Webster, 1983)[1].

In this paper we propose a novel packaging and cooling scheme that addresses the shortcomings of conventional packaging and cooling techniques for high power devices, e.g. Disc Type Package.

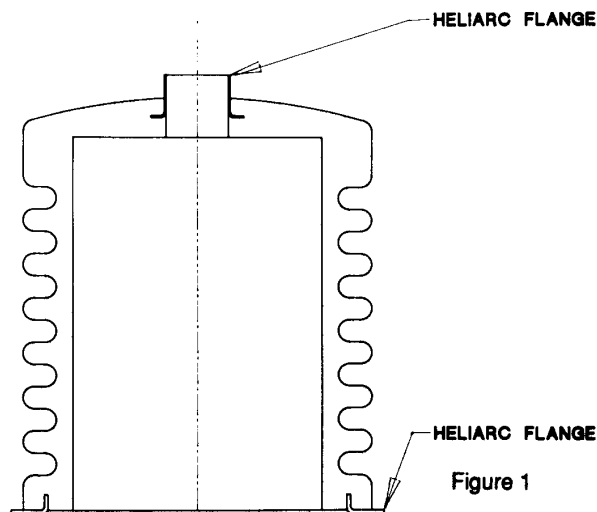
A. Benefits Include:

- 1) Operation of devices at higher ratings, e.g. substitution of 53mm devices for 77mm or 100mm devices with attendant cost savings.
- 2) Longer device life by virtue of lower junction temperatures [24] and reduced leakage, as provided by low thermal resistance construction and low boiling point coolants.
- 3) Improved system reliability as provided by subcooled nucleate boiling heat transfer wherein the heat exchange surface

is pinned at the boiling temperature of the coolant thereby providing uniform device junction temperatures system wide, from cabinet to cabinet and room to room;

- a) Simplifies parallel operation of devices.
 - b) Thermal resistance change with transient or steady state operation should be minimal and confine substantially to that due to the silicon temperature rise by virtue of the elimination of mechanical pressure contacts and their associated degradation of thermal fatigue life.
 - c) Combined with (2) provides suppression of hot spots.
 - d) Combined with (2) enables sustained surge current operation.
- 4) Compact, light weight construction results from efficient heat transfer from the silicon chip substrate, e.g. negligible lateral heat transfer resulting in a heat transfer surface, i.e. the substrate, that is the same size as that of the chip. The dielectric coolant is an excellent insulator, e.g. 38 KV/0.1" for FC72 and therefore package size is principally dictated by external electrical breakdown characteristics of the housing. For example, 60 Thyristors (4500V/2000A) could be seriesed to provide 120KV/2000A operation. It is estimated that package size might be about 30" long, 8" diameter and weigh about 50 lbs. Curved surface cooling could increase device rating to 8,000 A at the expense of the long term surge current safety margin.
- 5) Cost effective design is provided on two levels; more efficient heat transfer permits higher device ratings, that is a 53mm device can be substituted for a 77mm or 100mm device, e.g. a doubling or quadrupling of heat flux removal, alternatively or in combination, substantial performance margin can be achieved to permit sustained operation under surge current conditions. Compact design reduces system space requirements and uniform junction temperatures simplifies fail-safe circuit design. Incorporating a large number of devices in a hermetically sealed housing eliminates the individual device packaging costs, e.g. ceramic housing and associated component and labor costs.

II. CONSTRUCTION



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Figure 1 illustrates an insulating structure to house an assembly of power devices. The housing may be cast from a suitable epoxy such as is employed for HV insulators. Alternative insulating materials of construction include ceramic or glass. Flanges for heliarc welds are employed at both the positive and negative terminals to provide hermetic seals.

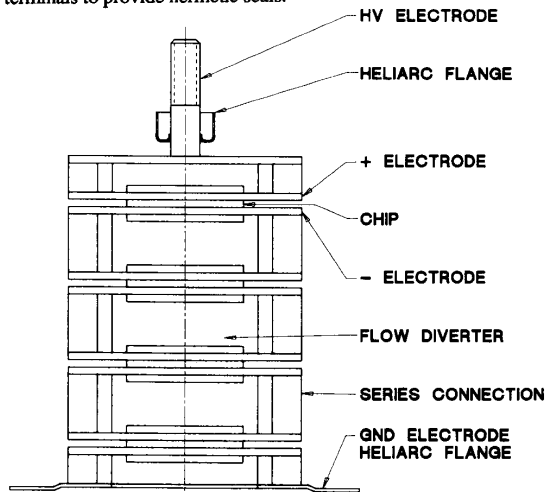


Figure 2 illustrates a preassembled stack of power devices shown for convenience as electrically connected in series. Alternative connections are parallel or a combination of series-parallel connections as might be employed for fail-safe operation.

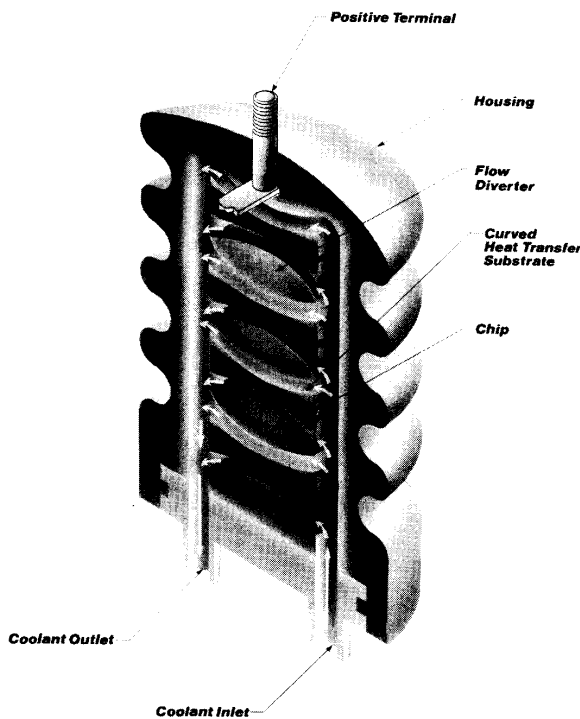


Figure 3 illustrates an angled cross section view of the stack of power devices of Fig. 2 mounted in the housing of Fig. 1. Coolant flow is shown progressing up the input coolant conduit and then branching off at 90° to progressively feed, in parallel, each of the

curved surface coolant channels. The coolant, being a dielectric, contributes to the electrical insulation of the package, e.g. 38KV/0.1" for FC72 fluorocarbon. In addition to cooling the device, coolant channels may be provided to divert a predetermined coolant flow over the device electrical interconnects to remove resistive heat losses. For clarity of illustration of the curved surface cooling concept, single curved surface cooling is shown. For power devices of large dimensions, periodic curved surface cooling as described in U.S. Patent #4,712,609 [23] would be employed.

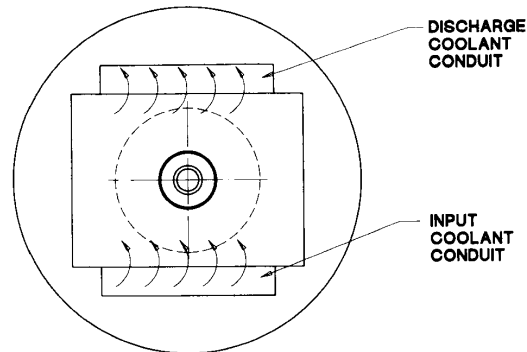


Figure 4 illustrates a top-down, cross section view of the packaged devices to show input and discharge coolant channels and the method for alignment of the preassembled power device stack within the housing.

A. Device Cooling

Conventional high power device cooling, e.g. Disc Type Packages, are reputed to have maximum heat flux removal rates of about 150 W/cm² and utilize massive and bulky clamping springs and insulators. Curved surface cooling has demonstrated 17 KW/cm² (Gambill & Greene, 1958)[2], 24 KW/cm² (Leslie et al 1983)[3] and 6.3 KW/cm² (Iversen and Whitaker 1984)[4]. Sub-cooled nucleate boiling heat transfer for flow over concave curved heat exchange surfaces offers substantial advantages. Curved surface sub-cooled nucleate boiling at the device heat exchange surface acts effectively as an infinite heat sink thereby maintaining a substantially constant heat sink temperature over a broad range of device heat fluxes from device to device and package to package independent of environment or location. Junction temperature is then the product of the heat flux and thermal resistance of a thin high thermal conductivity substrate, e.g. tungsten, plus the boiling point of the coolant, to which is added a superheat which depends upon the heat flux. Low boiling point fluorocarbon coolants, e.g. FC 87(32° C), or FC 72(56° C), enable high heat fluxes to be obtained while maintaining junction temperatures within acceptable limits. Thus, the heat transfer characteristics of the heat sink are no longer limited by mechanical contacts and thick heat sink members.

Device junction-to-fluid thermal resistance, O_{jf} is strongly influenced by the construction technique utilized. By employing diffusion bonded device construction, Glascock [1] demonstrated a substantial drop in junction -to -case thermal resistance, O_{jc} , over dry contact construction while still using Disc Type Package heat sinking. With liquid cooling, the dry sliding Disc Type Package heat sink contact is replaced with one of high compliance, negligible pressure and total contact for optimum performance. To simplify calculations and to provide a conservative design, the thermal resistance is calculated for single sided cooling. With double sided cooling, e.g. the use of structured copper, thermal resistance of about 60% of that for single sided cooling may be obtained. Using Nuegebauer's [13] junction to fluid specific thermal resistance O_{jf} of 0.1° C/Wcm² for bonded structures resulting in a thermal resistance of ~0.002° C/W for a 77mm device and ~.0012° C/W for a 100 mm device would provide a junction temperature rise ΔT of 15° C at the conventional maximum heat flux of 150 W/cm². Using FC72 (BP 56° C) as a coolant would result in a conservative junction temperature of 71° C (56° C + 15° C) plus superheat resulting in

lower leakage currents and enhanced device life. With double sided cooling, the thermal resistance of the 77mm devices would reduce to about 0.0012° C/W, as contrasted to greater than 0.01° C/W for Digital Type Package construction. Thermal resistance change with transient or steady state operation should be minimal and confined substantially to that due to the silicon temperature rise by virtue of the elimination of mechanical pressure contacts. Direct liquid cooling eliminates Disc Type Package related thermal stress and associated device life degradation.

B. Coolant Compatibility

With respect to coolant compatibility, the use of pure fluorocarbons, e.g. 3M, FC72 (BP 56° C) or FC87 (BP 32° C) appears to present no difficulties for device cooling. In an ongoing experiment, 3M has been operating bare chip devices in FC72 for over 4 1/2 years with no difficulties[5]. The Cray 3 will employ 3M fluorocarbon bare chip cooling. Virtually all prior work reported with fluorocarbon cooling has been with the chloro-fluorocarbon R-113 (BP 47° C). R113 has been reported as reacting with aluminum, copper and other similar metals, Eriksson et al[6]. The authors experience has been similar to the above with respect to copper and zinc in addition to experiencing corrosion of 400 series stainless steel. The pure fluorocarbons do not seem to present any difficulties for packaged or bare chip cooling nor do they appear to present a threat to the environment.

C. Reliability

A recent field study by the military provided a correlation between failure rate models and real life situations. From the data, device temperatures of 50-100° C are acceptable and 100-125° C are marginal [24]. Employing the temperature related failure rate table of Corman et al. [24] a comparison of relative failure rates for Disc Type Packages and curved surface cooling can be made. At a 110° C junction temperature for Disc Type Packages, the failure rate would be 10 times higher for CMOS and 4 times higher for BiPolar as compared to a 71° C (FC72) junction temperature, i.e. 150 W/cm², with curved surface cooling. At a 47° C (FC87) junction temperature, Disc Type Package failure rate would be 38 times higher for CMOS and 13 times higher for BiPolar as compared to curved surface cooling. At a junction temperature of 125° C, the Disc Type Package failure rate increases to 20X for CMOS and 6.4X for BiPolar over 71° C (FC72) curved surface cooling. Compared to 47° C operation (FC87), it is 80X higher for CMOS and 20X for BiPolar. A further failure mode is caused by thermal stresses which arise from 1) the clamping forces employed in Disc Type Package heat sinking, and 2) the high temperature excursions that Disc Type Packages undergo. In the proposed design, (1 above) is eliminated and (2 above) is minimized.

D. Extended Surge Current Tolerance

By increasing the coolant velocity and using a lower boiling point coolant, e.g. FC87 (BP 32° C), high surge currents can be tolerated for extended periods. For example, using FC87 (BP 32° C) and a 125° C max junction temperature, the permissible ΔT temperature rise is 93° C. Assuming 10° C for super-heat, the junction to heat exchange surface ΔT is 83° C. Using the junction to heat exchange surface specific thermal resistance O_{jc} of 0.1 C/W cm², the dissipated power is then 830 W/cm², an almost 6X increase over the Disc Type package maximum heat flux of 150 W/cm². For a 77mm device, this corresponds to a continuous dissipation of 38KW. With double sided cooling, the overall specific thermal resistance would decrease to about 0.06° C/W cm² thereby increasing the power dissipation for a ΔT of 83° C to about 1,380 W/cm² resulting in a 9X increase in the continuous surge current rating over standard operational ratings, e.g. a 2,000A rated device (150 W/cm² dissipation) could be operated at a sustained surge current of 18,000 A (1,3980 W/cm² dissipation). It should be noted that the above value of 1,380 W/cm² is well below the 6 KW/cm² demonstrated in the Iversen-Whitaker experiment[4].

E. Subcooled Nucleate Boiling

The process of subcooled, nucleate boiling is an especially efficient means of heat removal. Nucleate boiling traditionally produces extremely high heat fluxes, and when the liquid is subcooled we have the advantage of a heat transfer system in which

only a single phase is present in the entrance and exit streams. In subcooled boiling, the vapor bubbles leave the heat transfer surface and then collapse in the liquid as illustrated in Figure 5.

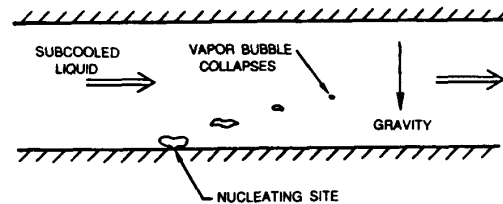


Fig. 5 Bubble Generation and Collapse During Subcooled Boiling

The high rates of heat transfer that occur during nucleate boiling are caused by the intense liquid motion that is generated by the growth and subsequent departure of the vapor bubbles from the heat transfer surface. These vapor bubbles, which generate the desired fluid motion, are also the source of the boiling crisis known as the critical heat flux or "burnout" condition. As the heat flux rises, the number of active nucleating sites (one of which is indicated in Figure 5) increases and the concentration of vapor bubbles in the neighborhood of the heat transfer surfaces increases. When sufficient bubble coalescence occurs, a stable vapor film develops and this usually leads to system failure. A high rate of vapor bubble removal from the heat transfer surface is the key to an efficient heat transfer process, and there are two principal mechanisms which can be used to enhance the rate of removal of vapor bubbles: 1) the effect of shear and turbulent dispersion caused by a forced flow of liquid past the heat transfer surface, and 2) the effect of an increased body force caused by an acceleration of the liquid normal to the heat transfer surface. This increase in the body force can be generated by flow past a concave surface.

The effect of forced flow on the critical heat flux has been reviewed by Katto [21] for a wide variety of configurations. While the critical heat flux depends on the flow configuration and the nature of the heated surface, one can conclude that the critical heat flux is roughly proportional to the square root of the liquid velocity, i.e.,

$$q_c \sim v^{1/2}, \text{ forced flow} \quad (1)$$

Clearly this is an approximation since experimental results can be found for which q_c is proportional to $v^{1/3}$ and for which q_c is proportional to $v^{2/3}$. For our purposes the representation given by Equation 1 is very attractive since it nicely describes the work of Gambill and Greene [2] and the follow-up study of Mayersak et al [7]. The effect of shear flow on the boiling process is extremely complex and not yet completely understood. However, it is clear that a turbulent shear flow tends to break vapor bubbles away from the heat transfer surface and transport them into the surrounding liquid. This enhances the boiling process and gives rise to an increase in the critical heat flux as indicated by Equation 1. While the critical heat flux must be avoided in the operation of most systems, Equation 1 is still quite useful since it describes the effect of a shear flow.

The influence of buoyancy effects on the nucleate boiling process is clearly evident in the process of "pool boiling" such as one observes when a pan of water is heated on a stove. In that case the buoyancy force, generated by the difference in the vapor and liquid densities and the gravitational force, causes the vapor bubbles to be propelled upwards away from the heat transfer surface. As early as 1950 Kutateladze [8] was able to deduce that the critical heat flux during pool boiling was proportional to the fourth root of the acceleration normal to the heat transfer surface, i.e.,

$$q_c \sim (a/g)^{1/4}, \text{ pool boiling} \quad (2)$$

Here a represents the acceleration and g is the standard gravitational acceleration, i.e., one "gee". Kutateladze's result was corroborated by the hydrodynamic stability analysis of Zuber [9] and confirmed by the experimental studies of Usiskin and Siegel [10]. The implications of Equation 2 for micro-gravity environments are obvious, and a survey of this general problem was presented by Siegel [11] in 1967.

F. Flow Past Curved Surfaces

When a liquid flows past a concave surface at which nucleate boiling occurs, two flow phenomena influence the critical heat flux. First, the boiling process is enhanced because of the shear generated by the tangential fluid motion as illustrated in Figure 5 and suggested by Equation 1. In addition, the radial acceleration, $a = v^2/r$, gives rise to a buoyance force which propels the vapor bubbles away from the heat transfer surface as illustrated in Figure 6.

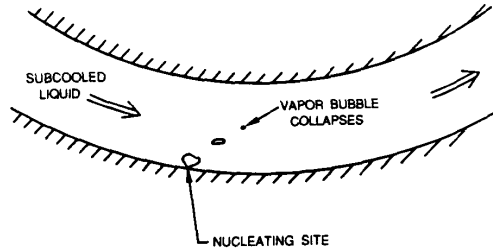


Figure 6 Bubble Generation and Collapse During Subcooled Boiling in a Curved Channel.

This effect was first investigated by Gambill and Greene [2] who used a swirl flow to achieve a critical heat flux of 17 kw/cm². This represented the highest known heat flux in 1958 and it clearly indicated that curved surface flow could be used to generate an exceptionally efficient boiling process. This exceptional efficiency results from the fact that the liquid velocity is used to generate a buoyance force that is normal to the heat transfer surface, whereas the drag force generated in linear flow is tangent to the heat transfer surface. Thus, with curved surface flow the vapor bubbles are subject to a force which propels them directly away from the heat transfer surface and into the subcooled liquid where they collapse.

The original discovery of Gambill and Greene [2] has been utilized by researchers at Hewlett-Packard Laboratories (Leslie et al [3]) to develop an enhanced brightness X-ray source. Using a cone-shaped anode with a concave radius of curvature of 2 centimeters and a fluid velocity of 10,000 cm/s, Leslie et al were able to obtain a heat flux of 25 kw/cm². However, this was not a critical heat flux and they suggested that their curved surface device was capable of producing a heat flux two to three times larger than the measured value of 25 KW/cm². In 1984, experiments were carried out by Iversen and Whitaker [4] using FC 77 as a coolant and a curved channel similar to that illustrated in Figure 6. With a radius of curvature of 2.5 centimeters and a velocity of 1340 cm/s ($a/g = 726$), a "near-burnout" heat flux of 6.3 kw/cm² was obtained.

On the basis of the studies of Gambill and Greene [2] and our own experimental work, we have developed an empirical equation for the critical heat flux for FC 77. This is given by

$$q_c = 3.6 \times 10^{-4} v^{1/2} \Delta T_{\text{sub}} [1 + (a/g)^{1/4}] \quad (3)$$

where q_c is in kw/cm², v is in cm/s, and the subcooling, ΔT_{sub} , in °C. When the radial acceleration is negligible, Equation 3 reduces to the form given by Equation 1. When a/g is large compared to one, we have

$$[1 + a/g^{1/4}] = (a/g)^{1/4} = (v^2/rg)^{1/4} \sim v^{1/2} \quad (4)$$

and under these circumstances the critical heat flux becomes a linear function of the velocity

$$q_c \sim v, \text{ forced flow past a concave surface} \quad (5)$$

A comparison of this result with Equation 1 clearly indicates a dramatic improvement in the efficiency of the boiling process, and this improvement is directly related to the radial acceleration that is generated by concave surfaces.

While the heat fluxes reported by Gambill and Greene [2], Leslie et al [3], and Iversen and Whitaker [4] are much higher than those

currently encountered in semiconductor cooling processes, the efficiency of curved surface nucleate boiling is not to be ignored. In the next section we indicate how this efficiency can be successfully utilized.

G. Semiconductor Cooling

In Figure 3 we have shown a power semiconductor cooling configuration [12]. The semiconductor chips are bonded to the substrated heat sinks with each having its own curved heat transfer surface. Alternatively, structured cooper or a suitable silicon substrate could be employed. The substrate shown is molybdenum or tungsten which provides a low thermal resistance and a good match in terms of coefficients of thermal expansion. From Table III of Neugebauer et al [13] we find the thermal resistance of the chip-solder-molybdenum assembly to be 0.110°C/(W/cm²). As an example of the use of this type of system, we consider the following conditions.

$$\text{fluid} = \text{FC 77}, \text{ radius of curvature} = 3 \text{ cm} \quad (6)$$

$$\text{fluid velocity} = 54 \text{ cm/s}, \Delta T_{\text{sub}} = 70^\circ \text{C}$$

For these conditions, which provide $a/g = 1$, we can use Equation 3 to predict a critical heat flux of

$$q_c = 370 \text{ watts/cm}^2 \quad (7)$$

If the heat flux at a semiconductor chip is 200 watts/cm², one could expect to safely operate under the conditions represented by Equation 6. The corresponding critical heat flux for a linear flow system ($a/g = 0$) would be 185 watts/cm² and such a system would burnout if the operating heat flux were 200 watts/cm².

The thickness of the flow channel illustrated in Figure 2 is a parameter in an optimization analysis; however, we have found that a channel depth on the order of 0.2 cm is generally acceptable. For this channel depth and a velocity of 54 cm/s, the pressure drop per centimeter of channel length is 0.004 psi when the channel is assumed to be hydraulically smooth. Under these circumstances a one meter long flow channel of the type illustrated in Figure 32 will give rise to a pressure drop on the order of one half pound per square inch. From these calculations we see that the pressure drop in the heat transfer channel is not a matter of serious concern. To emphasize this point, we remark again that experiments have been performed (Iversen and Whitaker [4]) in which a heat flux of 6300 watts/cm² was obtained in a curved channel system for which the pressure drop was 2.70 psi per centimeter.

As we mentioned earlier, it is the intense fluid motion generated by bubble growth and departure from the heat transfer surface that leads to the high heat fluxes associated with nucleate boiling. This intense fluid motion can also be obtained by the use of sufficiently high liquid velocities. A dramatic example of this is supplied by the work of Tuckerman [14] who used the classical method of extended surface heat transfer (Whitaker [15]) to obtain a heat flux of 790 watts/cm² using water in laminar flow. The absence of boiling would appear to give rise to a simpler heat transfer system; however, it leads to significant surface temperature gradients in the direction of flow, high pressure drops, and relatively high temperature driving forces. In Table I we have shown the experimental result for water from Tuckerman's thesis [14], the conversion of these results to FC 77, and the results predicted by Equation 3. Both the surface temperature rise in the direction of

flow and the pressure drop associated with Tuckerman's configuration indicate that laminar flow heat transfer is very inefficient compared to nucleate boiling at a curved surface. In calculating the Coriolis results listed in Table 1, we have used a radius of curvature of 3 cm, a liquid velocity of 143 cm/s, and a subcooling of 70°C. While our experiments and calculations have been carried out with FC 77, we expect that semiconductor cooling would be carried out with FC 72 which boils at 56°C. At a heat flux of 200 watts/cm² and a thermal resistance of 0.110°C cm²/watt, the temperature drop across the chip-solder-molybdenum assembly would be 22°C. If the steady-state superheat in the liquid were 5°C, the maximum chip temperature would be 83°C. If lower

Table 1 Comparison Between Laminar Flow cooling and Nucleate Boiling at a Curved Surface

	Tuckerman (Run#81F9)		Coriolis
	Water	FC 77	FC 77
surface temperature change in the direction of flow per centimeter	22°C	49°C	negligible
temperature driving force	35°C	108°C	~10°C
heat flux	790 watts/cm ²	790 watts/cm ²	790 watts/cm ²
pressure drop per centimeter	22 psi	36 psi	0.022 psi

temperatures are required one could use the fluorocarbon R11 which boils at 24° C or FC87 (32° C BP).

H. Temperature Overshoot

For most systems the superheat required to maintain steady nucleate boiling is on the order of 5-10° C; however, the transient superheat or "temperature overshoot" may be considerably larger. This effect was apparently first observed by Cory and Foust [16] in a study of pool boiling using ether, normal pentane, and Freon 113. They observed temperature overshoots on the order of 10-15° C, and their results were comparable to those obtained by Bankoff et al. [17] using methanol. A detailed study of temperature overshoot was carried out by Bergles and Chu [18] for porous metallic surfaces, and they found that the temperature overshoot for Freon R-113 was about double that of water for pool boiling conditions. The maximum observed temperature overshoot for R-113 was about 8° C.

Bar-Cohen and Simon [20] have pointed out that temperature overshoot can present a serious problem with the use of dielectric fluids such as FC77. Since the fluorocarbons characteristically have a low surface tension and contact angle they can penetrate deep into the pores and cavities that exist on any solid surface. In the absence of non-condensable gases, a re-entrant cavity must exist in order to produce a stable nucleating site. It seems clear that fluids with low surface tension and low contact angle will fill many potential nucleating sites, thus leaving only a small fraction of re-entrant cavities to provide nucleating sites. This situation is illustrated in Fig. 7 where we have shown a re-entrant cavity as a nucleation site. In the absence of non-condensable gases, the pressure of the vapor in the cavity can be expressed as

$$P_{vap} = P_{liq} - 2\sigma/R_n \quad (8)$$

Here P_{vap} represents the vapor pressure which can be determined as a function of the temperature by the Clausius-Clapeyron equation, P_{liq} represents a local ambient pressure in the liquid, and σ is the surface tension. The radius of curvature of the gas-liquid interface shown in Fig. 7 is designated by R_n , and it is important to note that the effect of surface tension is to reduce the pressure of the vapor in the cavity. It is this reduction of the pressure that generates a stable nucleating site.

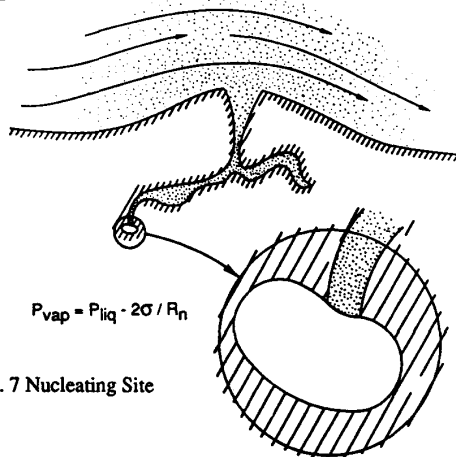


Fig. 7 Nucleating Site

When the temperature of the solid is increased, the pressure of the vapor will increase and the region occupied by the vapor will expand until the vapor-liquid interface reaches the position shown in Fig. 8. Under these circumstances, the effect of surface tension is reversed and the pressure of the vapor is given by

$$P_{vap} = P_{liq} + 2\sigma/R_o \quad (9)$$

The increase in the pressure of the vapor in going from the configuration shown in Fig. 7 to that shown in Fig. 8 is given by

$$\left\{ \begin{array}{l} \text{increase in vapor pressure} \\ \text{in going from the nucleation} \\ \text{site to the temperature} \\ \text{overshoot site} \end{array} \right\} = 2\sigma \left(\frac{1}{R_o} + \frac{1}{R_n} \right) \quad (10)$$

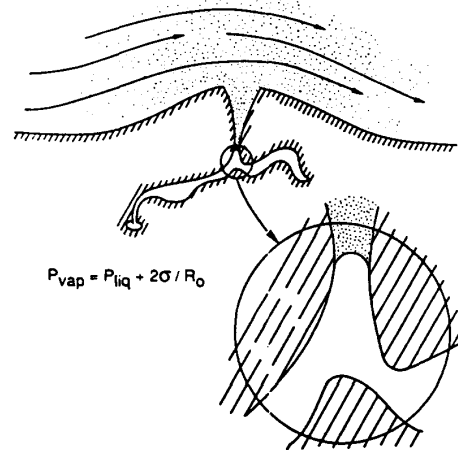


Fig. 8 Temperature Overshoot Site

The temperature associated with this change in pressure will be described by the Clausius-Clapeyron equation, and if R_o is relatively small one can expect a large temperature overshoot. A further increase in the temperature will force the vapor-liquid interface beyond the constriction illustrated in Fig. 8 and the steady superheat condition shown in Fig. 9 will be achieved. Under these circumstances the pressure of the vapor will be given by;

$$P_{vap} = P_{liq} + 2\sigma/R_s \quad (11)$$

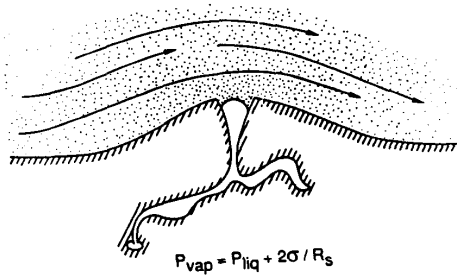


Fig. 9 Steady Superheat Site

however, one must remember that the condition illustrated in Fig. 9 is a dynamic condition and the radius of curvature, R_s , should be thought of as some appropriate average value. An equation analogous to Eq. 10 can be written as

$$\left\{ \begin{array}{l} \text{decrease in vapor pressure} \\ \text{in going from the temperature} \\ \text{overshoot site to the steady} \\ \text{superheat site} \end{array} \right\} = 2\sigma \left(\frac{1}{R_o} - \frac{1}{R_s} \right) \quad (12)$$

and a linearized version of the Clausius-Clapeyron equation can be used with this result to estimate the temperature overshoot as

$$\Delta T \sim \frac{2\sigma}{P_{liq}} \left(\frac{1}{R_o} - \frac{1}{R_s} \right) \frac{RT^2}{\Delta H_{vap}} \quad (13)$$

When R_o is small we see that the temperature overshoot, ΔT , will be large. Bar-Cohen and Simon [20] have presented some results for the influence of fluid velocity on ΔT , and those results suggest that the temperature overshoot is reduced with increasing velocities. At high velocities (17 m/s) they provide one example of a negligible temperature overshoot. Their observations suggest that the curved surface effects illustrated in Fig. 6 and represented in Eq. 3 are also at work at the scale of the surface roughness.

The interaction between a turbulent flow and the surface roughness has been reasonably well understood since the original experimental study of Nikuradse [22]. A simplistic description of turbulent flow in a closed conduit would separate the flow into a turbulent core and a laminar sublayer Whitaker[19]. As the Reynolds number

$$Re = \rho v D / \mu \quad (14)$$

increases, the thickness of the laminar sublayer (more precisely designated as the viscous sublayer) diminishes and the turbulent core begins to interact directly with the surface roughness. This situation is illustrated in Fig. 10. When the thickness of the laminar sublayer is less than the height of the surface roughness, the force that the fluid exerts on the wall is directly proportional to the fluid inertia, i.e.

$$\text{force} \sim \rho v^2 \quad (15)$$

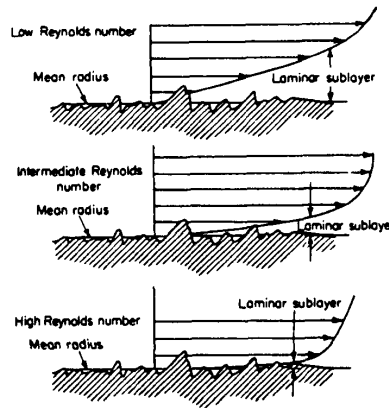


Fig. 10 Qualitative Description of Wall Effects in Turbulent Flow

This region of flow is known as the rough pipe region (Whitaker, Sec. 8.2, [19]). On the basis of numerous experimental studies of flow in pipes the thickness of the laminar sublayer can be estimated by

$$\delta \sim 50D/Re^{7/8} \quad (16)$$

This result can be used to construct an estimate of the local velocity which is given by

$$v_{loc} \sim \left(\frac{y}{\delta} \right) v, \quad y \leq \delta \quad (17a)$$

$$v_{loc} \sim v, \quad y > \delta \quad (17b)$$

Here y represents the distance measured from the mean radius illustrated in Fig. 4 and v represents the velocity in the turbulent core.

It should be clear at this point that the cavity illustrated in Fig. 9 is embedded in one of the roughnesses shown in Fig. 10. Because of the curved streamlines indicated in Fig. 9, the pressure in the liquid at the throat of the cavity is lowered by an amount which we estimate as

$$\left\{ \begin{array}{l} \text{pressure} \\ \text{reduction} \end{array} \right\} \sim \rho v_{loc}^2 h/r \quad (18)$$

Here h is the height of the protrusion illustrated in Fig. 9 and r is the radius of curvature of the streamlines over the protrusion. If we think of Eq. 9 as a static relation in which R_o is determined by the geometry of the cavity, we can use Eq. 18 in Eq. 2 to obtain

$$P_{vap} \sim P_o + 2\sigma/R_o - \rho v_{loc}^2 h/r \quad (19)$$

Here P_o represents the liquid pressure at the throat of the cavity in the absence of curved streamlines. We now return to Eqs. 17 and incorporate them in Eq. 19 to give

$$P_{vap} \sim P_o + 2\sigma/R_o - \rho v^2 h^3 / \delta^2 r, \quad h \leq \delta \quad (20a)$$

$$P_{vap} \sim P_o + 2\sigma/R_o - \rho v^2 h / r, \quad h > \delta \quad (20b)$$

If we approximate Eq. 16 by

$$\delta \sim 50D/Re \quad (21)$$

we can express Eq. 20a as

$$P_{vap} \sim P_o + 2\sigma/R_o - \rho v^4 \left(\frac{\rho^2 h^3}{2500 r \mu^2} \right), \quad h \leq \delta \quad (22)$$

This suggests that the pressure reduction is a very sensitive function of velocity when the height of the roughness is less than the thickness of the laminar sublayer.

To provide some idea of the importance of surface microcurvature, we estimate the Bernoulli effect in Eq. 20b as

$$\rho v^2 h / r \sim \rho v^2 \quad (23)$$

Taking the density to be that for FC77 (1.78 g/cm³) we find that Eq. 23 leads to

$$\rho v^2 h / r = \begin{cases} 0.005 \text{ atm} & , \quad v = 54 \text{ cm/s} \\ 0.50 \text{ atm} & , \quad v = 540 \text{ cm/s} \\ 5.0 \text{ atm} & , \quad v = 1700 \text{ cm/s} \end{cases} \quad (24)$$

From these calculations it is easy to see why temperature overshoot would decrease with increasing velocity, and why it might disappear entirely when the velocity is on the order of 17m/s (Bar-Cohen and Simon, [20]). To place these estimations in perspective we note that a pressure reduction of 0.5 atm translates to an 18°C reduction in the boiling point of water. Thus it seems possible that rough surfaces and high fluid velocities can be used to overcome the deleterious effects of temperature overshoot. Increased surface roughness and increased fluid velocity naturally produces an increased pressure drop; however, curved surface nucleate boiling is an especially efficient heat transfer process and increases in the pressure drop can be tolerated.

I. Safety Considerations

The boiling crisis to which we referred earlier could indeed become a crisis if some component of the cooling system failed. Pump failure or leaks in the cooling system could be easily detected in sufficient time to activate a back-up system, and the "worst case" system failure would appear to be an instantaneous loss of pump pressure. Under these circumstances fluid inertia would be sufficient to provide the necessary margin of safety (Whitaker [19]).

II. Conclusions

The design and construction of a novel compact, lightweight and cost effective packaging and cooling scheme for high power devices has been described. Benefits include improved device reliability, high ratings, sustained surge current tolerance and simplified fail-safe operation. The influence of surface curvature on the nucleate boiling process has been discussed from the point of view of cooling power semiconductor devices. The macro-curvature effects discovered by Gambill and Greene [2] are relatively well understood and can be utilized to construct superior heat sink devices for use with dielectric fluids. Micro-curvature effects appear to play an important role in the control of temperature overshoot, and if this is the case it may be possible to construct heat transfer channels that minimize temperature overshoot.

Nomenclature

a	acceleration (m/s ²)
g	gravitation acceleration (m/s ²)
h	height (m)
P_{vap}	vapor pressure (N/m ²)
P_{liq}	liquid pressure (N/m ²)
P_o	local ambient liquid pressure (N/m ²)
q_c	critical heat flux (kw/m ²)
r	radius of curvature of a fluid-solid surface (m)
R	radius of curvature of a vapor-liquid interface (m)
ΔT_{sub}	subcooling (K)
v	liquid velocity (m/s)
σ	interfacial tension (N/m)
ρ	liquid density (kg/m ³)

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