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SILICON HEAT PIPES FOR COOLING ELECTRONICS

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Abstract

The increasing power density of integrated circuits (ICs) is creating the need for improvements in systems for transferring heat away from the chip. In earlier investigations, diamond films were used to conduct heat from ICs and spread the energy across a heat sink. Our investigation has indicated that a 635 μm (25 mil) thick silicon substrate with embedded heat pipes could perform this task better than a diamond film. From our study, it appears that the development of a heat-pipe heat-spreading system is both technically and commercially feasible.

The major challenge for this heat-spreading system is to develop an effective wick structure to transport liquid to the heated area beneath the chip. This paper discusses the crucial design parameters for this heat-pipe system, such as the required wick properties, the material compatibility issues, and the thermal characteristics of the system. The paper also provides results from some recent experimental activities at Sandia to develop these heat-pipe heat spreader systems.

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Introduction

Integrated circuits currently operate at peak power levels of 10 Watts and power densities of 17 W/cm^2. In the next few years, these power levels are expected to approach 40 Watts at power densities of 25 W/cm^2. In other systems, such as concentrating photovoltaic arrays, there is also the possibility that heat loads are applied externally. New advances in heat removal technologies will be required to cool these electronics without resorting to pumped-liquid cooling systems.

A heat spreader is one of the advanced techniques that is currently being studied to improve heat rejection from integrated circuit (IC) chips. A heat spreader, as it is currently envisioned, is a thin substrate that transports heat away from the IC chip and spreads the energy over a larger area of a heat sink.

DISCLAIMER

Diamond films are being studied for this application because of the material's high conductivity. Diamond film is 50 times more conductive than the alumina materials that are commonly used as substrates and about 4 times more conductive than copper [Beck, Osman, and Lu, 1993]. It is estimated that diamond film substrates will be capable of passively dissipating heat from circuits with components that generate 15 W/cm^2 or more [Norwood, Worobey, and Sweet, 1994]. The main drawback to diamond substrates is the cost. Currently, an 0.5-mm thick layer of diamond substrate costs about $1000 per square inch [Hipsh, 1994]. This high cost has led investigators to seek alternative methods to form high-conductivity substrates.

The use of micro-heat pipes is one of the more promising technologies that is being applied to the development of advanced heat spreaders. Micro-heat pipes, as they were first described by Cotter [1984], are small ducts (usually with a triangular cross-section) on the order of 1-mm across that are filled with a working fluid. Heat is transferred in the tube by the evaporation and condensation of the working fluid. In the evaporator region of the tube, the fluid vaporizes and the vapor then travels to the cooled section of the tube and condenses. The condensed liquid collects in the corners of the triangular duct, and capillary forces pull the liquid back to the evaporator region. The fluid is in a saturated state, so the temperature inside of the tube is nearly isothermal.

Several investigators have modeled the performance of micro-heat pipes [Mallik, Peterson, and Weichold, 1992, Khrustalev and Faghri, 1993, Cao, Faghri, and Mahefkey, 1993, Longtin, Badran, and Gerner, 1992], and a few investigators have also constructed the pipes [Babin, Peterson and Wu, 1990]. In a heat-spreader application, the micro-heat pipes would be imbedded in a silicon substrate (or other substrate material) to effectively increase the conductivity of the substrate material. The main effect would be along the length of the pipe, however, by packing the micro-heat pipes closely together, the benefits will be realized in all directions.

Relying on the corner crevice to transport liquid to the heated evaporator section can place severe limits on the operating performance of a micro-heat pipe. Fluid basically has one path to migrate back to the evaporator. This limitation is avoided in a miniature heat pipe that uses a wick structure over the entire inner surface of the heat pipe. Plesch et al. [1991] tested a 7-mm wide by 2-mm thick by 120-mm long miniature heat pipe and found that it was possible to transfer up to 70 W along the length of the pipe with only a 35°C temperature drop. This power corresponded to a flux of about 35 W/cm^2. As a rough comparison, tests by Babin et al. [1989] demonstrated a maximum allowable heat flux of about 1 W/cm^2 with a micro-heat pipe that had a 1-mm by 1-mm cross-section [Cao and Faghri and Mahefkey 1993].

Sandia is investigating methods for forming miniature-heat pipes directly in a silicon substrate. Initial results of this investigation are presented in this paper. A preliminary assessment of the performance of miniature heat pipes is also presented.

Design

Three different heat spreader configurations were explored. These options were: (1) a series of independent tubular heat pipes imbedded in a flat substrate, (2) a single, flat heat pipe with a continuous vapor space, and (3) a flat evaporator surface with multiple vapor channels.

Independent Heat Pipes

Ideally, this heat spreader would consist of a series of parallel tubes that run through the substrate as illustrated in FIG 1. A wick structure lines the inner walls of the tubes, and each tube is evacuated, charged with enough working fluid to saturate the wick and then individually sealed.

For this preliminary investigation, it is assumed that the heat spreader is made of a 635 μm (0.025") thick flat silicon plate. To maximize conduction perpendicular to the substrate, the diameter of heat pipe, D, should be as large as possible. Mechanical stress in the narrow region between the wall of the heat pipe and the surface of the substrate will be the limiting factor on the diameter. If it is assumed that the maximum pressure in the heat pipe is 60 psig and the working stress of silicon is 1000 psi, then a 584 μm (0.023") diameter heat pipe tube is acceptable.

The distance between the tubes determines the effective conductivity in the plane of the spreader substrate. Parallel to the tubes the conductivity is essentially infinite because of the isothermal nature of the heat pipes. In a direction perpendicular to the tubes, the conductivity can be estimated by treating the substrate as a network of series resistances. If the heat pipes have a diameter D, and they are located a distance C from center to center, then the conductivity in the plane of the substrate, perpendicular to the heat pipe is approximately,

$$k_{\perp} = k_s\left(\frac{C}{C-D}\right), \qquad (1)$$

where k_s is the conductivity of silicon (≈1.3 W/cm °C @ 50°C). For a tube diameter D=0.023" (584 μm) and a spacing C=0.025" (635 μm), the planar-perpendicular conductivity is 12.5 times more conductive than silicon, that is $k_{\perp} \approx 16$ W/cm°C. This can be compared with the conductivity of a diamond of about 14 W/cm°C. A more precise model would consider the true shape of the intersection between the heat pipes and the effective conductivity of the fluid saturated wick inside of the heat pipe, but this calculation offers an indication of the conductivity in the heat spreader substrate.

Performance of a heat pipe can be limited by high vapor velocities at low operating temperatures, boiling in the wick at high fluxes, and the ability of the wick to distribute liquid over the evaporator surface. For the heat-spreader systems, the third of these limiting factors places the largest restraint on the system.

The wick structure inside of the heat pipe returns liquid to the evaporator region through capillary pumping. Capillary capabilities of the wick must exceed hydrostatic and frictional pressure drops or the wick will dry out and the heat pipe will cease to function. Heat pipes directly under the IC chip will be under the greatest thermal loading, therefore, those particular pipes will be the ones that limit performance of the heat spreader. Frictional pressure drops can be determined by applying Darcy's Law [Bird Stewart, and Lightfoot, 1960] to flow through the wick [Dunn and Reay, 1982]. Because of the high length-to-diameter ratio *(L/D)* of the pipe, the pressure drop through the wick is predominantly caused by flow in the axial direction, *x*. Darcy's Law applied along the length of the heat pipe can be written as,

$$\frac{dP}{dx} = \frac{\mu V_a}{\kappa_w}, \tag{2}$$

where μ is the absolute viscosity, V_a is the axial velocity through the wick, and κ_w is the permeability of the wick. If vapor condenses uniformly along the heat sink, then the mass flow rate in the wick, $\dot{m}$, will increase linearly along the length of the pipe. The axial liquid velocity in the wick is therefore approximately,

$$V_a = \frac{\dot{m}}{\rho \pi D\, \delta} = \frac{q\left(L_{in}/L\right) C\, x}{\rho\, h_{fg}} \frac{1}{\pi D\, \delta}, \tag{3}$$

where q is the heat flux from the IC and L_{in} is the length of the heated section. With this velocity profile, Darcy's equation can be integrated along the length of the heat pipe to determine the pressure drop in the wick. Based on this analysis, the maximum frictional pressure drop is

$$\Delta P_a = \left(\frac{q\, \nu}{h_{fg}}\right)\left(\frac{C\, L_{in}\, L}{2\pi D\, \kappa\, \delta}\right). \tag{4}$$

This pressure drop would be combined with the hydrostatic pressure drop to determine the final pressure distribution in the wick.

Capillary pumping forces counteract the pressure drop to transport liquid across the wick structure. The capillary pressure is given by the expression,

$$P_c = \frac{2\, \sigma \cos\theta}{r_e}, \tag{5}$$

where σ is the surface tension of the liquid, θ is the wetting angle, and r_e is the effective pore radius of the wick. If the hydrostatic pressure drop is neglected, and it is assumed that the liquid wets the substrate perfectly (θ=0), then the capillary pumping can be set equal to the axial frictional pressure drop to determine limiting restraints on the wick properties as,

$$\frac{\kappa_w \, \delta}{r_e} = \frac{L^2}{4\pi} \left(\frac{q}{\sigma h_{fg} / \nu} \right) \left(\frac{C}{D} \right) \left(\frac{L_{in}}{L} \right) . \tag{6}$$

The term $\sigma \, h_{fg}/\nu$ has the units of heat flux and it represents the heat transport capability of a working fluid in relation to the frictional drop in the wick. For methanol at 50°C, $\sigma \, h_{fg}/\nu$ has a value of 4540 kW/cm², and for acetone and ethanol the term has values of 2970 and 2130 kW/cm², respectively. For the case of a 13x13-mm IC setting in the middle of a 40x40-mm heat spreader, L_{in}=6.5 mm, and L=20 mm. If the IC rejects 40 Watts of power, then the wick properties must meet the condition,

$$\frac{\kappa_w \, \delta}{r_e} \geq 100 \ \mu\text{m}^2 \, . \tag{7}$$

These properties are well within the norm on existing wick materials. For instance, a 40 micron grade spherical powder packs to give a permeability of about 25 µm² and an effective pore radius of about 40 µm. If a 160 µm layer of this powder could be applied to the heat pipe's inner wall, then the above restrain on the wick would be met.

Forming small, longitudinal tubes through a silicon substrate and applying (or forming) a wick structure on the inner walls of the tubes is not an easy task. A patent by Weichold, Peterson, and Mallik [1993] describes a method of cutting grooves in a silicon substrate and then vapor depositing layers of copper or aluminum over the substrate to form small (~0.2-mm) grooves for micro heat pipes. A similar approach could be used to form channels for the series of independent heat pipes described in FIG 1, but the problem of applying a wick structure to the inner surface of the heat pipes still remains.

An attempt was made to form a porous structure on the surface by reactive ion etching. This process formed a pattern of 10-µm features on a silicon surface that were about 10-µm deep, but the permeability was far lower than acceptable. The surface also tended to flake away after the ion etching process and it was questionable if the process could be applied to a curved surface (such as in a tube). The need to form narrow tubes in the substrate material, and the need to apply a wick structure to the inner surface of the tube are two major drawbacks to producing a heat-spreader system with independent heat pipes.

<u>Flat Heat Pipe with Continuous Vapor Space</u>

In a flat heat-pipe system with a continuous vapor space, a single large cavity is formed in the substrate and a wick covers the interior surface. Post or rib structures are provided to support the cavity walls, and a wick structure is incorporated in the supports to transfer liquid from the condenser to the evaporator surface.

Gu, Chow and Baker [1994] recently tested a 1/2 inch thick flat-plate system for cooling electronics. The system consisted of two flat plates with a screen wire wick structure on the evaporator, and a studded surface on the condenser. Columns made of ceramic fiber bundles bridged the gap between the evaporator and the condenser. In their system, the electronic chips were mounted inside of the evaporator plate directly beneath the wick structure. With the condenser placed above the evaporator, Gu et al. observed a critical heat flux of about 6 W/cm^2 using dielectric fluids FC-70 and FC-72. Gu's work demonstrated feasibility of the system, but the use of screen wicks could limit the degree the system can be miniaturized.

Plesch et al. (1991) tested two flat heat pipes with a continuous vapor space system that were much closer to the miniature systems under consideration; the pipes were 2-mm thick, 7-mm wide and 120-mm long. One of the pipes had fine, transverse grooves to transport liquid around the circumference of the pipe, and two grooved ribs that ran the length of the pipe to transport liquid longitudinally. This wick system supported flux loads of about 8 W/cm^2. A second pipe Plesch et al. tested had all longitudinal grooves and it supported flux loads of about 35 W/cm^2.

In its simplest configuration, a flat heat pipe with a continuous vapor space could be made by etching or machining a cavity in two sections of substrate material; these cavities will form the heat pipe envelope when the substrate materials are bonded together. Grooves could be cut or etched on the interior surface to form a wick structure, and pylons of porous material could be set between the two substrates before bonding to serve as bridges between the wicked surfaces. Cutting the grooves in two directions will assist in the distribution of liquid, but the effective pore radius will be reduced because of the intersecting grooves.

One of the main advantages of a continuous vapor space system is that the thermal path from the heat source (i.e. the IC) and the evaporation surface is minimized. This should reduce the overall temperature drop through the system. A second advantage of the flat system, is that the wick system can be applied to a flat surface rather than the curved inner surface of a tube. This can be advantageous in both machining and etching processes. The third benefit is that liquid can reach the heated surface from multiple directions, so the operating limit for the system could be raised.

The main disadvantage to a flat heat-pipe system with a continuous vapor space is that the liquid return path is effectively lengthened. It is possible that gravity or cooling rates of the heat sink will cause condensate to preferentially collect in a location furthest from the heat source. This problem can be mitigated by adding wick-covered ribs or posts in the system, but of course, this solution adds to the complexity of the system.

<u>Flat Heat-Pipe Evaporator with Segmented Vapor Space</u>

The segmented vapor-space system that is illustrated in FIG 2 has several practical advantages over a continuous vapor-space system. The segmented approach retains some of the strength advantages of the independent heat-pipe system, and the segmented system also has an inherent connection between the evaporator surface and the condenser surface.

Sandia is currently pursuing a design where a bi-directional, grooved wick structure is formed in a flat substrate, and a series of troughs are etched in a second substrate. The two substrates are joined at the edges and the system is then charged with the working fluid through a hole in the surface. By heating the substrate, excess working fluid is boiled away and the non-condensable gases are swept out of the system. After the proper charge is established, the hole is sealed, and the heat pipe is then cooled. The vapor passages are not completely independent; small ducts between the vapor passages are provided to assist in the purging of non-condensable gases.

The simple design of this system does have a few draw-backs. First of all, the condenser surface is not wicked so the system will rely somewhat on gravity to return liquid to the evaporator wick. An orientation with the heat sink above the evaporator will be preferred for optimum performance, however, results from computer models have shown that the system can compensate for rather large deviations from this orientation. Similar to the continuous vapor space system, it is also possible that the condensate can preferentially collect far from the heat source. Segmenting the vapor-space, though, will make the system less sensitive to an uneven cooling rate across the heat sink.

Sandia is now performing a series of tests on components that will be used to construct a segmented vapor-space heat spreader. FIG 3 shows a wick structure that was formed on a substrate by cutting grooves in a silicon substrate with a diamond wafer saw. The pillars that are shown in FIG 3 are roughly 75 μm tall by 50 μm wide, and the spacing is about 63 μm. Tests on a substrate with 100-μm wide wick pillars showed that the permeability is on the order of 100 μm^2. Wicking height tests showed that the effective pore size was on the order of 80 μm, so the system is close to the acceptable limit given in EQ 7.

Bonding techniques for substrate materials have also been investigated. Wells which could serve as vapor passages were first KOH etched in a silicon wafer to a depth of 0.5 mm. A second wafer that had holes aligned with the well pattern was then bonded to the etched wafer. Bonding was accomplished by depositing a 1-μm layer of boron-phosphorous-silicate glass (BPSG) to the wafer without wells, followed by cleaning and furnace annealing in O_2 for several hours at 850-1000°C. Several bonded samples were tested with a vacuum helium leak detector and found to be hermetic to the lowest limit of the detector (10^{-8} std cc/s). Cutting the wick structure in the wafer can leave debris, and it has not been determined if the wafers can be bonded after the wick structure is in place.

Future Activities

From a preliminary analysis, it appears that a silicon heat-pipe system should out-perform a diamond substrate in distributing energy from electronic circuits. Sandia is now engaged in the fabrication of a complete silicon heat-pipe heat-spreader system. The focus will be on building a system with a flat wick structure and a segmented vapor space with no wick structure on the condenser. This system does present some performance compromises in the range of acceptable orientations, but compromises appear to be outweighed by the simplicity of fabrication.

Several issues must still be resolved. Methanol is currently the preferred working fluid, but tests must be performed to determine the compatibility of methanol with the silicon, or if an oxide layer is needed to protect the silicon and prevent gas generation in the heat pipe. Filling and sealing techniques must also be established and proven. There are also concerns about the effectiveness of bonding techniques in a wafer with a machined wick structure.

Additional testing must also be conducted to determine the performance of the wick structure. Tests have been performed in air with a heater mounted to a wicked substrate and it was found that the system was effective in dissipating heat at temperatures near the boiling point of the working fluid. Tests will soon be performed with the wick substrates in a vacuum chamber with an atmosphere of saturated methanol.

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Nomenclature:

C = heat pipe tube spacing (m)
D = heat pipe tube diameter (m)
L = tube length (m)
L_{in} = heater length (m)
P = pressure in wick (Pa)
P_a = axial pressure distribution (Pa)
P_c = capillary pressure (Pa)
V_a = axial velocity (m/s)
h_{fg} = heat of vaporization (kJ/kg)
$k_\perp$ = conductivity perpendicular to tube (W/m°C)
k_s = substrate conductivity (W/m°C)
r_e = effective pore radius
x = axial distance (m)
δ = wick thickness (m)
κ_w = permeability (m^2)
μ = absolute viscosity (kg/m s)
ν = kinematic viscosity (m^2/s)
ρ = liquid density (kg/m^3)
θ = wetting angle

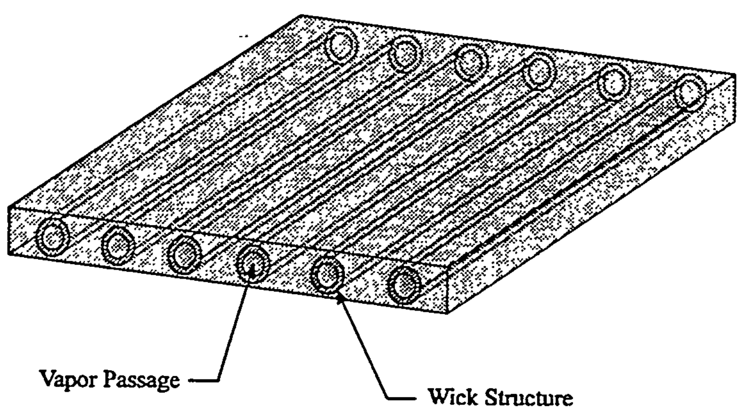

FIG 1. Independent parallel heat pipes in a substrate.

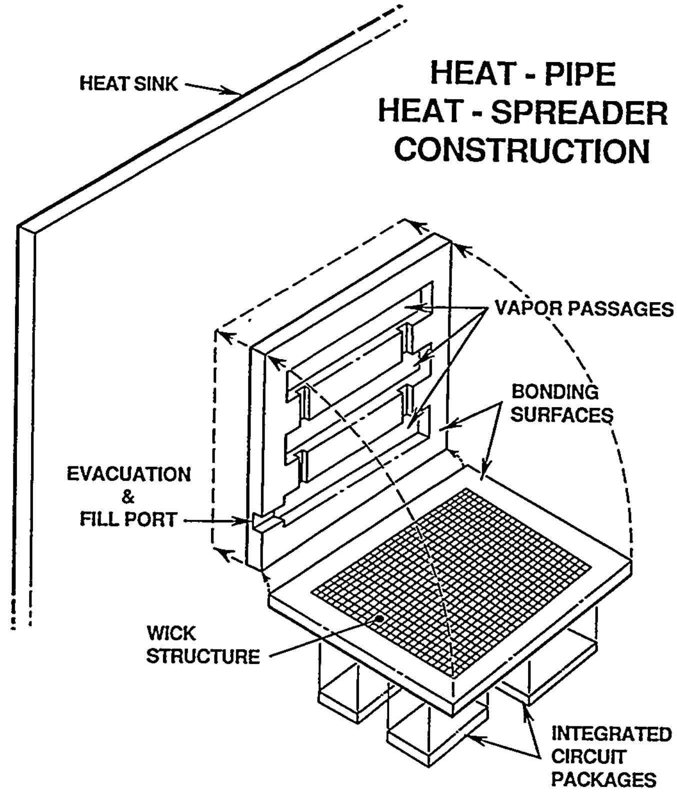

FIG 2. Flat heat-pipe evaporator with a segmented vapor space.

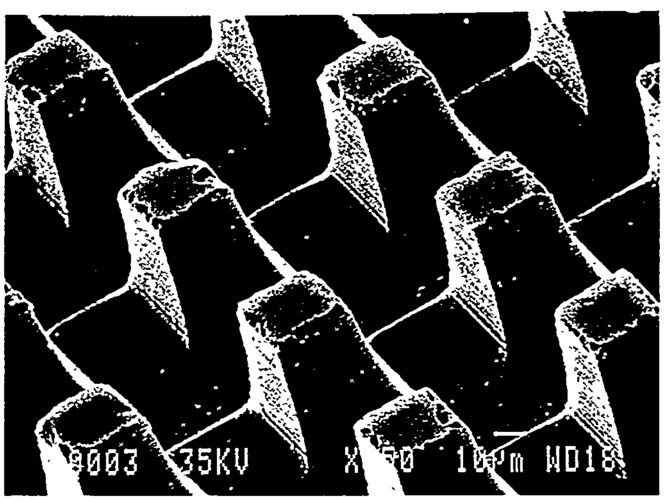

FIG 3. Micrograph of a sawed wick structure in a silicon substrate.