EXTREME HOTSPOT HEAT FLUX THERMAL MANAGEMENT VIA THIN-FILM EVAPORATION FROM MICROSTRUCTURED SURFACES

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ABSTRACT

We demonstrate an extreme heat flux thermal management solution targeted towards cooling hotspots with $640 \times 620 \mu\text{m}^2$ footprint. Our heat dissipation strategy utilizes thin-film evaporation by incorporating micropillar wicks that allow passive fluidic transport *via* capillarity in addition to maximizing the evaporation area by extending the three-phase contact line. With our wick design, we dissipated $\approx 5.8 \text{ kW/cm}^2$, the largest heat flux reported to date when compared to past thin-film evaporation studies with similar size hotspots. Our experimental results indicate that thin-film evaporation is a promising thermal management strategy for the next generation microprocessors, power amplifiers and radio-frequency devices, where cooling hotspots is a significant challenge.

INTRODUCTION

Thermal management is a primary design concern for numerous power dense equipment such as microelectronic devices, power amplifiers, electric vehicles, solar energy converters, and radar and advanced military avionics [1]. These devices generate large amounts of waste heat from sub-millimeter areas during operation. The heat generation can exceed 1 kW/cm^2 [2, 3] causing hotspots which are localized regions with extreme heat flux and exceedingly high temperature [3]. Furthermore, the heat generation is spatially and temporally non-uniform [4], making the task of removing the waste heat from the device footprint a challenging task for thermal engineers using existing cooling technology [5]. More importantly, the heat generation and the associated rise in junction temperature can adversely impact device performance and reliability [6, 7]. Consequently, many cutting-edge technologies are becoming increasingly dependent upon the ability to efficiently cool hotspots [2, 3].

Traditional air and single-phase liquid cooling solutions use stacks of highly conductive solid-state thermal spreaders such as copper to distribute the heat load over a larger area before ultimately dissipating it *via* convection [8-11]. However, convective air cooling is fundamentally limited by the inferior thermophysical properties of air [1] while single-phase liquid cooling is limited by the low coefficient of performance (COP) due to high pumping power that is required for transporting the working fluid. Furthermore, these systems are not well suited to address extreme heat fluxes ($>1 \text{ kW/cm}^2$). Consequently, a new paradigm in extreme heat flux hotspot thermal management is required for the advancement of the microelectronic industry.

An efficient strategy to remove chip-level heat loads that exceed 1 kW/cm^2 for high performance systems is to utilize a phase-change-based approach where the high latent heat of vaporization can be harnessed when the working fluid changes phase from liquid-to-vapor. Furthermore, phase-change-based thermal solutions reduce the size, weight, and overall power consumption making them suitable for microelectronic devices. Such solutions including microchannel flow boiling [7], jet impingement [12], and spray cooling [13], however, are fundamentally limited by instabilities [14]. Accordingly, flow restrictors have been proposed for microchannel flow boiling [15], but lead to significant increase in pumping power which can limit

their appeal as a thermal management strategy. A potential phase-change-based cooling strategy for high heat flux devices is thin-film evaporation from enhanced surfaces. Thin-film evaporation [16, 17] has received significant interest in the last few decades due to recent advances in microfabrication technology which facilitated the use of engineered surfaces for heat transfer applications. Thin-film evaporation relies on the phase-change of a thin liquid film from an extended meniscus near the three-phase contact line. The extended meniscus increases the evaporation area, decreases the conduction resistance, and improves the thermal and fluidic transport near the thin-film region [18]. By controlling the thin-film region and the solid-liquid interaction, ultra-high heat fluxes can be dissipated *via* thin-film evaporation.

The present work is aimed at using thin-film evaporation based devices with microstructured wicks for extreme heat flux thermal management. We designed and fabricated test devices that manipulate the solid-liquid interface and the liquid-to-vapor phase transition. The test devices were also designed to closely emulate high performance electronic devices by (a) incorporating multiple hotspots on the same device, and (b) superposing hotspots with background heating. To dissipate this heat, the microstructured wicks relied on capillarity for fluidic transport, *i.e.*, the working fluid was transported to the evaporative area by creating a pressure gradient along the flow direction. The pressures gradient was in direct response to the variation in the meniscus shape along the flow direction and therefore no external pumping power was required for operation. Lastly, by integrating the cooling solution directly on our device, we were able to combine the heat spreader and the heat dissipation technology into a single chip in direct contact with the heat source.

DEVICE DESIGN

Our design leverages microstructured wicks with well-defined silicon micropillar arrays in a square pattern. The overall dimension of the test device was $3 \times 3 \text{ cm}^2$ with the center $1 \times 1 \text{ cm}^2$ microstructured area for evaporation (Fig. 1a). We fabricated these devices from a silicon wafer using contact photolithography and deep-reactive-ion-etching. We varied the diameters (D), heights (H), and center-to-center spacings (L) of the micropillars (scanning electron micrograph, Fig. 1b) from 5-12, 40-100, and 12-20 μm , respectively for systematic characterization of fluidic and thermal transport within the micropillar wick. In this work we discuss results for one set of geometry ($D=8 \mu\text{m}$, $H=84 \mu\text{m}$, and $L=20 \mu\text{m}$), however, similar trends were observed across the entire parametric space. The microstructured surface ($1 \times 1 \text{ cm}^2$) was surrounded by a 6 mm water reservoir (Fig. 1c). The working fluid, de-ionized (DI) water, was passively transported from the reservoir to the evaporative area due to the variation in the shape of the liquid meniscus (*i.e.*, capillary pressure) along the flow direction (Fig. 1c). On the back side of each test device, we incorporated thin-film (100 nm) platinum lines which served both as heaters and temperature sensors (Fig. 1d). The heaters/sensors were calibrated in a convection oven prior to the experiments. The resistance and temperature were linearly correlated with $<1\%$ RMS error. To emulate realistic operating conditions in microelectronic devices, each test device incorporated a background heater (H0) which

provided uniform heating for the $1 \times 1 \text{ cm}^2$ microstructured area and three local hotspot heaters (H1, H2, and H3) which provided concentrated heat load over a $640 \times 620 \mu\text{m}^2$ area (Fig. 1d).

Figure 1: (a) Image of front side of the device. The square and the three circles represent the back side location of the background and hotspot heaters, respectively. (b) Scanning electron micrograph of the microstructured surface. Inset showing pillar diameter (D), height (H), and center-to-center spacing (L). (c) Schematic cross-section A-A showing operating principle of our device. The pressure gradient from the edge to the center of the microstructured area was utilized for fluidic transport (i.e., capillary-wicking). (d) Back side of the device showing thin-film (100 nm) heaters/sensors. The three circles indicate the location of the three hotspot heaters (H1, H2, and H3, each $640 \times 620 \mu\text{m}^2$) while the square indicates the location of the background heater (H0, $1 \times 1 \text{ cm}^2$). The location of the background heater on the back side matches the location of the microstructured area on the front side. The hotspot heaters were located 3 mm apart along the longitudinal centerline. All temperatures were measured at the back side of the test device.

EXPERIMENTAL SETUP

Samples were wet cleaned (acetone, methanol, isopropanol, and DI water), dried using compressed nitrogen gas, and plasma treated (PDC-001, Harrick Plasma) in an oxygen environment for 30 min before each experiment. All experiments were conducted at saturated conditions ($\approx 24^\circ\text{C}$ and $\approx 3 \text{ kPa}$) inside a stainless steel environmental chamber ($16 \times 16 \times 16 \text{ in}^3$, Fig. 2). After attaching the different feedthroughs (electrical, thermocouple, and liquid), the system was pumped down using a vacuum pump (Pascal 2010 SD, Adixen) to remove noncondensable gases. The chamber was then allowed to reach saturated conditions by injecting degassed DI water into the chamber *via* a liquid feedthrough. The chamber temperature was measured by inserting a calibrated J-type thermocouple *via* a thermocouple feedthrough. All thermocouples were calibrated prior to experiment using a refrigerated water bath heat exchanger (Lauda RE200, Lauda-Brinkmann). A capacitance manometer (740C Baratron, MKS Instruments, Inc.) was used to measure the chamber pressure. High purity DI water (Chromasolv 34877, Sigma-Aldrich) was used for all experiments. The DI water which was degassed *via* boiling before the experiment was stored in a stainless steel canister outside the environmental chamber. During the experiments, the water inside the canister was maintained at $\approx 60^\circ\text{C}$ and $\approx 20 \text{ kPa}$ using a closed-loop temperature feedback system which used a PID controller (CN7800, Omega Engineering). The pressure difference

between the canister and the chamber ($\approx 17 \text{ kPa}$) was utilized to transport water from the canister to the environmental chamber by overcoming the hydrostatic head and the viscous losses in the valve and tubing. The flow rate of the incoming water to the chamber was measured using a liquid flow meter (L-50CCM-D, Alicat Scientific). The incoming water was actively cooled to $\approx 24^\circ\text{C}$ before entering the environmental chamber using a water bath heat exchanger (Lauda RE200, Lauda-Brinkmann). During the experiments, the evaporating water vapor was allowed to condense on the internal walls of the environmental chamber. The condensate was removed from the chamber *via* a drain valve at the end of each experiment. The chamber has visual access and images were acquired at 1000 frames per second using a high speed camera (Phantom v7.1, Vision Research). Data was collected for analysis using a data acquisition system (cDAQ-9174, National Instruments). A DC power source was used to supply the voltage required for heating and temperature sensing *via* an electrical feedthrough. The electrical voltage and current were measured using a digital multimeter (Model 2000, Keithley Instruments).

Figure 2: Schematic (not-to-scale) experimental setup. The working fluid, DI water, was stored outside the stainless steel environmental chamber in a canister at 60°C and 20 kPa . The incoming water to the chamber was cooled to the chamber temperature ($\approx 24^\circ\text{C}$) using a water bath heat exchanger. The T, P, and V indicate the thermocouple, pressure gage, and liquid flow meter, respectively. During the experiments, the water vapor was allowed to condense on the internal walls of the environmental chamber. At the end of each experiment, the condensate was removed *via* the drain valve and the chamber was made ready for the next experiment.

RESULTS

We experimentally characterized capillary-limited thin-film evaporation with our test devices and experimental setup. We first created a single hotspot using the thin-film heater H2 and simultaneously measured the temperature. The local temperatures 3 mm away from the hotspot were also measured using H1 and H3, while the background temperature (over the microstructured area) was measured using H0. During the experiments, the heat flux was increased in an incremental fashion. Each heat flux was maintained for $\approx 100 \text{ s}$ (Fig. 3a). The temporal data was time-averaged to obtain a single data point for the subsequent figures shown here. The heat flux was increased until the hotspot dried out due to liquid starvation (i.e., when the viscous losses exceed capillary pressure). We dissipated $\approx 5.8 \text{ kW/cm}^2$ when the hotspot temperature was $\approx 260^\circ\text{C}$ (Fig. 3a-b). This heat flux is an order of magnitude higher when compared to past thin-film evaporation studies with similar size hotspots [18, 19]. More importantly, the background temperature and local temperatures 3 mm away from the hotspot were $< 50^\circ\text{C}$ (Fig. 3b). The measurement uncertainty in our experiments for temperature ($\pm 2.5^\circ\text{C}$) and heat flux ($\pm 0.3 \text{ kW/cm}^2$) was estimated

via standard error propagation analysis by combining the random (statistical, one standard deviation) and system errors [20].

Without background heating, we created multiple hotspots on our test devices and examined the hotspot dryout heat flux. Two hotspots were created by activating H1 and H3 while three hotspots were created by activating H1, H2, and H3. The hotspot temperatures were measured at the location where the heat was generated. For example, when H1 and H3 were active hotspots, the hotspot temperatures at the respective locations were measured using H1 and H3. Our experimental results show that the hotspot dryout heat flux decreased monotonically when multiple hotspots were created on the microstructured surface (Fig. 3c-d). The dryout heat flux which was $\approx 5.8 \text{ kW/cm}^2$ when a single hotspot (H2) was active (red, Fig. 3c), decreased to $\approx 4.2 \text{ kW/cm}^2$ per heater when two hotspots (H1, H3) were active simultaneously (blue, Fig. 3c). This dryout heat flux decreased further to $\approx 2.9 \text{ kW/cm}^2$ per heater when three hotspots (H1, H2, H3) were active simultaneously (black, Fig. 3c). Furthermore, when examining the heat flux as a function of hotspot temperature (Fig. 3c), the results followed a similar trend when one (H2), two (H1, H3), and three (H1, H2, H3) hotspots were active at the same time. For example, when the heat flux was $\approx 2.9 \text{ kW/cm}^2$, the hotspot temperature remained $\approx 132^\circ\text{C}$ (Fig. 3c), whether we activated a single hotspot (H2), two hotspots (H1, H3), or three hotspots (H1, H2, H3). Unlike the hotspot dryout heat flux which progressively decreased as the number of active hotspots increased on the microstructured surface (blue, left ordinate, Fig. 3d), the total heating power increased from 23 W to 35 W (red, right ordinate Fig. 3d) for one hotspot (H2) and three hotspots (H1, H2, H3), respectively.

In actual microelectronic devices, hotspots coexist with background heating. Therefore, we characterized the effect of background heating on the hotspot dryout heat flux (Fig. 3e-f). A

single hotspot was created at the center of the microstructured surface by activating H2 while superposing a uniform background heat flux. Background heat flux was provided by uniformly heating the entire $1 \times 1 \text{ cm}^2$ microstructured area using thin-film heater H0. During experiments, the background heat flux was kept constant and the hotspot dryout heat flux was increased in a stepwise manner until the microstructured surface dried out due to liquid starvation. This procedure was repeated for different background heat fluxes. We observed that, the hotspot dryout heat flux decreased monotonically as background heat flux increased (Fig. 3f). The hotspot dryout heat flux which was $\approx 5.8 \text{ kW/cm}^2$ without background heating (red, Fig. 3e) decreased to $\approx 4.0 \text{ kW/cm}^2$ (blue, Fig. 3e) and $\approx 2.9 \text{ kW/cm}^2$ (black, Fig. 3e) by superposing a 10 W/cm^2 and 20 W/cm^2 background heat flux, respectively. Unlike the hotspot dryout heat flux which decreased with background heating (blue, left ordinate, Fig. 3f), the total heating power from the microstructured surface increased from 23 W to 31 W (red, right ordinate, Fig. 3f), when background heat flux increased from 0 and 20 W/cm^2 .

We attribute the high heat fluxes in our study to the micropillar wick design which separates the vapor transport from the liquid transport. Our test devices allowed unimpeded vapor transport by allowing the phase change to occur at the top of the liquid-vapor interface. Furthermore, the micropillar wick surrounding the hotspot region assisted liquid transport by generating capillary pressure owing to the meniscus shape. The capillary pressure was utilized for passive fluidic transport which avoided the need for an external pump. Moreover, we eliminated conventional heat spreaders by incorporating the cooling solution directly on our devices. Even higher heat fluxes can be achieved by optimizing the dimensions of the micropillars, indicating that thin-film evaporation from engineered surfaces is a promising strategy for cooling hotspots.

Figure 3: (a) Temperature as a function of time for a typical experiment. The heat flux was increased in a stepwise manner until dryout. (b) Heat flux as a function of temperature as measured by the different sensors (H0, H1, H2, and H3). A dry island formed at the hotspot H2 (inset, scale bar = 10 mm) when viscous losses exceed capillary pressure in which case the rate of mass loss via evaporation exceeded the rate at which liquid was delivered to the evaporative area. (c) Heat flux as a function of hotspot temperature when one (H2), two (H1, H3) and three (H1, H2, H3) hotspots were activated concurrently. The arrows indicate the hotspot dryout heat flux. (d) The presence of concurrent multiple hotspots decreased the hotspot dryout heat flux (blue, left ordinate), but increased the total heating power (red, right ordinate). (e) Heat flux as a function of hotspot temperature when the hotspot (H2) was superposed with uniform background heat flux. The arrows indicate the hotspot dryout heat flux. (f) When the hotspot (H2) was assisted by a 10 W/cm^2 and 20 W/cm^2 uniform background heat flux, the hotspot dryout heat flux decreased (blue, left ordinate), but the total heating power increased (red, right ordinate).

CONCLUSIONS

We experimentally characterized hotspot cooling via thin-film evaporation using well-defined silicon micropillar wicks. Our test devices were designed to supplement spatially distributed hotspots with background heating to closely resemble actual microelectronic devices during operation. Fluidic transport was implemented via capillarity where the working fluid, DI water, was transported due to the pressure gradient along the flow direction. This was facilitated by using microstructured surfaces. Furthermore, the thermal transport near the thin-film region was enhanced by extending the liquid meniscus and three-phase contact line. With our devices, we dissipated $\approx 5.8 \text{ kW/cm}^2$ from a $640 \times 620 \text{ }\mu\text{m}^2$ hotspot area at $\approx 260 \text{ }^\circ\text{C}$ hotspot temperature. More importantly, the average temperature over the entire $1 \times 1 \text{ cm}^2$ evaporative area and the local temperatures 3 mm away from the hotspot were significantly low ($< 50 \text{ }^\circ\text{C}$). The high heat fluxes attained in our experiments are attributed to the capillarity generated by our tailored microstructured surfaces. The micropillar arrays can be optimized to achieve higher heat fluxes. Our experimental results show that thin-film evaporation is a promising on-chip thermal management approach for cooling hotspots in high performance systems which generate heat loads in excess of 1 kW/cm^2 .

The hotspot dryout heat flux decreased by concurrently creating multiple hotspots which were spatially distributed over the $1 \times 1 \text{ cm}^2$ microstructured surface. Similarly, the hotspot dryout heat flux decreased by assisting the hotspot with background heating. Despite the decrease in the hotspot dryout heat flux, the total heating power increased by spatially distributing the hotspots over the microstructured surface as well as by assisting the hotspot with a uniform background heat flux. The results of this study provide insights into the design of high performance evaporator wicks which are capable of dissipating extreme heat fluxes ($> 1 \text{ kW/cm}^2$). The understanding can be used to improve the design of wicking structures which are commonly used in two-phase thermal management devices such as heat pipes and vapor chambers.

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