Local Heat Transfer Coefficient Measurements of Flat Angled Sprays Using Thermal Test Vehicle

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Abstract

Impingement cooling methods, such as spray cooling and jet impingement have demonstrated the capability of cooling high heat flux surfaces while maintaining a low thermal resistance. Most spray cooling and jet impingement experiments attempt to measure the average heat transfer coefficient, even though it is known that heat transfer coefficients are known to change as a function of distance from the impact zone. Secondly, most experiments are done on thick uniformly heated surfaces although most electronic devices are very thin (<0.2mm) and generate heat very non-uniformly with very large peak heat fluxes (>1000W/cm^2) over very small areas (<0.25mm^2). In this study an accurate measurement of the uniformity of the spray cooling thermal solution was attained using an Intel supplied thermal test vehicle. The heater block is a thin silicon chip (<0.25mm thick and 7cm^2 in surface area) delivering a uniform heat flux to 70W/cm^2. The platform also has the ability to power large peak heat fluxes (>1000W/cm^2) over small areas (<0.25mm^2). Experiments using jet impingement with flat spray nozzles angled to the surface were conducted with water, methanol, and HFE-7000. The axial heat transfer coefficient variation was measured under uniform heat loading. Finally, the measurements are compared to modified models from the literature with good agreement.

Keywords

Spray Cooling, Jet Impingement, Thermal Management, Heat Transfer

Nomenclature

α	Angle (°)
β	Correction factor for angled impingement
υ	Fluid kinematic viscosity (m^2/s)
d	Nozzle diameter (m)
D	Tube Diameter (m)
h	Heat transfer coefficient (W/m^2-K)
H	Nozzle to surface distance (m)
k	Fluid thermal conductivity (W/m-K)
m	Power law factor for single slot nozzle correlation
$\overline{Nu}$	Spatially averaged Nusselt number for angled flat spray cooling
$\overline{Nu_M}$	Spatially averaged Nusselt number from Martin Correlation
$\overline{Nu_T}$	Spatially averaged Nusselt number from Tawfek Correlation
Nu_x	Local Nusselt number for flat angled spray cooling
$\Pr$	Fluid Prandtl number
Re	Jet Reynolds Number
V	Velocity of fluid exiting nozzle
x	Axial position from fluid impact region (m)
y	Transverse direction from nozzle (m)

1. Background and Motivation

Single phase jet impingement heat transfer has been given a great deal of attention in the literature due to its ability to dissipate high heat fluxes at low thermal gradients. [1] Jet impingement has been used in many applications ranging from turbine blade cooling to plasma facing components to high density microelectronics applications. [2] Gas impinging jets were first studied with respect to turbine blade cooling. However, heat transfer coefficients orders of magnitude higher than gas jets can be obtained using liquid jets. For the case of liquid jets, two types exist. One type is a submerged jet. In this case the nozzle ejects water through a pool of liquid. The second type is a free surface jet, where the liquid jets through stagnant air before hitting the surface. The experimental data acquired in this study is of the free surface variety using liquids.

In the current study we are also limiting ourselves to flat sprays that are angled to the surface (as opposed to spraying normal to the surface). A diagram of the spray cooling system studied is shown in Figure 1. The motivation behind this study was unique to a semi-conductor debug process, where the back side surface of the chip needs to be accessible while the chip is heated under typical thermal loadings experienced in real world computer processors. The nature of the process requires removal of any heat sink and thermal interface material traditionally used to cool the back side of the silicon chip. Only direct fluid technologies are capable of cooling the chip while the back side of the silicon is accessed. The goal of this study was to measure the local temperature of the chip under thermal loading using the proposed flat angle spray cooling system. To take these local measurements under realistic thermal conditions and chip geometries an Intel supplied thermal test vehicle was utilized. The thermal test vehicle is unique in is ability to measure accurate real time temperatures, while generating the necessary heat flux through the silicon chip surface. In the required process, the chip junction temperature was required to be controlled within 2°C between -10°C and 110°C. Three fluids were studied using the proposed cooling system to meet the various temperature ranges: water, methanol and HFE-7000.

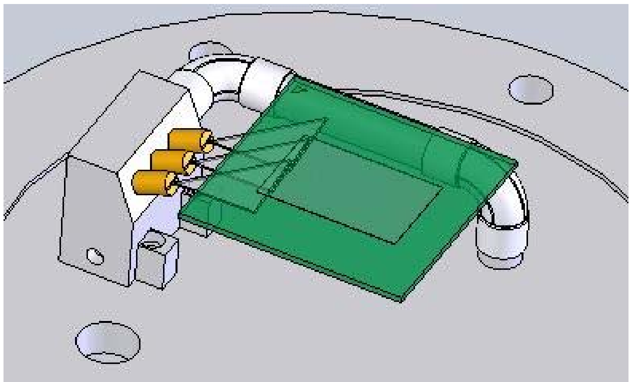

Figure 1: A diagram illustrating the spray cooling system studied in this paper is shown. Liquid exits an array of nozzles forming a flat spray. The spray impacts the surface of the silicon chip (shown as semi-transparent) at an angle to the surface, directly on the edge of the chip. The chip itself is held in place by a metal clamp, not shown.

2. Modeling of Local Heat Transfer Coefficients with Flat Angled Sprays

Multiple models exist in the literature to predict heat transfer coefficients in jet impingement systems. [3-5] A well referenced model in the literature, especially for electronics cooling applications is the Martin correlation for single slot nozzles. [6,7] The Martin correlation for single slot nozzles is know to work well for predicting average heat transfer coefficient for jet impingement systems. However, some modifications need to be made to the spatially average model to make comparisons to the local heat transfer coefficients measurements acquired in this study. The current study also requires adjustments to be made for the effect of angling the nozzles. The Martin single slot nozzle correlation is for nozzles angled normal to the surface. Tawfek has proposed a correlation for impingement on a tube that includes a power law factor describing the effect on the heat transfer coefficient due to nozzle angle. A model is proposed here to model flat angled spray cooling heat transfer coefficients by combing the Martin model with the power law angle effect of the Tawfek model. To capture the local heat transfer coefficient variations, the combined model (which is spatially averaged) is than differentiated to calculate local values of the Nusselt number.

A summary of the Martin model for single slot nozzle cooling and its application to the proposed study is described. The Nusselt number for jet impingement is shown in Equation 1. The Nusselt number for the Martin single slot nozzle correlation is calculated using terms for physical property values, flow conditions and geometric considerations as shown in Equation 2. The physical property values are accounted for by a Prandtl number effect. The flow conditions are characterized by the jet Reynolds number, described by Equation 4. The power law factor, m, for the Reynolds number is also a function of geometric considerations as described by Equation 3. The geometric considerations include the length of the heated surface, L, distance between the nozzle and impact zone, H, and the hydraulic diameter of the nozzle, d. It should be noted for slot nozzles the hydraulic diameter is equal to twice the width of the slot. However, in the current study a flat spray nozzle was used with an elliptical shape. An effective hydraulic diameter was used as specified by the nozzle vendor.

$$\overline{Nu} = \frac{hd}{k} \tag{1}$$

$$\frac{\overline{Nu_M}}{\Pr^{0.42}} = \frac{3.06}{\frac{x}{d} + \frac{H}{d} + 2.78} \mathrm{Re}^m \tag{2}$$

$$m = 0.695 - \left[\left(\frac{x}{2d}\right) + \left(\frac{H}{2d}\right)^{1.33} + 3.06\right]^{-1} \tag{3}$$

$$\mathrm{Re} = \frac{Vd}{\upsilon} \tag{4}$$

In the current application, available space allows only an angled solution due to the required access to the chip surface. The jet will not impinge perpendicular to the surface, but at an angle of approximately 20°. The Tawfek correlation provides a power law factor accounting for the angle of the impinging nozzle which must be included [8-10]. Although the complete Tawfek model is not as accurate as the Martin model for electronics applications, the power law factor was separated from the complete Tawfek correlation and combined with the Martin correlation. The Tawfek correlation containing the angle factor and the range of applicability for the angled factor are shown in Equation 5 and Equation 6.

$$\overline{Nu_T} = 0.142\,\mathrm{Re}^{0.71}\,\alpha^{0.194}\left(\frac{H}{d}\right)^{-0.14}\left(\frac{d}{D}\right)^{-0.35} \tag{5}$$

$$20° \le \alpha \le 90° \tag{6}$$

The complete model for local heat transfer coefficient measurement is described here. The power law factor, β, derived from the Tawfek correlation is shown in Equation 7. The factor is normalized to 90° to match the angle at which the Martin correlation is applicable (normal to the surface). To create a spatially average flat angled spray model, the Nusselt number correlation from Martin is multiplied by the angled factor in Equation 8. Some manipulation of the model was completed to transform the spatially averaged model into a local Nusselt number model. The spatially averaged model is described by Equation 9 as an integral of the local Nusselt number over the length of the heated surface divided by the total length of the surface. Figure 4 diagrams the axial position of the fluid, x. The desired local Nusselt number can be solved for by multiplying by the length, x, and differentiating. The end result is shown in Equation 10. A numerical solution was used to calculate local Nusselt numbers and the corresponding local heat transfer coefficients.

$$\beta = \left(\frac{\alpha}{90^\circ}\right)^{0.194} \tag{7}$$

$$\overline{Nu} = \overline{Nu_M}\,\beta \tag{8}$$

$$\overline{Nu} = \frac{\int_0^x Nu_x dx}{x-0} \tag{9}$$

$$Nu_x = \frac{d\left(\overline{Nu}x\right)}{dx} \tag{10}$$

3. Test Setup and Thermal Test Vehicle

A spray cooling test section was integrated with an Intel supplied 34980A thermal test platform to acquire data in this study. The thermal test platform is an integrated thermal testing system including the thermal test vehicle, power supplies, data acquisition system, and Labview interface. The thermal test vehicle (TTV) is a silicon chip with two large embedded resistive heaters, four micro-heaters for delivering peak heat fluxes and 29 resistance temperature detectors (RTD's). The organic substrate measures 37.5 mm by 37.5 mm and the silicon die measures 33 mm by 22 mm. The die is less than 0.25mm thick. The TTV is capable of supplying a uniform heat flux of 70W/cm^2 across its surface as well as peak heat fluxes over 1000W/cm^2 over areas less than 0.25mm^2 at four locations on the die. An Agilent data acquisition system acquires the temperature readings, measures the voltage drop across and current through the resistive heaters in the die and calculates the corresponding uniform heat flux provided to the silicon. Each test vehicle is independently calibrated in a thermal bath filled with fluorocarbon at -10°C, 50°C, and 90°C. The accuracy of the temperature readings of the RTD's are accurate to within 0.1°C.

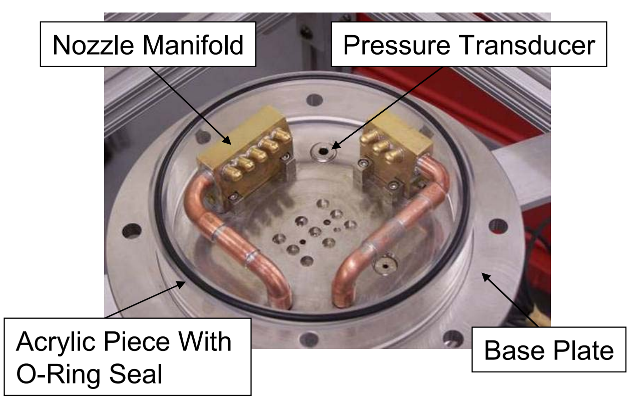

Figure 2: Photograph showing spray cooling system hardware. Not shown is the thermal test vehicle which sits above the nozzle arrays centered between the arrays. There is also a plate which created a liquid tight seal around the cooling solution and test vehicle.

The thermal test vehicle sits above a flat angled nozzle array as shown previously in Figure 1. The nozzles are at a 20° angle to the TTV. Also shown is the fluid exiting the nozzles making a flat spray pattern as the fluid impacts the edge of the chip. The nozzles were tilted at a 5° angle (rotational) to avoid the intersection of flow before the spray impacts the chip. The spray exits the flat spray nozzle at a 20° angle to the surface of the chip. The nozzles were overlapped 50% in order to have flow distributed uniformly over the edge of the die. As the fluid exits the die it is collected and recirculated through the system by a pump. A photograph of the cooling system test section is shown in Figure 2. The fluid exiting the test section drained to a heat exchanger system to cool the fluid to the desired temperature. The system was capable of cooling the methanol and HFE-7000 fluids to temperatures below -35°C.

4. Experimental Data

Local spray cooling heat transfer coefficient test data were acquired at uniform power input for three fluids: water, methanol and HFE-7000. Water was used to test at ambient conditions while HFE-7000 and methanol were tested at sub-ambient conditions. Water is the fluid of choice for it superior physical properties. However, it is limited to keeping the chip at above ambient conditions because of freezing concerns at lower temperatures. Methanol and HFE-7000 were chosen for the sub-ambient conditions because of their low freezing points. Methanol is expected to have higher heat transfer coefficients than HFE-7000, but flammability, toxicity and chemical compatibility make it less attractive as a cooling medium. HFE-7000 has inferior thermophysical properties to methanol, but is inherently safe and chemically compatible with the chip. HFE-7000's dielectric properties are also more attractive for direct cooling of electronic chips.

Water was sprayed at a total flow rate of 1000mL/min through the three 0.020" diameter nozzles at a temperature of 26.7°C. A heat flux of 42W/cm^2 was applied to the die uniformly. Methanol was sprayed to the chip at a flow rate of 1250mL/min and a -31.6°C inlet temperature. HFE-7000 was sprayed at a total flow rate of 1300mL/min at -29.7°C. The uniform heat flux applied to the die for the methanol and HFE-7000 testing was 14W/cm^2.

Figure 3 shows a plot of the experimental heat transfer coefficient data versus axial position (in the x-direction) away from the impact zone for the three fluids at their respective test conditions. A diagram of the coordinate system used is shown in Figure 4. The thermal test vehicle has RTD's spread across the die at various x-y coordinates. There is expected to be some variation in the heat transfer coefficient in the direction along the impact zone of the spray (y-direction) due to non-uniform velocity profiles. However, this variation is neglected and only variation in the x-direction is studied. Position 0.0cm signifies the location of the impact of the spray. At the impact zone the maximum heat transfer coefficient is measured for all three fluids. The heat transfer coefficient then decreases as the fluid travels away from the impact zone due to the decrease in momentum of the fluid in the boundary layer and resulting increase in thickness of the boundary layer.

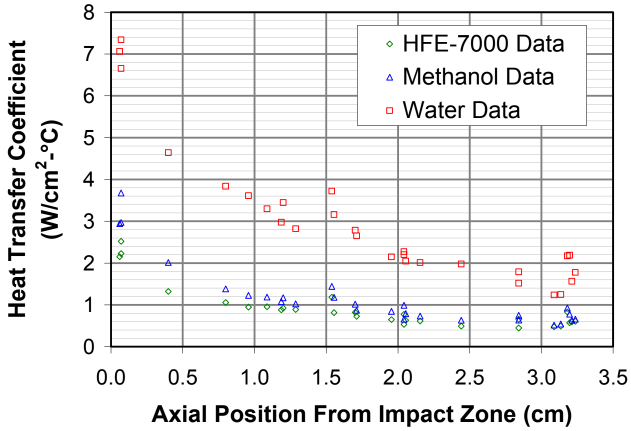

Figure 3: Summary plot of experimental local heat transfer coefficients for all three fluids. The highest heat transfer coefficients were measured for water. Methanol heat transfer coefficients were higher than that of HFE-7000 at the impact zone, but matched those of HFE-7000 near the end of the chip surface.

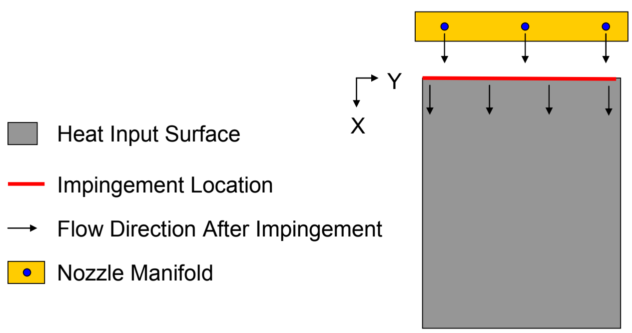

Figure 4: Diagram illustrating the impact region location of the spray on the chip surface and direction of resulting flow. The coordinates referred to in Figures 3, 5, 6 and 7 are also indicated.

The experimental water data are plotted against the local heat transfer coefficient model predictions in Figure 5. Overall there was good agreement with water data in terms of both slope and absolute value. However, there was a tendency for the model to over predict most measured values by approximately 25%. At the impact zone, heat transfer coefficients over 7 W/cm^2-°C were measured. Towards the end of the test chip (3.3cm) heat transfer coefficient values approached 1 W/cm^2-°C.

The experimental methanol data are plotted against predictions in Figure 6. The model predictions for methanol tended to over predict values by 50%. The maximum heat transfer coefficient in the impact zone was 3.6 W/cm^2-°C. The minimum measured heat transfer coefficient was 0.5 W/cm2-°C.

The experimental HFE-7000 data are plotted against predictions in Figure 7. The model predictions for HFE-7000 tended to over predict values by 25%. The maximum heat transfer coefficient in the impact zone was 2.5 W/cm^2-°C. The minimum measured heat transfer coefficient was 0.45 W/cm^2-°C. Interestingly, the measured values for the heat transfer coefficient of HFE-7000 at the end of the chip were very similar to that of methanol, even though methanol had a significantly higher heat transfer coefficient in the impact zone. It is expected that the increased viscosity of methanol causes a faster rate of momentum loss in the boundary layer near the wall compared to HFE-7000. The momentum loss causes a boundary layer thickening and reduction in heat transfer coefficient.

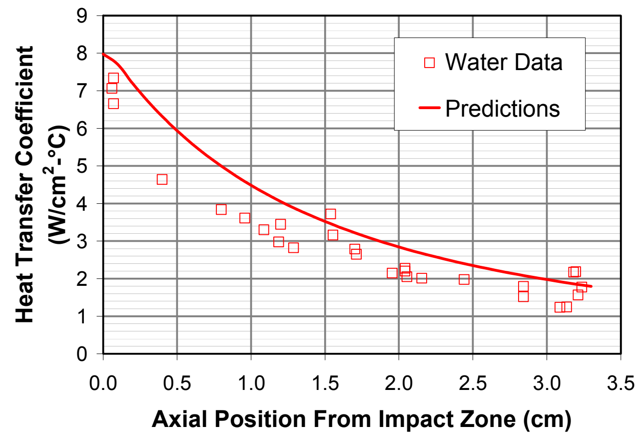

Figure 5: Local spray cooling data for water are shown, illustrating the decrease in heat transfer coefficient as a function of length. Comparison to model predictions is also shown.

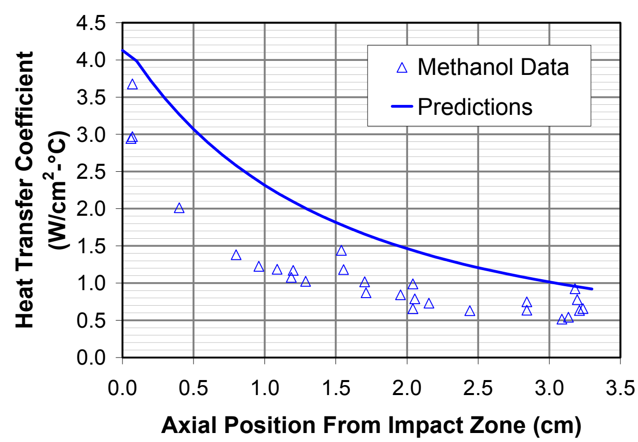

Figure 6: Local spray cooling data for methanol are shown, illustrating the decrease in heat transfer coefficient as a function of length. Comparison to model predictions is also shown.

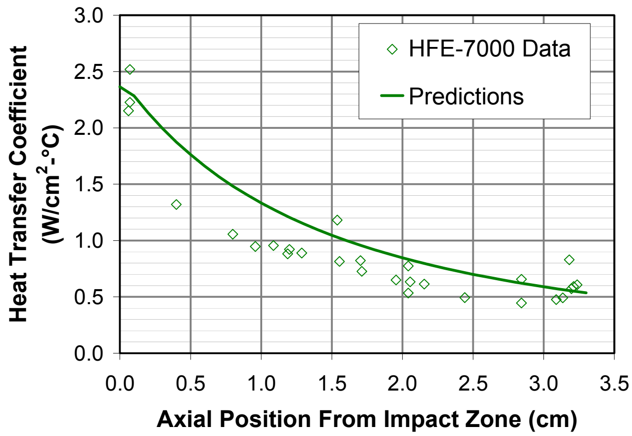

Figure 7: Local spray cooling data for HFE-7000 are shown, illustrating the decrease in heat transfer coefficient as a function of length. Comparison to model predictions is also shown.

5. Conclusions

Experimental data were acquired for a flat spray cooling system with nozzles angled to the surface of a silicon chip. The use of a thermal test vehicle allowed for local accurate measurements of the heat transfer coefficient across the chip. Data was acquired for water, methanol and HFE-7000 at ambient and sub-ambient temperatures. Water performed the best, and would be chosen for applications requiring ambient junction temperatures and above. Methanol slightly outperformed HFE-7000 in the sub-ambient temperature ranges. However, HFE-7000 is the fluid of choice in the cold temperature range because the better performance associated with methanol does not justify it's the flammability, toxicity and chemical compatibility concerns. A local heat transfer coefficient model for the flat angled spray cooling system was derived by combining a Martin slot nozzle correlation with a Tawfek model for angled impingement. The spatially averaged model was than differentiated to determine a local heat transfer coefficient model. The model was compared to experimental data with good success.

Acknowledgments

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References

1. Incropera, F.P. et al, Heat transfer in electronic and microelectronic equipment, Ed. A. E. Bergels, Hemisphere Publishing Corporation, (New York 1990), pp. 407-444.
2. Lee, D.Y., Vafai, K., "Comparative analysis of jet impingement and microchannel cooling for high heat flux applications," Int. J. Heat Mass Transfer, 42, pp. 1555-1568, 1999.
3. Wright, W.B., 2004, "An evaluation of jet impingement heat transfer correlations for piccolo tube applications," April 2004, AIAA 2004-0062, pp. 1-17, 2004.
4. Ma, C.F., Gan, Y.P, Tian, Y.C., Lei, D.H., Gomi, T., "Liquid jet impingement heat transfer with or without boiling," J. of Thermal Science,2(1), pp. 32-49,1993.
5. Wen, M., Jang, K., "An impingement cooling on a flat surface by using circular jet with longitudinal swirling strips," Int. J. Heat Mass Transfer, 46, pp. 4657-4667, 2003.
6. Martin, H., "Heat and mass transfer between impinging gas jets and solid surfaces," Adv. in Heat Transfer, 13, pp. 1-60, 1977.
7. Bar-Cohen, A., "Thermal management of electronic components with dielectric liquids," Int. J. JSME, 36(1), pp. 1-25, 1993.
8. Tawfek, A.A., "Heat transfer studies of the oblique impingement of round jets upon a curved surface," Heat and Mass Transfer, 38, pp. 467-475, 2002.
9. Tawfek, A.A., "Heat transfer and pressure distributions of an impinging jet on a flat surface," Heat and Mass Transfer, 32, pp. 49-54, 1996.
10. Yang, G., Choi, M., Lee, J.S., "An experiment study of slot jet impingement cooling on concave surface: effects of nozzle configuration and curvature," Int. J. Heat Mass Transfer, 42, pp. 2199-2209, 1999.