Heat Transfer in Evaporation of Dielectric Liquid Films in Different Atmospheres Using Silicon Testchips

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Abstract

In this paper we will discuss a possible solution for cooling a large silicon detector system. The idea is to use evaporation cooling from liquid films deposited directly on the VLSI circuits. The intention is to feed liquid to each individual VLSI, using a combination of a liquid pipe line and a wick structure. The liquid which is deposited on the VLSI is then evaporated into an atmosphere which is consisting of vapor and possibly nitrogen.

In the paper we will be concerned with the heat transfer from liquid films deposited on silicon test chips. The experiments have been performed under different atmospheres. We have used both purely vapor and a mixed vapor and nitrogen atmosphere. The total pressure and thereby the liquid saturation temperature was varied during the different experiments. To increase the heat transfer coefficient at low heat fluxes we have also introduced different modifications of the silicon surface in some of the experiments.

Introduction

The cooling of a silicon innertracker detector for the proposed Large Hadron Collider (LHC) at Cern is a most demanding task. This detector system consists of six or seven layers of double sided silicon detector briquettes, over a total area of more than 10m^2 . The whole detector will be installed inside a "cylinder" having a diameter less than 0.8 m, and a total length of between 1.0 and 1.5 m. There is about 10 million individual detectors, and in the order of 100 000 VLSI circuits for signal amplification and read out [1]. A total power of between 10 and 15 kW is mainly dissipated by the readout circuits (VLSI), with a heat flux of approximately 0.4 W/cm^2 . The temperature in the detector system must be kept within a small temperature interval which is in the order of a few degrees. This is partly due to mechanical requirement, which is a dimensional stability in the order of $10\mu\text{m}$. Probably more important is the annealing effects in the silicon detectors themselves, changing their electrical characteristics due to the influence of an extremely high radiation flux.

To reduce the interaction (absorption and scattering) between the radiation and the mechanical structure, a minimum of mass is required in the detector system. From the same reason, materials with a low atomic number (Eg. low Z-materials) must be used. A high magnetic field ($\sim 20\,000$ Gauss), with a B-vector parallel to the cylinder axis is also present.

At the present stage different cooling principles have been proposed for the detector. This includes forced convection cooling using helium gas or water, and evaporative cooling using dielectric liquid.

In this paper we will discuss one possible solution using evaporative cooling from liquid films deposited directly on the VLSI. The intention is to feed liquid to each individual

VLSI, using a combination of a pipe line and a wick-structure. The liquid is then evaporated into an atmosphere which is consisting of vapor and nitrogen. The paper will deal with the heat transfer from the liquid films deposited on silicon test chips both from a conceptual and an experimental point of view.

The experiments have been performed under different atmospheres, where we have used both pure vapor and a mixed vapor and nitrogen. Forced convection of the atmosphere has not been introduced, but the total pressures have been varied during the experiments. To increase the heat transfer coefficient at low heat fluxes we have introduced different modifications of the silicon surface in some of the experiments.

The use of a wick structure is proposed due to the problems involved in manufacturing and maintenance of a large number (100 000) nozzles for impingement cooling. A single wick can be used for the cooling of an array of VLSI circuits. The reliability will probably also increase due to the redundancy of different liquid flow paths.

Nomenclature

a : thermal diffusivity [m^2/s]
 A : area of test chip [m^2]
 c_p : specific heat capacity
 d : thickness of liquid layer
 g : acceleration of gravity [m/s^2]
 h : convection heat transfer coefficient [$\text{W/m}^2\text{K}$]
 h_{fg} : heat of vaporation [J/kg]
 l : characteristic length [m]
 l_p : perimeter of the heat source (test chip) [m]
 M_A : molecular weight [kg/kmol]
 p : pressure [Pa]
 q'' : heat flux [W/m^2]
 Q'' : heat flow [W]
 Ra : Rayleigh number (dimensionless)
 T : temperature [K]
 V'' : volume flow [m^3/s]
 β : expansion coefficient [$1/\text{K}$]
 μ : dynamic viscosity [kg/sm]
 ν : kinematic viscosity [m^2/s]
 ρ : density [kg/m^3]
 σ : surface tension [N/m]

Subscripts

sat : saturated
l : liquid
v : vapor

Theory

Two different modes of cooling must be considered when cooling a VLSI circuit by a thin (in the order of a few

hundred microns) liquid layer flowing on the surface. At low heat fluxes, the heat is transported into the liquid by diffusion and convection. Generally the heat absorbed by the liquid will increase the internal energy (the temperature) of the liquid. The increased temperature will also increase the evaporation from the liquid surface. Analysis shows however that the effect of evaporation directly from the liquid surface will correspond to a heat transfer coefficient which is three orders of magnitudes less than what is given by the heat capacity of the liquid [4]. This is due to a stationary vapor atmosphere in the present experiments. The effect of surface evaporation can therefore be neglected.

When the heat flux increases vapor bubbles are formed inside the liquid, at or very close to the heater surface. These introduce a new mode of cooling, where the combined heat of evaporation and the increased mass transport increases the cooling capabilities significantly. This mode is usually called nucleate boiling.

The liquid used during these experiments was the Fluorocarbon liquid FC-87 made by 3M. The liquid properties are given in Table 1. Given a low surface tension liquid as in the present experiments, the point of transition into nucleate boiling will be very dependent on the surface structure and material properties. This effect is well known from pool boiling [2,3], where it has been a serious limitation in the use of evaporation cooling in electronic systems.

Expansion coeff.	$1.57 \cdot 10^{-3} [1/K]$
Thermal diffusivity	$2.9 \cdot 10^{-8} [m^2/s]$
Heat of evaporation	$88.5 [J/kg]$
Dynamic viscosity	$4.47 \cdot 10^{-4} [kg/sm]$
Kinematic viscosity	$2.6 \cdot 10^{-7} [m^2/s]$
Density liquid	$1750 [kg/m^3]$
Density vapor	$20 [kg/m^3]$
Surface tension	$8.9 \cdot 10^{-3} [N/m]$
ds/dT	$1.0 \cdot 10^{-4} [N/m]$

Table 1. Liquid (vapor) properties of FC-87 at 1 atm.

Before we continue with our discussion of nucleate boiling, we will look into what happens during the first mode of cooling.

Impingement cooling

During the impingement cooling, a free liquid surface is exposed on top of the VLSI. This temperature gradient creates surface tension induced convection in the liquid. The Marangoni number (B) which is used to characterize this type of convection is given by [5]

$$B = \sqrt{F((\delta\sigma/\delta T)\Delta T d, \rho, \nu)} \quad (1)$$

An estimate of the buoyancy driven heat convection can be found by the Rayleigh's number which is defined as [6]

$$Ra = \sqrt{F(g\beta\Delta T d^3, \nu)} \quad (2)$$

The ratio B/Ra in Fig. 1 shows the relative importance between surface tension driven and buoyancy driven convection. With a liquid layer thickness in the order of

two or three hundred microns the heat transfer due to buoyancy can therefore be neglected.

Figure 1. The ratio B/Ra using FC-87 at room temperature.

During impingement cooling, convective currents are introduced due to the impingement velocity. Both the surface tension induced convection and the convection introduced by the impingement will participate in the mixing of the liquid on the VLSI. Due to these combined convective currents the liquid temperature will probably be quite homogenous when it leaves the heater surface.

To calculate the heat transfer coefficient in impingement cooling we also assume an infinite good contact between the heater surface and the liquid. The heat flux is then given by

$$q'' = \sqrt{F(c_p V'' \rho, A)} \quad (3)$$

Wick cooling

A "wick" in this paper consists of a cotton cylinder with one flat end on the silicon chip surface. Introducing this wick the liquid velocity in the direction vertical to the test chip surface is reduced by two orders of magnitude. Also the free surface of the liquid on the Si chip which is outside the wick structure is reduced. Due to this very much reduced flow speed, and the reduced free surface we assume the convective currents to be negligible. The result is a laminar flow, with very little mixing between the different "layers". Thermal diffusion will therefore be the dominating source for heat-transfer during wick cooling.

To discuss the effect of thermal diffusion, let us consider a simplified physical model. A liquid layer of thickness d is in contact with the heating surface. Initially there is a uniform temperature T_0 in the system. At time $t=0$, the heating surface temperature is raised to T_1 . The temperature gradient across the liquid layer in z-direction, and the heat flux into the liquid can now be solved as a function of time [6]. This is used to evaluate the total amount of heat absorbed by the liquid during a time $t = t_0$. The liquid is initially at a temperature T_0 when brought in contact with a heat source having a uniform temperature of T_1 . We choose the time t_0 to be equal to the total transport time of the liquid

flowing across the heat source. Then we have a measure for the total heat flow seen by the liquid.

In Figure 2 we show a simplified drawing of what we mean by liquid transport time. Remembering that the perimeter of the perimeter of the total heat source is $4 \cdot l$ which gives a liquid cross-section of $4 \cdot l \cdot d$ at the edges. The transport distance is $l/2$. The transport time is then given by

$$t_0 = \sqrt{F(2l^2 \cdot d, V'')} \quad (4)$$

Figure 2. Simplified drawing one half of a symmetrical one dimensional model for calculating liquid transport time

Due to the inherent uncertainties of this model we choose to neglect the effects going from a one-dimensional to a two-dimensional flow. The total heat flow dissipated into the liquid during the time $t = t_0$, is given by

$$Q''(t_0) = Q''(t_\infty) [1 - (8/\pi^2) \exp(-\pi^2 \alpha t / (2d)^2)] \quad (5)$$

where $Q''(t_\infty)$ is equal to the total heat flow given a homogenous temperature has been obtained in the liquid.

$$Q''(t_\infty) = c_p V'' \rho (T_1 - T_0) \quad (6)$$

Nucleate boiling in liquid film

In the present experiments the liquid film thickness is several orders of magnitude bigger than the diameter of the bubble nucleus formed in the initial nucleation phase. The initial bubble dimension is typically less than $1 \mu\text{m}$ using FC-87. It is therefore not likely that the nucleation process is significantly changed compared to pool boiling.

On the other side, the bubble diameter at the departure from the test chip is bigger than the liquid film thickness. The bubble speed at the departure will therefore be significantly reduced. This is due to the reduced buoyancy forces and the surface tension of the liquid film surrounding the bubble. This will reduce the effective mass flow, and therefore also the heat-transfer coefficient significantly compared to pool boiling.

Experimental

Square test chips [2] measuring 11.80 mm in both directions have been designed and manufactured at Center for Industrial Research. These test chips contain a large area resistor for power dissipation. A temperature sensitive resistor with a temperature coefficient of approximately $1000 \text{ ppm}/^\circ\text{C}$ is located in the center. Each temperature sensitive resistor has been calibrated individually prior to use. An enclosed test container has been made, in which the experiments have been performed. The temperature in the container was controlled by water flowing in channels inside the container wall.

Prior to the experiments the container and the liquid were degassed. This was done by heating the container until the vapor pressure was well above the atmospheric pressure. Due to the high density of Fluorinert vapor the air was forced to the highest part of the container. Opening the outlet valve by a small amount, the air was forced out by the internal pressure in the module. This procedure was repeated several times to minimize the content of non-condensable gas.

The cooling liquid was supplied to each test chip using a small pump and a syringe needle. The needle had an inner diameter of 0.25 mm . In the different experiments the flow speed was controlled by changing the pumping speed. The flow speed was varied between 0.08 and 0.20 ml/s (equal to $0.14 - 0.35 \text{ g/s}$) in the different experiments. Using this set-up, we obtained a form of impingement cooling, with a liquid flow speed between 1.6 and 4.0 m/s . In some experiments a wick structure was added between the needle and the test chip. This reduced the vertical liquid flow velocity to less than 0.005 m/s . In Fig. 3 we find a simplified drawing of the test setup.

So far in the experiments we have used a plain silicon surface covered with silicon dioxide and approximately 100 nm of gold. We have also used a surface coated with $15 - 50 \mu\text{m}$ stainless steel particles. These particles were glued to the silicon with an adhesive layer of approximately $20 \mu\text{m}$.

One of the experiments was performed with nitrogen at a partial pressure of 0.6 bar introduced into the test chamber. Apart from this one experiment, all the other experiments were conducted with the liquid at the saturation temperature.

Water cooled test chamber

Figure 3. A simplified drawing of the test setup

Results
In Fig. 4, 5, 6, 8 and 9 we find the measured results of heat flux versus the difference between the test chip and the inlet liquid temperature. This difference is labeled over-temperature in the figures. In all the results an over-temperature of 0°C corresponds to a test chip temperature

equal to the saturation temperature of the liquid. The only exception is the two curves in Fig. 6 where nitrogen was introduced into the test container.

The figures clearly show two different cooling modes with a substantially difference in the slope of the curve of heat flux versus over-temperature. At heat-fluxes below 1-2 W/cm², the heat was transported into the liquid by thermal diffusion or convection without any nucleate boiling. In fact the over-temperature in this mode corresponds to the superheat of the liquid. At higher heat-fluxes, the introduction of nucleate boiling changed the slope of the curve of heat flux versus over-temperature. In this mode nucleate boiling become the dominating mode of heat transfer.

Erosion of the "coated" test chips was observed after some hours of impingement cooling. The stainless steel powder in the splash zone was removed, but there were no sign of erosion on the silicon itself.

Figure 4. Wick cooling at different liquid temperatures (60°C, 26°C and 12°C) both on "non-coated" and "coated" test chips. The liquid flow speed was 0.18 ml/s.

Thermal diffusion and convection

From Fig. 5 we see that during the thermal diffusion / convection mode the impingement cooling had a heat-transfer coefficient which was roughly 50% higher than when using a wick structure. From the results it seems as if the "surface" of the test chip had no significant influence on the heat transfer

coefficient during this cooling mode, Fig. 4. The presence of nitrogen did not either seem to make any difference in the heat transfer coefficient. The small difference seen in Fig 6 is due to a 15% reduced liquid flow speed.

Transition to nucleate boiling

The transition from convective to nucleate boiling heat-transfer was obtained with a heat flux of between 0.7 and 2.5 W/cm². This corresponds to a liquid superheat of between 10 and 25°C. The transition was influenced by:

- A) The use of impingement or wick cooling (Fig. 5)
- B) The "surface" of the testchip (Fig. 4)
- C) The degree of sub-cooling of the liquid (Fig. 6)

It was generally observed that impingement cooling gave a smooth transition into nucleate boiling for both the "non-coated" and the "coated" test chips, see Fig. 5 and 9. The transition occurred however at lower over-temperatures and heat-fluxes for the test chips which were "coated" compared to the "non-coated".

Figure 5. Impingement cooling versus wick-cooling, at a liquid flow speed of 0.18 ml/s.

In the similar experiments with wick structures, there were observed large differences in behavior between the "non-coated" and the "coated" test chip. The "coated" test chip showed a very smooth transition into nucleate boiling. This transition occurred at a heat-flux and over-temperature which were lower than observed for any of the tests on impingement

cooling. For the "non-coated" test chips the transition was on the contrary much more dramatic. In this case the transition also occurred at much higher heat-fluxes and over-temperatures. We observed significant "thermal overshoots" during these experiments, Fig. 4 and Fig. 5. In Table 2 we have compared the observed over-temperatures before nucleate boiling for all experiments.

	Over-temperature
Wick "non-coated"	15-25 °C
Wick "coated"	10-18 °C
Imp. "non-coated"	18-22 °C
Imp. "coated"	15-18 °C

Table 2. Observed over-temperature before nucleate boiling

In the experiment where nitrogen was introduced to the chamber, the transitions into nucleate boiling were shifted to higher heat-fluxes and temperatures, Fig. 6.

Figure 6. The influence of a partial pressure of 0.6 bar of nitrogen. The flow speed was 0.18 ml/s during the experiment without, and 0.15 ml/s in the experiment with nitrogen

Nucleate boiling

When nucleate boiling was introduced the slope of the heat-flux versus temperature curve (dq''/dT) increased by a factor of approximately 4 using a wick. The difference in heat-transfer coefficient between "un-coated" a "coated" testchip was small.

During impingement cooling the change in slope was somewhat different. The change was a factor of 2.5 - 3 on "non-coated" test chips. This gave a quite similar heat transfer coefficient to what was observed for wick cooling. On coated test chips however the changes in the slope of the heat-flux versus temperature were increased by a factor of approximately 4. The result was a significantly increased heat transfer coefficient on the "coated" surface, Fig. 5. In Table 3 the slopes of the heat flux versus temperature have been listed for the different experiments, and compared to similar data from pool boiling.

	"Heat transf. coeff." (dq''/dT)
Wick "non-coated"	$\approx 0.35 \text{ W/cm}^2\text{K}$
Wick "coated"	$\approx 0.35 \text{ W/cm}^2\text{K}$
Imp. "non-coated"	$\approx 0.35 \text{ W/cm}^2\text{K}$
Imp. "coated"	$\approx 0.6 \text{ W/cm}^2\text{K}$
Pool boiling	$\approx 3.0 \text{ W/cm}^2\text{K [2]}$

Table 3. Observed slope of heat-flux versus over-temperature during nucleate boiling

Discussion

Wick cooling

A model for the heat transfer during wick cooling has been discussed earlier, Eq. 4-6. The reduction in heat transfer coefficient which was expected from this model was clearly observed in Fig. 5. In Fig. 7 we have calculated of the ratio $Q''(t_0)/Q''(t_\infty)$ from Eq. 5 for a liquid flow rate of 0.18 ml/s.

Figure 7. The ratio $Q(t) / Q(\infty)$ for a liquid flow rate of 0.18 ml/s

In this calculation we have used the actual dimensions of the chip. From this figure we can see that a 50% increase in heat transfer coefficient ($Q''(t_0)/Q''(t_\infty) \approx 0.67$) during impingement cooling corresponds to a liquid thickness of approximately 0.15 mm. The actual observed film thickness was probably in the order of 0.2 mm. Taking into consideration the two-

dimensional flow pattern of the liquid would have given a better agreement between calculated and observed values.

Impingement cooling

In Fig. 8 we have plotted the heat flux versus temperature for impingement cooling (from Fig. 5) and compared it to the theoretical results from Eq. 3. The measured result is very close to what was predicted by this model. This confirms the assumption that the convective currents due to surface tension and the impingement have given a homogeneous mixing of the liquid.

Figure 8. Experimental and theoretical heat flux versus temperature for impingement cooling.

Transition to nucleate boiling

During wick cooling on the smooth surface a significant over-temperature and hysteresis effect was observed. Experiments on rough surfaces showed no noticeable hysteresis effect at all. This can be explained by the fact that nucleation centers were created at the stainless steel particles. This was much the same behavior as have been observed in "pool-boiling" [2]. This was expected due to the thickness of the liquid layer which was order of magnitudes higher than the diameter of the bubble nucleus formed in the initial nucleation phase.

Impingement cooling gave a much smoother transition to nucleate boiling. This can probably be due to that small "pockets" of vapor were embedded in the liquid in the splash zone.

In the experiment where a partial pressure of 0.6 bar of nitrogen was introduced into the container the saturation temperature of the liquid was changed from approximately 30°C until 42°C. This accounts for the observed shift in the transition temperature.

Nucleate boiling

The heat transfer coefficients observed during nucleate boiling were significantly smaller than what we previously have observed in pool boiling. This was expected due to that the liquid film thickness was small compared with the bubble size at departure from the test chip.

We observed an increased heat transfer coefficient during impingement cooling on "coated" surfaces. This can be explained by the assumption that the combination of high liquid flow-speed and stainless steel particles will introduce many nucleation centers. Generally it is assumed that the heat-transfer coefficient is a monotone increasing function of the number of nucleation centers [7].

Figure 9. Heat transfer from impingement cooling at different temperatures (10°C and 28°C). The liquid flow speed was 0.15 ml/s

Conclusions

The experiments have shown that impingement generally gave better results for cooling than using a wick. On the other hand, for a large array of VLSI circuits wick cooling

will probably be more convenient both from a manufacturing and a reliability point of view.

During impingement cooling we observed a good agreement between the theoretical and the measured results. The same was also the case for wick cooling.

The heat transfer coefficients during nucleate boiling were significantly reduced compared to pool boiling. This is probably because the bubble diameter at departure from the test chip was larger than the thickness of the liquid layer.

Due to a stagnant layer of vapor during the experiments, evaporation from the surface of the liquid was of negligible influence on the results.

Further work

The work so far has shown that it could be possible to cool a silicon innertracker detector for the proposed Large Hadron Collider (LHC). To be feasible for this application the liquid flow over the heat sources must be significantly decreased. To make a detector system based on transpiration cooling a well controlled flow of inert gas must be purged across the evaporating liquid. This will increase the convective mass transport dramatically. On the other hand, to emphasize nucleation boiling, further work must be done to reduce the over temperature before transition to nucleate boiling.

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