Thermal Management of Low Volume Complex Electronic Systems

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Abstract Advances in the field of electronics have resulted in a significant increase in density integration, clock rates, and emerging trend of miniaturization of modern electronics. This resulted in dissipation of high heat flux at the chip level. In order to satisfy the junction temperature requirements in terms of performance and reliability, improvements in cooling technologies are required. The task of maintaining acceptable junction temperature by dissipating the heat from the integrated circuit chips is a significant challenge to thermal engineers. Much work has been done on cooling one hot spot with one heat sink, but there has not been as much investigation into cooling multiple hot spots with a single heat sink. A circuit board with a specific geometry and chip arrangement will be cooled using a liquid-cooled cold plate.

Keywords Thermal management · Cold plate · Junction temperature · Pressure drop · Thermal resistance

1 Introduction

The authors concluded that thermal management must be taken into account early in the design process. Room must be made for the cold plate to fit the electronics, and components that do not need to be cooled should not interfere with the cold plate.

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Electronics cooling is viewed in three levels, which are non-separable. First, the maintenance of chip temperature at a relatively low level despite high local heat density. Second, this heat flux must be handled at system or module level. Finally, the thermal management of the computer machine room, office space, or telecommunication enclosure. The thermal design of the system is influenced by the key drivers such as chip size, power dissipation, junction temperature, and ambient air temperature.

The high chip temperature results in thermal failures such as mechanical stresses, thermal de-bonding, and thermal fracture. The failure in electronics during operation occurs mainly due to temperature. Therefore, thermal management is a key enabling technology in the development of advance electronics. The main constraint for any thermal management is the cost. Therefore, the cooling technology must be cost-effective and keep pace with the reduction in overall package and system cost per function. The cost of cooling is also recognized as a factor playing important role in maintaining competitiveness.

2 Earlier Research

There has been a dramatic shift in cooling high-power devices in the industry during the past decade. Although it is quite difficult to make a distinction based on total power dissipation, it seems that beyond a range of about 1,500 W dissipation, there are many physical and design constraints that may dictate a shift toward liquid as the preferred medium.

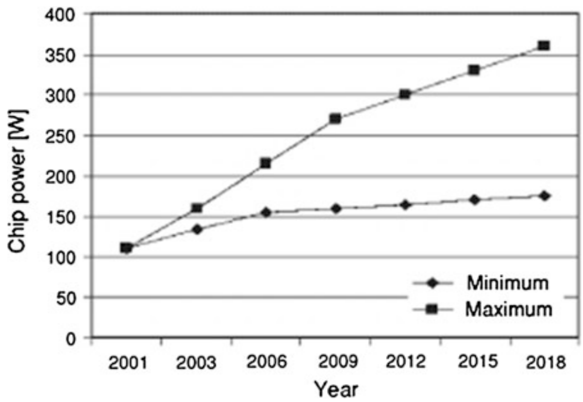
Nottage (1945) suggested that the heat sink fin and channel might be thought of as a type of heat exchanger. The solid fin is considered as a hot stream. The flow stream direction relative to heat flow direction plays a significant role in determining the heat transfer effectiveness of a fin-fluid arrangement. The counterflow arrangement has the greatest potential to achieve high effectiveness [Evolution of cold plate design].

Jones and Smith (1970) performed a similar study for rectangular fin assemblies facing upward and downward in relation to the gravity vector. In both studies, the correlation equations were restricted to a fixed range of geometric and flow conditions limiting their use as general-purpose design tools [Evolution of cold plate design].

The need for liquid cooling has resulted in a major paradigm shift for many electrical system designers and manufacturers. This brought the realization that liquid cooling offers a better, and often the only feasible, solution. The compactness of the cold plates and the supply and return lines, lower power consumption, and reduction in noise levels are some of the attractive features.

The use of microchannels as a viable cooling solution was first proposed in 1981 by Tuckerman and Pease, who designed and tested an integral, water-cooled

Fig. 1 High-performance chip power trend



heat sink and achieved a high heat flux of 790 W/cm^2 with a temperature rise of 71 °C above the inlet water temperature. Microchannel heat sinks are one of the effective cooling methods for high-power density and compact electronic devices according to Zhao. Further confirmation was provided by studies done by Chien-Hsin.

The available literature on the cold plate design is primarily focused on relatively low power dissipation. A number of thermal management solutions in use for cooling power electronic modules in automotive applications are reviewed in Nakayama et al. (1984) and Garg and Velusamy (1986). The coolant in such cooling must be accomplished with a low temperature difference between the semiconductor and the coolant (Fig. 1).

3 Research Objectives

The objective of this project is to design and fabricate a highly effective, liquid-cooled cold plate for high heat flux cooling. This project is unique in that the main objective was to accommodate chips of different heights and power densities.

The cold plate requirements are listed below. The cold plate must work in low-temperature environments, as well as environments where both the magnitude and direction of the gravity force are constantly changing, while remaining inexpensive.

The cold plate must accommodate the complex geometry of the multi-chip circuit board, and different chip heights must be accommodated while maintaining good conduction contact.

Requirement	**Current Design**	**Standard Goal**	**Stretch Goal**
Power Dissipated by the module	16.6W	200W	400W
Highest Power Concentration	2.96W/in2 (0.46W/cm2)	64.5W/in2 (10W/cm2)	258 W/in2 (40 W/cm2)
Highest Single Power Device	2.7W	60W	150W
Fluid Used	N/A	50/50 Propylene Glycol/Water	PAO
Pressure Drop	N/A	20 psi (138 kPa)	2.5psi (18 kPa)
$T_{max,module}$-$T_{fluid,inlet}$	30 C	10 C	3 C
Maximum Module Thickness	.125 in(.318cm)	.25 in (.635 cm)	.1 in. (.25 cm)
Inlet Fluid Temperature Range	N/A	0 C to 4 C	40 C to 60 C

Fig. 2 Project objectives

4 Experimental Setup

A test section was designed and constructed to evaluate the capabilities of the cold plate. The test section was designed to emulate the geometry and thermal characteristics of the circuit board.

It can be seen that Chips 1–4 have the highest heat flux. Attention in this project was focused on finding a thermal solution capable of cooling those chips. Anything with a smaller heat flux should be cooled just as effectively by the same cold plate (Fig. 2).

4.1 Test Section Description

The test section consists of a rectangular G10 fiber glass composite board 88 mm wide, 225 mm long, and 2.50 mm thick. G10 is a fiberglass that is often used in low-temperature applications. Four copper squares were made. Three of the

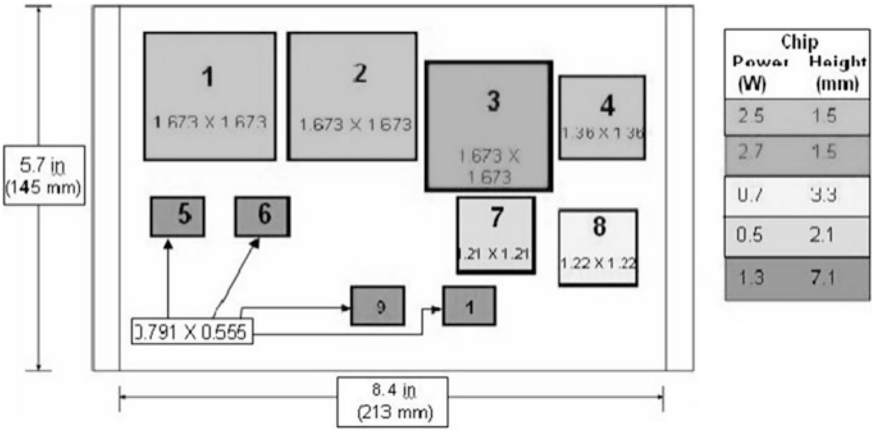


Fig. 3 Test section

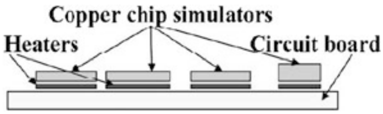


Fig. 4 Test section schematic

squares are 42.5 mm on a side and 2 mm thick. The fourth square is 34.5 mm on a side and 3.5 mm thick. These copper squares are the same size and shape as Chips 1–4 in Fig. 8. Each copper square has a heater attached to the back of it for heat generation (Fig. 3).

4.2 Test Section Drawings and Pictures

A schematic drawing of the test section is shown in Fig. 4.

A picture of the fabricated test section is shown in Fig. 5, including the thermocouples on the surface of the copper chip simulators. These thermocouples measured the temperature of the copper chip simulators at the interface between the copper chip simulators and the cold plate.

5 Experimental Procedure

The experimental procedure for the two cold plates is discussed in this section. The procedures do not vary much because most of the data, except for the pressure drop, were taken from the test section. The same test section was used for each experiment.

The test section and cold plate were then placed in contact. The chiller water was circulated through the cold plate at 5 °C for fifteen minutes. Once the chiller

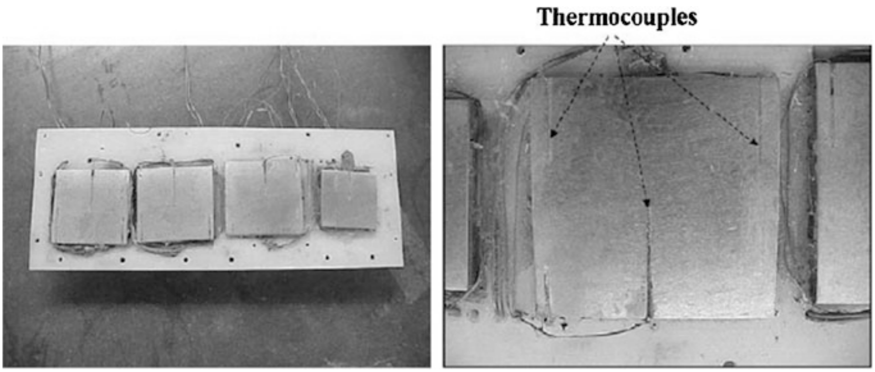


Fig. 5 Test section

water reached steady state and flow rate, power was applied to the heaters. The pressure drop across the cold plate and the temperatures of each thermocouple were recorded both by computer and by hand. The three temperatures recorded for each chip were then averaged to get an average chip temperature. Each steady state reading produced one data point for each copper chip simulator.

5.1 Cold Plate Setup

The cold plate is a U-type cold plate with pressure ports on the inlet and outlet of the cold plate to measure pressure drop (Fig. 6).

5.2 Channels

The cold plate consists of 12 parallel channels fed by the inlet header. Figure 7 shows cross-sectional views of the cold plate revealing the channel geometry.

The channel was compressed to the point where the fins which extended only partway into the channel became joined and the channel walls bent. The direction of the bending of the walls in the compression was random and uncontrollable with the process used to create the compression.

Figure 7 shows the final cold plate. The cold plate has a compression to accommodate a taller chip and pressure ports on both the inlet and outlet.

5.3 Modifications Based on Setup 1: Setup 2 Cold Plate

The first-generation cold plate was constructed and tested. A great non-uniformity was found in the thermal resistances of the chips.

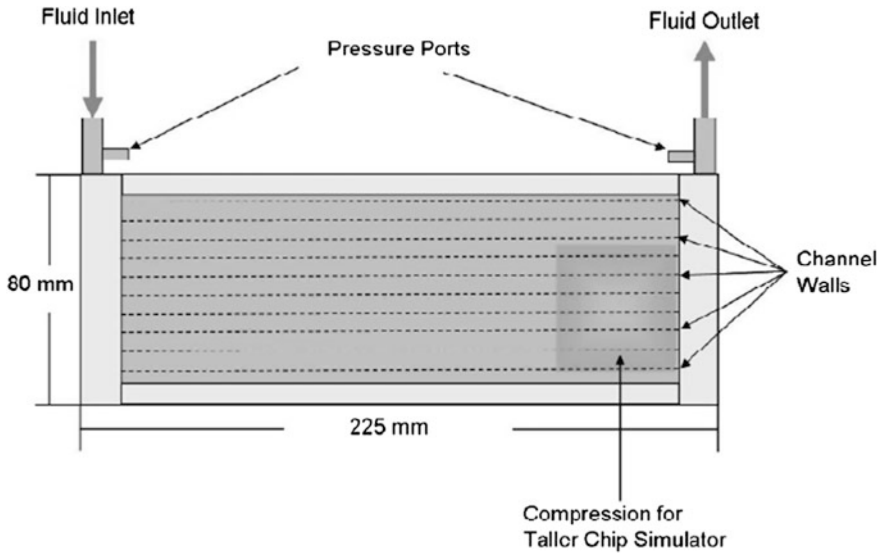


Fig. 6 Cold plate setup 1

Fig. 7 Aluminum section of cold plate

The second-generation cold plate was designed with long headers and short channels, both to take advantage of the entrance effect while also decreasing the pressure drop. The channels of the second-generation cold plate had to be smaller in order to satisfy the thermal resistance stretch goal. Since the pressure drop through a closed channel increases as the hydraulic diameter decreases, the pressure drop through the second-generation cold plate channels was expected to increase.

The modified design reduced the flow length through the channels and thus lowered the pressure drop. Figure 8 illustrates the second-generation cold plate.

The second generation had a smaller hydraulic diameter than the channels of the first-generation cold plate. Smaller channels were chosen for several reasons.

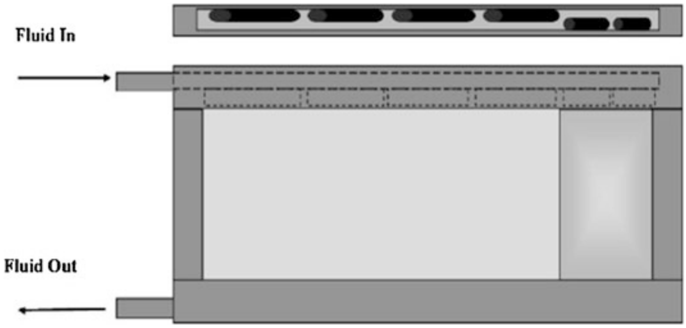


Fig. 8 Fluid path setup 2

The first reason is that the Nusselt number based on the channel hydraulic diameter,

$$\mathrm{Nu}_d = \mathrm{hd}_h/k$$

converges to a constant value, 4.36, in fully developed, laminar, closed channel flows exposed to a constant heat flux. This means that channels with a smaller hydraulic diameter have larger convection coefficients.

The second reason is that channels were smaller in the second-generation cold plate inorder to meet the goal for maximum allowed cold plate thickness. The first-generation cold plate met the standard goal for cold plate thickness, but did not satisfy the stretch goal. As the channel dimension shrinks, the pressure drop in the channels increases.

The channel flow length also had to be shortened to help meet the stretch goal for pressure drop. Since the channels were smaller than the previous cold plate, a new technique was used to accommodate the taller chip height.

Compressing the channels would not have been possible because the channels were not thick enough to accommodate the required depth of the compression. Instead, a common header and fluid inlet and outlet paths were used, but to accommodate the taller chip height, the connecting slot and the slots for fluid distribution were offset within the header. This eliminated the need to compress the channels while still accommodating the taller chip height and remaining easy to design and manufacture.

5.4 Cold Plate Setup 3

5.4.1 Modifications from earlier design

The cold plate designs so far were full contact type. This forced us to keep the design bulky and required extra rigidity from the test section to hold it together.

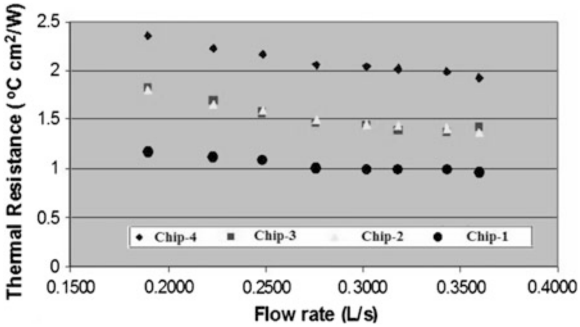


Fig. 9 First-generation cold plate thermal resistance

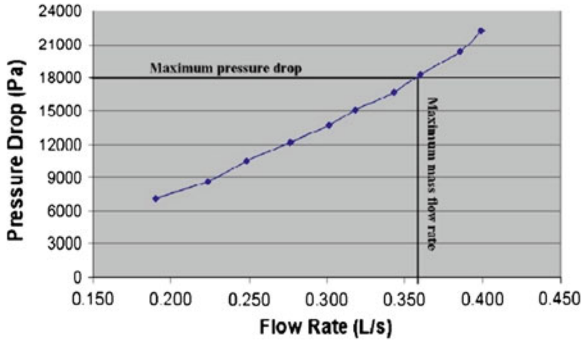


Fig. 10 First-generation cold plate pressure drop

Material of construction remained aluminum as in the earlier designs. At the same time, selective coating was provided to the external structure of the cold plate to reduce emissivity and justify the cooling aspect.

In the new cold plate, flow length was increased slightly, allowing some pressure drop across the section.

6 Results and Discussions

6.1 *First-generation heat transfer*

Figure 9 is a graph showing the first-generation cold plate thermal resistance as a function of the flow rate through the cold plate.

Figure 9 shows that the first-generation cold plate meets the standard goals for thermal resistance while adhering to the pressure drop restriction. The most important feature of the graph is the uneven distribution of the thermal resistances of each of the copper chip simulators.

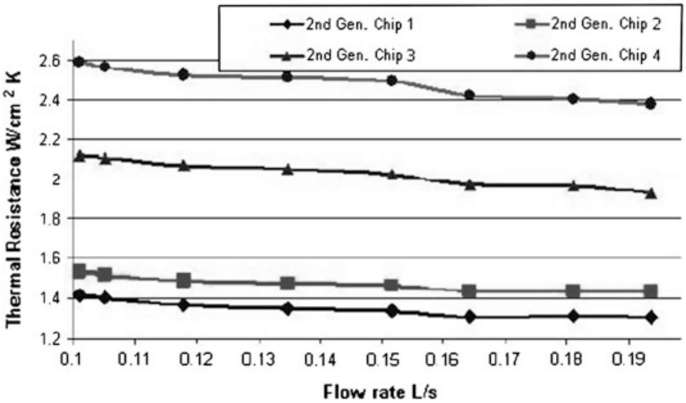
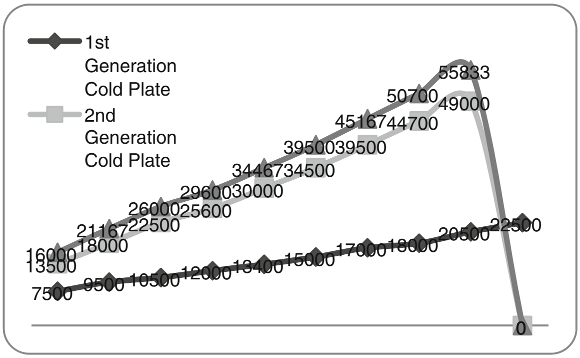


Fig. 11 Second-generation cold plate thermal resistance

Fig. 12 Pressure drop



The maximum pressure drop allowed by the standard goal is shown along with the corresponding flow rate. The thermal resistance goal was satisfied while also satisfying the pressure drop standard goal (Fig. 10).

When compared to the first-generation results, several differences present themselves. The first difference is that the thermal resistance non-uniformity has become greater. Some thermal resistances have increased (Chip 1 and Chip 3), some have decreased (Chip 2), and the thermal resistance for Chip 4 has not changed (Fig. 11).

6.2 Pressure Drop

The pressure drop from the second-generation and the third-generation cold plates is measured as shown in Fig. 12.

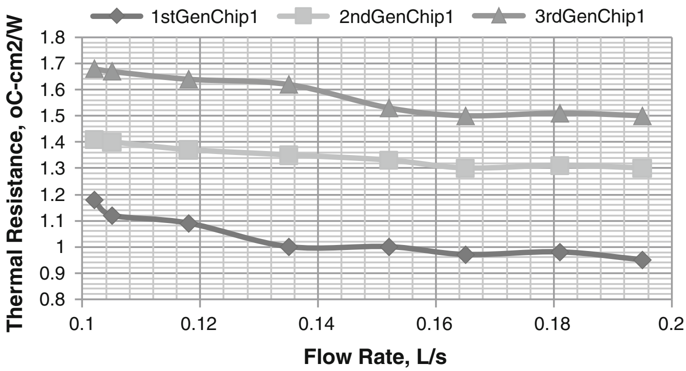


Fig. 13 Thermal resistance comparison for cold plates (Chip 1)

Figure 13 shows that the pressure drop for the second-generation cold plate is greater than the pressure drop for the first-generation cold plate and that for the third-generation cold plate is greater than both previous versions. This is a result of the second- and third-generation cold plates having smaller channels than the first-generation cold plate and the third-generation cold plate having smaller channels than the second.

6.3 Comparisons

There are two important differences in heat transfer between the different setups of cold plates. The first difference is that the cold plates cooled different chips differently.

Switching from one generation of cold plate to another caused the thermal resistances of some chips to increase while causing the thermal resistances of other chips to decrease. The second important difference is that while the range of thermal resistances was about the same for the three cold plates, the range of the second- and third-generation cold plates had higher average temperatures than the first-generation cold plate.

Overall, the second and third setups of cold plates were less effective at cooling the entire test section.

Figures 13, 14, 15, 16 are comparisons of the thermal resistances of Chip 1 and Chip 4. From these figures, the trend is clear that at a given flow rate, the first-generation cold plate is a better heat exchanger than the second-generation cold plate.

The thermal resistance data presented from each chip are composed of three separate thermal resistances: thermal interface resistance, conduction resistance through the cold plate, and convection resistance from the cold plate to the water.

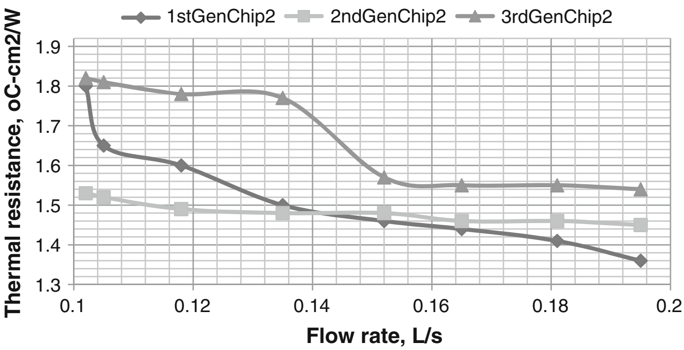


Fig. 14 Thermal resistance comparison cold plates (Chip 2)

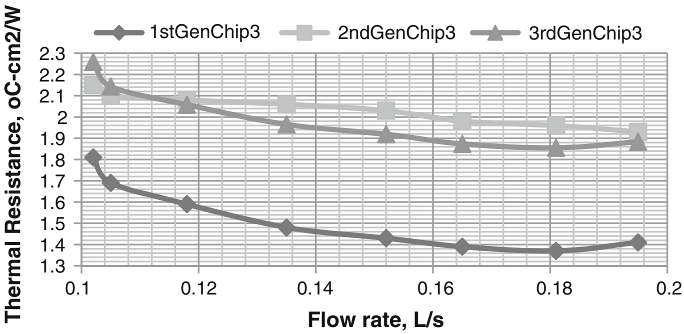
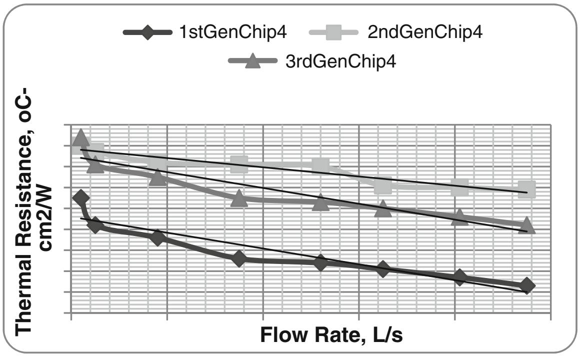


Fig. 15 Thermal resistance comparison cold plates (Chip 3)

Fig. 16 Thermal resistance comparison for cold plates (Chip 4)



Of these resistances, the only one that changes with flow rate are the convection resistance. Therefore, it must be the convection resistance that causes both the high initial thermal resistance and causes it to fall more rapidly in the first-generation cold plate. The source of this trend is believed to be flow maldistribution in the header. Numerical simulations were performed on this cold plate, which showed that there was a jet effect in the header due to the small diameter and high velocity. This jet effect caused fluid to skip over the first few channels in the cold plate resulting in more fluid in the other channels, thus giving the other channels a higher mass flow rate.

It is believed that Chip 2 benefited from this effect because it was cooled by the channels in the middle of the cold plate. As the jet effect increased, more fluid was delivered to the channels cooling Chip 2 in the first-generation cold plate, resulting in a drop in the total thermal resistance.

7 Conclusions

The experimental boundary conditions are well characterized and specified, and the setup is verified to produce good results via a verification of the energy balance achieve and a comparison of single phase measurements to predictions. We observe that:

- The heat transfer coefficient peaks at a thermodynamic vapor quality of approximately 20 % and decreases significantly with increasing vapor quality.
- The heat transfer coefficient shows a strong increase with increasing heat flux.
- The influence of the saturation pressure in the investigated range is negligible.

The header of the second-generation is an elongated version of the first-generation cold plate header. The fluid inlet and outlet paths are the same diameter; however, the paths in the second-generation header are longer.

The difference in pressure drop between the two cold plates has to do with both the header design and the channels. The second-generation cold plate was designed so that most of the pressure drop would take place in the channels.

Due to the high flow rate and the design of the fluid distribution slots, the flow through the channels was not even, and a large amount of the pressure drop is believed to have occurred due to the high velocity in the fluid inlet and outlet paths. This increased the flow maldistribution in the cold plate.

7.1 Heat Transfer Performance

The heat transfer differences are also believed to be strongly dependent on the flow distribution in the header and the channels. The entrance effect alone could not explain the difference in thermal resistance for the first-generation cold plate.

This was further proven by the second-generation cold plate. If the entrance effect were the cause of the improved heat transfer, all the chips in the second-generation cold plate would have benefited from it.

Uneven flow rate in the channels would account for the uneven cooling performance. Channels receiving little or no flow would not cool the chip as effectively. The average temperature for each chip was not very far from the individual thermocouple readings.

This suggests that it is a flow problem and that many channels are affected by it, not just one or two scattered channels.

Today's advanced microelectronic technology is pushing semiconductor devices to their thermal limits, underscoring the importance of a well-designed reliable heat sink. Unlike fuses and semiconductors that can be standardized in size and performance, high-power thermal management is always customized for each individual application.

To select the thermal management method that best meets the needs of application at hand, the OEM should collaborate closely with the heat sink manufacturer at the earliest possible stages of the design project. After carefully considering the application's mechanical, hydraulic, power, and cost requirements, the OEM and heat sink manufacturer will together be able to design a solution that delivers the optimum combination of performance, size, and cost.

Once the fluid flow problem had been dealt with, the solution of the heat transfer problem was undertaken. In recognition of the fact that the needed computer resources for a cold plate whose stream wise dimension is very large are not generally available, a simplified solution strategy was developed and implemented. This strategy takes full advantage of the periodic nature of the problem and, at first, focuses attention on the solution for one of the periodic modules that is representative of the others in the array.

That solution is necessarily conjugate. From the solution to the conjugate problem, heat transfer coefficients are extracted, which facilitate the subsequent heat conduction solution for the temperature distribution in the bounding walls of the entire array. These heat transfer coefficients were subjected to an accuracy check that ensures the quality of the final results.

To illustrate the use of the method, a problem that typifies the thermal management of electronic equipment by means of a cold plate was formulated and solved numerically. The results of the numerical solution were interpreted from the standpoint of the quality of the overall performance.

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