

ABSTRACT

Title of Document: An Innovative Thermal Management
Solution for Cooling of Chips with
Various Heights and Power Densities

Timothy Walter McMillin, Master of Science,
2007

Directed By: Professor Michael Ohadi, Ph.D.
Department of Mechanical Engineering

The challenges and benefits of using a liquid-cooled cold plate to cool a multi-processor circuit board with complex geometry were explored. Two cold plates were designed, fabricated, and tested experimentally. Thermal interface resistance was experimentally discovered and confirmed with numerical simulations.

A circuit board simulator was constructed. This simulator was meant to mimic a multi-processor circuit board with heat sources of different surface areas, heights, and heat dissipations. Results and discussions are presented in this thesis.

THERMAL MANAGEMENT SOLUTIONS FOR LOW VOLUME COMPLEX
ELECTRONIC SYSTEMS

By

Timothy Walter McMillin

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Advisory Committee:
Professor Michael Ohadi, Chair
Associate Professor Tien-Mo Shih
Assistant Professor Bao Yang

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Dedication

To my family, thank you all for your support.

Acknowledgements

First I would like to thank Dr. Michael Ohadi for giving me the opportunity to work for my master's degree at the University of Maryland. I would like to thank Dr. Michael Ohadi, Dr. Serguei Dessiatoun, and Dr. Amir Shooshtari for their help and guidance with the research.

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Chapter 1: Introduction

Broadly speaking, electronic cooling systems can be classified into Air-cooled and Liquid cooled systems. In the following a brief description and advantages/disadvantages of each category are provided.

Air-Cooled Systems

Current Applications of Air-Cooled Systems

Air cooling is still the most common method of heat dissipation for thermal management of electronics. In an air cooling set-up, a heat sink is the only heat exchanger and transfers heat directly from the heat source to the surrounding air. Heat sinks are the most commonly employed, cost effective electronics thermal management hardware in air cooling. Heat sinks come in several shapes and varieties. The main parameters of interest in heat sink design are convection type (forced or natural), heat sink geometry, and heat sink material. These three parameters serve to determine the maximum rate of heat rejection achieved by the heat sink.

Benefits of Air Cooling

There are many benefits to air cooling, but three of the main benefits are reduced cost, simplicity of design, and increased reliability. Air-cooled systems have at most two components: the heat sink and the fan. Heat is transferred directly from the source to the heat sink and is dissipated to the surrounding air. It is this simplicity which results in reduced cost and increased reliability. Pictures of a typical air-cooled heat sink with and without a fan are shown below in Figure 1.

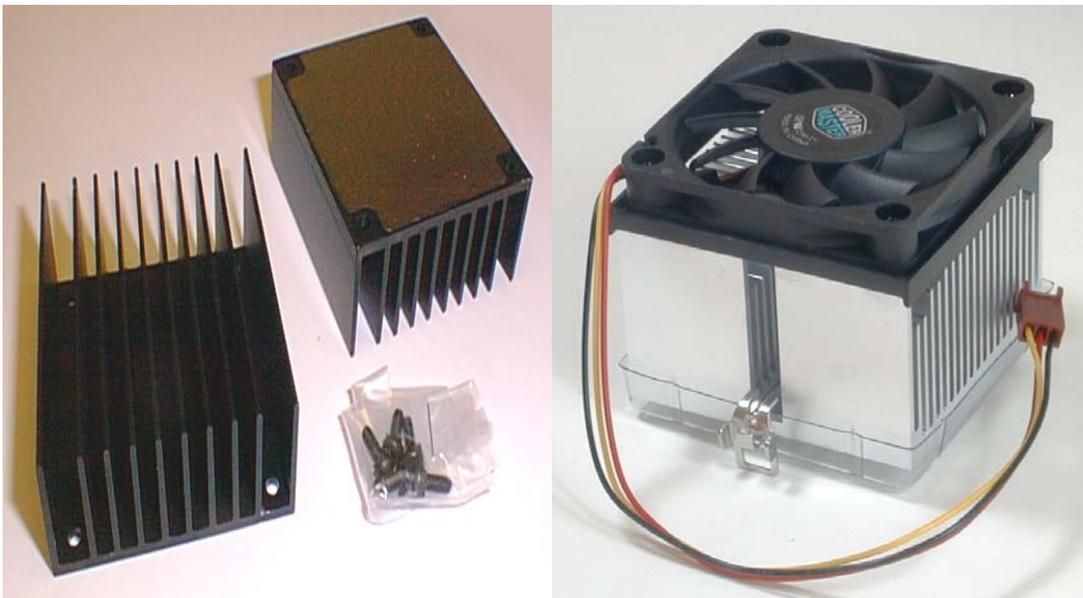


Figure 1: Air-cooled heat sink (a) without fan and (b) with fan [1]

Limits of Air Cooling

As microprocessors increase in speed, their heat dissipation also increases. An Intel Pentium 4 2.40 GHz processor dissipates 58 Watts of heat. Using this power dissipation, a simple case study is performed below to estimate the heat sink temperature necessary to dissipate 58 Watts of heat to air at standard temperature and pressure [2]. The geometry is shown first, followed by the assumptions used in the calculations. Finally the results will be presented. The heat sink geometry is shown below in Figure 2.

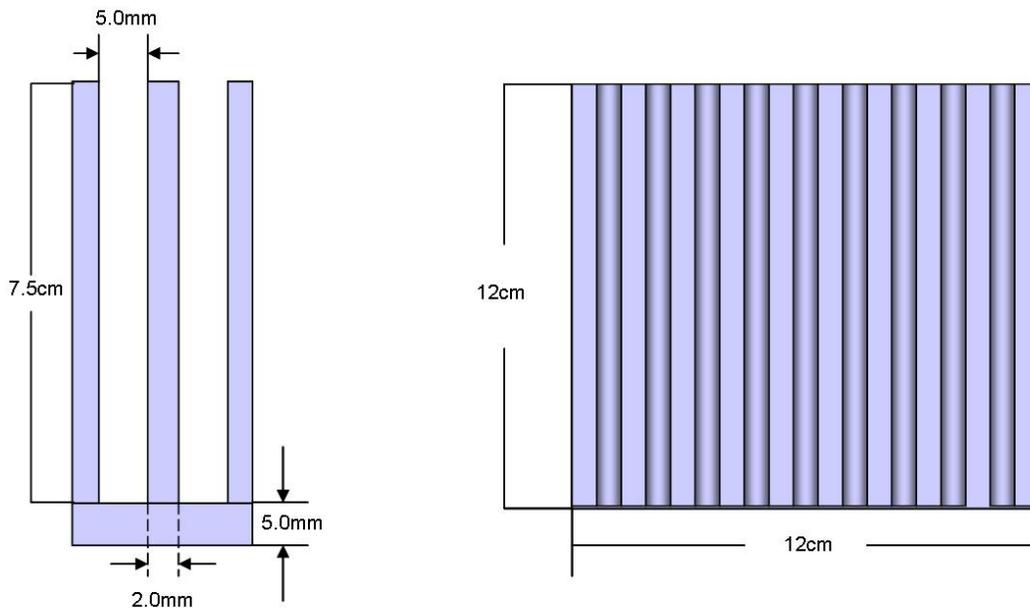


Figure 2: Heat Sink Geometry Used for Case Study

The dimensions are similar to the dimensions of a typical Pentium 4 air-cooled heat sink. For simplicity a uniform heat flux condition is imposed over the entire surface of the heat sink. This neglects

the temperature variation in space due to conduction thermal resistance. The flow in each unit cell is modeled as flow over three flat plates—two vertical and one horizontal—and the flow is assumed to move parallel to the fins and base plate. A correlation for calculating the Nusselt number for flow over a flat plate of length x , Nu_x , was obtained from Incropera's *Heat and Mass Transfer*, 6th Edition [3]:

$$\text{Equation 1: } Nu_x = 0.453Re_x^{1/2} Pr^{1/3}$$

The Nusselt Number, Nu_x , is defined below in Equation 2:

$$\text{Equation 2: } Nu_x = \frac{hx}{k}$$

A typical computer fan is capable of providing a flow rate of 89.39 CFM at a velocity of 0.042m/s, a power consumption of 6W, and a noise level of 32dB [4]. Air properties were evaluated at room temperature, 25°C. The Reynolds (Re) number for this case is 12117 and Nu_x was calculated to be 44, leading to a heat transfer

coefficient $\bar{h}_x = 19 \frac{W}{m^2 K}$. Assuming a heat dissipation of 58W and an ambient air temperature $T_\infty = 25^\circ C$, the heat sink surface temperature would have to be $44^\circ C$ to dissipate the required heat. Using air cooling, as the heat dissipation increases either the temperature or the fan speed must also increase. Increasing the temperature is undesirable, however, because it will reduce the reliability of the microprocessor and lead to earlier chip failure. Increasing the fan speed is also undesirable because the reliability of the fan will decrease and the noise will quickly reach unacceptable levels, especially for the home consumer. While a $44^\circ C$ chip surface temperature is feasible, a 32dB noise level is unacceptable for some markets and air cooled systems are reaching the heat dissipation limit as a thermal management solution for electronics cooling.

Liquid-Cooled Systems

Current Applications of Liquid-Cooled Systems

Liquid cooling has entered the market as a viable thermal management option. In a liquid-cooled system a secondary fluid acts as a heat spreader to more efficiently remove heat from the

source before it is dissipated to the air. Heat generated by electronic components is transferred first to the secondary fluid and then to the air via a heat exchanger. In all cooling methods the final heat rejection will be to the surrounding air. Liquid cooling use began with high power microprocessors and power electronics. As home computers become more powerful, liquid cooling has begun to penetrate that market as well. Liquid cooling provides a quiet, efficient, low-energy method of heat dissipation.

Overview of typical liquid-cooled cold plate designs

The most basic definition of a cold plate is a thermally conductive metal shell with liquid flowing inside it. One side of the conductive metal shell is placed in contact with the heat source. Heat is conducted through the metal shell and removed through convection by the fluid flowing on the other side of the shell. Variation in cold plate designs occurs mainly in the shape of the conducting shell and the fluid path.

The most basic cold plate is a conduction cold plate. In a conduction cold plate, the entire cold plate is a solid piece of metal. Heat is conducted to the edge of the cold plate, where it is removed by convection. In a conduction cold plate no fluid flows through

the cold plate. Figure 3 and Figure 4 below are examples of a conduction cold plate and a convection cold plate.

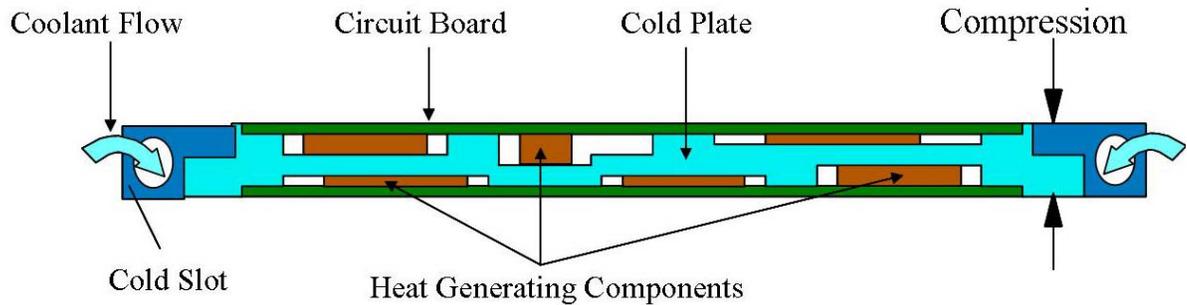


Figure 3: A conduction cold plate

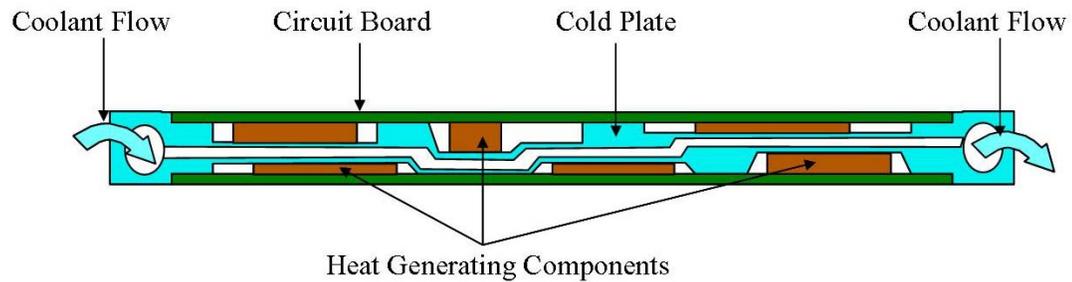


Figure 4: A convection cold plate

The second type of cold plate design consists of a tube which winds back and forth through a metal block. Fluid is pumped through the tube, and heat is removed from the block which conducts heat away from the source. This type of cold plate is referred to as a tubed cold plate. A picture of a tubed cold plate is shown below in Figure 5.



Figure 5: A tubed cold plate [5]

Another type of cold plate is called a flat tube cold plate. The tubes from the tubed cold plate are removed, and smaller channels are cut directly into the metal block or shell, which is in contact with the heat source. Flat tube cold plates offer lower thermal resistance in a more compact design. A picture of a flat tube cold plate is shown below in Figure 6.



Figure 6: A flat tube cold plate with both Z and U type configurations [6]

Benefits of Liquid Cooling

The main benefit of liquid cooling is the ability to dissipate higher heat fluxes at lower temperature differences. Returning to the case study done for air cooling, but changing the fluid to water while keeping the geometry and free stream velocity the same,

results in $\bar{h}_x = 84 \frac{W}{m^2K}$, or a nearly five-fold increase in heat

transfer. Slowing the water velocity to 0.02 m/s would result in a

convection coefficient of $\bar{h}_x = 19 \frac{W}{m^2K}$. The result is the same heat

dissipation, less power input, and quieter operation.

This thesis covers the challenges and benefits of adapting the liquid cooling method to create a cold plate for use in multi-heat source cooling. Challenges include accommodating complex geometries, minimizing costs, and minimizing thermal interface resistance.

Chapter 2 of this thesis will introduce and describe the project including circuit board geometry, and project goals for heat dissipation, pressure drop, and cold plate thickness.

Chapter 3 will discuss the experimental apparatus and data collection system and how it was manufactured.

Chapter 4 is a description of the first generation cold plate and the method used in its manufacture.

Chapter 5 is a description of the second generation cold plate and the method used in its manufacture.

Chapter 6 is a discussion of the origins of thermal interface resistance and the methods and experiments used to minimize the thermal interface resistance in this project.

The results and a discussion of the results will be presented in Chapter 7.

Chapter 8 will summarize the conclusions and suggest future work in this field.

Chapter 2: Project Description

Project 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. 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 the liquid-cooled cold plate. Figure 7 below shows the geometry and heat dissipation of the first-generation circuit board. The cold plate is being designed to cool the second-generation circuit board. The geometry will remain the same but the heat dissipation will increase with the second-generation circuit board.

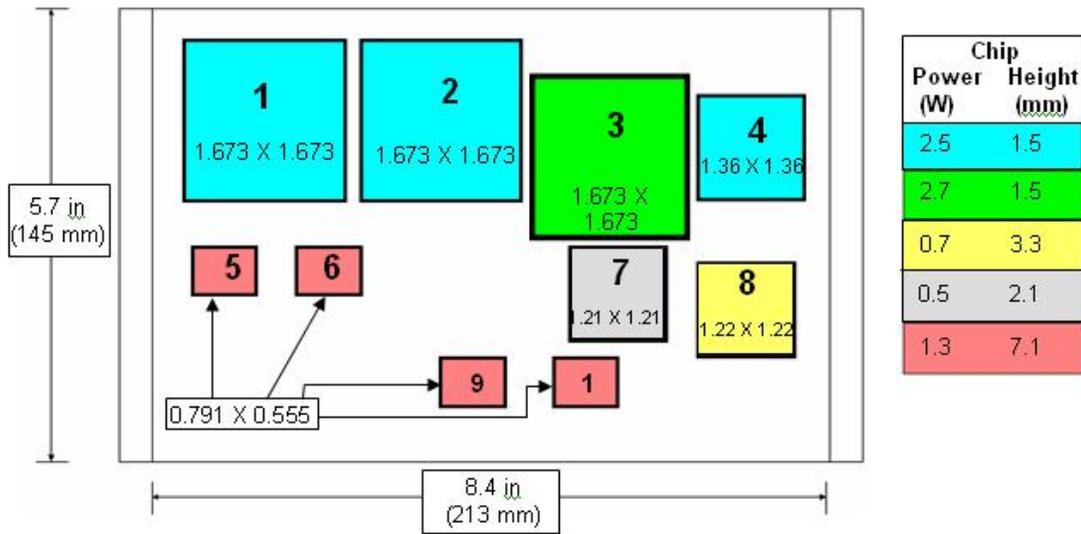


Figure 7: Geometry of previous-generation circuit board

The cold plate requirements provided by the sponsor are listed below in Table 1. 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/in ² (0.46W/cm ²)	64.5W/in ² (10W/cm ²)	258 W/in ² (40 W/cm ²)
Highest single power device	2.7 W	60W	150 W
Fluid Used	N/A	50/50 Propylene Glycol/Water	PAO
Pressure Drop	N/A	20 psi (138 kPa)	2.5 psi (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 40°C	-40°C to 60°C

Table 1: Standard and Stretch Goal Requirements for the Cold Plate

Anticipated Challenges

Table 1 shows that the requirements are grouped into two steps. As the requirements become more stringent the challenges will become tougher. A discussion of the challenges in meeting these requirements follows.

Standard Goal Challenges

The first-generation cold plate was designed to meet the standard goals. The greatest challenge in meeting the list of standard goals was meeting the heat flux dissipation of $10 \frac{W}{cm^2}$.

The maximum module thickness required attention in design but posed no real challenge, and neither did the 20 psi pressure drop. The first-generation cold plate met or exceeded all standard goal requirements.

Stretch Goal Challenges

The biggest challenge in meeting the stretch goals was the heat flux requirement, $40 \frac{W}{cm^2}$ coupled with the temperature difference, $3^\circ C$. The thermal resistance of most interface materials exceeds the total allowable thermal resistance, $0.075 \frac{^\circ C}{W/cm^2}$, of the stretch goal. The second-generation cold plate was designed to meet all the stretch goal requirements, but it did not.

Chapter 3: Experimental Apparatuses and Procedure

Test Section Requirements

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 provided by the sponsor. The circuit board geometry is shown again in Figure 8. From Figure 8 it can be seen that chips 1-4 have the highest heat flux. Attention in this project 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.

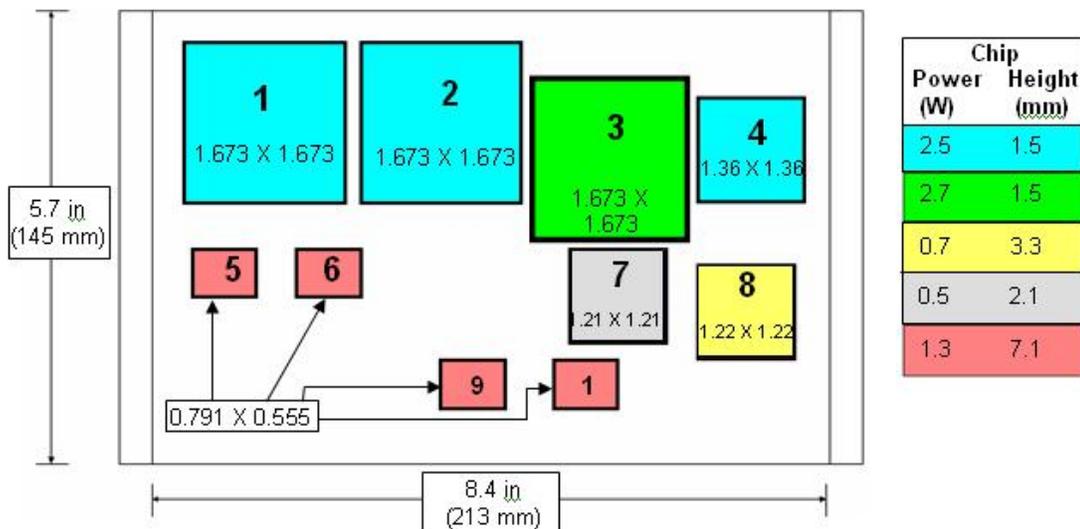


Figure 8: Circuit board geometry

The test section was designed to simulate real world conditions as closely as possible. The test section is made of 4 heat sources of the same size and shape as chips 1-4 attached to a fiberglass board. A fiberglass board was used because it has mechanical and thermal characteristics similar to that of the circuit board provided by the sponsors. Similar thermal characteristics are important because heat spreading by conduction will occur. Some of the heat will be dissipated on the back side of the circuit board. It is important that this spreading and conduction be present in the experiment as well. Mechanical characteristics are important because the cold plate is mounted to the fiberglass board, not the chips themselves. The thermal interface resistance between the chips and the cold plate is heavily dependent on the force pushing the two together. As more pressure is applied the board will flex and bend. This flexing and bending will also affect the thermal interface resistance and it is important that this effect be included in the experiment.

Test Section Description

Dimensions

The test section consists of a rectangular G10 fiber glass composite board 88mm wide, 225mm long, and 2.50mm thick. G10 is a fiberglass which is often used in low temperature applications. Four copper squares were made. Three of the squares are 42.5mm on a side and 2mm thick. The fourth square is 34.5mm on a side and 3.5mm thick. These copper squares are the same size and shape as chips 1-4 in Figure 8. Each copper square has a heater attached to the back of it for heat generation.

Methods

The following section will discuss the methods of fabrication and instrumentation for the test section.

Fabrication

The test section is made up of five main parts all of which were hand made. The four copper squares were cut from oxygen

free copper plates. An end mill was used to make sure all 4 sides were flat, smooth, and parallel. The two large flat surfaces were ground down on a milling machine as well.

After each square was cut, a 0.75mm slitting saw was used to make 0.75mm wide and 0.75mm deep slits in the sides which would be in contact with the cold plate. The constantine wire from a T type thermocouple was then soldered into each slit. The thermocouples were used to measure the temperature of the surface of the copper chip simulators that were in contact with the cold plate. This is analogous to measuring the temperature at the outside of the package of a semiconductor. The slits were cut as small as possible so as not to impede the heat transfer from the chip to the cold plate as well as to make sure that the thermocouples were measuring the temperature as closely as possible to the chip simulator surface.

Attachment

Once the chip simulators were machined the thermocouples were attached. A 0.5mm hole was drilled in each slit where the thermocouple tip was placed. The tip of each thermocouple was

placed into the hole and the wire was held in place with Kapton tape. The chip simulators were then placed on a hot plate and heated to 240°C. A lead based solder with a melting temperature of 220°C was used to attach the thermocouples. An acid based soldering flux was placed in the slit and the melted solder then flowed into the slit filling in the space around the wire and bonding it to the chip simulator. The chip simulators were made out of copper and the junction of a T-type thermocouple is a copper-constantine junction. The copper-constantine junction was created between the constantine wire and the copper chip simulator in the slit.

Omegatherm thin film heaters were then attached to the back of the copper chip simulator. The heaters came with adhesive already applied by Omegatherm. After the heaters were attached they were bonded to the G10 fiberglass composite board using RTV silicone glue. A silicone based glue was used to minimize the heat transfer from the back (the side not in contact with the cold plate) in order to reduce the uncertainty in the energy balance.

Test Section Drawings and Pictures

A schematic drawing of the test section is shown below in Figure 9. The methods of fabrication and attachment have been discussed in previous sections.

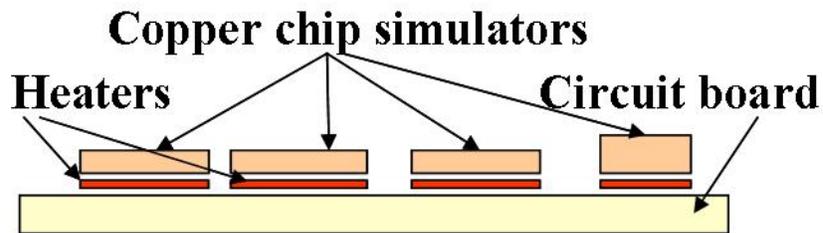


Figure 9: Test section schematic

A picture of the fabricated test section is shown below in Figure 10, 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.

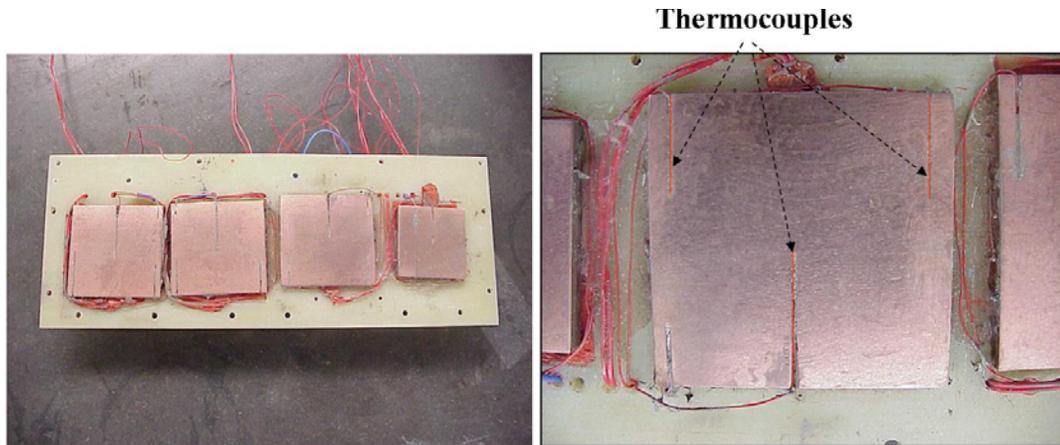


Figure 10: Test section used for data collection

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, was taken from the test section. The same test section was used for each experiment.

Interface Preparation

As discussed in the previous chapter, interface resistance can have a huge effect on cold plate performance. As a result consistent preparation of the two thermal interfaces is essential to obtaining reliable and repeatable data. Before the test section and cold plate were joined each surface was cleaned with isopropyl alcohol and wiped clean with a paper towel.

Arctic Silver 5 thermal grease was then applied to the test section copper chip simulators. A razor blade was used to ensure the layer of thermal grease was smooth and uniform on each chip. The layer of thermal grease was approximately 1mm thick. The test section and cold plate were then placed in contact, and the screws were tightened to hold the two together. The screws in the middle of the cold plate were tightened first, and the screws at the corners were tightened last to help squeeze out any excess thermal grease from the interface.

Chiller Preparation

The chiller was then turned on, and water was circulated through the cold plate at 5°C for fifteen minutes. Once the chiller water reached steady state and flow rate, power was applied to the heaters. Once the setup reached steady state, 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.

Chapter 4: First-Generation Cold Plate

Description

Geometry

Figure 11 shows the external details of the cold plate. The cold plate is a U-type cold plate with pressure ports on the inlet and outlet of the cold plate for measuring the pressure drop.

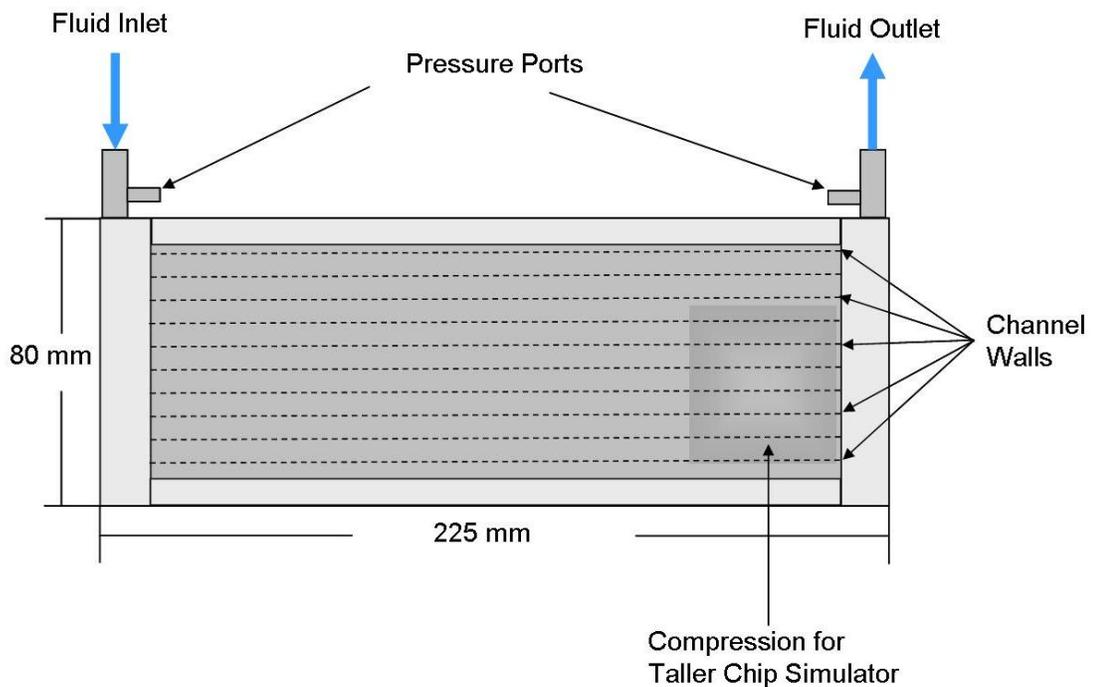


Figure 11: Cold Plate External Details

Channels

The cold plate consists of 12 parallel channels fed by the inlet header. Figure 12 and Figure 13 below show cross-sectional views of the cold plate revealing the channel geometry.



Figure 12: Extruded aluminum section of cold plate

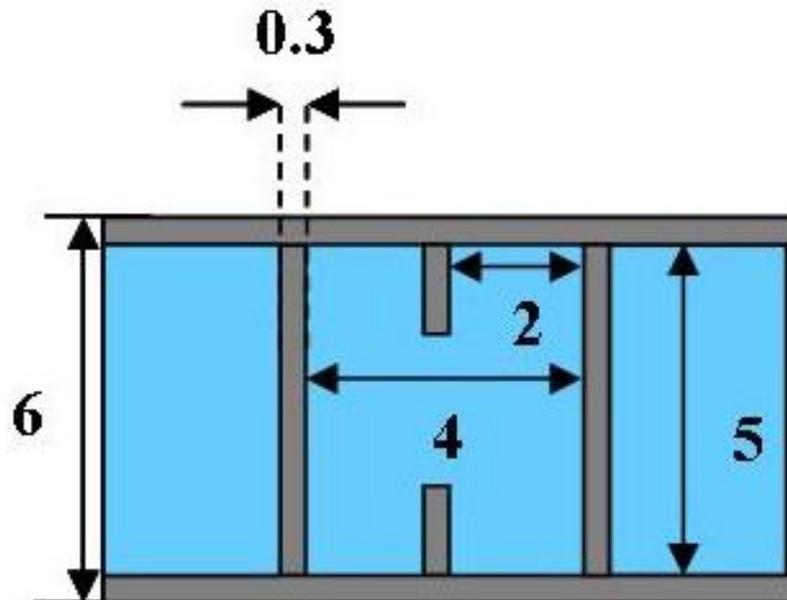


Figure 13: Cross-sectional view of the cold plate showing channel geometry
(All dimensions in millimeters)

A typical cross-section of the cold plate beneath the compression is shown below in Figure 14.

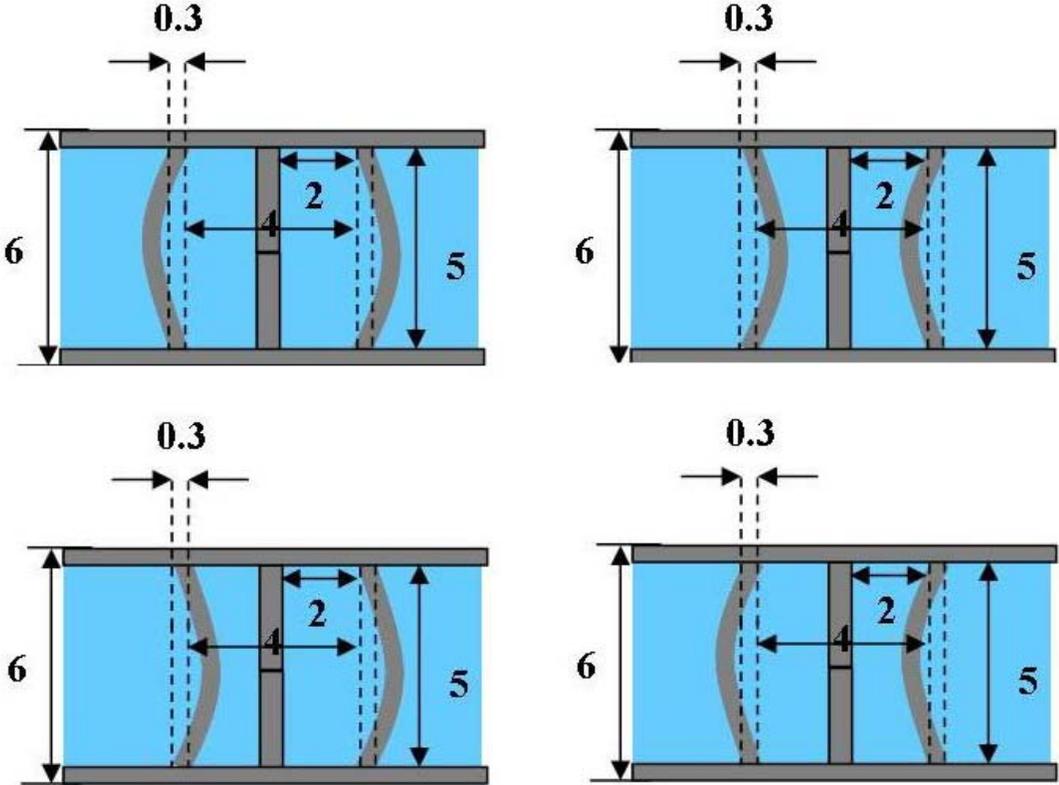


Figure 14: The bend of the channel walls after compression

The channel was compressed to the point where the fins which extended only partway into the channel became joined and the channel walls bent. Figure 14 shows the possible channel cross-sections after bending. The direction of the bending of the walls in the compression was random and uncontrollable with the process used to create the compression.

Materials

The channels of the cold plate were made from extruded aluminum. The headers and frame were machined from solid aluminum blocks. The inlet, outlet tubes, and the pressure port outlet tubes are stainless steel. Aluminum was chosen because of its high thermal conductivity, machinability, and low cost.

Manufacturing

Methods

The headers for the cold plate were cut from solid aluminum bars. First, rectangular header bars 80mm long, 15mm wide, and 10mm thick were cut using a band saw. All six sides were then made flat, parallel, and perpendicular to each other by placing the cut bars in a vise and running an end mill over each side. Each rectangular bar was cut oversize on the band saw by 1mm to leave enough extra material for the end mill operation.

Once all the faces of the header were square and flat, a 7.50mm-diameter hole 7.5cm long was drilled into the rectangular pieces. These holes served as the fluid inlet and outlet paths of the headers. A picture of the header is shown below in Figure 16.

Finally, an end mill was used to cut slots in the face of the header to distribute the fluid and allow it to enter the channels of the extruded aluminum section. Two types of slots were cut into the header. First, a larger slot was cut so that the extruded aluminum section could be inserted into the headers. This slot was designed to support and join the headers to the extruded aluminum section. This slot was shallow and was not deep enough to connect with the hole drilled in the header in the previous step. The internal dimensions of this slot were cut slightly smaller than the external dimensions of the aluminum extrusion that was joined with the header in this slot. The idea was to create a press fit to help seal the cold plate. The third step in the header fabrication process is shown below in Figure 17. Within the large shallow slot, smaller deeper slots were cut which connect with the hole drilled in the previous step. The purpose of these smaller, deeper slots was to create a path for the fluid to flow from the inlet holes of the header,

through the header, and into the aluminum extrusion. The inlet header and outlet header were cut identically to simplify the manufacturing process. Figure 15 through 18 illustrate the steps in the header manufacturing process.

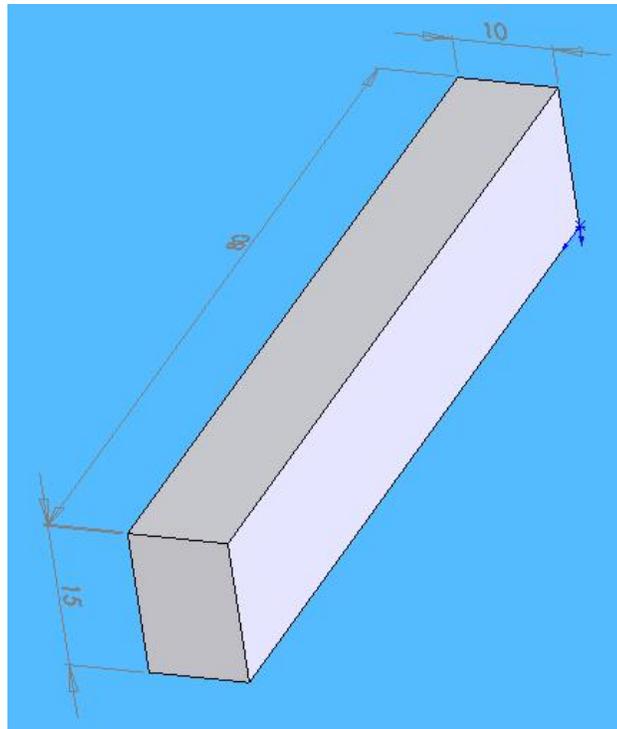


Figure 15: The first step in header fabrication: cutting an aluminum bar to size and squaring off the faces

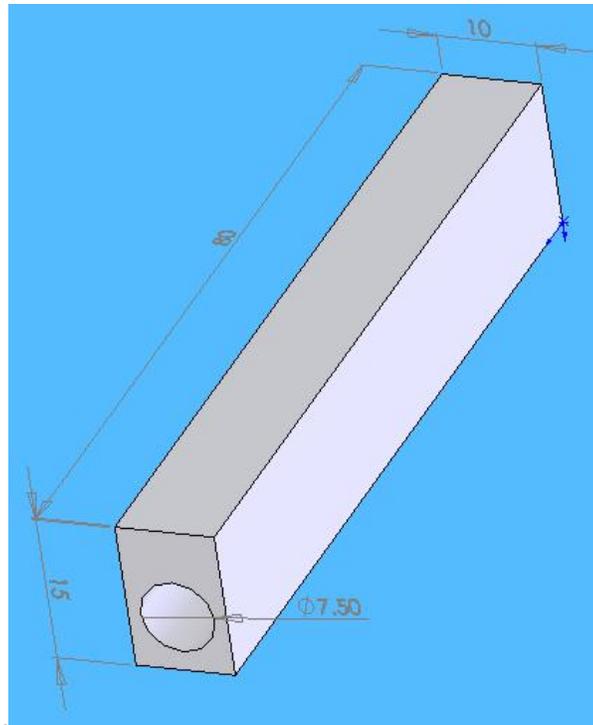


Figure 16: The second step in header fabrication: fluid inlet and outlet holes

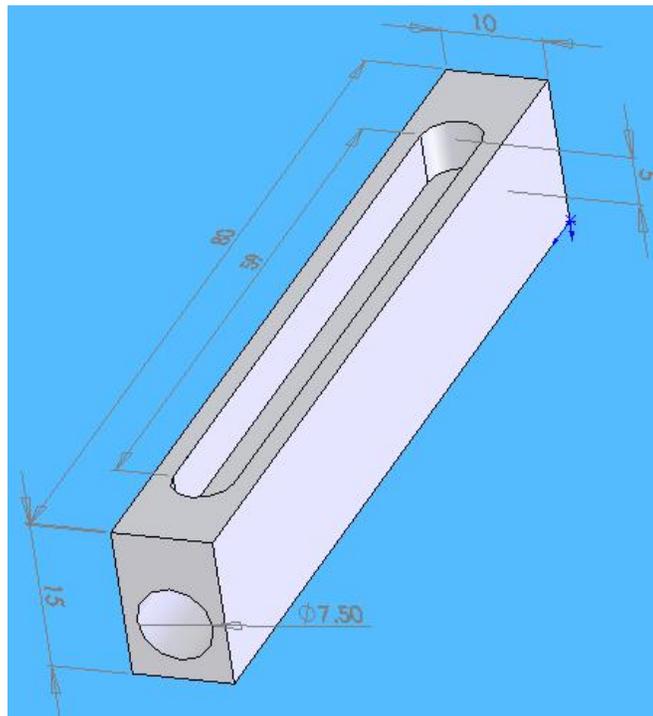


Figure 17: The third step in header fabrication: cutting a slot to insert the extruded aluminum section into

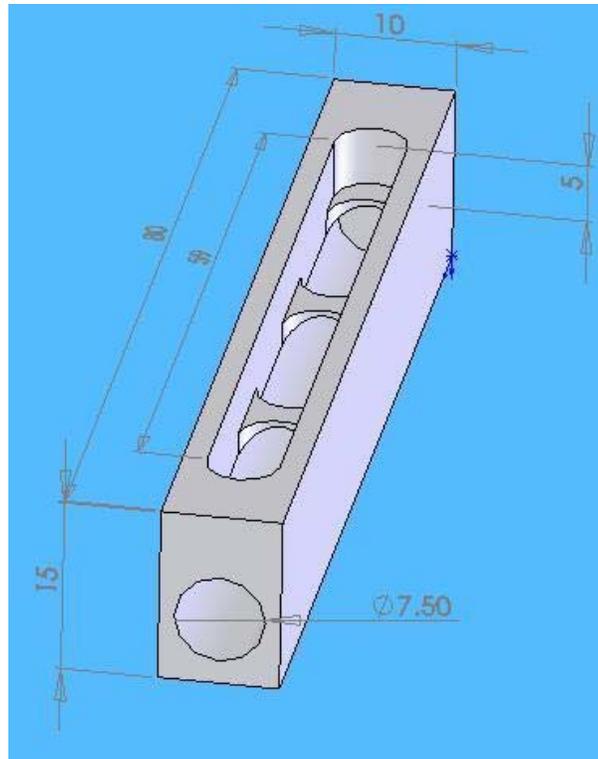


Figure 18: The fourth step in the header fabrication process: cutting slots for the fluid paths

A piece of extruded aluminum with a cross section shown below in Figure 19 was placed in a vise.



Figure 19: Cross section of the extruded aluminum section showing channel geometry

A hydraulic ram was used to create a compression in the cold plate which would accommodate the taller height of one of the chips in the test section. The process of compressing the channel walls and surface of the cold plate must be accounted for in the selection of the heat exchanger. The strength of the cold plate will be reduced by the compression process. If the walls of the cold plate are thick enough, this reduction in strength will not affect the reliability of

the cold plate. The cold plate must be chosen with thick enough walls if this compression method is to be used. The cost will be a slight increase in the conduction resistance through the cold plate but the benefit is that the strength and reliability of the cold plate will be negligibly affected.

After the compression was made, the extruded aluminum section which formed the body of the cold plate was cut on a band saw. The ends of the extrusion were also squared off using an end mill. The final cut was very slow and shallow to remove any damage caused by the band saw to the ends of the extrusion.

Joining

Once the aluminum extrusion and the headers were cut, the aluminum extrusion was inserted into the slots in the header. A two-part epoxy was used to attach the aluminum extrusion to the headers and seal the cold plate. First, the headers and the aluminum extrusion were cleaned using isopropyl alcohol to remove any dirt or grease, which would prevent the epoxy from bonding to the aluminum. Then the two part epoxy was mixed and applied around the seams where the aluminum extrusion and the headers were joined.

Figure 20 and Figure 21 below are pictures of the final cold plate. The cold plate has a compression to accommodate a taller chip and pressure ports on both the inlet and outlet. The second figure is a side view of the cold plate showing the thickness of the cold plate.

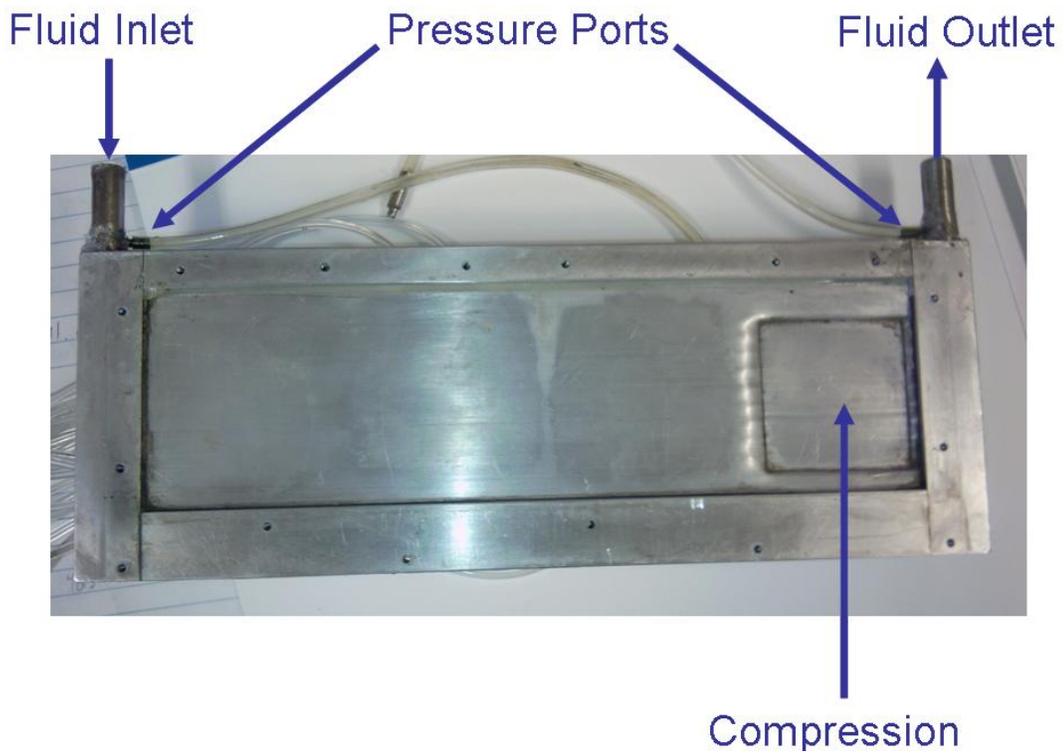


Figure 20: Complete first-generation cold plate

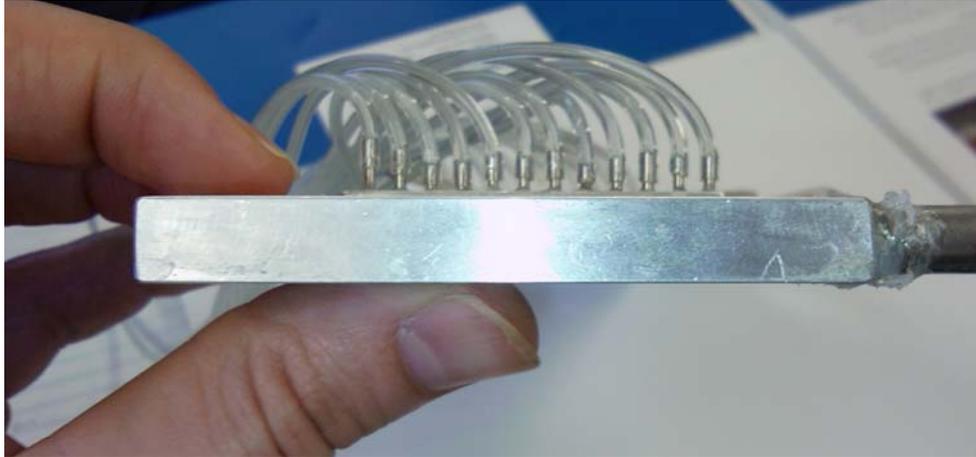


Figure 21: First-generation cold plate side view

Chapter 5: Second-Generation Cold Plate

Modifications based on First Generation Cold Plate Results

The first-generation cold plate was constructed and tested.

A great non-uniformity was found in the thermal resistances of the chips. Figure 22 below shows the chips and the flow path of the water for the first-generation cold plate.

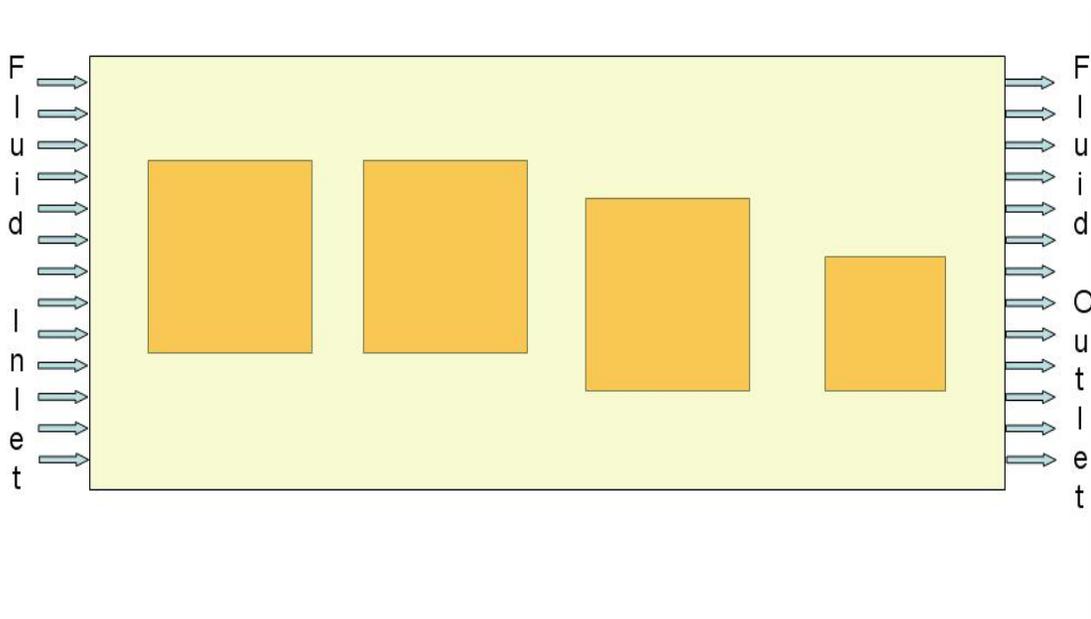


Figure 22: Fluid flow path

The chip closest to the inlet header had the lowest thermal resistance, and thermal resistance increased as the distance from the inlet header increased. It was thought that entrance effects

were affecting the convective heat transfer. 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 23 below is an illustration of the second-generation cold plate.

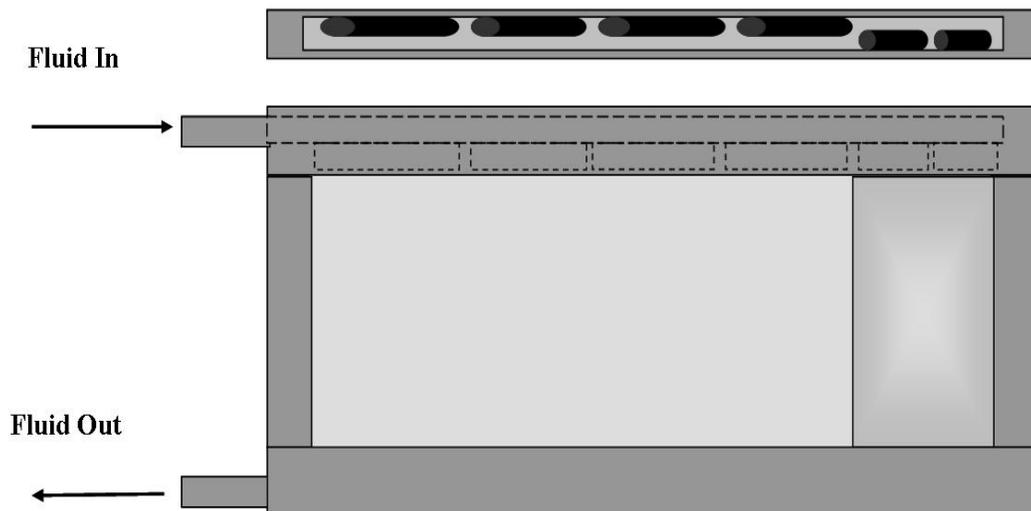


Figure 23: Second-generation cold plate

The Channels

The channels used in the second-generation had a smaller hydraulic diameter than the channels of the first-generation cold plate. Smaller channels were chosen for several reasons. The first reason is that the Nusselt number based on the channel hydraulic diameter,

$$\textbf{Equation 3: } Nu_d = \frac{hd_h}{k}$$

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 equation for convection thermal resistance is:

$$\textbf{Equation 4: } R''_{th,c} = \frac{1}{h}$$

It can be seen from Equation 4 that larger convection coefficients lead to lower convection thermal resistances. Figure 24 is a picture showing the dimensions of the channels used in the second-generation cold plate.

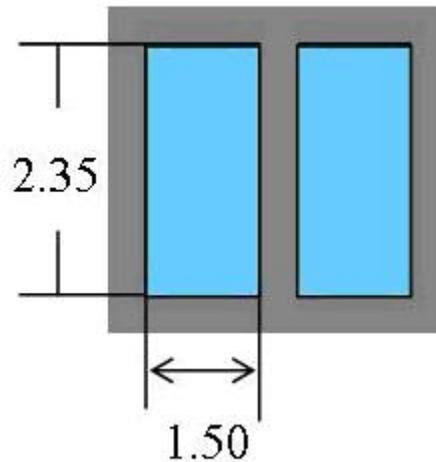


Figure 24: Second-generation cold plate channel dimensions in millimeters

The second reason the channels were smaller in the second-generation cold plate was to meet the stretch 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.

Materials

Like the first-generation cold plate, the second-generation cold plate uses aluminum headers and aluminum extruded channels as well. Again aluminum was chosen because of its low cost, good machinability, and high thermal conductivity.

Manufacturing

Headers

The goal of the second-generation cold plate was to create a cold plate with long headers and short channels. Creating long headers presented several difficulties in the manufacturing process. Two of the biggest difficulties in manufacturing came from the fact

that holes with extremely high aspect ratios had to be cut in the header.

For a hole of length L and diameter D, the aspect ratio of the hole is defined as

$$\text{Equation 5: } \frac{L}{D}$$

Holes with high aspect ratios are hard to drill or cut because the cutting occurs at the very end of the tool. This means the cutting force is applied to the end of the tool and the magnitude of the force is proportional to the diameter, D, of the hole. A high aspect ratio hole dictates that a large force be applied to the end of a long, narrow lever. The result of this is that cutting must be done slowly and carefully, or cutting tools will be broken. Both the fluid inlet and outlet paths of the headers as well as the slots for connecting the fluid inlet and outlet paths to the channels had high aspect ratios (42 and 10).

Figure 25 below shows the rectangular bars that were used to make the second-generation headers. The bars were cut from a sheet of aluminum, and all the sides were first made flat and parallel.

After the rectangular bars were prepared, the fluid inlet and outlet paths were cut into the headers. The fluid paths are circular holes 210mm long and 5.0 mm in diameter. The mill did not have the required vertical travel length to accommodate both the header and the drill bit. A right-angle transfer case had to be attached so the holes could be cut using the longer travel of the X axis of the mill. Figure 26 shows the headers after the holes had been cut.

After the holes were cut, the slot for connecting the channels to the header and holding the channels in place was cut using a 2.50mm diameter end mill. Figure 27 shows the headers with the connecting slot cut.

The final step in the header manufacturing was to cut the slots for distributing the fluid into the channels. These slots were cut inside the connecting slot and made using the same 2.50mm diameter end mill. Figure 28 shows the headers after these slots were cut.

Figure 25 through 28 illustrate the manufacturing process for the second-generation cold plate headers.

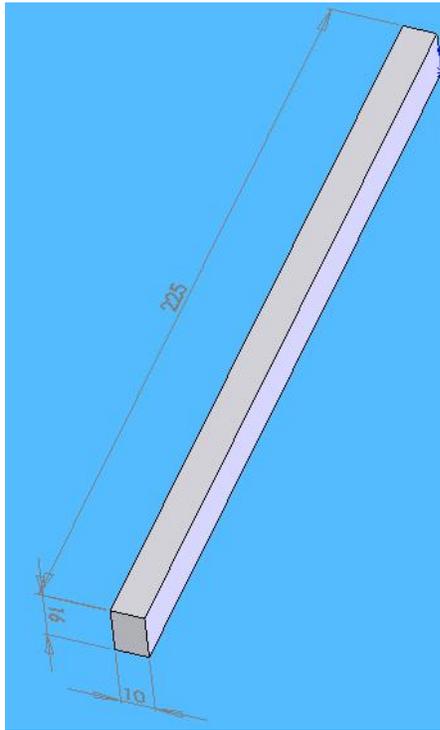


Figure 25: Second-generation header

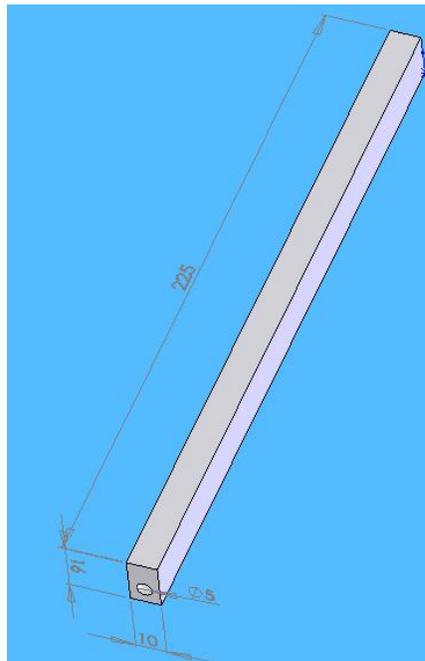


Figure 26: Second-generation header with fluid inlet and outlet holes

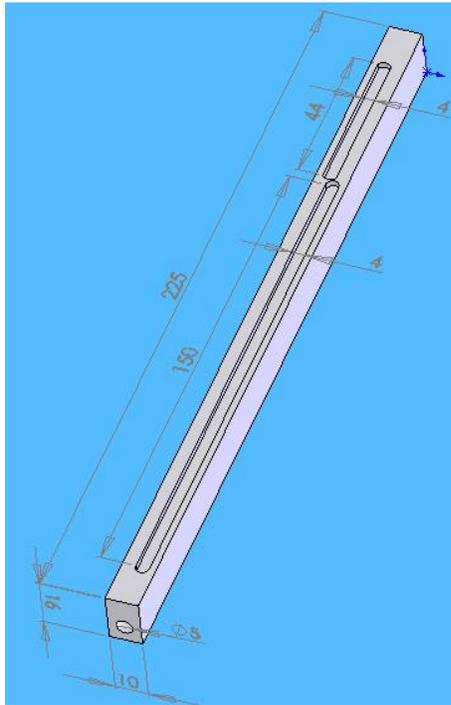


Figure 27: Second-generation header with connecting slot.



Figure 28: Second-generation header

Channel Joining

The extruded channels used in the second-generation cold plate were only available in 27.5mm wide extrusions. As a result, nine sections had to be joined together to create the cold plate.

Figure 29 below is a picture of one of these strips.

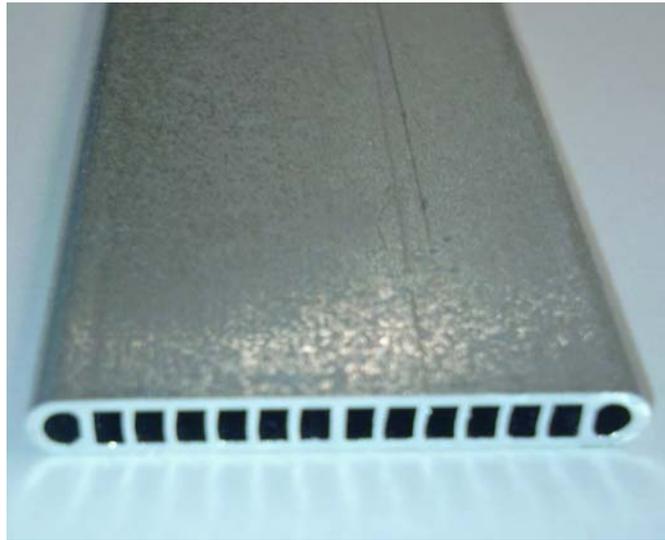


Figure 29: Second generation cold plate extruded aluminum channels

The rounded outside channels were removed and the edges were squared off. Sections were then glued together on the flat, squared-off edges using the same two-part epoxy that was used in the first-generation cold plate. Figure 30 below shows two sections of channels glued together.



Figure 30: Two sections of extruded aluminum channels glued together for the second-generation cold plate

Once all the sections were glued together, they were inserted into the cold plate header connecting slot and glued into place.

Due to the large number of seams that were glued together, sealing the cold plate was a big problem. Many leaks were found that had to be sealed. Leak checking and fixing was the most time consuming part of this process. In the future, finding one single extrusion would be a much better solution than trying to join many separate extrusions together.

After the cold plate was assembled, the surface was sanded thoroughly using progressively finer sand paper. The surface in contact with the copper chip simulators had to be flat and smooth to minimize the thermal interface resistance and to ensure an even

temperature profile. Figure 31 below show the complete second-generation cold plate.



Figure 31: Second-generation cold plate

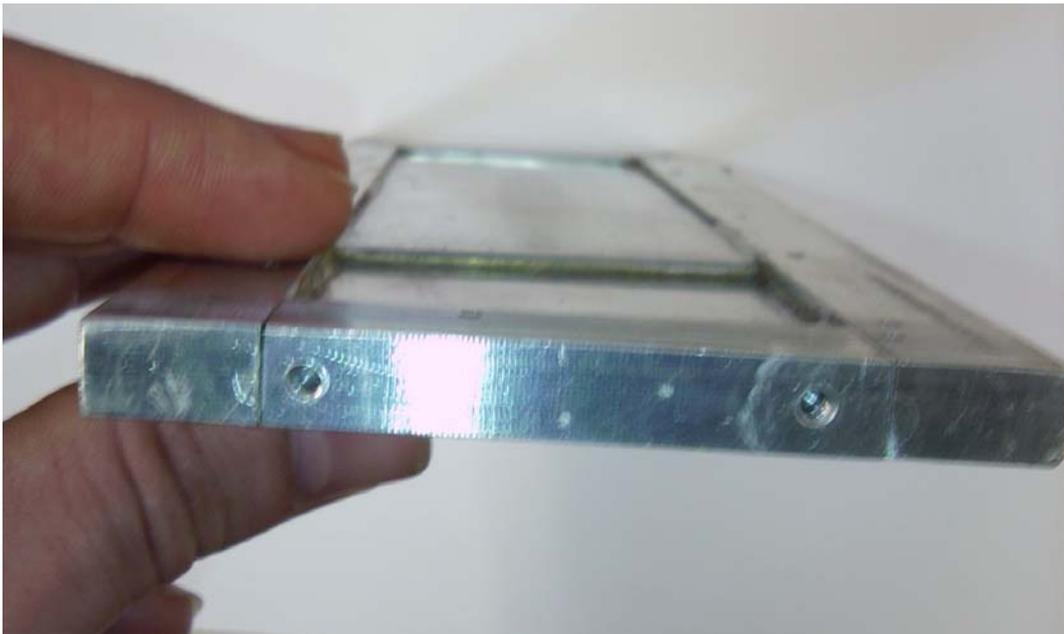


Figure 32: Second-generation cold plate side view



Figure 33: Second-generation cold plate channel offset to accommodate taller chip simulator

Chapter 6: Thermal Interface Resistance

Introduction to Thermal Interface Resistance

Thermal interface resistance is the thermal resistance between the interfaces of two separate solid materials. It is caused by surface defects in the material, and all solids have surface defects. Due to the crystalline nature of most solid materials, no interface is ever perfectly smooth and flat. At a microscopic level, the interface between two solid materials looks like the interface shown below in Figure 34.

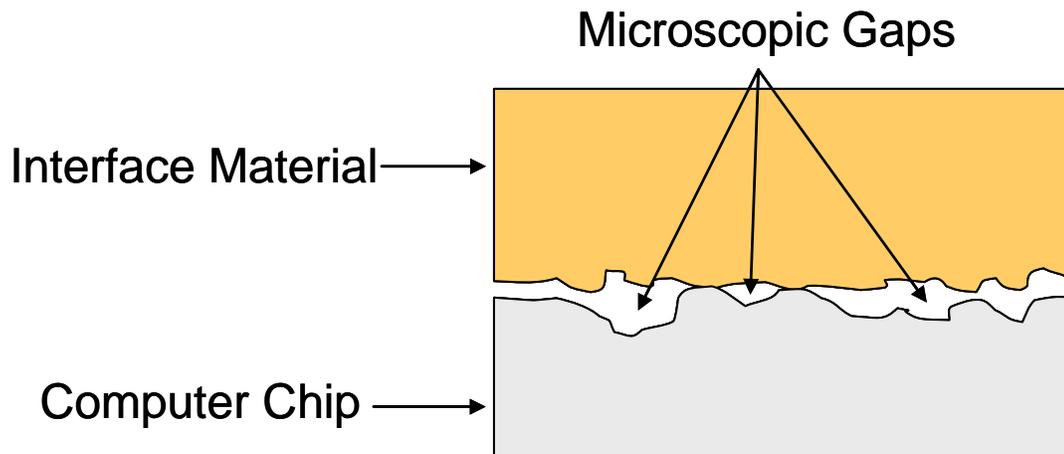


Figure 34: Interface between two solid materials

As shown in Figure 34 above, the interface of a solid material, even one that appears to be smooth and flat, is full of peaks and valleys. It is these peaks and valleys that create two

different coefficients of friction: kinetic and static. When two solid materials are placed in contact with each other some of the peaks and valleys contact each other. Heat is transferred across the interface from one material to another by conduction, only through the points which are in contact with each other. The equation for heat transfer is given below in Equation 6:

$$\text{Equation 6: } \dot{Q} = -kA_c \frac{dT}{dx}$$

where $k \left[\frac{W}{mK} \right]$ is the thermal conductivity of the material through

which heat is being conducted, and $A_c \left[m^2 \right]$ is the area

perpendicular to the temperature gradient $\frac{dT}{dx} \left[\frac{W}{m} \right]$. The thermal

resistance is obtained from rearranging Equation 6 into Equation 7:

$$\text{Equation 7: } \dot{Q} = \frac{dT/dx}{\frac{1}{kA_c}} = \frac{dT/dx}{R_{th}}$$

From Equation 7 it can be seen that a decrease in the cross-sectional area, A_c , causes an increase in the thermal resistance.

This is precisely what happens at an interface between two materials. Due to the rough nature of the surfaces, only a fraction

of the actual surface area of the materials is in contact, meaning A_c decreases, R_{th} increases, and the total heat transfer, ϕ [W] decreases for a given temperature gradient, $\frac{dT}{dx}$.

Methods of Minimizing Thermal Interface Resistance

There are numerous methods for minimizing thermal interface resistance, but they generally fall into two categories.

Mechanical Deformation

The first category for minimizing thermal interface resistance involves mechanically deforming the interface itself. This is done by applying a high pressure to the interface. This high pressure causes the two interfaces to smooth out by forcing them to deform each other. The result is a much higher contact area, A_c . A higher contact area leads to a lower thermal interface resistance and a higher rate of heat transfer for a given temperature gradient.

Deformable Materials

The second method for reducing thermal interface resistance involves adding a deformable material to the interface. In a rough

interface like the one shown in Figure 34 above, heat transfer occurs only where the two materials are in contact with each other.

The rest of the space is filled with air. These spaces are too small for natural convection cells to occur, so the heat transfer across these spaces is conduction through air. The thermal conductivity of air is $0.03 \frac{W}{mK}$, or 0.0075% of the thermal conductivity of copper ($400 \frac{W}{mK}$). This means that for a given cross-section and temperature gradient, less than 1% of the heat will be transferred through the air via conduction.

A deformable material is added to the interface to displace the air. With a deformable material added to the interface, heat transfer across the gaps in the interface occurs by conduction through the material. Any material that has a higher thermal conductivity than air will increase the heat transfer across the interface. There are several different types of deformable materials used to achieve this purpose.

The first type of deformable material is a polymer pad. These pads are made of solid elastic polymers. They generally have adhesives on their surfaces and deform easily. They are good

for situations where the distance between the two interfaces is large and the two interfaces cannot be pressed against each other. The polymer pad serves to bridge the gap between the two interfaces. Polymer pads usually have the lowest thermal conductivity of any interface material ($2 \frac{W}{mK}$ to $5 \frac{W}{mK}$) [7].

The second type of deformable material is thermal grease. Thermal grease transitions between a solid and a liquid at room temperature. It behaves as a very viscous liquid. At the elevated temperatures caused by microprocessor heat dissipation the grease becomes a liquid. Silicone-based thermal greases have thermal conductivities around $2-3 \frac{W}{mK}$ [8]. The thermal conductivity of thermal grease can be increased by adding metal particles to the grease. Thermal greases with metal particles can have thermal conductivities as high as $8 \frac{W}{mK}$. When metal particles are added to thermal grease it is important that the metal particles be extremely small. If the particles are on the same order of magnitude or larger than the surface roughness features, they will cause the interfaces to be further apart and increase the thermal interface resistance. When particles are smaller than the surface

roughness features they are able to fill the gaps created by the surface roughness. In this situation the interfaces will not be pushed apart, and conduction will be enhanced across the gaps in the interface. As a result, thermal interface resistance will decrease. These situations are illustrated below in Figure 35 and Figure 36.

Particles increase
interface separation

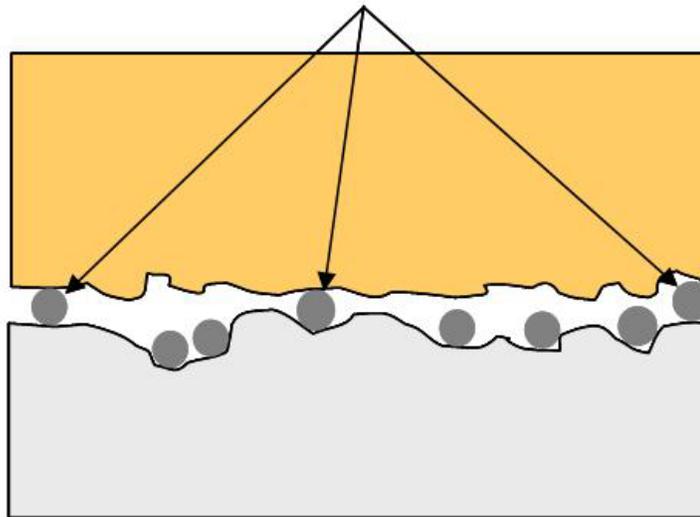


Figure 35: Thermal grease with particles larger than the surface roughness features increases thermal interface resistance

Small particles fill gaps
and decrease interface
resistance

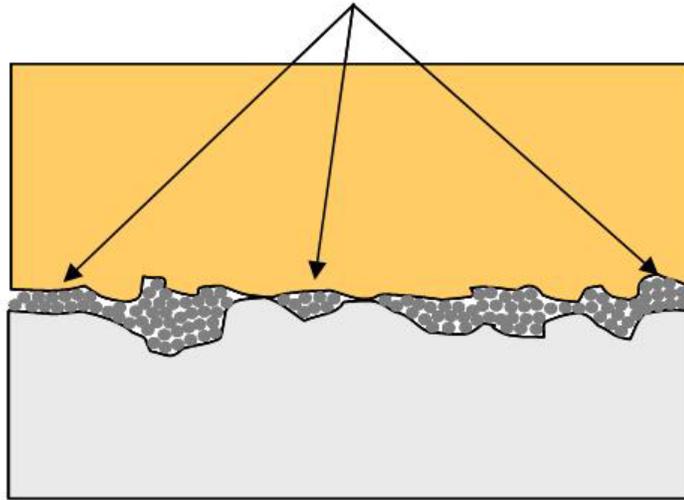


Figure 36: Thermal grease with particles smaller than the surface roughness features decreases thermal interface resistance

Thermal greases are most effective when used in situations where the interfacial separation distance is small and two interfaces are in contact with one another. The grease works best when it is applied to the two interfaces and then the two interfaces are squeezed together. The excess grease will be pushed out by squeezing the two interfaces together, leaving grease only in the places where two interfaces would not have made contact. The biggest disadvantage of this method is that the thermal cycling creates thermal stress between the interfaces and these stresses will push

the thermal grease out of the interface, causing the thermal interface resistance to increase.

The third type of deformable material is applied as a liquid at elevated temperatures but remains a solid at room and operating temperatures, like a solder. Soldering or brazing two interfaces together is the most effective way of reducing thermal interface resistance. The metal in the solder or braze compound acts similarly to the thermal grease described previously; however, it has a higher thermal conductivity $25 \frac{W}{mK}$ [9], resulting in a lower thermal interface resistance.

Thermal Interface Resistance Experiments

Experiments were performed to determine the thermal interface resistance of two different interface materials: a thermal grease and a thermal pad. These experiments were done because the stretch goal for the total thermal resistance of the cold plate is $0.075 \text{ } ^\circ\text{C}\text{-cm}^2/\text{W}$. Any thermal interface material used with the cold plate must result in a thermal interface resistance lower than $0.075 \text{ } ^\circ\text{C}\text{-cm}^2/\text{W}$ or it cannot be used to reach the stretch goal for thermal resistance. Two materials were tested to determine if they

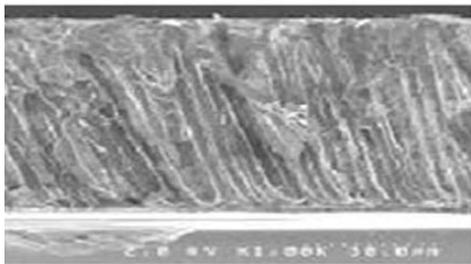
created a low enough thermal resistance to meet the stretch goal for total thermal resistance for this experiment. The thermal grease used in this experiment was Arctic Silver 5 [10]. Arctic Silver 5 is a thermal grease containing small silver particles in it to increase the overall thermal conductivity of the grease. It has thermal conductivity of $8.89 \frac{W}{mK}$. A picture of the thermal grease is shown below in Figure 37.



Figure 37: Arctic Silver 5 [10]

The thermal pad is a carbon fiber thermal pad. In this pad carbon fibers are suspended in a polymer matrix. The matrix is used to line up the fibers so they are perpendicular to the two interfaces. When the

fibers are perpendicular to the two interfaces, heat is transferred across the interface through the length of the fiber. The thermal conductivity of single carbon fiber is $1000\left[\frac{W}{mK}\right]$ [11]. The carbon fiber pads must be applied to the surface and heated under pressure to work effectively. A picture of the carbon fiber thermal pads is shown below in Figure 38.



Carbon fibers oriented vertically



Carbon fibers running into and out of the page

Figure 38: Carbon Fiber Thermal Interface Material [8]

Experimental Setup

The experimental setup is shown below in Figure 39. It consists of a heat source, a heat sink, and two brass rods with thermistors for measuring the temperatures on each side of the thermal interface. One of the thermistors was not working properly so data from it was not used. Since the temperature profile in the brass rod is linear, the trend was very clear from the remaining thermistors, and the 6th thermistor was not needed. From this device a linear temperature gradient can be found, and the deviation caused by the thermal interface can be used to calculate the thermal interface resistance.

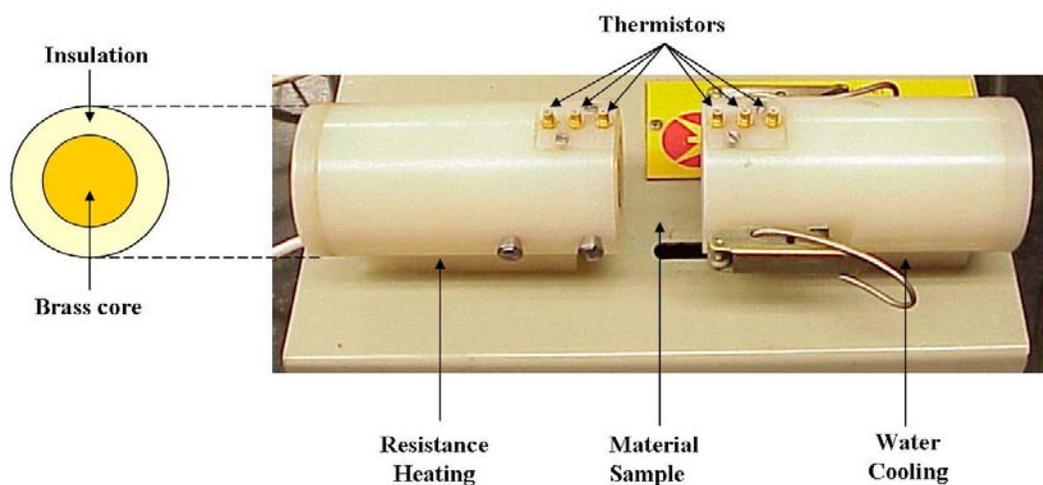


Figure 39: Thermal interface resistance experimental setup

Experimental Procedure

The experimental procedure for the thermal grease and the carbon fiber pad varies slightly because each material requires a different preparation method before it is applied to the brass rod interface.

In order to calculate the thermal interface resistance, the interfacial separation distance must be known. To create a constant and predictable interfacial separation distance with the thermal grease, small pieces of 0.2mm diameter copper wire were cut and laid flat between the two rods. The number of pieces and their size was kept very small so that the effect of conduction through the wire pieces could be neglected. Thermal grease was applied to the interface, and the wires were then carefully placed and evenly distributed to create the gap. The two brass rods were squeezed together using the clamps supplied by the device. Power was supplied to the heat source with a constant input of 9.5W. Cold water was supplied to the heat sink from a chiller at a temperature of 10°C.

To test the carbon fiber thermal pads, the two brass surfaces were first cleaned with alcohol. A 0.1778mm thick carbon fiber

pad was placed between the two brass rods. The brass rods were stacked on top of each other, but the clamp was not attached.

Weights were stacked on top of the brass rod to apply pressure to the carbon fiber thermal pads. A pressure of 20psi was applied to the carbon fiber thermal pads while the interface was maintained at a temperature of 60°C for 20 minutes, as described in the instructions that came with the thermal pads.

Results

Arctic Silver 5

The results for the Arctic Silver 5 thermal grease experiments are shown below in Figure 40. Figure 40 shows that the interface resistance decreases as the contact pressure increases, but the decrease in thermal interface resistance lessens as the contact pressure is increased. It means that increasing the contact pressure between two thermal interfaces is a viable means of decreasing the thermal interface resistance, but only to a certain point. Eventually the reliability of other components will begin to decrease as a result of the increased pressure, and any gains made

by lower thermal interface resistances will be negated. The stretch

goal for thermal resistance in this project was $0.075 \frac{^{\circ}\text{C}}{\text{W}/\text{cm}^2}$.

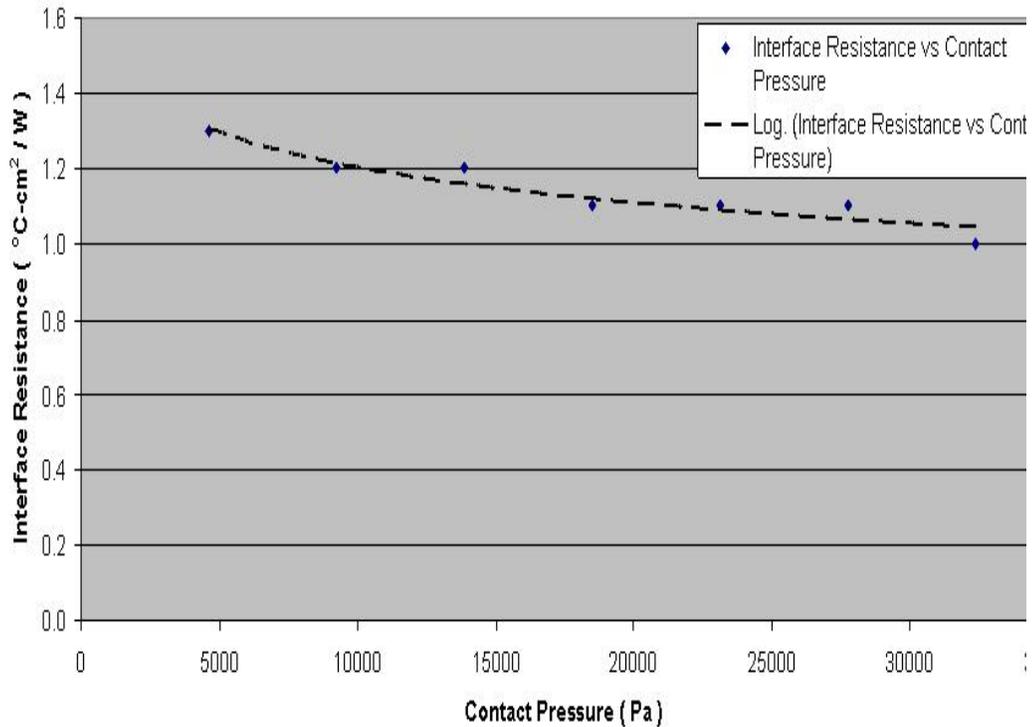


Figure 40: Thermal interface resistance results for Arctic Silver 5

The use of Arctic Silver 5 thermal grease results in a thermal interface resistance more than ten times higher than the stretch goal for total thermal resistance. Therefore, Arctic Silver 5 is not an acceptable thermal interface solution for meeting the stretch goal for thermal interface resistance in this project and other options were explored.

Carbon Fiber pads

Carbon fiber thermal pads were tested with the same experimental setup. A carbon fiber thermal pad 0.1778mm thick was used in the experiment. The carbon fiber thermal pads had considerably better performance than the thermal grease. No detectable temperature difference could be found across the thermal interface for all contact pressures. The maximum power was applied to the heater, and water was supplied to the heat sink at 5°C. When significant figures are ignored, the carbon fiber thermal pads have an estimated thermal resistance of

$0.0098 \frac{^{\circ}\text{C}}{\text{W}/\text{cm}^2}$, or less than a tenth of the stretch goal requirements.

Better data for this experiment is not available due to equipment limitations.

Thermal Interface Conclusions

Numerical simulations were also performed by a colleague which did not take the thermal resistance into account [12]. When experiments with the cold plate were run with the Arctic Silver thermal grease, the difference between the simulated thermal resistance and the actual thermal resistance in the cold plate

experiment matched the values found in the thermal interface resistance experiments.

Finding the correct thermal interface material is essential to meeting total package thermal resistance goals. Interface thermal resistance can account for a significant portion of the total thermal interface resistance with current micro channel cold plates.

Chapter 7: Results and Discussions

This chapter will discuss the experimental procedure and results for the two cold plates.

First-Generation Cold Plate Results

First-Generation Heat Transfer

Below is a graph showing the first-generation cold plate thermal resistance as a function of the flow rate through the cold plate. The thermal resistance, R_{th} is defined below in Equation 8.

$$\text{Equation 8: } R_{th} = \frac{T_{chip} - T_{water,in}}{\dot{Q}}$$

where T_{chip} is the temperature of the copper chip simulator measured by the thermocouple, $T_{\text{water, in}}$ is the temperature of the water coming from the chiller, and q is the heat flux.

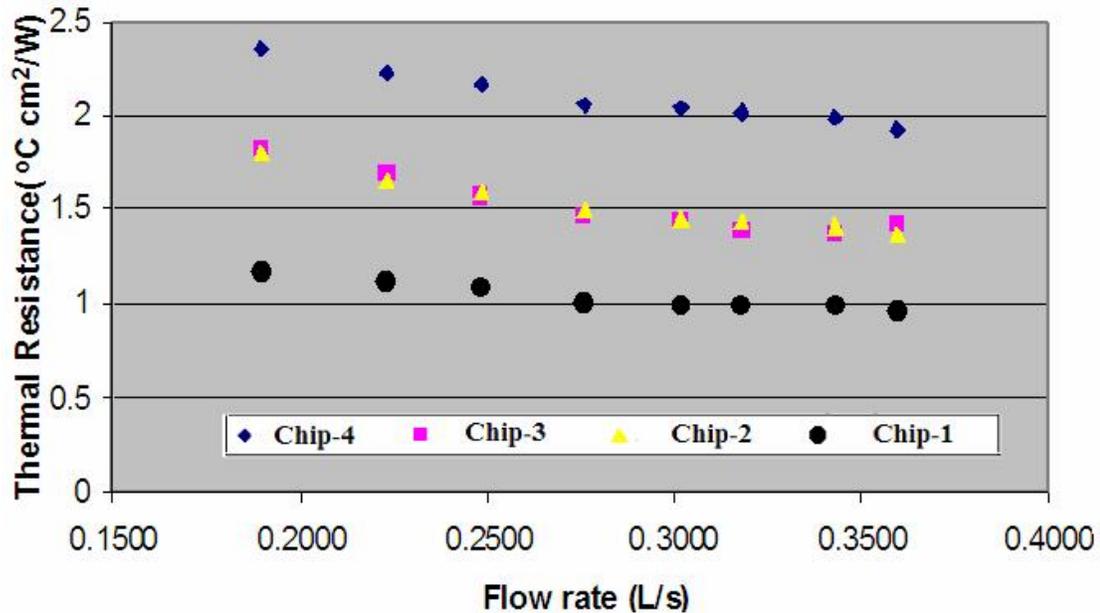


Figure 41: First-generation cold plate thermal resistance

The graph above 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. Figure 42 below shows the test section with chips 1-4 marked so that meaning can be given to the data presented above.

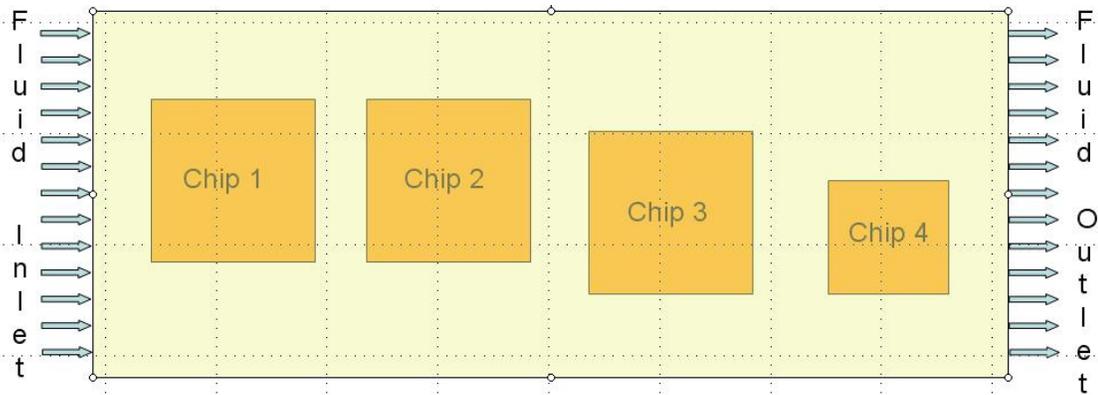


Figure 42: Test section schematic

It was expected that chip 4 would have the lowest thermal resistance. Chip 4 is the taller chip. A compression was made in the cold plate to accommodate the taller chip height. This compression should have made the channels under the compression smaller. The water flowing through the smaller channels under Chip 4 should have been flowing faster, and the thermal resistance was expected to be lower as a result.

Given the trend of increasing thermal resistance with increasing distance from the cold plate entrance, it was assumed that the entrance effect was decreasing the thermal resistance of the chips near the cold plate entrance.

First-Generation Pressure Drop

Figure 43 below shows the pressure drop across the cold plate as a function of the flow rate through the cold plate. 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.

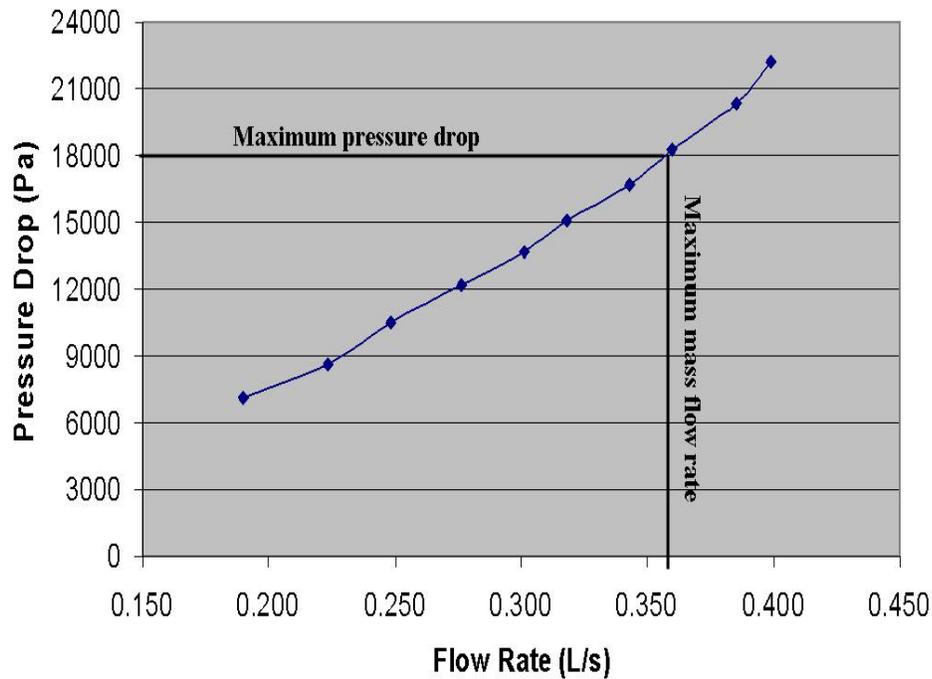


Figure 43: First-generation cold plate pressure drop

Second-Generation Cold Plate

The design features for the second-generation cold plate are shown below. The second-generation cold plate used long headers and short channels in an attempt to reduce the pressure drop.

Figure 44 below is an illustration showing the second-generation cold plate design details.

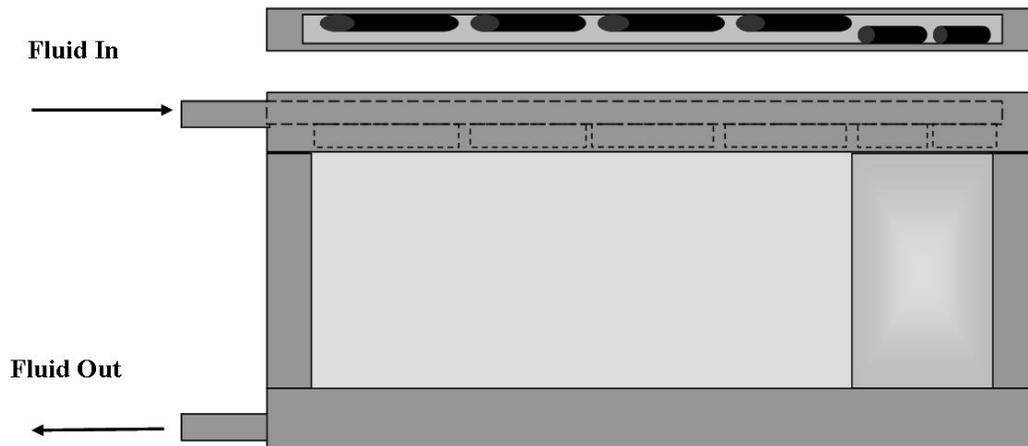


Figure 44: Second-generation cold plate

Second-Generation Heat Transfer Results

The thermal resistance for all four copper chips simulators is shown below in Figure 45.

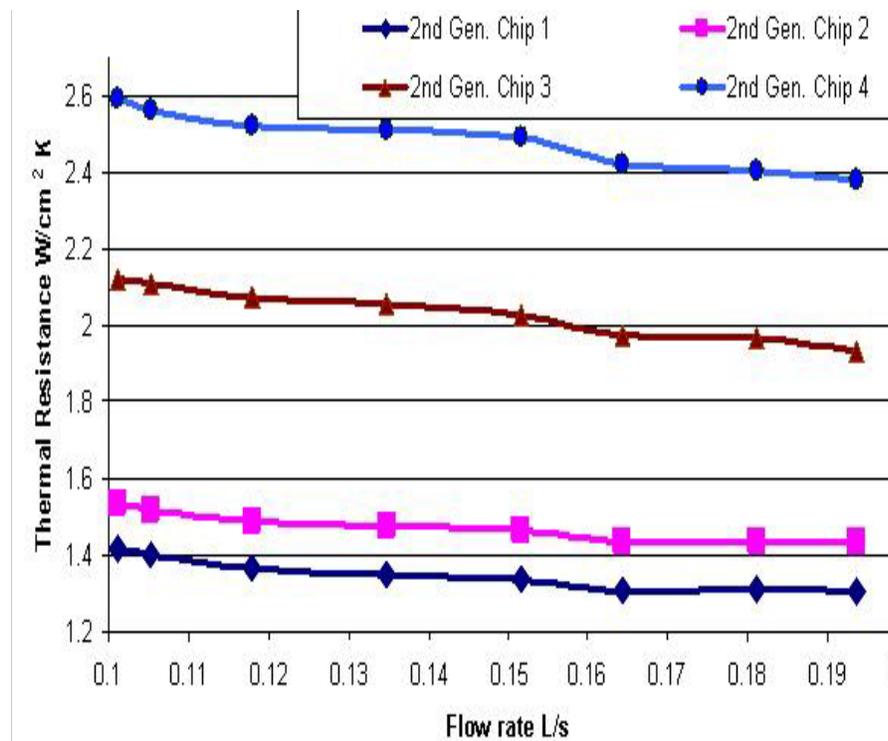


Figure 45: Second-generation cold plate thermal resistance

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. An explanation of these results will be discussed in the next chapter of this thesis.

Second-Generation Pressure Drop

The pressure drop across the cold plate was measured. It is shown below in Figure 46. The pressure drop from the first-generation cold plate is shown for comparison.

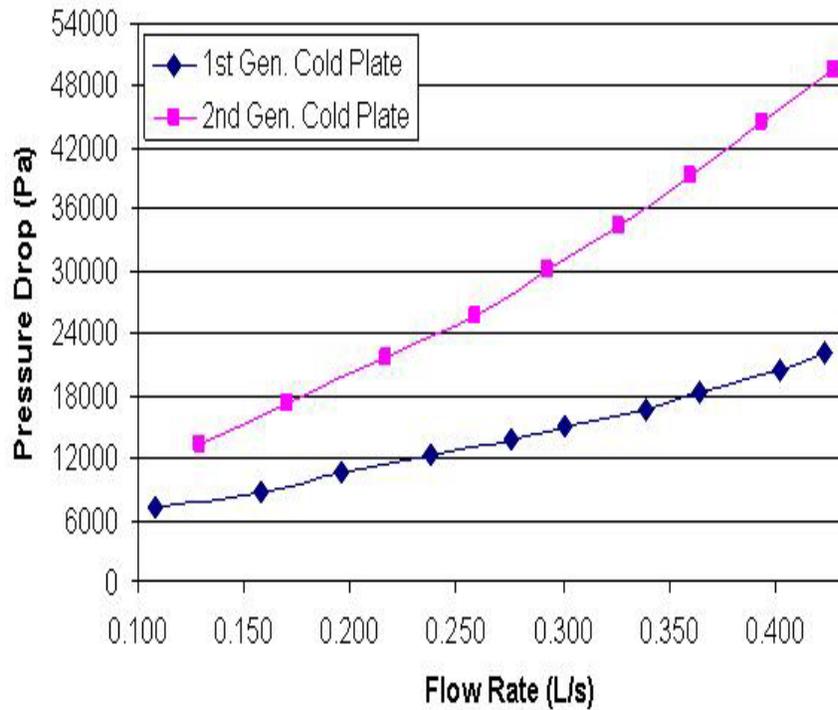


Figure 46: Second-generation cold plate pressure drop

Figure 46 shows that the pressure drop for the second-generation cold plate is greater than the pressure drop for the first-generation cold plate as a result of the second generation cold plate having smaller channels than the first generation cold plate.

Comparison of the Results

Differences in Heat Transfer

There are two important differences in heat transfer between the first- and second-generation cold plates. The first difference is

that the two 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 two cold plates, the range of the second-generation cold plate had higher average temperatures than the first generation cold plate. Overall, the second-generation cold plate was less effective at cooling the entire test section. The second generation cold plate caused higher chip surface temperatures due to a combination of flow mal-distribution, which reduced the convection heat transfer effectiveness, and increased thermal interface resistance as a result of the differences in manufacturing.

Figure 47, Figure 48, Figure 49, and Figure 50 are comparisons of the thermal resistances for each chip. 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 only deviation from this trend appears in the results from Chip 2. The results for Chip 2 show that the

first generation cold plate has higher thermal resistance initially, but it drops as the flow rate through the cold plate is increased. The thermal resistance data presented from each chip is 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. Of these resistances, the only one that changes with flow rate is 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 [12], 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 affect 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.

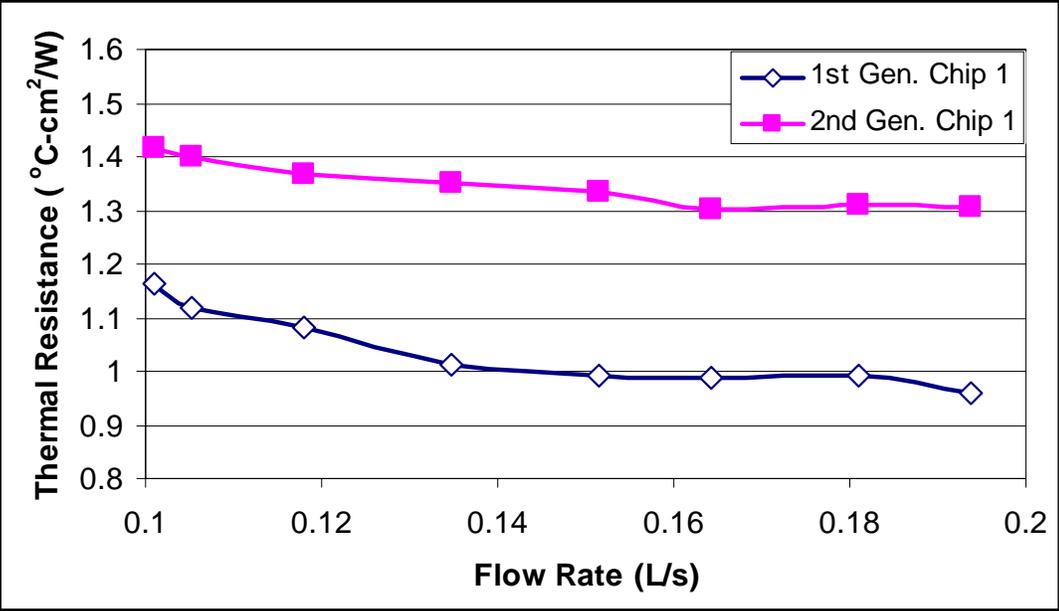


Figure 47: Thermal resistance comparison for first and second generation cold plates (Chip 1)

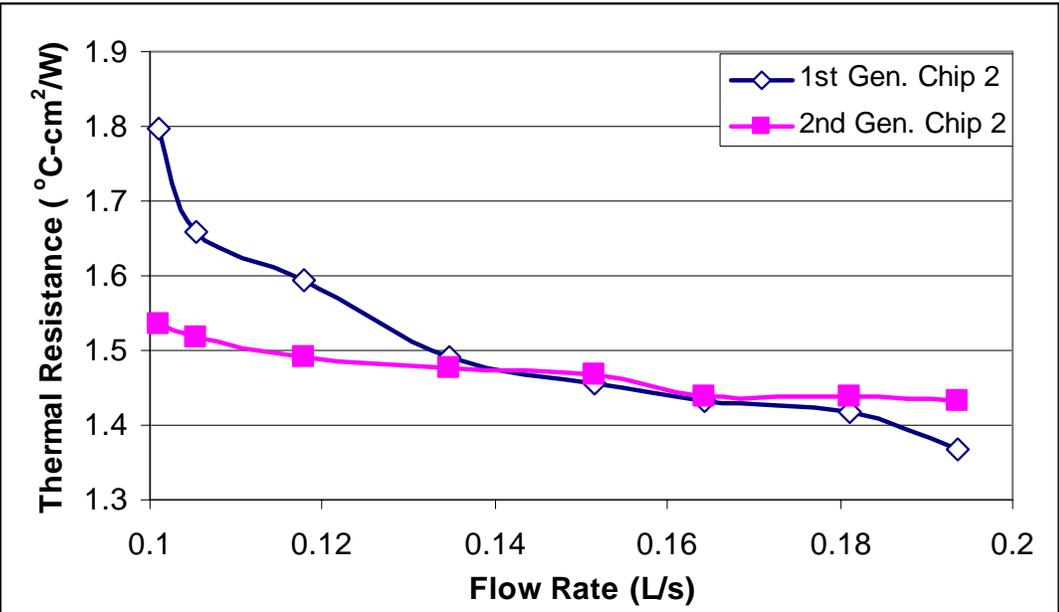


Figure 48: Thermal resistance comparison for first and second generation cold plates (Chip 2)

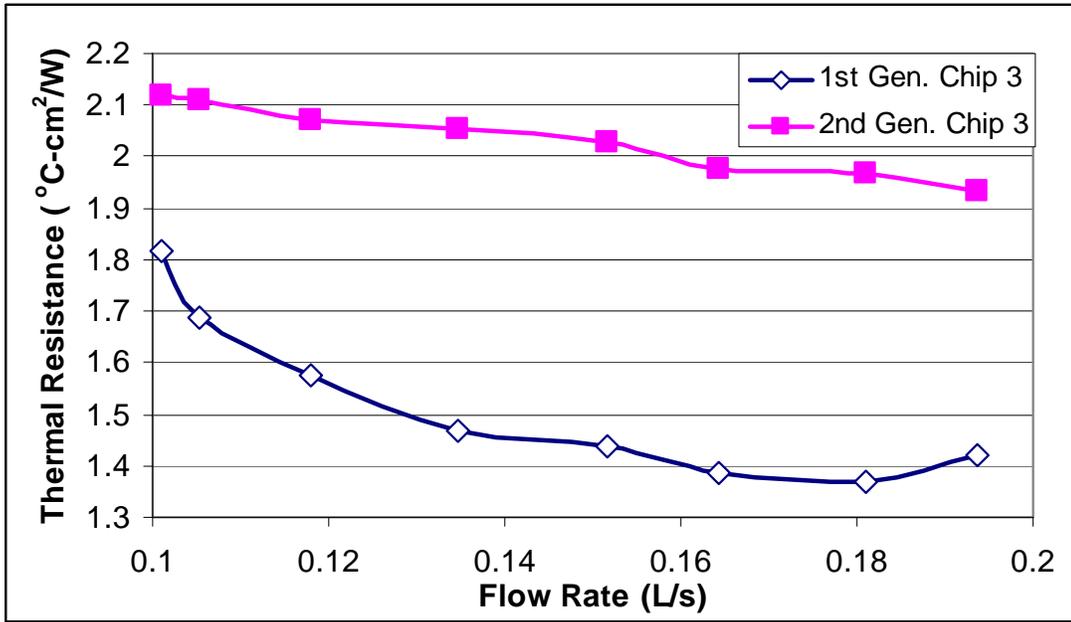


Figure 49: Thermal resistance comparison for first and second generation cold plates (Chip 3)

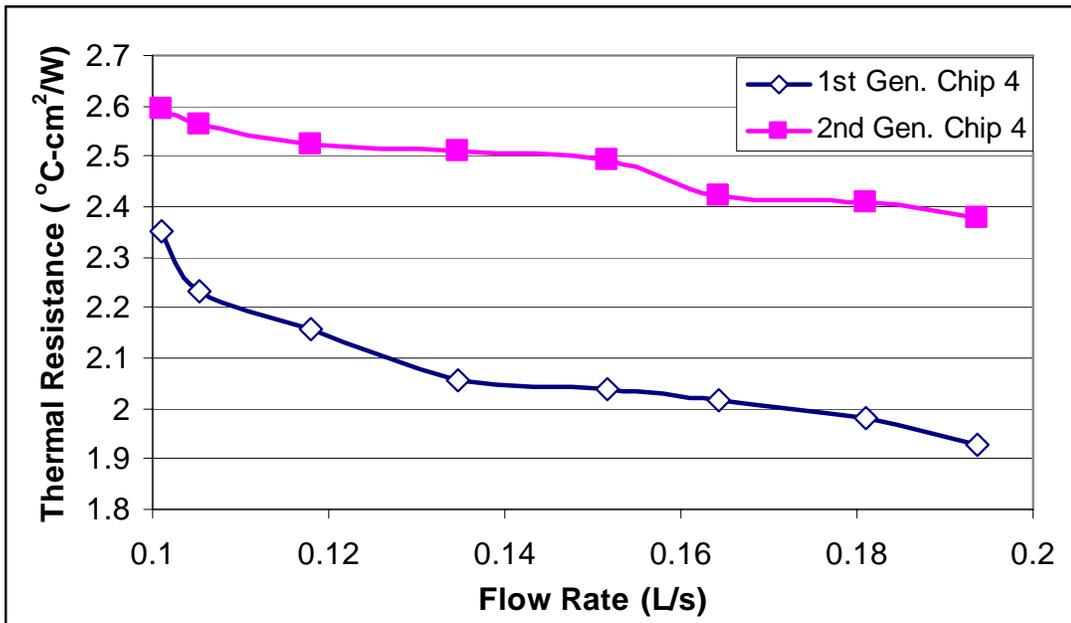


Figure 50: Thermal resistance comparison for first and second generation cold plates (Chip 4)

Comparison of Pressure Drops

Figure 46 shows that the pressure drop in the second-generation cold plate is significantly higher than the pressure drop in the first-generation cold plate. This is predominantly because the second-generation cold plate has longer headers, and smaller but shorter channels.

Chapter 8: Conclusions and Suggested Future Work

This chapter will discuss the conclusions drawn from the results of the previous chapter as well as offer suggestions for possible future research work in this area.

Explanation of the differences in the performance

Most of the differences in performance can be attributed to the flow distribution within the headers and the cold plate.

Pressure drop performance

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

It was thought that this would reduce the flow mal-distribution. 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 mal-distribution in the cold plate. Making the fluid inlet and outlet holes larger would have helped, but the thickness limitation prevented that. Changing the size and length of the connecting channels which distributed the fluid from the inlet channel to the outlet channel would have decreased the mal-distribution as well.

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.

If liquid cooling is to be used, circuit boards must be designed to accommodate the cold plate. Extra room must be made around hot components to accommodate the cold plate and allow it to fit well to the hot components. This will reduce thermal interface resistance. Circuit board components that do not need to be cooled should be placed and designed so that they do not interfere with the cold plate. Finally, hot components must be placed on the board in a way that will simplify the flow distribution and cold plate design.

In order to achieve these goals, thermal management must be taken into account early in the design process, not after the circuit board has been made. 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. The biggest gain in heat transfer performance that can be made by considering the

thermal management solution in the design process lies in the reduction of thermal interface resistance. Designs must allow for precise mating of the cold plate to the chip to minimize thermal resistance. . Making the semiconductors flat, parallel, and not surrounded by obstructions will greatly aid in reducing thermal interface resistance. Since thermal interface resistance is a function of the contact pressure, designing circuit boards to withstand a higher contact pressure would further reduce the thermal interface resistance.

Heat exchangers should also be designed to minimize thermal interface resistance. Heat exchangers should be smooth, flat, and rigid. This will reduce the thermal interface resistance in two ways. Making the surface as smooth as possible will reduce the interface resistance. If the heat exchanger is flat and rigid higher contact pressures can be used which will further reduce thermal interface resistance. A lot of work has been done on these issues with air-cooled heat exchangers and many of the techniques used to reduce the thermal interface resistance of air-cooled heat exchangers can be used reduce the thermal interface resistance of liquid cooled heat exchangers. Finally, because flow distribution

is a problem, designing the circuit board layout with liquid cooling in mind will make it easier to get an even and uniform flow distribution. Component placement should be decided both by electric signal concerns and thermal management concerns.

Proposed Future work

Explore differences in Heat Transfer and Pressure Drop

Most of the future work in this field lies in exploring the cause behind the difference in heat transfer and pressure drop between the two generations, namely quantifying the effect of any flow mal-distribution. The flow distribution must be better understood and controlled to maximize the heat transfer and reduce the pressure drop. In this project, work could be done looking at the effects of enlarging the fluid inlet and outlet paths as well as examining the effect of changing the cold plate from a u-type flow cold plate to a z-type flow cold plate.

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