CFD MODELING AND PARAMETRIC STUDY OF VAPOR CHAMBERS AS HEAT SPREADERS FOR HIGH-POWER ELECTRONIC DEVICES

by

DHANRAJ ARUN PATIL

Presented to the Faculty of the Graduate School of
The University of Texas at Arlington in Partial Fulfillment
of the Requirements
for the Degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

THE UNIVERSITY OF TEXAS AT ARLINGTON

December 2015

Acknowledgements

I would like to thank Prof. Dereje Agonafer for the guidance and constant motivation he has provided me over the past two years of my research and study at The University of Texas at Arlington. It would not have been possible to accomplish this milestone without his invaluable advice and support.

I would also like to thank Prof. Haji-Sheikh and Prof. Kent Lawrence for serving on my thesis committee. I wish to express my gratitude towards Ms. Sally Thompson and Ms. Debi Barton for patiently helping me in all office activities.

Words can never do justice to how much thankful and grateful I am to my parents and my sister. I wouldn’t be in this position if it weren’t for my parents Arun and Bharati Patil, who put ceaseless efforts so that I could realize my dream of pursuing higher studies. I dedicate my work to my family.

November 24, 2015

Abstract

CFD MODELING AND PARAMETRIC STUDY OF VAPOR CHAMBERS AS HEAT SPREADERS FOR HIGH-POWER ELECTRONIC DEVICES

Dhanraj Arun Patil, MS

The University of Texas at Arlington, 2015

Supervising Professor: Dereje Agonafer

Technological developments have resulted in a growing demand for high power electronic devices. Although these high power devices meet the high performance requirements, they also generate a very large amount of heat which adversely affects their operating efficiency. Most part of the heat within a package is generated at the chip and hence it is imperative to keep junction temperature as low as possible. This is commonly achieved by using heat sinks mounted directly on top of the package.

Heat sinks provide a significant enhancement in cooling performance by increasing the surface area. However, base thickness of heat sinks limits its effectiveness and becomes a bottleneck. This results in hot spots and consequently increases thermal resistance.

Using a vapor chamber can reduce thermal resistance by better spreading heat across the heat sink base. This work presents a parametric study of vapor chambers as heat spreaders and discusses the merits of using this technology especially in high power devices through CFD modeling.

Table of Contents

List of Illustrations

List of Tables

Chapter 1

Introduction to Thermal Management of Electronics

1.1 Introduction to Electronics Packaging

Electronics Packaging is a multi-disciplinary approach for enclosing electronic components or their assemblies so as to facilitate in their protection from physical damage and harsh environmental conditions such as moisture and contaminants, as well as provide a path for effective heat dissipation. Some other functions of a package include facilitation of repair and rework operations, electrical testing and power distribution to various elements within the package. The performance of any electronic device depends to a great extent on how effectively it performs electrical functions. However, it also depends significantly on how efficiently the heat generated in the device is removed out of it. Electronic Packaging is essentially a strategy used to cool these electronic devices so as to maintain their temperatures within the operating range.

Cost-effectiveness, feature-rich, visual appeal and robust performance are some of most desirable attributes of modern electronic gadgets. To keep up with these requirements, several parameters such as size, design, material etc need modifications. This substantiates necessity of Electronics Packaging to achieve above mentioned goals.

1.2 Hierarchy in Electronics Packaging

Electronics Packaging is carried out at different levels depending on enormity of the electronic system under consideration. These levels are commonly recognized as follows:

1) First Level Package/Device: This level involves packaging of Chips, Single or Multi, which consists of Dies and Transistors.

2) Second Level Package/PCB: This level involves packaging of a Printed Circuit Board (PCB), which carries components like chips, capacitors, inductances, resistors etc.

3) Third Level Package/Box or Server: This level involves packaging of PCB assemblies, daughter cards, power supply cables, cooling devices like fans, and other peripherals.

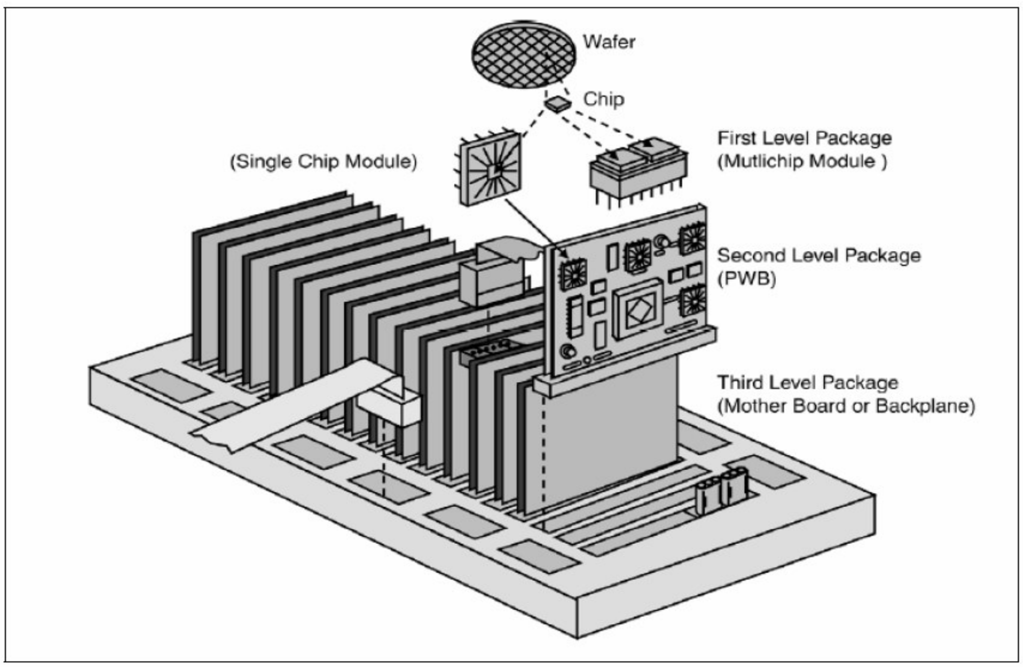

Figure 1-1 Hierarchy in Electronics Packaging

Chip is the most fundamental component in an electronic device; it is manufactured from a silicon wafer. The chip is then mounted onto the substrate through either wire bonding or Flip Chip Attach (FCA). This comprises the first level package. Several such chips, along with other miniature devices, are then placed onto a Printed Circuit Board (PCB) to form an assembly. This comprises the second level package. The

PCB is then enclosed in a box with daughter cards, storage devices, power cables etc which comprises the third level package.

1.3 Thermal Management at different levels

The advent of high-power electronic devices has prompted need for better thermal management. Different strategies that are commonly used for cooling of electronic packages are air cooling, liquid cooling (Water or Oil immersion) and two-phase cooling. The strategy that needs to be used for packaging at these different levels depends on several factors like effectiveness, weight, size, physical and chemical effects, reliability and cost.

1.3.1 First/Device level cooling

The first level package consists of Chip, Die and Substrate which is enclosed within a casing. Conduction is the primary mode of heat transfer at this level and hence conduction of heat from Chip to Case surface is our foremost concern. This could be done by using Heat Sinks mounted directly on top of case surface, and by using high conductivity Thermal Interface Materials (TIM) which have enough viscosity to confirm to the surfaces in contact and avoid gaps which lead to contact resistance.

1.3.2 Second/PCB level cooling

The second level package comprises PCB and other components contained in an enclosure where heat dissipation from PCB takes place by conduction whereas from the heat sink takes place by convection. The convective mode of cooling can be achieved either by air flow over heat sinks or immersion in dielectric liquids like oil.

1.3.3 Third/Server level cooling

The third level package comprises of several servers placed on a rack. These are typically cooled by employing refrigeration systems. Dry air may be blown over the servers by using huge blowers, or evaporative cooling systems consisting of wet cooling pads may be used to blow moist air, depending on thermal management requirements.

1.4 Latest trends in Electronics Cooling

As mentioned earlier, we are facing a rising demand for high-power electronic devices. The devices are dissipating increasing amounts of heat while at the same time their sizes are decreasing. This restricts surface area available for heat transfer and hence leads to rise in junction temperature thereby creating hot spots. Heat sinks have proved to be quite effective, however they still have large spreading resistance. In 1942, R.S Gaugler of General Motors invented a novel technology known today as “Heat Pipes”. However, it was only further developed by George Grover of Los Alamos Laboratory in 1963, by implementing it to cool a nuclear reactor used to generate electricity in space. Heat pipes have proved to achieve heat transfer rates as high as 50000 W/m^{o}K. This is approximately a hundred times more than conventional heat sinks. However, Heat pipes have a limitation of only 1-D heat flow. They cannot be used to transfer heat in multiple directions. Researchers have recently invented a new form of heat pipes known as Vapor Chambers. These are basically “Flat Heat Pipes” which have large surface area as compared to thickness. Vapor chambers can transfer heat in all directions, and thereby reduce spreading resistance when integrated into heat sinks.

Chapter 2

Fundamentals of Vapor Chambers

2.1 Introduction

Vapor Chamber is a 2-phase heat transfer device which uses medium such as water or any other fluid to transfer heat from one point to the other. The fundamental operation of Vapor chamber is similar to that of Heat Pipes, except that heat pipes transfer heat in one direction, whereas vapor chamber can achieve heat transfer in multiple directions. Both of them rely on phase change of liquid to significantly increase their thermal conductivity.

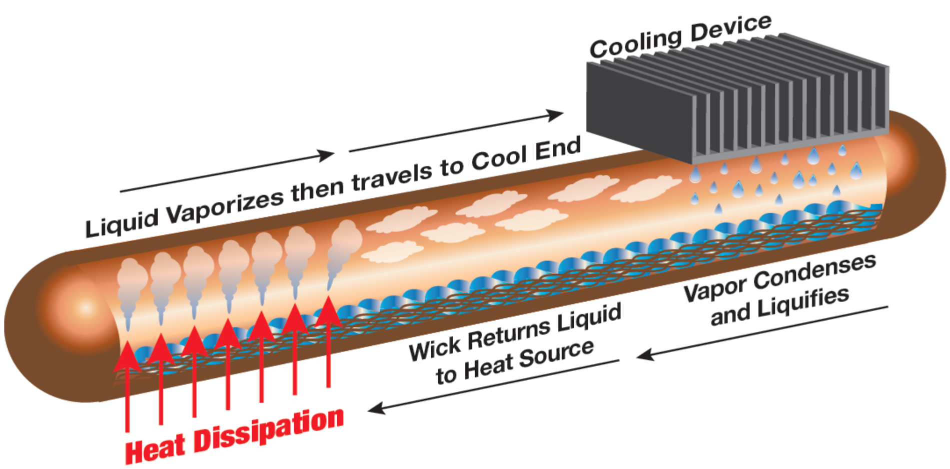

Figure 2-1 Concept of Heat Pipe

Vapor Chamber is basically a vacuum sealed container which has a very small thickness as compared to its surface area, and carries a small quantity of working fluid inside it (typically water). The low pressure inside the chamber allows fluid to vaporize at a temperature much lower than its normal boiling temperature. When heat is applied, fluid at the evaporator immediately vaporizes and rushes to fill the entire volume of the chamber (driven by vapor pressure difference). When the vapor comes in contact with the cool condenser surface, it condenses. The condensed fluid is transported back to evaporator by an internal wick structure through capillary pressure.

The working fluid is contained by wick structure that envelopes the inner surface of the chamber. The thermal conductivity of the working fluid is several thousand times than that of a solid copper block of equivalent dimensions and also has high latent heat of vaporization. Hence the fluid can carry large amount of heat from one surface to the other even at low temperatures. Also, since the vapor can travel in all directions, vapor chambers can spread heat evenly over the bottom surface of heat sinks. This facilitates faster heat dissipation and avoids hot spots.

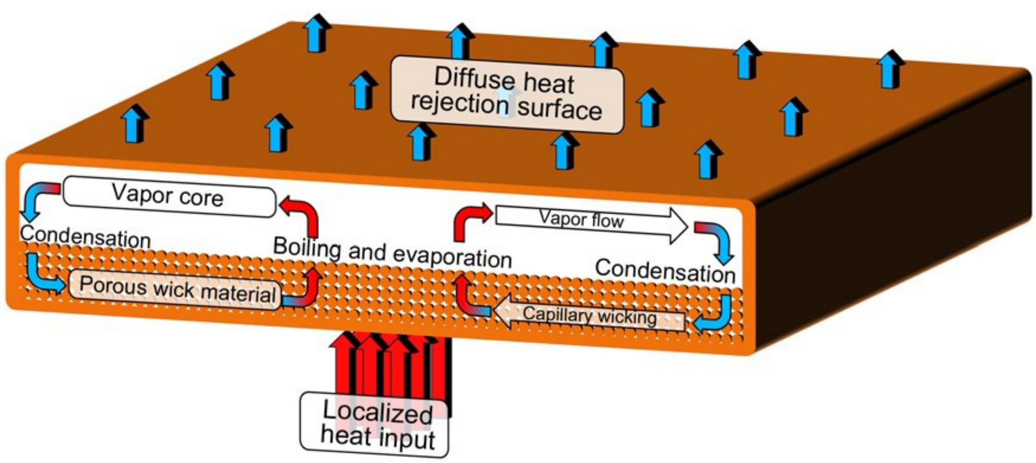

Figure 2-2 Concept of Vapor Chamber

A Vapor Chamber is typically integrated into heat sinks of either copper or aluminum material. Stamped fins are soldered to vapor chamber and can accommodate various vapor chamber footprints. This assembly is then placed directly on top surface of Chip. The hotter surface in contact with the chip is called Evaporator whereas the cooler surface in contact with heat sink base is called Condenser. The chip surface and vapor chamber bottom surface are held in contact by a Thermal Interface Material (TIM) so as to avoid any air gaps and minimize contact resistance.

2.2 Types of Wick Structures used in Vapor Chambers

Wick is the most critical part of vapor chamber because over 90% of the temperature drop in the evaporator section occurs in the wick if the thickness of the wick is same as the thickness of the copper wall of the vapor chamber [7]. The desirable characteristics of wick include capability to hold sufficient amount of working fluid, provide enough capillary pressure to transport the fluid from condenser back to the evaporator, as well as distribute liquid around evaporator section to any areas where heat is likely to be received [1]; this is especially important in case of cooling multi-chip modules. Furthermore, since the overall thermal resistance of evaporator section is largely affected by wick, it should also have good thermal conductivity. The effectiveness of wick is determined by the capillary pressure it can generate; the capillary pressure needs to be more than the vapor pressure and pressure drop in fluid across wick thickness. This is governed by several parameters depending on the type of structure of wick. Three different types of wick structures that are widely used in vapor chambers are explained in Table 2-1 along with their properties and applications.

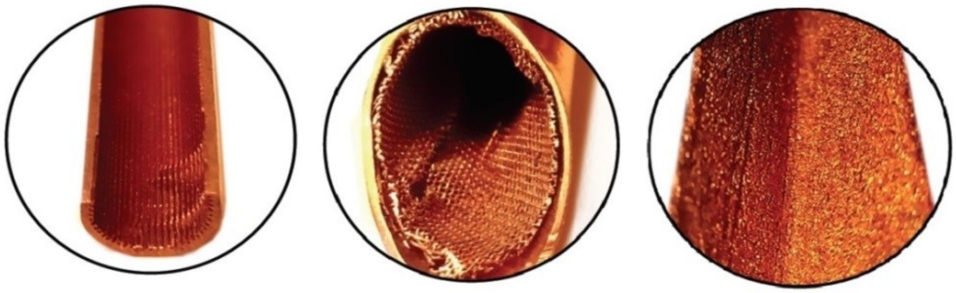

Figure 2-3 Types of Wick Structures

Table 2-1 Characteristics of different wick structures

	Power Density	Thermal Resistance	Orientation
Grooved Wick	< 20 W/cm^2 Lowest performance and cost. Must be gravity aided/neutral	0.35 – 0.22 $^oC/W/cm^2$	+90^0 to 0^0
Wire Mesh/Screen	< 30 W/cm^2 Good for very thin Heat Sink. Limited Bending	0.25 – 0.15 $^oC/W/cm^2$	+90^0 to -5^0
Sintered Powder Wick	< 500 W/cm^2 Good for freeze/thaw and bent shapes Small heat sources up to 1000 W/cm^2	0.15 – 0.03 $^oC/W/cm^2$	+90^0 to -90^0

The effectiveness of Grooved wick depends on width and height of micro-channels as well as gap between them. The surface tension acting on the fluid filled in between the gaps leads to capillary force. The parameters that decide effectiveness of Wire Mesh/Screen wick and Sintered Powder wick are porosity, permeability, wettability and volume fraction of solid phase to liquid phase.

2.3 Effective Thermal Conductivity of Wick

As mentioned in the earlier section, the thermal conductivity of wick is a vital factor affecting the overall thermal resistance of evaporator section, and consequently the performance of vapor chamber. The water acts as a thermal barrier because of much lower thermal conductivity as compared to copper powder. There are several methods used to compute the effective thermal conductivity of wick structure. The wick and water can be assumed to be either in series or parallel [1].

For series assumption, we have relation

$$K_w = \frac{1}{(1-\varepsilon)/K_s \ + \varepsilon/K_l} \quad (1)$$

For parallel assumption, we have relation

$$K_w = (1-\varepsilon)K_s + \varepsilon K_l \quad (2)$$

The effective thermal conductivity of Sintered Wick can be calculated by using the relation derived by Maxwell [2] as follows

$$K_w = \ K_s\left[\frac{2 + K_l/K_s - 2\varepsilon(1 - K_l/K_s)}{2 + K_l/K_s + \varepsilon(1 - K_l/K_s)}\right] \quad (3)$$

Where, k_s is the thermal conductivity of solid phase

k_1 is the thermal conductivity of liquid phase and

ε is the volume fraction of solid phase = 1 – porosity of medium

Gorring, R.L. and Churchill, S.W. [3] show that the above expression agrees reasonably well with experimental results. Kozai, H. *et al*. [4] has confirmed the applicability of above equation for Wire/Screen wicks as well.

The values of effective wick thermal conductivities calculated using above equations are shown in Table 2-2.

Table 2-2 Values of effective thermal conductivities of wick calculated from different equations

Eq. (1)	Eq. (2)	Eq. (3)
185.11	1.153	147.159

For computational analysis purposes, it is thus safe to assume wick thermal conductivity anywhere within the range as calculated using above equations as shown in the table. Vadakkan U. *et al.* [5] has assumed effective thermal conductivity of wick as 40 W/m^{o}K.

2.4 Working Fluid

A vapor chamber fundamentally works on 2-phase process. The working fluid is the medium which undergoes this phase change. Any fluid changes its phase from liquid to vapor, when its temperature is equal to or greater than its boiling point at a pressure known as vapor pressure. Higher the vapor pressure, higher is the boiling point. Electronic devices require to be operated at temperatures in the range of 30^{0}C to 70^{0}C. Hence, it is important to choose a working fluid which has boiling point in this range, if it has to undergo phase change process. The high amount of heat dissipation from evaporator to condenser is a result of this phase change, since the fluid carries heat equivalent to its latent heat of evaporation. The latent heat of evaporation of a fluid is the amount of heat absorbed or released by the fluid, during a constant-temperature process, to undergo a change of phase from liquid to vapor or vice versa. The higher the latent heat of evaporation, the greater is the heat transfer capability of the fluid.

2.5 Types of Working Fluids

The desirable properties of working fluids to be used especially in Vapor Chambers include high latent heat of evaporation, specific heat capacity, chemical stability with a wide range of materials, and most importantly low boiling point in the range of operating temperature of the electronic device. Many of the working fluids available at present do not possess all of the above desirable properties. Hence, the choice of fluid should be made according to the application. Table 2-3 shows properties of fluids that are commonly used for phase change applications [1].

Table 2-3 Types of fluids used in phase change applications

Fluid	Boiling Point at 14.69 psi	Vapor Pressure for boiling point of $45^{o}C$ (psi)	Latent Heat of Vaporization (kJ/kg)
Ammonia	-33	228.9	1369
Pentane	35	24.65	348
Acetone	56	9.847	518
Methanol	64.6	6.45	1100
Ethanol	78.37	3.33	846
Water	**100**	**1.38**	**2264.76**

Choice of fluids depends primarily on operating temperature range specified for the device, which typically is $30\text{-}80^{0}C$, because it is very important that the fluid undergoes change within that temperature range. The phase change of fluid occurs at constant temperature, which is known as the boiling point of the fluid. Boiling point of fluid depends on the vapor pressure acting on fluid. Boiling point of various fluids at atmospheric pressure of 14.69 psi is shown in the table. We can see that the boiling temperature of most fluids is higher than desirable boiling point of around 45^0C. Hence, we need to reduce the vapor pressure of any chosen fluid such that its boiling point is reduced to 45^0C. The pressures that need to maintained within the chamber in order to do so is shown in table. Fluids like Ammonia and Pentane require very high pressures whereas water requires very low pressures. Water, however, seems to be the fluid which meets most desirable properties like high latent heat and specific heat capacity. It is also chemically stable with a wide range of materials and has minimum safety issues when used in electronics if the designing is proper. Furthermore, it costs lesser than other fluids and is readily available.

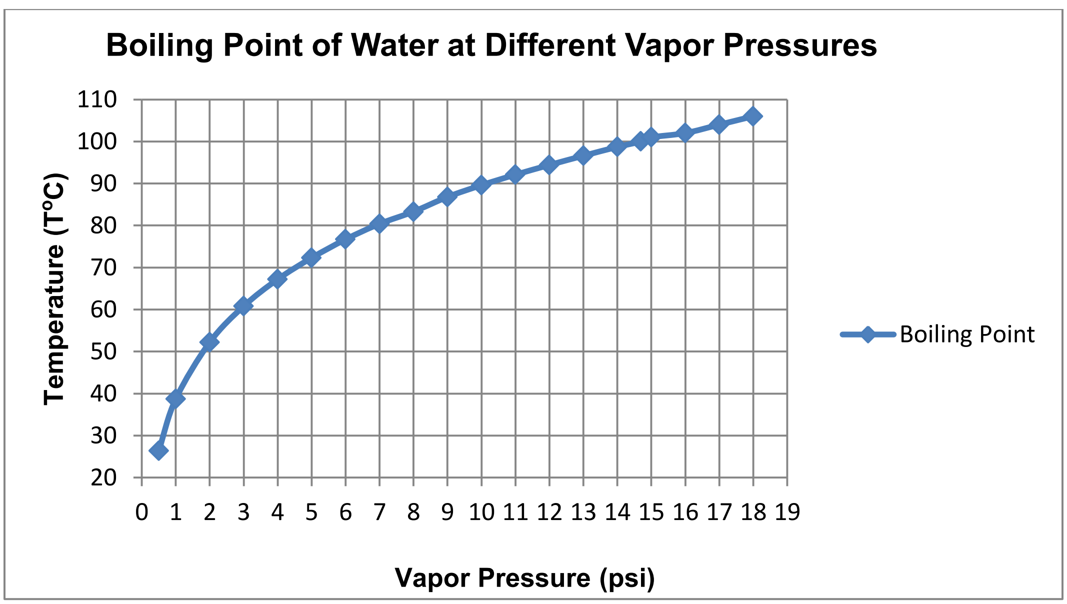

Figure 2-4 Boiling Point of Water at different Vapor Pressures

2.6 Effective Thermal Conductivity of Vapor Space

Vapor space is the region of highest thermal conductivity and hence highest heat flux and lowest thermal resistance within a vapor chamber. Vapor is mainly responsible for uniformly distributing heat over the base of heat sink. According to Prasher [6], the effective thermal conductivity can be calculated using a relation based on ideal gas law and Clapevron equation for incompressible laminar flow, given by

$$K_{vapor_{VC}} = \frac{L^2 p_v \rho_v d_v^2}{12 R \mu_v T^2}$$

Where,

$L =$ Latent Heat of evaporation (J kg^{-1})

$p_v =$ Vapor Pressure (N m^{-2})

$\rho_v =$ Vapor Density (kg m^{-3})

$d_v =$ Thickness of Vapor Space (m)

$R =$ Gas Constant per unit mass (J kg^{-1})

$\mu_v =$ Viscosity (kg m^{-1} s^{-1})

$T =$ Temperature of Vapor (^{0}C)

Theoretically calculated effective thermal conductivity comes out to be around 11260 W/m^{0}K. However, experimental results show that thermal conductivity of Water Vapor thermal conductivity can be safely assumed as high as 30000 W/m^{o}K [6]. Also, small changes in vapor thermal conductivity will not significantly affect evaporator resistance [6].

2.7 Vapor Chamber performance limits

Thermal performance of vapor chambers depends on some parameters which need to be determined through experimental data for different configurations and thermal models to predict it accurately. In the case of vapor chambers designed as heat spreaders, two such limitations need to considered, namely, the capillary and the boiling limits [6]. Other performance limits, such as entrainment and sonic limits which exist in heat pipes, are not as critical in case of vapor chamber heat spreader because of its geometry.

The capillary limit depends on the fluid pressure drop required to return the liquid from the condenser to the evaporator. For proper operation, the total fluid pressure drop must be lower than the fluid capillary pressure in the evaporator section. The fluid capillary pressure can be estimated using the following equation [1]:

$$\Delta P_c = 2\sigma \frac{\cos\theta}{r_p}$$

Where, σ is the surface tension of the liquid, θ is the wetting angle, and r_p is the effective pore radius of the wick. The fluid pressure drop can be estimated using a form of Darcy's equation [7]

$$\Delta P_l = \frac{1}{K}\mu V_s \Delta x$$

Where, K is the wick permeability, μ is the viscosity of the liquid, V_s is the liquid surface velocity, and Δx is the distance between the condenser and the evaporator edge.

Chapter 3

Computational Fluid Dynamics (CFD)

3.1 Introduction

Computational Fluid Dynamics (CFD) is a tool which provides qualitative (and sometimes even quantitative) prediction of fluid flow by means of Mathematical Modeling, Numerical Methods and Software tools (like Fluent or Icepak). Computers are used to perform complex and time consuming calculations required to simulate the interaction between liquids and gases at surfaces defines by initial and boundary conditions. This reduces processing time required to perform these calculations analytically.

The fundamental process that goes behind computational fluid dynamics analysis involves solving Navier-Stokes equations for viscous flow by means of a finite-difference numerical technique. These equations typically involve integrals and partial derivatives. Computational Fluid Dynamics is the art of replacing these integrals and derivatives with discretized algebraic forms, which are then solved to obtain numbers for the flow field values at discrete points in time and/or space. The solution results thus help us identify hot spots or regions which require more cooling and thereby also help us optimize our design for thermal efficiency.

3.2 History of Computational Fluid Dynamics

One of the earliest attempts made at performing the type of calculations that resemble modern CFD are those by Lewis Fry Richardson in early 1940, in the sense that these calculations made use of finite differences that divided physical space in cells. The first work involving use of computers to model fluid flow, as governed by Navier-Stokes equations, was performed a group led by Francis H. Harlow at Los Alamos National Lab in 1957 [8]. The first paper which involved a three-dimensional model was published by John Hess and A.M.O Smith of Douglas Aircraft in 1967. Since then, several CFD codes developed by researchers began to be widely used ultimately leading to numerous commercial packages like Fluent and Icepak.

3.3 Methodology used by commercial packages for CFD analysis

Numerous commercial packages are available today to perform computational fluid flow analysis. The approach used by any of these software is the same. The methodology followed is briefly explained as below [9].

Preprocessing:

- The geometry of problem is defined and a model is created
- Type of analysis is defined (Steady state or Transient)
- Type of flow is defined (Laminar or Turbulent)
- Initial Conditions and Boundary Conditions are given wherever required (Temperature, Velocity, Heat Flux, Power etc.)
- Governing equations are defined based on type of analysis (Conduction, Convection, Radiation etc.)
- Model is meshed with desired type of mesh depending on application (Uniform, Non-uniform, Coarse, Fine etc.)

Solution:

- Meshing is the foremost thing to be done before we run the solver
- Appropriate element size and type of mesh is chosen so as to improve mesh quality
- Mesh sensitivity analysis may be done so as to make sure the mesh quality is high enough to get more accurate results

- Model is first checked for errors like improper contact, insufficient parameters etc.
- The computer runs the solution and calculates initial values at different locations within the geometry.
- These values are then reverse substituted for the given number of iterations to achieve convergence.

Post processing:

- This is step involves plotting results (Object face and Plane Cut)
- Monitor points may be created at various locations to find exact values of different parameters (like Temperature, Velocity, Heat Flux etc.) at that location

Chapter 4

Computational Fluid Dynamics (CFD) modeling of Vapor Chamber

4.1 Importance of Computational Modeling

The first level/chip level electronic package focusses on cooling of chips which are the primary source of heat generation. It is important to know the performance achieved by vapor chambers before they are deployed into high-power devices because their manufacturing involves huge costs. Hence, a number of tests need to be done to find out their effectiveness for a particular application. However, testing different configurations of vapor chambers only adds to the costs involves. To save time as well as cost involve in testing several configurations, we can make use of computational tools available and optimize different parameters so that we only need to manufacture the vapor chamber model which gives the most efficient cooling performance. Computational modeling is thus important for any engineering application.

In this study, computational modeling of Vapor Chamber is carried out using advanced CFD software ANSYS Icepak. Computational Fluid dynamics is employed to develop a conduction model of the vapor chamber and thereby analyze various components involved for temperature rise when subjected to Power. This model is compared to a simple copper block model to judge the effectiveness and merits of Vapor Chamber technology over conventional copper spreaders.

4.2 Model Setup in Ansys Icepak

The 3-D model of Vapor chamber designed using ANSYS Icepak is shown in Figure 4-1. A baseline model is developed and then parametric study is performed by changing several parameters which will be explained in later sections. The package consists of several components that sit on top of each other. The dimensions of these components and the corresponding material properties are shown in Table 4-1.

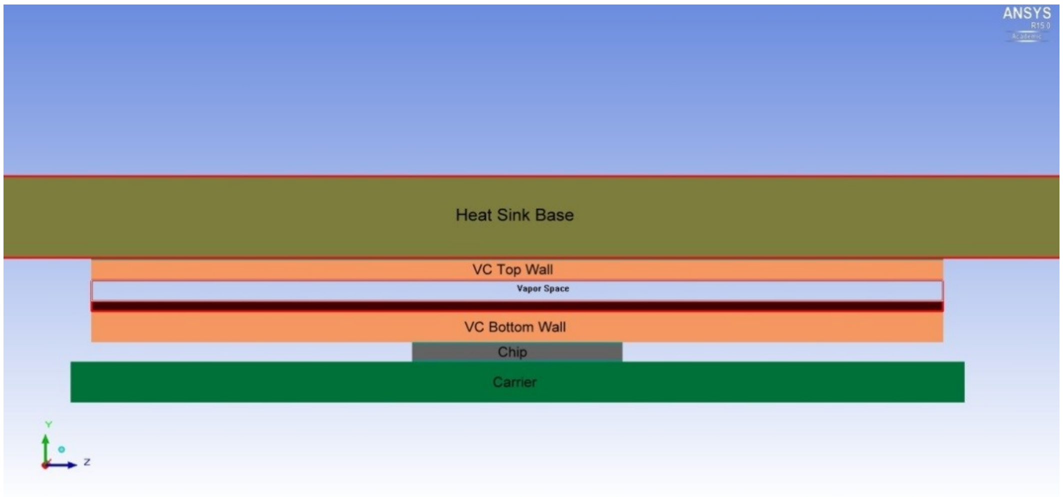

Figure 4-1 CFD Model of Vapor Chamber in Icepak

Table 4-1 Dimensions and Material Properties of components for Baseline Case

Part	Model	Material/Thermal Conductivity (W/m^{o}K)	Dimension (mm)
Carrier	Block	Ceramic/15	42.5 x 42.5 x 2
TIM 1	Plate	Thermal Paste/4.5	9 x 9
Chip	Block	Silicon/148	9 x 9 x 0.785
TIM 2	Plate	Thermal Paste/3.5	9 x 9
VC Bottom Wall	Block	Copper/387.6	37.5 x 37.5 x 1.5
Wick	Block	Sintered copper powder/50	37.5 x 37.5 x 0.5
Vapor	Block	Water Vapor/30000	37.5 x 37.5 x 1.0
VC Top Wall	Block	Copper/387.6	37.5 x 37.5 x 1.0
TIM 3	Plate	Thermal Paste/3.5	37.5 x 37.5
HS Base	Block	Aluminum/240	90 x 90 x 4

The bottom-most part is the carrier/substrate which carries the chip. The source is placed at the center of chip and has a Thermal Design Power (TDP) of 100 W. The Vapor Chamber bottom wall is in contact with the top surface of chip via Thermal Interface Material (TIM1). This is done to avoid air gaps and minimize contact resistance. The wick structure is modeled as solid block of copper material with effective thermal conductivity of 50 W/m^{o}K. The effective thermal conductivity is calculated using empirical relation provided in section 2.3. The vapor space is modeled with fluid material as water vapor having effective thermal conductivity of 30000 W/m^{o}K.

4.3 Parts in ANSYS Icepak

Choosing appropriate part for a component is very important because the type of part you use greatly affects the boundary conditions and hence thermal behavior. For example, if you choose a Block instead of Plate for a component which does not conduct heat along the thickness direction but rather along the plane (2-D), you will get completely different – and mostly inaccurate – results. This section describes the parts that were – and should be – used to create the model in Icepak [9].

4.3.1 Cabinet:

Cabinet is the space under consideration for analysis. All the components which are placed within the boundaries of cabinet are only considered to exist. We can also specify the wall types for all six sides depending on the application. The wall types that can be assigned are Wall, Opening and Grille.

4.3.2 *Block:*

This is the most commonly used part in a wide range of models, because blocks conduct heat in all directions unless, of course, a boundary condition prevents from doing so. In current model, Blocks were used to model Carrier, Chip, Vapor Chamber Walls, Wick, Vapor Space and Heat Sink Base since all these components conduct heat in all directions, in real-life scenario. However, some boundary conditions were applied to a few components. The wick and vapor have dominant heat conduction in thickness direction as far as vapor chambers are concerned. Hence, the heat flux in other directions was restricted. Also, the heat sink is assumed to have a convective heat transfer coefficient of 1400 W/m^2K, which is a typical value for fan cooled heat sinks using air as medium. This boundary condition was placed at the top of heat sink base rather than modeling entire heat sink, so as to minimize complexity.

4.3.3 *Plate:*

The plate is typically a part which has negligible thickness as compared to its surface area. From thermal point of view, plates are used for components which are assumed to conduct heat only along its surface area (in plane) and exhibit minimum heat conduction along its thickness. Hence, in current model, this part was used to model Thermal Interface Materials (TIM). As the name suggests, the purpose of Thermal Interface Materials is only to minimize contact resistances at the interface of different components and not to conduct heat. However, they conduct very small amount of heat along its surface. Hence, using plates to model them makes more sense than modeling them as resistances.

4.3.4 Source:

A source is something that generates heat. In electronic devices, the primary component that generates heat is the Chip. The amount of heat generated depends on the electrical power consumed by the chip. The amount of heat generated may not be exactly same as the power, since some of the heat is used to raise the temperature of the chip itself as result of its specific heat capacity. However, since the difference is very small, the amount of heat generated is considered equivalent to the actual power input to the chip in order to simplify analysis. In current model, the source is placed at the center of the chip, and has the same footprint as chip.

4.4 Setup for analysis

This section explains various settings that were used for analysis.

4.4.1 Basic Parameters

- Variables solved: Temperature
- Radiation is turned off
- Ambient temperature is set at 39^{o}C
- Type of analysis: Steady State

4.4.2 Solution Settings

- Energy convergence criteria: 1e-15
- Discretization scheme for temperature: Second order
- Solver type: W
- Precision: Double

4.4.3 Meshing

- Mesh type: Mesher-HD
- Max element size: 0.4 mm
- Mesh parameters: Normal
- Minimum elements in gap: 4
- Minimum elements on edge: 3
- Max size ratio: 2
- Number of elements: 924685

4.5 Results of Analysis

4.5.1 Plane Cut Temperatures

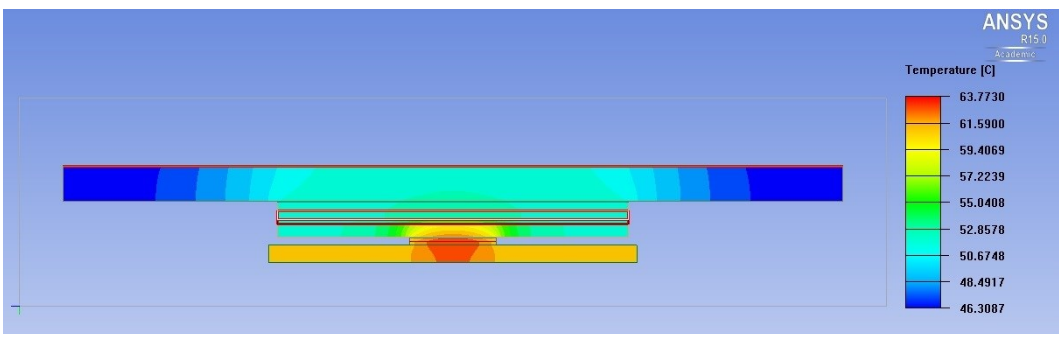

Figure 4-2 Plane cut temperature for vapor chamber model

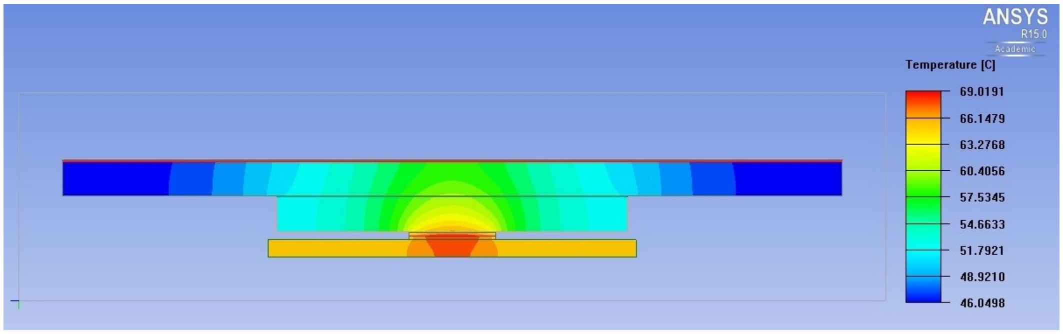

Figure 4-3 Plane cut temperature for copper spreader model

4.5.2 *Object Face Temperatures*

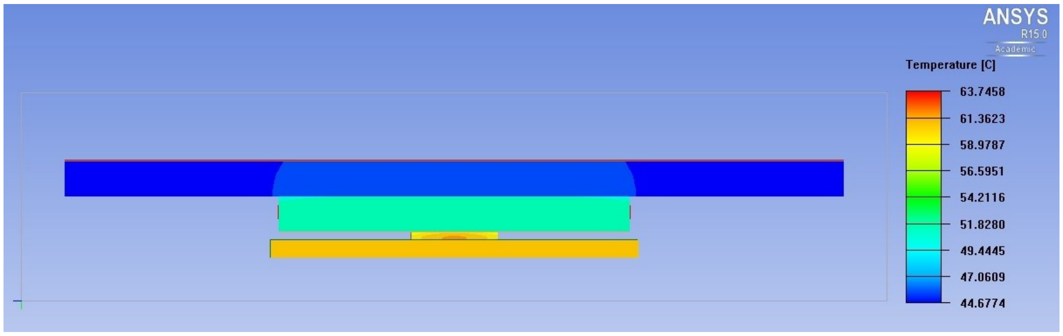

Figure 4-4 Face temperatures for vapor chamber model

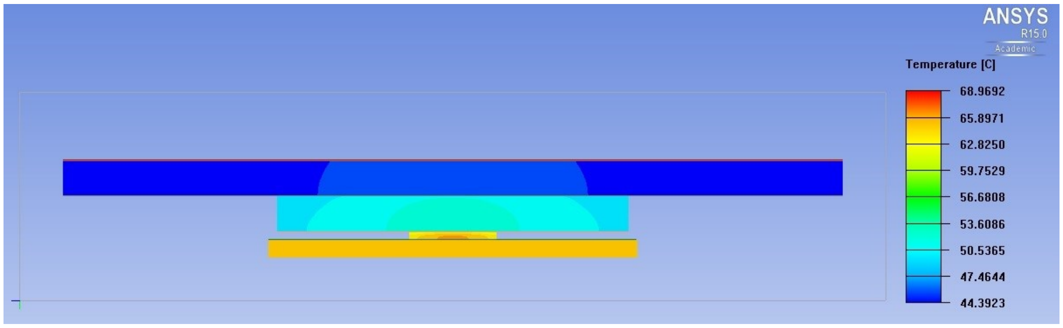

Figure 4-5 Face temperatures for copper spreader model

4.5.3 Temperature distribution

The temperature distribution over the surface of different components for both vapor chamber and copper spreader model were compared. Figure 4-6 shows the temperature at various locations.

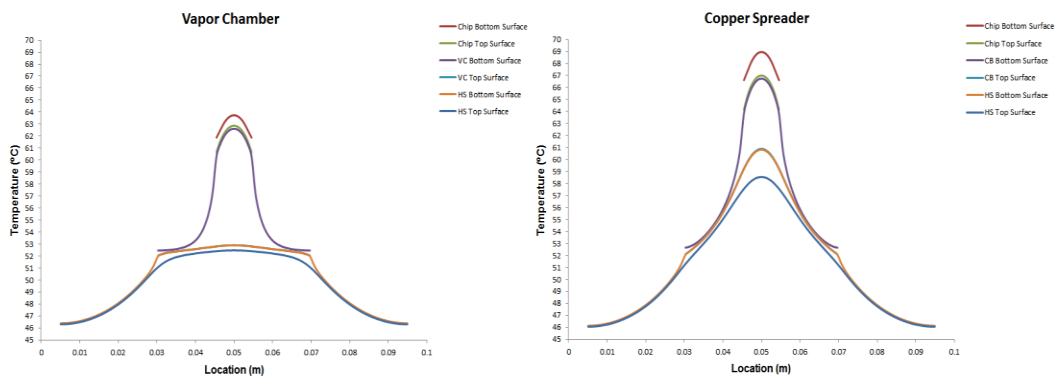

Figure 4-6 Temperature distribution over the surface of different components

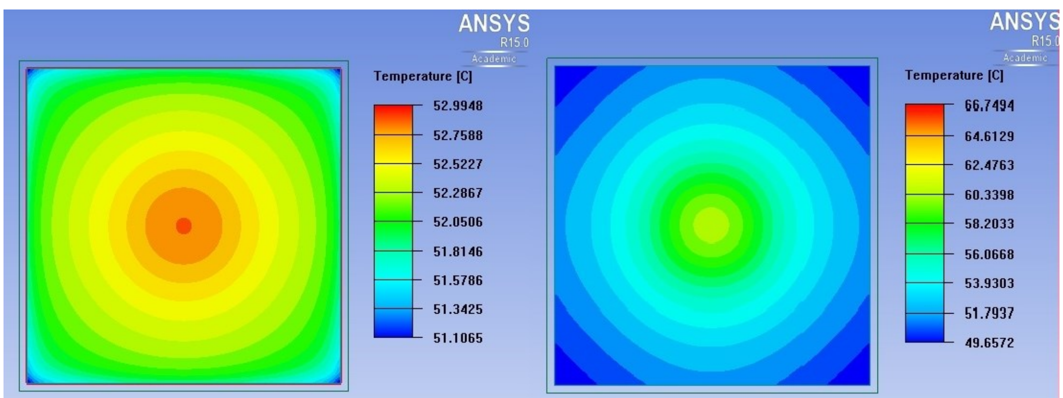

Figure 4-7 Top surface temperature distribution of vapor chamber and copper spreader

Figure 4-7 shows temperature distribution over the top surfaces of vapor chamber on the left and copper spreader on the right. The temperature is observed to be

~8^{o}C lower in vapor chamber model as compared to copper spreader model. Moreover, the temperature variation for vapor chamber model is less than 2^{o}C, whereas for copper spreader model it is ~17^{o}C. Hence, it can be implied that vapor chambers are better heat spreaders than copper.

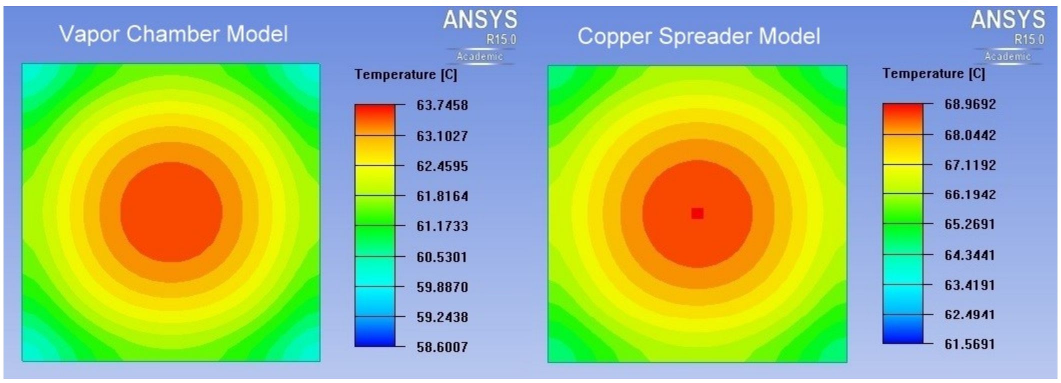

Figure 4-8 Junction temperatures for vapor chamber and copper spreader models

Figure 4-8 shows temperature distribution over top surfaces of chip in both cases of using vapor chambers and copper spreaders. It can be clearly seen that we can achieve ~5^{o}C lower junction temperature if vapor chambers were used in place of copper spreaders. The junction-to-ambient thermal resistance (θ_{JA}) is 0.247^{o}C/W for vapor chamber model and 0.299^{o}C/W for copper spreader model. The vapor chamber model has ~17% less thermal resistance than copper spreader model. In other words, vapor chambers can achieve ~17% better thermal performance than copper spreaders.

4.6 Intel's Thermal Test Vehicle (TTV) data for 6th Gen Core i7 processor

The dimensions of the chip used in the current study were chosen same as that of the 6th Gen Core i7 processor which intel released in the Q1 of 2015. This was done so as to validate the CFD model with actual Thermal Test Vehicle (TTV) thermal profile for the 6th Gen Core i7 PCG 2015D processor published by Intel [11] as shown in Figure 4-9.

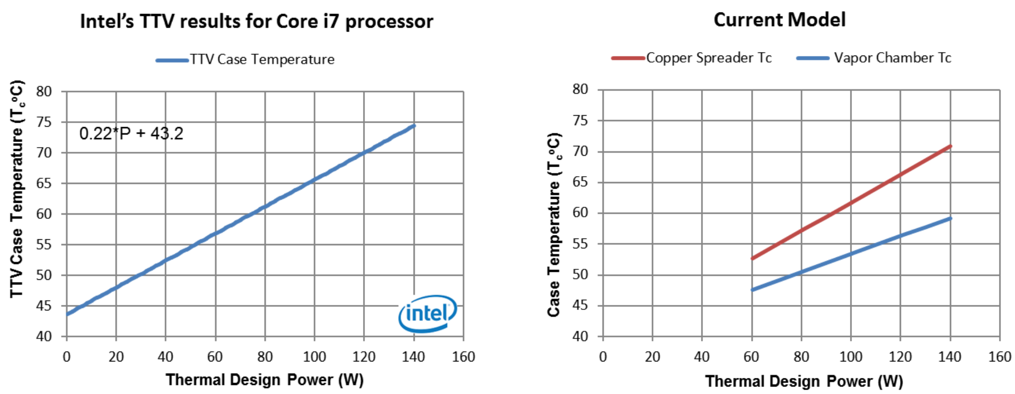

Figure 4-9 Comparison of Junction temperatures of vapor chamber and copper spreader models with Intel's core i7 processor TTV results

The analysis results of current model agree well with the TTV results. Copper spreader model showed almost exact trend of variation in junction temperatures with power as the Intel's processor. The vapor chamber model, however, showed better performance. The case temperature for vapor chamber model was lower than that for copper spreader model at any given power level by ~5°C. These results only reiterate the fact that vapor chambers have better thermal performance than copper spreaders.

Chapter 5

Parametric Study of Vapor Chambers

Vapor chambers have about eleven different components which have different characteristics and properties. The properties like thermal conductivity, heat flux, convective heat transfer coefficient of heat sinks, coupled with dimensional changes have a significant effect on the performance of Vapor chambers. It is time consuming and tedious to quantitatively predict the end result. Also, it is important from engineering point of view to optimize these parameters so that our final design could have maximum possible efficiency. Hence, parametric study of vapor chambers is performed so as to examine effect of changes in various parameters on thermal behavior of the vapor chamber. The results of this parametric study can further serve as a guideline for vapor chamber optimization.

The parameters that are studied in this work include:

- Effect of changes in width of spreader
- Effect of changes in thermal conductivities of Wick and Vapor
- Effect of changes in convective heat transfer coefficient of heat sink
- Effect of changes in source footprint
- Effect of variation in heat sink size

5.1 Effect of Changes in width of spreader

The baseline case has width of spreader as 37.5mm x 37.5mm. The width of both vapor chamber and copper spreader were changed in steps of 5mm x 5mm to study the changes in Junction as well as Case temperatures. The trend is shown in Figure 5-1. The benefit of using vapor chamber increases as width increases. At smaller widths, however, Copper spreader outperforms Vapor Chamber. For the current configuration, we can clearly see the cross-over point around 15.5mm x 15.5mm.

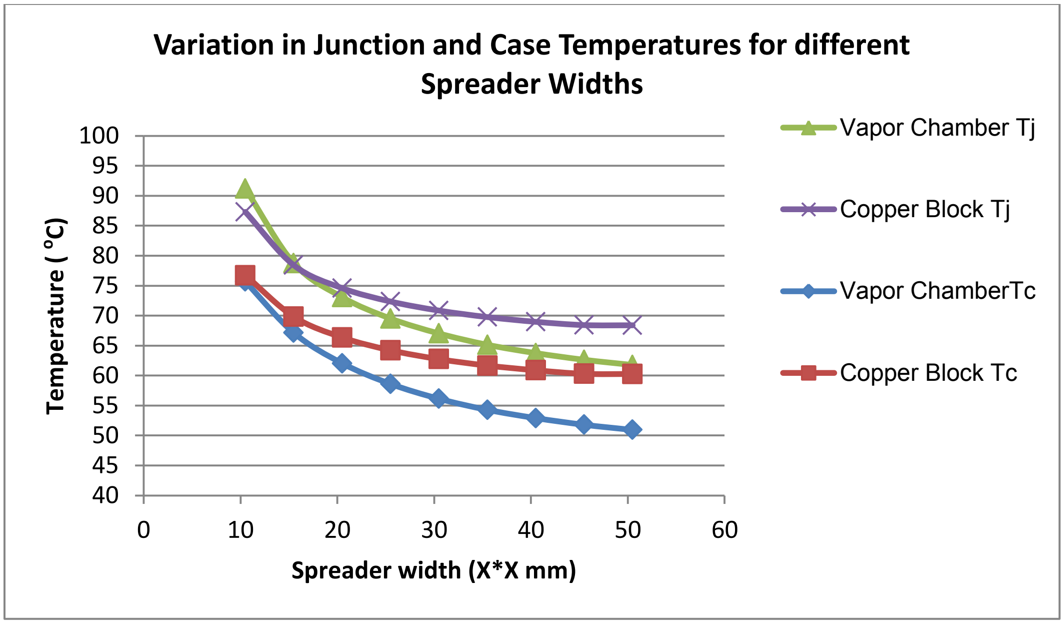

Figure 5-1 Effect of spreader width on thermal performance of both Vapor Chamber and Copper Spreader

Thus we can imply that vapor chamber is particularly useful in applications which require large area for heat transfer because of high heat-flux or low heat transfer coefficients as in case of air cooling.

5.2 Effect of changes in thermal conductivities of wick and vapor

As discussed in section 2.3 and section 2.6, there is still no definite method available to accurately predict values of effective thermal conductivities of wick and vapor space, because of the dependence of their values on geometry and phase chase. Hence, we have to rely on assumption from different theories and experimental work. However, in order to understand the effects of variations in the values of effective thermal conductivities, a parametric study was performed.

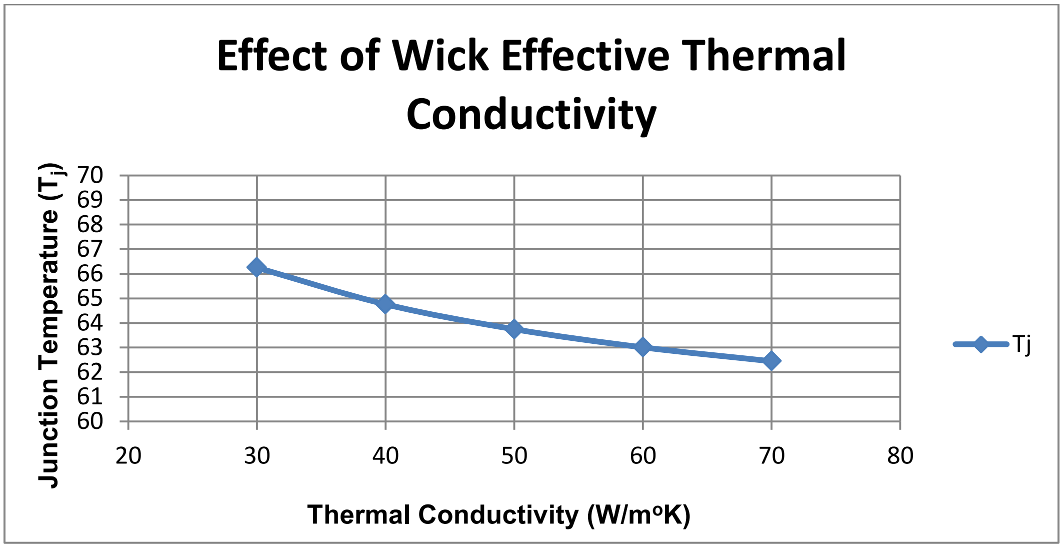

Figure 5-2 Effects of wick thermal conductivity

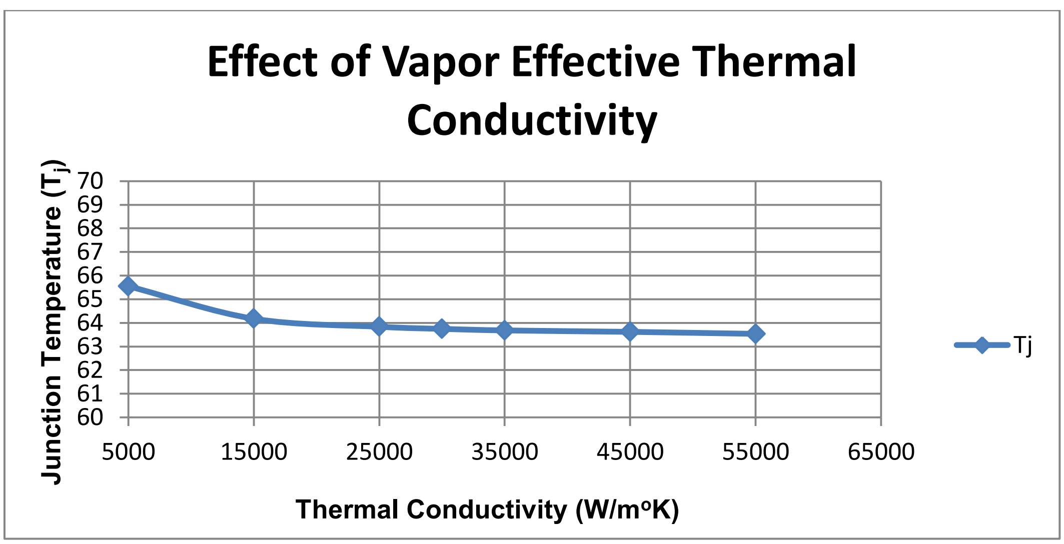

Figure 5-3 Effects of vapor thermal conductivity

Figure 5-2 shows variation of junction temperature with wick conductivity. In this case, the vapor thermal conductivity was kept constant at 30000 W/m^{o}K whereas wick conductivity is varying. Figure 5-3 shows variation of junction temperature with vapor thermal conductivity. In this case, the wick thermal conductivity is kept constant at 50 W/m^{o}K whereas vapor thermal conductivity is varying. We can clearly observe from the graph that the effective thermal conductivity of wick more significantly affects thermal performance than vapor thermal conductivity. Hence, considering uncertainties that still exist in predicting the exact value, a conservative approach is necessary to be adopted while designing vapor chambers.

5.3 Effect of changes in convective heat transfer coefficient of heat sink

Heat sink acts as the condenser which dissipates heat from system to the surrounding. The rate of dissipation depends on the convective heat transfer coefficient, which is different for different types of cooling methods. For air cooled heat sinks, the range of heat transfer coefficients is from 10-1500 W/m2K. Water based cooling technologies like cold plates can provide heat transfer coefficients in the range of 2000-15000 W/m^{2}K. However, air cooling is the most preferred method of cooling even today. Hence, a convective heat transfer coefficient of 1400 W/m^{2}K was considered for baseline study and this boundary condition was applied at the top surface of heat sink.

Figure 5-4 and Figure 5-5 show Junction and Case temperatures respectively for both cases of using vapor chamber and copper spreader for a range of convective heat transfer coefficients. As heat transfer coefficient increases, the junction temperature decreases rapidly at lower values and gradually at higher values for both cases.

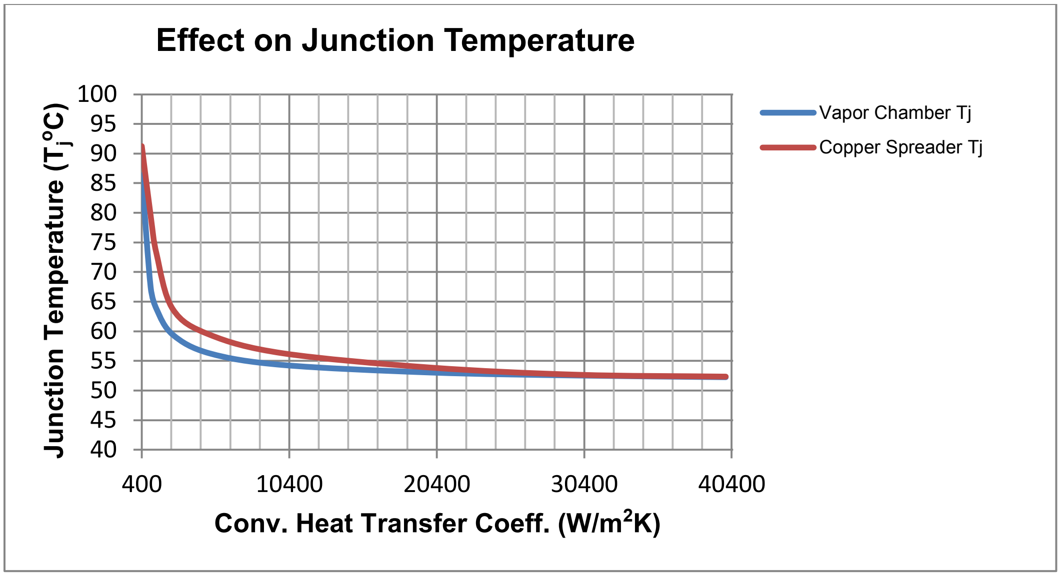

Figure 5-4 Effects of convective heat transfer coefficient on Junction temperature

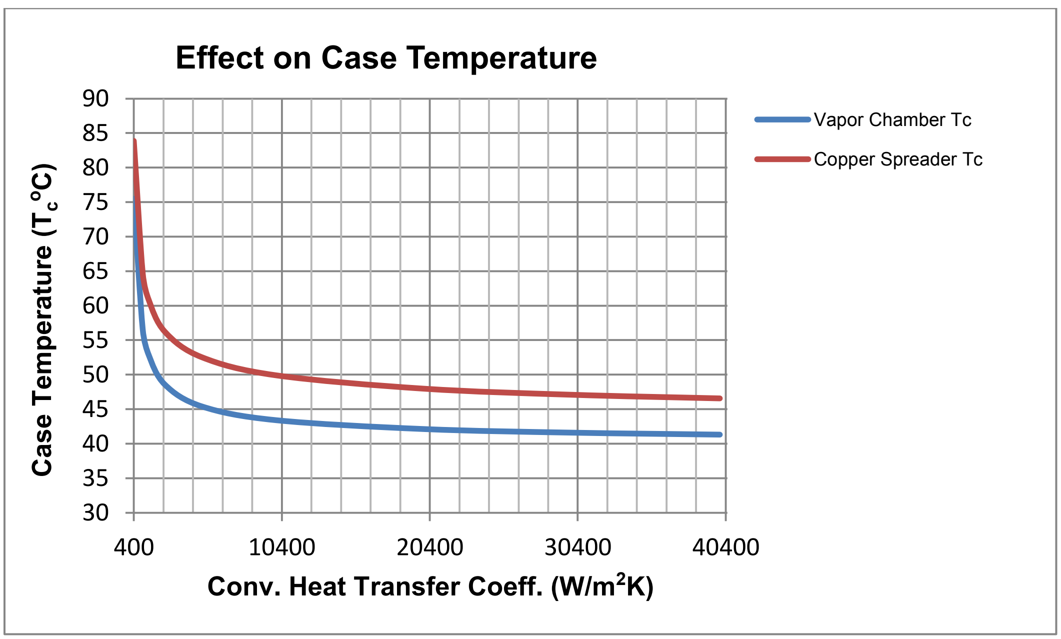

Figure 5-5 Effects of convective heat transfer coefficient on Case temperature

The junction temperature is lower for vapor chamber case than copper spreader at low heat transfer coefficients, but for very high coefficients, it is approximately same for both cases. This is probably because at high heat transfer coefficients, heat flow becomes one dimensional thus reducing lateral spreading. Thus we can conclude that it is more beneficial to implement vapor chambers for air-cooled heat sink applications which usually have very low heat transfer coefficients.

5.4 Effects of changes in heat source footprints

Technological advances have made it possible to fit more number of transistors on small dies. As a result, the size of chip is reducing whereas the power they consume is increasing. This situation leads to higher junction temperature and heat flux. Heat spreading becomes very critical in such cases. The baseline case considered has a source footprint of 10 mm x 10 mm. This is typical size of some of latest processor chips. The source size was varied in steps of 5 mm to examine the junction to ambient resistance θ_{JA} at a power of 100 W.

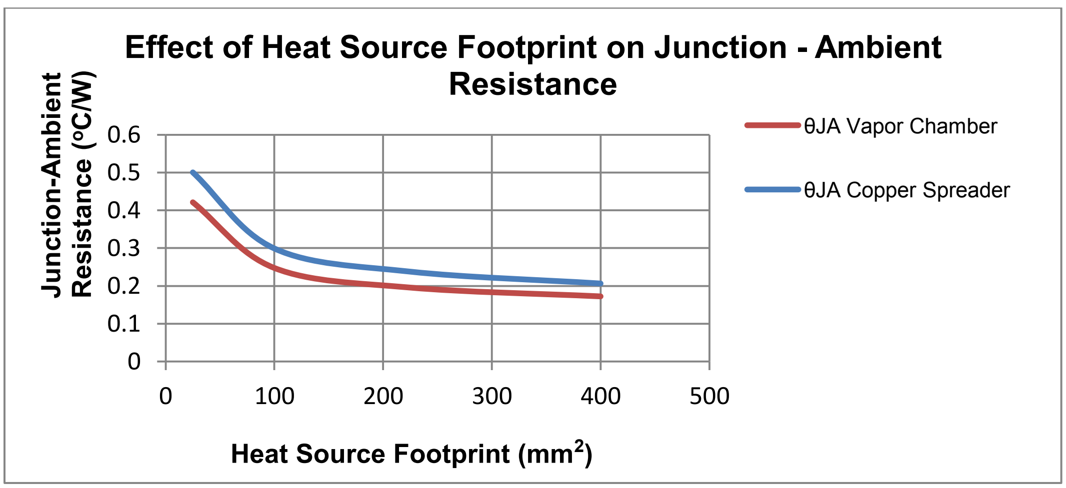

Figure 5-6 Junction to Ambient Resistances for different heat source footprints

From Figure 5-6, it is observed that the junction to ambient resistance (θ_{JA}) is higher for smaller footprint areas and sharply decreases as area increases until 100 mm for both cases. After this point, however, the change in resistance is very less. The further decrease in resistance becomes dependent on factors other than the source size.

5.5 Effect of variations in size of heat sinks

Size of heat sink required to dissipate heat depends on several factors like type of cooling medium, number of fins required, material used for heat sinks etc. The primary goal for any application is to be able to utilize as smaller and lighter heat sink as could be possible to maintain junction and case temperatures within operating temperature limits specified by device manufacturer. It is easy and efficient to cool heat sinks which have uniform temperature over its base.

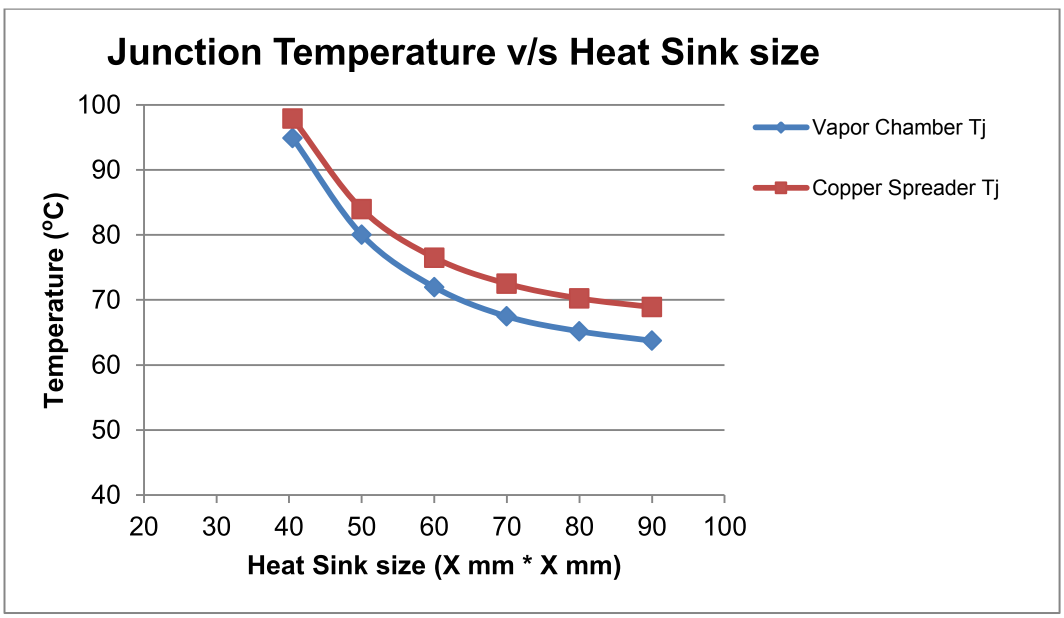

Figure 5-7 Variation in junction temperature with heat sink size

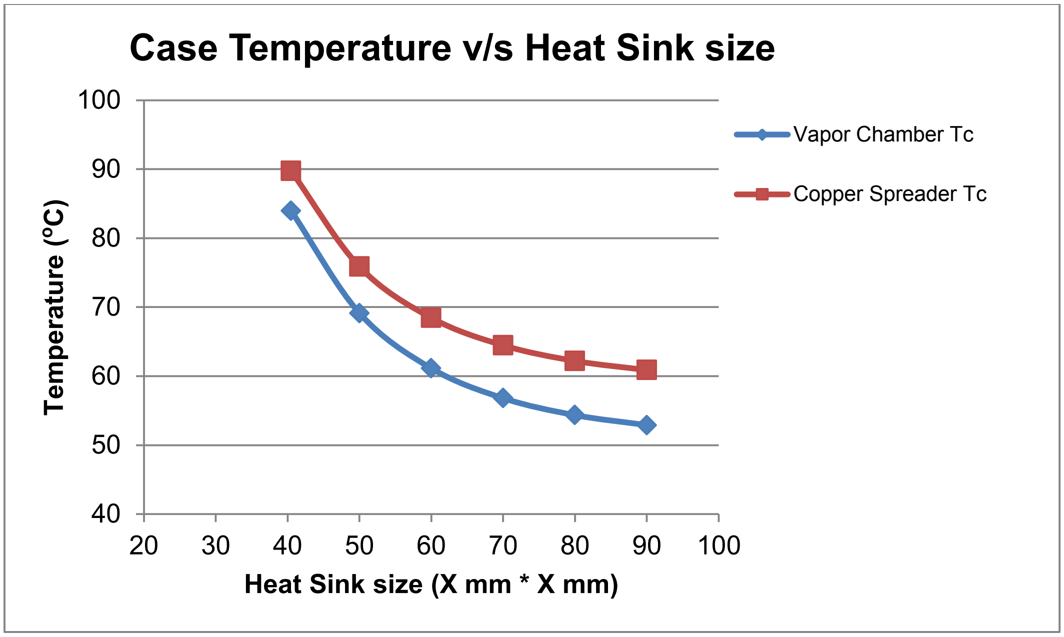

Figure 5-8 Variation in case temperature with heat sink size

Figure 5-7 and Figure 5-8 show junction and case temperature variations respectively with heat sink sizes. It can be clearly seen that as heat sink size increases, the junction as well as case temperature decreases, for both cases. Hence it is advisable to use heat sink sizes at least 1.5 times the spreader size Furthermore, the temperatures for vapor chamber are lower than copper spreader. In other words, we can say that size of heat sink required to maintain junction and case temperature within required limits is more if a copper spreader is used instead of vapor chamber.

Chapter 6

Conclusion

6.1 Summary

A Computational Fluid Dynamics (CFD) model of vapor chamber was developed and its thermal performance was compared to that of copper spreaders. It was observed that vapor chambers achieved 17% better performance than copper spreaders in terms of overall thermal resistance. This is mainly because of 2-phase process in vapor chambers which significantly increases rate of heat transfer. However, one interesting observation was that copper spreaders performed better at smaller spreader width, less than 15 mm x 15 mm in size. The results of baseline case were compared with Intel's TTV data for the 6th Gen Core i7 processor, and they were seen to agree well with the computational model.

A parametric study was carried out to examine the effect of different parameters on the thermal behavior of both vapor chamber and copper spreader. Based on results, the following conclusions have been drawn.

- The junction temperature increases more rapidly with increase in power input for copper spreader case than vapor chambers
- At small spreader widths, copper spreader out-performs vapor chamber whereas at large spreader widths, vapor chamber shows considerably better performance than copper spreader
- Overall thermal resistance of the package is predominantly affected by the wick structure rather than the working fluid
- Vapor chambers perform better in situations where heat transfer coefficient is very low, as in case of air-cooled heat sinks, because of its heat spreading ability

- Vapor chambers can help reduce size of heat sink required to achieve same degree of cooling as possible if copper spreaders were used

6.2 Future work

Further work would include design of experimental setup to perform experimentation to compare real-life data with the computational data obtained in this study. A more detailed 2-phase computational modeling is suggested to more accurately simulate phase change process within the wick structure and vapor space. New types of wick structures need to be developed which would have better performance in terms of capillary limits.

References

[1] David Reay, Peter Kew, *Heat Pipes: Theory, Design and Applications*, Butterworth–Heinemann Publications, Fifth edition 2006

[2] Maxwell J.C., *A Treatise on Electricity and Magnetism*, Vol. 1, 3rd edition, reprinted by Dover, New York, 1954

[3] RL Gorring, SW Churchill, Thermal conductivity of heterogeneous materials. Chemical Engineering Progress, Vol. 57, No.7, July 1961

[4] H Kozai, H Imura, K Takashima, Effective thermal conductivity of screen wicks. Proc. 3rd Internat. Heat Pipe Symposium, Tsukuba, 1989

[5] Vadakkan U., Chrysler G., and Sane S., 2005, "Silicon/Water Vapor Chamber as Heat Spreaders for Microelectronic Packages", IEEE SEMI-THERM Symposium, pp. 182-186, Mar 2005.

[6] Prasher R., "A Simplified Conduction Based Modeling Scheme for Design Sensitivity Study of Thermal Solution Utilizing Heat Pipe and Vapor Chamber Technology", J. Electronic Packaging, Vol. 125, pp. 378-385, 2003.

[7] Minhua Lu, Larry Mok, R.J. Bezama, A Graphite Foam Based Vapor Chamber for Chip Heat Spreading

[8] *Wikipedia:* https://en.wikipedia.org/wiki/Computational_fluid_dynamics

[9] ANSYS Icepak User's Guide Release 15.0, November 2013

[10] ANSYS Icepak Tutorials Release 15.0, November 2013

[11] 6th Generation Intel® Processor Datasheet for S-Platforms

Biographical Information

Dhanraj Patil received his Bachelor's degree in Mechanical Engineering (B.E) from the University of Pune, India, in 2012, before joining the Master's Degree (M.S) program in Mechanical Engineering at the University of Texas at Arlington, in Fall 2013. He has been actively involved in research work at the Electronics MEMS & Nano-electronics Systems Packaging Center (EMNSPC) lab at UTA. His academic career demonstrated his deep interest in thermal sciences, and he developed a keen interest in thermal management of electronics while working in a team headed by Dr. Dereje Agonafer. After his graduation, Dhanraj plans to pursue his career in the field of Mechanical Engineering and is open to challenging opportunities along the lines of his interests.