

COMPUTATIONAL STUDY OF IMPACT OF THERMAL SHADOWING
IN NON-DIRECTED FLOW FOR AIR AND OIL COOLED SERVERS.
FORM FACTOR STUDY AND CUSTOMIZATION OF HEAT SINK FOR
DATA CENTER APPLICATION

by

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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 2016

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Acknowledgements

I would like to utilize this opportunity to thank my research guide Dr. Dereje Agonafer for his constant support, continuous guidance and encouragement over one and half years with my research. I would also thank him to provide me to take critical decisions and allow me to attend conferences and present my work.

I would like to thank Dr. Ratan Kumar and Dr. Haji-Sheikh for taking their valuable time to serve on my thesis committee. Also, I would like to thank Ms. Sally Thompson and Ms. Debi Barton for their invaluable support and timely inputs in various educational matters.

I am obliged to thank Tejas Bhongale for sharing his expertise and inputs on critical points. I would like to thank all my friends in the EMNSPC team and in the University for helping me and supporting me throughout my research.

My special thanks to my parents Mr. Hemant Bhatt and Mrs. Reena Bhatt, my sister Richa, my uncle Mr. Mayur Trivedi for serving my motivation and support during the hard time. I am completely obliged to them for giving me a freedom to peruse my goals in life. I would like to acknowledge my almighty Lord Swaminaryan and Hariprasad Swamiji who have been constantly showing me the right direction throughout my life. They have provided me the courage and inspiration to fight against tough situations and it was impossible to complete without them.

December 07,2016

Abstract

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The University of Texas at Arlington, 2016

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In today's networking world, the usage of Servers and Data centers has been increasing significantly with a corresponding increase in power. The data center energy efficiency largely depends on thermal management of servers. Currently, air cooling is the most widely used thermal management technology in data centers. However, air cooling is starting to reach its limits due to high powered microprocessors and packaging. To overcome these limitations of air cooling in data centers, liquid cooling methods are starting to gain more traction. Thermal shadowing is the effect in which temperature of a cooling medium increases by carrying heat from one server and results in decreasing its heat carrying capacity due to a reduction in the temperature difference between the maximum junction temperature of successive heat sinks and incoming fluid. Thermal Shadowing is a challenge for both

air and low velocity oil flow cooling and as such, both air and low velocity dielectric flow cooling technologies need to be optimized to get high energy efficiency. Low velocity oil flow cooling can be an effective technology for high thermal loads and high heat capacitance.

Form factor study of 3rd generation server is another area of research in which impact of form factor on a maximum junction temperature and thermal resistance at the server level is documented. This work is to provide an insight to increase the rack density by reducing form factor of an existing server. This work can open up to more heat load per rack. The heat sink is a critical part for cooling effectiveness at server level. This work is to provide an efficient range of operation for heat sink with numerical and computational modelling of third generation open compute server. Optimization of heat sink can allow to cool high power density servers effectively. A parametric study is conducted and the thermal efficiency has been optimized.

Table of Contents

Acknowledgements.....	iii
Abstract.....	iv
List of Illustrations	viii
List of Tables	x
Chapter 1 INTRODUCTION.....	1
1.1 Introduction to Data Center	1
1.2 Data Center Cooling Methods	2
1.3 Motivation of the work	7
Chapter 2 SPECIFICATION OF THE SERVER UNDER STUDY	8
2.1 Server Description	8
2.2 Base line server specifications	10
2.3 Baseline Oil Cooled Server Specification	13
Chapter 3 IMPACT OF THERMAL SHADOWING.....	15
3.1 Thermal Shadowing	15
3.2 Validation and grid independent study of an air cooled server	17
3.3 Assumptions and validation of an oil cooled server	20
3.4 Thermal resistance calculation.....	23
Chapter 4 FORM FACTOR STUDY	26
4.1 Definition of Form Factor.....	26
4.2 Parametric Study of Form Factors	27
Chapter 5 APPLICATION AND OPTIMIZATION OF HEAT SINKS	28

5.1	Type of Heat Sinks	28
5.2	Modification of Baseline Server	30
Chapter 6 RESULTS.....		33
6.1	Impact of Thermal Shadowing in Air and Oil Cooled Server	33
6.2	Form Factor Study	36
6.2.1	Thermal performance of 2U server.....	36
6.2.2	Thermal performance of 1.5U server.....	38
6.2.3	Thermal performance of 1U server.....	39
6.3	Heat sink optimization	41
6.3.1	Pin fin heat sink optimization	41
6.3.2	Plate fin heat sink optimization.....	42
Chapter 7 CONCLUSION		44
Chapter 8 FUTURE SCOPE.....		46
References.....		47
Biographical Information.....		48

List of Illustrations

Figure 1.1 Hot aisle/ Cold aisle Approach.....	3
Figure 1.2 Cold plate top view.....	4
Figure 1.3 Oil immersion cooling	6
Figure 2.1 Open compute server top view	9
Figure 2.2 Side view of baseline server	10
Figure 2.3 CFD modeling of baseline server	10
Figure 2.4 Baseline duct	11
Figure 2.5 Baseline duct dimensions	11
Figure 2.6 Experimental setup of an air cooled server	12
Figure 2.7 Experimental setup of an oil cooled server	13
Figure 2.8 Baseline CFD model of an oil cooled server.....	14
Figure 3.1 Thermal shadowing phenomenon	15
Figure 3.2 ASHRE defined recommended zones for data center cooling	18
Figure 3.3 Grid independent study.....	20
Figure 3.4 Heat sink dimension	22
Figure 3.5 Calculation of thermal resistance	24
Figure 4.1 Rack unit comparision.....	26

Figure 5.1 Types of heat sinks	28
Figure 5.2 Extruded cut heat sinks.....	30
Figure 5.3 Pin fin heat sinks	31
Figure 5.4 Grid independent study for plate fin and pin fin heat sink	31
Figure 6.1 Thermal resistance VS Inlet air velocity	33
Figure 6.2 Maximum junction temp. VS Inlet air velocity.....	33
Figure 6.3 Partical trace of temperature in non-directed air cooled server.....	34
Figure 6.4 Thermal resistance VS Oil flow rate	35
Figure 6.4 Thermal resistance VS Oil flow rate	35
Figure 6.5 Maximum junction temp. VS Oil flow rate.....	35
Figure 6.6 Thermal resistance of 2U server	36
Figure 6.7 Maximum junction temperature of 2U server	37
Figure 6.8 Thermal resistance of 1.5U server	38
Figure 6.9 Maximum junction temperature of 1.5U server	38
Figure 6.10 Thermal resistance of 1U server	39
Figure 6.11 Maximum junction temperature of 1U server	40
Figure 6.12 Thermal resistance VS Pin radius	41
Figure 6.13 Thermal resistance VS Pin height	41
Figure 6.14 Thermal resistance VS Fin thickness	42
Figure 6.15 Thermal resistance VS Plate height	42

List of Tables

Table 1.1 Physical properties of heat transfer fluids.....	5
Table 2.1 Open compute server specifications	8
Table 3.1 Pressure drop of the CFD model.....	19
Table 3.2 Validation of an oil cooled CFD model.....	23

Chapter 1

INTRODUCTION

1.1 Introduction to Data Center

Data Center is the part of the industry, which facilitates to store, process and manage the important data by housing the servers mounted in racks. In today's networking world utilization of servers and data center has been increasing significantly. Data Center is the part of the industry, which facilitates to store, process and manage the important data by housing the servers mounted in racks. In today's networking world utilization of servers and data center has been increasing significantly. Increasing demands of processing and storage of data causes corresponding increase in power density of servers. Computer system dependency has been increased and that has encouraged the rapid growth of data centers in leading business units like banking, education, transportation, social media and many more. Data center unit is one of the leading power consumption sectors of an industry. To fulfill the demands of data storage and data processing, corresponding increase in power density of servers are needed. Computer system dependency has been increased and that has encouraged the rapid growth of data centers in leading business units like banking, education, transportation, social media and many more. Data center unit is one of the leading power consumption sectors of an industry.

As per an article by Jim Witkin, which was written in 2012 about 1.5 percent of all electricity generated worldwide goes to power datacenters. The

attendant greenhouse gas emissions, some 188 million tons of carbon dioxide per year, match the emissions of about 33 million passenger vehicles [1]. Increased demand of data centers affects the environment that triggers the engineer to develop efficient datacenter technology. Increase in the power density of the servers directly increases the amount of heat generation. To ensure the reliability and efficient working characteristics of the server component, servers must be cooled continuously and efficiently. The data center energy efficiency largely depends on thermal management of servers.

1.2 Data Center Cooling Methods

At present, there are various methods of data center cooling used for maintaining data center facility under permissible temperature range. Two major cooling techniques, which are as follow.

1. Air cooled servers
2. Liquid cooled servers

1. Air Cooled servers

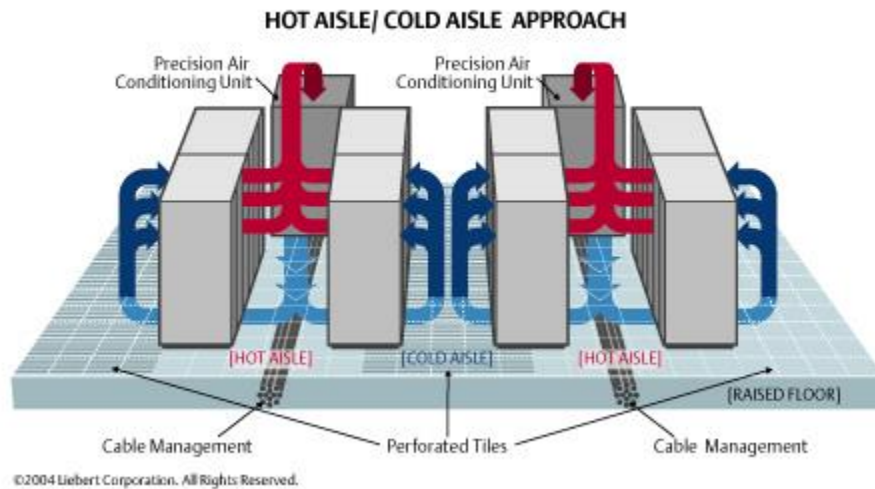


Figure 1.1 Hot aisle/ Cold aisle Approach

Air cooled configuration is the most widely used cooling technique for data center applications. Air cooled servers work on forced convection of air over heat transfer component called a heat sink. The Cooling system of an air-cooled server consists of axial fans to handle the air volume and heat sinks to increase the heat transfer. Basically, heat sinks are the component that transfers the heat from heat generated by the processor to ambient air through three modes of heat transfer that is conduction, convection and radiation. Fans are used to manage the airflow that can carry the heat from the server component from inlet to outlet. Fans are controlled according to the change in the temperature of the server component. Exhaust hot air from the server outlet enters to the hot aisle and then will be directed to the Computer Room Air Conditioning (CRAC) unit to cool the air up to the required temperature. That cooled air will be supplied back to the

cold aisle of the cooling system. Design of air cooled data center room is shown in the Figure 1.1 [2].

2. Liquid Cooled server

Currently, air cooling is the most widely used thermal management technology in data centers. However, air cooling is starting to reach its limits due to high powered microprocessors and packaging. Moreover, air cooled servers consume a larger space as it must house the ducting system and fans. To overcome these limitations of air cooling in data centers, liquid cooling methods are starting to gain more traction. A liquid such as water or oil will be used as a cooling medium instead of air. Liquid cooled servers are subdivided into two categories depending upon the cooling fluid used.

- a. Water cooled servers
- b. Oil cooled servers
- a. Water Cooled Server

Water cooling technique is an indirect heat transfer technique as we cannot allow water to get in touch with server components. Cold plate is used to transfer the heat generated by processor to water.

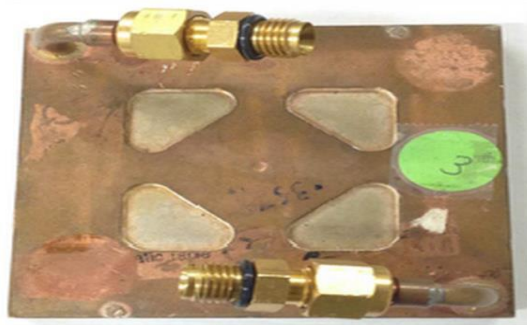


Figure 1.2 Cold plate top view

As shown in the figure cold plate was introduced as a passive heat transfer device. Cold plate has a bottom surface made of copper and is placed on the processor. The gap between the bottom surface of cold plate and the processor is filled with heat spreading material. Heat generated by the processor is conducted to the cold plate which increases the temperature of the copper plate. Heat will be transmitted to the water flowing into the cold plate through convection. Water is then directed to the chillers so that water can be cooled down for the next cooling cycle. The heat sink is replaced with the cold plate in the server. The advantage of using a cold plate compared to heat sink is it requires less space so that it allows the designer to make server design compact. Moreover, the cold plate configuration requires less power compared to air cooling system. Several research is going on to optimize the current cold plate and develop the concept of dynamic cold plate. [9]

b. Oil Cooled Serve

Type of Fluid	Heat Capacity (kJ/kg K)	Heat Conductivity (W/m K)	Kinematic Viscosity ($\times 10^{-6} m^2/s$)	Density (Kg/m^3)
Air	1.01	0.02	0.016	1.225
Water	4.19	0.58	0.66	1000
Oil	1.67	0.13	16.02	849.3

Table 1-1 Physical properties of heat transfer fluid

As shown in the Table 1-1 thermal mass (Density \times Heat capacity) of oil is higher than the air. Thermal mass carried by water is the highest but as it is used as a passive heat transfer medium, it reduces its efficiency of heat dissipation. Oil cooled servers further subdivided according to fluid motion.

a. Oil immersion cooling

In Oil immersion cooling, servers are immersed in oil and heat is carried by the oil on constant basis. Warm oil will be replaced with cold oil by the time interval.



Figure 1.3 Oil Immersion cooling

b. Low velocity oil flow cooling

In low velocity oil flow cooling, will be forced and forced convection takes place over the heat sink. The pump is used to provide the driving force to the oil. Flow meter is used to restrict the flow rate in recommended limit.

1.3 Motivation of the work

Very small saving in power consumption at server level can be converted into significant amounts at facility and infrastructure level of data center industry. To fulfill the cooling demand of high power servers, oil cooling is one of the emerging techniques. It has been reported that “Thermal shadowing” is an important phenomenon that has significant impact on cooling efficiency for air cooled servers in previous research. [3] It is very important to analyze the impact of thermal shadowing in oil flow cooling. There are very few studies which consider impact of thermal shadowing in oil flow cooling and also compare the impact of thermal shadowing between oil cooled and air cooled servers. This research includes computational study of impact of thermal shadowing, form factor study and geometrical optimization of heat sink for oil cooling applications. The main objective is to reduce power consumption compared to air cooled server and cost savings through reduction in material cost of heat sinks.

Chapter 2

SPECIFICATION OF THE SERVER UNDER STUDY

2.1 Server Description

The server taken under study is an Intel based third generation open compute server. [4] It comprises two microprocessors having a design power density of 115 W each. As shown in the figure server houses four DIM block and each having four DIMMS of 8GB memory capacity. Intel based server has the form factor 2U open rack unit. It has dimensions of 807mm×171mm×89 mm.

CPU 1	115 W
CPU 2	115 W
Chassis foot print	520 × 171 × 89 mm
CPU footprint	50 × 50 mm

Table 2.1 Open compute server specifications

The server contains two fans as it is originally designed for air cooling. The fan has the dimensions as 80mm×80mm×38mm. The speed of the fan is controlled by

the change in CPU die temperature. Heat sinks are mounted on the CPU to transfer the heat from processor to air.

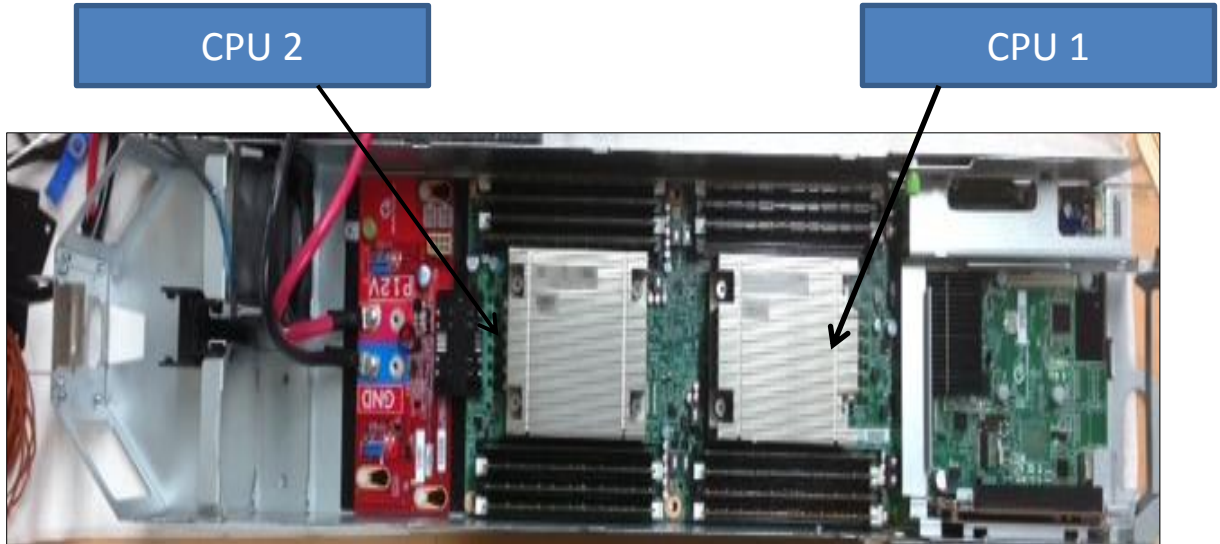


Figure 2.1 Open compute server top view

It can be described that the CPU 2 will remain in the thermal shadow of the CPU 1. Server is enclosed with the top cover mounted on the chassis body.

2.2 Base line server specifications

Baseline server has the same dimension except it had the power density of 95W for each CPU instead of 115W. Ducting system is designed to fit on the top

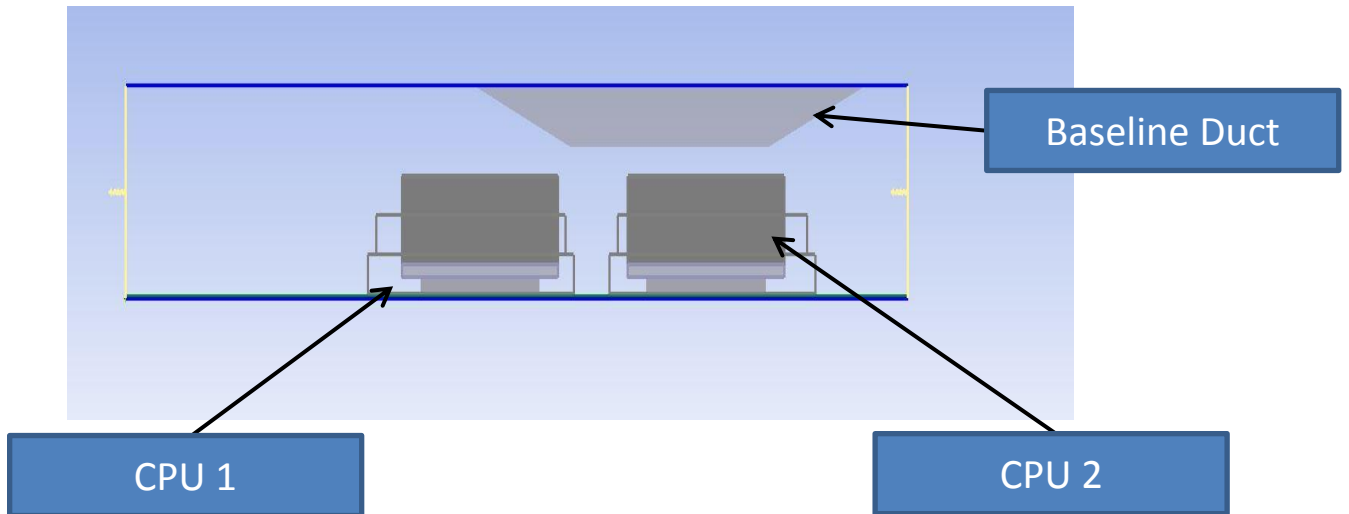


Figure 2.2 Side view of baseline server

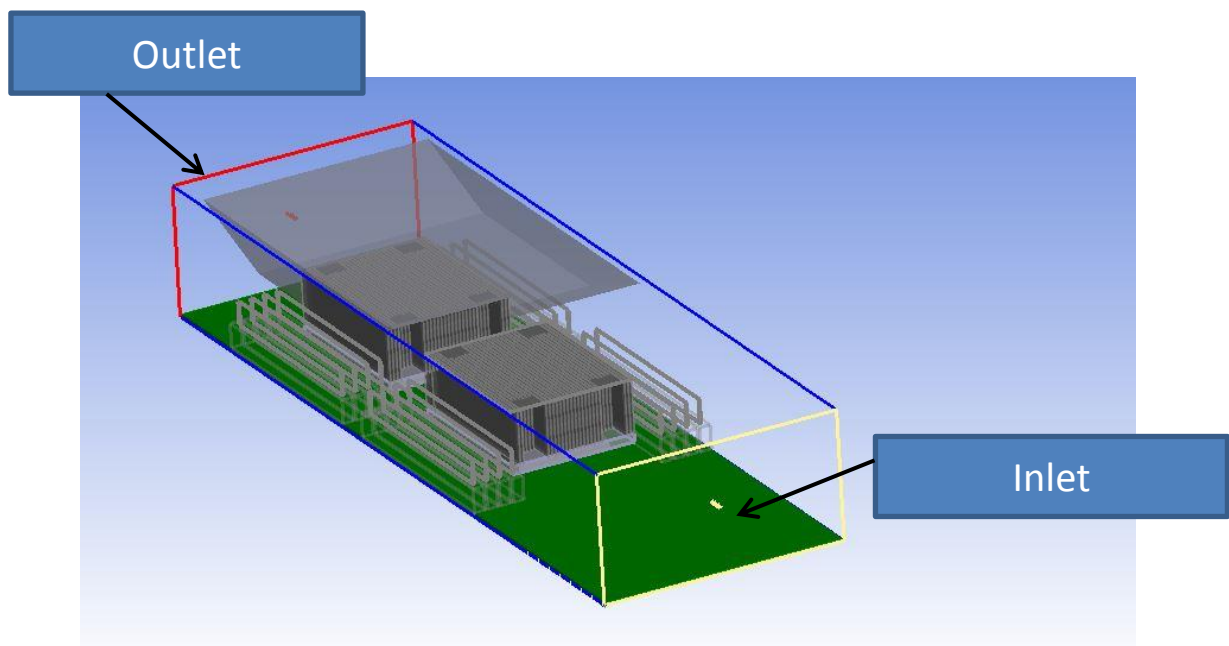


Figure 2.3 CFD Modeling of baseline server

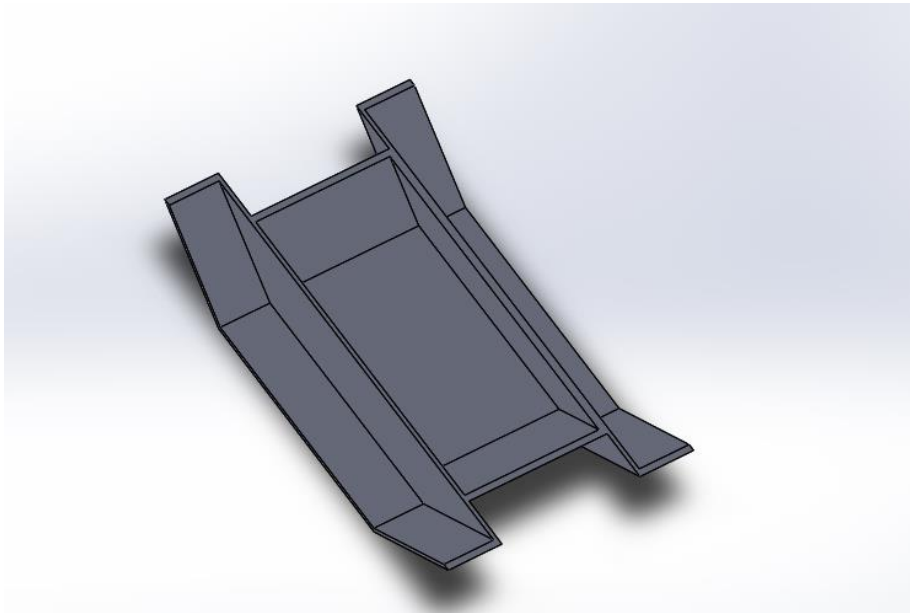


Figure 2.4 Baseline duct

The CFD model has been developed using the computational tool called Ansys ICEPAK 17.0 and duct geometry is imported from modeling tool SolidWorks.

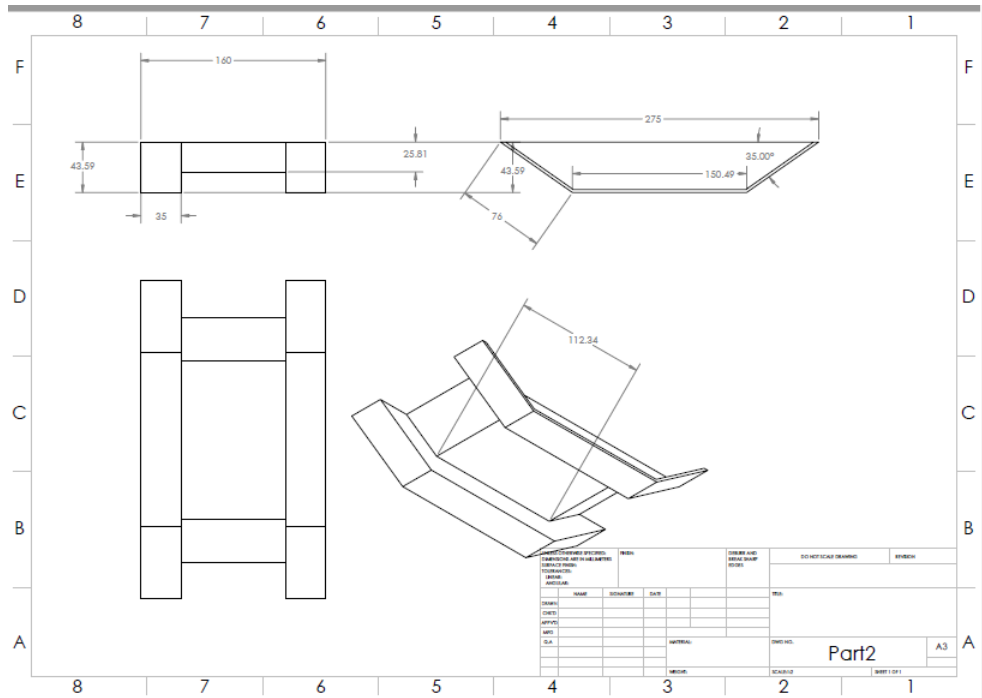


Figure 2.5 Baseline duct dimensions

Figure 2.5 shows the detail dimension of baseline duct. It is made of aluminum. As shown in Figure 2.3 one side of the server cross-section is considered as air inlet and the opposite side is selected as air outlet. Experiment on air cooling was performed by previous Master student Divya Mani. She had connected the server inlet with airflow bench. The desired air quantity is supplied using airflow bench and data of pressure drop and temperature had been documented. Figure 2.6 shows the experimental setup of an air-cooled server along side with an air flow bench. Thermocouples are placed at critical positions inside the server and are connected with DAQ units. [3]

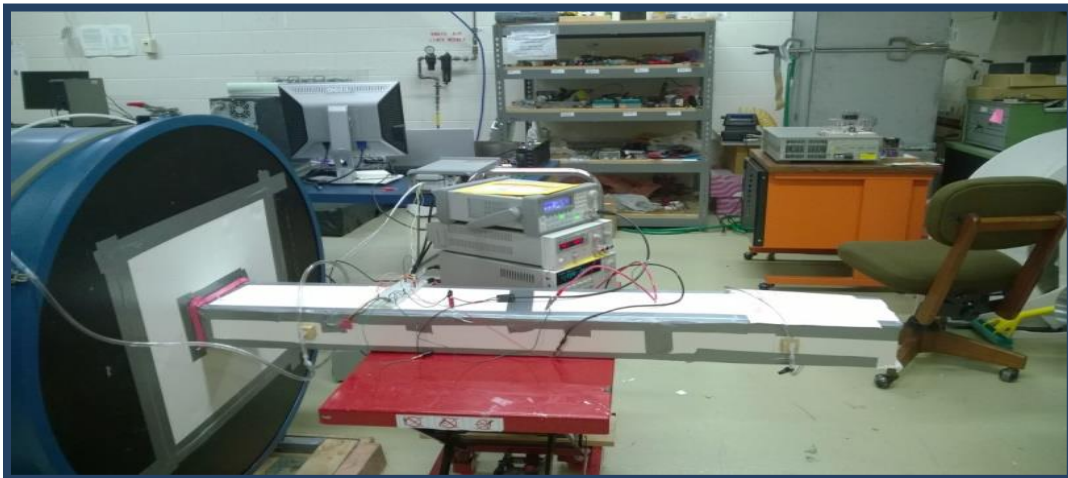


Figure 2.6 Experimental setup of an air-cooled server

The baseline single server had been tested at various inlet velocity and varying the processor usage.

2.3 Baseline Oil Cooled Server Specification

An oil cooled server has the same specifications as an air-cooled server with some updates. For study of an oil cooled server, processor has the power density of 115W each. For observing the behavior and flow patterns of an oil, aluminum chassis was replaced with transparent acrylic ceiling. An oil cooled system does not require fans and ducting system; hence they were removed for an experiment. Pumps are used to maintain the flow of a white mineral oil. Flow meter is inserted in the oil flow path to maintain the oil flow.

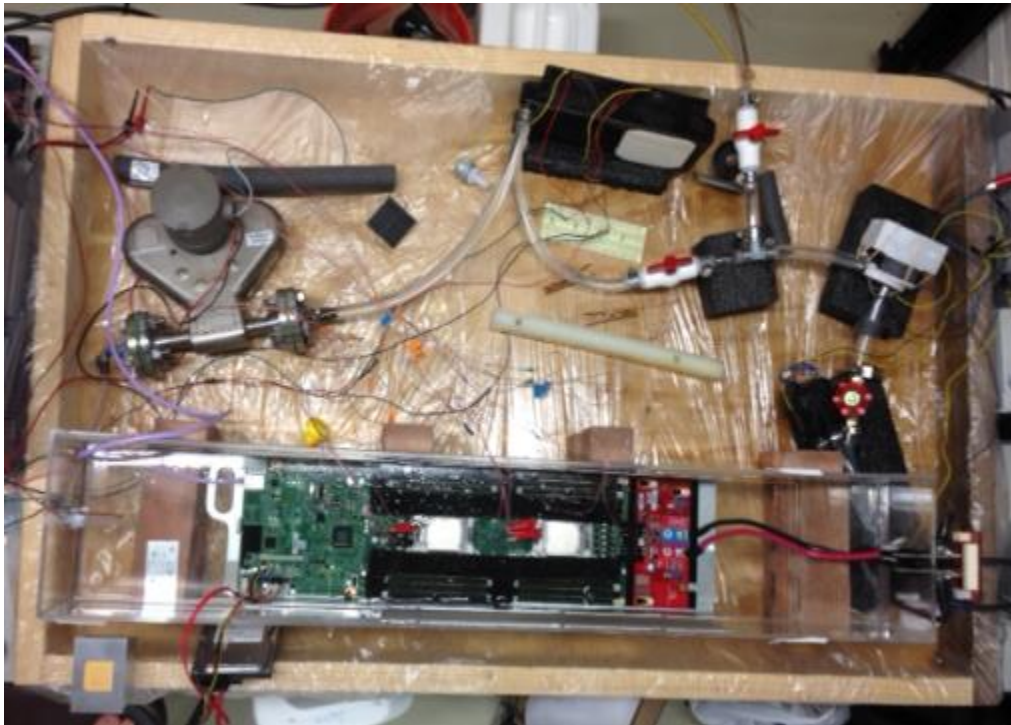


Figure 2.7 Experimental setup of an oil cooled server

Figure 2.7 [5] is an experimental setup of an oil cooled server. Thermocouples are placed near the CPU block to measure the maximum junction temperature of the

server. Oil was flown at one end of the server and hot oil is taken out at the other end of the server. The experiment was performed using the cooling medium as white mineral oil by previous master student Trevor Mc.Williams and thermal resistance and temperature data had been documented. [5]

Baseline oil cooled server has been developed using a computational tool as Ansys ICEPAK. The fans and ducting system has been removed. Server dimensions are kept similar as an air-cooled server except the height of the ceiling is kept as two fin space above the height of the plate fin heat sink.

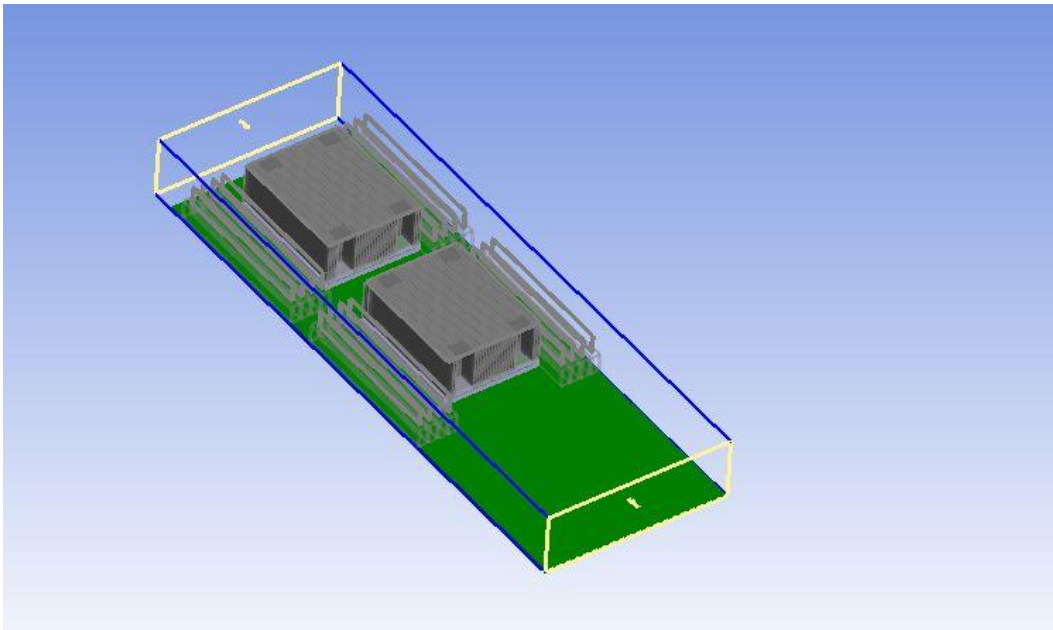


Figure 2.8 Baseline CFD model of an oil cooled server

Chapter 3

IMPACT OF THERMAL SHADOWING

3.1 Thermal Shadowing

Thermal shadowing is the phenomenon in which temperature of a cooling medium increases by carrying heat from one power source and results in decreasing its heat carrying capacity due to a reduction in the temperature difference between the maximum junction temperature of successive heat sinks and incoming fluid. Thermal shadowing causes the localized increase in temperature of an object (heat sink and CPU) that stays in thermal shadow of the other object.

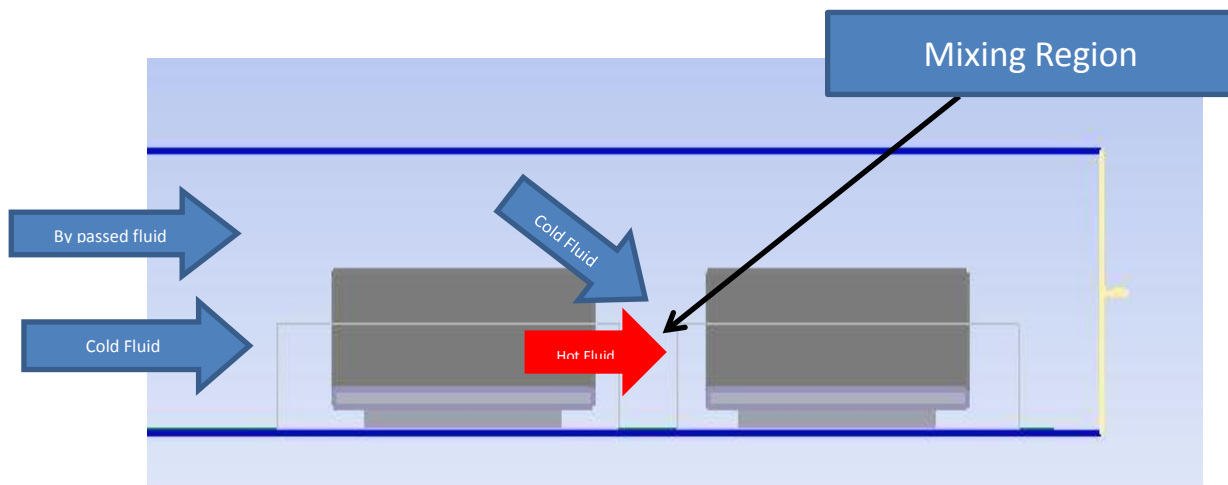


Figure 3.1 Thermal shadowing phenomenon

Figure 3.1 illustrates the concept of thermal shadowing. The fluid enters from one end of the server; some amount of fluid encounters the first heat sink and carries the heat. That increases the temperature of the cold fluid. Some amount of fluid will bypass the first heat sink. The tip clearance and span wise

spacing are the major factors of server design, which govern the flow bypass around heat sink. Tip clearance is the space between maximum heat sink height and the top of the server ceiling. The hot fluid that carries the heat from first heat sink mixes with the bypass cold fluid. The mixed fluid stream then enters to the second heat sink. Due to increased inlet temperature of fluid for second heat sink, reduces the temperature difference with downstream temperature and that also reduces heat transfer. Basically, it causes a localized increase of junction temperature of an object which stays in thermal shadow.

This phenomenon can be understood using energy balance equation.

$$M_1h_1 + M_2h_2 = M_3h_3 \dots\dots\dots (1)$$

Where

- | | |
|-----------------------------|------------------------------|
| M1 = mass of cold fluid | M3 = mass of mixed fluid |
| h1 = enthalpy of cold fluid | h3 = enthalpy of mixed fluid |
| M2 = mass of hot fluid | |
| h2 = enthalpy of hot fluid | |

To find the temperature of mixed air that enters in the second heat sink, enthalpy will be taken as product of heat capacity and temperature.

Hence, eq. (1) can be written as

$$M_1C_pT_1 + M_2C_pT_2 = M_3C_pT_3;$$

$$T_3 = \frac{M_1T_1 + M_2T_2}{M_3}$$

The temperature after mixing of fluid stream is T_3 . Temperature of mixed fluid stream T_3 is higher than the inlet fluid temperature ($T_3 > T_1$). To reduce the

impact of thermal shadowing ducting system has been designed so that it can direct the bypass flow towards the shadowed object and minimize the impact of thermal shadowing. To compare the impact of thermal shadowing, non-directed flow is considered in the study. This research has been done to analyze the various parameters like maximum junction temperature, thermal resistance and inlet flow velocity for a system without ducting. From comparative study between air and oil cooled server one can get to know that which parameters can affect the cooling system design and power consumption.

3.2 Validation and grid independent study of an air-cooled server

Validation of the CFD model with actual experimental data and previously developed CFD model in other computational tool is mandatory for an accurate and precise results of future simulations. As mentioned earlier, Ansys ICEPAK has been used as a computational analysis tool for this study. To validate the model boundary condition is very important. Boundary condition should be kept same as used for an experiment.

Boundary condition used for validation as inlet air temperature ($T=24.5^{\circ}\text{C}$) and relative humidity is in range of ASHRE defined recommended range. Figure 3.2 shows the allowable and recommended zones for air cooling method. [10]

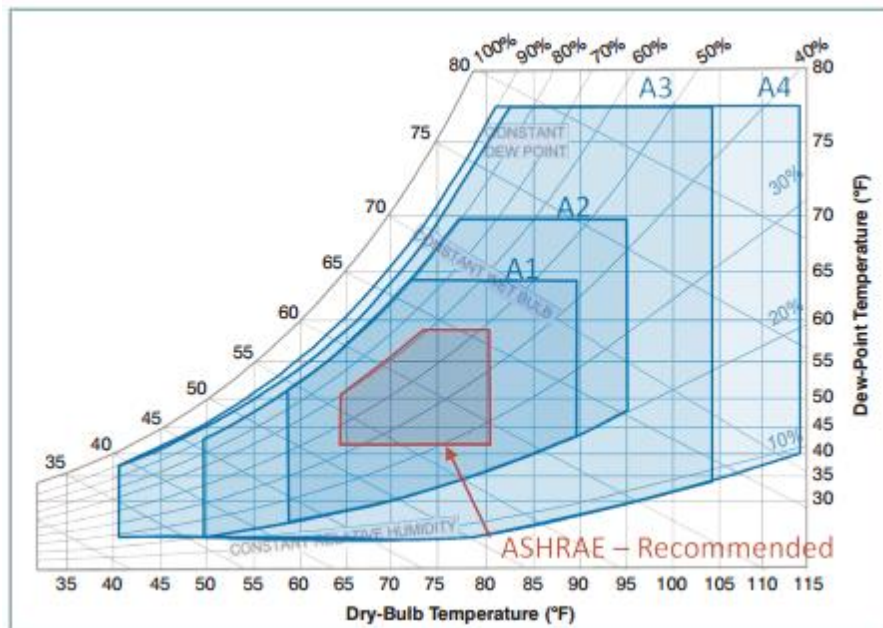


Figure 3.2 ASHRE defined recommended zones for data center cooling

Temperature and pressure drop has been obtained keeping the same boundary condition used for an experiment and compared with the previously documented CFD results. [3] For validation purpose power density of each server is kept as 95W and base line ducting system has been incorporated.

Once the boundary condition is applied and exact environment has been created, the next and very important stage of the process is simulations. ICEPAK basically runs the solver according to boundary condition and solves the Navier-Stokes equation on each node and up to predefined number of equations. Here, for an air-cooled server CFM had been varied from 0 to 100 with an interval of 20. [3] TO decide that which model is used for the simulation process, Reynolds number is very important dimensionless number. For an air-cooled server, it

came out as $Re \geq 4000$ for selected inlet CFM range. [3] To justify the condition, turbulent model with zero equation has been used. Pressure drop has been noted and compared with the previous results. It has come out with the maximum of $\pm 10\%$ error with the actual results.

Flow rate (CFM)	Previous CFD Pressure Drop (in/H₂O)	ICEPAK CFD Pressure Drop (in/H₂O)	Error Percentage
0	0	0	0
20	0.034	0.038	10.52
40	0.106	0.104	-1.88
60	0.214	0.218	1.834
100	0.549	0.556	1.3

Table 3.1 Pressure drop of the CFD model

For the accuracy of the model grid independent study has been carried out for an air-cooled server. For the grid independent study, processor power is kept as 95W each, inlet air temperature is taken as 24.5° C and inlet air velocity is kept constant as 1m/s.

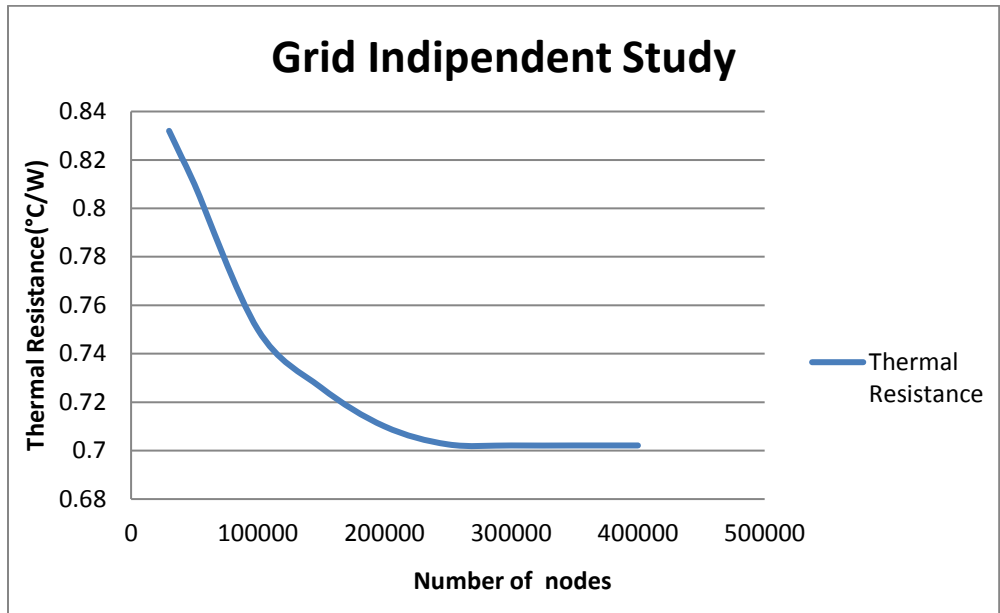


Figure 3.3 Grid independent study

It can be commented from the Figure 3.3 that thermal resistance decreases as we increase the number of nodes in the server and there isn't major variation after number of nodes exceeds 250000. For the rest of the research, total number of nodes is taken in the range of 250000 to 400000.

3.3 Assumptions and validation of an oil cooled server

To validate the oil cooled server, some material properties need to be specified. The material property of the server component will remain same as an air-cooled server but fluid properties have been changed as white mineral oil is used in place of an air. There are different types of mineral oil can be used, Novec is one of them. Novec oil has some limitations and because of that white mineral oil has been used. First, it has boiling temperature ranges from 34°C to 49°C hence for high temperature cooling applications the Novec oil has the tendency of two

phase cooling. From the material cost point of view, Novec oil has higher price compare to white mineral oil. These two drawbacks are eliminated in using of white mineral oil. In order to take above mentioned factors, white mineral oil has been used for the rest of the study.

There are some physical properties of white mineral oil that should be considered for computational analysis.

- Density – 849.3 Kg/m³
- Thermal conductivity – 0.13 W/m K
- Specific heat – 1680 J/kg k
- Thermal Diffusivity – 9.166E-8 $\frac{m^2}{s}$
- Over all heat transfer co-efficient – 50-30 W/m² K

Apart from the above mentioned physical properties, white mineral oil has another very important physical property called dynamic viscosity. Dynamic viscosity of white mineral oil is temperature dependent property. It varies with change in the temperature. Change in dynamic viscosity can be obtained using the empirical co-relation. [6]

$$\mu = C_1 \times \text{Exp} \left[\frac{2797.3}{T + 273.2} \right]$$

Where,

μ = the dynamic viscosity in centipoise

T = the temperature in °C

C₁=coefficient for scaling

(a value of 0.001383 was determined using the 40°C value from STE Oil data sheets as a reference point) [6]

It is very important to make the decision of the solver model. To decide that Reynold's number must be calculated and based on that decision will be taken.

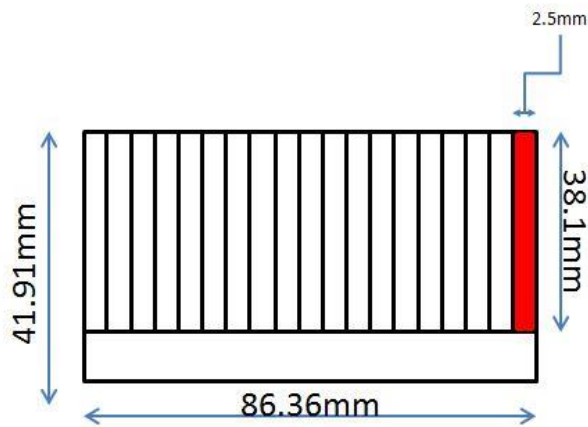


Figure 3.4 Heat sink dimension

Volume flow rate = Area of inlet × Velocity

$$Re = \frac{\rho V D}{\mu} \dots\dots\dots (3.1)$$

$$Re = 256$$

Where,

$$D = \frac{2bh}{b+h}$$

D= Hydraulic Diameter

b= channel width

h= fin height

V= velocity of fluid

$$\begin{aligned} \text{Pumping power} &= \text{pressure drop} \times \text{mass flow rate} \\ &= 1.94 \text{ W} \end{aligned}$$

Flow condition is kept same that had been used for an actual experiment. Inlet oil temperature is taken as 30°C, volume flow rate is kept constant of 1 lpm, by using the server cross sectional area and volume flow rate, velocity of an incoming oil is calculated. Server cross section in this case is taken as (171mm × 55mm), gauge pressure has the value of 6 psi.

Using the equation (3.1) Reynold's number came out as 256. Laminar model is used to solve the naiver stoke equation and get the results. CFD result is then compared with the experimental result documented by Trevor Mc.Williams.[5]

Flow rate(lpm)	experimental results	CFD Results	Error Percentage
0.3	70	71.23	1.72
0.4	68.74	68.86	0.174
0.5	67.43	66.87	-0.83

Table 3.2 Validation of an oil cooled CFD model

3.4 Thermal resistance calculation

Thermal resistance is the key factor for designing a cooling system. It is one of the major parameter that needs to be carefully designed and optimized. Thermal resistance is calculated for the heat transfer components. Thermal resistance can be calculated using the below mentioned formula.

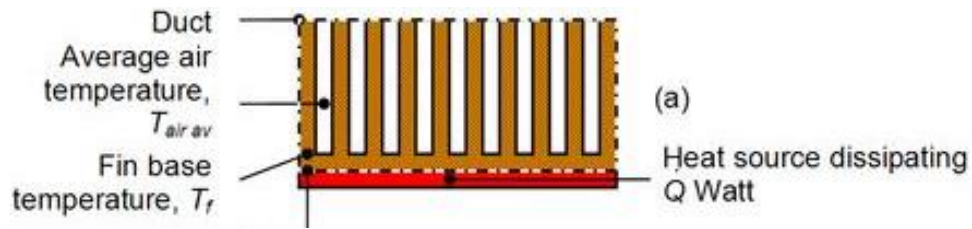


Figure 3.5 Calculation of thermal resistance

$$\text{Thermal Resistance} = \frac{T_j - T_a}{\text{Heat Dissipation}}$$

Where,

T_j = Junction Temperature

T_a = Incoming fluid temperature

Heat dissipation from processor is taken in watts.

Comparison of Thermal Shadowing

Comparison of the impact of thermal shadowing between non-directed flow type air and oil cooled server is important to design the cooling system. For comparison, the boundary condition is kept similar that has been used for validation except the power of the processor is taken as 115W each instead of 95W used for air cooled server.

Keeping inlet fluid temperature constant, inlet fluid velocity has been varied in a specified range. For air cooled server inlet air velocity changes from 1m/s to 1.6m/s with an increment of 0.1 m/s[3] . For white mineral oil inlet

temperature is kept as 30°C and volume flow rate changes from 0.3 lpm to 2.5 lpm.

The maximum junction temperature and thermal resistance are compared for an air cooled and oil cooled servers. Non-directed flow pattern is applied that means no ducting is included for this study.

Chapter 4

FORM FACTOR STUDY

4.1 Definition of Form Factor

Form Factor study of Facebook's third generation open compute server includes comparison of thermal resistance and maximum junction temperature of 2U, 1.5U and 1U for low velocity oil cooled servers. Form factor defines the height of the server. In the data center industry, U is considered as standard rack unit. It has the value of 44.45mm. Purpose of this study is to analyze the change in maximum junction temperature and thermal resistance when form factor of the server is reduced. [8]

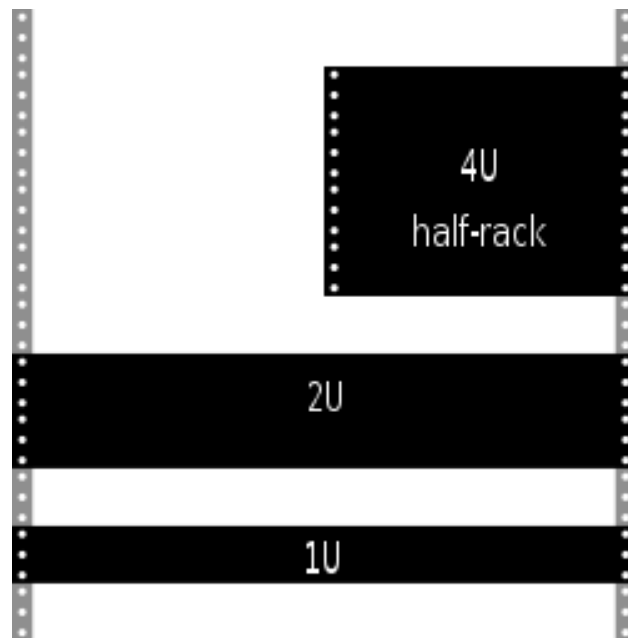


Figure 4.1 Rack unit comparison

4.2 Parametric Study of Form Factors

At first, server is kept same as it is designed. The server is basically designed for air cooling application and it has by default form factor as 2U. Ansys ICEPAK has the provision to change the parameters and solve the Navier-Stokes equations. Two major parameters have been considered are inlet oil temperature and oil flow rate. Inlet oil temperature has been varied from 30°C to 50°C. This range has been selected from the previous research experiment carried out on oil immersion cooling. [7] Volume flow range for an incoming oil is kept from 0.3 lpm to 2.5 lpm. Maximum value of Reynold's number for this study is 656 obtained at the volume flow rate of an oil as 2.5 lpm. It can be concluded that even at maximum volume flow rate of 2.5 lpm Reynold's number does not exceed 2000 hence, laminar model is used for solution of parametric trials.

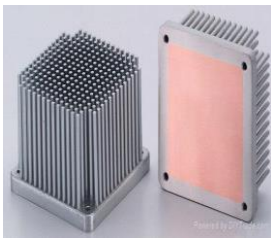
Similar procedure repeated for form factor 1.5U and 1U. The Cabinet height is changed according to the value of form factor. Impact of thermal shadowing increases as we reduce the form factor of the server. The major purpose of the study is to get an insight to increase the existing rack density by reducing its form factor. This study is very useful to predict the behavior of cooling fluid at various form factors. Small improvement at server level cooling can be useful for a significant amount of savings at the facility level.

Chapter 5

APPLICATION AND OPTIMIZATION OF HEAT SINKS

5.1 Type of Heat Sinks

Heat sinks are passive components those are used for heat transfer. Parallel plate heat sink, pin fin heat sink and extruded cut plate fin heat sink are the major three types of the heat sinks available in the market. Actual model with the parallel plate heat sink was studied and validated with the experimental results. [5]



Pin fin heat sink



Parallel plate heat sink



Plate fin heat sink

Figure 5.1 Types of heat sinks

Heat sinks are designed to increase the heat transfer area between heat generating component and the cooling medium. To increase the efficiency of heat sinks, heat spreaders are used between heat source and secondary heat exchanger. Heat spreader is the heat exchanger that transfers the heat with the more favorable surface area and geometry than the source. Heat spreader is made of very high

thermal conductive material so that maximum heat can be transferred. Generally, copper or aluminum is used for heat spreader. Plate type heat spreaders are popular and readily available in the market. Heat sinks are also built from the materials having high thermal conductivity. Copper and aluminum alloys have favorable heat transfer characteristics, including good thermal conductivity and thermal performance. Hence they are most widely used materials for heat sink manufacturing. At present, research work is going on in customization of heat sinks. Nowadays, it is possible to develop the customized heat sinks depending upon your thermal performance and its applications. The performance of parallel plate heat sink has been studied and validated so, this study includes thermal performance and optimization of pin fin and extruded cut plate fin heat sinks for the server under study. TO evaluate the thermal performance of the heat sinks, dimension less numbers like Reynold's number and Nusselt number are obtained. These two numbers are driving parameters to decide the boundary condition and thermal performance of the heat sinks.

To evaluate the thermal performance, Ansys ICEPAK is used. CFD model has been developed for plate and pin fin and then using multiparameter optimization, geometry of the heat sinks has been optimized. In this study, server form factor has been taken as 1U. The flowrate of an oil is kept at 1 lpm and inlet oil temperature is taken as 30°C and mass fix mass flowrate technique is applied.

5.2 Modification of Baseline Server

To develop the CFD model of the server having plate fin and pin fin heat sinks respectively, the base line server design needs to be modified. The parallel plate heat sinks are replaced by pin fin or extruded cut plate fins.

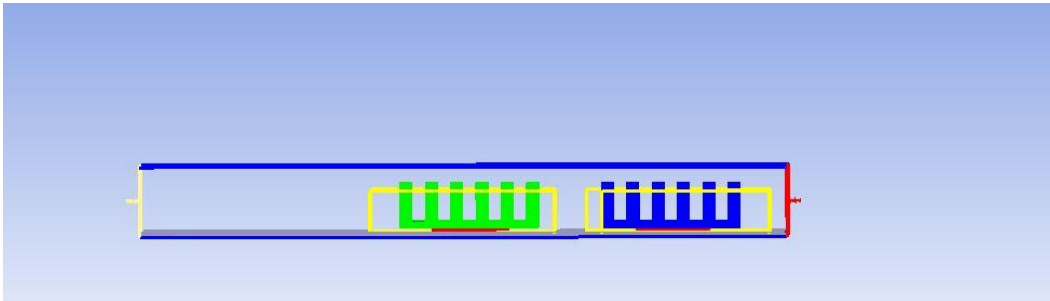


Figure 5.2 Extruded cut heat sinks

Figure 5.2 shows the geometry of extruded cut heat sinks are implemented in place of parallel plate heat sinks. Initially, 25 fins were used to optimize the heat sinks. The power of the sources was kept constant as 115W. The inlet oil flow rate is taken as 1 lpm and inlet oil temperature is kept as 30°C. The parametric study is carried out by varying the fin thickness and fin height. The optimized heat sink has the minimum thermal resistance at applied boundary condition. The results of the parametric study are explained further in detail. The aluminum is used as the heat sink material in this study. The heat generated by the DIMS are neglected because it has very low value of power compare to source power.

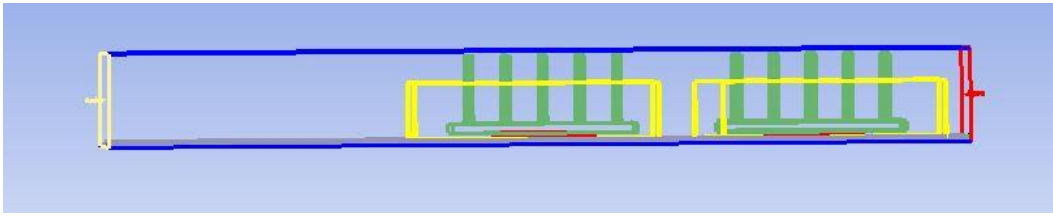


Figure 5.3 Pin fin heat sinks

Figure 5.3 shows the model of pin fin heat sink. Instead of the plate fins, pins of aluminum are implemented. Study of Nusselt number and Reynold's number are included in it. In the geometric optimization of the pin fin heat sink, radius and height of pin fins are studied and optimized for oil flow application for the same boundary condition that has been used for extruded cut fin heat sink. The grid independent study has been carried out for accuracy of the result for optimization.

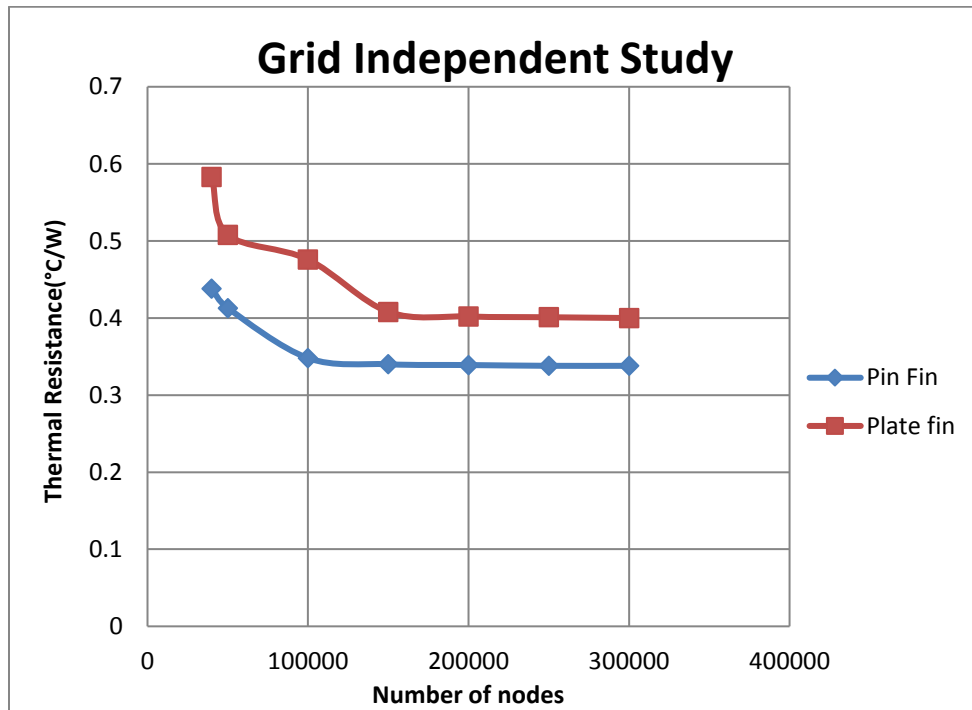


Figure 5.4 Grid independent study for plate fin and pin fin heat sink

Thermal resistance is the driving parameter for the thermal performance of the heat sink. To assure the accuracy of the results, grid independent study is carried out and from Figure 5.4 it can be said that for both plate fin and pin fin heat sinks, thermal resistance remains constant once the total number of nodes reaches to 150000. Total number of the study for the optimization is taken between 150000 to 250000. The inlet oil flow rate is kept as 1 lpm and temperature as 30° C.

Chapter 6

RESULTS

6.1 Impact of Thermal Shadowing in Air and Oil Cooled Server

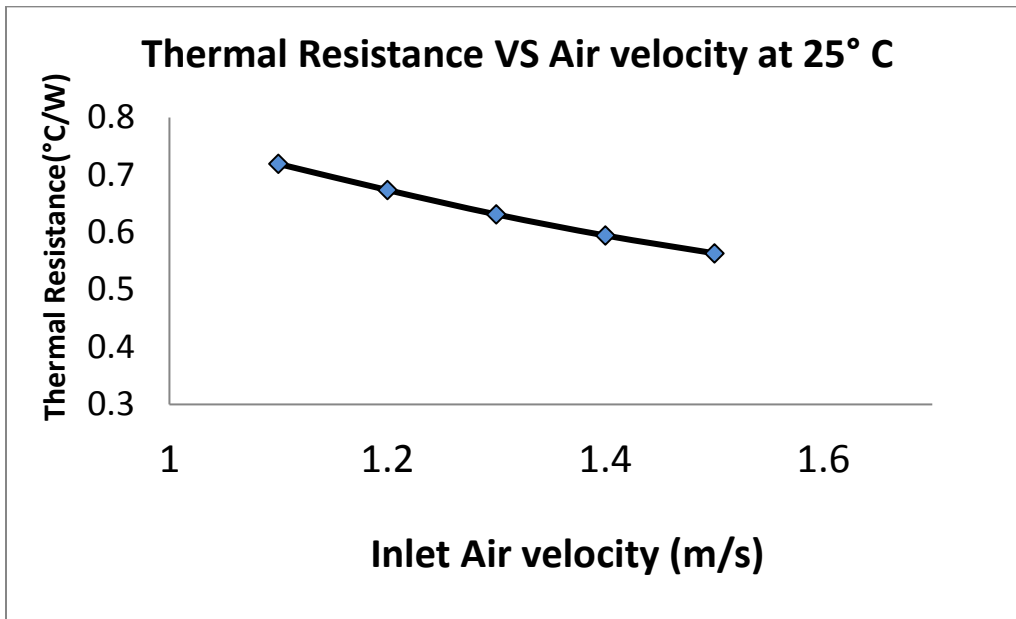


Figure 6.1 Thermal resistance Vs Inlet air velocity

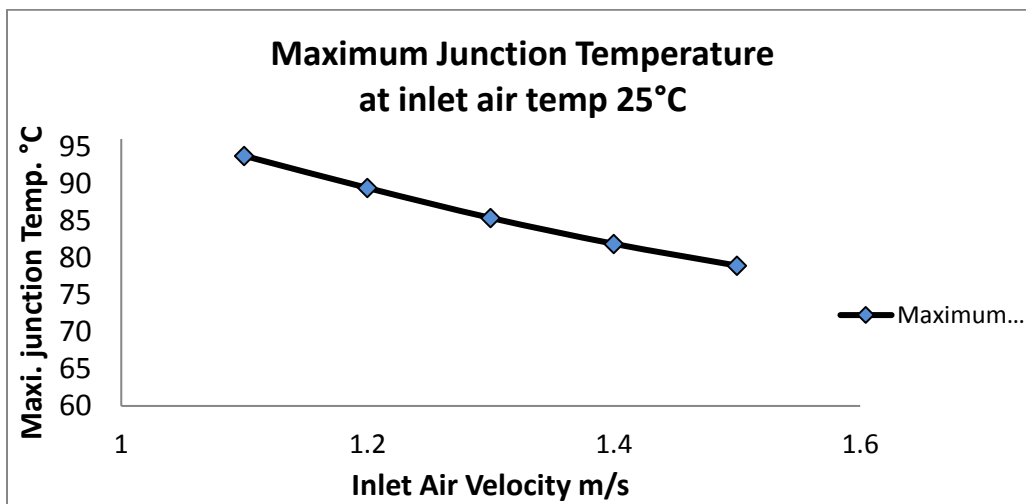


Figure 6.2 Maximum junction temp. Vs Inlet air velocity

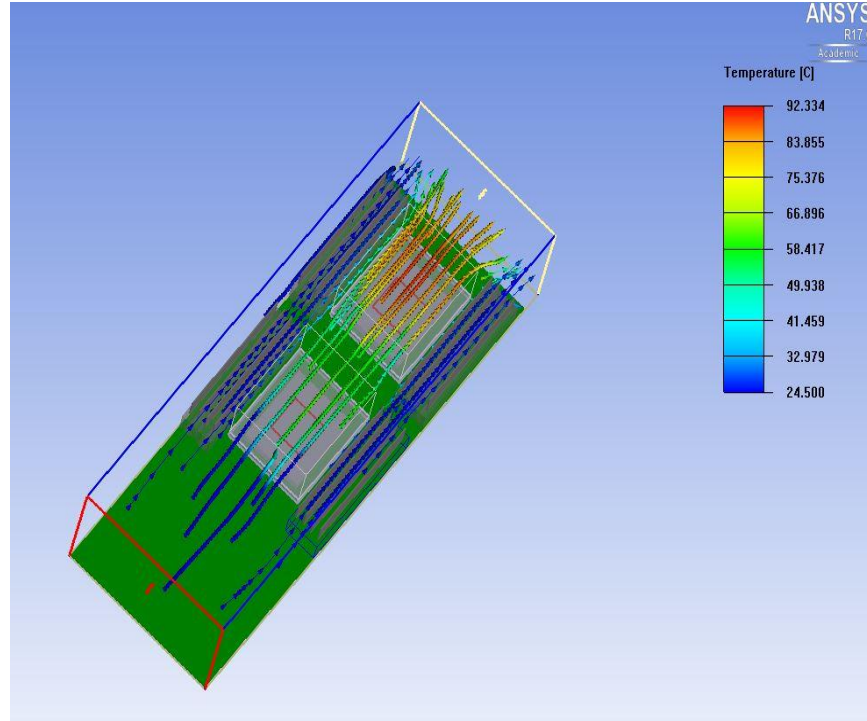


Figure 6.3 Particle trace of temperature in non-directed air cooled server

The change in maximum thermal resistance and junction temperature according to inlet air velocity is shown in Figure 6.1 and 6.2 respectively. As expected both quantities decrease with increase in inlet oil velocity. From the Figure 6.2, the maximum junction temperature reaches up to 95°C , it leads towards the thermal shadowing effect on the heat sink 2. The temperature of the server found to be at 80°C even though inlet air velocity is kept as 1.5 m/s . The results clearly indicate that ducting system is mandatory if air cooling technique is used. In the Figure 6.3, the blue vectors of particle trace get converted into green that shows the increase in air temperature and further changes into red near the

source 2. It can be stipulated that if flow is not directed towards the heat sink 2, there is considerable impact of thermal shadowing in air cooled servers.

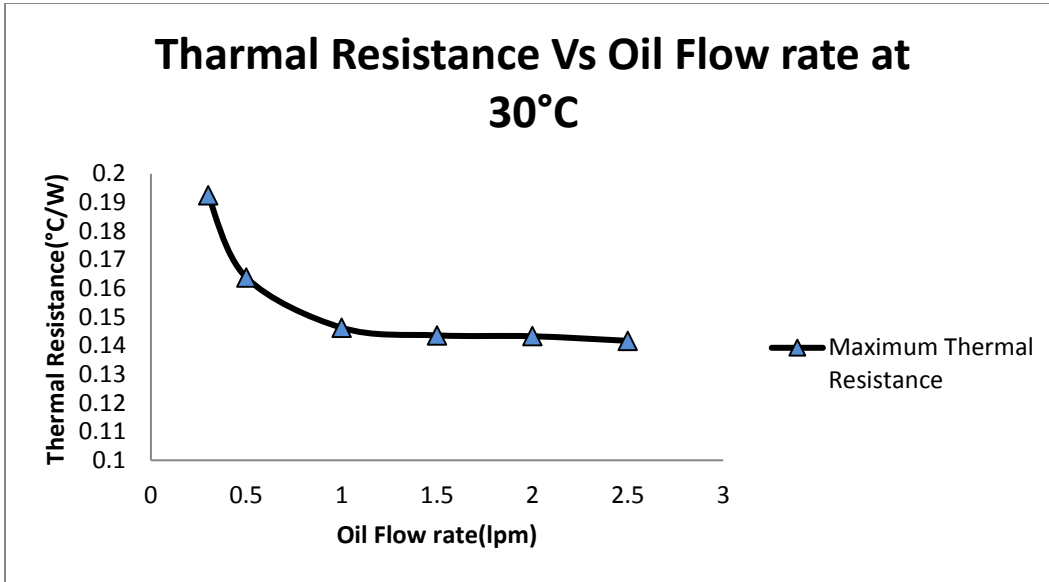


Figure 6.4 Thermal resistance VS Oil flow rate

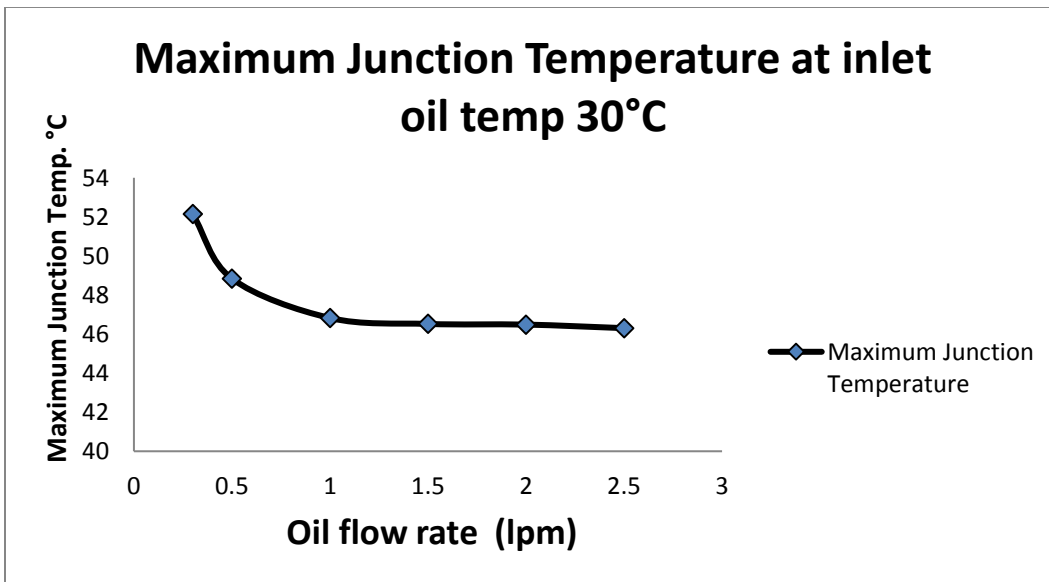


Figure 6.5 Maximum junction temp. VS Oil flow rate

The obtained results are focused on thermal performance of an oil cooled server. This results are obtained by keeping similar form factor and server power that has been used for an air-cooled server. As compare to air cooled server, the maximum junction temperature reaches to 52°C even though at very low oil flow rate of 0.3 lpm. The maximum thermal resistance has the value of 0.2 °C/W. Thermal resistance and junction temperature of the system decreases with increase in oil flow rate. The value of temperature and thermal resistance remain constant once the flow rate goes beyond 1 lpm.

6.2 Form Factor Study

This section of study includes the computational results of the three different form factors. This study shows the change in trend of thermal performance the server with variation in inlet boundary condition.

6.2.1 Thermal performance of 2U server

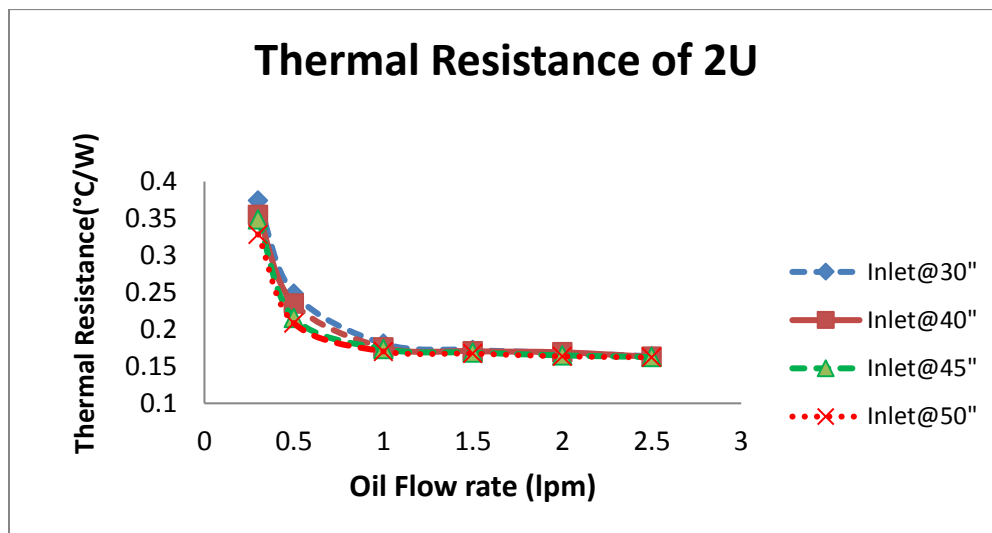


Figure 6.6 Thermal resistance of 2U server

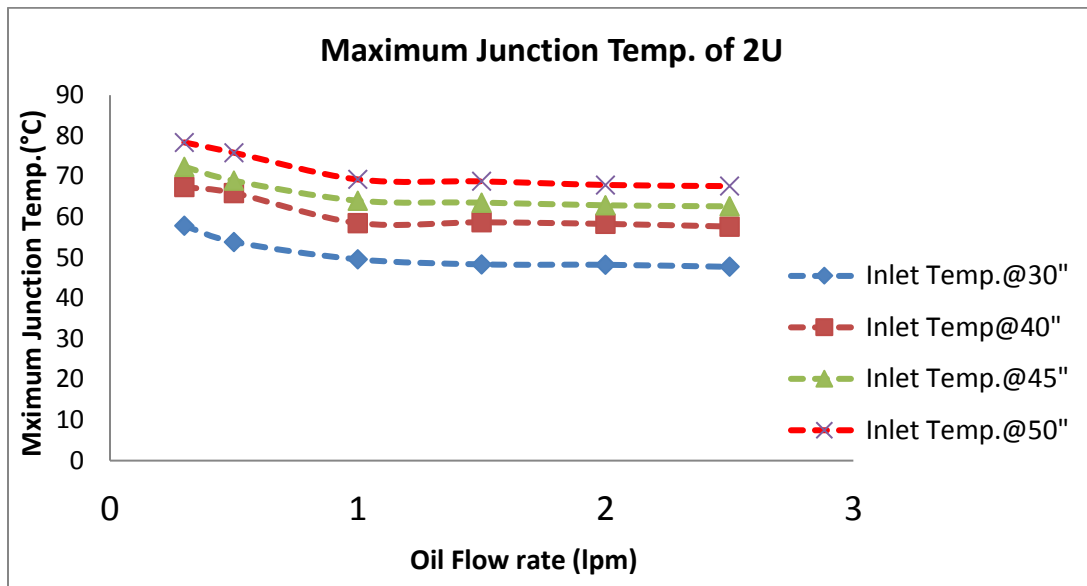


Figure 6.7 Maximum junction temperature of 2U server

Figure 6.6 and 6.7 describes the thermal performance of server having the form factor 2U. The study includes the multivariable parametric study of the server. Inlet oil temperature and oil flow rate are two variables that changes during the analysis. It can be depicted from Figure 6.6 that thermal resistance of the system decreases with increase in inlet oil temperature and completely opposite trend has been obtained for maximum junction temperature. The maximum junction temperature increases with increase in inlet oil temperature. It reaches to 80°C when the temperature of incoming oil is 50°C. The maximum thermal resistance is obtained when the incoming oil temperature is 30°C and oil flow rate is 0.3 lpm.

The range of oil flow rate and inlet oil temperature is taken from the previous experimental results and suggestions. [7]

6.2.2 Thermal performance of 1.5U server

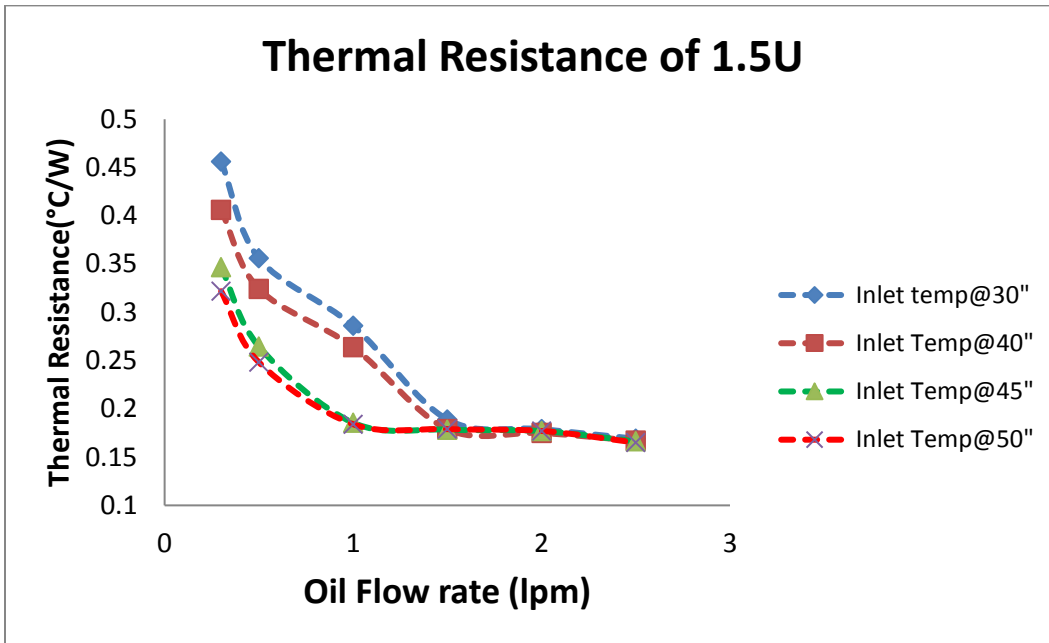


Figure 6.8 Thermal resistance of 1.5U server

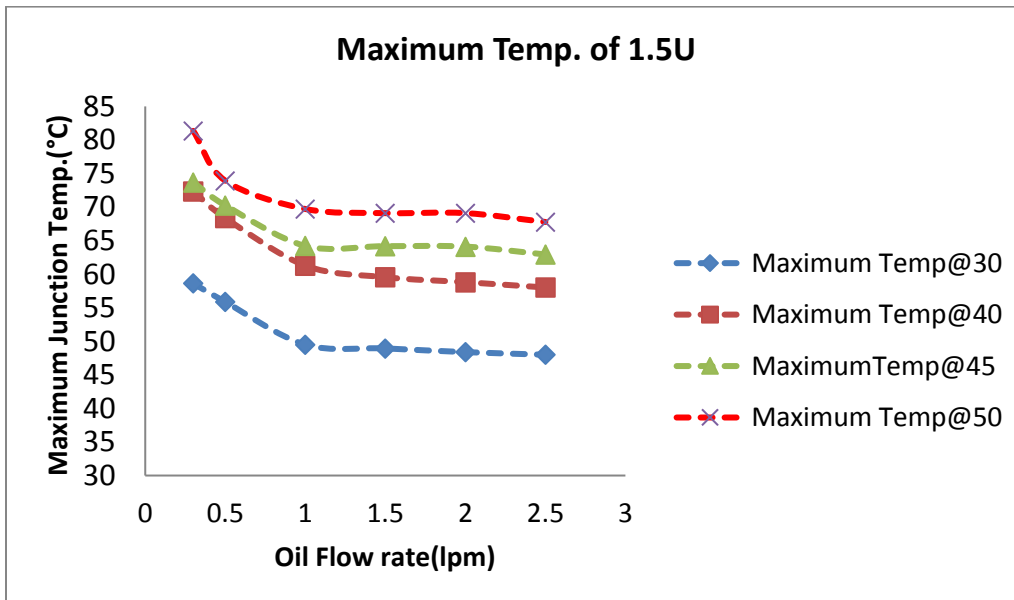


Figure 6.9 Maximum junction temperature of 1.5U server

Figure 6.8 shows the change in thermal resistance when the form factor is reduced from 2U to 1.5U, overall value of maximum thermal resistance is increased from 0.35 to 0.46 °C/W. The value of thermal resistance is decreasing when there is an increase in inlet oil flow rate. The value of thermal resistance will remain constant after the oil flowrate goes beyond the value of 1.5 lpm and its value is 0.18 °C/W.

Figure 6.9 explains the change of junction temperature when inlet oil temperature and oil flow rate is varied for 1.5U server form factor. The Junction temperature has the constant value for inlet oil flow rate greater than 1 lpm. The junction temperature has the maximum value of 83°C.

6.2.3 Thermal performance of 1U server

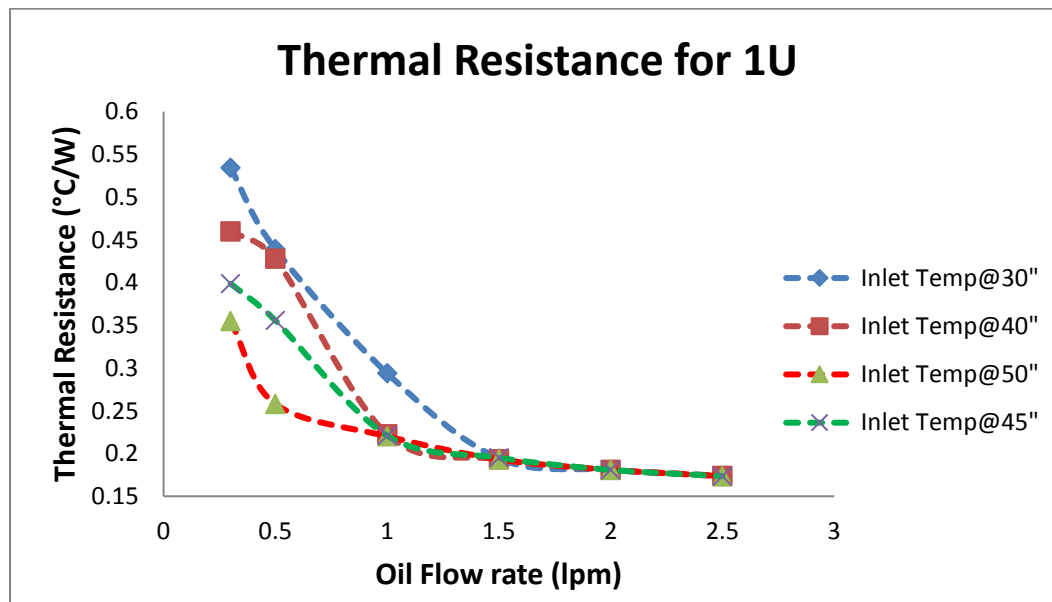


Figure 6.10 Thermal resistance of 1U

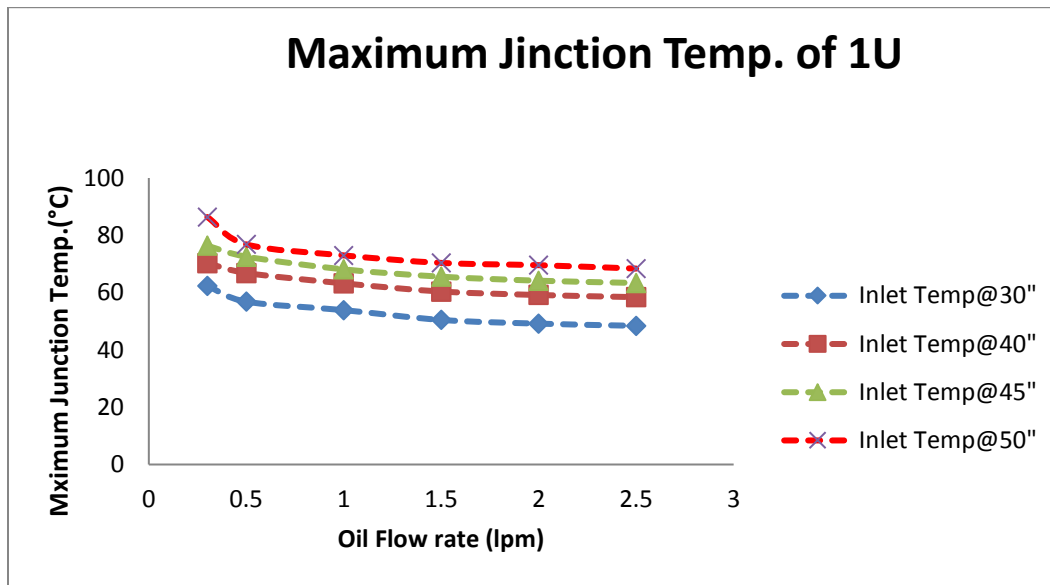


Figure 6.11 Maximum junction temperature of 1U

As expected, the maximum junction temperature and thermal resistance increases as compared to 1.5U and 2U form factor. Maximum junction temperature reaches to 95 °C when inlet oil temperature is kept at 50 °C. The trend of variation in temperature and thermal resistance remains constant once the inlet oil flow rate goes beyond the 1 lpm both quantities become near to constant. There isn't major change in their values after 1 lpm.

6.3 Heat sink optimization

6.3.1 Pin fin heat sink optimization

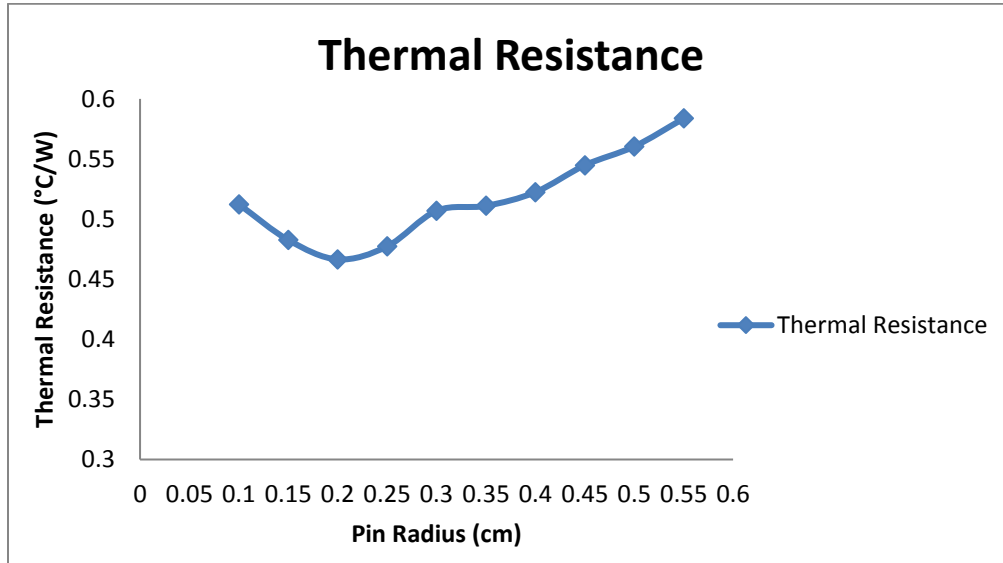


Figure 6.12 Thermal resistance Vs Pin radius

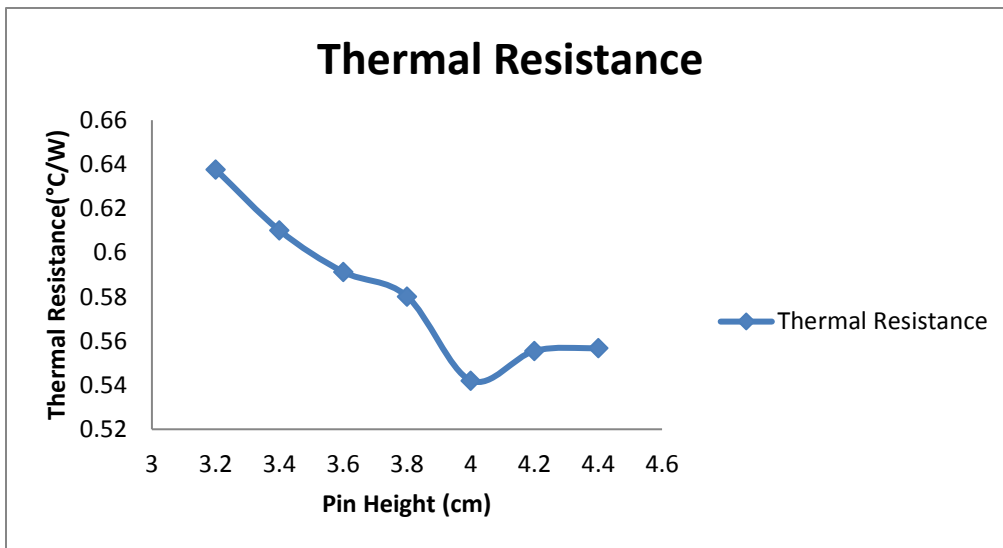


Figure 6.13 Thermal resistance Vs Pin height

Figure 6.12 and 6.13 shows the optimum range of pin radius and pin height respectively. From the Figure 6.12 and 6.13 optimum range of pin radius is from 0.2 to 0.25cm and optimum range of pin height is from 3.8 cm to 4.0 cm. In the mentioned range the optimized thermal performance can be obtained for the given server specifications for oil cooling applications.

6.3.2 Plate fin heat sink optimization

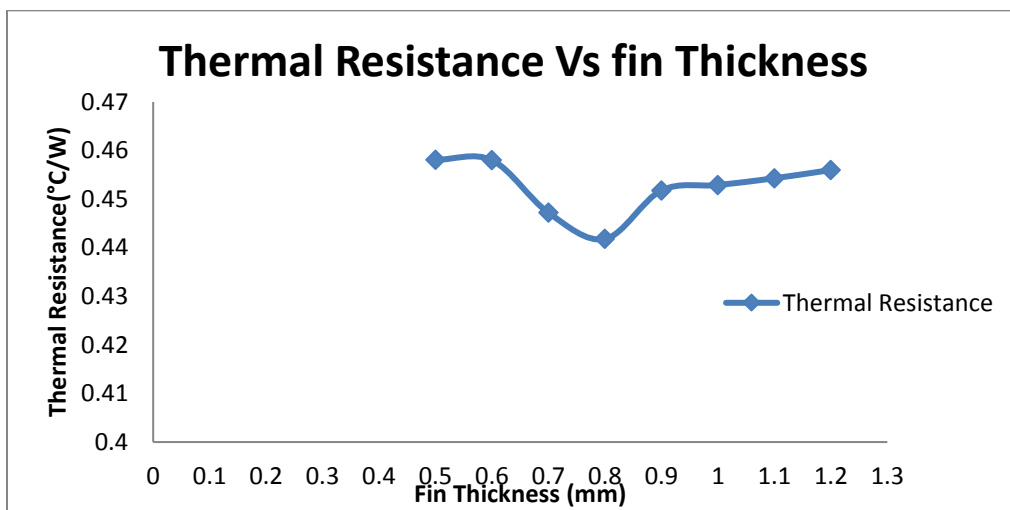


Figure 6.14 Thermal resistance VS Fin thickness

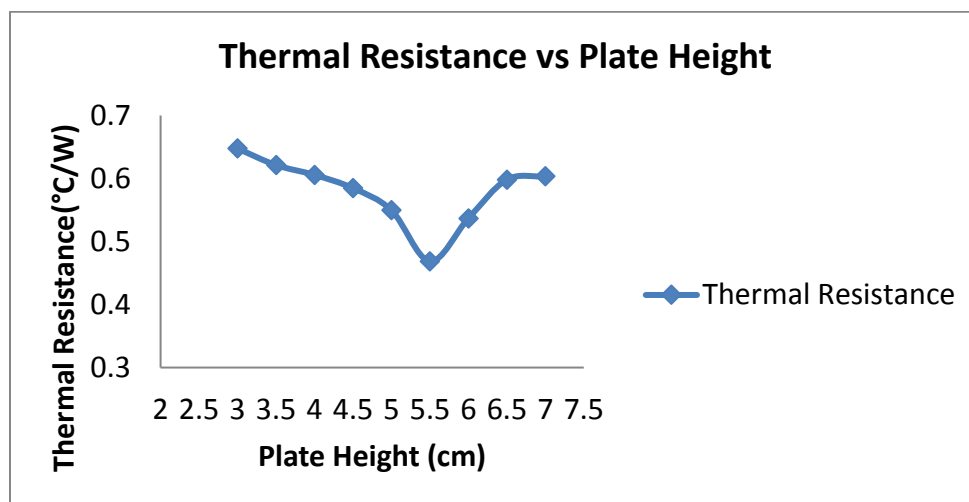


Figure 6.15 Thermal resistance vs Plate height

Figure 6.14 and 6.15 exhibits the optimized results for fin thickness and fin height for extruded cut plate fins. Figure 6.14 and 6.15 illustrate that optimum range of plate thickness should be from 0.5mm to 0.7mm and optimum range of plate height should be from 5cm to 6 cm to get the minimum thermal resistance for the given server.

Chapter 7

CONCLUSION

The study carried out here, opens new vision to improve the cooling system of highly demanding data center technology. The major conclusion and findings are summarized below.

1. From the obtained results, it can be concluded that ducting system can be eliminated altogether for an oil cooled server. The impact of thermal shadowing can be neglected in an oil cooled server compared to air cooled servers with similar power density.
2. Oil cooling system is capable enough to keep the system temperature below permissible limit of temperature. For an effective cooling of server having the form factor of 1U and 1.5U, it is advisable to maintain the oil flow rate greater than or equal to 1 lpm depending upon power density of server.
3. 12-13.4% savings in power consumption by server can be achieved as fans are removed which is an essential part of air cooling system and are replaced with pump for circulation of cooling oil.
4. Rack density can be increased by replacing 2U servers with 1U servers.
5. Optimized heat sinks have better thermal performance than the existing parallel plate heat sinks.

6. From material cost point of view by optimization of the heat sink there is 21.46 % saving compare to existing parallel plate heat sink geometry, considering material base cost as 2.6 \$ / kg.

Chapter 8

FUTURE SCOPE

This study can open the following scope to be accomplished.

1. Optimized heat sink can be experimentally validated with the computational data.
2. In order to cool very high power servers, modular system for oil cooling can be adopted to eliminate the thermal shadowing effect.
3. Instead of white mineral oil low viscosity DSI opticool heat transfer fluid can be used to achieve reduced pumping power and circulation cost. Evaluate the thermal performance using above suggested low viscosity oil.

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Biographical Information

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