CFD ANALYSIS OF THERMAL SHADOWING AND OPTIMIZATION OF HEAT SINKS IN 3RD GENERATION OPEN COMPUTE SERVER FOR SINGLE PHASE IMMERSION COOLING APPLICATIONS

by

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Abstract

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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. 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 very high-powered microprocessors and packaging. To overcome the limitations of air cooling in data centers, operators are moving towards immersion cooling using different dielectric fluids. 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. In this study, the impact of Thermal

Shadowing between different Dielectric Fluids is compared. The results of dielectric fluids, Mineral Oil and Synthetic Fluid EC100 are compared. The heat sink is a critical part for cooling effectiveness at server level. This work also provides an efficient range of operation for heat sink with 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.

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Chapter 1

INTRODUCTION

1.1 Introduction to Data Center

Data center is a part of the industry that encourages to store, process and deal with critical information by lodging the servers mounted in rack. Additionally, it enables the industry to use that information in a right manner. As the expansion of industries grows significantly, the tradition of data centers and servers raises rapidly. The growing demands of handling and storing the data in today's major sectors like education, networking, banking and production is the major cause for substantial consumption of power. Due to this high-power density of servers, the amount of heat generated increases which in turn requires efficient cooling.

1.2 Data Center Cooling Methods

The most common techniques used for cooling of data centers are

1. Air cooled servers
2. Liquid cooled servers

1.2.1 Air Cooled Servers

The prominent factor of data center is the dissipation of heat from the server. Air cooled arrangement is the most widely used technique for cooling this dissipated heat. The mechanism used in this technique is the forced convection of air over the heat sink. Heat sink is the heat exchanger that helps in transferring the heat generated from the processor to the outside medium. Based on the amount of heat, conditioned air is passed over the server. This airflow which carries heat from inlet to outlet is regulated by a device called axial fan. The hot air from the server outlet enters hot aisle and it is then directed into Computer Room Air Conditioning (CRAC) unit which cools down the air to the essential temperature. This cooled air is again sent into the cold aisle. The detailed process of air cooled server can be shown in the figure below [1].

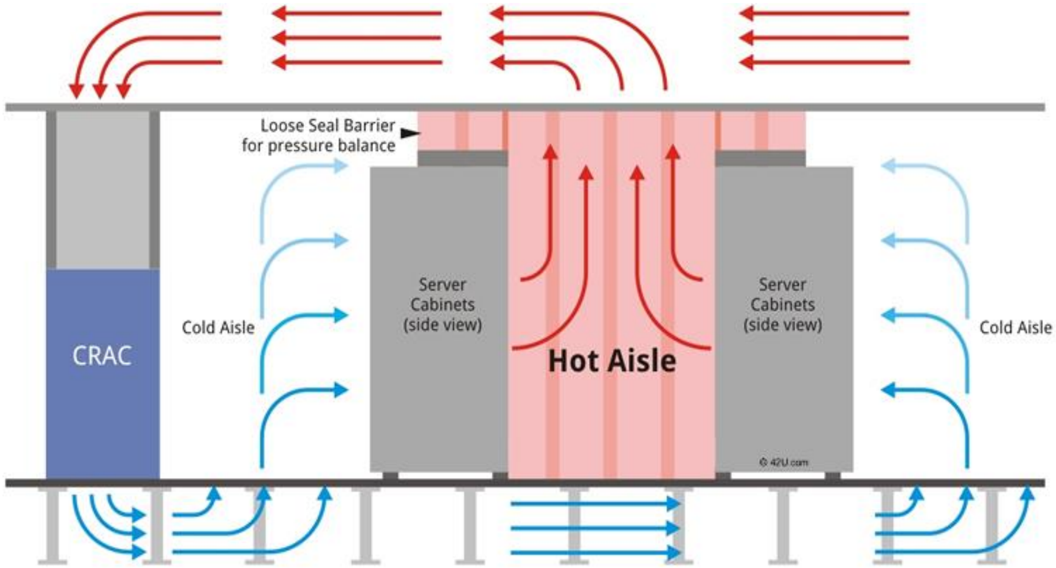

Figure 1 Hot Aisle / Cold Aisle Arrangement

1.2.2 Liquid Cooled Servers

Liquid cooling technique has been developed to overcome the restrictions and design constraints of air cooled servers. Air cooled server has its own limitations due to high powered packaging. As the air has poor conductivity of heat, it requires extra sources like fins attached to maximize surface area for transferring heat. Also, they consume larger space to accommodate fans and ducting system. Whereas in liquid cooling we can reduce the space occupied by fans. Liquid cooling is a method of heat dissipation where a processor is immersed in a fluid. The fluid used in liquid cooling can be categorized into two types.

a. Water cooled servers

b. Dielectric fluid cooled servers.

1.2.2.a Water Cooled Servers

There are two types of Water cooled systems-Immersion systems and In-direct immersion systems. In Immersion systems, the servers are submerged into water completely. This is not a suggested method as water conducts electricity when it gets in contact with server components. In In-Direct systems cooling method, a passive heat transfer device cold plate as shown in figure below is used. This is an indirect method of heat transfer as this cold plate is used to transfer heat generated from processor to water. The bottom surface of cold plate is made of copper and it is placed on heat producing components. The heat generated is then conducted to cold plate which increases the temperature of copper plate. This heat is transmitted to water that is flowing into cold plate through convection. The water is cooled down by chillers and recycled again [2].

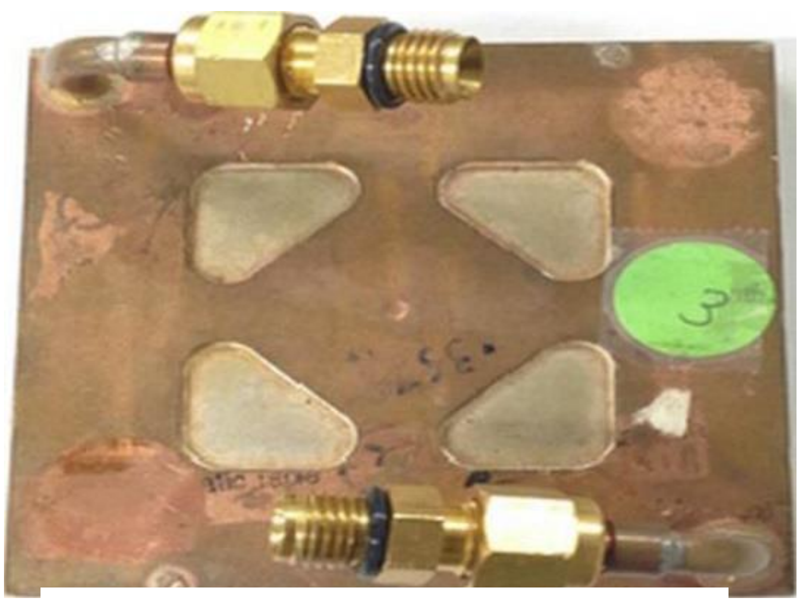

Figure 2 Cold Plate

1.2.2.b Dielectric fluid cooled servers

Even though the heat carrying capacity of water is high, water cooled servers are not recommended as water is a good electrolyte, good conductor of electricity and the design is complex. So, the choice of fluid should be non-conductive or Dielectric. In this technique, the server is immersed in Dielectric fluid and the heat is carried by the fluid. Flow meter is used to regulate the flow rate of the fluid. In this study two types of fluids were used. One is the widely used White Mineral oil and the other is the synthetic fluid Electro Cool 100 (EC-100). The physical properties of different types of fluids are compared for this study.

Figure 3 Dielectric Fluid Immersed Server

Table 1 Properties of different fluids

Type of Fluid	Heat Capacity (KJ/Kg K)	Density (Kg/m^3)	Kinematic viscosity (X 10^{-6} m^2/s)	Heat Conductivity (W/m K)
Air	1.01	1.225	0.16	0.02
Water	4.19	1000	0.66	0.58
White Mineral Oil	1.67	849.3	16.02	0.13
Synthetic fluid (EC-100)	2.165	803.78	13.22	0.1378

It is clear from the Table1 that the Thermal mass (Density x Heat Capacity) of Dielectric liquid is high compared to air. Although the thermal mass of water is the highest, the efficiency of heat dissipation is less as it is a passive heat transfer medium. Hence, we consider White Mineral Oil and Synthetic Fluid as a part of our study.

1.3 Motivation of the Work

Small saving in power utilization at server level can be transformed into critical amounts at industry level. To satisfy the demand for cooling of high power servers, oil cooling is one of the developing strategies. It has been identified that "Thermal shadowing" is an essential feature that has huge effect on cooling effectiveness for air cooled servers in past studies. So, it is essential to examine the effect of thermal shadowing in oil cooling. There are very few researches which consider effect of thermal shadowing in oil cooled servers. This study consists of computational analysis of effect of thermal shadowing and the comparison of the effect of thermal shadowing for different Dielectric fluids. Heat sink characterization is an important factor in determining product development. Determination of the optimum number of fins and fin specifications is also studied in this research work. The advantages of an optimized heat sink are reduction in material cost, increase in server efficiency and can allocate more servers per rack. The objective of this study is to propose an optimal heat-sink design in terms of thermal-performance and operating-cost.

Chapter 2

SPECIFICATION OF THE SERVER

2.1 Description of the Server

Third Generation Open Compute Server is considered in this study [9]. It has four DIM blocks each having four DIMMS of 8GB memory. It consists of two microprocessors with design power density of 115 W each. It has two CPU's each having a dimension of 50mm X 50mm. The length of the chassis is 166.2mm, width is 511mm and the height varies for different form factors as follows.

Table 2 Height of the chassis for rack unit and open rack unit

	RACK UNIT(mm)	OPEN RACK UNIT(mm)
1U	44.5	48
1.5U	66.5	72
2U	89	96

The Open Compute Server has a form factor of 2 Open rack unit. The footprint of the server under study is 166.2mm x 511mm x 96mm. The server is enclosed with a top cover on the chassis body. The top view of the server is presented below. It can be described that CPU2 will remain in thermal shadow of CPU1.

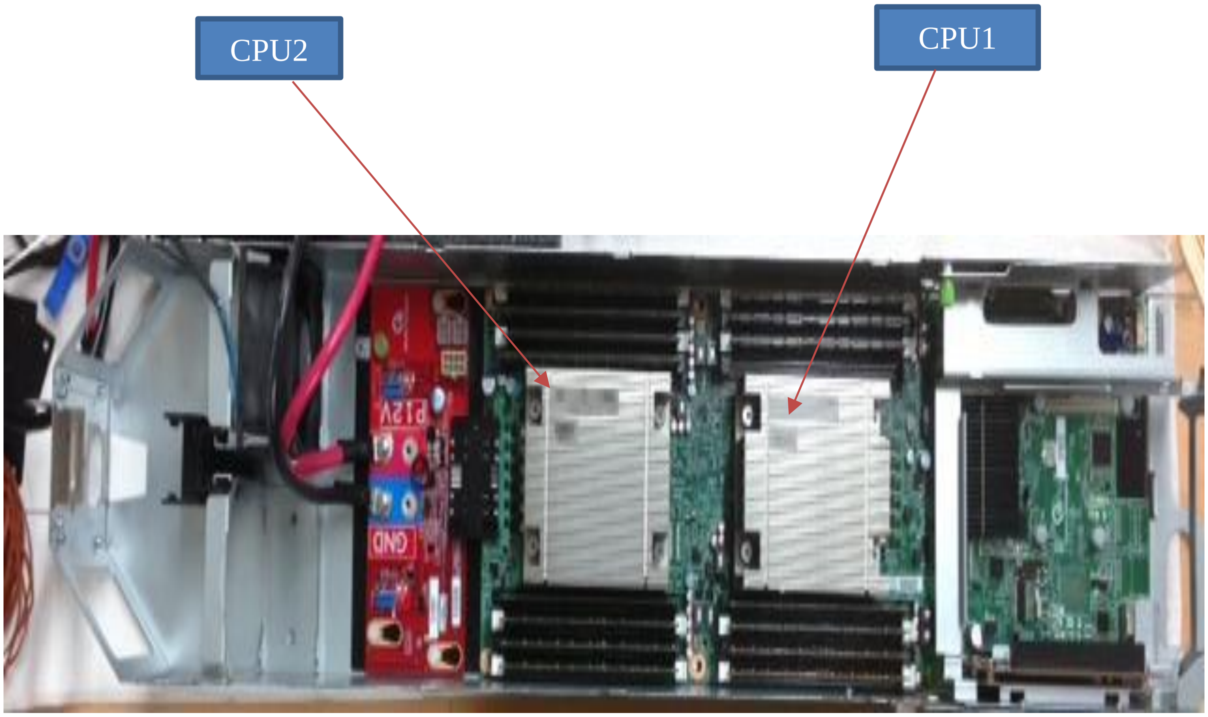

Figure 4 Top view of Open Compute Server

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

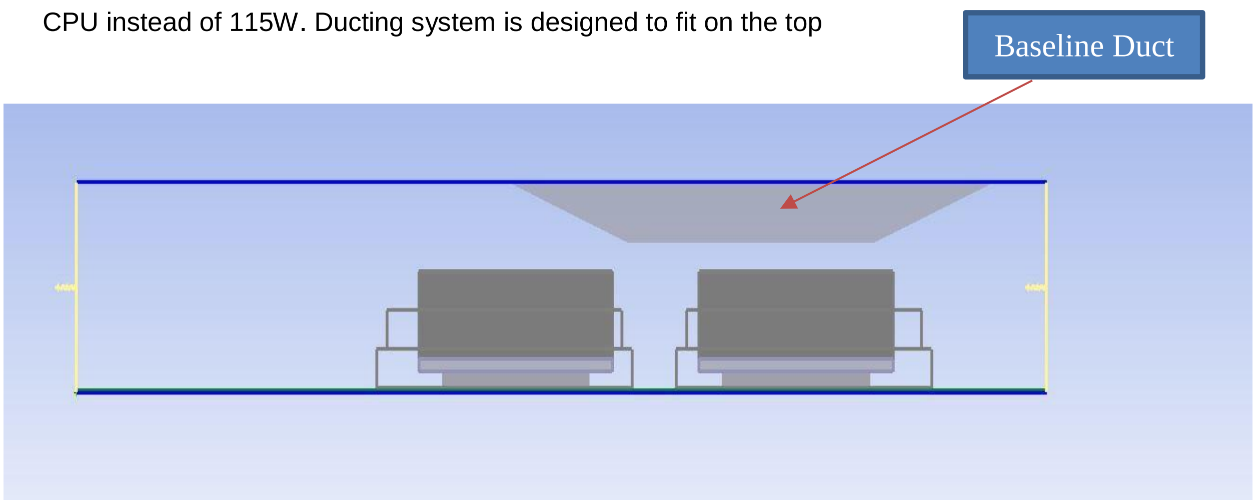

Figure 5 Side view of Baseline Server

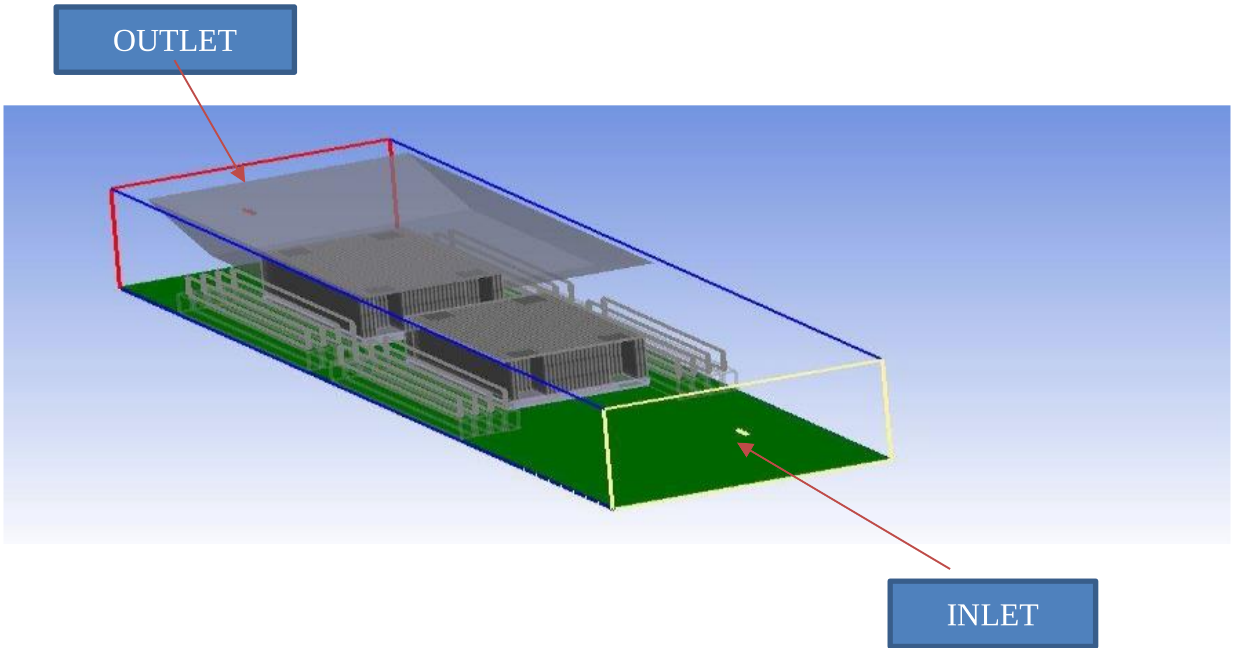

Figure 6 CFD model of Baseline Server

As shown in Figure 6, 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 [8]. 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.

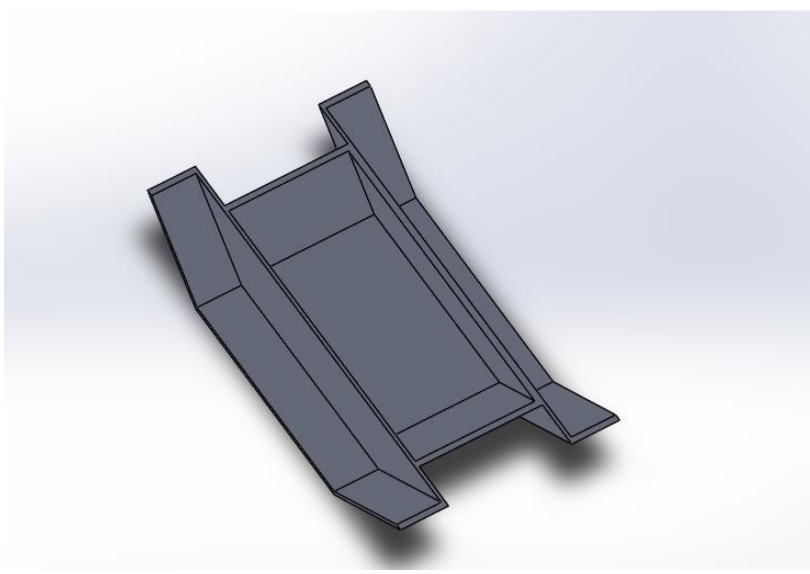

Figure 7 Baseline Duct

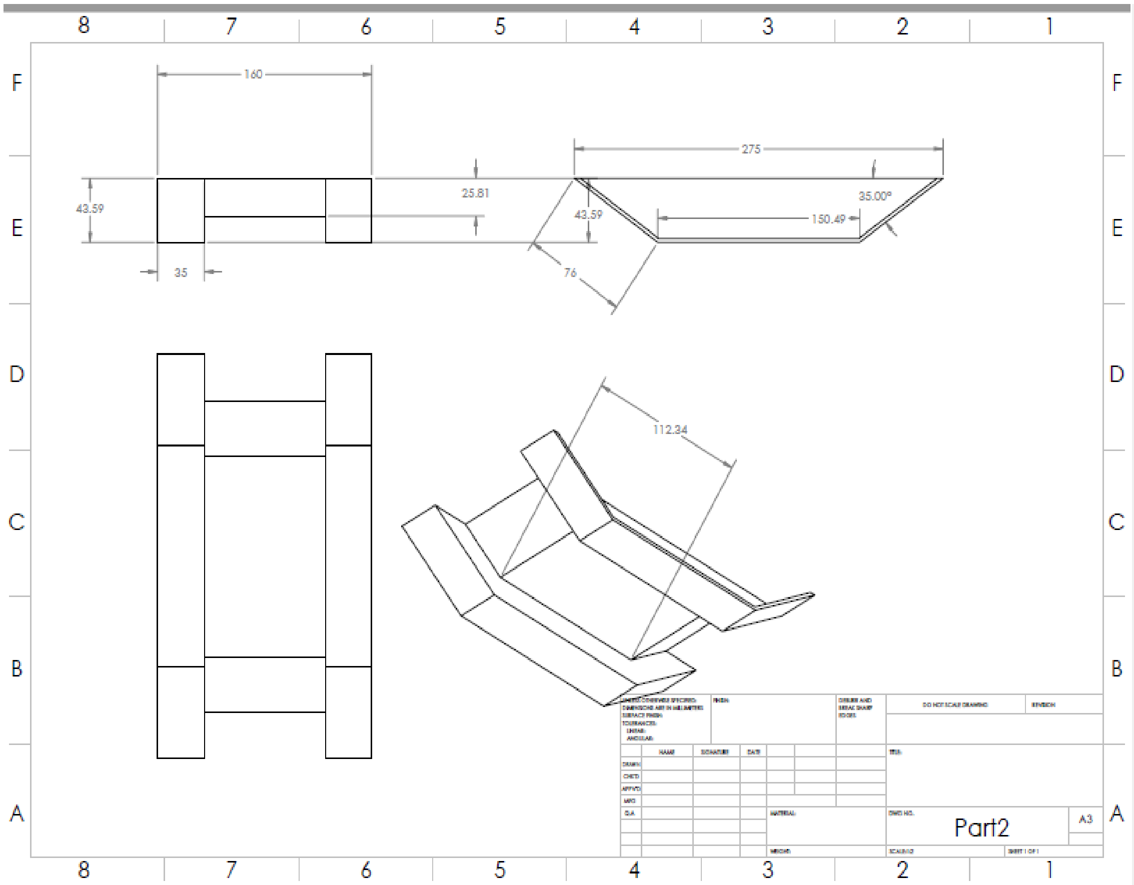

Figure 8 Baseline duct dimensions

Figure 8 shows the dimensions of the baseline duct. It is clear from the above dimensions that the duct consumes more space in the model. So, if the duct is removed, more servers can be accommodated in that space. So, in this study, for the baseline model duct is removed and the results are validated.

2.3 Oil Cooled Baseline Server Design

Baseline oil cooled server has been developed using a computational tool Ansys ICEPAK with the specifications mentioned for 2 Open Rack Unit Server. In the below geometry, Cabinet max y is the inlet and min y is the outlet for the flow of fluid.

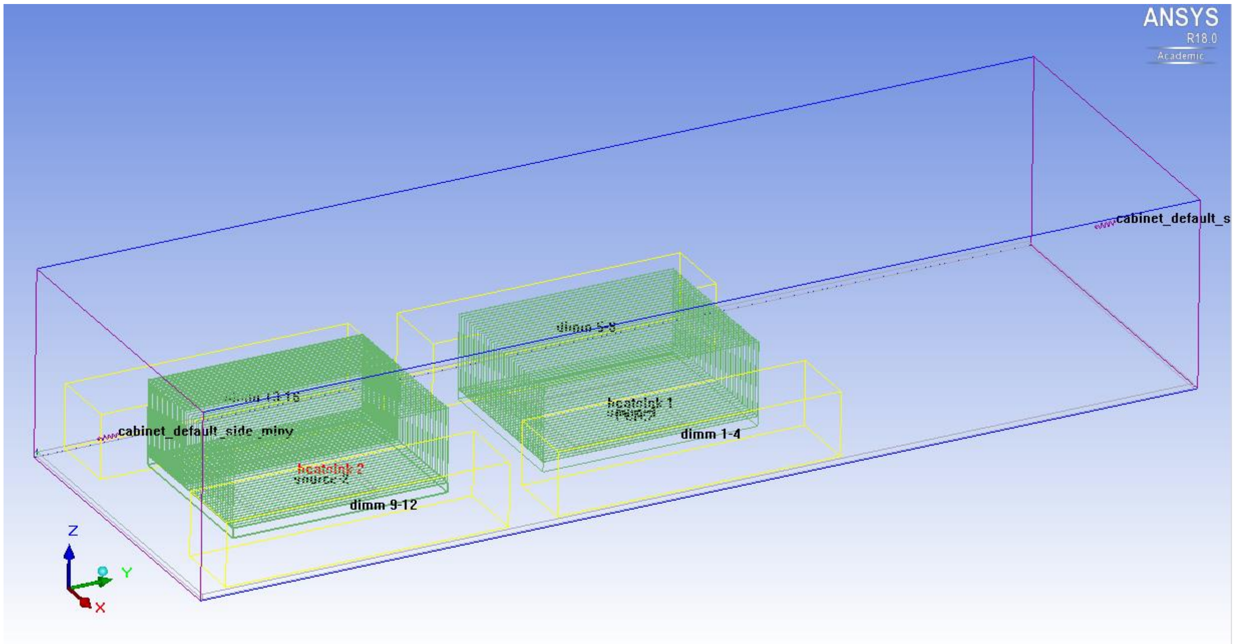

Figure 9 CFD model of a 2 Open Rack Unit Oil Cooled Server

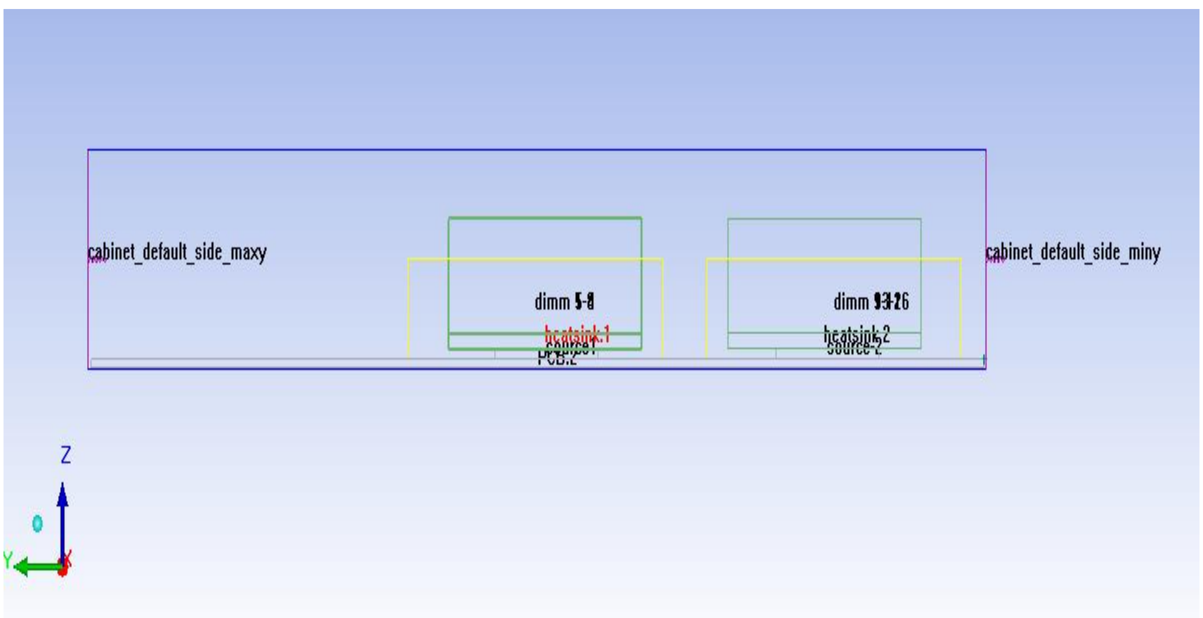

Figure 10 Positive X-direction Orientation of Server

2.4 Area Calculation

Area for different Open Rack Unit servers can be calculated by the following formula

Area = Length x Height …………………… (Equation 1)

1 Open Rack Unit:

Length= 166.2mm, Width= 511mm, Height= 48mm

Area of 1 Open Rack Unit server = Length x Height

= 166.2 mm x 48 mm

= 7977.6 mm^2

= 7977.6 x 10^{-6} m^2

1.5 Open Rack Unit:

Length= 166.2mm, Width= 511mm, Height= 72mm

Area of 1.5 Open Rack Unit server = Length x Height

= 166.2 mm x 72 mm

= 11966.4 mm^2

= 11966.4 x 10^{-6} m^2

2 Open Rack Unit:

Length= 166.2mm, Width= 511mm, Height = 96mm

Area of 2 Open Rack Unit server = Length x Height

= 166.2 mm x 96 mm

= 15955.2 mm^2

= 15955.2 x 10^{-6} m^2

2.5 Volumetric Flow Rate Conversion

In this study we consider Volumetric Flow Rate of 0.5 Liters Per Minute (LPM) to 2 LPM.

Table 3 Conversion of Volume Flow Rate from LPM to m^3/sec.

LPM	$m^3/sec.$
0.5	8.33E-06
1	1.6667E-05
1.5	2.5E-05
2	3.333E-05
2.5	4.1667E-05

2.6 Velocity Calculation

By using the volume flow rate and cross-sectional area, velocity of the fluid can be calculated.

Volume flow rate = Area x Velocity ……… (Equation 2)

⇨ Velocity (V) = Volume flow rate / Area

Table 4 Velocity Calculations for different Flowrates

LPM	m^3/sec	1 Open Rack Unit	1.5 Open Rack Unit	2 Open Rack Unit
0.5	8.33E-06	0.0010442	0.00069639	0.00052209
1	1.6667E-05	0.0020896	0.0013928	0.0010448
1.5	2.5E-05	0.0031337	0.002089	0.00156689
2	3.333E-05	0.00417795	0.002785	0.0020889
2.5	4.1667E-05	0.005223	0.003482	0.0026115

Chapter 3

EFFECT 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. Below figure shows the concept of Thermal Shadowing [3].

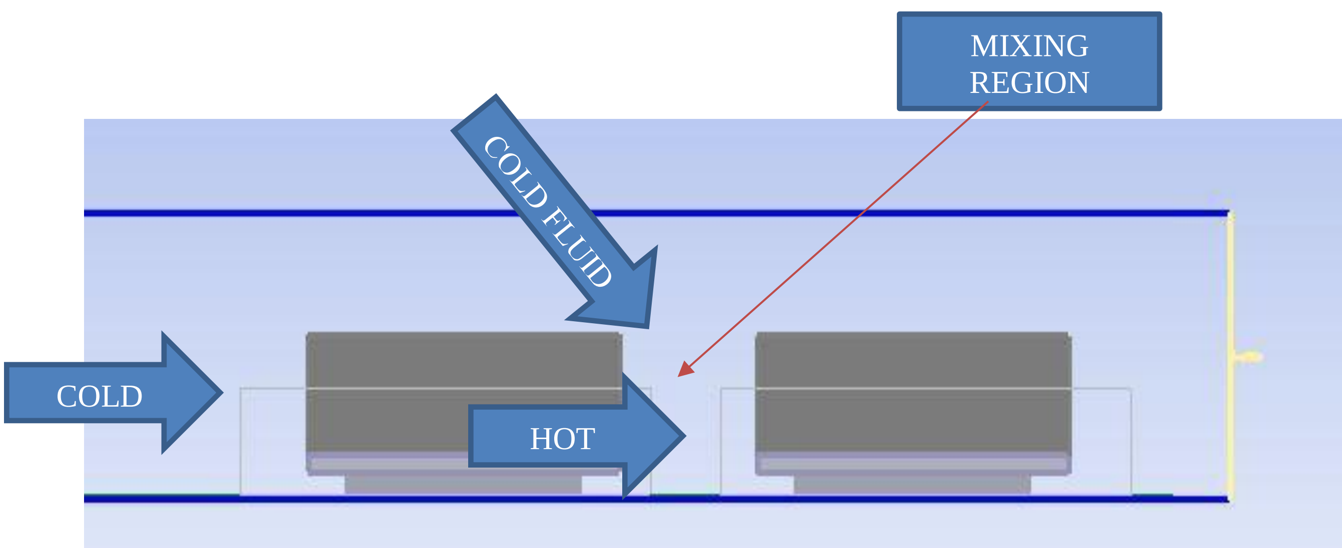

Figure 11 Thermal Shadowing Phenomenon

The cold fluid enters from one end of the server, some amount of fluid come across the first heat sink and carries the heat. This 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 explained using energy balance equation

$$M1h1 + M2h2 = M3h3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(Equation 3)}$$

Where

M1 = mass of cold fluid

h1 = enthalpy of cold fluid

M2 = mass of hot fluid

h2 = enthalpy of hot fluid

M3 = mass of mixed fluid

h3 = enthalpy of mixed fluid

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

$$M1CpT1 + M2CpT2 = M3CpT3$$

$$T3 = (M1T1 + M2T2) / M3$$

The temperature after mixing of fluid stream is T3. Temperature of mixed fluid stream T3 is higher than the inlet fluid temperature (T3 > T1). In air cooled servers, 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.

In this study, to compare the impact of thermal shadowing, non-directed flow is considered. This research has been done to analyze the parameters like Maximum Junction Temperature, Thermal Resistance.

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℃) and relative humidity is in range of ASHRAE [13] defined recommended range. Figure 12 shows the allowable and recommended zones for air cooling method.

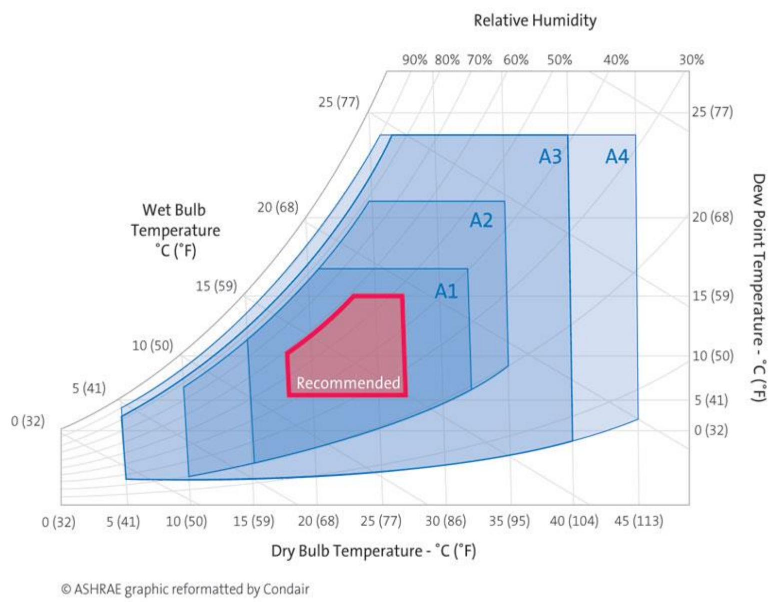

Figure 12 ASHRAE Recommended zones for Data center cooling

Temperature and pressure drop has been obtained keeping the same boundary condition used for an experiment and compared with the previously documented CFD results. 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 naiver-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. To decide that which model is used for the simulation process, Reynolds number is very important dimension less number. For an air-cooled server, it came out as $Re \geq 4000$ for selected inlet CFM range. 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 ±10 % error with the actual results.

Table 5 Pressure Drop of CFD Model

Flow Rate (LPM)	Previous CFD Pressure Drop (in/H_2O)	ICEPAK CFD Pressure Drop (in/H_2O)	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

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℃ and inlet air velocity is kept constant as 1m/s.

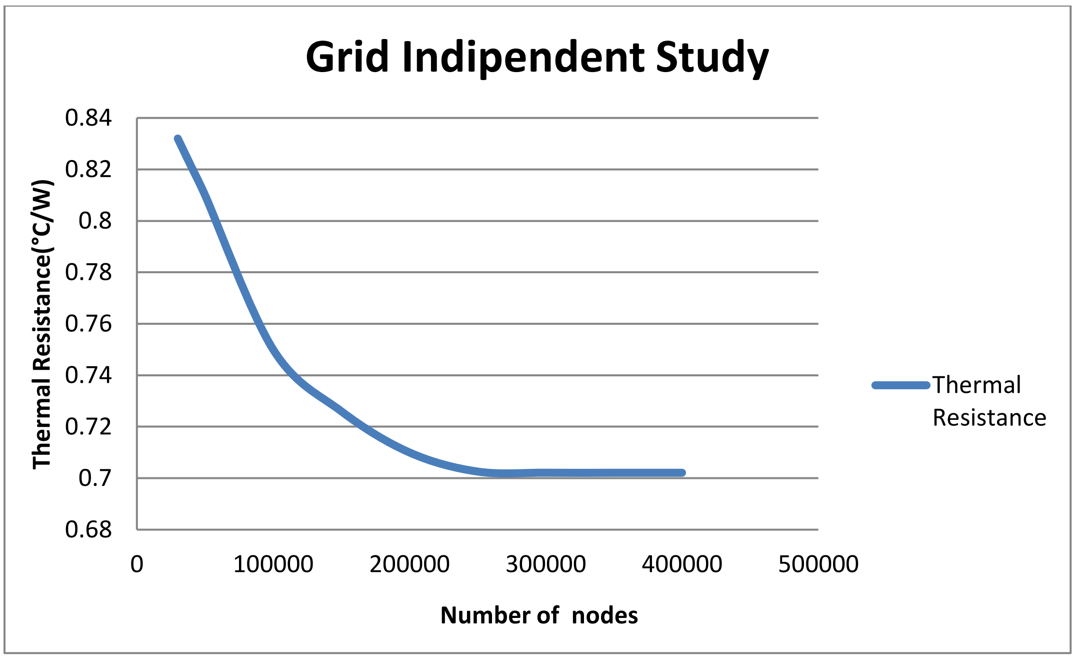

Figure 13 Grid Independent Study

It is clear from the above graph that the 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 Validation and Grid Independent Study of 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 to be changed in place of air. There are different types of Dielectric Fluids, But in the study White Mineral Oil and Synthetic Fluid EC100 is used by considering cost factors and High temperature cooling applications.

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. 2 Open Rack Unit Server cross section in this case is taken as 511mm x 166.2mm x 96mm.

The formula below can calculate Hydraulic Diameter,

$$D=\frac{2\,b\,h}{(b+h)} \text{ ……………… (Equation 4)}$$

Where,

b= channel width = 511 mm

h= channel height = 96 mm

D = (2 x 511 x 96) / (511 +96)

= 161.6 mm = 0.1616 m

3.3.1 Reynolds number calculation for White Mineral Oil

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

Density – 851.515 Kg/m^3

Thermal conductivity – 0.13 W/m K

Specific heat – 1680 J/kg k

Thermal Diffusivity – 9.166E-8 m^2/s

Molecular Weight – 150 Kg/ K Mol

Over all heat transfer co-efficient – 50-30 W/ $m^2\ K$

At 30℃, the Dynamic Viscosity (μ) of White mineral oil is 0.01405 Kg/m s

The Kinematic Viscosity (v) is 1.65E-05 m^2/s

Reynolds number can be calculated using the below formula

$$Re=\frac{V\,D}{\nu} \ldots\ldots\ldots\ldots\ldots\ldots \text{(Equation 5)}$$

Where Re is the Reynolds number,

V is the Velocity,

D is the Hydraulic Diameter,

v is the Kinematic Viscosity.

Substituting Hydraulic Diameter, Velocity and Kinematic Viscosity in eq 5,

Re= (0.0010448 x 0.1616) / (1.65E-05)

Re = 10.23

Prandtl number can be calculated using the below formula

$$P= \frac{\mu\, C_P}{\lambda} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(Equation 6)}$$

Where P is the Prandtl number,

μ is the Dynamic Viscosity,

C_P is the Specific Heat

λ is the Thermal Conductivity

Substituting Dynamic Viscosity, Specific heat and Thermal conductivity in eq 6,

P= (0.01405 X 1670) / 0.13

P= 180.488

Table 6 Change in properties of White Mineral Oil due to Temperature

Temperature (℃)	Dynamic Viscosity (kg/m-s)	Kinematic viscosity (m^2/s)	Reynolds Number	Prandtl Number
30	0.01405	1.65E-05	10.23	180.488
40	0.01046	1.23E-05	13.72	134.37
45	0.00909	1.07E-05	15.77	116.77
50	0.00794	9.35E-06	18.05	101.99

3.3.2 Reynolds number calculation for Synthetic Fluid

Density – 803.78 Kg/m^3

Thermal conductivity – 0.1378 W/m K

Specific heat – 2165.9 J/kg k

Molecular Weight – 350 Kg/ K Mol

At 30℃, the Dynamic Viscosity (μ) of Synthetic Fluid is 0.01062 Kg/m s

The Kinematic Viscosity (v) is 1.322E-05 m^2/s

Re=$\frac{V\ D}{\nu}$

Re = (0.0010448 x 0.1616) / 1.322E-05

Re = 12.77

Prandtl number can be calculated using the below formula

P= $\frac{\mu\ C_P}{\lambda}$ = (0.01062 X 2165.9) / 0.13787

P= 166.837

Table 7 Change in properties of Synthetic Fluid due to Temperature

Temperature (℃)	Dynamic Viscosity (kg/m-s)	Density (kg/m^3)	Specific heat (J/kg k)	Thermal Conductivity (W/m K)	Kinematic Viscosity (m^2/s)	Reynolds Number	Prandtl Number
30	0.01062	803.78	2165.9	0.13789	1.322E-05	12.77	166.837
40	0.00767	796.98	2203.2	0.13730	9.63E-06	17.53	123.077
45	0.00662	793.58	2221.9	0.13702	8.34E-06	20.24	107.349
50	0.00576	790.18	2240.5	0.13673	7.29E-06	23.16	94.385

Reynolds number is less than 2000 for both White Mineral Oil and Synthetic Fluid. So Laminar model is used to solve the naiver stoke equation. CFD results documented by Chinmay Bhatt [12] are compared with the experimental results documented by Trevor Mc.Williams [10] chifor Validation.

Table 8 Validation of an Oil Cooled CFD Model

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

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.

Thermal Resistance = (Tj – Ta) / Heat Dissipation

Where,

Tj = Junction Temperature

Ta = Incoming fluid temperature

Heat dissipation from processor is taken in watts.

3.5 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. The previous results of air cooled server are used for comparison with this study. For oil cooled server, inlet fluid temperature is kept varied from 30℃ to 50℃, inlet fluid velocity has been varied for specified flow rate changes from 0.5 LPM to 2.5 LPM.

The maximum junction temperature is compared for an air cooled and oil cooled servers. Also, a comparison between White Mineral Oil and Synthetic Fluid can also be observed in this study.

Chapter 4

OPTIMIZATION OF HEAT SINKS

4.1 Types of Heat Sinks

Heat sinks are passive components those are used for heat transfer. The major three types of heat sinks are Parallel plate heat sink, pin fin heat sink and extruded cut plate fin heat sink. The baseline model with the parallel plate heat sink was used in this study [4].

Parallel Plate Heat Sink   Plate Fin Heat Sink   Pin Fin Heat sink

Figure 14 Types of Heat Sinks

Heat sinks are designed to increase the heat transfer area between heat producing 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. Parallel 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. The results of these Pin fin and Plate fin are then compared with the existing Parallel plate heat sink. To evaluate the thermal performance of the heat sinks, dimension less numbers like Reynold's number and Prandtl 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℃ and mass fix mass flowrate technique is applied.

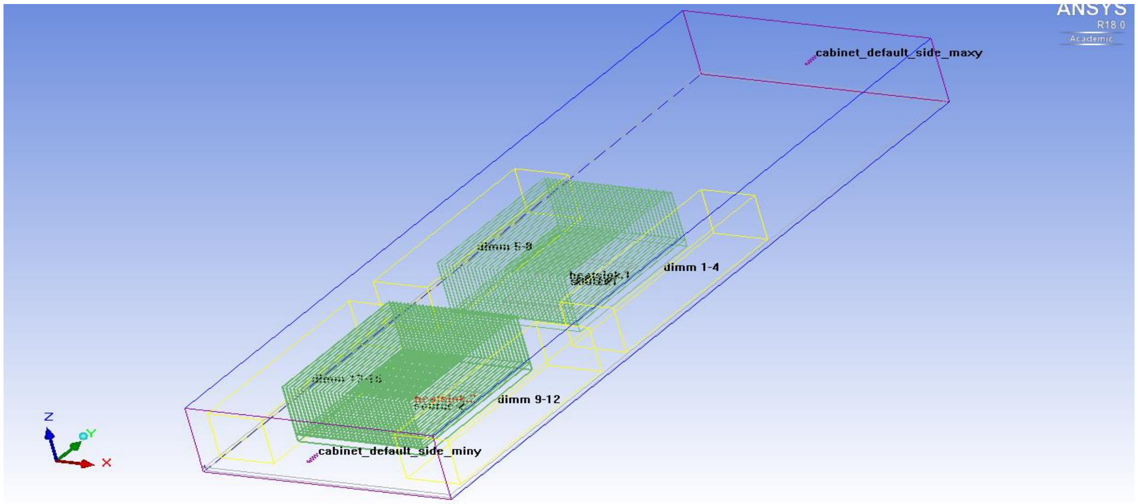

Figure 15 Baseline Parallel Plate Heat Sink Server

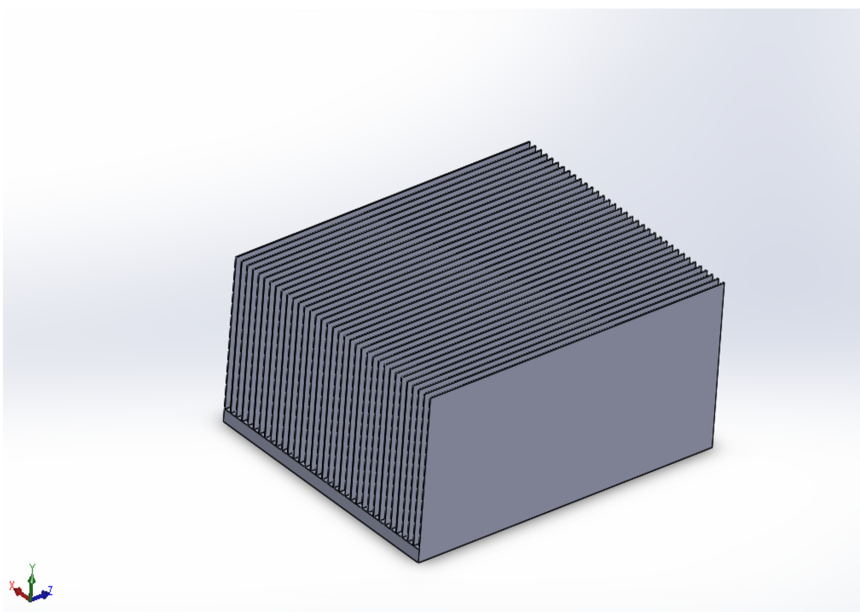

Figure 16 : CAD model of existing Parallel Plate Heat Sink

Table 9 Specifications of existing parallel plate heat sink

Thickness	0.3 mm
Height	41 mm
Length	110 mm
Number of fins	35

Volume of this heatsink can be calculated using the formula,

Volume = Thickness x Height x Length …………… (Equation 7)

Volume = 0.3 x 41 x 110 = 1353 mm^3

For 35 fins,

Volume = 35 x 1353

= 47355 mm^3

4.2 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. In this optimization study, for all the simulations, we consider 1 Open Rack Unit Server at 30℃ and the Power of the CPU is considered as 65W. The existing parallel plate heat sink has 35 fins which is complex. So, the existing parallel plate heat sink is modified, and the Fin count is reduced to 15. Parametric study was conducted on this optimized model to validate the results with existing baseline server model. The results of this study are explained in detail in the next chapter.

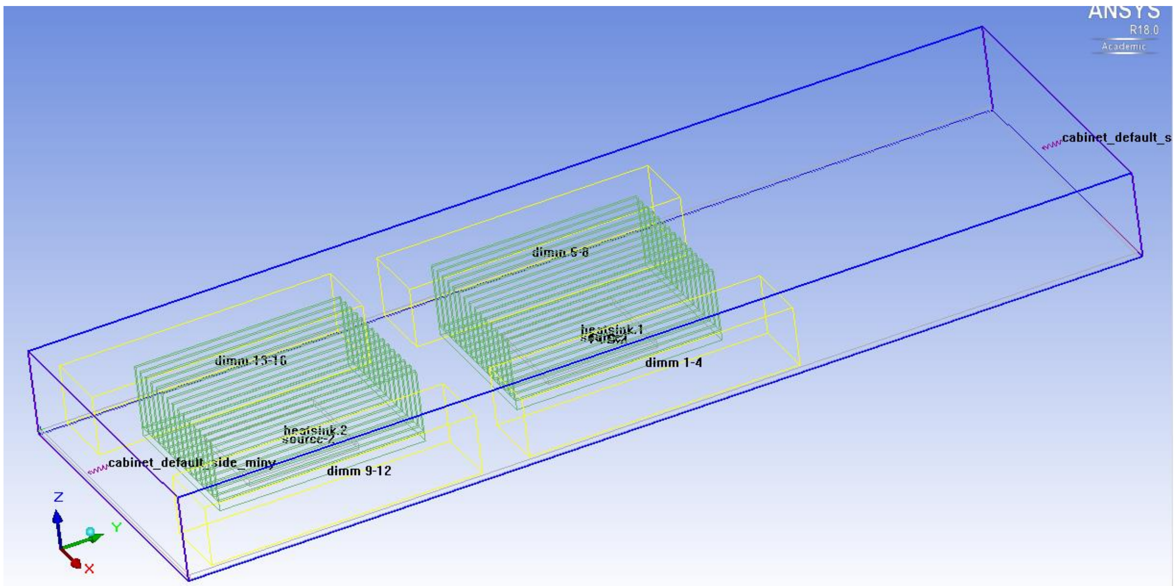

Figure 17 Optimized Parallel Plate Heat Sink Server

Similarly, the parallel plate heat sink is replaced with cut plate fins and pin fins. For cut plate fins, fin count is kept constant as 42 and the parametric study is conducted by varying the plate height and plate thickness. For pin fin heat sink, parametric study is conducted by varying pin radius and pin height by keeping the number of fins constant as 25. The same boundary conditions are used for this study.

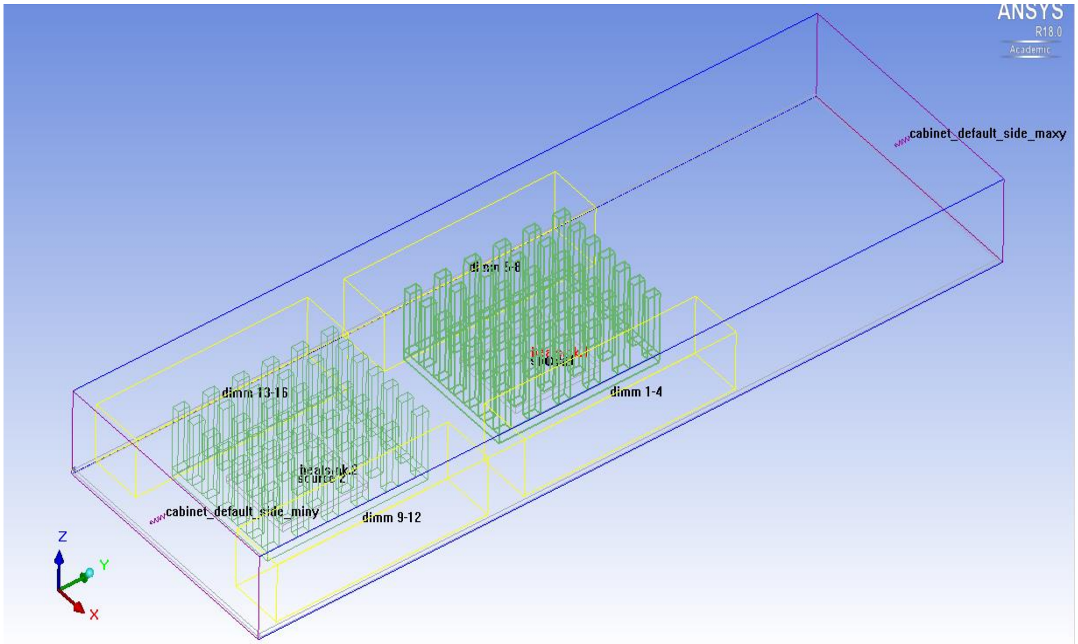

Figure 18 Extruded Cut Plate Fins Heat Sink

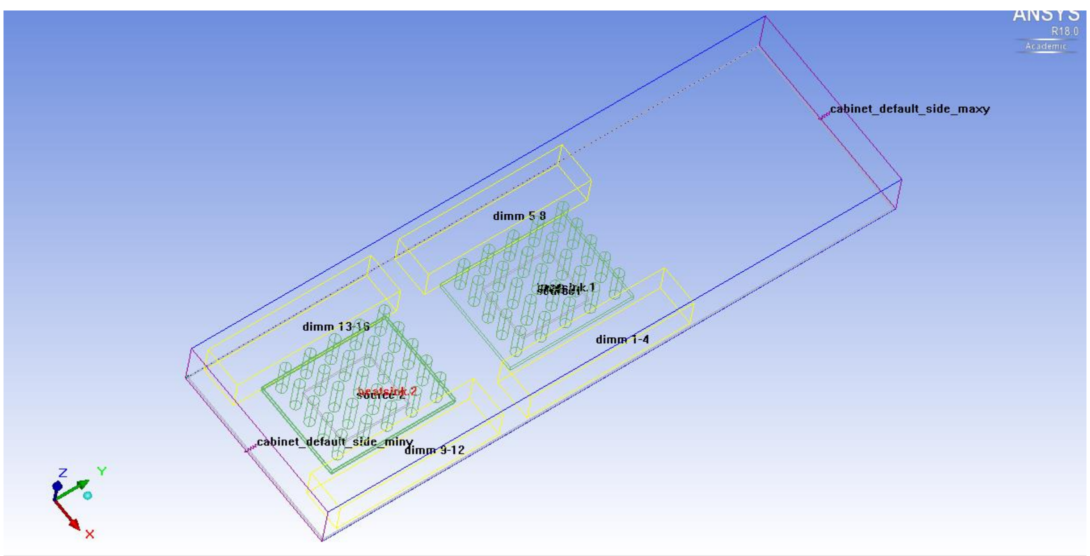

Figure 19 Pin fin Heat Sinks

The grid independent study has been carried out for accuracy of the result for optimization.

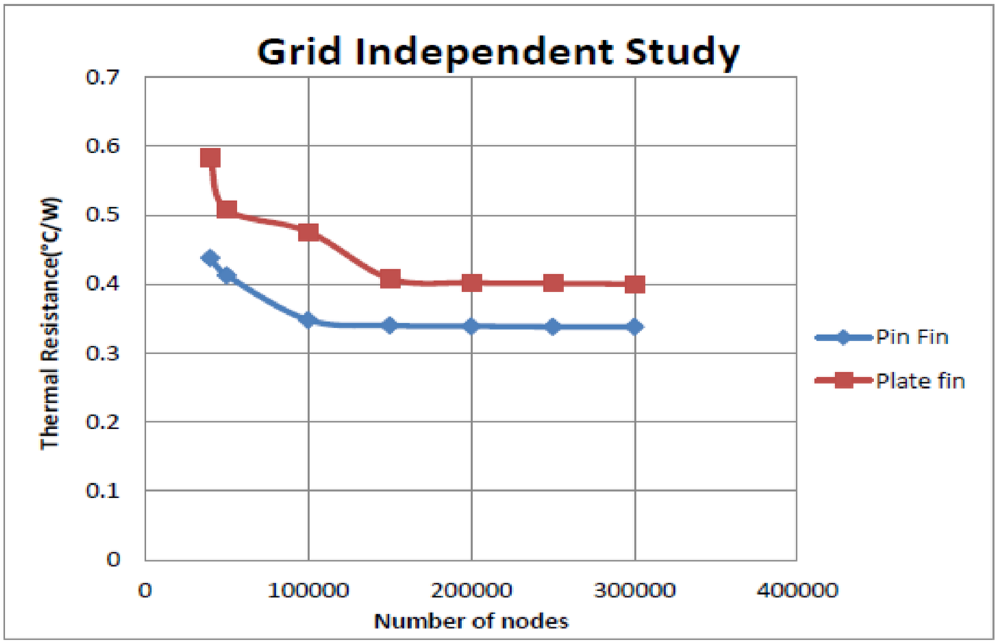

Figure 20 Grid independent study for plate fin and pin fin heat sink

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 20, 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℃.

Chapter 5

RESULTS

5.1 Impact of Thermal Shadowing in Air and Oil Cooled Servers

5.1.1 Thermal Shadowing in Air Cooled Server

Inlet temperature is considered as 25℃ and Flow rate as 1LPM. These results are documented by Divya Mani. These are used for our validation with an oil cooled server.

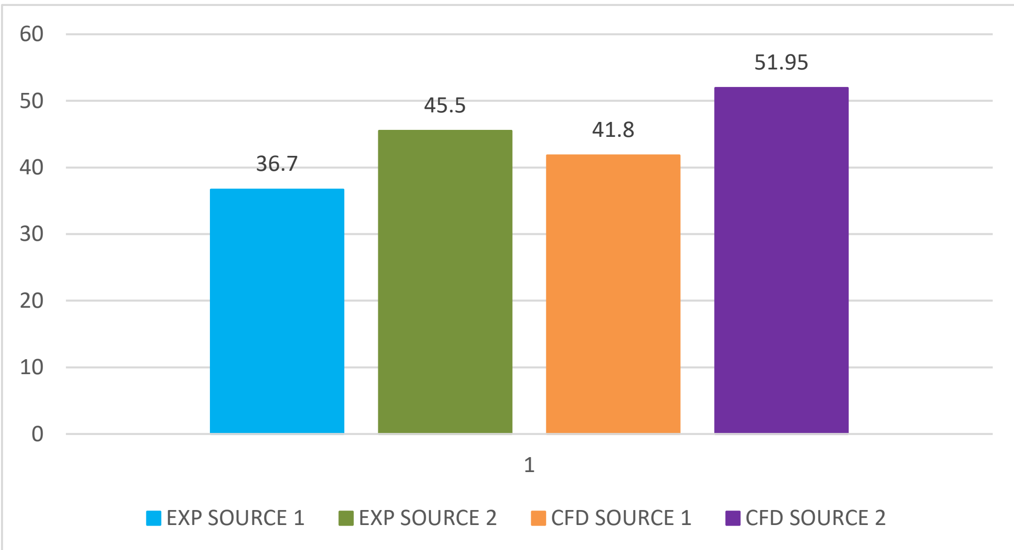

Figure 21 Experimental and CFD results of an Air-Cooled Server

From the above figure 21, it can be clearly seen that for experiment, the temperature of source 1 is 36.7℃ and the temperature of source 2 is 45.5℃. The temperature difference between source 1 and source 2 is 8.8℃. If we consider CFD results, the temperature of source 1 is 41.8℃ and the temperature of source 2 is 51.95℃. The temperature difference between source 1 and source 2 is 10.15℃.

5.1.2 Impact of Thermal Shadowing in Oil Cooled Server

5.1.2.a Thermal Shadowing in 2 Open Rack Unit Server using White Mineral Oil

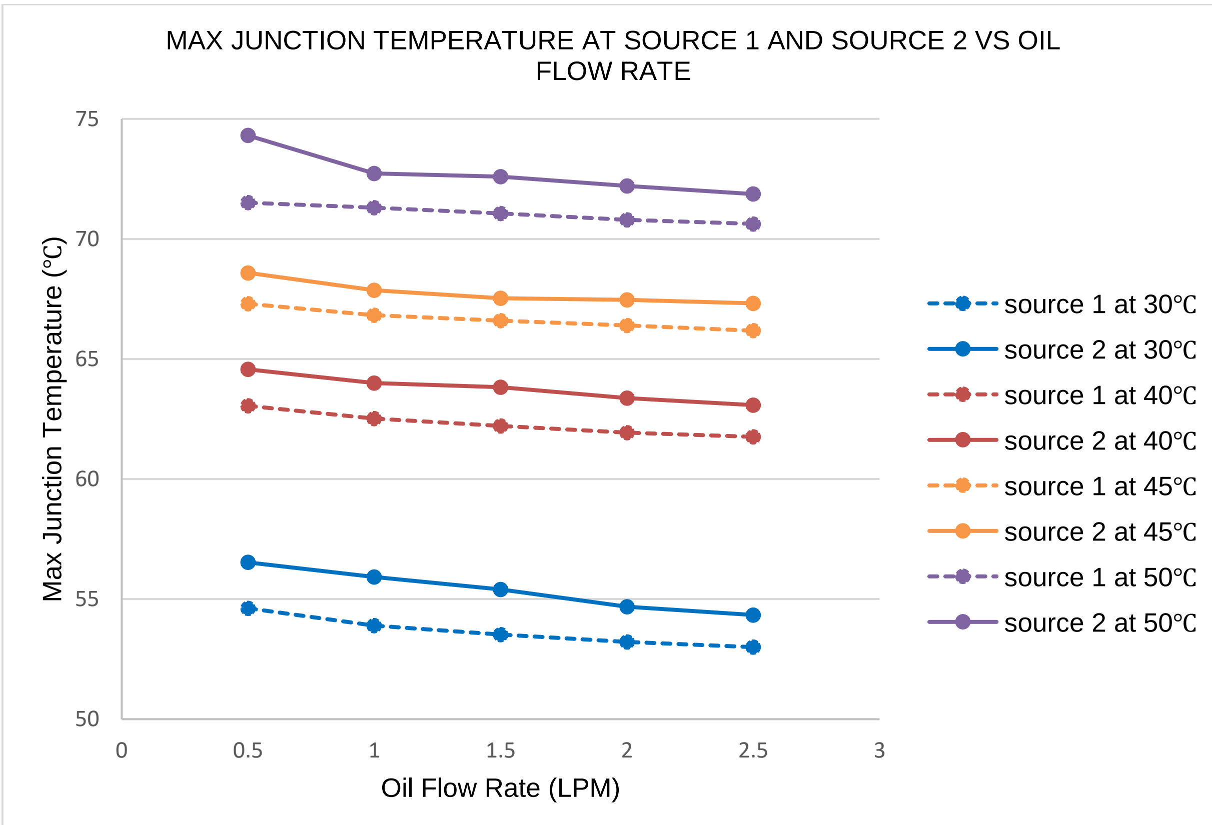

Figure 22 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for 2 Open Rack Unit Server using White Mineral Oil

Table 10 Temperature difference between source 1 and source 2 for 2 Open Rack Unit Server at different temperatures for 1LPM using White Mineral Oil

	Temperature at Source 1 (℃)	Temperature at Source 2 (℃)	Temperature Difference (℃)
At 30℃	54.62	56.53	1.91
At 40℃	63.05	64.57	1.52
At 45℃	67.3	68.59	1.29
At 50℃	71.51	74.31	2.8

It is clear from the above Table 10 that the temperature difference between source 1 and source 2 is less than 3℃. Whereas the temperature difference in an air-cooled server is 10.15℃. So, the impact of thermal shadowing in an Oil Cooled Server using White Mineral Oil can be neglected.

5.1.2.b Thermal Shadowing in 2 Open Rack Unit Server using Synthetic Fluid

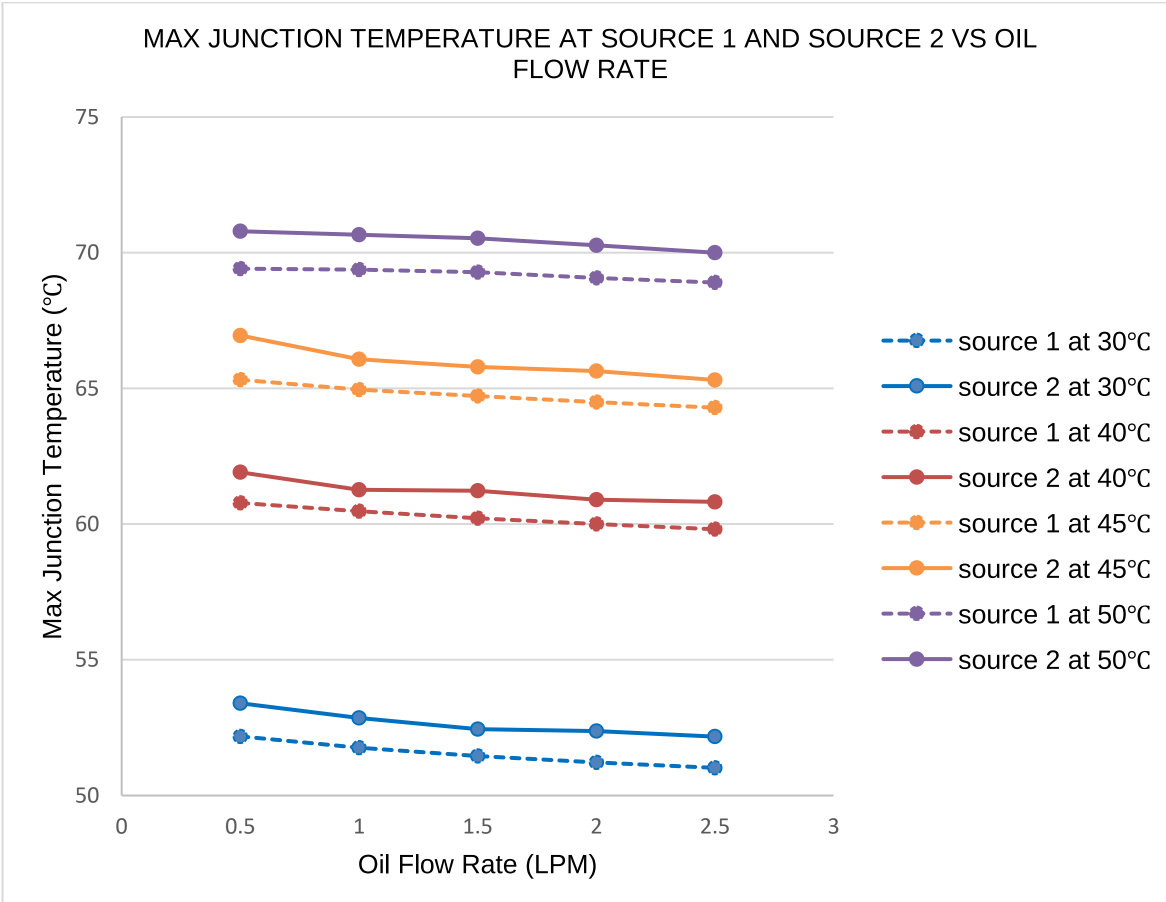

Figure 23 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for 2 Open Rack Unit Server using Synthetic Fluid

Table 11 Temperature difference between source 1 and source 2 for 2 Open Rack Unit Server for 1LPM at different temperatures using Synthetic Fluid

	Temperature at Source 1 (℃)	Temperature at Source 2 (℃)	Temperature Difference (℃)
At 30℃	52.18	53.4	1.22
At 40℃	60.78	61.91	1.13
At 45℃	65.32	66.95	1.63
At 50℃	69.41	70.79	1.38

It is clear from the above Table11 that the temperature difference between source1 and source2 is less than 2℃. Whereas the temperature difference in an air-cooled server is 10.15℃. So, the impact of thermal shadowing in an Oil Cooled Server using Synthetic Fluid can be neglected.

5.1.2.c Thermal Shadowing in 1.5 Open Rack Unit Server using White Mineral Oil

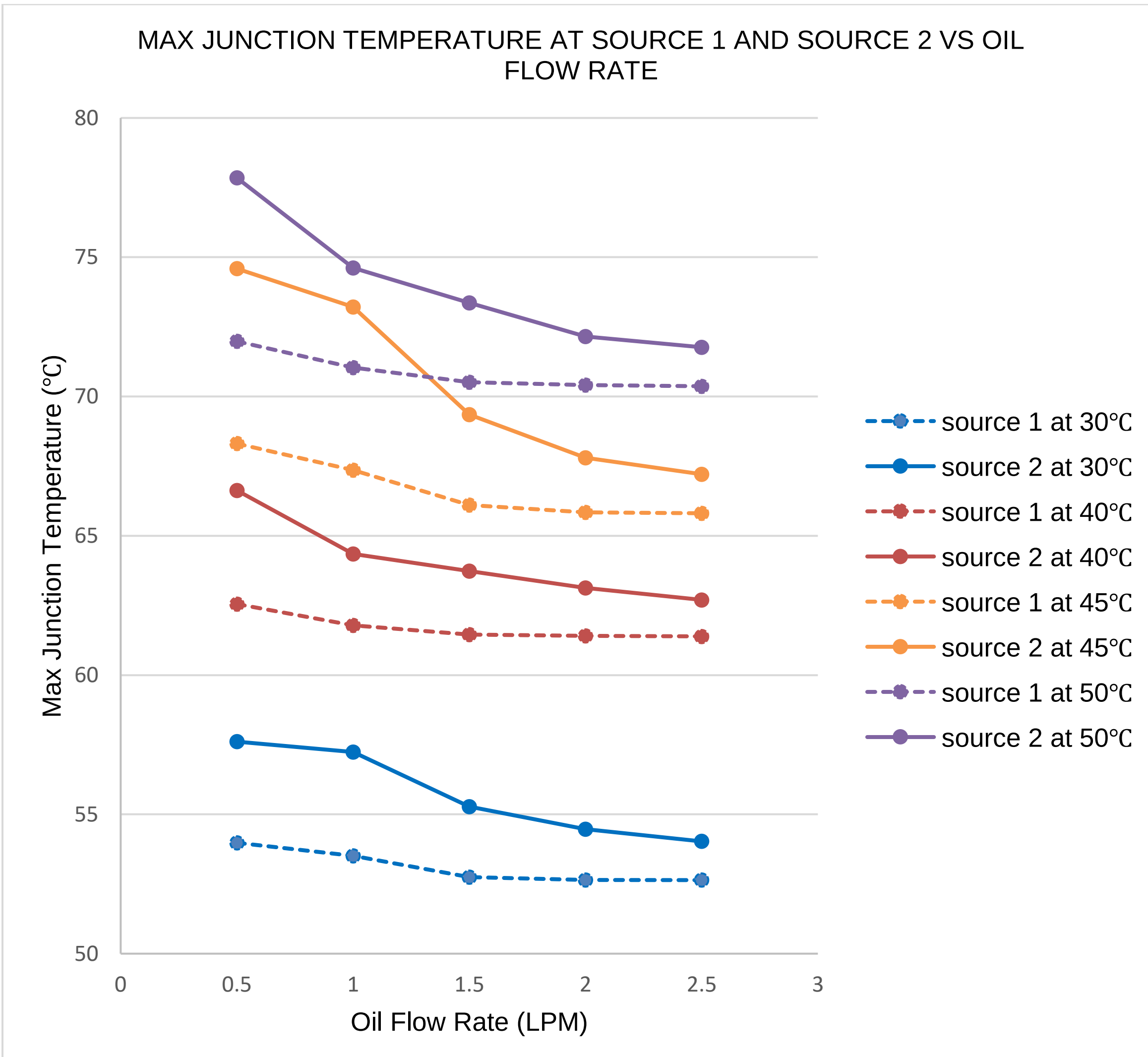

Figure 24 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for 1.5 Open Rack Unit Server using White Mineral Oil

Table 12 Temperature difference between source 1 and source 2 for 1.5 Open Rack Unit Server for 1LPM at different temperatures using White Mineral Oil

	Temperature at Source 1 (℃)	Temperature at Source 2 (℃)	Temperature Difference (℃)
At 30℃	53.98	57.61	3.63
At 40℃	62.55	66.63	4.08
At 45℃	68.31	74.59	6.28
At 50℃	71.98	77.85	5.87

It is clear from the above Table 12 that the temperature difference between source 1 and source 2 is less than 7℃. Whereas the temperature difference in an air-cooled server is 10.15℃. So, the impact of thermal shadowing in an Oil Cooled Server using White Mineral Oil can be neglected.

5.1.2.d Thermal Shadowing in 1.5 Open Rack Unit Server using Synthetic Fluid

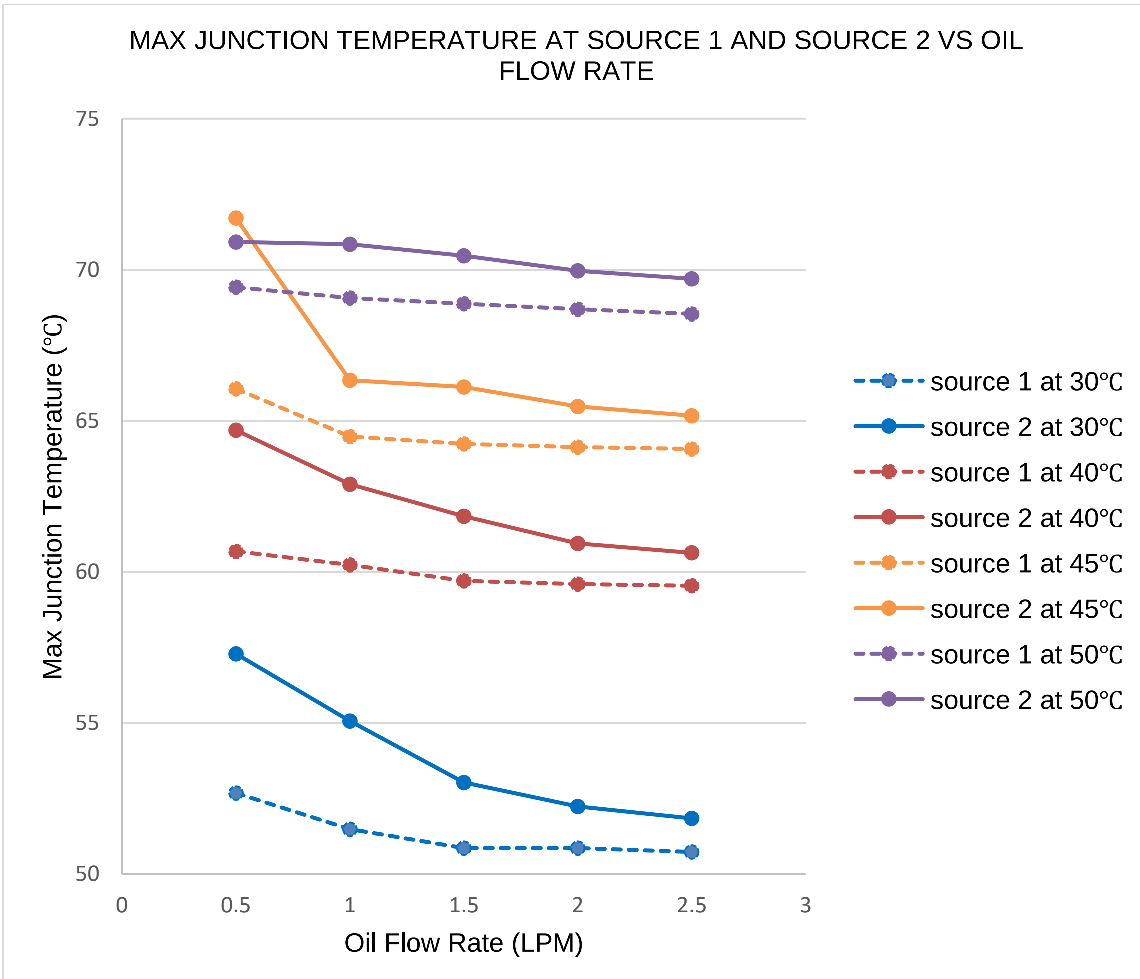

Figure 25 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for 1.5 Open Rack Unit Server using Synthetic Fluid

Table 13 Temperature difference between source 1 and source 2 for 1.5 Open Rack Unit Server for 1LPM at different temperatures using Synthetic Fluid

	Temperature at Source 1 (℃)	Temperature at Source 2 (℃)	Temperature Difference (℃)
At 30℃	52.68	57.29	4.61
At 40℃	60.68	64.69	4.01
At 45℃	66.06	71.71	5.65
At 50℃	69.42	70.92	1.5

It is clear from the above Table 13 that the temperature difference between source 1 and source 2 is less than 6℃. Whereas the temperature difference in an air-cooled server is 10.15℃. So, the impact of thermal shadowing in an Oil Cooled Server using Synthetic Fluid can be neglected.

5.1.2.e Thermal Shadowing in 1 Open Rack Unit Server using White Mineral Oil

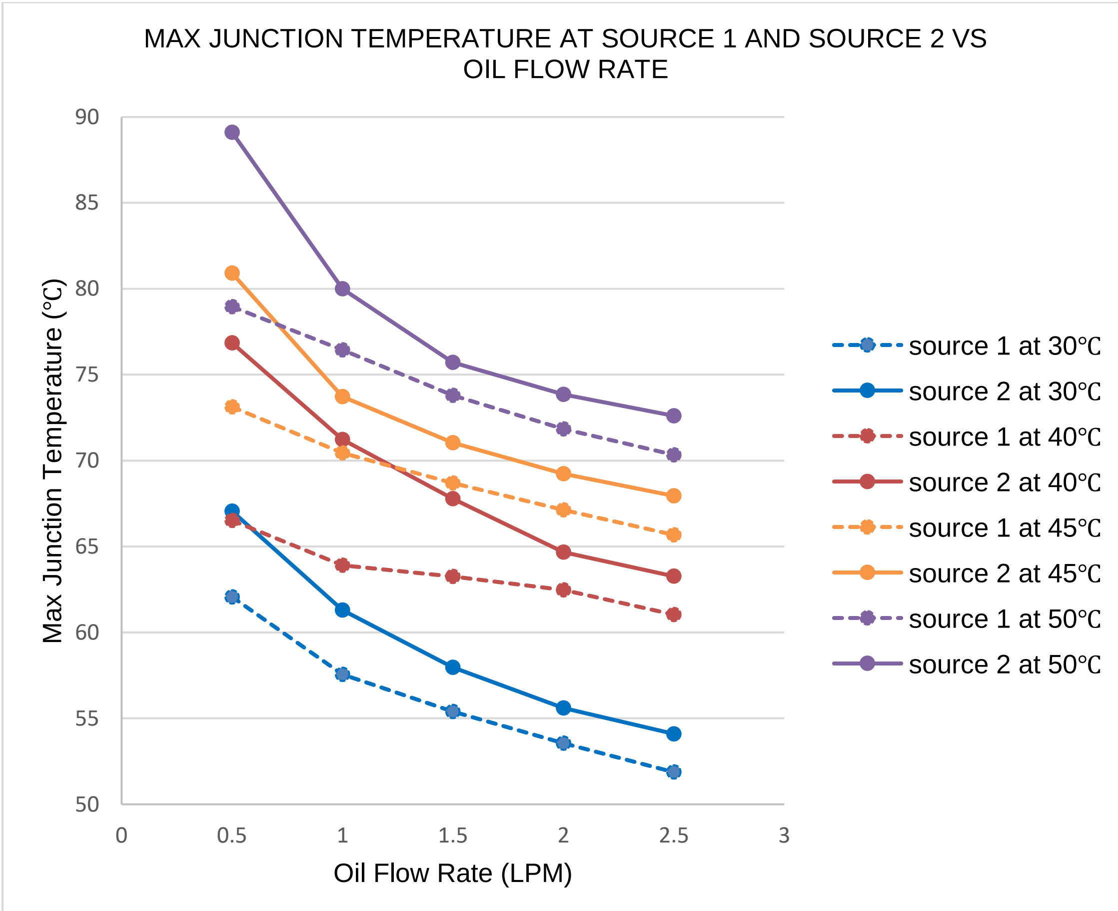

Figure 26 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for 1 Open Rack Unit Server using White Mineral Oil

Table 14 Temperature difference between source 1 and source 2 for 1 Open Rack Unit Server for 1LPM at different temperatures using White Mineral Oil

	Temperature at Source 1 (℃)	Temperature at Source 2 (℃)	Temperature Difference (℃)
At 30℃	62.07	67.06	4.99
At 40℃	66.52	76.85	10.33
At 45℃	73.13	80.91	7.78
At 50℃	78.96	89.11	10.15

It is clear from the above Table 14 that the temperature difference between source 1 and source 2 is less than 10.3℃. Whereas the temperature difference in an air-cooled server is 10.15℃. So, the impact of thermal shadowing in an Oil Cooled Server using White Mineral Oil can be neglected as it is similar to that of an air-cooled server having directed flow using ducting system.

5.1.2.f Thermal Shadowing in 1 Open Rack Unit Server using Synthetic Fluid

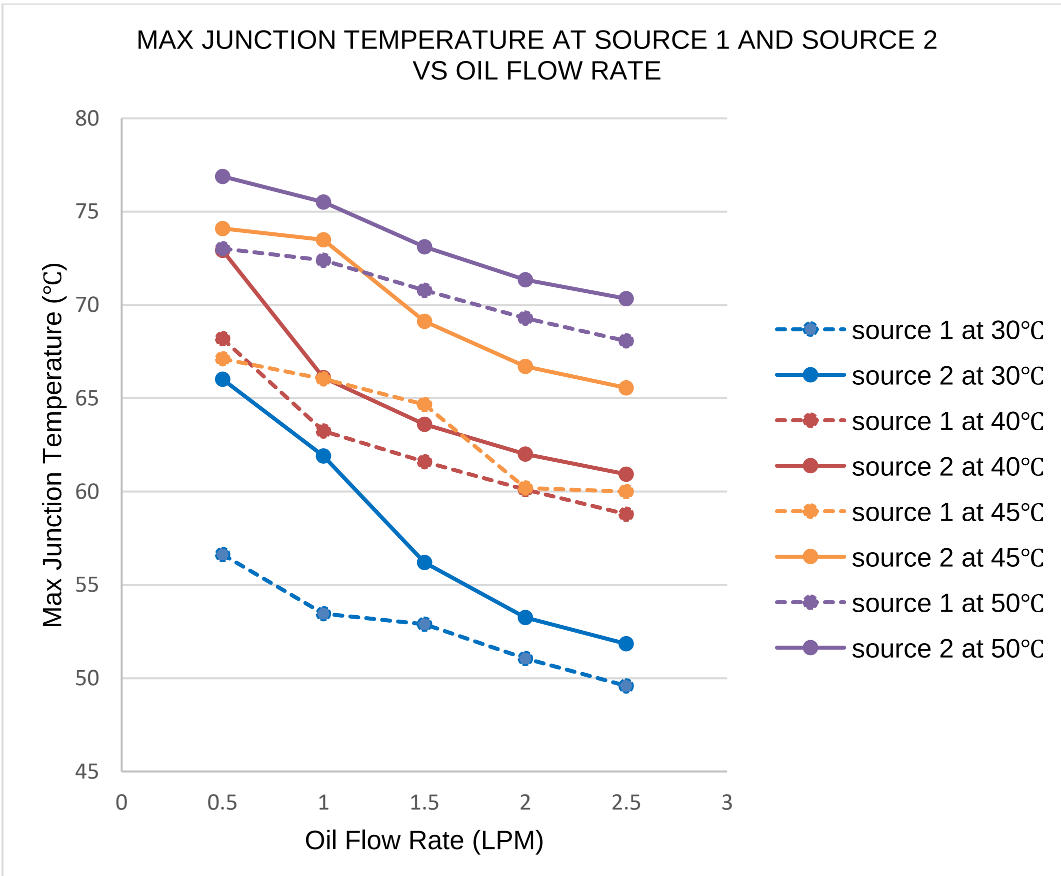

Figure 27 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for 1 Open Rack Unit Server using Synthetic Fluid

Table 15 Temperature difference between source 1 and source 2 for 1 Open Rack Unit Server for 1LPM at different temperatures using Synthetic Fluid

	Temperature at Source 1 (℃)	Temperature at Source 2 (℃)	Temperature Difference (℃)
At 30℃	56.63	66.02	9.39
At 40℃	68.19	72.92	4.73
At 45℃	67.11	74.1	6.99
At 50℃	73.02	76.89	3.87

It is clear from the above Table 15 that the temperature difference between source 1 and source 2 is less than 9.39℃. Whereas the temperature difference in an air-cooled server is 10.15℃. So, the impact of thermal shadowing in an Oil Cooled Server using Synthetic fluid can be neglected as it is like that of an air-cooled server having directed flow using ducting system.

5.1.3 Comparison of Impact of Thermal Shadowing in 1 Open Rack Unit, 1.5 Open Rack Unit and 2 Open Rack Unit Servers at 30℃ using White Mineral Oil and Synthetic Fluid

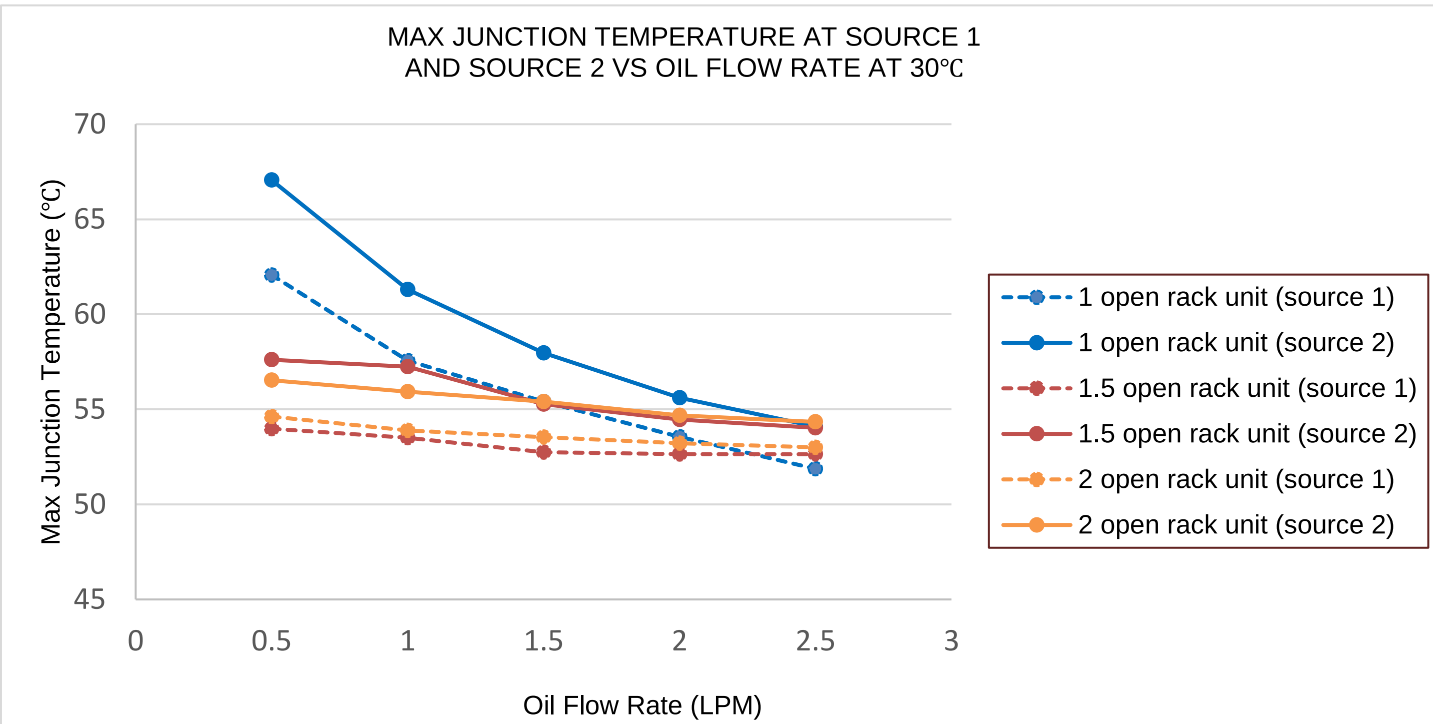

Figure 28 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for different Open rack unit servers at 30℃ using White Mineral Oil

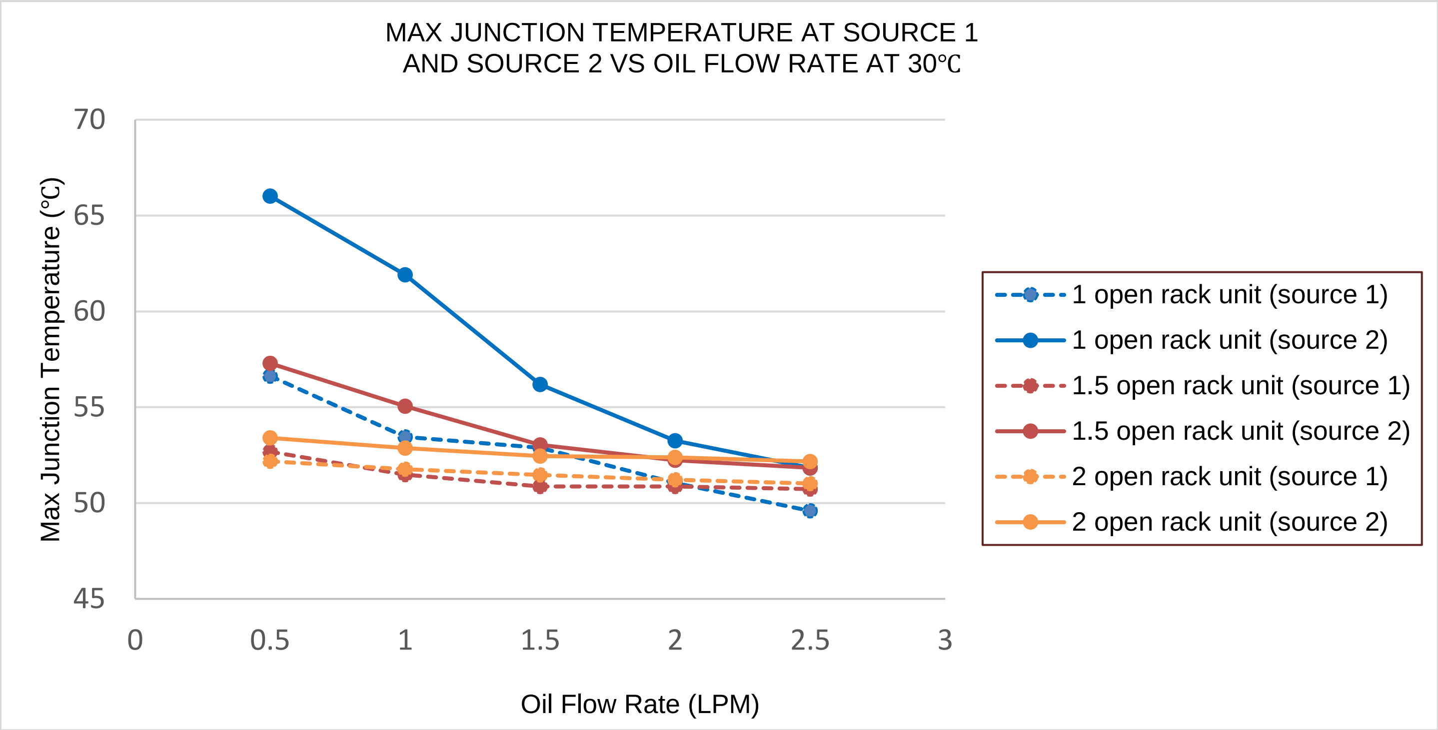

Figure 29 Maximum Junction Temperature at source 1 and source 2 vs Oil flow rate for different Open rack unit servers at 30℃ using Synthetic Fluid

Using White Mineral Oil, From the figure 28, the Impact of thermal shadowing can be neglected for all kinds of Open Rack Unit Servers. We can observe that the temperature difference between the source 1 and source 2 for 2 Open Rack Unit Server is less compared to that of 1.5 Open Rack Unit Server and 1 Open Rack Unit Server.

Similarly using Synthetic Fluid, From the figure 29, the Impact of thermal shadowing can be neglected for all kinds of Open Rack Unit Servers. We can observe that the temperature difference between the source 1 and source 2 for 2 Open Rack Unit Server is less compared to that of 1.5 Open Rack Unit Server and 1 Open Rack Unit Server.

Table 16 Temperature difference for Air cooled server and Oil Cooled Server using different fluids at 30℃, 1LPM of 2 Open Rack Unit Server

Server	Temperature Difference (℃)
Air cooled Server	10.15
Oil Cooled Server using White Mineral Oil	1.91
Oil Cooled Server using Synthetic Fluid	1.22

From table 16, the impact of thermal shadowing can be neglected for Oil cooled servers when compared to that of an air-cooled server. It is also clear that the temperature difference is even lesser for Synthetic Fluid when compared to White Mineral Oil.

5.2 Optimization of Heat Sinks

5.2.1 Results with Existing Parallel Plate Heat Sink

For 1 Open Rack Unit Server, at 30℃, 1LPM, with power 65W, with fin count 35 and Thickness of 0.3mm, the Temperature and Thermal Resistance are as follows.

Table 17 Temperature and Thermal Resistance of Source 1 and Source 2 for Mineral Oil and Synthetic Fluid with an existing parallel plate Heat Sink

	Temperature at Source 1	Temperature at Source 2	Thermal Resistance at Source 1	Thermal Resistance at Source 2
Mineral Oil	49.19	48.82	0.3093	0.3181
Synthetic Fluid	46.86	46.67	0.2736	0.2792

From the above data, we can calculate the volume by

Volume = Thickness x Height x Length

Volume = 0.3 x 41 x 110 = 1353 mm^3

For 35 fins,

Volume = 35 x 1353

= 47355 mm^3

5.2.2 Results with Optimized Parallel Plate Heat Sink

The number of fins is reduced to 15, with the same boundary conditions as that of the existing parallel plate heat sink. The fin count is kept constant as 15 and the height is kept constant as 41mm, the dimension of the baseline model. The plate thickness is varied from 0.25mm to 3.25mm. After conducting parametric study for this condition, we get Thermal resistance as follows.

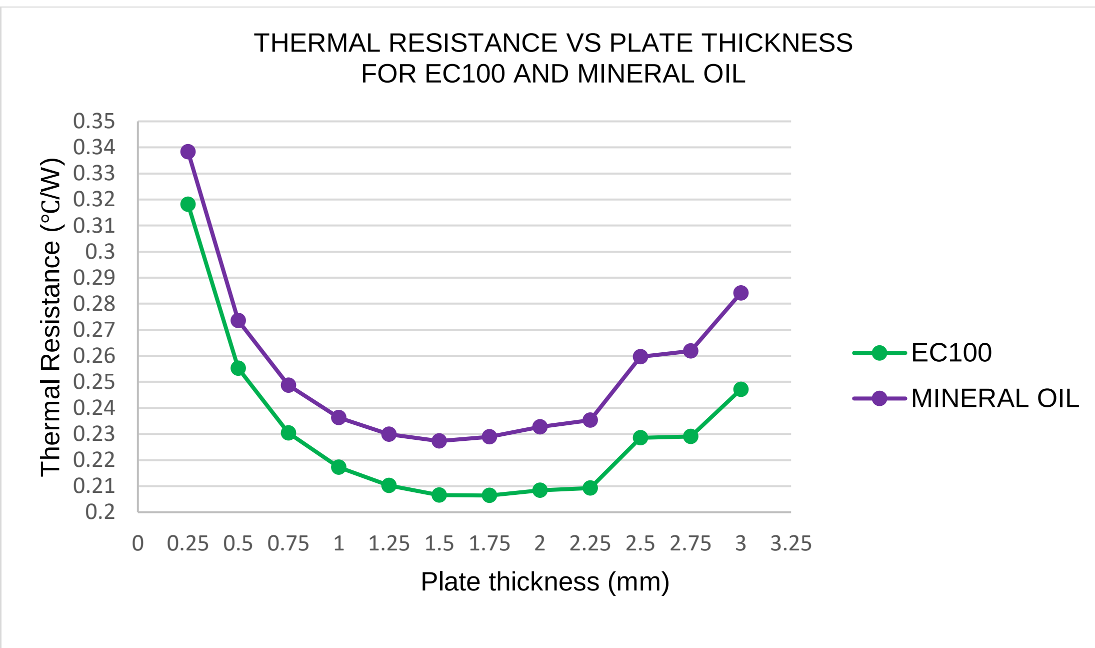

Figure 30 Graph of Thermal Resistance vs Plate thickness for Mineral Oil and Synthetic Fluid of an Optimized Parallel Plate Heat Sink

It is clear from the Figure 30 that the Thermal Resistance decreases for both White Mineral Oil and Synthetic Fluid when the thickness is from 0.25mm to 1.5mm. So, any thickness in between 0.25mm to 1.5mm can be considered for the study. In this study, the thickness of 0.25mm is considered. Now keeping the plate thickness 0.25mm as constant, the plate height is varied from 20mm to 41mm. The thermal resistance curve for this height parametric study is shown below.

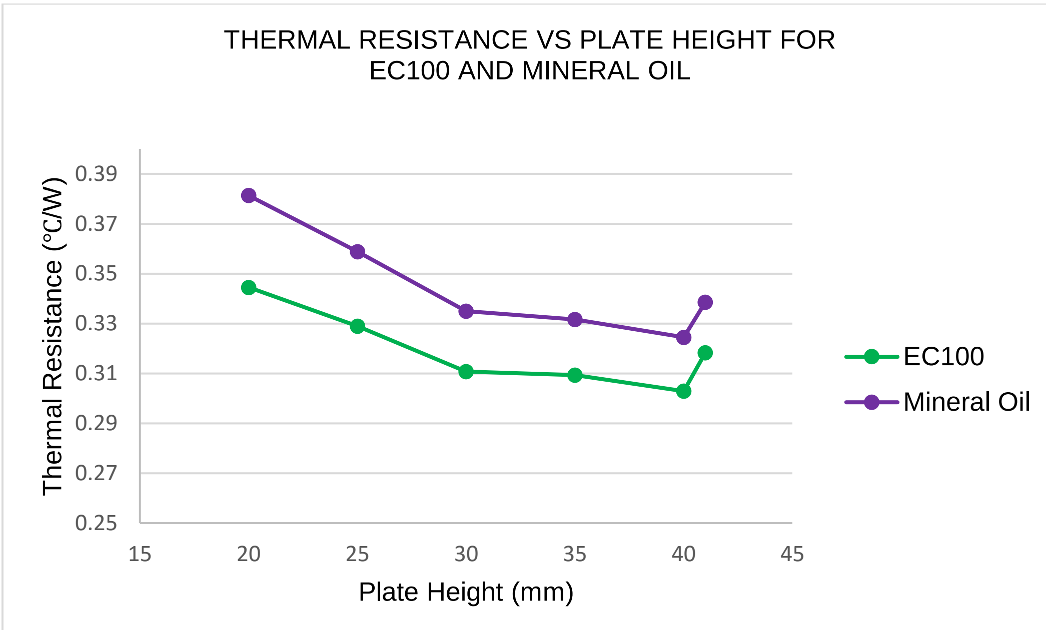

Figure 31 : Graph of Thermal Resistance vs Plate height for Mineral Oil and Synthetic Fluid of an Optimized Parallel Plate Heat Sink

It is clear from the Figure 31 that the Thermal Resistance decreases for both White Mineral Oil and Synthetic Fluid when the height is from 20mm to 40mm. So, any height in between 20mm to 40mm can be considered for the study. In this study, the height of 20mm is considered. Now the simulation is performed for this optimized model of plate thickness 0.25mm and plate height 20mm and is compared with the results of an existing parallel plate heat sink.

Table 18 Temperature and Thermal Resistance of Source 1 and Source 2 for Mineral Oil and Synthetic Fluid with an optimized parallel plate Heat Sink

	Temperature at Source 1	Temperature at Source 2	Thermal Resistance at Source 1	Thermal Resistance at Source 2
Mineral Oil	51.63	52.57	0.3813	0.3659
Synthetic Fluid	49.49	50.32	0.3445	0.3303

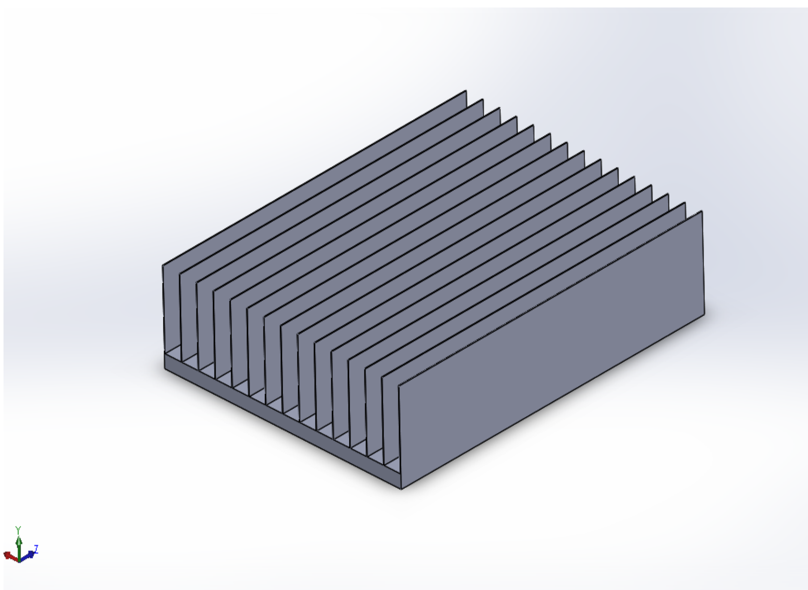

Figure 32 CAD model of an optimized parallel plate heat sink

Table 19 Volumetric comparison of existing parallel plate heat sink and optimized parallel plate heat sink

	Existing Parallel Plate heat sink dimensions	Optimized Parallel Plate heat sink dimensions
Thickness	0.3 mm	0.25 mm
Height	41 mm	20 mm
Length	110 mm	110 mm
Volume	Thickness x Height x Length =1353 mm^3	Thickness x Height x Length =550 mm^3
Number of fins	35	15
Volume for total fins	35 x 1353 =47355 mm^3	15 x 550 =8250 mm^3

It is clear from the Table 19 that the volume of the optimized model is very less when compared with the base model. The volume is almost decreased by 5 times, which implies that the material used is reduced by 5 times. So, the material cost can be reduced significantly. Also, from Table 18, It is clear that the temperature and thermal resistance are within the specified range.

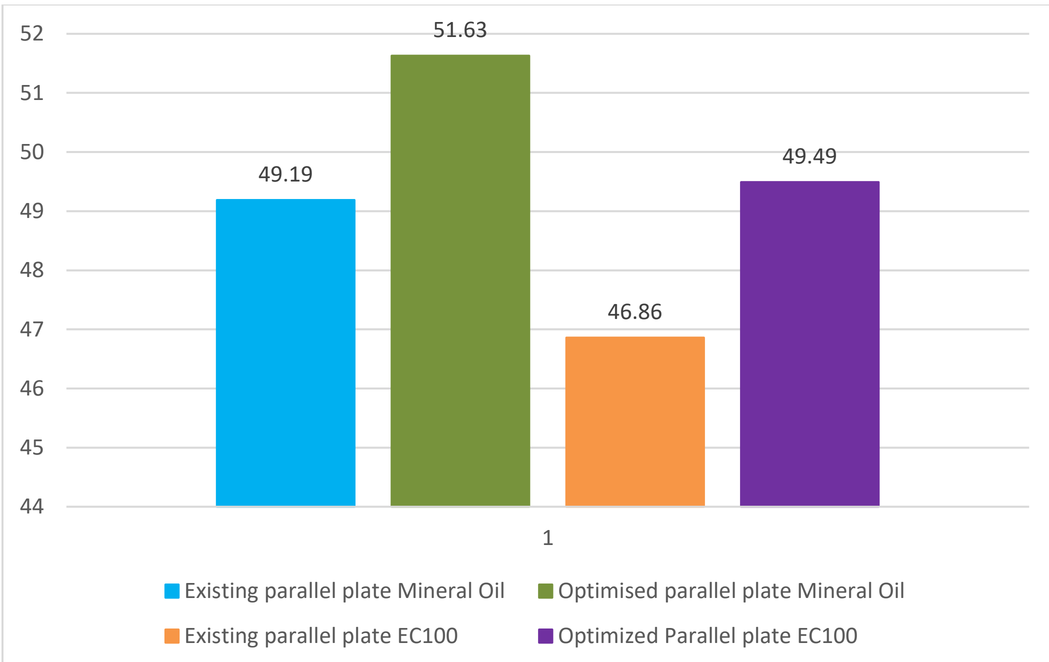

Figure 33 Comparison of Temperature at Source 1 for an existing and optimized parallel plate model using white mineral oil and synthetic fluid.

From Figure 33, It can be seen that there is very less variation in Temperature. Although there is increase in temperature of Optimized model, the rise of temperature is less than 5℃. This can be neglected as the temperature is less than the Max Junction Temperature specified by INTEL for 65W power. So, the optimized heatsink has better thermal performance and the material cost can be reduced.

5.2.3 Results with Optimized Plate fin Heat Sink

The parallel plates are replaced with plate fins in this study. The boundary conditions are kept same as that of the existing parallel plate heat sink. The plate fin count is kept constant as 42 and the height is kept constant as 41mm, the dimension of the baseline model. The plate thickness is varied from 4mm to 10mm. After conducting parametric study for this condition, we get Thermal resistance as follows.

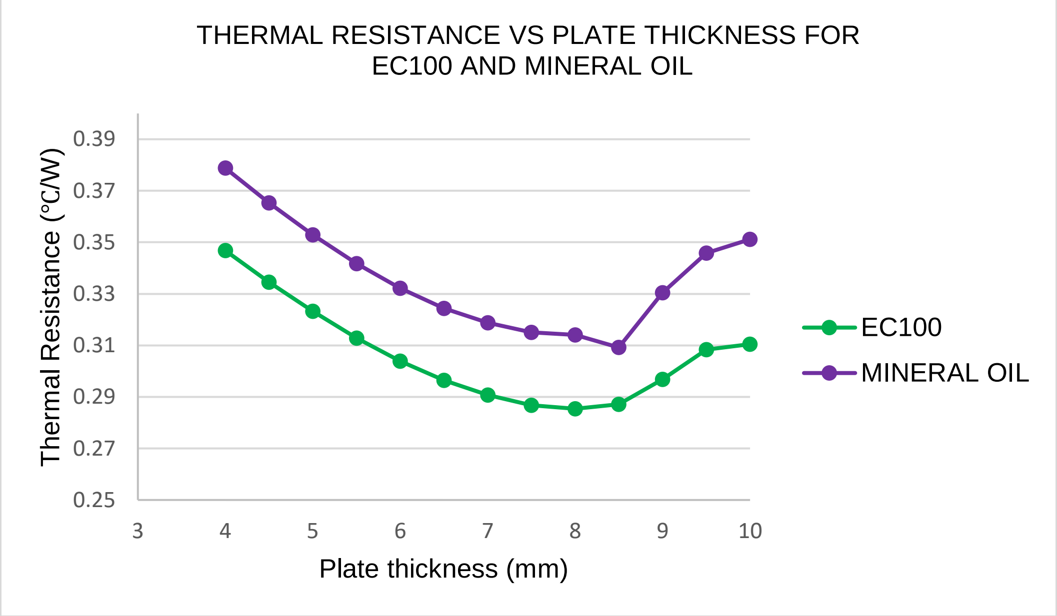

Figure 34 Graph of Thermal Resistance vs Plate thickness for Mineral Oil and Synthetic Fluid of an Optimized Plate fin Heat Sink

It is clear from the Figure 34 that the Thermal Resistance decreases for both White Mineral Oil and Synthetic Fluid when the thickness is from 4mm to 7.5mm. So, any thickness in between 4mm to 7.5mm can be considered for the study. In this study, the thickness of 4mm is considered. Now keeping the plate thickness 4mm as constant, the plate height is varied from 20mm to 41mm. The thermal resistance curve for this height parametric study is shown below.

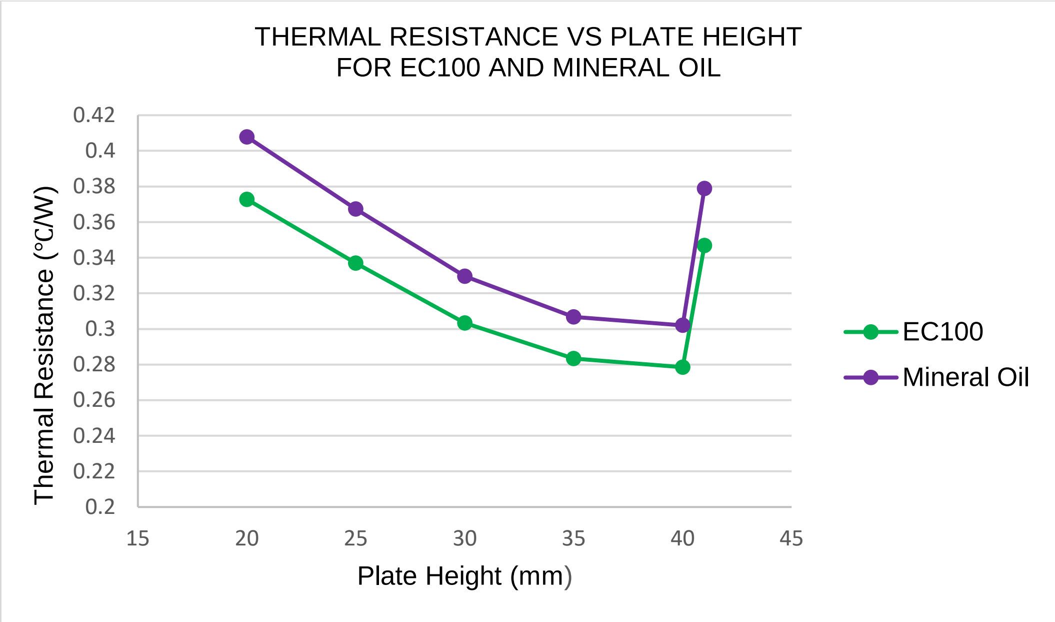

Figure 35 Graph of Thermal Resistance vs Plate height for Mineral Oil and Synthetic Fluid of an Optimized Plate fin Heat Sink

It is clear from the Figure 35 that the Thermal Resistance decreases for both White Mineral Oil and Synthetic Fluid when the height is from 20mm to 40mm. So, any height in between 20mm to 40mm can be considered for the study. In this study, the height of 20mm is considered. Now the simulation is performed for this optimized model of plate thickness 4mm and plate height 20mm and is compared with the results of an existing parallel plate heat sink.

Table 20 Temperature and Thermal Resistance of Source 1 and Source 2 for Mineral Oil and Synthetic Fluid with an optimized plate fin Heat Sink

	Temperature at Source 1	Temperature at Source 2	Thermal Resistance at Source 1	Thermal Resistance at Source 2
Mineral Oil	54.14	53.51	0.4079	0.428
Synthetic Fluid	51.79	51.38	0.3728	0.3878

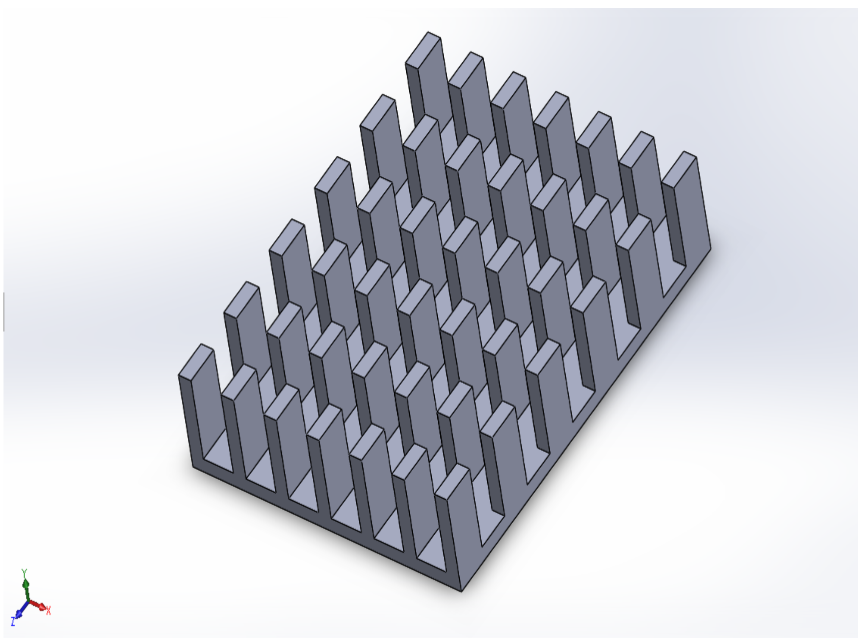

Figure 36 CAD model of an optimized plate fin heat sink

Table 21 Volumetric comparison of existing parallel plate heat sink and optimized plate fin heat sink

	Existing Parallel Plate heat sink dimensions	Optimized Plate fin heat sink dimensions
Thickness	0.3 mm	4 mm
Height	41 mm	20 mm
Length	110 mm	10 mm
Volume	Thickness x height x length =1353 mm^3	Thickness x height x length =800 mm^3
Number of fins	35	42
Volume for total fins	35 x 1353 =47355 mm^3	42 x 800 =33600 mm^3

It is clear from the Table 21 that the volume of the optimized model is very less when compared with the base model. So, the material cost can be reduced significantly. Also, from Table 20, It is clear that the temperature and thermal resistance are within the specified range.

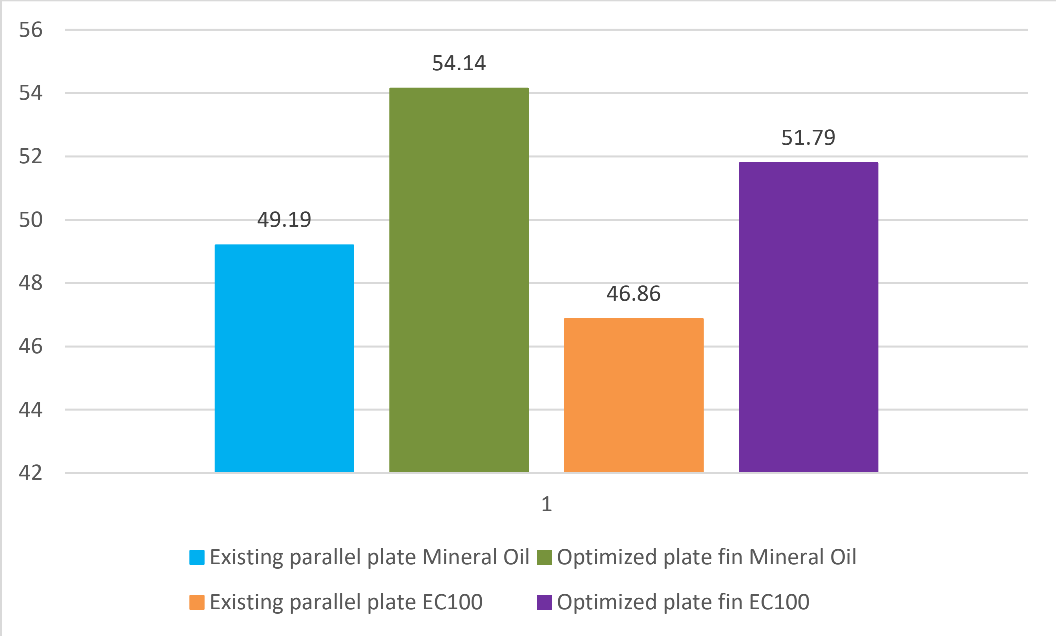

Figure 37 Comparison of Temperature at Source 1 for an existing and optimized plate fin model using white mineral oil and synthetic fluid.

From Figure 37, It can be seen that there is very less variation in Temperature. Although there is increase in temperature of Optimized model, the rise of temperature is less than 7℃. This can be neglected as the temperature is less than the Max Junction Temperature specified by INTEL for 65W power. So, the optimized heatsink has better thermal performance and the material cost can be reduced.

5.2.4 Results with Optimized Pin fin Heat Sink

The parallel plates are replaced with pin fins in this study. The boundary conditions are kept same as that of the existing parallel plate heat sink. The pin fin count is kept constant as 25 and the height is kept constant as 41mm, the dimension of the baseline model. The pin radius is varied from 2.5mm to 8mm. After conducting parametric study for this condition, we get Thermal resistance as follows.

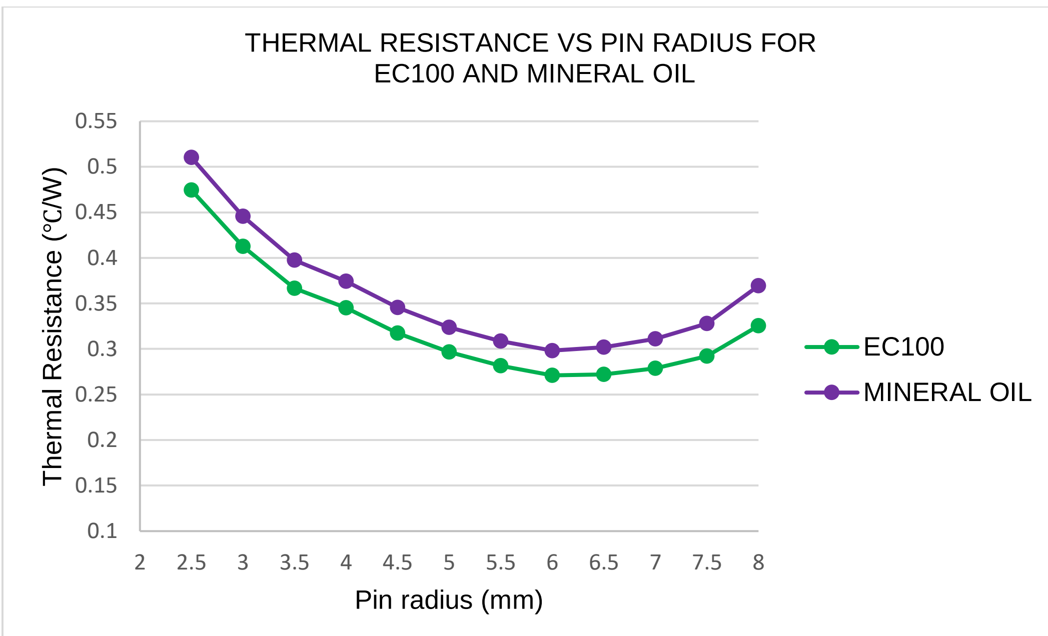

Figure 38 Graph of Thermal Resistance vs Pin radius for Mineral Oil and Synthetic Fluid of an Optimized Pin fin Heat Sink

It is clear from the Figure 38 that the Thermal Resistance decreases for both White Mineral Oil and Synthetic Fluid when the radius is from 2.5mm to 6mm. So, any radius in between 2.5mm to 6mm can be considered for the study. In this study, the radius of 3mm is considered. Now keeping the pin radius 3mm as constant, the pin height is varied from 20mm to 41mm. The thermal resistance curve for this height parametric study is shown below.

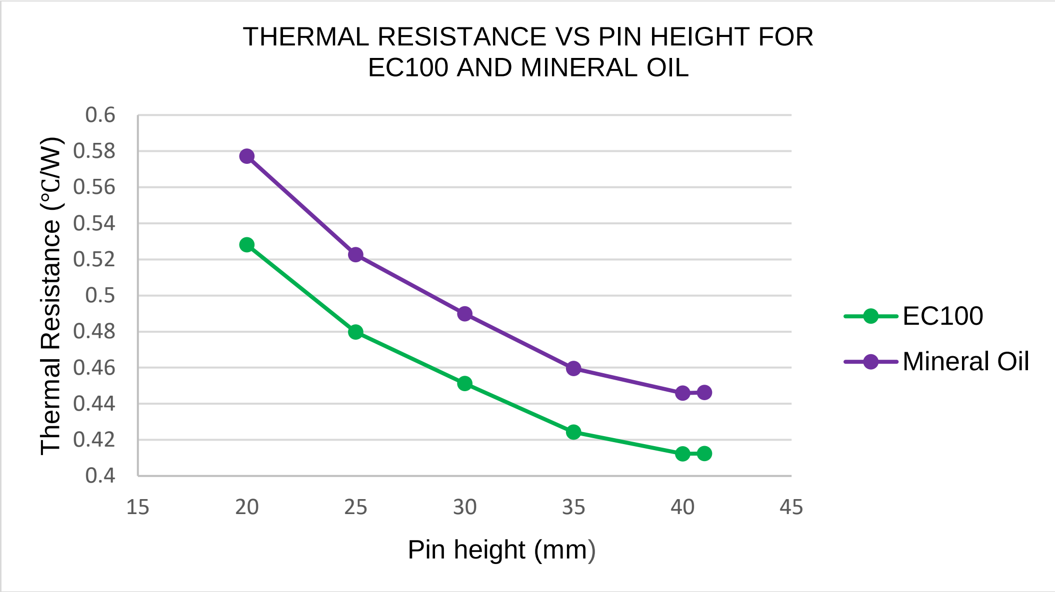

Figure 39 Graph of Thermal Resistance vs Pin height for Mineral Oil and Synthetic Fluid of an Optimized Pin fin Heat Sink

It is clear from the Figure 39 that the Thermal Resistance decreases for both White Mineral Oil and Synthetic Fluid when the height is from 20mm to 40mm. So, any height in between 20mm to 40mm can be considered for the study. In this study, the height of 20mm is considered. Now the simulation is performed for this optimized model of pin radius 3mm and pin height 20mm and is compared with the results of an existing parallel plate heat sink.

Table 22 Temperature and Thermal Resistance of Source 1 and Source 2 for Mineral Oil and Synthetic Fluid with an optimized pin fin Heat Sink

	Temperature at Source 1	Temperature at Source 2	Thermal Resistance at Source 1	Thermal Resistance at Source 2
Mineral Oil	61.31	60.94	0.5772	0.6022
Synthetic Fluid	58.36	58.1	0.5281	0.5488

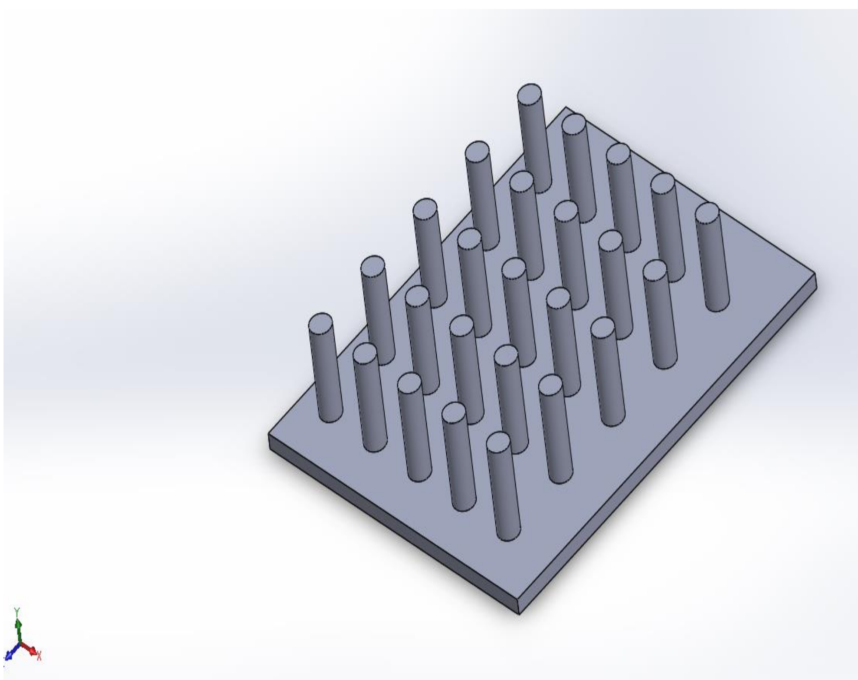

Figure 40 CAD model of an optimized pin fin heat sink

Table 23 Volumetric comparison of existing parallel plate heat sink and optimized pin fin heat sink

	Existing Parallel Plate heat sink dimensions	Optimized Plate fin heat sink dimensions
Thickness	0.3 mm	-
Height	41 mm	20 mm
Length	110 mm	-
Radius	-	3 mm
Volume	Thickness x height x length =1353 mm^3	Π x $radius^2$ x height =565.2 mm^3
Number of fins	35	25
Volume for total fins	35 x 1353 =47355 mm^3	25 x 565.2 =14130 mm^3

It is clear from the Table 23 that the volume of the optimized model is very less when compared with the base model. So, the material cost can be reduced significantly. Also, from Table 22, It is clear that the temperature and thermal resistance are within the specified range.

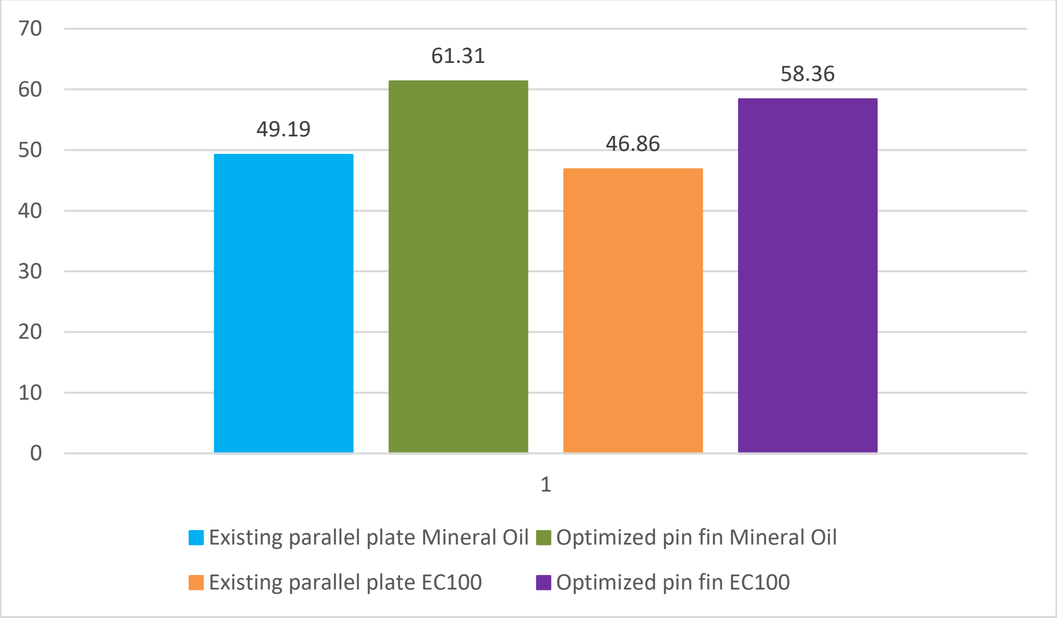

Figure 41 Comparison of Temperature at Source 1 for an existing and optimized pin fin model using white mineral oil and synthetic fluid.

From Figure 41, It can be seen that there is very less variation in Temperature. Although there is increase in temperature of Optimized model, the rise of temperature is less than 10℃. This can be neglected as the temperature is less than the Max Junction Temperature specified by INTEL for 65W power. So, the optimized heatsink has better thermal performance and the material cost can be reduced.

Chapter 6

CONCLUSIONS

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

- It is clear from the study that the Impact of Thermal Shadowing of the Oil Cooled Server can be neglected as the Temperature difference between Source 1 and Source 2 is 3℃ or lesser which can be neglected when compared to Air Cooled Server.
- So, we can conclude that the ducting system can be removed in Oil Cooled Server.
- Savings in power consumption by the 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.
- Also, the Impact of Thermal Shadowing of the EC100 is even lesser compared to White Mineral Oil.
- There is no significant Temperature change for Pin fin heat sink and Plate fin heat sink when compared with the existing parallel plate heat sink.
- As the geometry of parallel plate heat sink is complex, if we replace that with either Pin Fin or Plate Fin heat sink, material cost can be reduced.

Chapter 7

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Biographical Information

Ravya Dandamudi was born in Andhra Pradesh, India. She received her Bachelor of Engineering in Mechanical Engineering with an excellent grade of 4 GPA from Velagapudi Ramakrishna Siddhartha Engineering College, Andhra Pradesh, India in the year 2016. She started her Master of Science from University of Texas at Arlington in Fall 2016. During her master's program she conducted her research in the field of thermal management under Dr. Dereje Agonafer. She has indulged herself in various industry collaborated projects and gained extensive experience working in laboratory environment. She has worked with computational characterization of rack mount servers. Ravya received her Master of Science in Mechanical Engineering from University of Texas at Arlington in May 2018.