Design and CFD analysis of a tank solution for natural convection based single-phase immersion cooling of servers

By

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Abstract

DESIGN AND CFD ANALYSIS OF A TANK SOLUTION FOR NATURAL CONVECTION BASED SINGLE-PHASE IMMERSION COOLING OF SERVERS

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The issue of thermal management of data centers is becoming even more challenging with increasing chip power densities. This has been primarily driven by increasing demands in artificial intelligence and machine learning-based applications, increased usage of cloud-based services, and high-performance computing. The development of these applications has led to a corresponding increment in high performance and high power density CPUs and GPUs. Liquid cooling technologies are outperforming the traditional air cooling approach in dissipating high heat fluxes. Out of these technologies, single-phase immersion cooling offers advantages like substantially lower PUE values, simplified data center infrastructure, ease of deployment for edge data centers, improved reliability, and even temperature profile on the servers. This study focuses on the development of a natural convection-based single-phase immersion cooling solution for edge data centers. Eight servers were immersed in a tank filled with mineral oil were analyzed using CFD. The tank is directly cooled with integrated heat

exchanging plates to cool down the hot fluid. The inlet flow rate and temperatures of the coolant in the plates were varied to determine the optimal value of the required flow rate. The baseline results of the air-cooled heat sink were also compared with the improved heat sink better suited for natural convection. The CPU temperatures and the temperatures of auxiliary components on the motherboard were also analyzed. A pumping power comparison of the baseline case where the entire tank is cooled using forced convection was also performed.

Table of Contents

List of Illustrations

List of Tables

List of Equations

Chapter 1

INTRODUCTION

1.1 Introduction to Data Center

A data center is a department in an enterprise that comprises of multiple elements which provide a safe, secure location for data storage. Most of IT operation depends upon these data centers and it has become a crucial part of the business. Any downtime in these systems can lead to loss of time and money. In earlier days, only one supercomputer was used for processing the data. As the size of the equipment shrunk and got cheaper and the demand for processing increased, multiple server clusters were used. These servers can be called as industrial counterparts to our home computers. The clustered servers can be housed in a room, entire building, or group of buildings or can be connected to communication networks so that they can also be accessed remotely. As the need for processing increases, thermal management of these clusters has become important.

1.2 Data Center Cooling Methods

Common techniques used for cooling of data centers are

1. Air cooled servers
2. Liquid cooled servers

1.2.1 Air Cooled Servers

This is the most widely used technique used to cool datacenters. In this technique, the air is forced through the heated IT equipments. The system utilized in this procedure is the forced convection of air over the heat sink [7]. The heat sink is a kind of heat exchanger which removes heat from mechanical or electrical equipment through conduction and transfers it to a fluid medium. Conditioned air is supplied inside the data center depending upon the heat of the IT components. The axial fan regulates the airflow from inlet to outlet in a data center. The computer Room Air Conditioning (CRAC) unit is responsible for cooling down the air to a suitable temperature [4]. The air picks up heat from the server components and moves to the hot aisle from where it enters the CRAC [5]. The cooled air from CRAC is supplied using elevated floor infrastructure to the cold aisle [6]. The detailed process of air cooled server can be shown in the figure below (1).

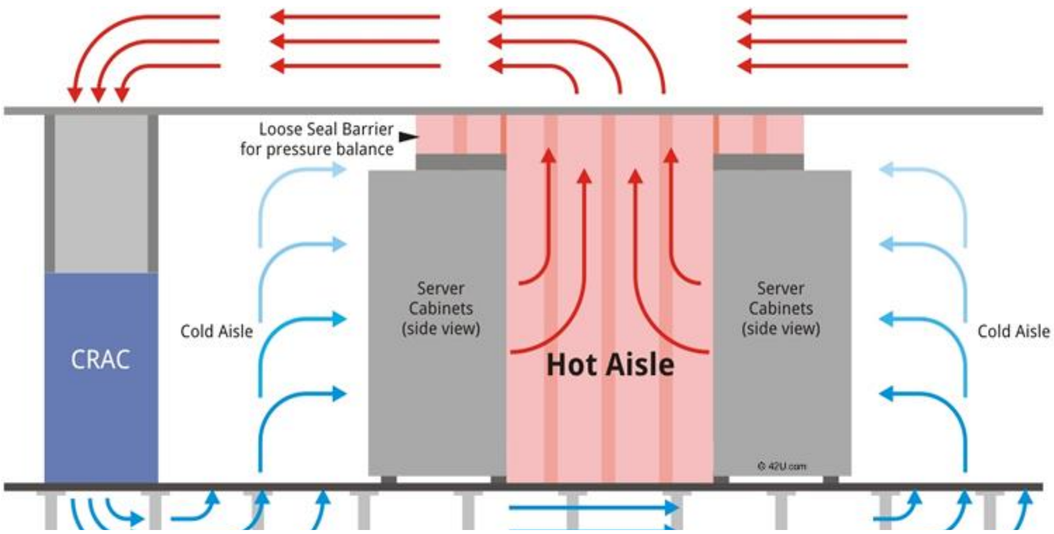

Figure 1 Hot Aisle / Cold Aisle Arrangement

1.2.2 Liquid Cooled Servers

As rack densities continue to increase and more heat is produced, pushing the air-cooled systems to their maximum economic capabilities. Due to high powered microprocessors and packing air cooled systems cannot provide the desired cooling for the ITE. Liquid cooling technique can be used to overcome the drawbacks and design constraints of air cooled servers. Air has poor conductivity of heat compared to that of a liquid, it requires extra attachments like fins to maximize surface area for heat transfer. Also, air carrying various contaminants can damage the IT systems [8]. Humidity in the air can cause lead to problems regarding corrosion [9]. Also, large infrastructure and space are required in air cooling as it has large fans and ducts. Therefore the industry is looking forward to this technique. 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 servers – immersion technique and In-direct immersion techniques. In the immersion technique, the servers are placed in a water bath for heat transfer. But as water is conductive it can damage the systems. There are also certain reliability issues in using this method. The other technique is In-direct systems cooling, in which a passive heat transfer cold plate can be used as shown in Figure (2). This type of cold plate is used to remove heat from the processor and transfer it to the flowing water. The bottom surface of the cold plate is made of copper which is housed on the CPU or heat producing component. Heat is conducted to the cold plate and the temperature of the cooper base increases [2]. This heat is then transferred to the water flowing in the cold plate through convection [16]. The flow rate of fluid entering coldplate can be controlled using a flow control device [17]. This heated water can then be cooled using a heat exchanger or chiller and can be recirculated.

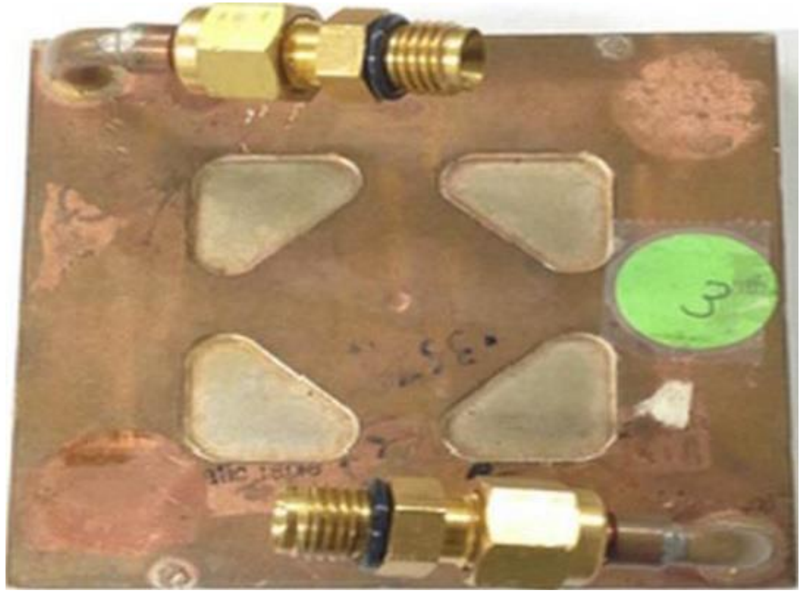

Figure 2 Cold Plate

1.2.2.b Dielectric fluid cooled servers

Even though water has better thermal conductivity than air, it cannot be allowed to come in direct contact with the servers. As water is conductive, which may cause damage to the servers. Therefore, a dielectric fluid having high heat carrying capacity can be used for immersion of servers [15]. Two phase immersion cooling also includes the use of dielectric fluid, where the phase change takes place of the dielectric fluid [3]. Mineral oil and synthetic fluid Electro Cool 100 (EC-100) are widely used and also considered for this study. The physical properties of different types of fluids are compared for this study [12].

Figure 3 Dielectric Fluid Immersed Server

Table 1 Properties of different fluid

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 can be seen from Table1 that the thermal mass (Density x Heat Capacity) of Dielectric liquid is high compared to air.

1.3 Motivation of the Work

In the current era of crypto-currency, 5G, AI and, IoT(Internet of Things) there is an ever increasing need for processing power which in turn is heating up the IT equipments more and more. To satisfy this demand of cooling of high performance servers, immersion cooling is one of the developing strategies. There were very little available research found which considered immersion cooling using natural convection. One of the studies based on such a setup focused on servers that were powered at around 145W [18]. It mainly focused on testing different types of fluid that can be used inside the tank and it did not consider higher CPU powers or the type of heat sink solution. This study consist of computational analysis of the natural convection for high powered servers up to 400W and its thermal shadowing along with different types of heat sinks. Optimized for forced convection air cooling from the earlier studies, plate fin and pin fin heat sinks were used for comparison. The objective of this study is to propose such a setup for practical application and defining various factors like power input, type of heat sinks, thermal shadowing and hot swapping.

Chapter 2

SPECIFICATION OF THE SERVER

2.1 Description of the Server

The servers used in this study are third generation open compute servers. Each server has four DIM blocks and each having four DIMMS of 8GB memory. Two microprocessors with a design power density of 115W are mounted on the server [10]. The CPU has a dimension of 50mm × 50mm and the chassis is 166.2mm long, 511mm in width and the height varies for different form factors.

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 server has 2 open rack unit form factors and a dimension of about 166.2mm×511mm×96mm [11].

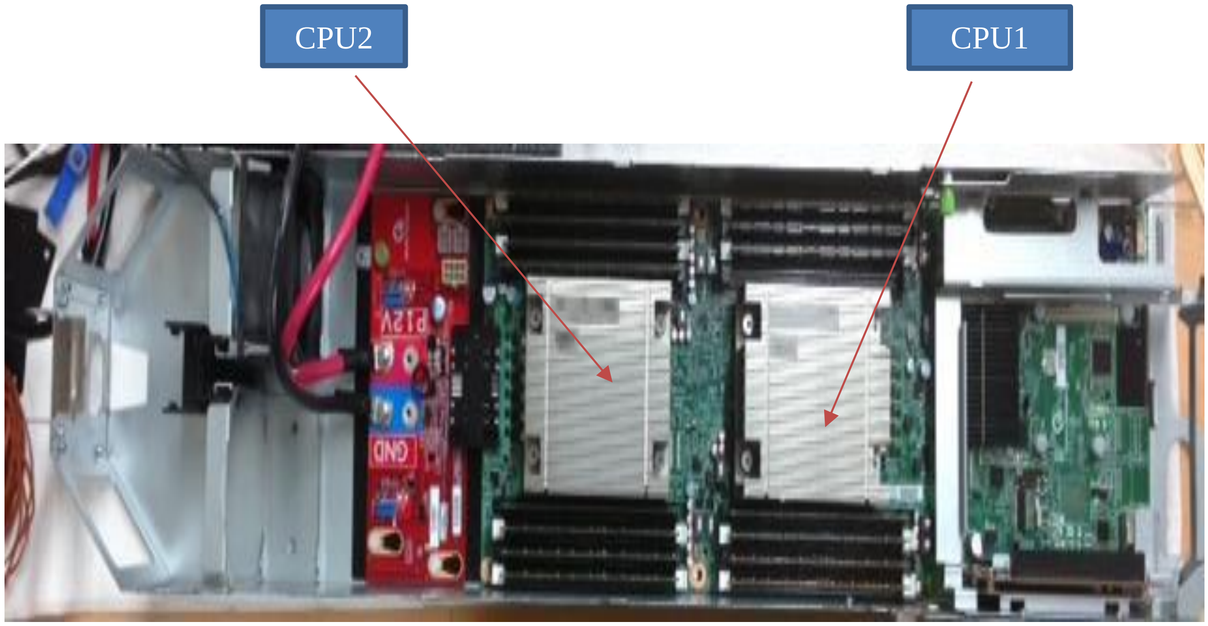

Figure 4 Top view of Open Compute Server

2.2 Base line server specifications

The baseline server has the same dimension except its power density can be varied for each CPU. Baffles are designed as an obstruction to direct the flow of the fluid.

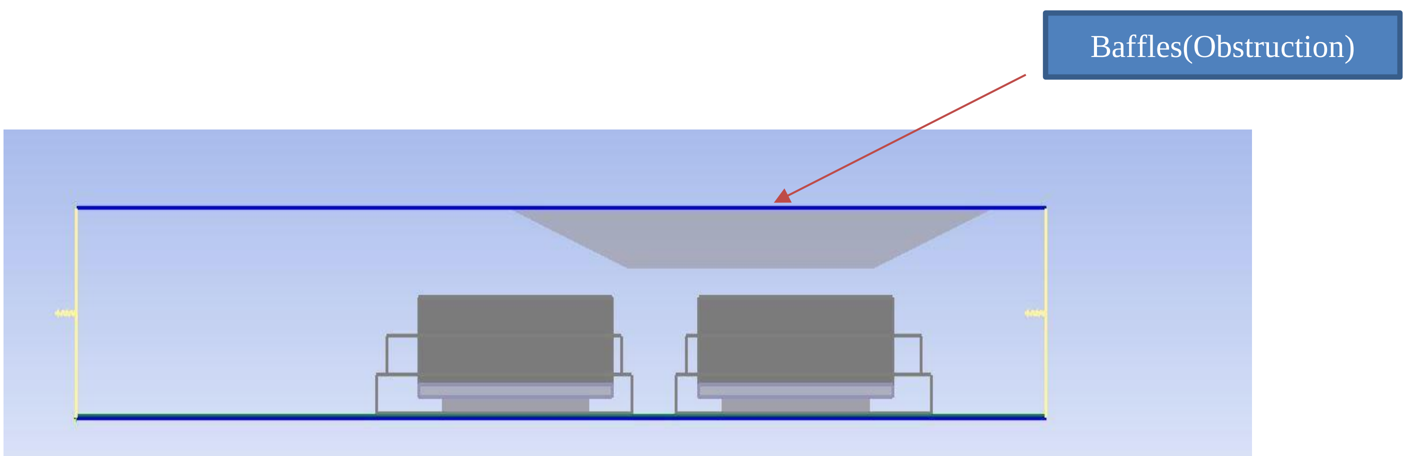

Figure 5 Side view of Baseline Server

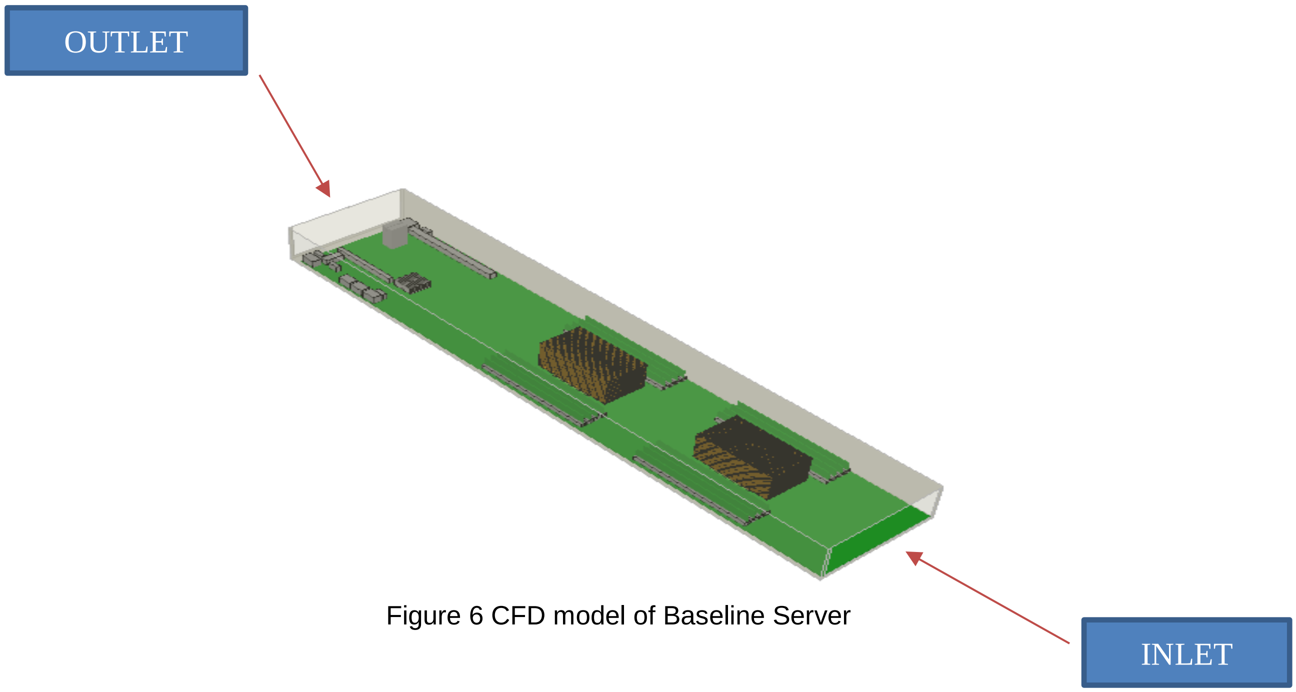

Figure 6 CFD model of Baseline Server

As shown in Figure (6), one side of the server cross-section is considered inlet for the fluid and the opposite side is the outlet. The fluid moves through the DIMMS and the heat sinks are mounted on the microprocessor. The fluid picks up the heat from the IT equipment and moves through the outlet.

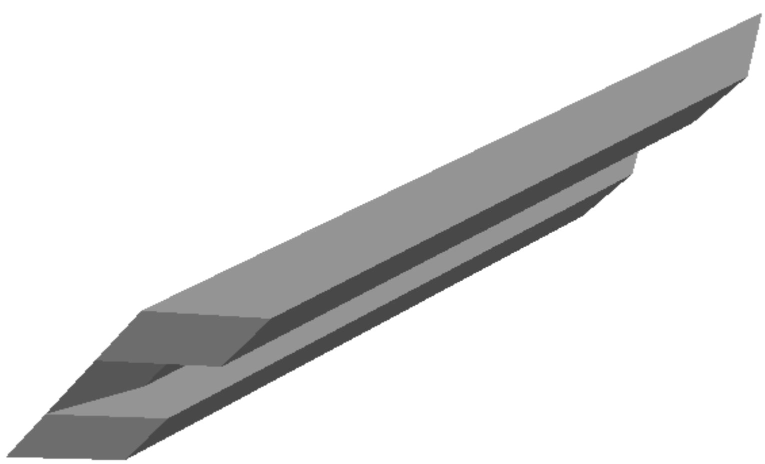

Figure 7 Baseline Duct

Figure 8 Positive X-direction Orientation of Server

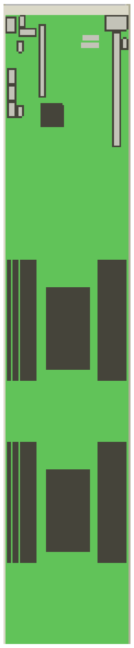

Figure 9 Top View of the server

2.3 Tank Design

A chassis design was created to represent a tank in a computational tool 6SigmaET. The tank dimensions are 1000mm×800mm×200mm and eight servers are placed into the tank. Two heat exchangers are placed into the tank between the servers. The tank is completely filled with dielectric fluid either Mineral Oil or EC-100. The servers were immersed into the bathtub without any fan or a pump for the circulation of the dielectric fluid.

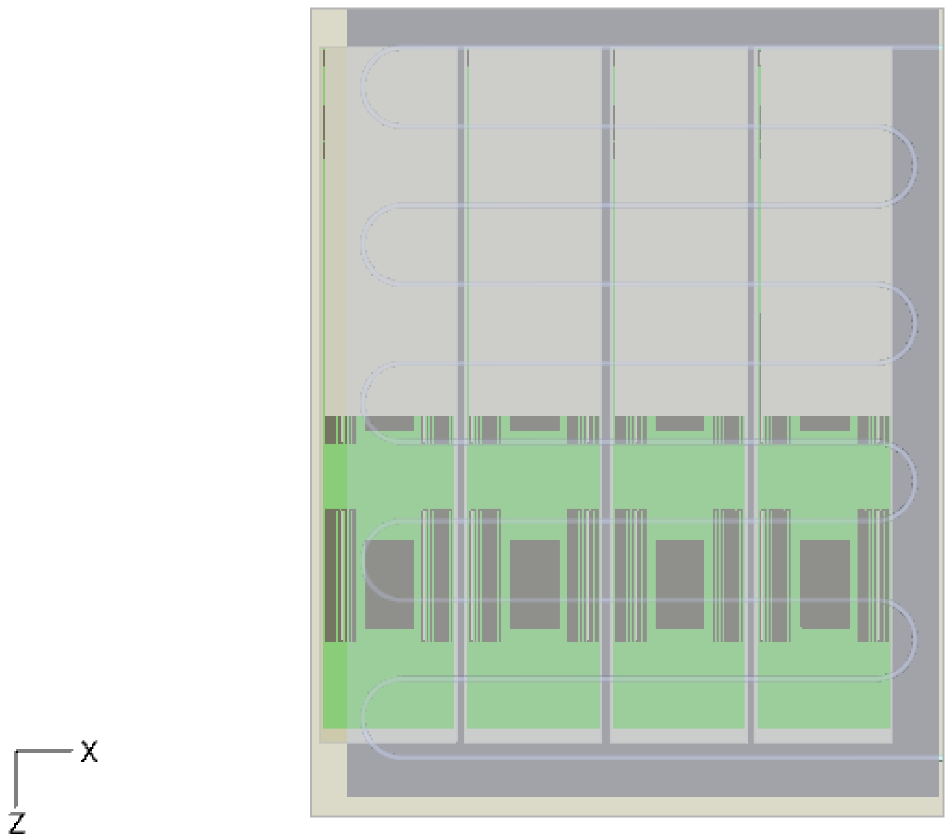

Figure 10 Positive Y-direction orientation of the tank

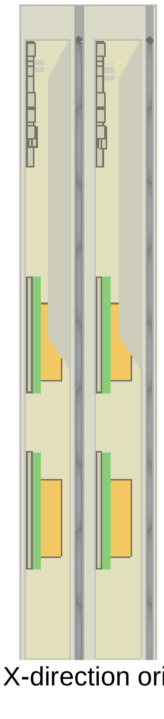

Figure 11 Positive X-direction orientation of the tank

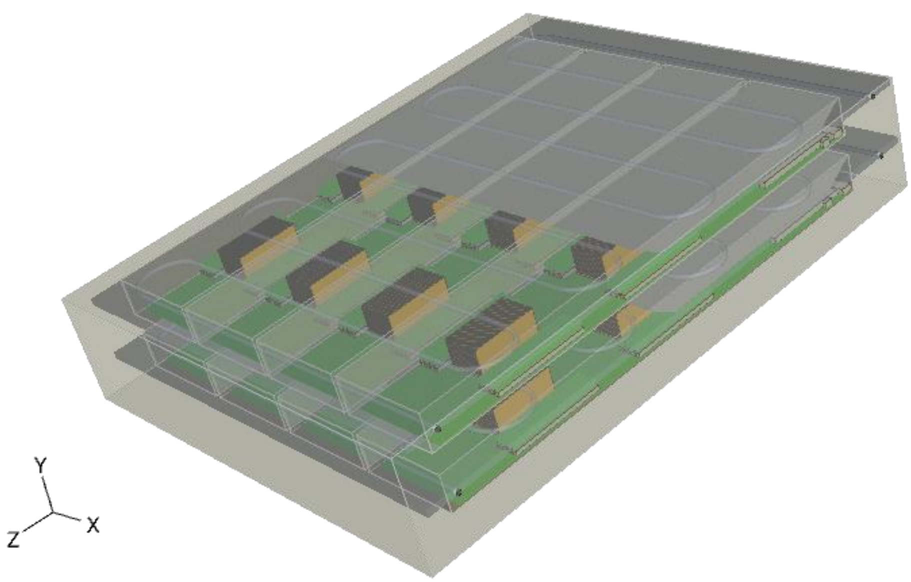

Figure 12 Isometric View of the server

2.4 Volumetric Flow Rate Conversion

In this study, we consider a Velocity of 0.005m/s, 0.1m/s, and 1m/s flowing through the pipes of heat exchangers.

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

Volumetric flow rate = Area x Velocity ……… (Equation 1)

The area of the pipe is calculated as the diameter of the pipe is 8mm

Area = $(\pi/4)$ x Diameter^2 ……… (Equation 2)

Area = 0.00005027 m^2

Table 2 Flow rate conversions

Velocity(m/s)	Volumetric Flow Rate(m^3/s)
0.005	0.00000025135
0.1	0.000005027
1	0.00005027

Chapter 3

METHODOLOGY

3.1 Thermal Shadowing

Thermal shadowing is a phenomenon in which the heat carrying capacity of the cooling medium decreases as its temperature increases after it picks up heat from the first power source. This happens due to a reduction in the temperature difference between the maximum junction temperature of successive power sources and the cooling fluid. Thermal shadowing causes an increase in temperature of a localized component that stays in the thermal shadow of other components. The below figure explains the concepts of thermal shadowing,

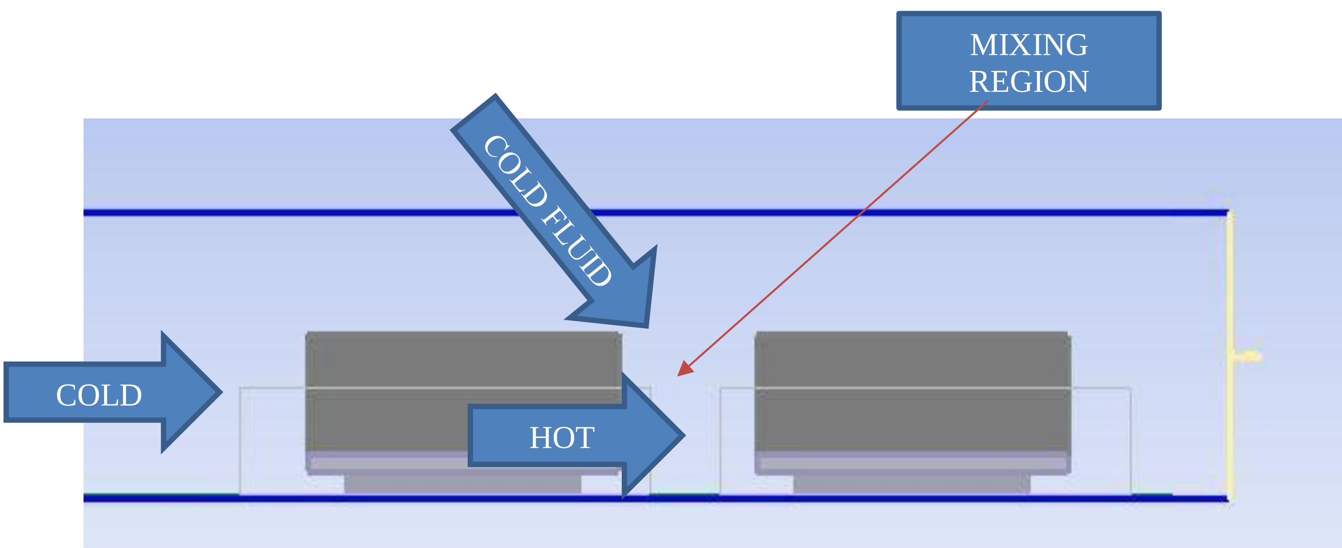

Figure 13 Thermal Shadowing Phenomenon

As we can see from the above figure, the fluid entering from the cold side moves through the first heat sink and carries its heat. Some of the fluid bypasses the first heat sink without picking up heat due to factors like tip clearance and span wise spacing. Tip clearance is a gap between the top point of the heat sink and the ceiling. The heated fluid then mixes with the bypassed fluid due to which there is an overall increase in the temperature of the inlet fluid of the second heat sink. This causes a localized increase in the junction temperature of the heat sink which stays in thermal shadowing [14].

The phenomenon of thermal shadowing can be explained using this energy balance equation.

$M1h1 + M2h2 = M3h3$ ………………………… (Equation 3)

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

Enthalpy can be taken as product of heat capacity and temperature to find the temperature of mixed fluid that enters the second heat sink

$M1CpT1 + M2CpT2 = M3CpT3$

$T3 = (M1T1 + M2T2) / M3$

T3 is the temperature after mixing of fluid stream. T3 is higher than the inlet fluid temperature ($T3 > T1$). To reduce the impact of thermal shadowing, baffles have been designed so that it can direct the bypass flow towards the heat sink and duce the impact of thermal shadowing.

3.2 Grid Independent Study

This study is conducted on a computational software named 6SigmaET. A grid independence study is carried out to validate the mesh count and for optimum result. It is important to validate the model boundary condition. Baseline boundary conditions are used for grid independence study. The inlet temperature of the water flowing through the heat exchangers is 20℃. The velocity of the water is considered as 0.1 m/s. The default dielectric fluid temperature is 25℃. Properties of Mineral oil were considered for the dielectric fluid.

Table 3 Grid independence

No. of Elements (Million)	Mean Temperatures (°C)
15.6	43.6
22.8	41.9
54.1	41.7

The points are plotted on the graph

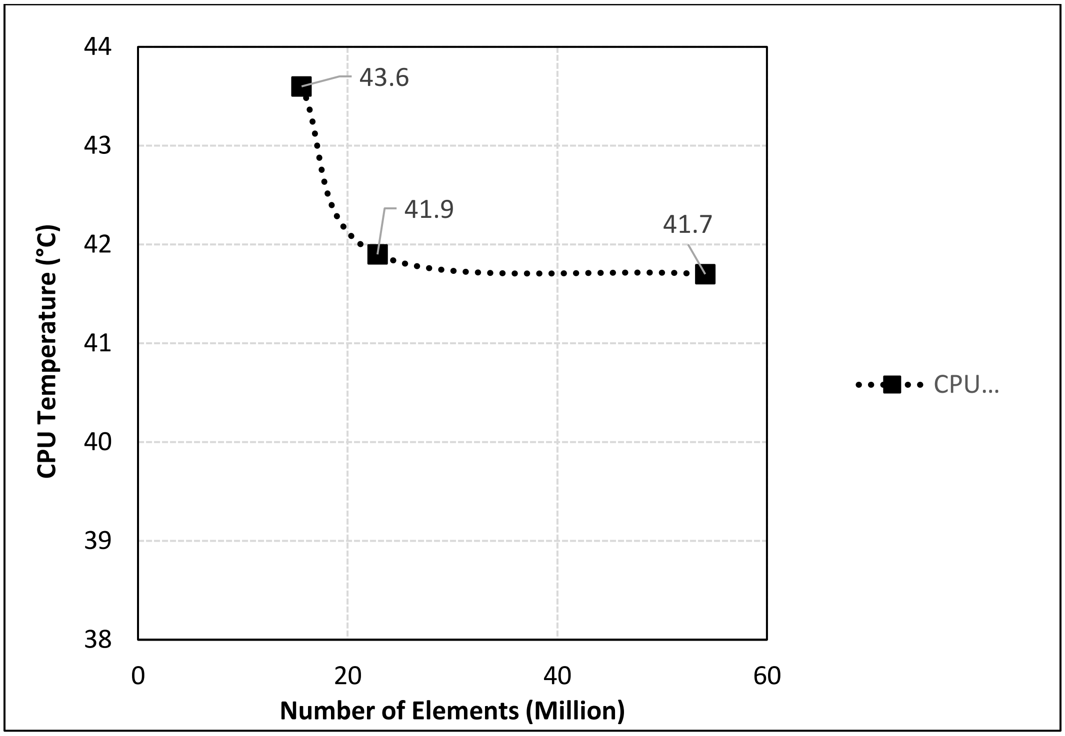

Figure 14 Grid Independent Study

It is clear from the above graph Figure (14) that as grid count increases the average temperature of the server decreases. After a certain number of nodes, there isn't major variation. For optimum results, the total number of nodes is taken in the range of around 22800000

3.3 Temperature dependent properties

The properties of the dielectric fluid in the tank change as the temperature of the tank changes. As the temperature increases the viscosity of the dielectric fluid decreases due to which the circulation of the fluid through the tank increases. The temperature dependent properties of Mineral Oil [13] and EC-100 are given in Table (4) and Table (5).

Temperature (℃)	Dynamic Viscosity (kg/m-s)	Kinematic viscosity (m^2/s)
30	0.01405	1.65E-05
40	0.01046	1.23E-05
45	0.00909	1.07E-05
50	0.00794	9.35E-06

Table 4 Change in properties of White Mineral Oil due to Temperature

Table 5 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)
30	0.01062	803.78	2165.9	0.13789	1.322E-05
40	0.00767	796.98	2203.2	0.13730	9.63E-06
45	0.00662	793.58	2221.9	0.13702	8.34E-06
50	0.00576	790.18	2240.5	0.13673	7.29E-06

Reynolds number is less than 2000 for both White Mineral Oil and Synthetic Fluid. So Laminar model is used to solve the Naiver Stokes equation.

3.3 Types of Heat Sinks

Heat sinks are the important components that are used for heat transfer. There are three major types of heat sinks - Parallel plate heat sink, extruded cut plate fin heat sink, and pin fin heat sink. A parallel plate heat sink was used in the baseline server in this study.

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

Figure 15 Types of Heat Sinks

Heat sinks are designed to remove heat from the power source by increasing the heat transfer area. Heat spreaders are used between heat source and secondary heat exchanger. The heat spreader is the heat exchanger that transfers the heat with a 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 as a heat spreader. Parallel Plate type heat spreaders are popular and readily available in the market. Heat sinks are also built from materials having high thermal conductivity. Copper and aluminum alloys have favorable heat transfer characteristics, including good thermal conductivity and thermal performance. Hence they are the most widely used materials for heat sink manufacturing. At present, research work is going on in the 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.

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

Table 6 Specifications of existing parallel plate heat sink

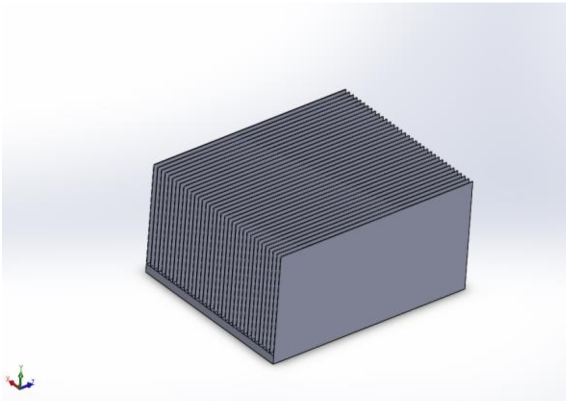

Figure 16: CAD model of existing Parallel Plate Heat Sink

Volume of this heatsink can be calculated using the formula,

Volume = Thickness x Height x Length …………… (Equation 4)

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

For 35 fins,

Volume = 35 x 1353

= 47355 mm^3

3.4 Modification of Baseline Server

The baseline server is modified for different types of heat sink. Plate heat sink and pin fin heat sink which are optimized for forced air cooling were used instead of the tradition parallel plate heat sink. For the optimized heat sink to increase the narrow region of the heat sink for the water to flow, the distance between the fins were increased and the number of fins were reduced to 20. For the pin fin the height of the fins was 20mm and the number of fins were 15×15.

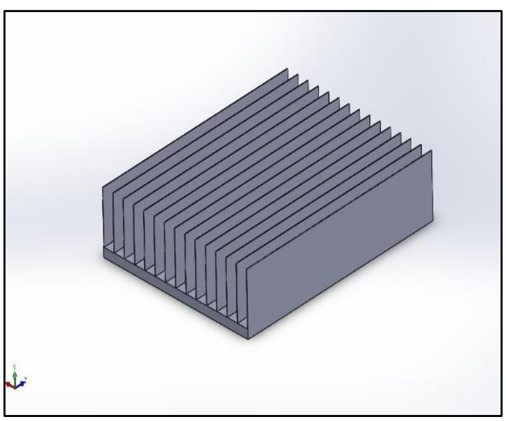

Figure 17 CAD model of optimized parallel plate heat sink

Table 7 Specifications of optimized parallel plate heat sink

Thickness	0.25mm
Height	30mm
Length	110mm
Number of fins	20

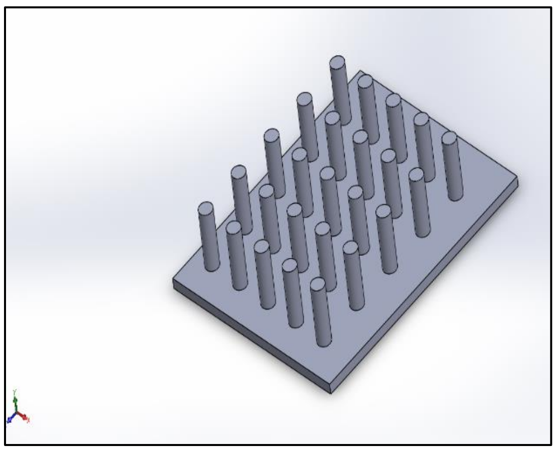

Figure 18 CAD model of pin fin heat sink

Table 8 Specifications of pin fin heat sink

Radius	1.5
Height	20
Number of fins	15 x 15

CHAPTER 4

RESULTS

4.1 Results for baseline simulation

Baseline simulation was carried out using Mineral oil as the dielectric fluid. These simulations were done for low powered servers of around 100W. Several cases were conducted by varying the velocity and the temperature of the water flowing through the heat exchanger.

Cases	Velocity (m/s)	Inlet Temperature (ºC)	Power (Watt)	Di-electric fluid Temperature (ºC)	Heat Sink (type)	Dielectric Fluid K= 0.13789 (W/m K)
Case 1	0.005	20	100	25	Parallel plate	Mineral Oil
Case 2	0.1	20	100	25	Parallel plate	Mineral Oil
Case 3	1	20	100	25	Parallel plate	Mineral Oil
Case 4	0.005	25	100	25	Parallel plate	Mineral Oil
Case 5	0.1	25	100	25	Parallel plate	Mineral Oil
Case 6	1	25	100	25	Parallel plate	Mineral Oil
Case 7	0.1	35	100	25	Parallel plate	Mineral Oil
Case 8	0.005	35	100	25	Parallel plate	Mineral Oil
Case 9	1	35	100	25	Parallel plate	Mineral Oil

Table 9 Simulation cases for low powered servers

These cases were simulated and the graph was plotted for the same.

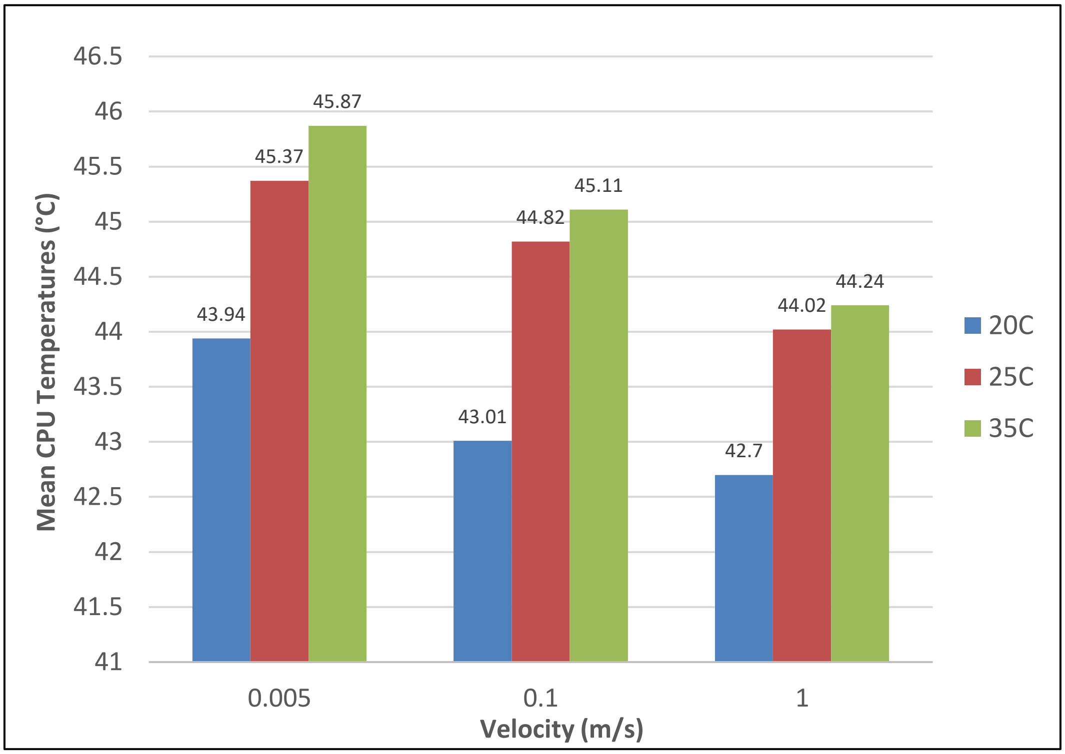

Figure 19 Velocity vs Mean temperature
Comparison of low power servers

From the above graph Figure (19) we could see that as the water temperature at the inlet of the heat exchanger decreases and the velocity of the water increase, better cooling efficiency is obtained.

As the temperature of the dielectric fluid was 25ºC and for some cases, the inlet water through the heat exchanger was supplied at 35ºC, the water supplied tends to drop the temperature as it moves inside the tank.

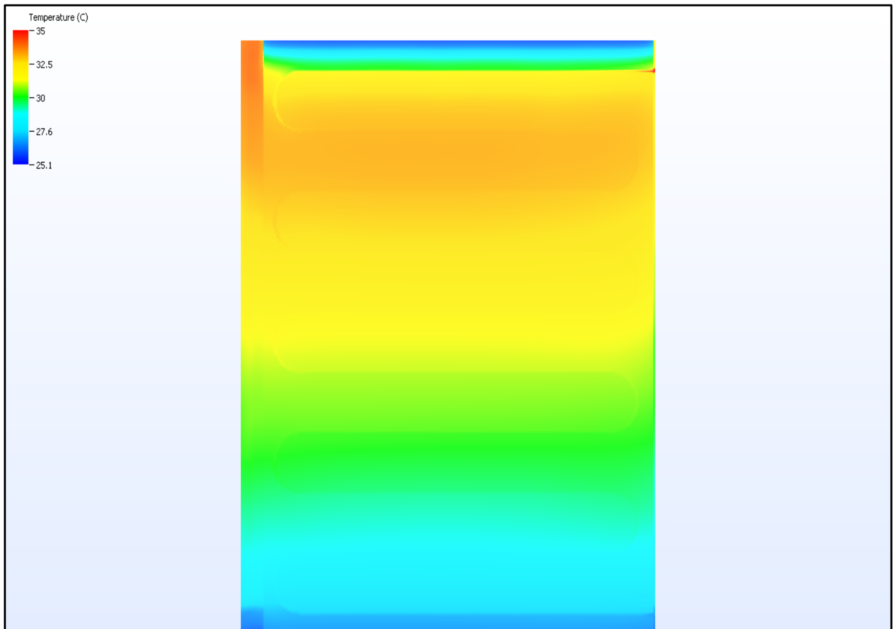

Figure 20 Result Plane plotted on the heat exchanger

Therefore, it can be observed from Figure (20) that for low powered servers of around 100W, no heat exchanger is necessary for such a solution. Due to natural convection the hot fluid moves upward and the cold fluid at the bottom due to the density difference. This circulation of dielectric fluid takes place in the tank continuously. The heat produce by the low powered servers is not as much and the natural convection is enough for such a tank solution, considering the fluid temperature is 25 ºC.

Table 10 Temperature obtained at different velocities

Velocity (m/s)	Pumped Return (°C)
0.005	27
0.1	28.1
1	32.4

4.2 Baseline results with no heat exchanger

The solution was tested for which the heat changers were absent and the cooling was only dependent on natural convection. The dielectric fluid was maintained at 25ºC for the initial stage. The power source were at 100W and at 100% utilization. The picture below shows the representation of the tank solution with the heat exchanger.

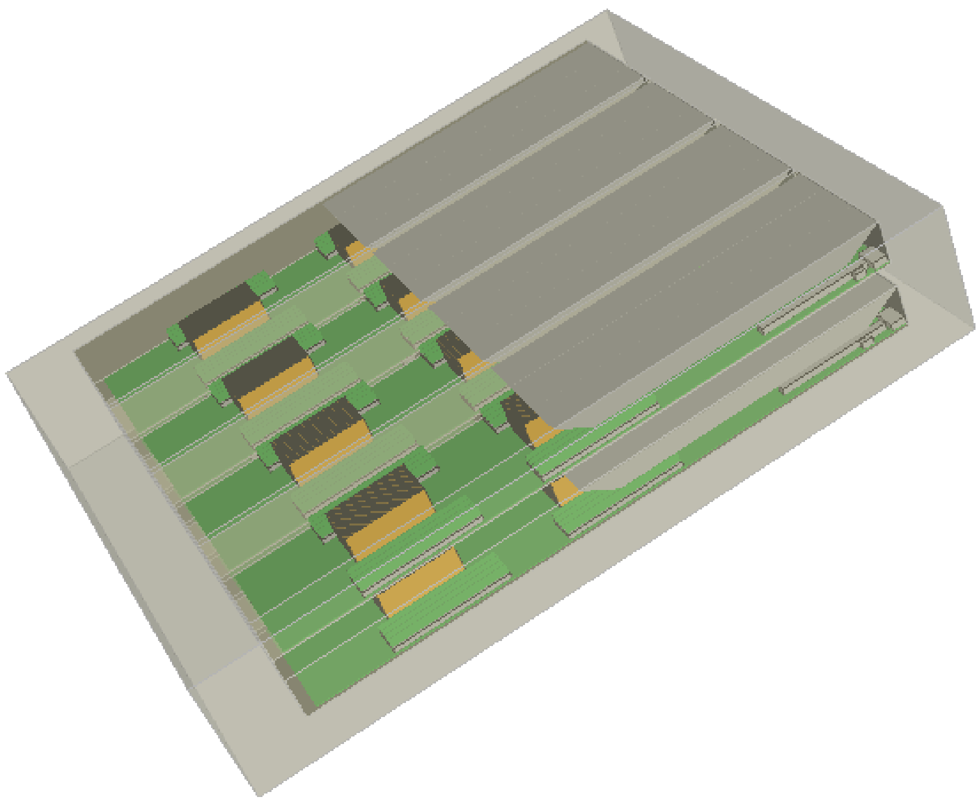

Figure 21 Isometric view of the tank with no heat exchanger

As just for natural convection, the maximum temperature observed at the component was 49.4 ºC. The maximum junction temperature is below the maximum temperature the components and the server can withstand. Therefore it could be said that natural convection would be a sufficient cooling method for a similar tank solution where low powered servers are used of around 100W. The results in Figure(22) and Figure(23) shows the temperature plot for the two resultant planes.

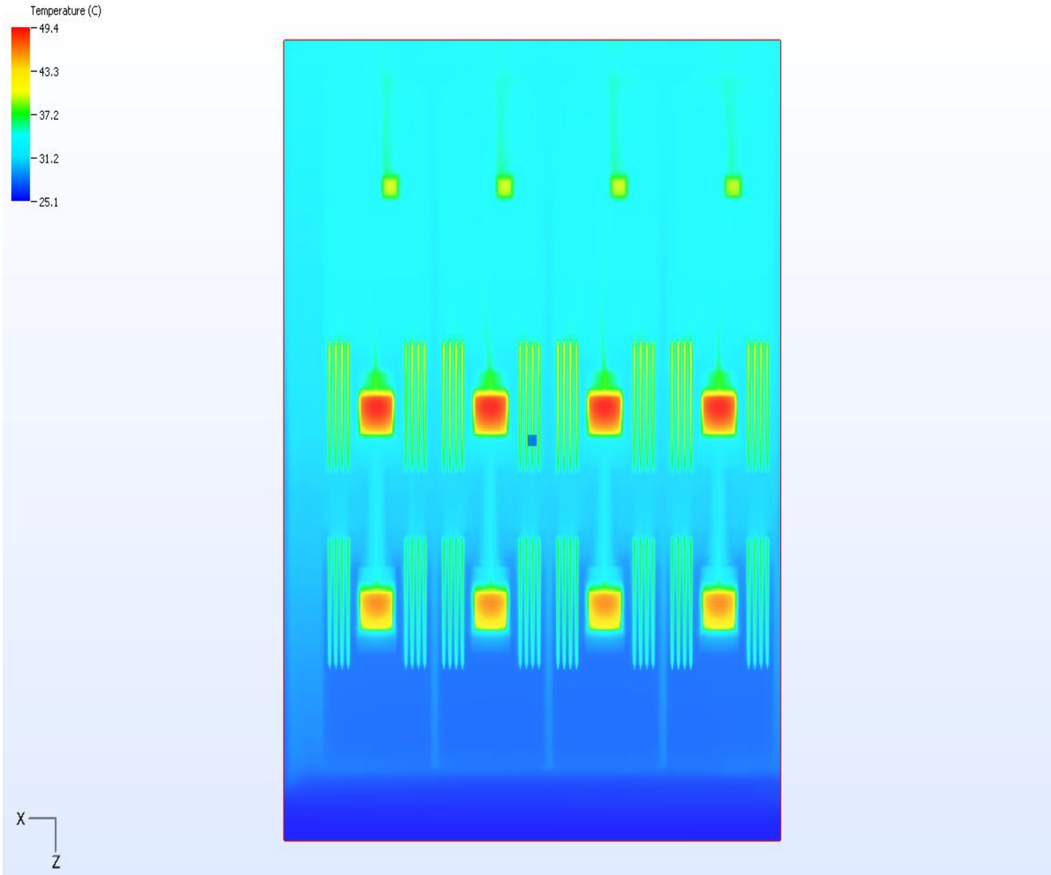

Figure 22 Temperature distribution for front server

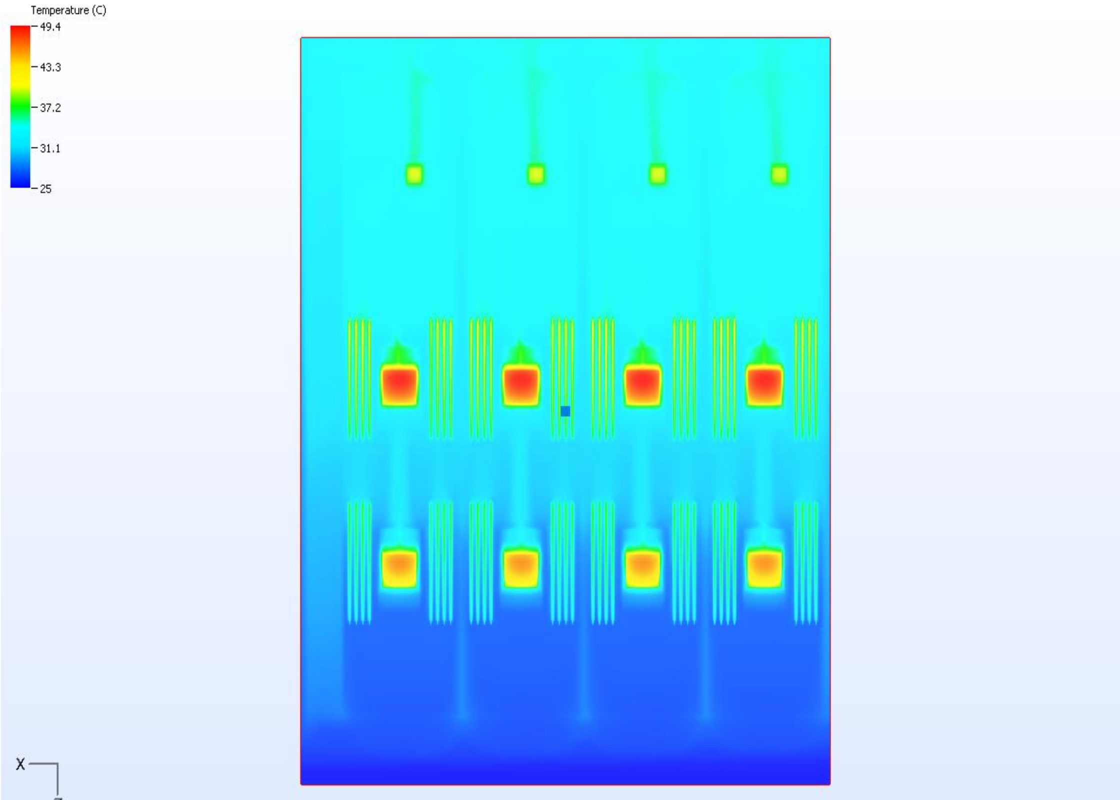

Figure 23 Temperature distribution for back server

4.3 Results for High Powered Servers

The tank solution was also tested for high powered servers of around 200W, 300W and 400W. Nowadays, High powered servers are used for such edge datacentre solution. The dielectric fluid was at 30 ºC at the initial stage. The velocity and the inlet temperature at the heat exchanger inlet was kept constant at 0.1m/s and 35 ºC respectively.

Table 11 Simulation cases for high powered servers

Cases	Velocity (m/s)	Water inlet Temperature (°C)	Power (Watt)	Di-electric Fluid Temperature (°C)
Case 10	0.1	35	200	30
Case 11	0.1	35	300	30
Case 12	0.1	35	400	30

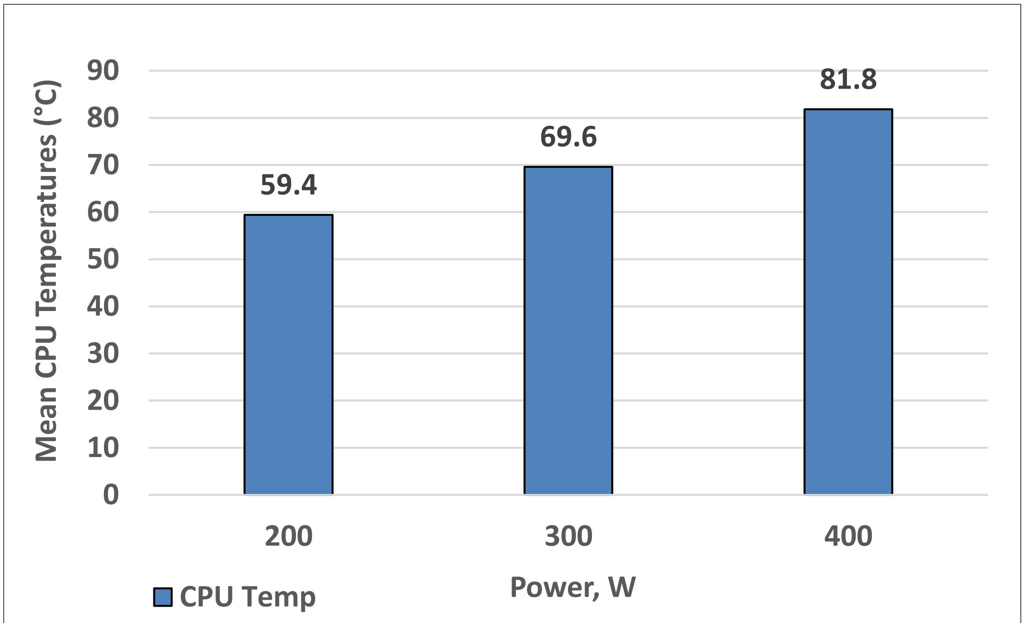

Figure 24 Mean CPU temperatures for high powered servers

As we can observe from Figure (24), the mean temperature obtain when all the eight servers were at 200W at 100% utilization was 59.4 ºC and for 300W it was around 69.6 ºC. The mean temperature for the 300W power usage model was 81 ºC. Therefore, this tank solution cannot be used for servers of 400W as the mean junction is higher and not desirable. The natural convection technique is not able to cool these high powered servers and external components will be required for the circulation of the dielectric fluid. From the observation of the above graph, it can be concluded that this tank solution can be ideal for servers powered up to 200W-300W.

4.4 Power Difference Results

To validate some real life cases a power difference between two consecutive power sources was simulated. In this scenario, one power source was set to 100% utilization and the consecutive one was set to 10% utilization from the same server.

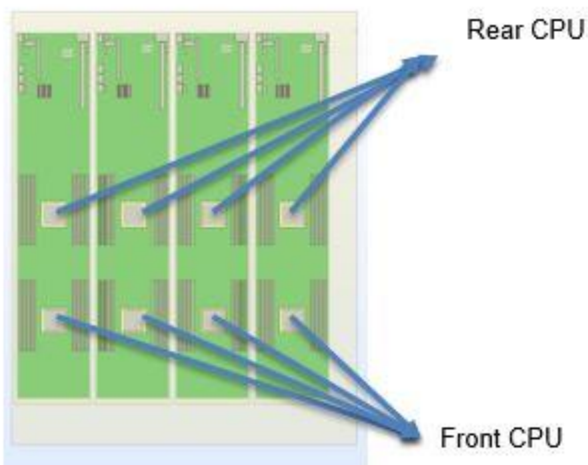

Figure 25 Front and Back CPU in the server

In the first case, the front power source was set to 100% utilization and the rear power source to 10% utilization. In the next case, the front power source was set to 10% utilization and the rear was set to 100% utilization. This can usually happen when there is less load or the two power sources were performing a different task. Both cases are then compared to when all power sources are running at 100% utilization.

The velocity and the inlet temperature of water into the heat exchanger as kept constant at 0.1m/s and 35 ºC respectively. The power was kept at 300W and the dielectric fluid initial temperature was 30 ºC. EC-100 engineering fluid was used for this scenario as it has better cooling efficiency for high powered servers.

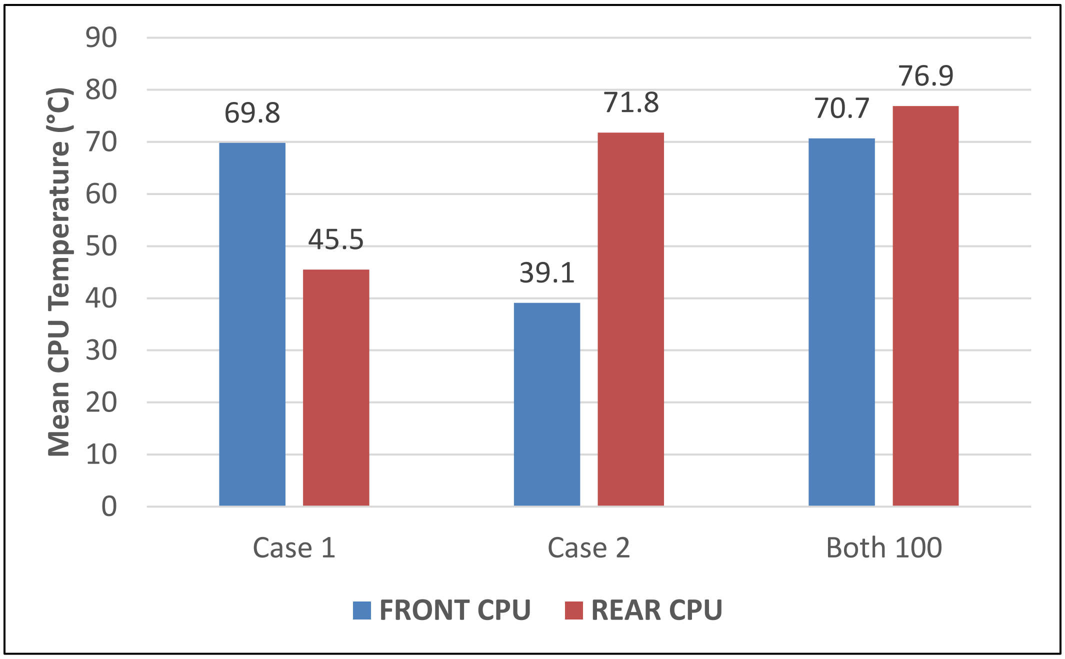

Figure 26 Comparison of mean temperatures in front and back CPU

4.5 Thermal shadowing results

A study for thermal shadowing was conducted to consider its effects on such a tank based solution with multiple servers present in the tank. The velocity and temperature of the water flowing into the heat exchanger is 0.1m/s and 35 ºC respectively. The dielectric fluid used was EC-100 and its initial temperature was maintained at 30 ºC.

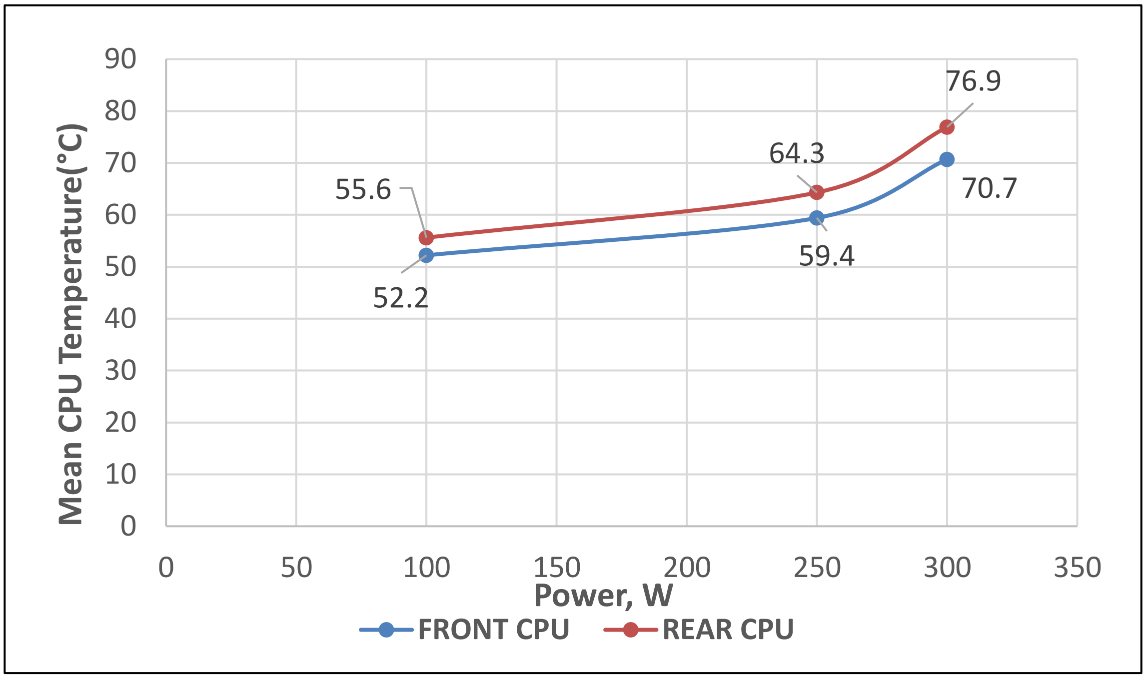

Figure 27 Thermal Shadowing between the CPU

In Figure (27), the difference between the temperatures of the front and the rear power source can be observed. The figure indicates that when the power of the server increases that effect of thermal shadowing also increases

4.6 Types of Heat Sink Results

The traditional parallel plate heat sink was compared with the optimized heat sink for forced convection in air cooling which was validated from earlier study. For this study velocity and the temperature of the water was 0.1m/s and 20 ºC respectively. Ec-100 was the type of dielectric fluid used and the initial temperature was at 30ºC. Figure (27) shows the comparison between the three heat sink for such a tank based solution with natural convection.

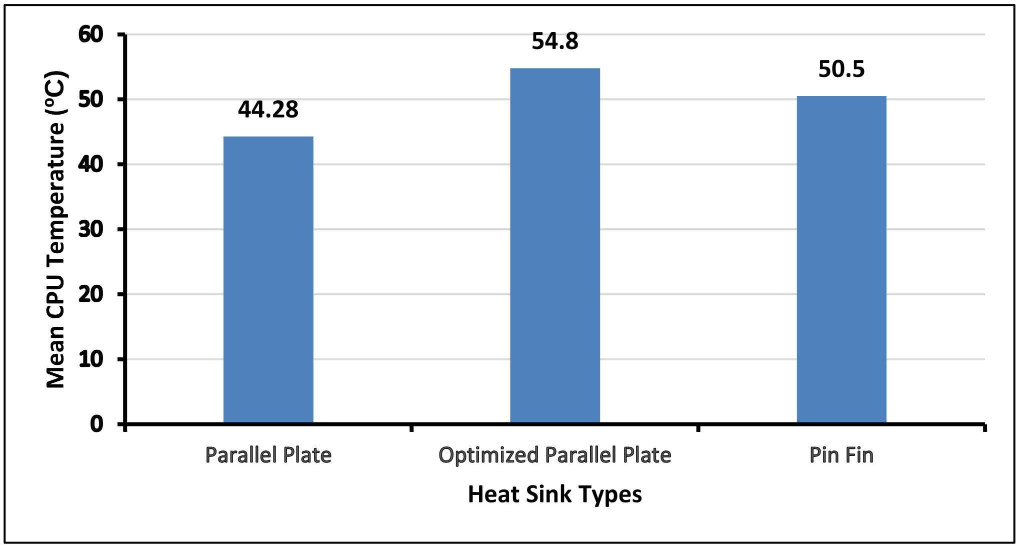

Figure 28 Mean CPU temperature between different heat sinks

The above graph shows the distribution for types of heat sink vs the mean temperatures obtained. As we could see that the lowest temperatures are observed for the traditional parallel plate heat sink as compared to the other two. Pin fin also performed well and the highest temperature was observed in an optimized parallel plate heat sink. Traditional parallel plate proved to have better efficiency in removing heat from the CPU’s

4.7 Hot Swapping Results

In this scenario, two of the servers are removed from the setup while the rest of the servers are still operating. This type of simulation is mainly done to simulate real life cases where servers or IT components are removed because of maintenance and failure. Boundary conditions for this case were assumed 0.1m/s velocity of the inlet water and 35ºC temperature. Power supplied to the 300W and the dielectric fluid temperature 30ºC. EC-100 was the kind of dielectric fluid considered for this high powered servers.

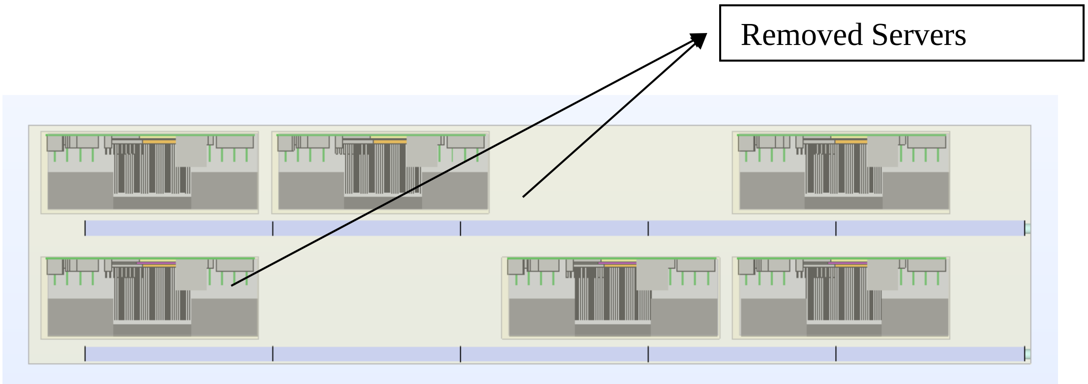

Figure 29 Hot Swapping

Figure (29) represents the case of hot swapping. The two servers are removed from the tank based solution and the rest of the servers are at 300W with 100% utilization. This change in the setup can change the flow for natural convection and can cause temperature variation. The temperature variations are plotted in the below Figures (30) and Figure (31).

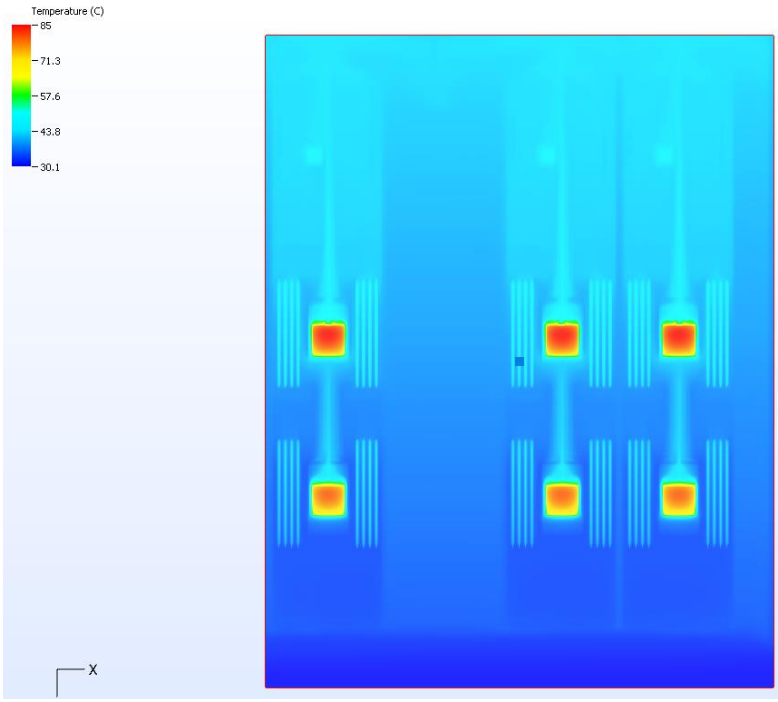

Figure 30 Resultant plane for front server

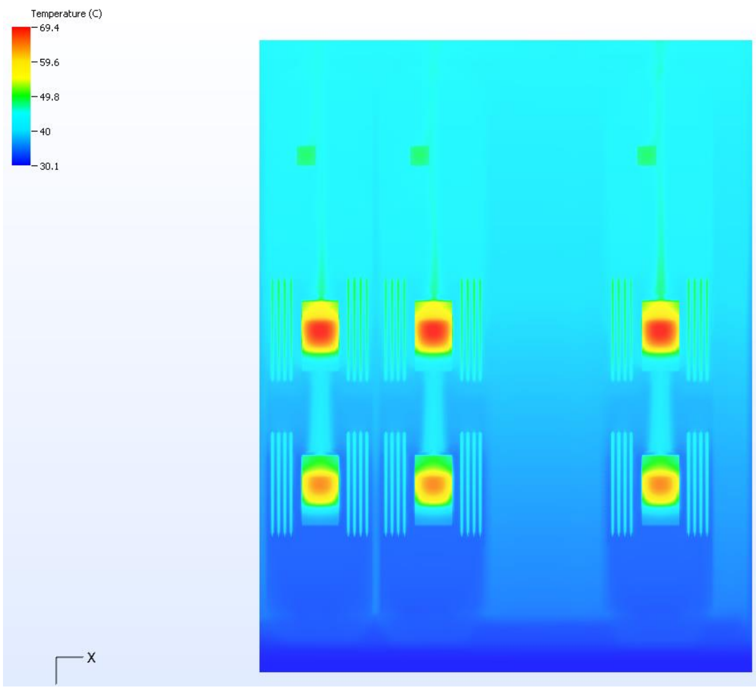

Figure 31 Resultant plane for back server

4.8 No Heat Sink Results

In the earlier study, Figure (32) relationships between CPU power applied and CPU junction temperature with heatsink and without heatsink was plotted [18]. The temperature obtain in this study for no heat sink case for 60W power was about 90ºC. To validate the extend of the power input to which the setup can sustain a similar study was performed on this model for 100W and 150W CPU power. The inlet water velocity was taken as 0.1m/s while its temperature was 35ºC. Dielectric fluid was EC-100 with its temperature of 30ºC

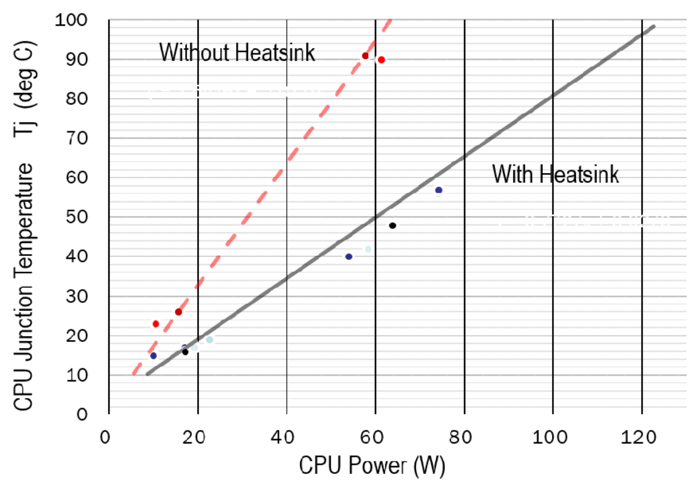

Figure 32 Relationships between CPU power applied and CPU junction

For 100W CPU power with no heat sink, the maximum temperature of 117 ºC was recorded and at 150W CPU power maximum temperature of around 137 ºC was observed. As the obtained temperature is higher than the desired junction temperatures, we could conclude that this setup can be used along with heat sinks for a power input of more than 100W.

CHAPTER 5

CONCLUSIONS AND FUTURE WORK

Immersion cooling is proving to be an effective method for overcoming thermal challenges in electronic cooling industry. Implementation of single phase immersion has made heat removable process from high-performance computational systems more energy efficient. This technique provide substantial advantages for reliability issues such as contamination of the IT equipment and fan acoustics. Single phase immersion cooling has various advantages over both air cooling and two phase immersion cooling. Less complexity and cost as compared to air cooling are observed in single phase immersion cooling [20]. As complex system and special dielectric fluid that change phase are required for two phase immersion system [19]. Also the global warming potential is lower in single phase immersion cooling. With the use of natural convection in single phase the efficiency can be increases with removal of pump hereby decreasing the PUE.

The study conducted here, opens new vision to improve highly demanding data centre cooling technologies. It is clear from the study that for a low powered CPU of around 100W and dielectric fluid of 25°C, heat exchanger is not required unless the supply temperature of water is below 25°C. So, we can conclude that this type of cooling solution can be implemented for serves with CPU power of 300W-350W.Baseline parallel plate heat sink in natural convection performed better than optimized parallel plate and pin fin heat sink (Forced Convection). Also, For Hot Swapping, significant temperature difference is observed between the two planes of the result. For natural convection, the maximum power that can be cooled without heat sink is around 75W in this study.

Chapter 6

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