Investigation and Optimisation of Heatsink Design for Single-Phase Immersion Cooling

SCHOOL OF MECHANICAL ENGINEERING

UNIVERSITY OF LEEDS

MECH3890 – Individual Engineering Project

Investigation and Optimisation of Heatsink Design for Single-Phase Immersion Cooling

PRESENTED BY Liam Thomas Huckle

If the project is industrially linked, tick this box and provide details below

COMPANY NAME AND ADDRESS:

STUDENT DECLARATION (from the "LU Declaration of Academic Integrity")

I am aware that the University defines plagiarism as presenting someone else's work, in whole or in part, as your own. Work means any intellectual output, and typically includes text, data, images, sound or performance. I promise that in the attached submission I have not presented anyone else's work, in whole or in part, as my own and I have not colluded with others in the preparation of this work. Where I have taken advantage of the work of others, I have given full acknowledgement. I have not resubmitted my own work or part thereof without specific written permission to do so from the University staff concerned when any of this work has been or is being submitted for marks or credits even if in a different module or for a different qualification or completed prior to entry to the University. I have read and understood the University's published rules on plagiarism and also any more detailed rules specified at School or module level. I know that if I commit plagiarism I can be expelled from the University and that it is my responsibility to be aware of the University's regulations on plagiarism and their importance. I re-confirm my consent to the University copying and distributing any or all of my work in any form and using third parties (who may be based outside the EU/EEA) to monitor breaches of regulations, to verify whether my work contains plagiarised material, and for quality assurance purposes. I confirm that I have declared all mitigating circumstances that may be relevant to the assessment of this piece of work and that I wish to have taken into account. I am aware of the University's policy on mitigation and the School's procedures for the submission of statements and evidence of mitigation. I am aware of the penalties imposed for the late submission of coursework.

Signed LHuckle

Date: 08/05/2022

CONTENTS

Abstract

Datacentres use a large quantity of power and it one the major costs of running a data centre. Server cooling is the reason for the high power consumption and a large proportion is just to cool the Infrastructure. As such, better and more efficient methods of cooling are desirable.

A method to implement better cooling efficiency is Immersion cooling, whereby the components are submerged in a highly thermally conductive dielectric fluid. The advantage over using air is that a reduced pump speed can be used and overall power consumption is reduced.

An investigation into the possible heatsink geometries that would be best suited for immersion cooling was conducted. A reference intel chipset and heatsink was used. The first of four heatsink designs was based off of this reference design.

Subsequent designs were then examined to see if it was possible to optimise the design. Ultimately, the reference design was not improved upon but more testing is needed to optimise the design.

Chapter 1 - Introduction

1.1 Introduction

The demand for data centres is surging. Current estimates suggest that the compound annual growth rate (CAGR) for the data centre services market is 13.69% over the next 5 years (Mordor Intelligence, 2020). The data centre services market expected value by 2026 is USD 105.6 billion (Mordor Intelligence, 2020). Thus, there is great investment potential in this sector. Data centres are expensive to run, and the growth rate of data centres energy consumption has slowed to around 4% per year (A. Shehabi et al., 2016). This is only due to technological advancements and implementation of more efficient operating methods. The specific power draw percentage for cooling of data centres is estimated at 38% of the total power use (C. Nadjahi et al., 2016). Evidently there is demand for better ways to cool heat generating computer chips on an industrial level to reduce operating costs and reduce the carbon footprint of data centres.

Chipsets are getting more powerful and as a result produce more heat per unit area. Traditional air-cooling methods can be effective but are costly to operate and maintain. Immersive cooling methods are more effective at transferring heat away from chipsets whilst also producing a better power usage effectiveness (PUE) than air cooling.

Immersion cooling submerges entire server racks in a thermally conductive but electrically non-conductive fluid (Dielectric fluid). The fluid may move under natural convection or forced convection. Immersion cooling is cheaper to run than air-cooling due to fewer moving parts, meaning fewer points of failure and therefore cheaper maintenance costs.

The performance and efficiency of cooling methods is assessed through a Thermofluids analysis. Thermofluids governs the theoretical potential for cooling methods and can be simulated with the aid of Computer Fluid Dynamics (CFD) software. The CFD software used in this investigation is Ansys FLUENT. This software is able to model fluid flows, heat transfer and turbulence. All of which are relevant parameters for single-phase immersion cooling.

The project focuses on the optimisation of heatsink design by varying the heatsink geometry within the geometric confines of a standard 1U server rack, commonly found in data centres. Although typical immersion cooling systems have a proprietary server mounting solution, basing the geometry confines to that of a 1U server rack is logical as there is less probability of sizing issues if it similar to the industry standard.

1.1 Aim

To evaluate and optimise an existing heatsink design for use in a forced convection single phase immersive cooler.

1.3 Objectives

Objectives to be satisfied:

1. Investigate literature focused on single phase heatsink design.
2. Research an industry used heatsink and the typical operating parameters.
3. Identify a suitable dielectric fluid to be used in the analysis.
4. Model an existing forced convection, single phase cooling heatsink in CFD.
5. Adjust the existing heatsink design to optimise heat transfer.
6. Identify potential reasons for differences in design choice between the existing and the optimised heatsink.

1.4 Report Layout

- Chapter 1 - Introduces the aim, background and context of the project.
- Chapter 2 - Explains the theory of CFD and heat transfer.
- Chapter 3 - States the starting geometry and reference heatsink design.
- Chapter 4 - Provides the boundary conditions for the simulations.
- Chapter 5 - Explores the various heatsink geometries tested.
- Chapter 6 - Discusses and presents the conclusions of the analysis conducted on the various heatsink designs.

Chapter 2 - Heat Transfer and CFD Theory

2.1 CFD software

Computational Fluid Dynamics (CFD) is a valuable tool when assessing or evaluating a new design concept. CFD software can accurately simulate scenarios of a specified nature. CFD allows the user to alter and change certain variables in their simulation easily which is not possible with experimental methods. Immersion cooling is expensive, and it is not necessary to purchase a server, or run CPU under load or use an expensive dielectric fluid when an inexpensive and environmentally friendlier method of computer simulation is available. In addition to this, CFD removes environmental anomalies as the boundary conditions can be selected and variables can be fixed according to the user's choice. With the increasing computational power of computers at present, the ability to simulate high mesh density simulations in a short time period is perfectly viable. CFD software will save manufactures lots of money and time when researching and evaluating heatsink designs (Ramlan and Darlis, 2019). CFD allows a user to import a 3D model of their heatsink design, select a working fluid and select a heat source. The software then uses physics laws such as mass, momentum, and energy conservation to compute the analysis.

2.2 Heat Transfer Equations

Fourier's Law dictates the heat transfer rate of an object. It factors in a materials thermal conductivity, (k), the surface area of the object (A) and the temperature gradient/difference of the surface to the ambient surroundings (ΔT). The symbol for heat transfer rate is q and the Fourier equation is shown below.

$$q = -kA\Delta T$$

Heatflux is an adaption of the Fourier equation which removes the surface area term and its symbol is q" with units W/m^2. Ultimately CFD software will compute the energy/heat using the Navier-Stokes Equations, which accommodate for greater complexity and accuracy.

Thermal resistance is the impedance of the flow of heat through a material or object. A heatsink with a low thermal resistance is desired to maximise the transfer of heat away from hot components. The thermal resistance of a heatsink can be expressed as the ratio of the temperature difference (difference between the ambient fluid temperature and the heat generating components) to the total heat transfer of the heatsink array.

$$R_{Total} = \frac{\Delta T}{q_t}$$

2.3 Fluid Flow Equations

Ansys FLUENT solves for mass and momentum conservation. When heat transfer is involved an additional energy equation is calculated. The combination of these equations is sometimes referred to as the Navier-Stokes Equations. It is a complex set of equations and is thus usually only calculated using computers. It includes in the momentum and mass continuity and the energy equation. The equations are shown below in a general form (NASA, 2021)

Navier-Stokes Equations
3 – dimensional – unsteady

Glenn Research Center

Coordinates: (x,y,z) Velocity Components: (u,v,w)

Time : t Density: ρ Total Energy: Et

Pressure: p Stress: τ

Heat Flux: q Reynolds Number: Re Prandtl Number: Pr

Continuity:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0$$

X – Momentum:

$$\frac{\partial(\rho u)}{\partial t} + \frac{\partial(\rho u^2)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} + \frac{\partial(\rho uw)}{\partial z} = -\frac{\partial p}{\partial x} + \frac{1}{Re_r}\left[\frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z}\right]$$

Y – Momentum:

$$\frac{\partial(\rho v)}{\partial t} + \frac{\partial(\rho uv)}{\partial x} + \frac{\partial(\rho v^2)}{\partial y} + \frac{\partial(\rho vw)}{\partial z} = -\frac{\partial p}{\partial y} + \frac{1}{Re_r}\left[\frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{yz}}{\partial z}\right]$$

Z – Momentum

$$\frac{\partial(\rho w)}{\partial t} + \frac{\partial(\rho uw)}{\partial x} + \frac{\partial(\rho vw)}{\partial y} + \frac{\partial(\rho w^2)}{\partial z} = -\frac{\partial p}{\partial z} + \frac{1}{Re_r}\left[\frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z}\right]$$

Energy:

$$\frac{\partial(E_T)}{\partial t} + \frac{\partial(uE_T)}{\partial x} + \frac{\partial(vE_T)}{\partial y} + \frac{\partial(wE_T)}{\partial z} = -\frac{\partial(up)}{\partial x} - \frac{\partial(vp)}{\partial y} - \frac{\partial(wp)}{\partial z} - \frac{1}{Re_r Pr_r}\left[\frac{\partial q_x}{\partial x} + \frac{\partial q_y}{\partial y} + \frac{\partial q_z}{\partial z}\right]$$
$$+ \frac{1}{Re_r}\left[\frac{\partial}{\partial x}(u\,\tau_{xx} + v\,\tau_{xy} + w\,\tau_{xz}) + \frac{\partial}{\partial y}(u\,\tau_{xy} + v\,\tau_{yy} + w\,\tau_{yz}) + \frac{\partial}{\partial z}(u\,\tau_{xz} + v\,\tau_{yz} + w\,\tau_{zz})\right]$$

Figure 1: Navier-Stokes Equations (NASA, 2021)

2.4 Turbulence Models and Limitations

Turbulence due to its inherently unsteady nature is extremely difficult and computationally taxing to calculate. CFD software can use many different turbulence models, in which each try to best approximate turbulence, but they make many assumptions to create equations that are time efficient. The most common turbulence model is the K-epsilon model. This is the model used in all simulations for the project. This model uses two partial differential equations. The equations account for turbulent kinetic energy and rate of dissipation of turbulent kinetic energy. The K-epsilon model, however, assumes that turbulence is isotropic meaning that the ratio between mean rate of deformations and Reynold’s stress is uniform in all directions. Although there are more accurate turbulence models, this one was selected due to the simplicity of input variables needed and the lower computing power needed.

Chapter 3 - Reference Heatsink Geometry

3.1 Description of Server Racks

Data centres have a standardised set of computer/server form factors. Servers are usually setup in a cabinet housing many servers racks. Each server rack is assigned a form factor. The standard makes use of a 'Rack Unit' and the size of server racks are sized as a multiple of this 'Rack Unit'. The size of a Rack Unit is 44.45mm in height and the width of most server rack cabinets is standardised. Therefore when describing server rack dimensions only the Rack Unit is varied.

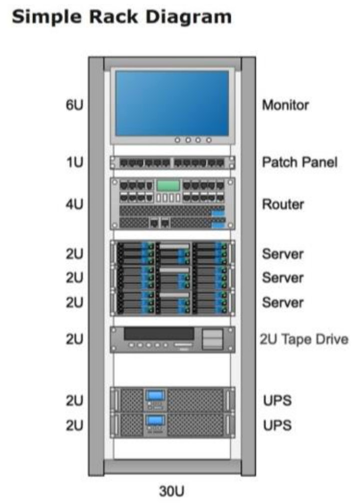

Figure 2: Server Rack Diagram (EdrawMax, 2022)

3.2 Typical CPU and Heatsink Pairings

There are great number of CPUs to be used in servers, however, Intel's Xeon Range of CPUs are the most widely used Chipset for datacentres (Electronic Design, 2018). Intel's Xeon range has many offerings with CPUs designed for 1U and 2U servers racks, low performance and high performance chipsets and they also have reference heatsinks for each CPU model. CPUs and reference heatsinks designed for the 1U high performance servers will serve as the benchmark for the project. All relevant dimensions and chip TDP (thermal design power) values were derived from these specific Intel Xeon Chipsets.

2.3 CPU Thermal Design Power

Typical Xeon chips used in high performance 1U servers have a TDP of between 150W to 165W. Thus, for the simulations a heatflux that aligns within this wattage range will be selected to replicate the heat output of the server grade CPU.

2.4 Heatsink Dimensions and Materials

The reference heatsink for the aforementioned chip is of a low profile and simple geometry. It is made out of aluminium alloy which is a common choice for heatsinks due to its great thermal properties and low cost. The material used in a heatsink is vital as it will directly affect the thermal resistance of the heatsink. When geometry of a heatsink is fixed and the fluid medium and flow rate is also fixed, the material choice for the heatsink is the biggest factor in influencing the total heat dissipation. Aluminium is the material of choice within the industry for multiple reasons. Although there are other materials with a higher thermal conductivity such as copper, aluminium is far less expensive (Gabrian, 2020). Aluminium is easy to manufacture heatsinks out of through extrusion. A common aluminium used in industry is 6063 Aluminium alloy with a high thermal conductivity of 201-218 W/(mK) (N. Quesnel, 2019).

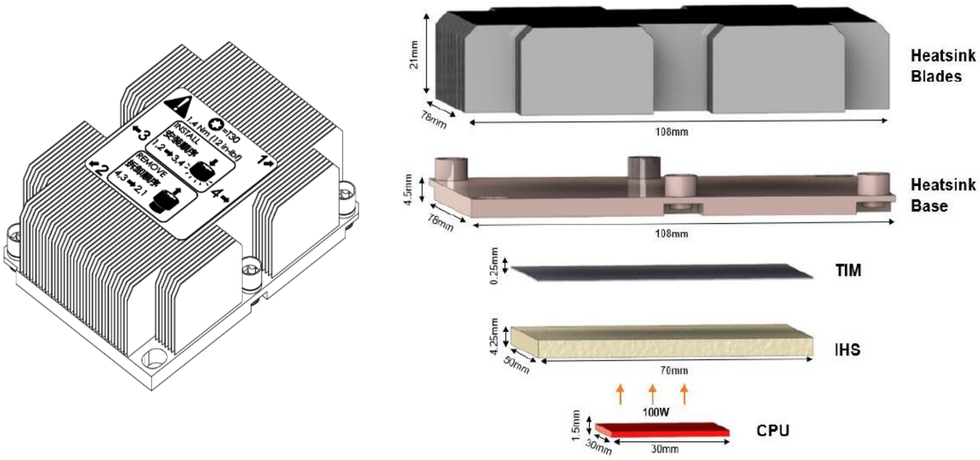

Figure 3: 1U Narrow High Performance Heatsink Dimensions (Intel Xeon Brochure, 2020)

2.5 Heatsink geometry

The primary objective of a heatsink is to increase the effective surface area in contact with a working fluid to maximise heat transfer (P. Teertstra et al., 2000). Heatsinks work by transferring thermal energy from a higher temperature object to a second object (fluid) at a lower temperature but with a higher heat capacity (A. Soni). Heatsinks essentially try to bring the hotter objects into thermal equilibrium with the cooler surroundings. Designing the optimal heatsink depends on various factors such as cost, available airflow, size constraints and the final product will be a balance of these factors (P. Teertstra et al., 2000). There are many potential geometries that heatsink manufactures could adopt, and specific to server racks, plate-fin and pin-fin arrays are most common due to their economic production cost (A. Soni, 2016).

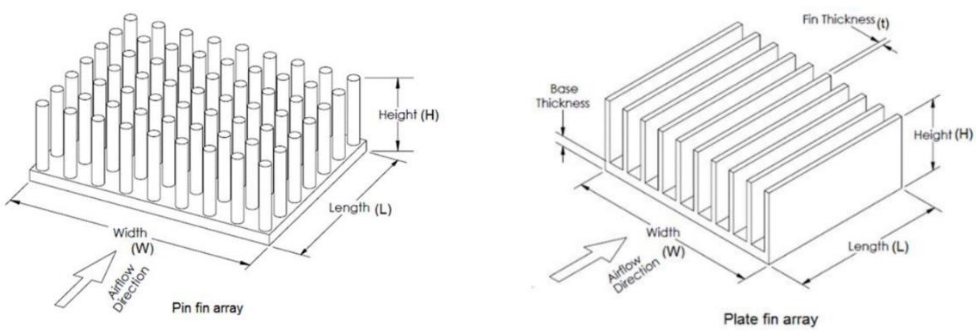

Figure 4: Heatsink Geometry

Plate-fin arrays typically have a greater surface area than pin-fin arrays, however pin-fins usually have a greater heat transfer coefficient due to the reduction of growth of the thermal boundary layers (A. Soni, 2016). The performance of competing heatsinks can be evaluated by the total heat dissipation for each heatsink for a specified base temperature and ambient temperature. There is already a wealth of published knowledge about plate-fin optimisation, however less so in regard to other geometry types (A. Soni). In addition to this, heatsinks with forced convection in air is well documented whereas heatsinks immersed in dielectric fluids are far less researched. The differences between the fluid mediums of air and dielectric fluids present a challenge when designing a heatsink for immersion cooling. Experimental results by Eiland et al (2014) show that heatsinks designed for forced convection air cooling are effective with mineral oil and a low pump setting, however, it was suggested that more performance could be achieved with a heatsink designed for the oil. Dielectric fluids are far more viscous than air making the formation of turbulent flow harder (Dymyd et al., 2020). This is particularly important as turbulent flow is more effective at removing heat than laminar flow (Dymyd et al., 2020). It is suggested that heatsinks in higher viscous fluids benefit from a larger fin pitch (distance between fins) than compared to fin pitches of air-cooled heatsinks (Dymyd et al., 2020).

Chapter 4 - Controlled Variables and Boundary Conditions

4.1 Fluid Mediums

The working fluid will have to be thermally conductive but electrically insulative. Fluids of this nature are known as dielectric fluids. With regards to single-phase cooling, the working fluid must remain a liquid at all times, thus a fluid with a high boiling point exceeding that of the system heat production is required. To effectively remove heat from the system the fluid must be able to absorb heat easily and whilst not climbing greatly in temperature. The fluid must therefore have a high thermal conductivity and a high specific heat capacity. Synthetic fluids are very useful as they can be made to be environmentally friendly, non-flammable, non-corrosive and highly efficient as a working fluid. Common fluids in use for immersion cooling are the 3M™ Fluorinert™ range. ElectroCool® EC-100 Dielectric Coolant is another fluid that was used in a precious study by Shah, et al (2019). Due to the availability of data for the EC-100 fluid this was the fluid used in all simulations.

Table 1: Comparison of Fluid Mediums

	EC-100 @40°C	Air @40°C
Density (kg/m^3)	830	1.127
Kinematic Viscosity (kg/m-s)	0.01	1.702E-05
Thermal Conductivity (W/mK)	0.136	0.026
Specific Heat (J/kg-K)	2209	1007

4.2 Heatsink materials

The simulations had only two main solid parts, one being the CPU and the other being the heatsink. Both pieces are modelled as aluminum 6063 as this is a commonly used alloy in the industry.

4.3 CPU TDP

A CPU of 160W was replicated. This was modelled as a single layer on the bottom of the CPU mesh. Ansys requires the Wattage per unit area so the TDP was simply divided by the CPU area to find this input value.

4.4 Inlet and Outlet

The inlet has two main variables to assign, namely the temerature of the fluid medium and the velocity of the fluid. For all simulations a temperature of 30°C was used. Flowrates for oil based fluids are typically much slower than that of air due to the viscosity. A reasonable velocity of 1 m/s was selected and designed to replicate forced convection.The outlet is assigned a intilisation temperature of 30°C. the inlet is at the bottom of the fluid domain and the outlet at the top. This is to allow the heatsink to take advagntage of convective heat currents.

4.5 Boundary Walls

The walls of the testing chamber that encase the fluid flow are adiabatic.

4.6 Fluid Domain and General Layout

The fluid domain, cpu dimensions and heatsink baseplate dimension were kept constant for each simulation. The diagrams below illustrate the fluid domain and solid body dimensions that are common between models.

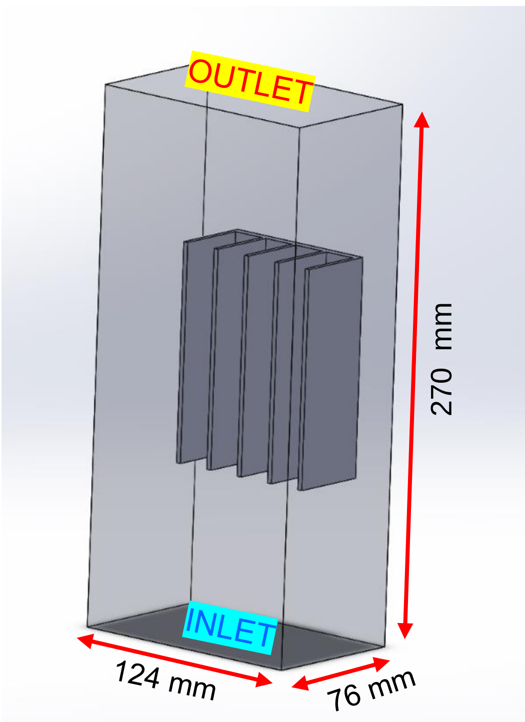

Figure 5: Fluid Domain

	Length (mm)	Breadth (mm)	Height (mm)
CPU	77.5	56.50	3.3
Baseplate	110	80	4.5

Table 2: CPU and Baseplate Dimensions

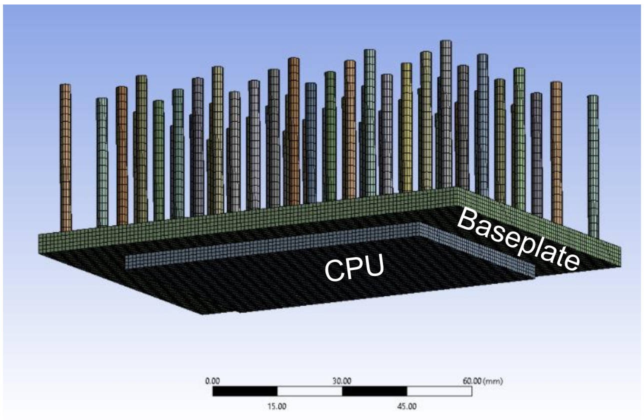

Figure 6: CPU and Heatsink Arrangement

Chapter 5 - Geometry Variations Tested

5.1 Introduction to Modelling Method

Ansys has a built in design modeler which enables the ability to create geometries, however, for this project SolidWorks was used instead. SolidWorks was used as it has a far more intuitive method of creating geometries. The Solidworks models were then imported into the Ansys software as a STEP file. Once imported into Ansys, the relevant surfaces and bodies were labelled and a mesh was applied.

Meshing is extremely important as this directly determines the accuracy of the results. A low element/node mesh will not provide the precision needed to model fluid flows correctly. The method in which the mesh is applied is also important. It is best practice to use mesh shapes that best suit the geometry so that the number of sharp or splintered cells is reduced. Ansys FLUENT student edition was used in the project for the simulations. This edition has an element limit of 512 000 elements. This is not ideal as to get really good meshes, typically 2 million elements achieved good accuracy for the various heatsinks. Consequently, each of the geometries modelled was meshed using different methods to suit the specific geometry types of each. The aim for each mesh was to minimise the skewness and maximize the element count within 512 000 elements.

Table 3: Mesh Statistics and Quality

Heatsink Type	High Density Plate Fins	Low Density Plate Fins	Low Density Pin Fin	High Density Pin Fin
Nodes	206989	144435	139351	160277
Elements	201719	149803	191003	510544
Skewness (max)	0.78	0.796	0.90	0.83
Skewness (Average)	0.08	0.127	0.28	0.30

5.2 High Density Plate Fins

For the first design iteration, a parallel fin geometry was created and intended to be similar to the reference heatsink from Intel. This first model has 9 plate fins. Each fin is 2.5mm thick and 35.5mm in height.

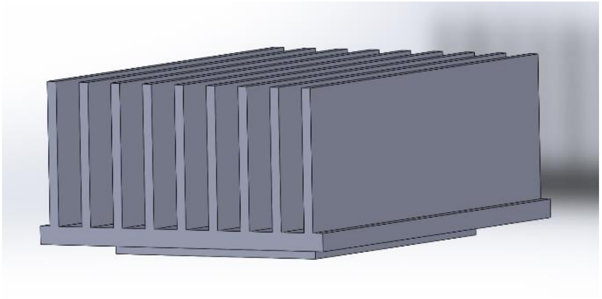

Figure 7: High Density Plate Fin

5.3 Low Density Plate Fins

This second model has a reduced plate count of 5 plate fins. Each fin is 3mm thick and 35.5mm in height. The intention was to investigate if a greater distance between fins was optimal as more viscous fluids typically benefit from a greater distance compared to low viscosity fluids.

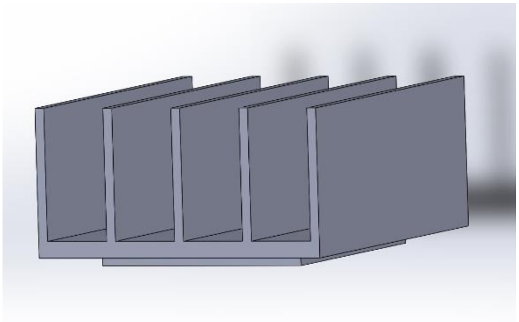

Figure 8: Low Density Plate Fin

5.4 Pin Fin Low Density / Crosshatch

This third model adopted a pin fin design. The pins are 2.5mm in diameter and 35.5mm in height. There are 50 pins in total. Pin fins offer great surface area per unit volume and also offer a greater chance of fluid mixing.

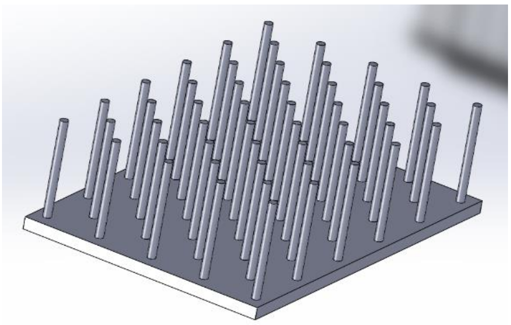

Figure 9: Low Density Pin Fin

5.5 Pin Fin High Density / Parallel

This fourth and final model is a high density pin fin model. The pins are 2.5mm in diameter and 35mm in height. There are 99 fins in total.

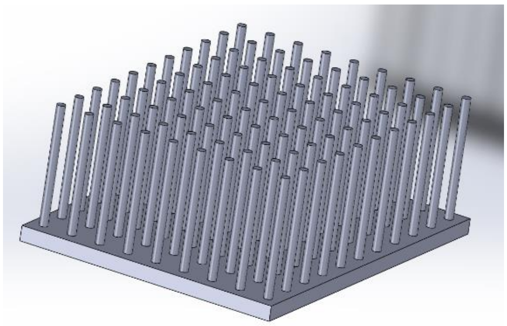

Figure 10: High Density Pin Fin

5.6 Approach to Testing and Optimising

The first model was intended to be similar to the reference intel heatsink. The priority here was to remain in a similar form factor and geometry type. Thus, plate fins were used in a high density. This provided great results and acceptable CPU temperatures of a maximum of 42.31°C.

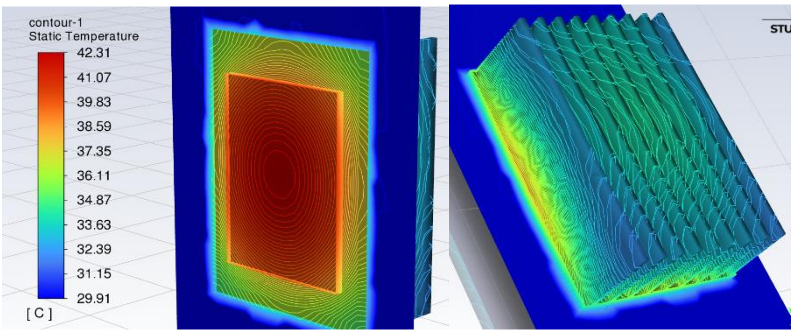

Figure 11: Heatmap of High Density Plate Fins

The second model was intended to optimise on the first design by providing a greater fin pitch to optimise fluid flow for the high viscosity fluid. The results were still acceptable but were in fact higher at a max of 48.83°C.

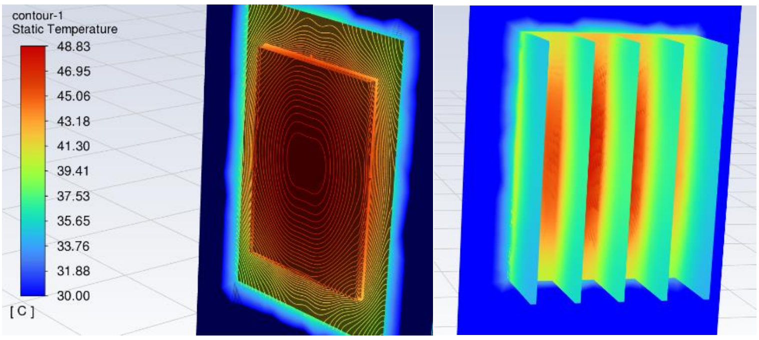

Figure 12: Heatmap of Low Density Plate Fins

The third model used a pin fin configuration. This was to test another commonly manufactured heatsink design. The results were acceptable but still higher than either of the plate fin geometries. The maximum was 56.80°C.

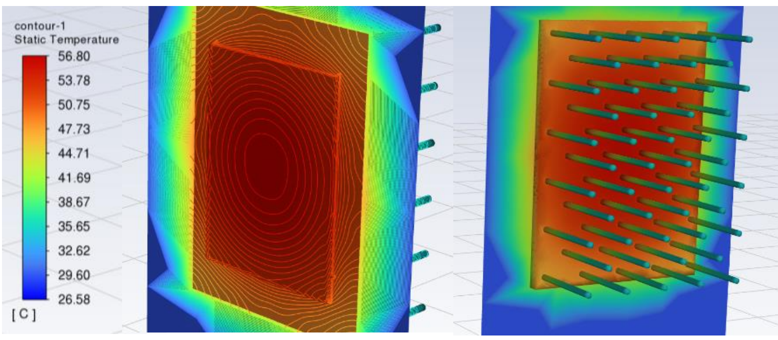

Figure 13: Heatmap of Low Density Pin Fins

The fourth and final model was a higher density arrangement of the third model as it was suspected that the pins did not outperform the plate design due to the reduced interface material and lack of fluid mixing. The higher density was expected to provide better mixing of the fluid. The results were positive and improved on the low density pins design but still did not outperform the plate fin design. The maximum temperature captured was 50.52°C.

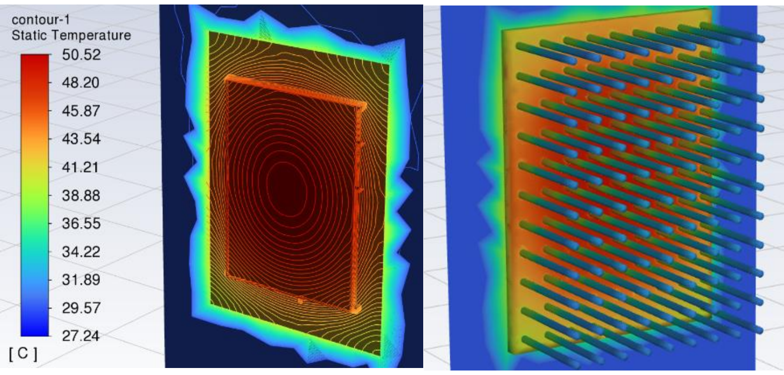

Figure 14: Heatmap of High Density Pin Fins

Chapter 6 - Conclusions

6.1 Achievements

- Identified and simulated real world geometry and boundary conditions
- Produced four varying geometries with realistic results

6.1 Discussion

The overall goal was to design a more optimised heatsink design for single phase immersion cooling. It is no surprise that employing a heatsink in a better heat conducting fluid such as EC-100 provided excellent performance. The investigation was to determine if there is a particular geometry that is optimal for the increased viscosity of the medium and reduced flow rate compared to air whilst conforming to similar size restrictions of the heatsink.

In section 3.3, it was highlighted that the best performing heatsink was a high density plate fin design. Other designs were made in comparison and attempts were made to optimise these designs but they did not outperform the first modelled design.

This is possibly due to the surface area benefit of the first design and there could potentially be an even more optimised density for the same design, but more testing would be required.

A limitation to the results is that the quality of the pin fin meshes was not as good as the plate fin designs. So the results could be erroneous. A limitation to the overall approach is that not enough geometries were examined and not enough background theory on the specific geometries was explored.

6.1 Conclusion

The findings of the project is that the key factors that influence the performance of heatsink design are the fluid flow rate, fluid conductivity and surface area of the heatsink and the related geometry of the design. An improved or optimised heatsink design to that of the reference was not obtained but the designs were still acceptable and would be able to perform in a single phase immersion system.

In future projects a greater variation of geometries and density of the designs should be included to get closer to an optimal design. If possible, a software package capable of greater mesh densities should be used for greater accuracy.

References

Gabrian. 2020. Copper vs. Aluminum Heatsinks: What You Need to Know. [Online]. [Accessed 23/11/2021]. Available from: https://www.gabrian.com/copper-vs-aluminum-heatsinks/

Gyarmarthy, K. 2020. What You Should Know About Data Center Cooling Technologies [Online]. [Accessed 20/11/2021]. Available from: https://www.vxchnge.com/blog/data-center-cooling-technology

Mordor Intelligence. 2020. Data center services market - growth, trends, covid-19 impact, and forecasts (2021 - 2026) [Online].[Accessed 18/11/2021] Available from: https://www.mordorintelligence.com/industry-reports/service-market-for-data-center

Nadjahi C, Louahlia H, Lemasson S, 2018, A review of thermal management and innovative cooling strategies for data center, Sustainable Computing: Informatics and Systems, 19, pp.14-28. ISSN 2210-5379

Nord, A. 2016. KTH Industrial Engineering and Management. Immersed Cooling of Electronics. EGI 2016-092 MSC
Open Compute. 2018. Immersion Cooling for Green Computing. [Online]. [Accessed 5/11/2021]. Available from: https://www.opencompute.org/files/Immersion-Cooling-for-Green-Computing-V1.0.pdf

Ramlan, I., Darlis, N., Ishak, I. A., Khalid, A. and Asyraf Zailani, M. A. 2019 "Comparison between Solidworks and Ansys CFX Flow Simulation on Aerodynamic Studies", Journal of Complex Flow, 1(2).

Shehabi, A, Smith, S.J, Horner, N, Azevedo, I, Brown, R, Koomey, J, Masanet, E, Sartor, D, Herrlin, M, Lintner, W. 2016. United States Data Center Energy Usage Report. Lawrence Berkeley National Laboratory, Berkeley, California. LBNL-1005775
Sheldon, R. 2020. Liquid cooling vs. air cooling in the data center.[Online]. [Accessed 19/11/2021]. Available from: https://searchdatacenter.techtarget.com/feature/Liquid-cooling-vs-air-cooling-in-the-data-center

Soni, A. 2016. Study of Thermal Performance between Plate-fin, Pin-fin and Elliptical Fin Heat Sinks in Closed Enclosure under Natural Convection. International Advanced Research Journal in Science, Engineering and Technology. 3(11). ISSN 2393-8021

Teertstra, P. Yovanovich, M.M. Culham, J.R. 2000. Analytical forced convection modelling of plate fin heat sinks. Journal of Electronics Manufacturing, 10(4), pp. 253-261.

Appendices

Residuals 1-4 of the Models