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PARAMETRIC MULTI-OBJECTIVE OPTIMIZATION OF COLD PLATE FOR SINGLE-PHASE IMMERSION COOLING

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ABSTRACT

The increasing demand for high-performance computing in applications such as the Internet of Things, Deep Learning, Big data for crypto-mining, virtual reality, healthcare research on genomic sequencing, cancer treatment, etc. have led to the growth of hyperscale data centers. To meet the cooling energy demands of HPC datacenters efficient cooling technologies must be adopted. Traditional air cooling, direct-to-chip liquid cooling, and immersion are some of those methods. Among all, Liquid cooling is superior compared to various air-cooling methods in terms of energy consumption. Direct on-chip cooling using cold plate technology is one such method used in removing heat from high-power electronic components such as CPUs and GPUs in a broader sense. Over the years Thermal Design Power (TDP) is rapidly increasing and will continue to increase in the coming years for not only CPUs and GPUs but also associated electronic components like DRAMs, Platform Control Hub (PCH), and other I/O chipsets on a typical server board. Therefore, unlike air hybrid cooling which uses liquid for cold plates and air as the secondary medium of cooling the associated electronics, we foresee using immersion-based fluids to cool the rest of the electronics in the server. The broader focus of this research is to study the effects of adopting immersion cooling, with integrated cold plates for high-performance systems. Although there are several other factors involved in the study, the focus of this paper will be the optimization of cold plate microchannels for immersion-based fluids in an immersion-cooled environment. Since immersion fluids are dielectric and the fluids used in cold plates are conductive, it exposes us to a major risk of leakage into the tank and short-circuiting the electronics. Therefore, we propose using the immersed fluid to pump into the cold plate. However, it leads to a suspicion of poor thermal performance and associated pumping power due to the difference in viscosity and other fluid properties. To address the thermal and flow performance, the objective is to optimize the cold plate microchannel fin parameters based on thermal and flow performance by evaluating thermal resistance and pressure drop across the cold plate. The detailed CFD model and optimization of the cold plate were done using Ansys Icepak and Ansys OptiSLang respectively.

Keywords: Hyperscale Datacenters, hybrid cooling, high-density racks, high-performance computing, Immersion cooling

NOMENCLATURE

TDP	Thermal Design Power
HPC	High-Performance Computing
CDU	Coolant Distribution Unit
CFD	Computational Fluid Dynamics
DLCS	Direct Liquid Cooling System
GWP	Global Warming Potential

1. INTRODUCTION

A data center is a physical facility where organizations store their critical applications and data. The design of a data center is built on a network of computing and storage resources that allow the delivery of shared applications and data. In the AI era, deep Learning, machine learning, and big data need enormous amounts of CPU power and computing resources. This implies that a large number of high-performance processors, such as high-performance CPUs, GPUs, FPGAs, and ASIC devices are required [1]. Due to the limitations of air cooling in dissipating growing power densities in servers, researchers have been driven to seek newer and more effective cooling alternatives [2-8]. The next radical change in the thermal management of data centers is to shift from conventional cooling methods like air cooling to direct liquid cooling (DLC) [2-8].

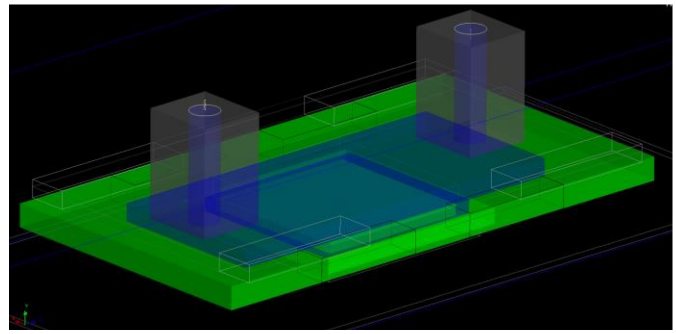
Submerging servers and IT equipment in a dielectric medium for cooling results in significant energy savings due to the high energy loads and density. Its heat capacity per volume is 1120–1400 times that of air [9]. Furthermore, the rack density as a function of Power Usage Effectiveness (PUE) shows that the rack power density for single-phase immersion cooling is nearly three times greater than for air cooling. While a conventional air-cooled system has a PUE of about 1.5, immersion cooling has a PUE of about 1.07, meaning a 36% reduction in power usage when employing immersion cooling [10]. The single-phase immersion cooling uses a dielectric fluid that helps in dissipating much higher heat from the components of the server. Some of its advantages include a high heat transfer coefficient, stable hydrodynamic flow, and the ability to directly cool hot components using the fluid. In single-phase immersion cooling, there is no phase change phenomenon, and the server along with its electronic components is immersed completely in the dielectric fluid. The heat transferred from the server components to the tank is cooled using an external heat exchanger or a coolant distribution unit (CDU) and finally discharged to the ambient through the primary side cooling units [11].

Thus, liquid cooling can assist to boost performance per watt while cutting total energy use [12-14]. A heat exchanger dumps the heat collected from the server modules to the facility water in a typical liquid-cooled data center server rack that utilizes a warm water direct liquid cooling system (DLCS). The heat exchanger's primary and secondary loops are the building facility water loop and the coolant loop to the server manifolds. The secondary loop coolant is routed through manifolds to cold plates, which are mounted on processor chips in individual servers with the thermal interface material (TIM) at the interface between the processor chips and the cold plates. The heat transfer coefficient ‘h’ between the entering coolant and the channel determines the amount of heat that can be taken up by the coolant going through the cold plate [15].

This paper focuses on the application of single-phase liquid immersion cooling along with a cold plate for high heat flux components. The cold plates typically have thin fin microchannels which transfer the heat from surfaces with high heat load to the fluid used in the system. It has two connectors at the end for the inlet and outlet flow. The fluid flows inside the cold plate, through the microchannels removing heat from the source. A baseline numerical model was created in Ansys Icepak with a cold plate attached to a heater with the whole setup immersed in a dielectric fluid. The same dielectric fluid is pumped through the cold plate as well. The baseline cold plate model was parametrically optimized for immersion cooling using ANSYS OPTISLANG with the fin spacings, fin thickness, and fin height as the variable parameters.

2. Computational Model

The cold plates are designed to suit each of the CPUs or GPUs in a server arrangement. A commercially available cold plate of overall dimensions – 110 mm x 85 mm x 41 mm, and with a base of - 55 mm x 55 mm x 4.5 mm, was chosen for this study. The CFD model of the server and cold plate setup is shown in Fig. 1.



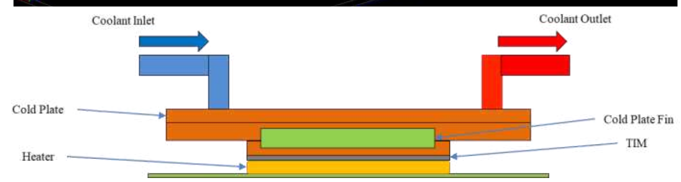


FIGURE 1: MODEL OF COLD PLATE & STACK-UP OF THE COLD PLATE ON THE HEATER.

The detailed CFD model of the cold plate was created using Ansys Icepak. The model stack-up consists of a ceramic heater (50 mm x 50 mm x 3 mm) simulating the heat load of an electronic chip CPU placed on top of a PCB. The fin thickness of the original cold plate was 0.2 mm, and the height was 3mm. A commercially available immersion cooling-specific Indium Heat-spring TIM (50 mm x 50 mm x 0.3 mm) was used as the interface material between the heater and the cold plate, and its thermal conductivity is 8 W/mK (calculated based on the data provided by vendor) which is slightly higher than the commonly used thermal grease. The cold plate was attached to the heater with a torque of 8 lbf-ft force which creates optimum contact and better heat transfer. The heater and cold plate setup are kept inside the domain containing immersion fluid. EC-100 is used as the dielectric coolant as that has high thermal conductivity and is compatible with components of the server with no or minimal risk of corrosion and has low GWP.

EC100 thermo-mechanical				
Temperature	Dynamic Viscosity	Density	Thermal conductivity	Specific Heat
°C	Kg/m-sec	Kg/m^3	W/m/K	KJ/kg-K
20	0.02193	845.86	0.1389	2.1317
25	0.01763	842.56	0.13853	2.1499
30	0.01439	839.26	0.13815	2.1683
35	0.01191	835.96	0.13778	2.1868
40	0.00998	832.66	0.1374	2.209
45	0.00847	828.7	0.13703	2.2243
50	0.00725	826.06	0.13665	2.2433

TABLE 1: ELECTROCOOL (EC) 100 THERMO-MECHANICAL PROPERTIES

The heater was assumed to represent a high-performance CPU that could reach a very high operating temperature during operation. Various sets of simulation were performed to validate the thermal resistance and pressure drop across the cold plate and use the data to optimize the cold plate design for immersion-based fluids to integrate the cold plate in immersion cooling. The boundary conditions were set as shown in Table 2.

Boundary Conditions	
Coolant Inlet Temperature	40°C
Coolant Type	EC-100
Coolant Flowrates	3 lpm
Heater Power	400 W
Server Flow	Natural Convection

TABLE 2: BOUNDARY CONDITIONS

Experimental Setup and Model Validation

To validate the CFD model, experimental data was collected for 400 W heater power with 3 sets of inlet temperatures as 25°C, 35°C and 45°C and a combination of different flow rates - 0.5 lpm, 1.0 lpm, 1.5 lpm, 2.0 lpm, 2.5 lpm & 3.0 lpm. EC100 coolant was used for experiments and validation. Figure 2 shows the schematic of the experimental setup.

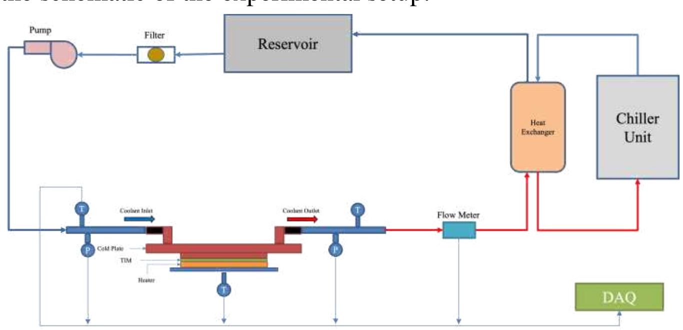


FIGURE 2: EXPERIMENTAL SETUP

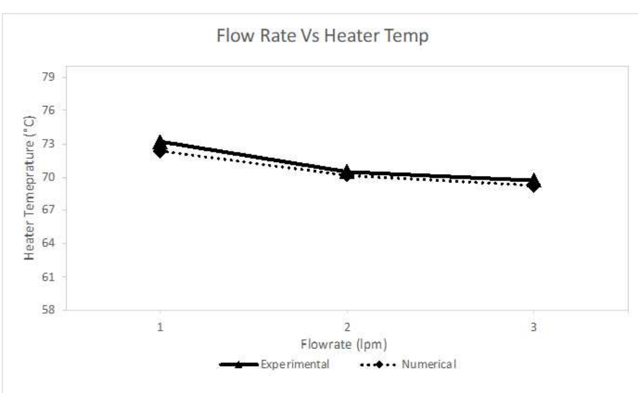


FIGURE 3: MODEL VALIDATION PLOT - HEATER TEMPERATURE VS. FLOW RATE

The model validation is shown in Fig. 3. Heater temperature was monitored for the same coolant inlet temperature and different flow rates. The experiments were conducted while the cold plate was in an air environment and the CFD model was validated with the same boundary conditions. The validated model was used to simulate the immersion fluid EC100 using OptiSLang to optimize the fin parameters (fin spacing, fin height, fin thickness) for the given boundary conditions.

Design Points for fin geometry			
No	Height	Thickness	Spacing
1	3	0.2	0.3
2	4	0.6	0.65
3	5	1.0	1
4		1.4	
Step Size	1	0.4	0.35
Total Discrete Values	3	4	3
Total No. of Design Points = 3*4*3 = 36			

TABLE 3: VARIABLE INPUT PARAMETERS

To optimize the fin parameters for the given inlet boundary conditions a set of 36 combinations were chosen as shown in TABLE 3. Three different values of fin heights and three different values of fin spacing with four distinct values of fin thickness are derived for the optimization variables. All the combinations models were simulated using Ansys Icepak coupled with Ansys OptiSLang to compare and get the optimized cold plate fin parameters.

4. RESULTS AND OBSERVATIONS

Thermal performance:

Fig. 4 shows thermal resistance as a function of fin thickness for different fin spacings at a fixed fin height of 3 mm. It was observed that the thermal resistance decreases as the fin thickness increases and this trend was observed for all fin spacings. Moreover, it is to be noted that the thermal resistance decreases as the fin spacings are increased. The fact that single-phase immersion fluids have higher viscosity can be attributed to the observation of decreasing thermal resistance with increased fin spacing as higher fin spacing, allows the viscous fluid to pass through the channel with ease preventing choking of the fluid at the channels. The decrease in thermal resistance with an increasing fin thickness was because of increased conduction heat transfer to the fins. These two reasons can be attributed to the reason why single-phase immersion heat sinks have higher pitch and fin thickness.

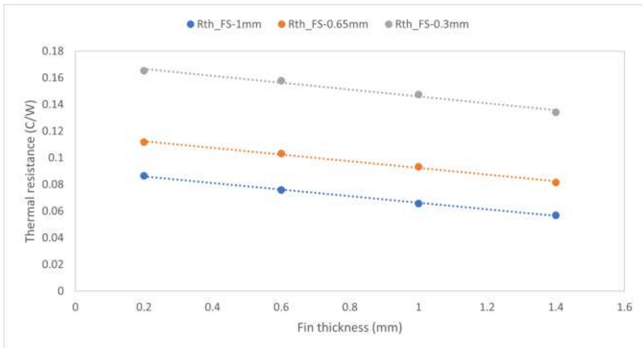


FIGURE 4: THERMAL RESISTANCE VS. FIN THICKNESS FOR DIFFERENT FIN SPACING (FS) AT A FIXED FIN HEIGHT OF 3 MM

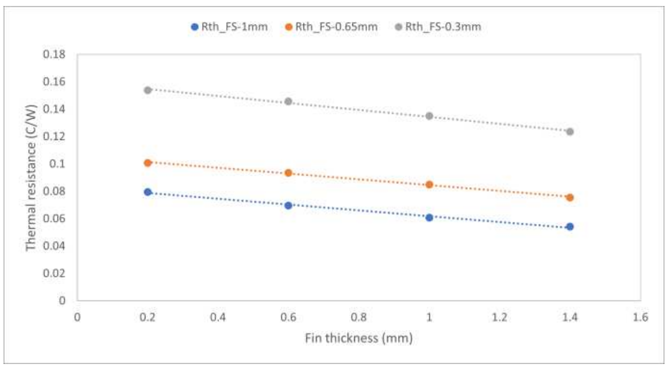


FIGURE 5: THERMAL RESISTANCE VS. FIN THICKNESS FOR DIFFERENT FIN SPACING (FS) AT A FIXED FIN HEIGHT OF 4 MM

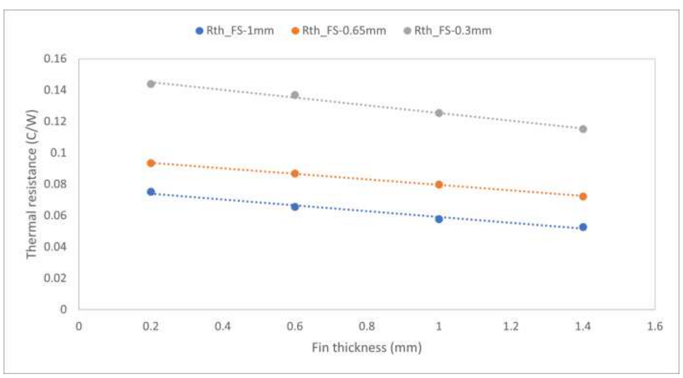


FIGURE 6: THERMAL RESISTANCE VS. FIN THICKNESS FOR DIFFERENT FIN SPACING (FS) AT A FIXED FIN HEIGHT OF 5 MM

It was observed that the trend from Fig.4 can be seen in Fig.5 and Fig.6 which show the thermal resistance as a function of fin thickness and fin spacing with fin height lengths of 4 mm and 5 mm respectively.

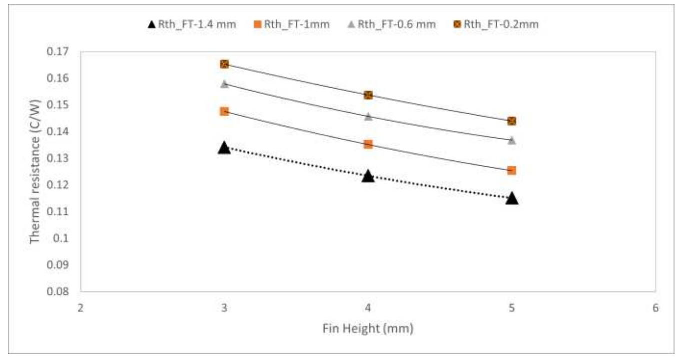


FIGURE 7: THERMAL RESISTANCE VS. FIN HEIGHT FOR DIFFERENT FIN THICKNESSES AT A FIXED FIN SPACING OF 1 MM

The above figure shows the thermal resistance vs fin height and fin thickness for a fixed fin spacing of 1mm and it was observed that the thermal resistance at the cold plate reduces when fin height increases, this is due to an increase in heat transfer surface area as the fin height increases thus enhancing the convection heat transfer mechanism.

Flow Performance:

The pressure drop across the cold plate as a function of fin thickness and fin spacing is shown in Fig.8. It can be seen that the pressure drop at the cold plate reduces dramatically when fin spacing increases (a maximum reduction of ~ 80% was observed). The fin thickness and fin height do not seem to affect the pressure drop significantly. The Coefficient of Prognosis (COP) matrix in Fig 9 forecasts the quality of the regression model for prognosis. This shows the change in pressure at the inlet and outlet of the cold plate with varying fin thickness, height, and spacing.

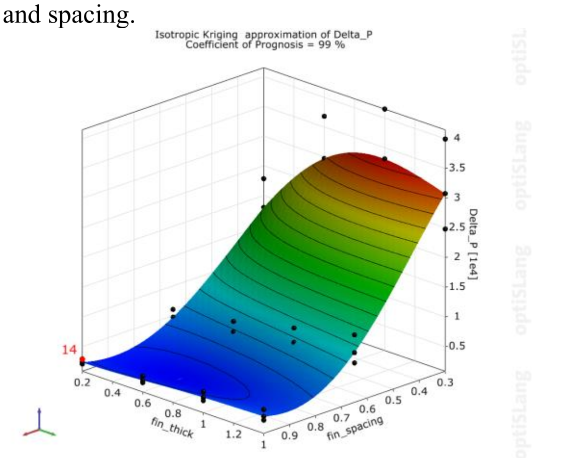


FIGURE 8: PRESSURE DROP ACROSS THE COLD PLATE AS A FUNCTION OF FIN THICKNESS AND FIN SPACING

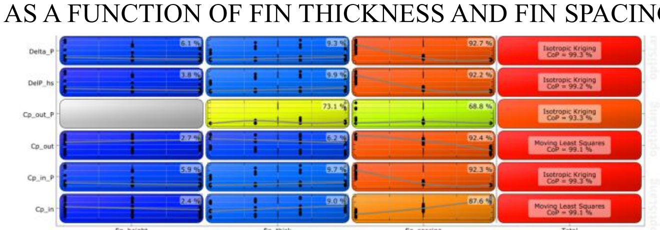


FIGURE 9: COP MATRIX OF RESULTS

Optimized Cold Plate:

From the 36 combinations of fin thickness, fin spacings, and fin heights the optimized cold plate was chosen which had the least thermal resistance. A comparison of the fin dimensions/ parameters of the optimized cold plate and the baseline cold plate used for optimization are shown in TABLE 4. As discussed in the thermal performance section previously it can be seen that an immersion-optimized cold plate's fins have higher spacing (owing to the viscous fluid), and higher fin thickness (to increase the conduction heat transfer to the fins).

Variables	Baseline Cold plate	Optimized Cold plate
Fin Spacing	0.3 mm	1.0 mm
Fin Height	3.0 mm	5.0 mm
Fin Thickness	0.2 mm	1.4 mm

TABLE 4: FIN PARAMETERS OF BASELINE COLD PLATE VS OPTIMIZED COLD PLATE

TABLE 5 shows the thermal performance of the baseline cold plate and the optimized cold plate. It was observed that the thermal resistance of the optimized cold plate was 0.025 °C/W.

Parameters	Baseline Cold Plate	Optimized Cold Plate
Heater Temperature (°C)	72	62
Inlet Fluid Temp (°C)	40	40
Δ T	32	37
Q (W)	400	400
Rth (°C/W)	0.08	0.025

TABLE 5: COMPARISON OF THERMAL PROPERTIES OF THE BASELINE COLD PLATE AND THE OPTIMIZED COLD PLATE

5. CONCLUSION

Also, with increasing edge deployments and expansion of 5G, immersion tanks operating at elevated ambient temperatures will be useful owing to simple cooling infrastructure and high heat capture capabilities. The major challenge in using cold plates in immersion tank environment is to set up the coolant delivery to the micro channels which involves internal pumps that supply the cold fluid from the tank directly to the cold plate. This adds on to additional power consumption which is a major trade-off with the heat transfer performance compared to heat sinks. Several conclusions drawn from the study are as follows:

• Using cold plates in an immersion-cooled environment, the heat transfer from the cold plate to the environment also contributes to enhanced heat transfer capability in cold plates.

• Optimization of cold plate fin geometries or heatsinks is purely based on the heat flux of the component and the thermo-mechanical properties of the fluid

• Immersion cooling supports provisioning even higher powered associated electronic components such as DIMMs

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