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# Optimization of the single-phase immersion cooling system: Impact of heat sink and coolant

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## ABSTRACT

With the development of data centers (DCs), traditional air cooling system was hard to meet cooling requirement. Single-phase immersion cooling system received more attention because of high performance and low energy consumption. However, the research for such technology is not enough. In this paper, the preferred heat sink is discussed, as well the coolant. Based on this, the heat sink is optimized according to different coolant. Results showed that, the performance of pin fin heat sink was better than that of micro-channel heat sink. Different coolants had different optimal fin spacing. The optimal fin spacing of FC3283 was 3 mm; for mineral oil and EC110, it was 5 mm. EC110 has the best performance in four coolants due to high thermal conductivity and low viscosity and density. The energy consumption can be reduced up to 0.22 W, compared to use mineral oil with 1 mm fin spacing.

**Keywords:** single-phase immersion cooling, heat sinks, fin spacing, coolants

## NONMENCLATURE

| | |
|---|---|
| *Abbreviations* | |
| DC | Data center |
| CPU | Central processing unit |
| *Symbols* | |
| u, v, w | Speed in *x*, *y*, *z* direction (m/s) |
| ρ | Density ($kg/m^3$) |
| P | Pressure (Pa) |
| $P_{in}$, $P_{out}$ | Pressure of inlet and outlet (Pa) |
| g | Gravitational acceleration ($m/s^2$) |
| t | Time (s) |
| $\vec{V}$ | Velocity vector (m/s) |
| T | Temperature (°C) |
| K | Thermal conductivity (W/(m·°C)) |
| $c_p$ | Heat capacity (J/(kg·°C)) |
| N | Number of CPUs |
| $T_{ave}$ | Average of CPU's temperature (°C) |
| $T_i$ | Temperature of CPU (°C) |
| W | Pump power (W) |
| V | Inlet flowrate ($m^3/h$) |

## 1. INTRODUCTION

With the rapid development of data center (DC), the energy consumption is received more attention. By 2025, the data center is predicted to consume 20% of global electricity, 40% of which was used to cooling system [1-2]. Moreover, the traditional cooling system, i.e., air cooling system, cannot meet the requirement of heat dissipation [3]. Therefore, finding a new cooling method is urgent. Single-phase immersion cooling directly put the servers into coolant to dissipate the heat [4]. It has the advantage of better cooling performance, lower energy, and higher stability [5].

Heat sink is also a key component in such cooling system. The type of heat sink and the parameter would affect its performance. For example, Li et al [6] optimized micro-channel heat sink by orthogonal method and found the optimal for fin height, thickness, and number were 30 mm, 1.5 mm, and 21, respectively. Sarangi et al. [7] suggested that large fin spacing is needed in immersion cooling. Through the optimization, central processing unit (CPU) temperature was decreased by 7 °C. Yang et al. [8] found uneven fin thickness for micro-channel heat sink could improve the Nusselt number. Moreover, the coolant, which is largely different with that of air, could also affect the performance. Finding a suitable parameter of coolant is the main work. For example, Shao et al. [9] compared the performance of four coolants. The results showed that coolant with lower viscosity could cool chips better. Niazmand et al. [10] compared the cooling performance of pure coolant and $Al_2O_3$ nanofluid. Results show that compared to pure coolant, 5% of nanofluid could decrease chip temperature by 3.5 °C.

However, the cooling performance can be simultaneously affected by the heat sink and coolant.

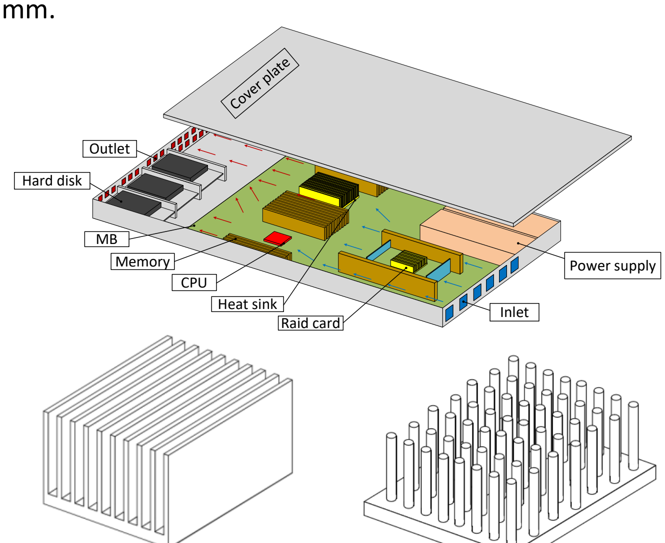
Their impact should be carefully considered. To bridge this knowledge gap, two types of heat sink (micro-channel heat sinks and pin fin heat sinks) and four coolants (FC43, FC3283, mineral oil and EC110) were employed to compare and discuss the cooling performance. The results obtained in this paper could provide guidance for designing and optimizing the liquid cooling heat sinks.

## 2. MODEL DESCRIPTION

### *2.1 Model description*

The server studied in this paper consists of two CPUs, twenty memories, six hard disks, one raid card and two power supplies. In order to simply the model, other components are ignored. A cover plate was embedded on the server, as shown in Fig. 1(a). The heat load of CPU is set as 120 W. Micro-channel heat sink and pin fin heat sink are used to compare their performance, as shown in Fig. 1(b). The detailed parameter for the model can be found in Table 1.

To discuss the effect of heat sink on cooling performance, the channel width and fin spacing is considered. Due to the limitation of heat sink, the channel width and fin spacing changes from 1 mm to 7 mm.




*(b) Micro-channel heat sink*  *(c) Pin fin heat sink*

*Fig.1 Models of server and heat sink*

*Table 1 Parameter of server and heat sink*

| | Item | Size (mm) | Number |
|---|---|---|---|
| Server | CPU | 40×35×3 | 2 |
| | Memory | 140×35×2 | 20 |
| | Raid card | 24×24×2 | 1 |
| | Cover plate | 741×420×2 | 1 |
| | Hard disk | 100×74×7 | 6 |
| | Power supply | 230×74×38 | 2 |
| | Shell | 741×420×44 | 1 |
| Heat sinks | Baseboard | 100×75×5 | - |
| | Fin height | 30mm | - |
| | Fin thickness | 3 mm | - |

Four coolants are used to explore its effect on cooling performance, that are FC3283, FC43, mineral oil, and EC110, respectively. The thermal properties are list in Table 2 [11-12].

Table 2 Thermal properties of four coolants

| Type | Viscosity (mPa·s) | Thermal conductivity (W/(m·°C)) | Heat capacity (J/(kg·°C)) | Density (kg/m$^3$) |
|---|---|---|---|---|
| FC3283 | 1.4 | 0.066 | 1100 | 1820 |
| FC43 | 4.7 | 0.065 | 1100 | 1860 |
| Mineral oil | 12 | 0.13 | 1670 | 849 |
| EC110 | 6.8 | 0.136 | 2212 | 832 |

The Creo Parametric and Floefd softwares are employed to establish model and conduct the simulation. The equation of continuity equation was [13-14]:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

$$\rho(\frac{D\vec{V}}{Dt})=\rho g-\nabla P+\mu\nabla^2\vec{V} \tag{2}$$

$$\frac{\partial(\rho T)}{\partial t}+\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}+\frac{\partial(\rho wT)}{\partial z}=\frac{\partial}{\partial x}(\frac{k}{c_p}\frac{\partial T}{\partial x})+\frac{\partial}{\partial y}(\frac{k}{c_p}\frac{\partial T}{\partial y})+\frac{\partial}{\partial z}(\frac{k}{c_p}\frac{\partial T}{\partial z}) \tag{3}$$

where, $u$, $v$, $w$ is the speed in $x$, $y$, $z$ direction, respectively. $\rho$ represents density, $P$ represents pressure, g represents gravitational acceleration and $t$ represents time, $\vec{V}$ represents velocity vector. T, k, $c_p$ represents inlet temperature, thermal conductivity and heat capacity, respectively.

### *2.2 Key performance indicators*

To assess cooling performance, average temperature and pump power are employed. Average temperature was the average of CPU temperature, which is the following [15]:

$$T_{ave}=\frac{1}{N}\sum_{i=1}^{N}T_i \tag{4}$$

where, $T_{ave}$ represents average temperature, $N$ is the number of CPUs, and $T_i$ is the temperature of CPUs.

Pump power can be obtained from pressure drop and inlet flowrate, as shown in the following [15]:

$$W=PV=(P_{in}-P_{out})V \tag{5}$$

where, $P_{in}$, $P_{out}$, and $V$ represents inlet pressure, outlet pressure, and inlet flowrate, respectively.

*2.3 Grid independence test and model validation*

The grid independence test is shown in Fig.2. With the number of grids increased, average temperature decreased. When grid number was larger than $113\times10^4$, the variation of average temperature was less than 1 °C. Therefore, this grid number was employed.

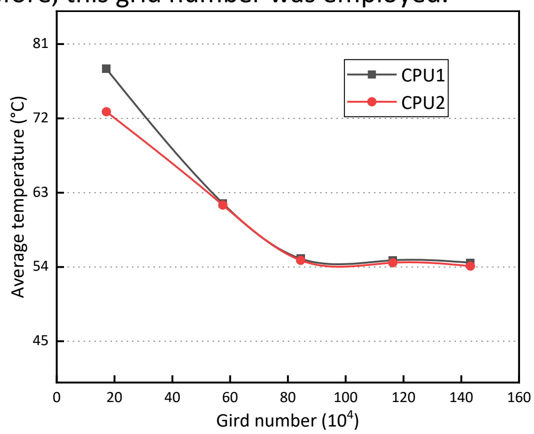



*Fig.2 Grid independence validation*

Experimental data from Chhertri et al. [16] is used to validate the proposed model. In the experiment, the heat load was set as 65 W and EC110 was employed. The flow rate ranged from 0.5 L/min to 2.5L/min with 40 °C. Fig. 3 shows the validation results. It can be found the simulation results showed good agreement with experimental data. The deviations of CPU temperature were less than 2 °C.

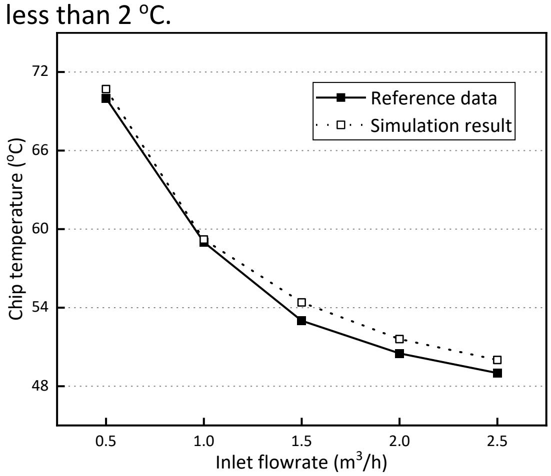



*Fig.3 Model validation*

## 3. RESULTS AND DISCUSSIONS

Fig. 4 compares the average temperature with different heat sinks and coolants, in which the channel width and fin spacing were 3 mm and inlet flowrate was 0.3 $m^3$/h. It was clear that, the average temperature by using pin fin heat sink was always lower than that of micro-channel heat sink. The reason was that flow resistance caused by pin fin heat sink was lower, which was conducive to flow. Therefore, pin fin heat sink was employed in the following content. Moreover, during the four coolants, EC110 showed the lowest average temperature, followed by FC3283, FC43, and mineral oil. This was mainly caused by EC110 has lower viscosity and higher thermal conductivity.

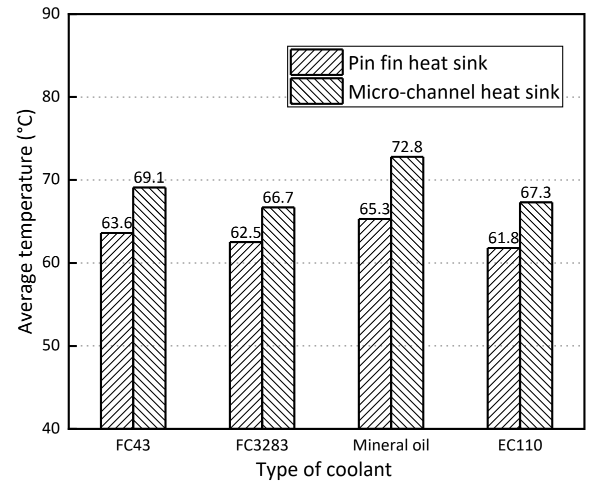



*Fig.4 Effect of heat sink and coolant on average temperature*

Fig. 5 shows the effect of flowrate on average temperature. It can be found the average temperature decreased with the increase of flowrate, but it changed slightly when the flowrate was larger than 0.3 $m^3$/h. This was mainly due to with degree of superheat between coolant and CPU decrease, heat is difficult to transfer. Moreover, since the highest viscosity, mineral oil had the highest average temperature; while EC110 had the lowest. For example, average temperature by using mineral oil was 6.2 °C higher than that by using FC43 at 0.1 $m^3$/h. In addition, when inlet flowrate increased to 0.5 $m^3$/h, average temperature by using mineral oil and FC43 was the same, this was caused by that mineral oil has higher thermal conductivity than FC43. Due to lower viscosity and higher thermal conductivity, EC110 was suggested.

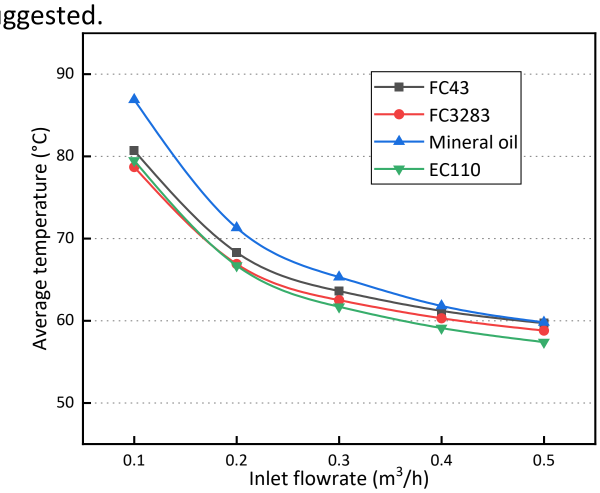



*Fig.5 Effect of inlet flowrate on average temperature*

Fig. 6 shows the impact of fin spacing on the average temperature. With the increase of fin spacing, average temperature would decrease firstly and then increase. The reason was that, with the increase of fin spacing, more coolant can flow through heat sinks but heat exchange area will decrease. Therefore, the optimal fin spacing ranged from 3 mm to 5 mm. Since the thermal property was different for different coolants. The optimal fin spacing was also different. For example, the optimal fin spacing was 3 mm for FC3283 and it was 5 mm for mineral oil and EC110.

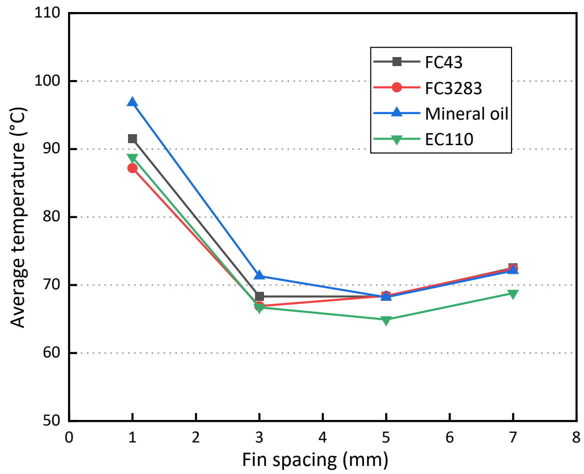



*Fig.6 Effect of different fin spacing on average temperature*

Fig. 7 shows the variation of pump power with different fin spacing and coolants, in which the average temperature was controlled at 70 °C. Although FC43 and FC3283 had better cooling performance when the fin spacing is less than 3 mm, pump power was still higher than mineral oil, which was due to the high density. Mineral oil had the highest viscosity in four coolants. More flowrate was need to decrease average temperature. Therefore, pump power caused by mineral oil was always higher than EC110. EC110 had the lowest pump power of 0.22 W when the fin spacing was 5 mm. Therefore, pin fin heat sink with 5mm of fin spacing and EC110 were recommended for such condition. Compare to use mineral oil with 1 mm fin spacing, the energy consumption can be reduced by 1.68 W.

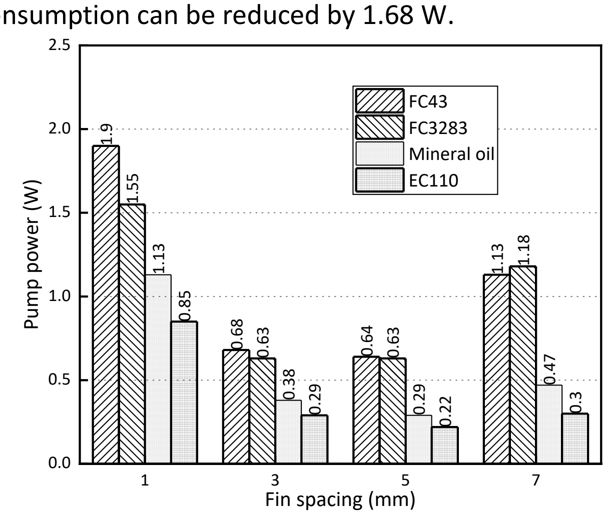



*Fig.7 Pump power under different fin spacing and coolants*

## 4. CONCLUSION

In this paper, the effect of parameter of heat sink and coolant on performance was studied. And two different type heat sinks were employed and compare their performance. Based on the results, it can be concluded as following:

(1) Compared to micro-channel heat sink, pin fin heat sink has less flow resistance, showing a lower average temperature.

(2) Different coolants had different optimal fin spacing. The optimal fin spacing of FC3283 was 3 mm; for mineral oil and EC110, it was 5 mm.

(3) EC110 has the best performance in four coolants due to high thermal conductivity and low viscosity and density. The energy consumption can be reduced up to 0.22 W, compared to that using mineral oil with 1 mm fin spacing.


## ACKNOWLEDGEMENT

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