



ARTICLE

Thermal Design Enhancement of a Single Phase Immersion Cooling System for Data Servers

Ali Anari Momenabadi¹, Ali Ahmadpour¹, Mohammad Reza Hajmohammadi¹
and Giulio Lorenzini^{2,*}

¹Department of Mechanical Engineering, Amirkabir University of Technology (Tehran Polytechnique), Tehran, Iran

²Department of Industrial Systems and Technologies Engineering, University of Parma, Parco Area delle Scienze 181/A, Parma, Italy

*Corresponding Author: Giulio Lorenzini. Email: giulio.lorenzini@unipr.it

Received: 14 May 2026; Accepted: 15 September 2026; Published: 27 October 2026

ABSTRACT: In this study, the thermal and hydraulic performance of a single-phase immersion cooling system for data servers was numerically investigated using three-dimensional CFD simulations in ANSYS Fluent. The flow was assumed to be laminar, incompressible, and steady, with Reynolds numbers ranging from 644 to 1932. The numerical model was validated against both experimental measurements and published numerical results, demonstrating good agreement. The effects of inlet–outlet arrangement, flow guide plate geometry, and Al_2O_3 nanofluid on the cooling performance were systematically investigated, and the optimized flow arrangement and flow guide plate were further evaluated in a dual-server configuration. Compared with the baseline T-type flow arrangement, the optimized inlet–outlet configuration produced only a modest improvement in thermal performance while slightly reducing the pressure drop. The optimized convex flow guide plate reduced the server case temperature by 10% with only a minor increase in pressure drop, whereas the addition of 2 wt.% Al_2O_3 nanoparticles to FC-40 further reduced the case temperature by 11.9%, yielding a thermal performance factor (TPF, indicating the balance between heat transfer enhancement and pressure drop penalty) of 1.11. For the dual-server configuration, the optimized flow arrangement and flow guide plates reduced the case temperatures of the first and second servers by 10% and 0.6%, respectively. These results demonstrate that combining geometric optimization with nanofluid enhancement provides an effective strategy for improving the thermal performance of single-phase immersion cooling systems for data servers.

KEYWORDS: CFD; heat sink; single-phase immersion cooling; data center cooling; nanofluid

1 Introduction

The rapid development of information and communication technologies has significantly increased the importance of data centers (DCs) as the backbone of modern digital infrastructure [1]. The continuous growth of cloud computing, artificial intelligence, and high-performance computing applications has resulted in increasing computational demands and higher power densities in data servers. During operation, a considerable amount of electrical energy is converted into heat, which must be effectively removed to ensure reliable operation, prevent thermal degradation, and maintain the lifetime of electronic components. Consequently, cooling systems account for a substantial proportion of the total energy consumption of data centers. It has been reported that data centers are responsible for approximately 1–2% of global electricity consumption, highlighting the urgent need for more efficient thermal management strategies [2].

Therefore, developing advanced cooling technologies with improved thermal performance and reduced energy consumption has become a critical requirement for next-generation data centers.

Currently, three major cooling approaches are employed in data centers: air cooling, indirect liquid cooling, and immersion cooling. Conventional air cooling relies on fans to circulate air through server components; however, its effectiveness decreases significantly with increasing heat flux, and approximately 40% of the total energy consumption of data centers may be associated with cooling systems [3]. Indirect liquid cooling improves heat removal capability by circulating a coolant through cold plates or closed-loop channels, but the additional thermal resistance between the coolant and electronic components, together with sealing requirements and system complexity, limits its further development. In contrast, immersion cooling provides direct contact between electronic components and a dielectric coolant, enabling more efficient heat transfer. Depending on the operating mechanism, immersion cooling can be classified into single-phase and two-phase systems. In two-phase immersion cooling, the coolant undergoes phase change through evaporation, offering excellent heat dissipation capability; however, its high cost, complex vapor management, and reliability concerns restrict large-scale implementation. Single-phase immersion cooling avoids phase-change-related issues by maintaining the coolant in a liquid state during operation and has demonstrated significant advantages in terms of simplicity, reliability, and scalability. Previous studies have shown that liquid-based cooling approaches without phase change can achieve substantially higher heat dissipation capability compared with air cooling [4]. Therefore, improving the thermo-hydraulic performance of single-phase immersion cooling systems has attracted increasing research attention.

Previous investigations on single-phase immersion cooling have primarily focused on optimizing operating conditions and coolant flow configurations to improve thermal performance. Muneeshwaran et al. [5] experimentally investigated the immersion cooling performance of a 1U server under different inlet and outlet arrangements, bypass flow conditions, flow rates, inlet temperatures, and heat sink designs. Their results demonstrated that the T-shaped inlet–outlet configuration provided the best thermal performance among the tested configurations, highlighting the significant influence of flow distribution on server temperature. Rokonzaman et al. [6] compared natural, forced, and mixed convection immersion cooling systems and reported that forced convection considerably enhanced cooling effectiveness compared with natural convection. Similarly, Kuncoro et al. [7] applied the Taguchi method to evaluate the effects of flow rate, inlet–outlet positions, and fan speed on cooling performance and energy consumption, revealing that flow-related parameters strongly influence both thermal behavior and system efficiency. Although these studies confirmed the importance of coolant flow management, most investigations were limited to evaluating individual flow parameters or several predefined configurations. A comprehensive optimization of inlet–outlet layouts considering both thermal performance and hydraulic characteristics remains insufficiently explored.

In addition to flow conditions, the arrangement of servers inside immersion tanks can significantly affect coolant distribution and temperature uniformity. Luiten [8] experimentally investigated the arrangement of three servers mounted on the tank wall under natural convection conditions and found that placing servers along the diagonal direction of the tank improved thermal performance. This study demonstrated that server positioning can influence the development of thermal gradients within immersion tanks. However, the investigation was conducted under natural convection conditions, and the interaction between server arrangement and forced coolant flow was not considered. Moreover, most previous studies have focused on single-server configurations, while the thermal interaction between multiple servers operating simultaneously remains less understood. Therefore, further investigation of multi-server immersion cooling

systems under forced convection conditions is necessary to evaluate flow interaction and prevent localized overheating.

To further improve coolant distribution and heat transfer performance, several researchers have explored modifications of the internal flow path. Li et al. [9] employed response surface methodology to optimize multiple parameters in a single-phase immersion cooling system and introduced flow dividers to modify the coolant pathway. Their results indicated that flow-path modification alone could reduce server temperature significantly, demonstrating the effectiveness of passive flow control structures. Zhang et al. [10] combined immersion cooling with jet impingement cooling and optimized geometric parameters, including jet-to-surface distance, nozzle diameter, and the number of jet nozzles, to achieve enhanced heat transfer with acceptable pressure losses. These studies confirmed that controlling coolant flow distribution through geometric modifications is an effective strategy for improving immersion cooling performance. However, existing approaches mainly rely on complex flow structures, such as flow dividers or jet impingement systems, which may increase manufacturing complexity and pressure losses. The influence of simple passive flow guide plates with different geometrical configurations on the thermo-hydraulic performance of immersion cooling systems requires further investigation.

Another promising approach for improving immersion cooling performance is the modification of coolant thermophysical properties through nanoparticle addition. Previous studies have demonstrated that metal oxide nanoparticles can enhance the thermal conductivity of base fluids and improve heat transfer characteristics [11–13]. Recent numerical investigations have further highlighted the potential of nanofluids for advanced thermal management applications. For example, studies on turbulent nanofluid flow considering complex effects such as oscillatory radiation, buoyancy, and Darcy–Forchheimer porous resistance demonstrated the significant influence of nanoparticle-enhanced fluids on heat transfer behavior [14]. However, most existing nanofluid studies have focused on conventional heat transfer systems rather than dielectric immersion cooling applications for data servers. In addition, although nanoparticle addition can improve thermal conductivity, the associated increase in viscosity may lead to higher pressure losses and additional pumping requirements. Therefore, evaluating the thermo-hydraulic trade-off of nanofluids in single-phase immersion cooling systems remains an important research topic.

Based on the above literature review, previous studies have demonstrated that the performance of single-phase immersion cooling systems can be improved through modifications of operating conditions, coolant flow distribution, and coolant thermophysical properties. However, several research gaps remain. First, although the influence of inlet–outlet arrangements has been investigated, a systematic optimization of flow configurations considering both thermal performance and hydraulic characteristics is still lacking. Second, previous flow enhancement techniques have mainly focused on complex structures, such as flow dividers and jet impingement systems, while the effects of simple passive flow guide plates with different geometrical designs on coolant redistribution and heat transfer enhancement have not been comprehensively evaluated. Third, most nanofluid studies have been conducted in conventional heat transfer systems, and the combined effect of nanoparticle-enhanced dielectric fluids and geometric optimization strategies in immersion cooling applications remains unclear. Furthermore, the thermal interaction between multiple servers under forced-convection immersion cooling conditions has received limited investigation. Therefore, the objective of the present study is to improve the thermo-hydraulic performance of a single-phase immersion cooling system for data servers through the combined application of geometric optimization and coolant enhancement strategies. A three-dimensional CFD model is employed to investigate four complementary approaches: (1) optimization of inlet–outlet layouts to improve coolant distribution, (2) implementation and geometric optimization of passive flow guide plates to enhance flow uniformity and heat transfer, (3) incorporation of

Al_2O_3 nanoparticles into FC-40 dielectric fluid to improve thermophysical properties, and (4) extension of the optimized cooling strategy from a single-server to a dual-server configuration. Unlike previous studies, which primarily examined these enhancement techniques independently, the present work systematically evaluates their combined and synergistic effects on the thermo-hydraulic performance of a single-phase immersion cooling system. The results provide useful insights into the design and optimization of efficient immersion cooling systems for high-power-density data centers.

2 Mathematical Modeling

2.1 Problem Description

In the present study, a single-phase immersion cooling (SPIC) system for data servers is numerically investigated. A 1U data server is placed inside an immersion tank, where the dielectric coolant FC-40 flows over the server to dissipate the generated heat. The baseline configuration considered in this study consists of the T-shaped inlet–outlet arrangement without a flow guide plate, which was adopted from the experimental study of Muneeshwaran et al. [5], where it was reported as the best-performing configuration among the investigated cases. Fig. 1 illustrates the geometry of the immersion tank, the server, and its major components, which are based on the configuration reported by Muneeshwaran et al. [5]. As shown in Fig. 1, the server is installed inside a rectangular immersion tank, with the CPU heat sink located at the center and two RAM heat sinks positioned on either side. The inlet and outlet diameters of the immersion tank are both 17 mm. The CPU heat sink is made of copper and has a base thickness of 4.5 mm, a fin height of 20.9 mm, a fin thickness of 0.25 mm, and a fin pitch of 2.2 mm. The RAM heat sink has a base thickness of 11 mm, a fin height of 23.1 mm, a fin thickness of 4.2 mm, and a fin pitch of 7.5 mm.

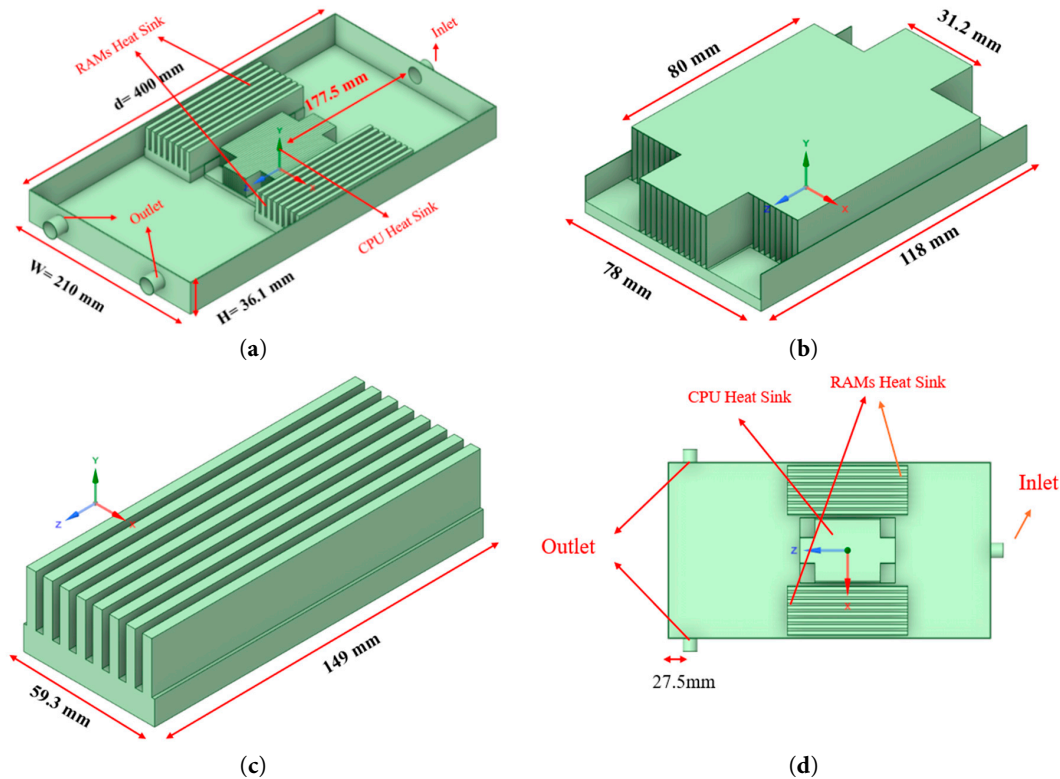


Figure 1: Geometry of the immersion tank with a single server: (a) immersion tank, (b) CPU heat sink, (c) RAM heat sink, (d) base configuration (T-type).

In the second part of this study, flow guide plates are introduced upstream of the CPU heat sink to modify the coolant flow distribution. Three different types of flow guide plates, including flat, concave, and convex configurations, are considered, as shown in Fig. 2. All guide plates have a constant width of 80 mm and are located at an x -position upstream of the CPU heat sink. For the flat guide plate, in addition to its installation position, the inclination angle (γ) and coverage height (h) are investigated as the main geometric parameters affecting its performance. Since the plate length (L) and coverage height (h) are geometrically related, the length of the flat guide plate is determined based on the selected coverage height. For the concave and convex guide plates, the effects of coverage height (h) and curvature radius (r), together with the plate position, are evaluated.

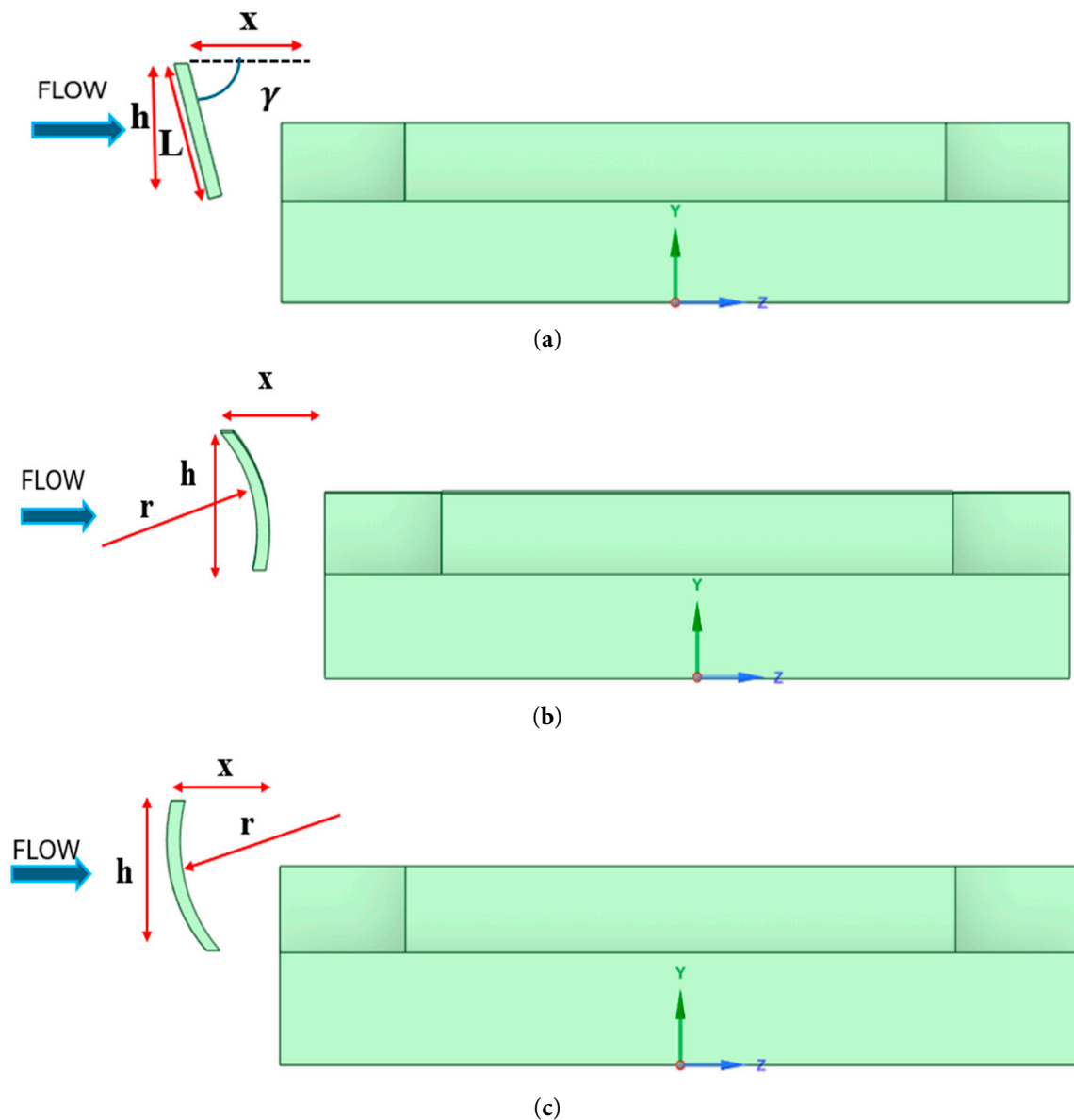


Figure 2: Geometry of the flow guide plate: (a) Flat plate, (b) Concave plate, (c) Convex plate.

In another part of this study, a dual-server immersion cooling system is analyzed. The baseline configuration of the dual-server system is defined as the original T-shaped inlet–outlet arrangement

without any flow guide plate. In this configuration, the first server is positioned 177.5 mm downstream from the inlet, while the second server is located 50 mm downstream from the first server, as shown in Fig. 3. During operation, the dielectric coolant directly contacts the electronic components and removes the generated heat through convective heat transfer. FC-40 is selected as the cooling fluid due to its electrical insulation capability, chemical stability, and low viscosity, which enable direct contact cooling of electronic components while maintaining reliable operation of the immersion cooling system. The thermophysical properties of FC-40 are presented in Table 1.

In the third part of this study, Al_2O_3 nanoparticles are dispersed into the FC-40 base fluid to enhance the heat transfer capability of the coolant. Al_2O_3 nanoparticles with an average diameter of 80 nm are selected due to their relatively high thermal conductivity, chemical stability, and non-metallic nature, which can improve the thermal transport characteristics of the base fluid while preserving the electrical insulation requirements of immersion cooling systems. The thermophysical properties of the Al_2O_3 nanoparticles are presented in Table 2.

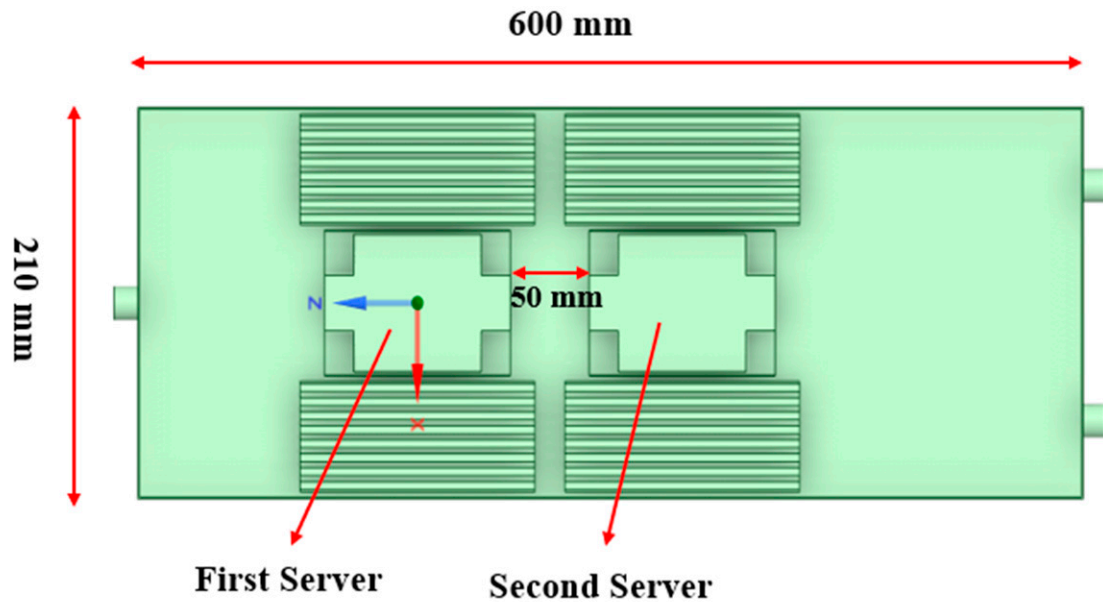


Figure 3: Geometry of the immersion tank with a two servers.

Table 1: Thermophysical properties of the FC-40 [5].

Properties		Values		
Density		1874.44 kg m ⁻³		
Specific heat	0°C	40°C	100°C	
	1014 J kg ⁻¹ K ⁻¹	1076 J kg ⁻¹ K ⁻¹	1169.4 J kg ⁻¹ K ⁻¹	
Kinematic viscosity	0°C	40°C	100°C	
	5e-6 m ² s ⁻¹	1.5e-6 m ² s ⁻¹	5.4e-6 m ² s ⁻¹	
Thermal conductivity	0°C	40°C	100°C	
	0.067 W m ⁻¹ K ⁻¹	0.0642 W m ⁻¹ K ⁻¹	0.0601 W m ⁻¹ K ⁻¹	

Table 2: Thermophysical properties of the Aluminum oxide (Al_2O_3) nanoparticles [13].

Properties	Values
Density	3970 (constant) kg m^{-3}
Specific heat	765 (constant) $\text{J kg}^{-1} \text{K}^{-1}$
Thermal conductivity	40 (constant) $\text{W m}^{-1} \text{K}^{-1}$

2.2 Governing Equations

In this study, the heat transfer process in the heat sink is investigated using a three-dimensional numerical model. The working fluid is assumed to be Newtonian, and the flow is considered laminar, incompressible, and steady. The effects of thermal radiation, phase change, chemical reactions, and viscous dissipation are neglected. Therefore, the continuity, momentum, and energy equations are solved for the fluid domain, as expressed in Eqs. (1)–(3). For the solid domain (heat sink), only the energy equation is solved, as presented in Eq. (4). The Reynolds number is calculated using Eq. (5). For the investigated flow rate range of 1–3 L/min, the Reynolds number varies from 644 to 1932. Since the Reynolds number remains below the critical value of approximately 2300 for internal flows, the flow is considered laminar, thereby justifying the use of the laminar flow model throughout the present study.

$$\nabla \cdot \vec{V} = 0 \quad (1)$$

$$\rho_f (\vec{V} \cdot \nabla \vec{V}) = -\nabla p + \nabla \cdot (\mu_f (\nabla \vec{V})) + \rho_f \vec{g} \quad (2)$$

$$\nabla \cdot (\vec{V} T) = \alpha_f \nabla^2 T \quad (3)$$

$$k_s \nabla^2 T = 0 \quad (4)$$

$$\text{Re} = \frac{\rho_f V_{in} D}{\mu_f} \quad (5)$$

In Eqs. (1)–(4), p is pressure, T is temperature, $\vec{g}$ is the gravity vector, $\vec{V}$ is the velocity vector, k_s is the thermal conductivity of the heat sink, and ρ_f , μ_f , and α_f are the density, viscosity, and thermal diffusivity of the fluid, respectively. In Eq. (5), V_{in} represents the average inlet velocity of the coolant, and D denotes the inlet diameter of the immersion cooling tank.

After the addition of aluminum oxide (Al_2O_3) nanoparticles to the FC-40 base fluid, the effective thermophysical properties of the resulting nanofluid are calculated using Eqs. (6)–(9) [13]. In these equations, ϕ represents the mass fraction of the nanoparticles, while the subscripts p , f , and nf denote the nanoparticles, base fluid, and nanofluid, respectively. In the numerical simulations, all thermophysical properties of the nanofluid are assumed to be temperature-dependent and are implemented as piecewise-linear functions over the investigated temperature range.

$$\rho_{nf} = \phi \rho_p + (1 - \phi) \rho_f \quad (6)$$

$$c_{p_{nf}} = \frac{\varphi \rho_p C_{p,p} + (1 - \varphi) \rho_f C_{p,f}}{\rho_{nf}} \quad (7)$$

$$\mu_{nf} = \mu_f (1 - \varphi)^{-2.5} \quad (8)$$

$$k_{nf} = k_f \left[\frac{k_p + 2k_f - 2\varphi(k_f - k_p)}{k_p + 2k_f + \varphi(k_f - k_p)} \right] \quad (9)$$

In evaluating the performance of the cooling system, the maximum server temperature, denoted as T_{case} , is considered the primary thermal performance indicator. Another important parameter is the thermal resistance, which is defined by Eq. (10) [5]. In Eq. (10), R_{th} , q , T_{in} , and T_{case} represent the thermal resistance, the server power, the inlet coolant temperature, and the maximum server temperature, respectively. Pressure drop is another key parameter for evaluating the hydraulic performance of the cooling system. Since heat transfer enhancement is generally accompanied by an increase in pressure drop, the thermal performance factor (TPF) is employed to evaluate the overall thermo-hydraulic performance. The TPF is defined according to Eq. (11) [15]. The exponent of $\frac{1}{3}$ is applicable under the constant pumping power criterion, in which the increase in heat transfer is evaluated together with the associated hydraulic penalty [15]. In Eq. (11), h_{ave} and h_{0ave} represent the average heat transfer coefficients of the modified and baseline configurations, respectively, while Δp and Δp_0 denote the corresponding pressure drops. The average heat transfer coefficient is calculated using Eq. (12) [5]. In Eq. (12), q , A_{base} , T_{case} , and T_{in} represent the server power, the base area of the heat sink, the maximum server temperature, and the inlet coolant temperature, respectively.

$$R_{th} = \frac{T_{case} - T_{in}}{q} \quad (10)$$

$$TPF = \frac{\frac{h_{ave}}{h_{0ave}}}{\left(\frac{\Delta p}{\Delta p_0} \right)^{\frac{1}{3}}} \quad (11)$$

$$h_{ave} = \frac{q}{A_{base}(T_{case} - T_{in})} \quad (12)$$

To solve the governing equations, appropriate boundary conditions are specified for all computational domains. At the tank inlet, a uniform velocity and a constant temperature of 35°C are imposed. The inlet velocity is determined from the prescribed volumetric flow rate, and simulations are performed for flow rates of 1, 2, and 3 LPM. At the tank outlet, a pressure outlet boundary condition with zero gauge pressure is applied. Since the heat generated by the RAM modules is significantly lower than that of the CPU, its contribution to the overall thermal performance is neglected [5,16,17]. A constant heat flux is imposed on the base surface of the CPU heat sink, where the heat flux is calculated by dividing the server power by the base area of the heat sink. Simulations are conducted for server power levels of 200, 300, 400, 500, and 600 W. The side walls of the computational domain are assumed to be adiabatic and satisfy the no-slip condition. At the solid–fluid interfaces, temperature continuity and heat flux continuity are imposed. The applied boundary conditions are summarized in Table 3.

Table 3: A summary of boundary conditions.

Location	Boundary Condition
Inlet	$u = U_i$ $T = T_i$
Outlet	$p_0 = 0$
Base of the CPU heat sink	$-k \frac{\partial T}{\partial n} = q''$
Side walls	$u = 0$ $\frac{\partial T}{\partial n} = 0$
Solid–fluid interface	$T_s = T_f$ $\frac{\partial T_s}{\partial n} = \frac{\partial T_f}{\partial n}$

2.3 Methodology

In the present study, computational fluid dynamics (CFD) simulations are performed to investigate the effects of different design parameters on the thermo-hydraulic performance of the single-phase immersion cooling system. The governing equations are solved using the finite volume method implemented in the commercial software ANSYS Fluent. Since the coolant flow is incompressible, the pressure-based solver is employed. Based on the Reynolds number analysis presented in Section 2.2, the flow is assumed to be laminar. Therefore, steady, incompressible, and laminar flow models are adopted throughout the numerical simulations.

The heat sink is assumed to be made of copper, while FC-40 and $\text{Al}_2\text{O}_3/\text{FC-40}$ nanofluid are used as the working fluids in the corresponding simulations. The thermophysical properties of both fluids are treated as temperature-dependent and are implemented using piecewise-linear functions. Pressure–velocity coupling is achieved using the SIMPLE algorithm. The convergence criteria are set to 10^{-3} for the continuity equation and 10^{-6} for the momentum and energy equations. The pressure equation is discretized using the Standard scheme, whereas the momentum and energy equations are discretized using the first-order upwind scheme.

3 CFD Analysis

3.1 Mesh Size Study

In the present study, an unstructured mesh is generated for both the fluid domain and the solid heat sink. The computational meshes are created using ANSYS Meshing, with a minimum element size of 0.5 mm. Fig. 4 illustrates the mesh distribution for the single-server configuration, and a similar mesh strategy is adopted for the two-server configuration. A mesh independence study is performed to ensure the accuracy and reliability of the numerical results. For the single-server configuration, the simulations are conducted at an inlet temperature of 35°C, a flow rate of 3 L/min, and a server power of 200 W. Based on the variation of the maximum server temperature (T_{case}), a mesh consisting of 7.8 million elements is selected as the grid-independent mesh. For the two-server configuration, the mesh independence analysis is performed at an inlet temperature of 35°C, a flow rate of 3 L/min, and a server power of 600 W. The results indicate that a mesh size of 14.4 million elements provides sufficient accuracy, and this mesh is adopted for subsequent simulations. The mesh independence results for the single-server and two-server configurations are presented in Tables 4 and 5, respectively.

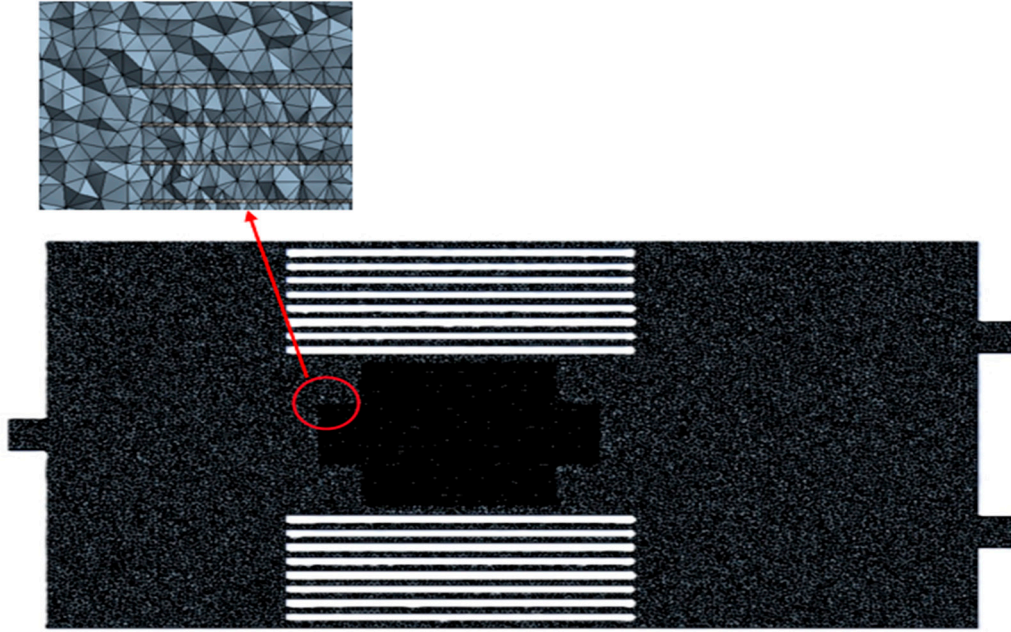


Figure 4: Applied mesh for the single-server configuration.

Table 4: Effect of mesh size on the single server case temperature when FC-40 is used as the coolant.

Number of Elements [million]	T_{case} [°C]	$Error\ T_{case} \times 100$ [%]
13	52.3	0
9	52.8	0.9
7.8	52.9	1.1
6	54.1	3.4

Table 5: Effect of mesh size on the two-server case temperature when FC-40 is used as the coolant.

Number of Elements [million]	T_{case1} [°C]	T_{case2} [°C]	$Error\ T_{case1} \times 100$ [%]	$Error\ T_{case2} \times 100$ [%]
16.7	86.8	100.1	0	0
14.4	87.8	100.7	1.1	0.6
12.1	89	101.9	2.5	1.8

3.2 CFD Validation

To ensure the accuracy of the simulation, the results are validated against previous studies. In this section, validation is performed using both numerical and experimental studies. These two validation cases denoted by “VC-1” and “VC-2” are presented as follows,

- Numerical simulation of the single-phase immersion cooling process using a dielectric fluid in a data server (VC-1):*** To validate the numerical model, the present simulations are first compared with the numerical results reported by Chhetri et al. [16], who investigated the single-phase immersion cooling of two electronic chips using EC-100 dielectric fluid. Their study adopted assumptions of laminar, incompressible, and steady-state flow with temperature-independent fluid properties, which are consistent with the governing assumptions of the present model. For the validation case, the inlet

temperature and heating power are set to 40°C and 90 W, respectively, following the reference study. The maximum junction temperature of the electronic chips is then compared with the results reported by Chhetri et al. [16]. As shown in Fig. 5, excellent agreement is achieved over the investigated flow rate range, with a maximum deviation of only 4.4%. This close agreement confirms the accuracy and reliability of the present numerical model.

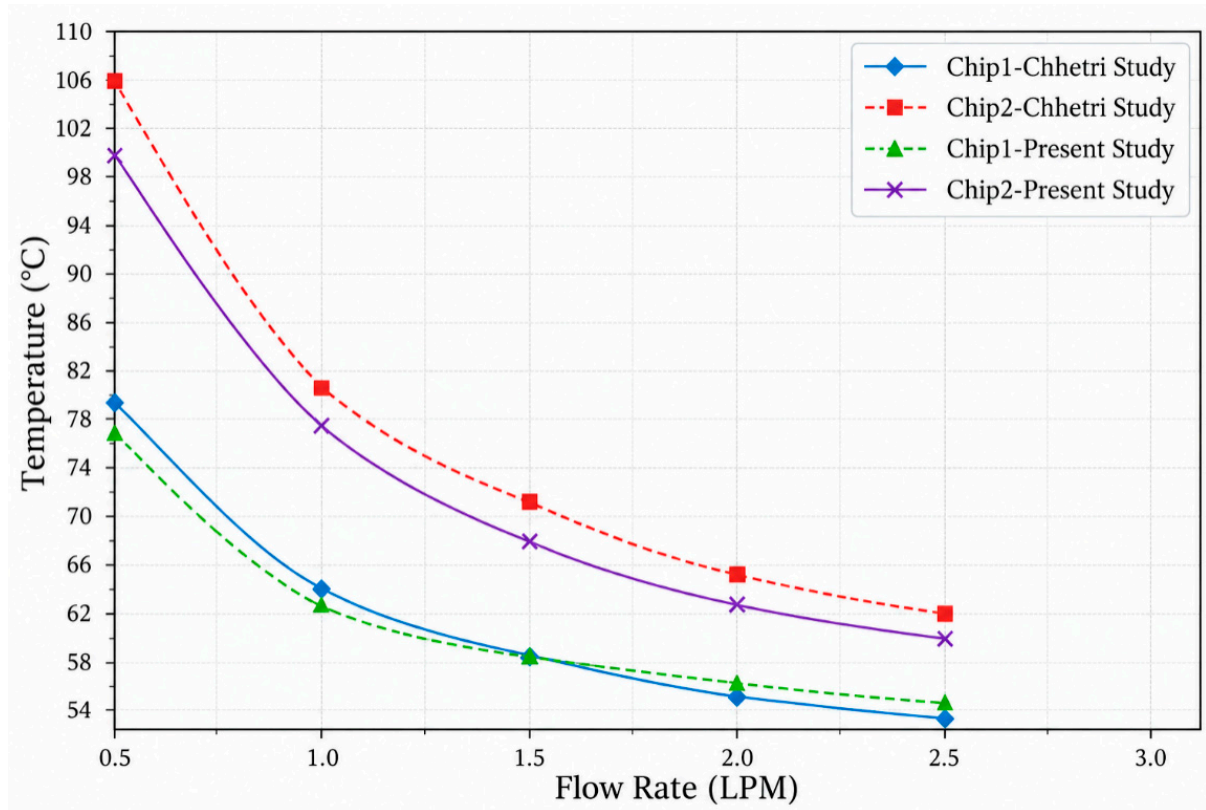


Figure 5: Comparison of the current numerical solutions and the numerical data of Chhetri et al. [16].

- Performance analysis of single-phase immersion cooling system of data center using FC-40 dielectric fluid (VC-2):*** A second validation is performed using the experimental study of Muneeshwaran et al. [5], who investigated the single-phase immersion cooling of a 1U data server using FC-40 as the dielectric coolant. This study is particularly suitable for validation because it employs the same cooling fluid and a server configuration similar to that considered in the present work. The experiments were conducted under laminar, incompressible, and steady-state flow conditions for different flow rates and server power levels to evaluate the server temperature and thermal resistance. For the validation case, the variation of the server case temperature with power is obtained at an inlet flow rate of 3 LPM and an inlet temperature of 35°C, following the experimental conditions reported in Ref. [5]. As shown in Fig. 6, the present numerical results are in good agreement with the experimental data, with a maximum deviation of 7.7%. The satisfactory agreement obtained from both validation cases demonstrates that the numerical model is capable of accurately predicting the thermo-hydraulic performance of the single-phase immersion cooling system.

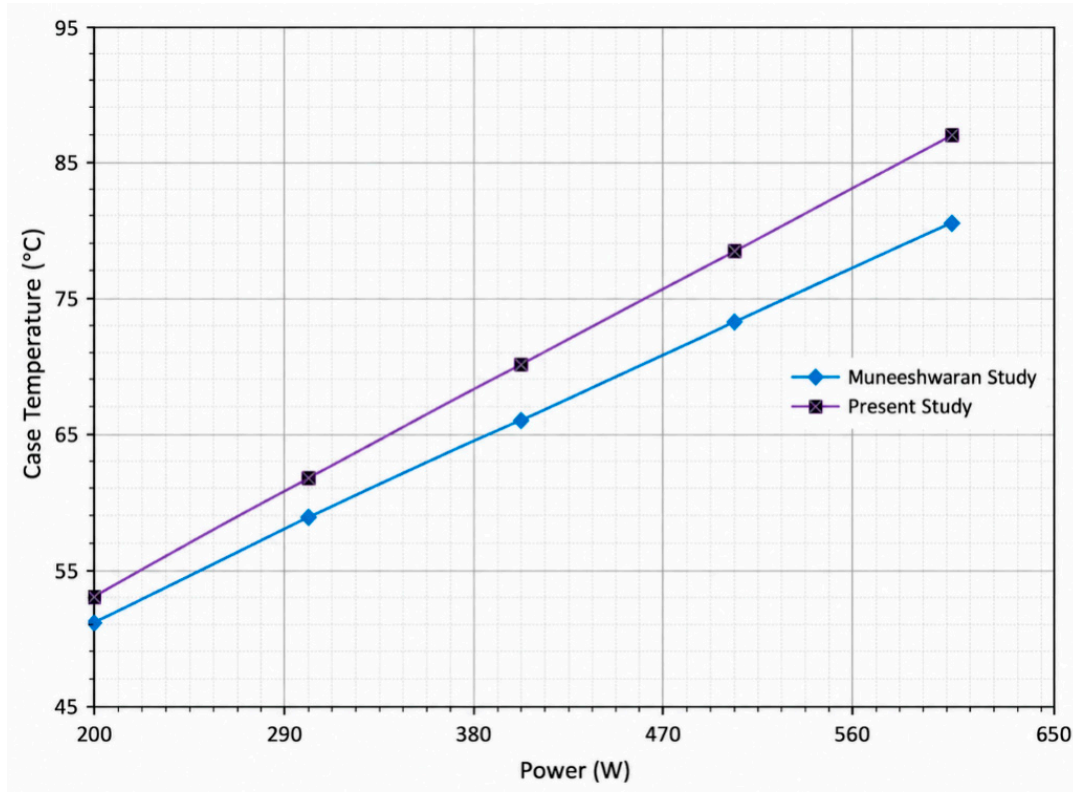


Figure 6: Comparison of the current numerical solutions and the experimental data of Muneeshwaran et al. [5].

4 Results and Discussion

In this section, the thermo-hydraulic performance of the single-phase immersion cooling system is systematically evaluated through a step-by-step optimization procedure. The primary objective is to reduce the server temperature while minimizing the associated pressure drop. First, the influence of the inlet–outlet configuration on the flow distribution and cooling performance is investigated to identify the optimal flow layout. Next, different flow guide plate configurations are evaluated by considering their position, geometry, and structural parameters to improve the coolant distribution around the CPU heat sink. Subsequently, the effect of Al_2O_3 nanoparticles on the thermo-hydraulic performance of the optimized cooling configuration is examined. Finally, the applicability of the optimized design is assessed for a two-server immersion cooling system by investigating the effect of the flow guide plate on the cooling performance of both servers.

4.1 Effect of Inlet–Outlet Configuration on Cooling Performance

The inlet–outlet configuration plays a key role in determining the flow distribution within the immersion cooling tank and, consequently, the cooling performance of the server. Since the CPU heat sink is located near the center of the tank and represents the primary heat source, positioning the inlet at the tank center directs the incoming coolant toward the CPU region, thereby enhancing convective heat transfer. Previous studies have shown that, for the present server geometry, the T-type configuration provides the best cooling performance among conventional inlet–outlet arrangements [5]. Therefore, the T-type configuration, illustrated in Fig. 7a, is selected as the baseline case for all subsequent comparisons in this section. The cooling performance of the proposed configurations is evaluated by comparing their case

temperatures with that of the baseline configuration, and the corresponding percentage improvement is calculated using Eq. (13).

$$\text{percentage improvement (percentage temperature reduction)} = \frac{T_{\text{case } T\text{-type}} - T_{\text{case}}}{T_{\text{case } T\text{-type}} - T_{\text{in}}} \times 100 \quad (13)$$

To investigate the effect of inlet–outlet arrangement on the cooling performance, two alternative flow configurations with a centrally located inlet are considered. In the first configuration, a single inlet and a single outlet are placed at the center of the tank, allowing the coolant flow to be directly directed toward the CPU heat sink. This arrangement is defined as the straight flow configuration and is illustrated in Fig. 7b. In the second configuration, one central inlet is combined with two outlets symmetrically positioned near the tank edges. The distance between each outlet and the corresponding tank edge is defined as a fraction of the tank width (W). This arrangement is referred to as the two-outlet configuration and is shown in Fig. 7c. The position of the outlets is described using the parameter xW , where x represents the fraction of the tank width. For instance, the $0.2W$ configuration (Fig. 7d) indicates that the inlet is located at the tank center, while the two outlets are positioned at a distance equal to 20% of the tank width from the tank edges.

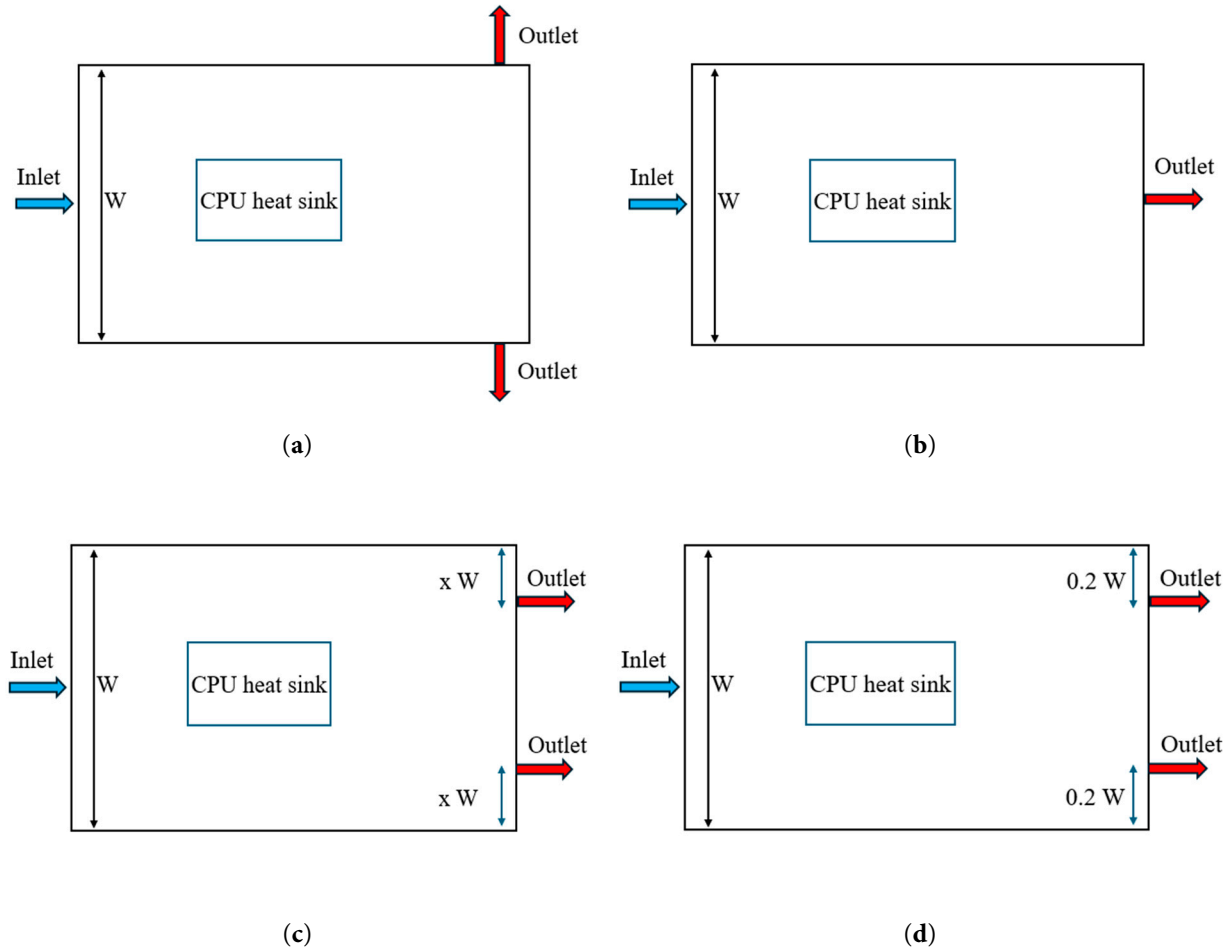


Figure 7: Different flow configurations; (a) T-type configuration, (b) Straight flow configuration, (c) Two-outlet flow configuration, (d) The $0.2W$ flow configuration is a type of two-outlet flow configuration.

To evaluate the effect of different inlet–outlet arrangements, simulations are performed at a flow rate of 3 L/min, a server power of 300 W, and an inlet temperature of 35°C. The corresponding results are summarized in Table 6. The straight flow configuration, which employs a single outlet, produces a considerably higher pressure drop than the two-outlet configurations because the coolant exits through a single flow path. In contrast, dividing the outlet flow into two passages reduces the flow resistance and, consequently, the pressure drop. The outlet location has only a marginal influence on the cooling performance, with all two-outlet configurations exhibiting similar case temperatures. Among the investigated cases, the 0.2W configuration provides the best overall performance, reducing the case temperature by 0.4% and the pressure drop by 0.6 Pa relative to the baseline T-type configuration. It should be noted that the baseline corresponds to the best-performing inlet–outlet configuration previously reported in the literature [5]. Therefore, the observed improvement demonstrates that further enhancement of the cooling performance can still be achieved through a simple modification of the outlet arrangement, without introducing additional cooling components or increasing the system complexity.

Table 6: Case temperature and pressure drop for different flow configurations.

Flow Configurations	T_{case} [°C]	Δp [pa]
T	61.6	33.8
Straight	61.7	93.5
0.1 W	61.6	33.2
0.2 W	61.5	33.2
0.3 W	61.7	33.4
0.4 W	62	33.2

4.2 Effect of Flow Guide Plate Configuration on Cooling Performance

In this section, the parameters affecting the performance of the flow guide plate are systematically investigated. First, the operating conditions under which the flow guide plate provides the greatest cooling enhancement are identified. For this purpose, a flat guide plate with $L = 20$ mm, $\gamma = 75^\circ$, and $x = 15$ mm is selected as the reference configuration, and its performance is evaluated under different server power levels and flow rates. Fig. 8 presents the variation of the case temperature with server power for the cases with and without the flow guide plate. As shown in Fig. 8, the case temperature increases with increasing server power because of the higher heat generation. The flow guide plate reduces the case temperature at all operating conditions by improving coolant delivery to the CPU heat sink. The reduction in case temperature ranges from 3.7% to 4.1%, with the maximum improvement observed at a server power of 600 W. At higher power levels, the larger temperature difference between the heat sink and the coolant increases the convective heat transfer potential, making the redirection of the coolant by the flow guide plate more effective. The influence of flow rate is summarized in Table 7, which shows that the guide plate provides the greatest cooling enhancement at a flow rate of 3 LPM. Therefore, the subsequent parametric study of the flow guide plate is carried out at a server power of 600 W and a flow rate of 3 LPM, where its influence on the cooling performance is most pronounced.

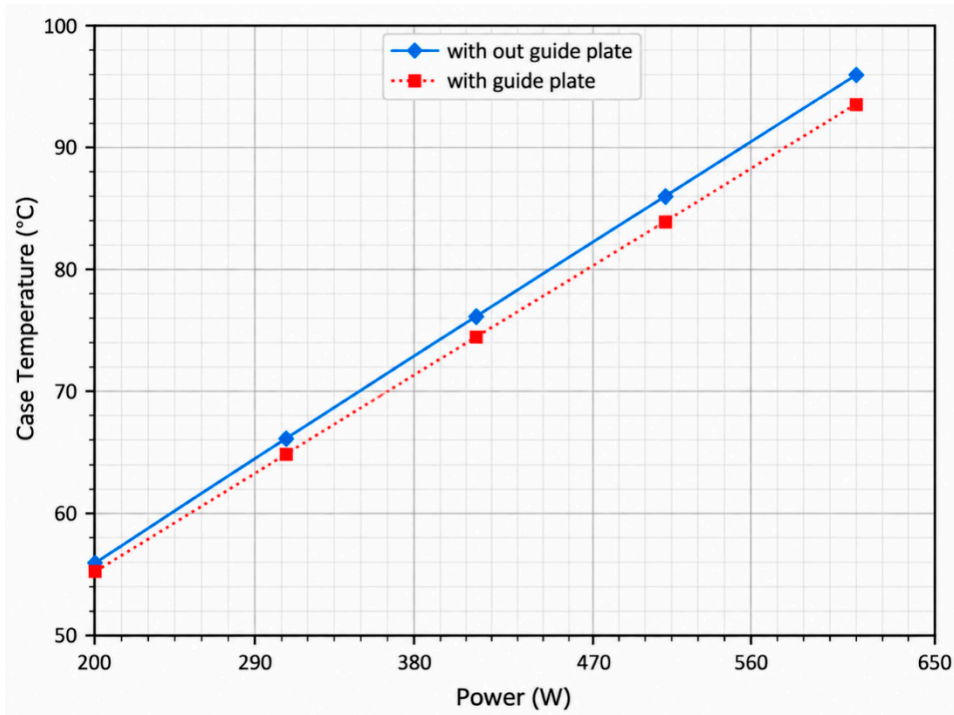


Figure 8: Case temperature versus power at $Q = 2$ LPM and $T_{in} = 35^\circ\text{C}$.

Table 7: Influence of the flow guide plate on case temperature at various flow rates for $Q = 3$ LPM and $q = 500$ W.

Flow Rate	Case Temperature without the Flow Guide Plate [$^\circ\text{C}$]	Case Temperature with the Flow Guide Plate [$^\circ\text{C}$]	Percentage Improvement [%]
1 LPM	105.8	104.9	1.3
2 LPM	86	84.2	3.5
3 LPM	79.5	76.8	6.1

To determine the optimal position of the flow guide plate, a flat plate with specifications of $L = 20$ mm and $\gamma = 75^\circ$ is placed at different x positions (as shown in Fig. 2a). Since excessive extension of the plate may interfere with the heat sink, x values greater than 15 mm are considered. The simulations are performed at a flow rate of 3 LPM and a server power of 600 W, and the results are summarized in Table 8. As observed, increasing the distance between the flow guide plate and the CPU heat sink reduces the cooling performance, resulting in a higher case temperature. When the plate is positioned farther from the heat sink, the coolant flow has more space to disperse before reaching the heat sink, which reduces the flow concentration in the CPU region and weakens the convective heat transfer. The flow behavior for different plate positions is illustrated by the streamlines shown in Fig. 9a–c. Without the flow guide plate (Fig. 9a), a portion of the coolant bypasses the CPU heat sink, limiting the heat removal capability. When the plate is positioned at $x = 15$ mm (Fig. 9b), the coolant is effectively redirected toward the CPU heat sink, increasing the flow concentration around the heat sink and enhancing heat transfer. However, at $x = 30$ mm (Fig. 9c), the larger distance between the plate and the heat sink allows the flow to spread before reaching the heat sink, resulting in weaker flow concentration and reduced cooling enhancement. Fig. 10 presents the variation of thermal resistance with the plate position. As shown in this figure, increasing the distance of the flow guide plate from the CPU heat sink increases the thermal resistance, indicating a reduction in

heat transfer capability. Consequently, the case temperature increases as the plate position moves away from the heat sink. Therefore, the optimal position of the flow guide plate is determined to be $x = 15$ mm.

Table 8: Effect of the flow guide plate position on case temperature at $Q = 3$ LPM and $q = 600$ W.

x [mm]	Case Temperature [$^{\circ}\text{C}$]	Percentage Improvement [%]
15	85.1	5.6
20	85.2	5.5
25	85.7	4.5
30	86.5	3

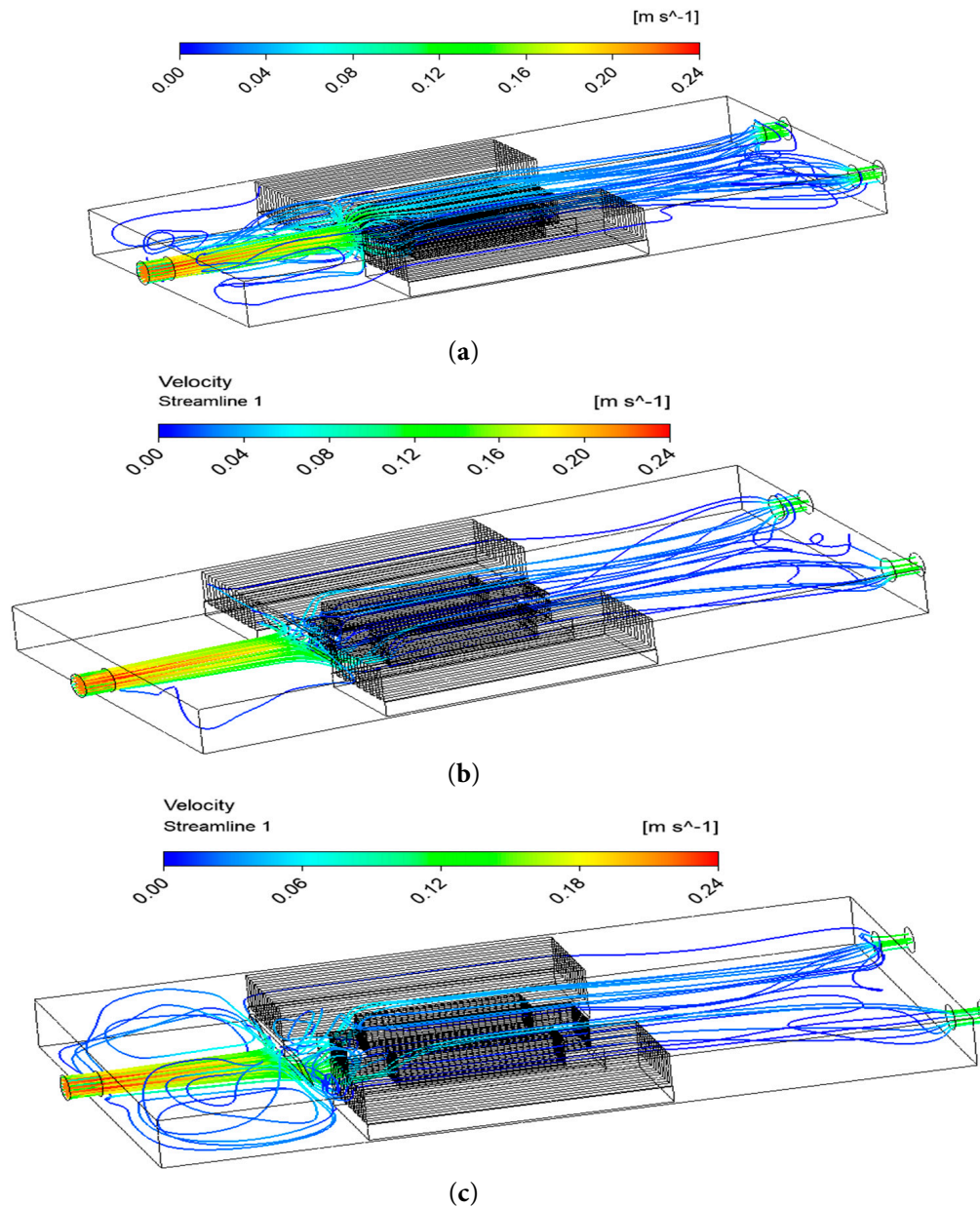


Figure 9: Stream line: (a) With out guide plate, (b) With flat guide plate at $x = 15$ mm, (c) With flat guide plate at $x = 30$ mm.

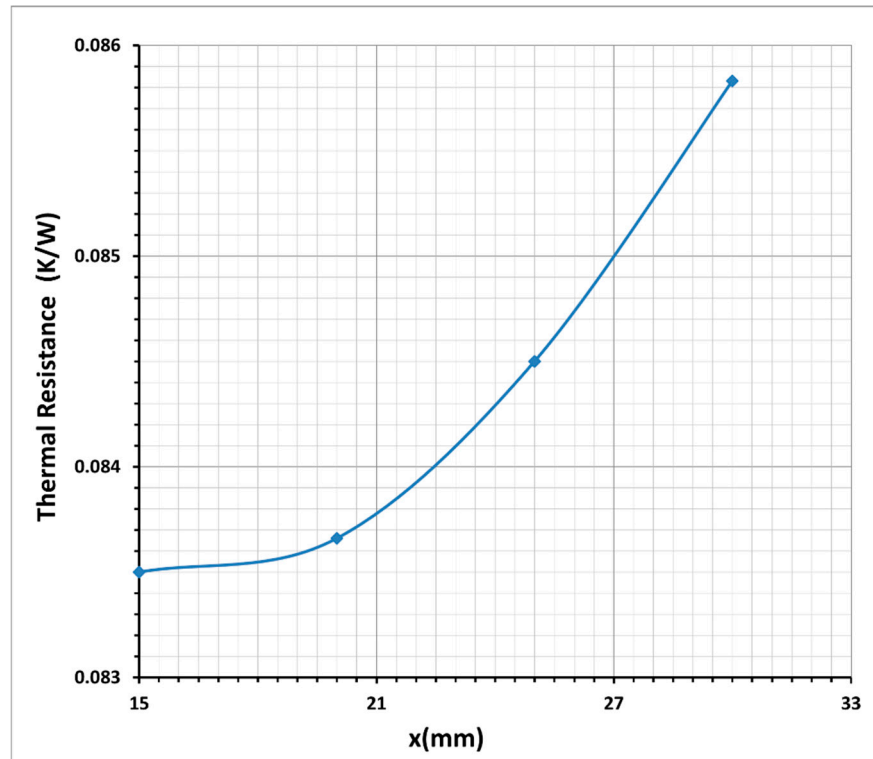


Figure 10: Thermal resistance versus flow guide plate position at $Q = 3$ LPM and $q = 600$ W.

After identifying the operating conditions at which the flow guide plate provides the greatest cooling enhancement and determining its optimal position, the effect of the plate geometry is investigated. Three guide plate geometries, namely flat, concave, and convex, are evaluated under identical operating and geometric conditions. The corresponding case temperatures and pressure drops are summarized in Table 9. As shown, the convex guide plate provides the lowest case temperature and pressure drop among the three configurations. Compared with the flat and concave plates, the convex geometry guides the incoming coolant more smoothly toward the CPU heat sink, increasing the amount of coolant flowing through the heat sink fins while minimizing unnecessary flow deviation. Consequently, the local convective heat transfer is enhanced without introducing a significant hydraulic penalty. Therefore, the convex guide plate is identified as the optimal geometry for the subsequent analyses.

Since the convex guide plate exhibits the best thermal performance while maintaining the lowest pressure drop among the investigated geometries, its geometric parameters are further optimized. As shown in Fig. 2c, the cooling performance of the convex guide plate is primarily governed by two parameters: the coverage height (h) and the curvature radius (r), while the plate position has already been optimized in the previous section. Fig. 11a presents the variation of the case temperature with the coverage height. The case temperature initially decreases with increasing h and reaches its minimum value at $h = 15$ mm, corresponding to a 9.8% reduction relative to the baseline configuration without a flow guide plate. Increasing the coverage height improves the ability of the guide plate to redirect the incoming coolant toward the CPU heat sink, thereby increasing the coolant flow through the heat sink fins and enhancing convective heat transfer. However, further increasing the coverage height provides no additional cooling benefit because the flow has already been effectively redirected toward the heat sink. The corresponding variation of thermal resistance is shown in Fig. 11b, where the minimum thermal resistance is also obtained

at $h = 15$ mm. The effect of the curvature radius is considerably smaller than that of the coverage height. As summarized in Table 10, the lowest case temperature is obtained at a curvature radius of $r = 11$ mm, indicating that the curvature radius has only a secondary influence on the cooling performance compared with the coverage height.

Table 9: Case temperature and pressure drop for the three flow guide plates, flat, convex, and concave, under geometric conditions of $h = 19$ mm, $x = 15$ mm, $r = 21$ mm, and $\gamma = 75^\circ$ at $Q = 3$ LPM and $q = 600$ W.

Plate Type	Case Temperature [$^\circ\text{C}$]	Pressure Drop [pa]
Flat plate	85.1	35.6
Concave plate	85	35.5
Convex plate	83.4	34.7

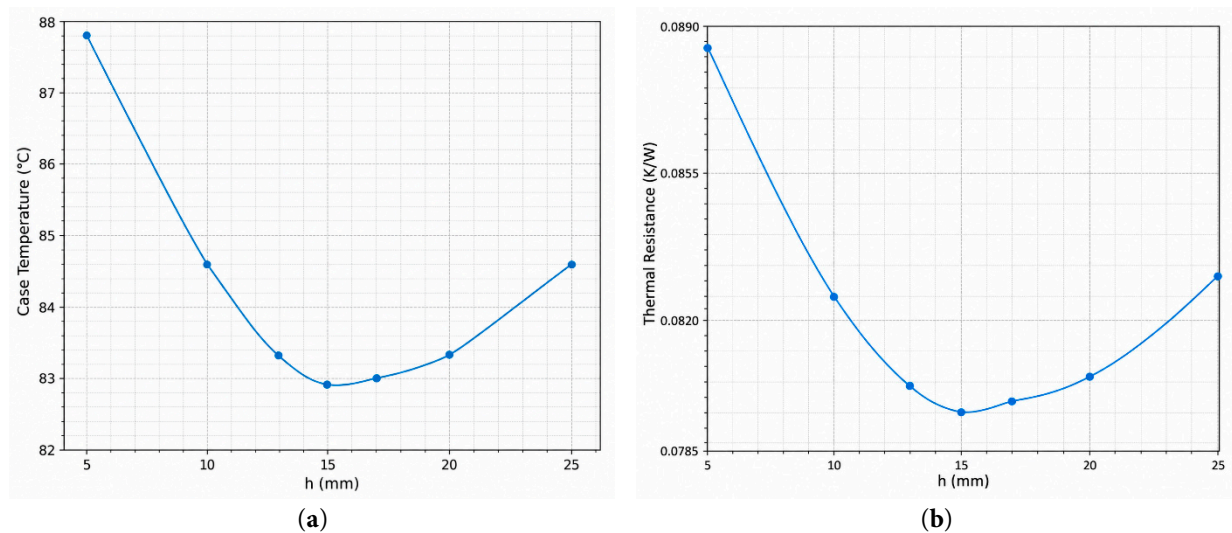


Figure 11: (a) Variation of case temperature with flow guide plate height, (b) Thermal resistance versus flow guide plate height.

Table 10: Effect of the flow guide plate curvature radius on case temperature under conditions of $x = 15$ mm, $h = 15$ mm, $Q = 3$ LPM, and $q = 600$ W.

Curvature Radius	Case Temperature [$^\circ\text{C}$]	Percentage Improvement [%]
9	82.9	9.8
11	82.8	10
16.5	82.9	9.8
21	83	9.6

In the baseline configuration, consisting of the T-type flow arrangement without a flow guide plate, the case temperature and pressure drop at a server power of 600 W and a flow rate of 3 LPM were 88.1°C and 33.2 Pa, respectively. Based on the optimization results, the 0.2W flow configuration was combined with the optimal convex flow guide plate ($x = 15$ mm, $h = 15$ mm, and $r = 11$ mm). Under these conditions, the case temperature decreased to 82.8°C , while the pressure drop increased only slightly to 34.5 Pa. The enhanced cooling performance can be attributed to the ability of the convex guide plate to effectively redirect the incoming coolant toward the CPU heat sink, increasing the coolant flow through the heat sink fins and

strengthening convective heat transfer. As a result, the optimized configuration achieved a 10% reduction in case temperature with only a 1.3 Pa increase in pressure drop. To evaluate the combined thermal and hydraulic performance, the thermal performance factor (TPF) was calculated and found to be 1.1. Since the TPF is greater than unity, the improvement in heat transfer outweighs the accompanying increase in pressure drop, confirming the effectiveness of the proposed flow guide plate design.

In addition to reducing the maximum server temperature, it is important to ensure that the introduction of the flow guide plate does not adversely affect the temperature distribution within the immersion tank. Fig. 12a presents the temperature contour of the baseline configuration without a flow guide plate. The temperature field is smoothly distributed throughout the tank, and no localized hot spots are observed around the CPU heat sink. Fig. 12b shows the corresponding temperature contour after introducing the optimized flow guide plate. Although the flow pattern is modified by the guide plate, the overall temperature distribution remains similar, and no additional hot spots are generated. At the same time, the maximum case temperature is noticeably reduced, demonstrating that the proposed flow guide plate enhances the cooling performance without adversely affecting the thermal stability of the immersion cooling system.

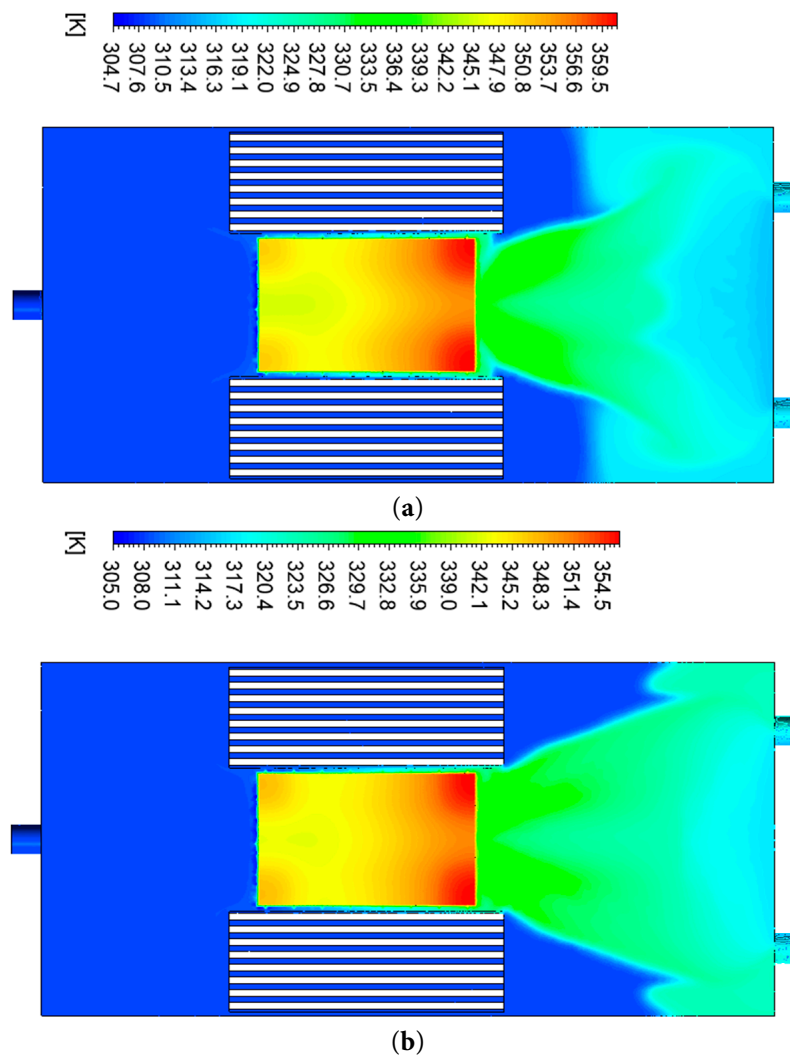


Figure 12: (a) Temperature contour of the server for the case without the flow guide plate at $Q = 3$ LPM and $q = 600$ W, (b) Temperature contour of the server in the presence of the flow guide plate at $Q = 3$ LPM and $q = 600$ W.

4.3 The Effect of Nanoparticles on the Thermal Performance of Server

In this section, aluminum oxide (Al_2O_3) nanoparticles are dispersed in the base fluid (FC-40) to further enhance the cooling performance of the optimized immersion cooling system. The nanoparticle mass fraction is varied from 0 to 2 wt%, and the corresponding thermophysical properties of the nanofluid are calculated using the correlations presented in Section 2. The simulations are performed for the optimized configuration obtained in Section 4.2, consisting of the 0.2W flow arrangement and the optimized convex flow guide plate, at a flow rate of 3 LPM and a server power of 600 W. Under these conditions, the simultaneous effects of geometric optimization and nanofluid application on the thermal performance of the cooling system are evaluated. As the nanoparticle concentration increases, the effective thermal conductivity, viscosity, and density of the nanofluid increase, whereas the specific heat decreases. The enhancement in thermal conductivity promotes heat transfer from the CPU heat sink to the coolant, while the accompanying increase in viscosity leads to a moderate increase in flow resistance. Therefore, the overall cooling performance is governed by the balance between these competing thermal and hydraulic effects.

As the nanoparticle mass fraction increases, the effective thermal conductivity of the nanofluid increases, promoting more efficient heat transfer from the CPU heat sink to the coolant. Although the addition of nanoparticles reduces the specific heat capacity of the nanofluid and increases its viscosity, the enhancement in thermal conductivity is the dominant effect under the investigated operating conditions. Consequently, the case temperature decreases as the nanoparticle concentration increases. As shown in Fig. 13a, the maximum reduction in case temperature is achieved at a nanoparticle mass fraction of 2 wt%, corresponding to an 11.9% improvement relative to the baseline configuration consisting of the T-type flow arrangement without a flow guide plate or nanoparticles. The increase in nanoparticle concentration also leads to a higher fluid viscosity, resulting in a gradual increase in pressure drop, as illustrated in Fig. 13b. To evaluate the combined thermal and hydraulic performance, the thermal performance factor (TPF) is presented in Fig. 13c. The TPF increases with nanoparticle concentration and reaches a maximum value of 1.108 at a mass fraction of 2 wt%. Since the TPF is greater than unity, the enhancement in heat transfer outweighs the accompanying increase in pressure drop. These results demonstrate that the combined application of the optimized flow configuration, the optimized convex flow guide plate, and Al_2O_3 nanoparticles provides the highest overall cooling performance among the investigated cases.

4.4 Effect of Flow Guide Plates on the Thermal Performance of a Two-Server Immersion Cooling System

In this section, a tank with a length of 600 mm is considered. As shown in Fig. 3, the first server is positioned 177.5 mm downstream of the inlet, while the second server is located 50 mm downstream of the first server. To improve the cooling performance of the two-server system, the optimized 0.2W flow arrangement and the optimized convex flow guide plate obtained in Section 4.2 are employed. The baseline case corresponds to the T-type flow arrangement without flow guide plates. Fig. 14 compares the case temperatures of the two servers at different power levels for the baseline configuration and the optimized configuration with two flow guide plates. The results demonstrate that the optimized flow guide plates provide a more significant cooling enhancement for the first server compared with the second server. The case temperature of the first server is improved by 10%, whereas the temperature improvement of the second server is limited to 0.6%. The greater improvement observed for the first server is attributed to the direct interaction between the redirected incoming coolant and the first CPU heat sink. The flow guide plate enhances the delivery of coolant toward the first server, increasing the local convective heat transfer rate. However, after passing over the first server, the coolant absorbs part of the generated heat, and its cooling

potential decreases before reaching the second server. Consequently, the influence of the second flow guide plate on the cooling performance of the second server is less significant, resulting in a smaller temperature reduction.

The hydraulic performance of the system is also evaluated by examining the pressure drop across the tank. The installation of two flow guide plates increases the pressure drop only slightly, from 37.4 Pa to 39.9 Pa, indicating that the additional flow resistance introduced by the plates is relatively small compared with the obtained thermal benefit. Fig. 15 presents the temperature contours of the two-server configuration with the flow guide plates installed. As shown, the optimized configuration reduces the maximum temperature of both servers without generating localized hot spots, demonstrating that the proposed flow guide plate design improves the thermal performance of the multi-server immersion cooling system without adversely affecting the overall temperature distribution.

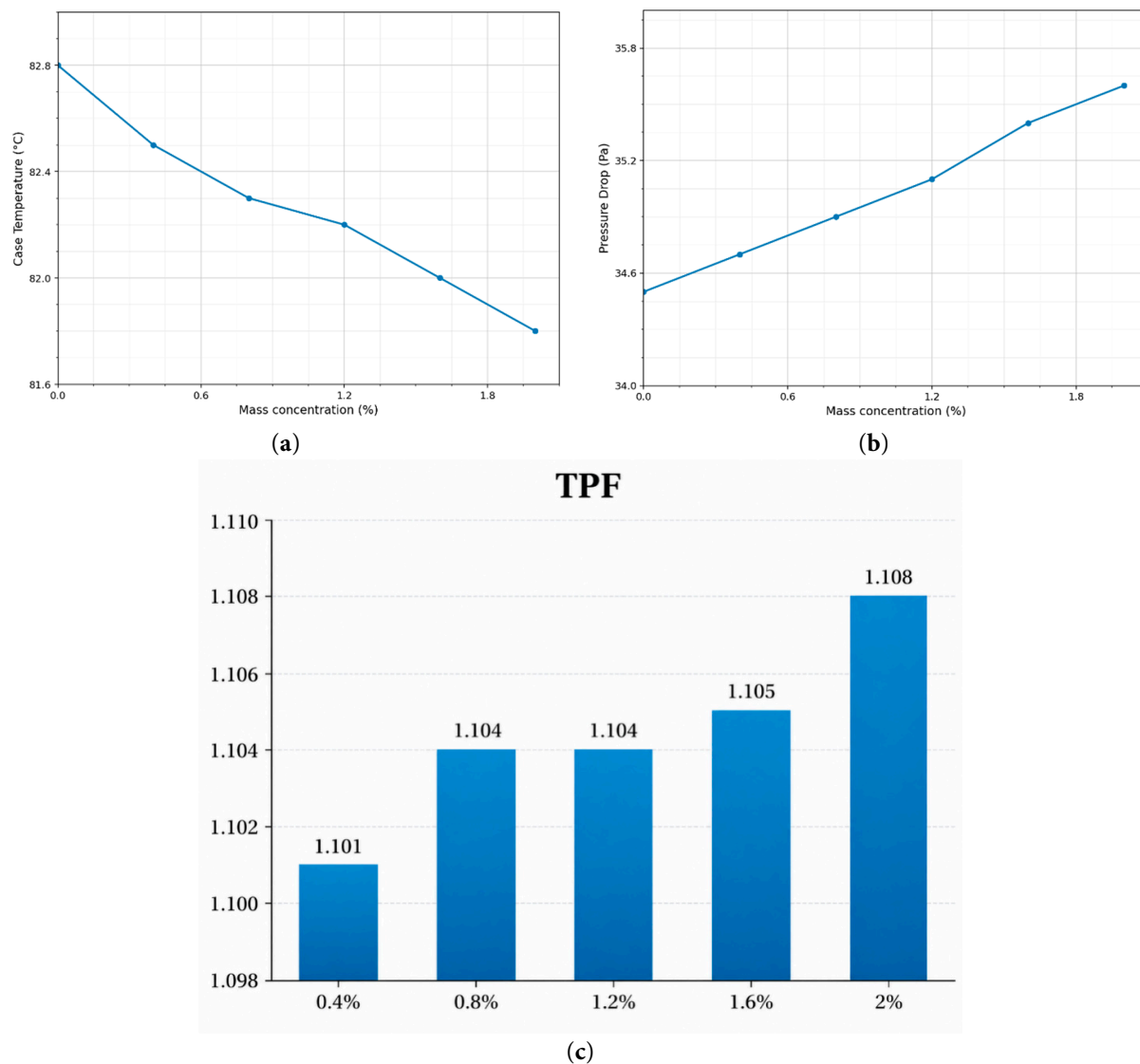


Figure 13: (a) Variation of case temperature with nanoparticle mass fraction, (b) Variation of pressure drop with nanoparticle mass fraction, (c) Bar chart of the thermal performance factor at different nanoparticle mass fractions.

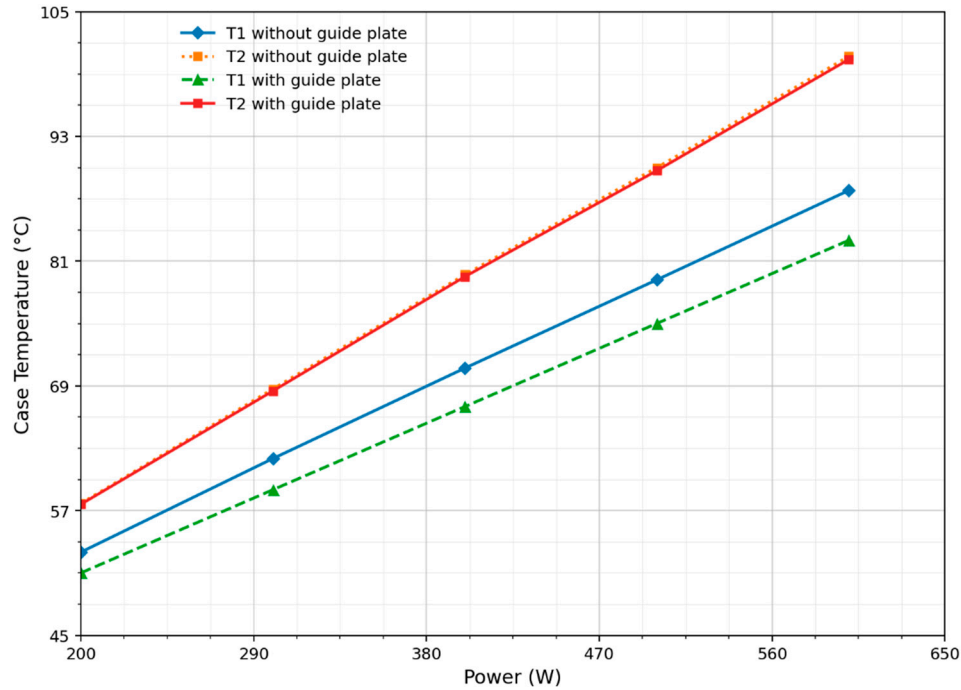


Figure 14: Case temperatures of the two servers versus power at $Q = 3$ LPM and $q = 600$ W.

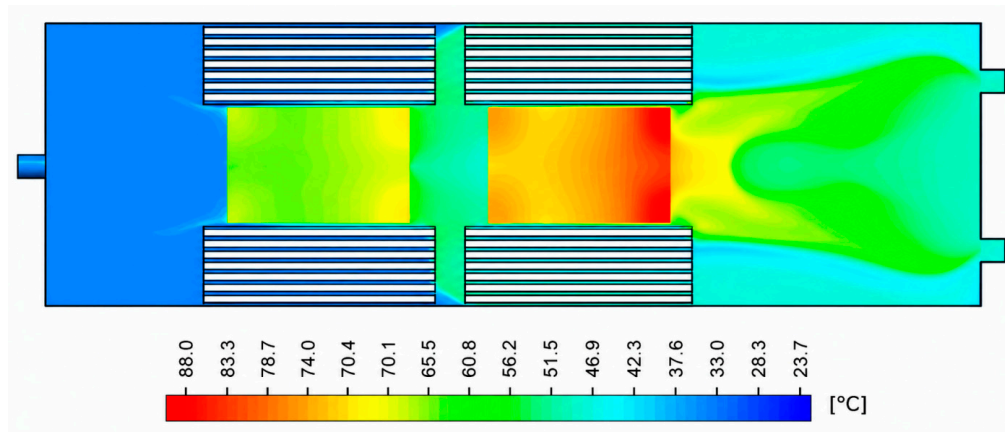


Figure 15: Temperature contour for the two servers at $Q = 3$ LPM and $q = 600$ W.

5 Conclusion

In the present study, the thermal and hydraulic performance of a single-phase immersion cooling system for data centers was enhanced through a comprehensive CFD investigation. First, the inlet–outlet flow arrangement was analyzed to identify the configuration that provides improved cooling performance with reduced flow resistance. Subsequently, a flow guide plate was introduced to redirect the coolant toward the CPU heat sink, and its geometric parameters were optimized to achieve the best thermal performance. Furthermore, Al_2O_3 nanoparticles were dispersed in the FC-40 base fluid to enhance the thermophysical properties of the coolant, and their effect was evaluated for the optimized single-server configuration. Finally, the applicability of the optimized flow arrangement and flow guide plate was investigated in a two-server immersion cooling system. The main findings of the present investigation can be summarized as follows:

1. Numerical analysis of different inlet and outlet configurations demonstrated that a single inlet combined with two outlets provides the lowest pressure drop among the investigated arrangements. The optimal configuration was obtained by locating the inlet at the center of the tank and positioning the two outlets at a distance of $0.2W$ from the side walls. Compared with the best configuration reported in previous studies, this arrangement resulted in a 0.4% reduction in case temperature and a 0.6 Pa decrease in pressure drop. Although the influence of the external flow arrangement on the thermal performance was limited, the optimized configuration provided a more favorable hydraulic condition for the immersion cooling system.
2. After optimizing the inlet–outlet arrangement, the effect of flow guide plates on the cooling performance was investigated. At a flow rate of 3 LPM and a heat load of 600 W, the maximum improvement was achieved using a convex flow guide plate with a coverage height of 15 mm, a curvature radius of 11 mm, and a position of 15 mm upstream of the CPU heat sink. Compared with the baseline configuration (T-type flow arrangement without a flow guide plate), this optimized design reduced the case temperature from 88.1°C to 82.8°C , corresponding to a 10% improvement, while increasing the pressure drop only slightly from 33.2 Pa to 34.5 Pa. The obtained results confirm that the flow guide plate effectively redirects the coolant toward the heat sink and enhances heat transfer with a limited hydraulic penalty.
3. The effect of Al_2O_3 nanoparticles on the performance of the optimized immersion cooling system was evaluated. Compared with the baseline configuration (T-type flow arrangement without a flow guide plate and nanoparticles), the addition of 2 wt.% Al_2O_3 reduced the case temperature from 88.1°C to 81.8°C , corresponding to an 11.9% improvement in thermal performance. Although increasing the nanoparticle concentration resulted in a higher pressure drop due to increased viscosity, the thermal performance factor reached 1.11, indicating that the enhancement in heat transfer outweighed the additional hydraulic resistance. These results demonstrate the potential of combining nanofluid enhancement with geometric optimization to further improve immersion cooling performance.
4. For the dual-server configuration, the optimized $0.2W$ flow arrangement and the convex flow guide plates obtained from the single-server analysis were applied to evaluate their effectiveness under a more complex thermal load distribution. Compared with the baseline configuration without flow guide plates, the installation of flow guide plates reduced the case temperature of the first server by 10%, while the improvement for the second server was limited to 0.6%. The different responses of the two servers were attributed to their positions relative to the coolant inlet. The first server directly receives the redirected incoming coolant, resulting in a stronger enhancement of convective heat transfer. In contrast, the coolant reaching the second server has already absorbed heat from the first server, reducing the cooling potential and limiting the impact of the second flow guide plate. The pressure drop increased only slightly from 37.4 Pa to 39.9 Pa, demonstrating that the optimized flow arrangement and flow guide plates can improve the thermal performance of multi-server immersion cooling systems with a limited hydraulic penalty.
5. Overall, the proposed strategy combining external flow optimization, internal flow guidance, and nanofluid enhancement provides an effective approach for improving the thermal performance of single-phase immersion cooling systems for data servers. Among the investigated approaches, the flow guide plate and Al_2O_3 nanofluid provided the most significant reductions in case temperature while maintaining acceptable hydraulic performance in the single-server configuration. In addition, the application of the optimized flow arrangement and flow guide plates demonstrated the potential for improving the thermal performance of the two-server immersion cooling system. The numerical results

highlight the effectiveness of the proposed design strategies for high-density server cooling; however, further experimental validation and investigation of long-term nanofluid stability are required for practical implementation.

Acknowledgement: Not applicable.

Funding Statement: The authors received no specific funding for this study.

Author Contributions: Ali Anari Momenabadi: conceptualization, methodology, investigation, numerical simulation, data analysis, and writing—original draft. Ali Ahmadpour, Mohammad Reza Hajmohammadi and Giulio Lorenzini: supervision, scientific guidance, and critical review and editing of the manuscript. All authors reviewed and approved the final version of the manuscript.

Availability of Data and Materials: The data supporting the findings of this study are available from the corresponding author upon reasonable request.

Ethics Approval: Not applicable.

Conflicts of Interest: Given his role as an Editorial Board Member/Guest Editor of this journal, Giulio Lorenzini had no involvement in the peer review of this article and had no access to information regarding its peer review. Full responsibility for the editorial process for this article was delegated to another journal editor. The authors declare no other conflicts of interest.

References

1. Yamin M. Information technologies of 21st century and their impact on the society. *Int J Inf Technol.* 2019;11(4):759–66. [\[CrossRef\]](#).
2. Cho J, Kim Y. Development of modular air containment system: Thermal performance optimization of row-based cooling for high-density data centers. *Energy.* 2021;231:120838. [\[CrossRef\]](#).
3. Ding T, Chen X, Cao H, He Z, Wang J, Li Z. Principles of loop thermosyphon and its application in data center cooling systems: A review. *Renew Sustain Energy Rev.* 2021;150:111389. [\[CrossRef\]](#).
4. Xu S, Zhang H, Wang Z. Thermal management and energy consumption in air, liquid, and free cooling systems for data centers: A review. *Energies.* 2023;16(3):1279. [\[CrossRef\]](#).
5. Muneeshwaran M, Lin YC, Wang CC. Performance analysis of single-phase immersion cooling system of data center using FC-40 dielectric fluid. *Int Commun Heat Mass Transf.* 2023;145:106843. [\[CrossRef\]](#).
6. Rokonzaman ASM, Erdem K, Şahin B, Özdemir MR. Experimental study of a single-phase immersion cooling system with natural and forced convection. *Int J Therm Sci.* 2025;214:109868. [\[CrossRef\]](#).
7. Kuncoro IW, Pambudi NA, Biddinika MK, Budiyo CW. Optimization of immersion cooling performance using the Taguchi Method. *Case Stud Therm Eng.* 2020;21:100729. [\[CrossRef\]](#).
8. Luiten W. Single phase passive hydrocarbon immersion cooling of high-power ICs. In: *Proceedings of the 2021 27th International Workshop on Thermal Investigations of ICs and Systems (THERMINIC)*; 2021 Sep 23; Berlin, Germany. p. 1–6. [\[CrossRef\]](#).
9. Li X, Xu Z, Liu S, Zhang X, Sun H. Server performance optimization for single-phase immersion cooling data center. *Appl Therm Eng.* 2023;224:120080. [\[CrossRef\]](#).
10. Zhang Y, Wang Y, Kosonen R, Khan DA, Wang X. Numerical research on the flow and heat transfer characteristics in the immersion jet cooling for servers. *Case Stud Therm Eng.* 2024;60:104748. [\[CrossRef\]](#).
11. Luo Q, Wang C, Wen H, Liu L. Research and optimization of thermophysical properties of sic oil-based nanofluids for data center immersion cooling. *Int Commun Heat Mass Transf.* 2022;131:105863. [\[CrossRef\]](#).
12. Makaoui A, Admi Y, Moussaoui MA, Mezhab A. Numerical study of influence of nanofluids on the optimization of heat transfer in immersion cooling systems. *Processes.* 2025;13(3):620. [\[CrossRef\]](#).
13. Niazmand A, Murthy P, Saini S, Shahi P, Bansode P, Agonafer D. Numerical analysis of oil immersion cooling of a server using mineral oil and Al₂O₃ nanofluid. In: *Proceedings of the ASME 2020 International Technical*

- Conference and Exhibition on Packaging and Integration of Electronic and Photonic Microsystems; 2020 Oct 27–29; Virtual. [[CrossRef](#)].
14. Alshammari S, Ullah Z, Alam MM, Ibrahim AO, Hamdoun HY, Abu-Zinadah H. Finite difference analysis of turbulent nanofluid and heat fluctuation with oscillatory radiation, gravity and Darcy-Forchheimer porous medium via vertical cone. *Chaos Solitons Fractals*. 2025;193:116126. [[CrossRef](#)].
 15. Bejan A, Kraus AD. *Heat transfer handbook*. Hoboken, NJ, USA: John Wiley & Sons, Inc.; 2003.
 16. Chhetri A, Kashyap D, Mali A, Agarwal C, Ponraj C, Gobinath N. Numerical simulation of the single-phase immersion cooling process using a dielectric fluid in a data server. *Mater Today Proc*. 2022;51:1532–8. [[CrossRef](#)].
 17. Shrigondekar H, Lin YC, Wang CC. Investigations on performance of single-phase immersion cooling system. *Int J Heat Mass Transf*. 2023;206:123961. [[CrossRef](#)].