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Evaluation of Local Thermal Environment in Operational Data Center

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ABSTRACT: When redeploying servers or installing new racks within an existing data center, assessing the local rack-level thermal environment is crucial to ensure target areas meet stringent cooling requirements. To date, limited research has addressed this specific issue. In this study, experimental measurements and CFD simulations were conducted to evaluate the local thermal environment of an operational data center. The Index of Mixing (IOM) was selected as the primary metric for thermal evaluation. To closely mimic real-world conditions, a 2U server simulator featuring adjustable air volume and heating power was constructed. Experimental results indicated that the four 2U servers experienced superior cooling in the lower section of the target rack. Furthermore, operating without blind plates resulted in a 718% increase in the IOM of the target rack. Additionally, a CFD simulation methodology was developed to evaluate the rack-level thermal environment. The CFD model comprised three racks, designated hot and cold aisles, and an overhead space to simulate the mixing of hot and cold airflows. A flow resistance model was employed to approximate the servers as porous media, with resistance characteristics calibrated using the experimental data. Five representative server layout scenarios were simulated. The numerical results demonstrated strong consistency with experimental data regarding the rack inlet and outlet temperatures. These findings demonstrate that the proposed server simulator and CFD modeling approach serve as highly effective tools for assessing local thermal environments within data centers.

KEYWORDS: Data center; server simulator; experimental measurement; CFD; thermal environment

1 Introduction

Driven by the rapid advancement of information technology (IT) and its deep integration across various business sectors, older data centers face mounting challenges regarding security, cost, and energy management. Over the past two decades, data center power consumption has surged, notably doubling in the United States from 2013 to 2020 [1]. Furthermore, a survey of over 100 data centers revealed that more than half suffered from low operational efficiency in their cooling systems [2]. Rack power density has also escalated due to changes in server population, arrangement, and type [3]. To adapt to evolving business needs through the consolidation and optimization of IT resources, many data centers are currently undergoing significant transformations. Consequently, the global data center transformation market is expected to grow from USD 13.09 billion in 2025 to USD 21.49 billion by 2030 [4]. Presently, air cooling remains the most well-established and widely adopted thermal management solution in the industry [5]. When upgrading these air-cooled facilities, conducting a rigorous thermal environment assessment of the target area is essential to ensure that the cooling requirements of the deployed devices are met.

The majority of investigations into the thermal environments of data centers have been conducted in small-scale facilities or those with uniform power densities. For instance, Nada et al. [6,7] conducted computational fluid dynamics (CFD) investigations on a small data center comprising two rows of racks. They examined the influence of factors such as airflow rate, perforated tile opening ratio, and server power density on the facility's thermal environment [6]. Their findings indicated that a rack's thermal environment heavily depends on its location within the array, with middle racks generally exhibiting better cooling performance [7]. Similarly, Tatchell-Evans et al. [8] developed a single-rack experimental setup and demonstrated that significant bypass airflow (approximately 20% of the air supply) can occur through server racks under typical pressure differences between cold and hot aisles. Other researchers have explored optimizing the thermal environment by employing plenum deflectors [9] and cold aisle containment systems [10]. However, these studies primarily focus on idealized data centers with uniform power densities, assuming that all servers operate at full load. Only a limited number of studies have accounted for variations in rack heat dissipation rates. Meng et al. [11] investigated optimization methods, such as adjusting the opening ratio of perforated tiles, to address uneven temperature distributions and chaotic air supply. Song [12] found that incorporating fan-assisted perforations enhances cooling efficiency by promoting under-rack recirculation. Additional studies have evaluated the optimization effects of various airflow management systems, such as rack-level flexible baffles [13], server-level lower-side terminal baffles [14], and rack-level composite baffle diversion systems [15]. Yet, these studies were predominantly conducted in small experimental settings and focused on thermal optimization rather than assessment. In operational data centers, rack power fluctuates frequently due to dynamic service demands. Consequently, the actual thermal environment is profoundly affected by uneven rack power densities, rendering it highly complex. An experimental study in a live data center revealed that the maximum temperature difference within a cold aisle containment system can reach approximately 14°C [16]. When rearranging racks or adjusting server configurations in active facilities, quickly and accurately evaluating the local thermal environment of the target area remains a significant challenge. Therefore, developing robust thermal environment assessment methods for medium-to large-scale operational data centers is urgently needed.

During experimental assessments, environmental parameters such as temperature and humidity may deviate from safe operating ranges, potentially causing active servers to fail or shut down. Utilizing server simulators instead of actual IT equipment to evaluate data center thermal environments effectively mitigates these risks. Furthermore, server simulators provide the significant advantage of allowing researchers to precisely quantify and control the experimental parameters of interest. Several studies have successfully employed this approach. For instance, researchers at Benha University utilized a physically scaled data center model equipped with four server simulators per rack [17]. Their work primarily investigated how the rack-level thermal environment is influenced by the opening ratio of perforated tiles, air conditioner placement, server location and power density, and overall rack position. Similarly, a team at the Georgia Institute of Technology developed a 10U server simulator to optimize rack-level CFD models [18]. In their design, heat was generated by seven parallel, finned electric heating tubes and dissipated via forced convection using four internal fans [19]. Based on this hardware, they also introduced a methodology to measure the sensitivity of airflow rates to differential pressure across a server rack [20]. Additionally, researchers at National Chiao Tung University in Taiwan, China, constructed simulated server racks using electric heaters and cooling fans [21]. The front of their simulated rack featured an "air wall" composed of 45 fans. By setting up a small-scale experimental data center with two rows of these simulated racks and a dedicated air conditioning unit, they systematically studied the effects of various operational factors on the local thermal environment [22–24].

Table 1 summarizes the server simulators currently utilized in data center thermal environment research. A common limitation of these existing simulators is their oversimplification. Specifically, previous studies have primarily focused on the heating power of the servers while neglecting their airflow resistance characteristics [17,18,21]. Furthermore, these simulators generally lack operational flexibility. To date, relevant investigations have been confined to small-scale laboratory settings and have focused predominantly on the phenomenological analysis of experimental results. Consequently, a comprehensive methodological framework for thermal environment assessment has not yet been established, and the associated preparatory work remains highly time-consuming. This inefficiency fails to meet the practical demands of operational data centers, which require rapid thermal evaluations. To the best of the authors' knowledge, no prior studies have deployed server simulators to conduct research within active, medium- to large-scale data centers.

Table 1: Summary of the primary server simulators utilized in existing literature.

Source	Components	Key Characteristics
Nada et al. [17]	A variable-speed fan and a flat electric heater (150 W).	10U size (1/6 length-scale ratio); suitable for specialized, physically scaled data center models.
Nelson [19]	Seven parallel finned electric heating tubes and four cooling fans.	10U size; heating power is adjustable in 250 W increments, up to a maximum of 5750 W.
Wang et al. [21]	Twelve U-shaped electric heating tubes per heating module.	Each rack comprises five integrated heating modules that cannot be disassembled.

CFD simulation is a widely used method for studying the thermal environments of data centers. Currently, the majority of research in this area focuses on room-level models [3]. Numerous studies have employed CFD to analyze various factors influencing data center thermal environments [6,7,9,10,25,26]. Regarding numerical model optimization, researchers have sought to improve both perforated tile models [27,28] and turbulence models [29,30]. For general simulation purposes, the standard k- ϵ turbulence model paired with standard wall functions is typically sufficient [3]. At the rack level, the validity of representing IT equipment using simplified black-box models has been widely discussed [31,32]. Furthermore, several transient CFD models for data center thermal simulations have been introduced. For instance, Choi et al. [33] developed ThermoStat, a CFD-based tool for analyzing temperatures in rack-mounted servers. Similarly, Ibrahim et al. [34] proposed a methodology to establish server temperature response curves for transient CFD simulations based on empirical data. Despite these advancements, modeling operational air-cooled data centers presents significant challenges due to highly complex rack power density distributions and airflow configurations. Establishing a comprehensive, facility-wide CFD model to evaluate a specific target area is prohibitively difficult and time-consuming. Such approaches fail to meet the practical demands of rapid facility upgrades, and the accuracy of the resulting simulations is often difficult to guarantee. Therefore, there is a critical need to develop local CFD simulation methodologies to guide targeted rack-level transformations in active data centers. To date, however, research addressing this specific requirement remains sparse.

To address the aforementioned gaps in the literature, this study experimentally and numerically investigates the local thermal environment of an operational data center. A 2U server simulator, featuring adjustable airflow and heating power, was employed for the experimental phase. The Index of Mixing (IOM)

was selected as the primary metric for evaluating the local thermal environment. By applying this index to the experimental data, the specific impacts of server placement and blind plates were quantitatively analyzed. Finally, building upon these empirical findings, a targeted CFD simulation methodology for evaluating the rack-level thermal environment of air-cooled data centers is presented and demonstrated.

2 Methodology

2.1 Data Center Description

The data center investigated in this study is located in Hangzhou, Zhejiang Province, China. It is a traditional air-cooled facility featuring a raised floor system. The data center spans an area of approximately 650 m², with a ceiling height of 3.6 m and a sub-floor plenum height of 0.7 m. Fig. 1 presents a top-down view of the facility's layout. The room houses eight rows of racks (designated A through H), which are arranged to form four cold aisles and five hot aisles without physical containment. Fifteen computer room air conditioning (CRAC) units, numbered 01 through 15, are positioned at the front and rear of the rack rows. Gray squares in the schematic denote individual rack positions. The experimental measurements were conducted within a specific test area situated centrally in row F.

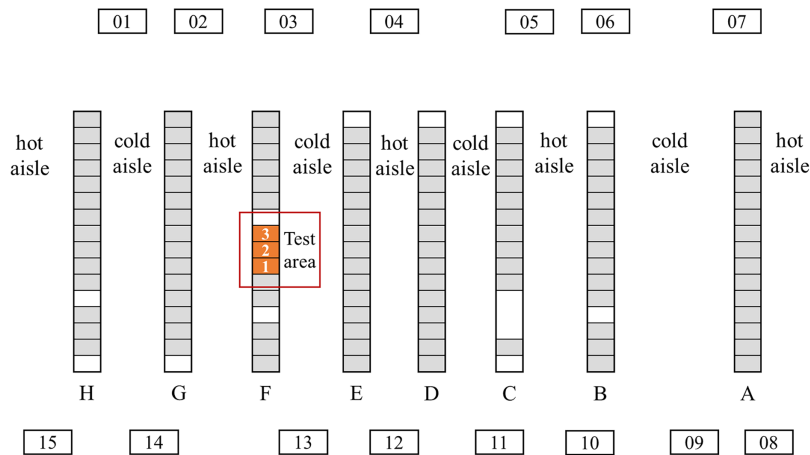


Figure 1: Layout of the data center.

The specific layout of the test area is illustrated in Fig. 2. The area comprises three standard 42U racks, numbered 1 through 3 from left to right. Rack 2 served as the primary test rack, while the adjacent racks (Racks 1 and 3) were kept empty and fully fitted with blind plates. In the adjacent rack immediately to the left of Rack 1, the lower section was populated with active servers. A 0.6-m-wide pedestrian aisle is located to the right of Rack 3. The supply air outlets consist of single-layer louvered grilles measuring 0.6 m × 0.6 m. The effective louvered area of each grille is 0.52 m × 0.3 m, with an airflow discharge angle of 60°.

2.2 Experimental Study

Server simulators can serve as effective substitutes for active IT equipment when assessing the thermal environments of data centers. In this study, by referencing the internal architecture of standard 2U rack servers and applying appropriate physical simplifications, a novel 2U server simulator was designed to closely mimic real-world hardware. The overall dimensions of the simulator are 550 mm × 435 mm × 88.9 mm. A top-down view detailing the internal composition of the simulator is presented in Fig. 3.

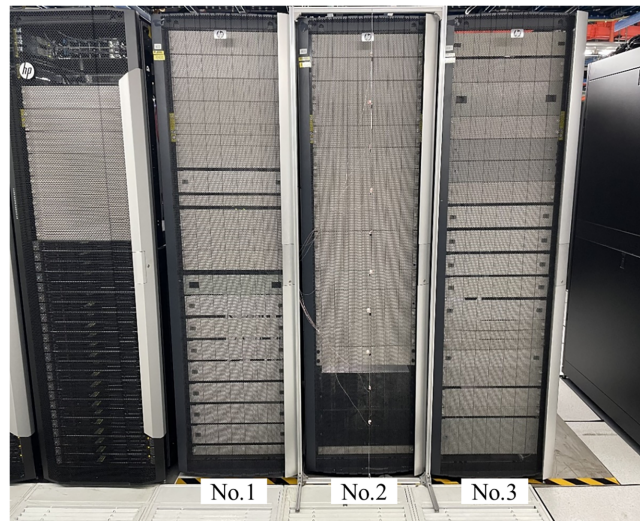


Figure 2: View of the test area.

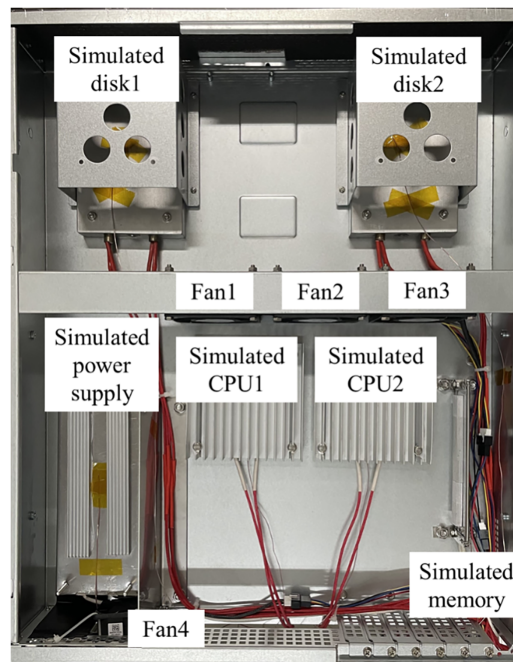


Figure 3: Composition of the 2U server simulator.

Each server simulator utilizes four variable-speed fans and six electric heating elements to accurately replicate the thermal behavior of an actual server. Specifically, the cooling system comprises three Delta PFB0812UHE fans and one smaller AVC DB04028B fan dedicated to the simulated power supply. As detailed in [Table 2](#), the specifications of the heating elements were determined based on the physical dimensions and power consumption of primary components found in standard servers [34,35]. With the exception of the simulated disk, which employs a cast aluminum heater, all other heating elements consist of ultra-thin mica plate heaters. To accommodate the high thermal output of the simulated CPU, its footprint was slightly enlarged relative to a standard physical CPU to ensure adequate heat dissipation, and corresponding

aluminum heat sinks were installed. The total maximum heating power of each simulator is 675 W. Furthermore, the heat output of all elements can be continuously adjusted using a variable regulated voltage power supply.

Table 2: Specifications of heating elements in the 2U server simulator.

Component	Dimensions	Power per Unit	Quantity
Simulated CPU	100 mm × 100 mm	150 W	2
Simulated disk	150 mm × 100 mm × 20 mm	125 W	2
Simulated memory	150 mm × 40 mm	45 W	1
Simulated power supply	220 mm × 80 mm	80 W	1

The experimental measurements were designed to accurately characterize the local thermal environment within the operational data center. Specifically, the following six parameters were measured: rack inlet and outlet air temperatures, surface temperatures of the simulated heating elements, overall surface temperature distributions of the server simulators, and the temperatures and airflow rates at the supply air outlets. As illustrated in Fig. 4, measurement points for the rack were established using two frames positioned 5 cm from the front and rear of the rack, in accordance with ASHRAE TC9.9 guidelines. Ten and thirty well-calibrated T-type thermocouples were evenly distributed across the inlet and outlet measurement frames, respectively. Additionally, a well-calibrated T-type thermocouple was affixed to the center of each heating element within the server simulator to monitor local surface temperatures. Temperature data were recorded at 5 s intervals, with the average values over the final 60 s adopted as the steady-state temperatures for each measurement point. Furthermore, the surface temperature distributions of the server simulators were captured using an infrared thermal imager. A hot-wire anemometer was employed to measure the temperatures and airflow rates at nine distinct points across each supply air outlet, with the final results calculated as the average of these nine readings. Finally, Table 3 details the technical specifications of the measurement instruments utilized in this study.

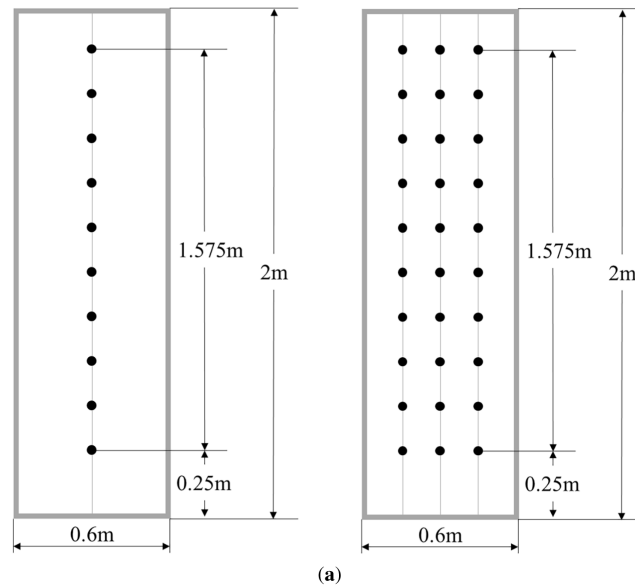


Figure 4: (Continued)



Figure 4: Schematic design and sensor layout of the rack temperature measurement frames. (a) Schematic diagram of the temperature measurement point layout for the server rack inlet and outlet; (b) Photograph of the physical measurement frames with actual thermocouple installation.

Table 3: Technical specifications of the primary measurement instruments.

Instrument	Measured Parameter(s)	Measurement Range	Accuracy
Testo 405i	Air velocity Air temperature	0 to 30 m/s 0°C to 50°C	$\pm 0.1 \text{ m/s} \pm 5\%$ (0 to 2 m/s)
Fluke TiX660	Surface temperature (Infrared imagery)	−40°C to 1200°C	$\pm 1.5 \text{ K}$ or $\pm 1.5\%$

To investigate the influence of server layout and the presence of blind plates on the rack thermal environment under partial-load conditions, ten distinct experimental cases were studied. These cases encompassed various simulator arrangements, both with and without blind plates installed. As illustrated in Fig. 5, the four server simulators were designated 1 through 4, ordered from top to bottom.

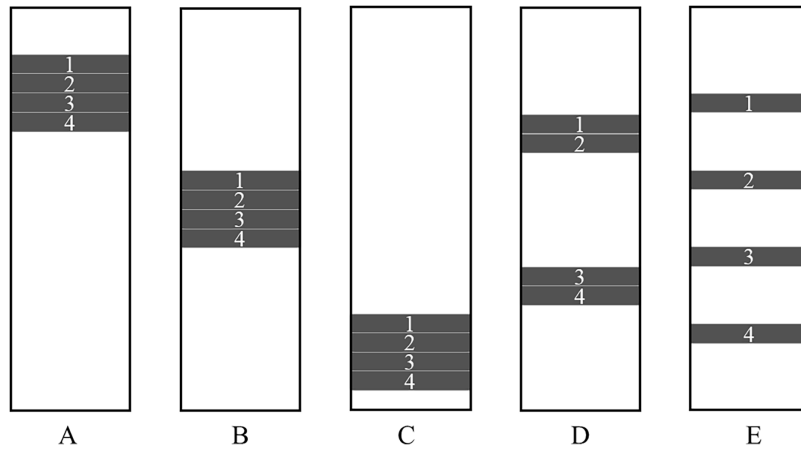


Figure 5: Schematic diagram of server simulator layouts. (A) Fully concentrated layout of all four servers at the rack's uppermost section; (B) Fully concentrated layout of all four servers at the rack's middle section; (C) Fully concentrated layout of all four servers at the rack's bottommost section; (D) Grouped dispersed layout with two server clusters at the upper-middle and lower-middle sections; (E) Fully dispersed layout with individual servers separated by unoccupied slots along the rack height.

Various indices exist for quantitatively assessing the airflow performance and thermal environments of data centers. The most commonly used evaluation metrics include the Supply Heat Index (SHI), Return Heat Index (RHI), Rack Cooling Index (RCI), and Return Temperature Index (RTI). However, these conventional indicators primarily rely on the average temperature of the rack. Consequently, they can mask local airflow temperature variations; an artificially low weighted-average temperature might fail to accurately reflect the true thermal distribution and the presence of local hot spots [3]. Therefore, these traditional indices are inadequate for the rack-level thermal environment assessment conducted in this study. Instead, IOM was selected to quantitatively characterize the degree of airflow mixing and the temperature distribution within the rack. A lower IOM value indicates a superior rack thermal environment, whereas a higher value signifies an overheated space and an increased probability of potential hot spots. Notably, an IOM value exceeding 1 indicates the occurrence of exhaust air recirculation within the rack [36]. Recently, this evaluation index has been successfully applied in several studies investigating data center thermal environments [37,38]. The mathematical definition of the IOM is expressed as follows:

$$\text{IOM} = \frac{T_{in_max} - T_{in_min}}{T_{out} - T_{in}} \quad (1)$$

In the equation, T_{in_max} is the maximum rack inlet temperature; T_{in_min} is the minimum rack inlet temperature; T_{out} is the average rack outlet temperature; T_{in} is the average rack inlet temperature.

2.3 Numerical Simulation

In this study, ANSYS Icepak was employed to evaluate the rack-level thermal environment of the data center. Based on the experimental configuration, a local CFD model was developed, as depicted in Fig. 6. Previous research has demonstrated that a three-rack model, with the target rack positioned centrally, yields greater accuracy than a single-rack model [39]. Consequently, the computational domain for this study encompassed the target rack and its two adjacent neighbors. Hot and cold aisles, each 0.6 m wide, were established at the front and rear of the racks, with free openings designated to represent the hot air exhaust boundaries. To account for the uncontained cold and hot aisles, a 0.15 m overhead clearance was incorporated

above the racks to simulate the mixing of hot and cold airflows accurately. The overall dimensions of the computational domain were $2.2 \text{ m} \times 1.8 \text{ m} \times 2.15 \text{ m}$.

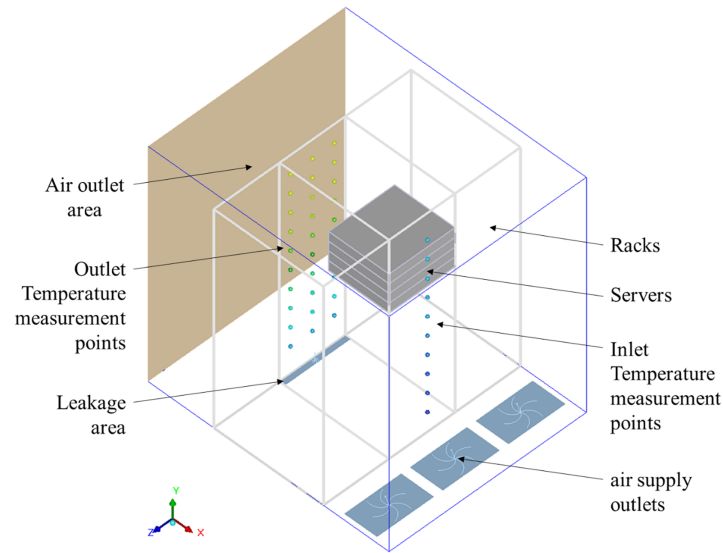


Figure 6: Computational domain and layout of the local CFD model.

The racks were standard 42U units, represented in the simulation using the enclosure model, with the front and rear doors modeled as grilles. For rack-level numerical simulations, simplifying the servers using a black-box approach is widely considered appropriate [3]. Consequently, a flow resistance model was adopted to approximate the servers as porous media, with specific resistance characteristics calibrated according to the experimental data. The physical thickness of the supply air outlets was neglected; instead, they were represented using a two-dimensional axial flow fan model measuring $0.52 \text{ m} \times 0.3 \text{ m}$, corresponding to the effective louver area of the actual supply outlets. Additionally, a 0.05 m wide floor gap located behind the target rack which permitted minor cold air leakage in the physical data center-was explicitly accounted for in the model. Based on empirical results from the test area and target rack, the parameter settings for the boundary conditions are summarized in Table 4. Because the operating conditions of the CRAC units and the sub-floor plenum airflow fluctuated naturally, the supply air temperature varied across different experimental cases. Therefore, the supply air temperature for each CFD simulation was specifically set to match the recorded experimental data of its corresponding case.

Table 4: Parameter settings for the CFD model boundary conditions.

Boundary Condition	Unit	Value
Supply air flow rate	m^3/s	0.345
Leakage airflow rate	m^3/s	0.06
Leakage air temperature	$^{\circ}\text{C}$	17
Server heat load	W	675
Ambient temperature	$^{\circ}\text{C}$	25
Front door opening ratio	–	0.02
Rear door opening ratio	–	0.9

To ensure computational efficiency while maintaining accuracy, the following simplifying assumptions were applied to the CFD numerical model: (1) the airflow was characterized as low-speed and treated as an incompressible viscous fluid; (2) the air satisfied the Boussinesq approximation; (3) the airflow was fully developed and modeled as steady-state turbulence; (4) the volumetric flow rate and temperature at all supply air outlets were defined as fixed constant values; (5) the physical interference of the rack temperature measurement frames and other experimental apparatus was neglected; (6) minor fluctuations in server power consumption were ignored; (7) radiative heat transfer from the building envelope and object surfaces was not considered; and (8) external heat transfer between cabinets, as well as the ambient heat loads from facility personnel and lighting, were omitted.

The local CFD numerical simulations utilized the standard $k-\epsilon$ turbulence model, which is widely recognized as sufficient for general data center calculations [3]. The governing equations were solved using the finite volume method. Specifically, the PRESTO scheme was employed to discretize the pressure term, as it is highly suitable for models incorporating porous media, perforated plates, and fans. All other terms were discretized utilizing a first-order upwind scheme. The convergence criteria were established as follows: residuals for continuity and the X, Y, and Z velocity components were required to fall below 10^{-3} , while the energy residual criterion was strictly set to 10^{-6} .

Given the regular geometry of the computational domain, a structured mesh was generated to ensure high grid quality. The mesh was locally refined around critical regions, specifically the supply air outlets and the servers. To verify grid independence, the maximum, minimum, and mean inlet and outlet air temperatures of the target rack were monitored across varying mesh densities, as shown in Fig. 7. The calculated temperatures achieved stability when the total number of grid elements exceeded 599,800. Consequently, this specific grid generation scheme was adopted for all subsequent numerical simulations. Finally, five experimental cases featuring blind plates were simulated, and the numerical results were systematically compared against the empirical data to validate the feasibility and accuracy of the local CFD model.

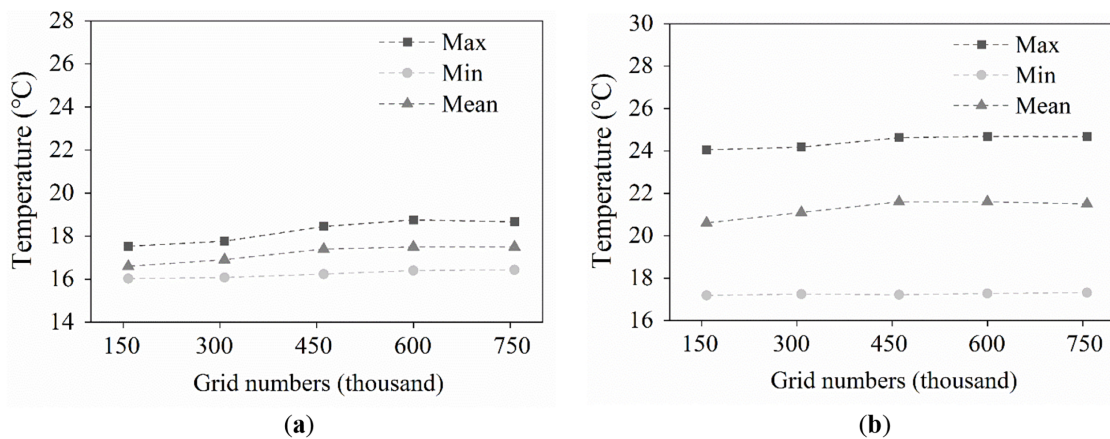


Figure 7: Rack temperature variations with grid number. (a) inlet air temperature; (b) outlet air temperature.

3 Results and Discussion

3.1 Effect of Server Layout and Blind Plates

Fig. 8a illustrates the rack inlet air temperatures across the different server layout cases. For all configurations utilizing blind plates, the inlet air temperature exhibited a gradual decrease from top to bottom. Notably, all measured inlet temperatures fell well within the 18°C to 27°C acceptable range stipulated by the Code for Design of Data Centers in China (GB50174-2017). Furthermore, because the maximum

recorded inlet temperature did not exceed 20°C, there is considerable margin to appropriately raise the supply air temperature, thereby reducing overall cooling energy consumption.

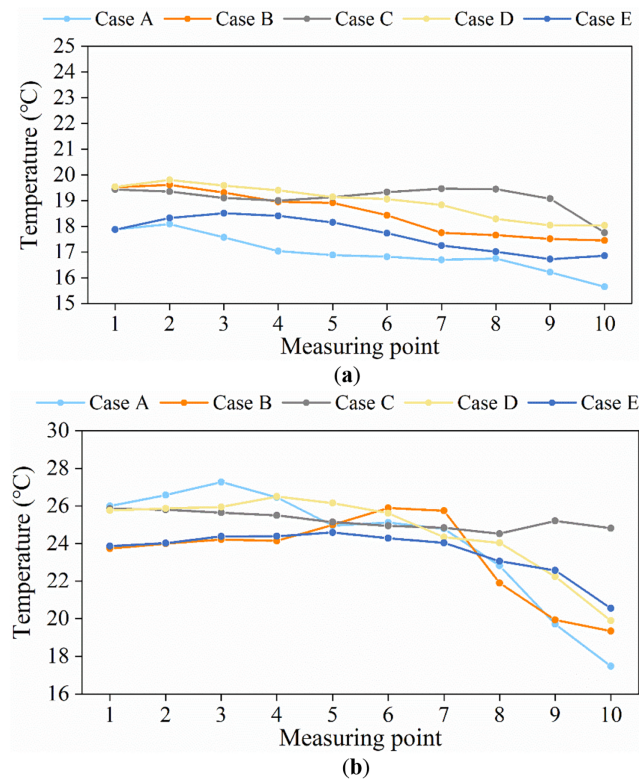


Figure 8: Rack air temperatures across the various server layout cases. (a) inlet air temperature; (b) outlet air temperature.

Fig. 8b presents the corresponding rack outlet air temperatures, calculated as the row-averaged values from the outlet measurement frame. The results indicate that the outlet thermal profile is primarily governed by two factors: the local inlet air temperature and the specific server layout. Similar to the inlet profile, the overall outlet temperature generally decreased from top to bottom; however, local temperature peaks clearly corresponded with the physical elevations of the active server simulators. In the centralized layout scenarios (Cases A, B, and C), these temperature spikes in the exhaust regions were particularly pronounced. Conversely, in Case E, where the servers were dispersed, the exhausted hot air mixed more extensively before exiting the rack. This enhanced mixing resulted in a much more gradual vertical temperature gradient, yielding a relatively uniform temperature distribution.

However, relying solely on inlet and outlet temperature profiles provides an incomplete assessment of the layout schemes. Therefore, IOM was utilized to rigorously evaluate the rack-level thermal environment. The calculated IOM values for each layout case are presented in Fig. 9. Across all cases, the IOM values remained below 1.0; this indicates that while some mixing of hot and cold air occurred within the target rack zone, detrimental hot air recirculation was largely absent [36]. Notably, Case C achieved the lowest IOM value of 0.28. From the perspective of airflow distribution and thermal management, this demonstrates that the four 675 W, 2U servers experienced optimal cooling when positioned in the lower section of the target rack. In contrast, Case B yielded an IOM of 0.44, significantly higher than the other configurations, which corresponds to an elevated risk of local hot spots and thermal insecurity. Consequently, while placing servers

in the middle of the rack might intuitively seem like a reasonable standard practice, the IOM analysis proves it is thermally unsuitable for this specific combination of server quantity and power density.

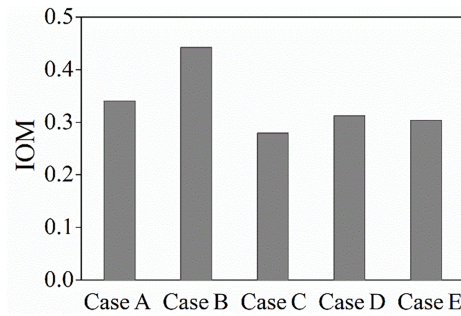


Figure 9: IOM values for the different server layout cases.

Fig. 10 presents the surface temperatures of the primary heating elements within each server simulator across the various layout cases. It should be noted that the reported CPU temperatures represent the average surface temperature of the two simulated CPUs housed within each simulator. Although minor temperature variations were observed among identical elements across different simulators, likely due to inherent experimental uncertainties and manufacturing tolerances, these discrepancies were deemed acceptable, as this study focuses on relative, rather than absolute, temperature comparisons between layout cases. Among the components, the simulated CPUs exhibited the highest surface temperatures, consistently exceeding 100°C and peaking at 120°C . Conversely, the simulated power supplies recorded the lowest temperatures, remaining entirely below 100°C . Ultimately, these results indicate that the macroscopic, rack-level server layout exerts a negligible influence on the local surface temperatures of the internal heating elements.

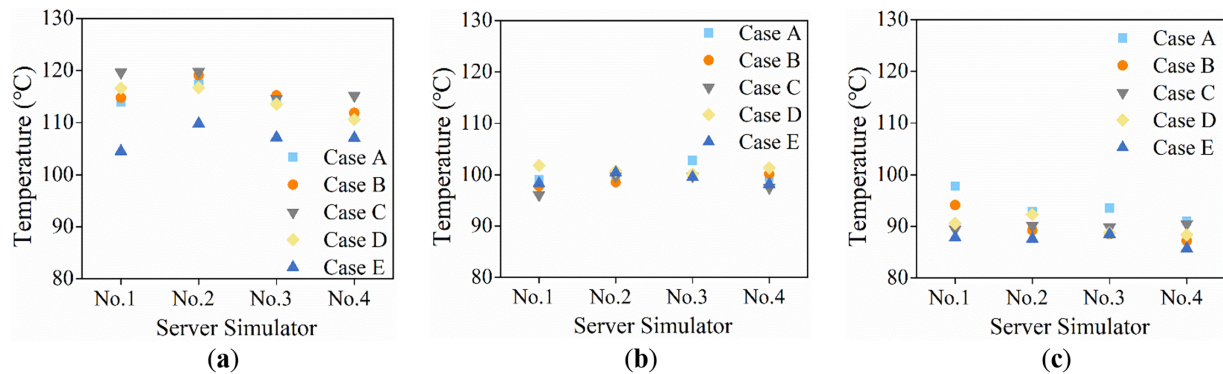


Figure 10: Surface temperature of heating element in different server layout cases. (a) simulated CPU; (b) simulated memory; (c) simulated power supply.

The macroscopic surface temperature distributions of the server simulators were captured using an infrared thermal imager. Observations from the cold aisle revealed that the infrared thermal profiles were highly consistent across all server layout configurations. A representative infrared thermogram illustrating the server simulator layout area at the front of the target rack is presented in Fig. 11. Generally, the surface temperatures at the air inlets of the server simulators remained notably low, whereas the areas immediately adjacent to the simulated disks exhibited the highest temperatures.

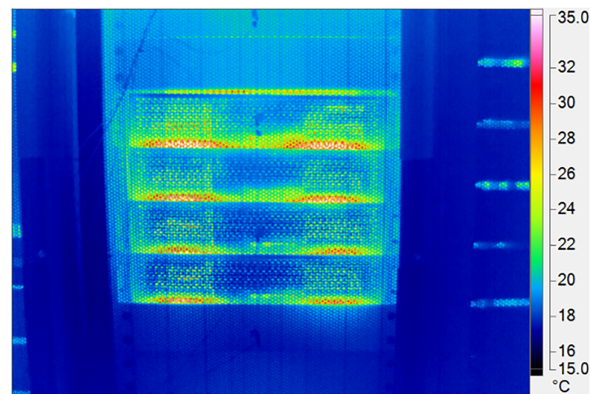


Figure 11: Infrared thermograms of the server simulator layout area at the front of the rack.

Fig. 12 presents the infrared thermograms of the server layout area at the rear of the rack. Generally, server simulators positioned higher within the rack exhibited correspondingly higher surface temperatures. This thermal stratification phenomenon is particularly evident in the thermograms for Cases D and E, which utilize distributed server configurations. Consequently, from the perspective of the rack's local thermal environment, as visualized by the infrared imaging, Case C emerges as the optimal layout. This visual evidence strongly corroborates the earlier conclusions drawn from the IOM analysis and the inlet and outlet temperature measurements regarding airflow mixing efficiency.

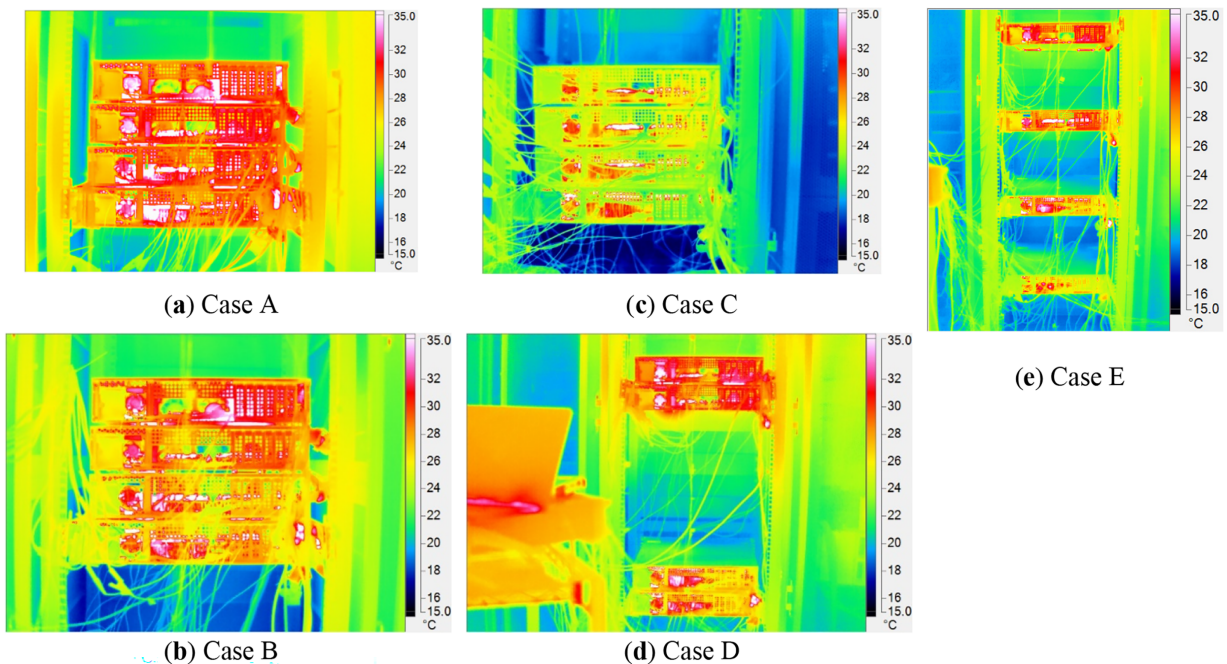


Figure 12: Infrared thermogram of the server simulator layout area at the rear of the rack. (a) Case A: Infrared thermogram of top-concentrated server layout; (b) Case B: Infrared thermogram of middle-concentrated server layout; (c) Case C: Infrared thermogram of bottom-concentrated server layout; (d) Case D: Infrared thermogram of grouped dispersed server layout; (e) Case E: Infrared thermogram of fully dispersed server layout.

Fig. 13a illustrates the inlet air temperatures of the target rack both with and without blind plates. The results indicate that blind plates exerted a particularly significant effect on inlet temperatures when the server

simulators were densely grouped. At identical measurement points, temperatures without blind plates could be up to 29% higher than those with panels installed. In Cases B and C, the absence of blind plates led to significantly elevated inlet air temperatures in the upper region of the server layout area. Under normal operating conditions, hot exhaust air flows out the rear door of the rack, driven by pressure differentials. However, if blind plates are omitted, this lower-density hot air rises to the upper sections of the rack or recirculates to the front, mixing with the cold supply air. Consequently, the inlet air temperature above the server layout area increased dramatically.

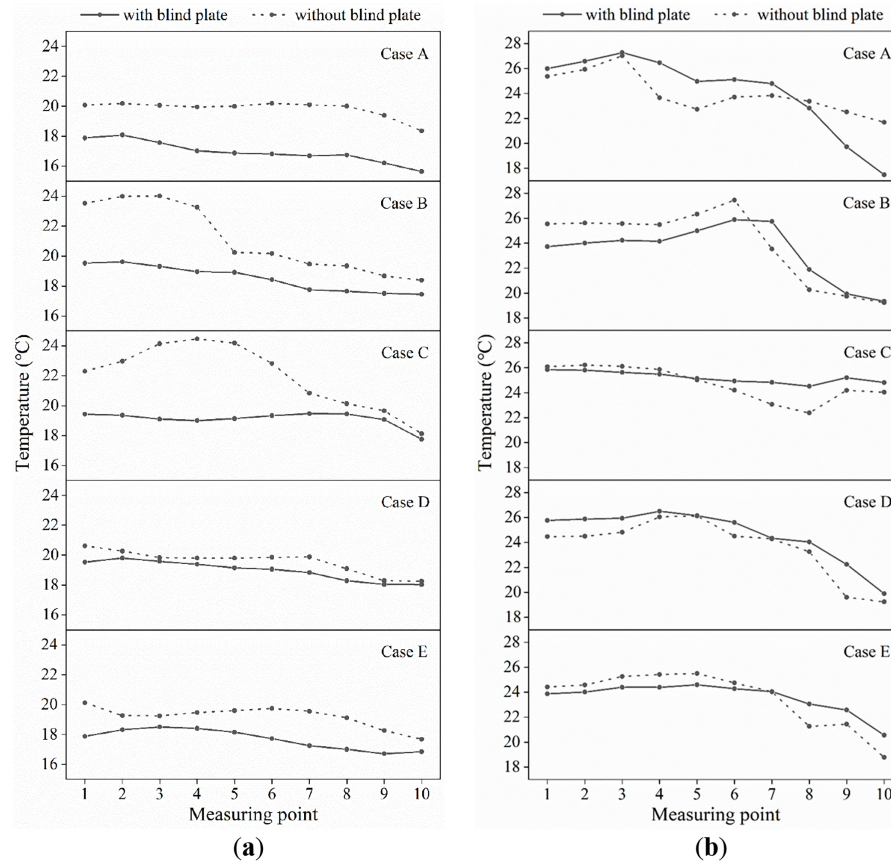


Figure 13: Air temperatures of the target rack with and without blind plates. (a) inlet air temperature; (b) outlet air temperature.

Fig. 13b presents the corresponding outlet air temperatures. Similar to the inlet profiles, the outlet temperatures with blind plates installed were generally lower and more uniformly distributed across all server layout cases. Without blind plates, the outlet temperatures at the bottom of the rack were typically low, a phenomenon primarily caused by the short-circuiting of cold supply air. Case A, however, presents a notable exception. As shown in Fig. 13b, in this unsealed configuration, the bottom outlet temperatures were actually elevated. This occurred because all server simulators in Case A were concentrated in the upper portion of the rack, leaving a massive unsealed void in the middle and lower sections. In this area, the severity of hot air recirculation completely overshadowed the effects of cold air short-circuiting.

Finally, Fig. 14 demonstrates the influence of blind plates on the IOM of the target rack. Removing the panels caused the IOM to surge by 31%, 358%, 718%, 84%, and 85% across the five respective layout cases. For Cases B and C without blind plates, the IOM values exceeded 2.0, indicating severe hot air recirculation and

a dangerous degree of hot and cold airflow mixing [36]. Among all configurations, the thermal environment in Case C proved most reliant on blind plates; upon their removal, the rack's thermal conditions rapidly deteriorated from a state of minor mixing into a critical exhaust self-recirculation failure.

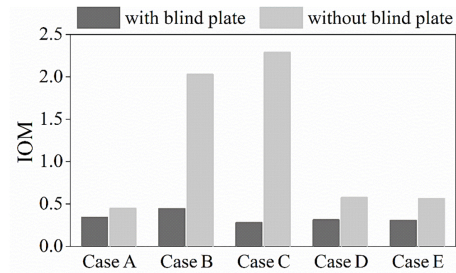


Figure 14: IOM in cases with or without blind plates.

Building upon the previous discussion regarding temperature variations among individual heating elements, Fig. 15 presents the overall average surface temperatures for each type of component across all simulators for every layout case. The data clearly indicates that component surface temperatures were consistently elevated in all configurations lacking blind plates. Consequently, the convective cooling efficacy of the supply airflow was significantly degraded when the blind plates were omitted. This finding perfectly aligns with the preceding analyses of the rack inlet and outlet air temperatures, as well as the calculated IOM values, further validating the critical role of physical containment in optimal thermal management.

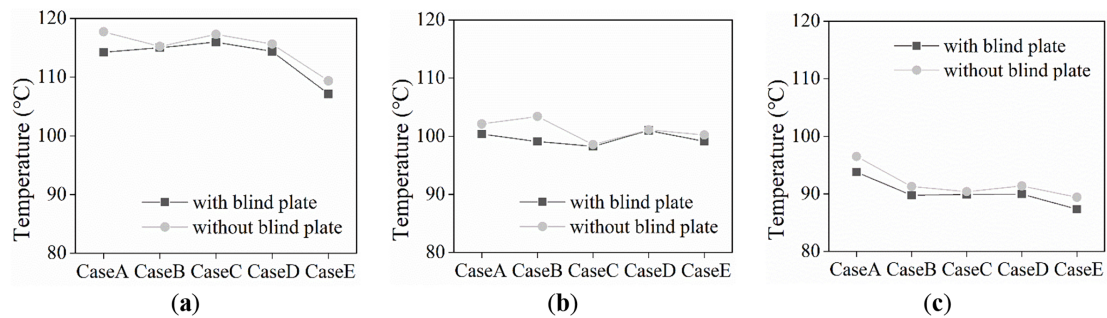


Figure 15: Surface temperature of heating element in cases with or without blind plates. (a) simulated CPU; (b) simulated memory; (c) simulated power supply.

Unlike previous server simulators that either cannot be disassembled from a rack [21] or feature dimensions incompatible with standard commercial racks [17,19], the 2U server simulators utilized in this experiment can be mounted directly into existing target racks. Combined with precision measuring devices, this allows for rapid evaluation of the rack's thermal environment and the determination of optimal layout configurations for specific server quantities and power densities. Furthermore, whereas earlier simulators featured internal structures vastly different from actual hardware, this novel 2U simulator was designed to closely mimic standard 2U rack servers. Consequently, its aerodynamic resistance closely reflects that of real servers. Previous research has demonstrated that a server's airflow impedance severely impacts rack air distribution and the overall thermal environment, often causing massive discrepancies between simulator experiments and real-world conditions [40]. Additionally, the simulators developed in this study enable real-time temperature monitoring of internal heating elements. This capability not only captures subtle thermal variations between different servers but also allows researchers to actively monitor component temperatures to ensure operational safety during experiments.

Racks operating at relatively low load ratios are highly common, particularly in large-scale data centers [41]. This study marks the first time server simulators have been deployed in an active, operational data center to investigate the thermal environment of a partially loaded rack. The findings demonstrate that server layout significantly impacts the rack's thermal profile. Specifically, the four 675 W, 2U servers achieved optimal cooling when local in the lower section of the target rack. Under this configuration, the rack yielded the minimum IOM value, indicating reduced airflow mixing and a superior overall thermal environment.

The presence of blind plates also exerts a profound impact on rack thermal environments. In operational data centers, the critical role of blind plates is frequently overlooked; they are often not reinstalled promptly following rack modifications, which rapidly degrades the local thermal environment. For racks housing only a small number of servers, installing blind plates is absolutely essential. These panels physically prevent the short-circuiting of cold supply air and the recirculation of hot exhaust air, thereby maintaining the target rack's inlet air temperature at an optimally low level.

3.2 Rack Thermal Environment Simulation

Compared to experimental measurements, numerical simulations offer distinct advantages, including high flexibility and cost efficiency. In actual air-cooled data centers, thermal environments are often highly complex due to uneven rack power distributions [16]. Establishing a local CFD model to evaluate the target thermal environment is a highly efficient, labor-saving approach; however, few researchers have thoroughly explored this method. While Fulpagare et al. [39] developed a three-rack transient CFD model to simulate how rack outlet temperatures shift with varying thermal loads, their model is not optimized for steady-state conditions. Furthermore, while their simulation trends aligned with experimental data, notable numerical discrepancies persisted between the simulated and measured values.

To address these gaps, this paper presents a numerical simulation methodology specifically designed to evaluate the local thermal environment of operational data centers. A local CFD model is proposed for assessing rack-level thermal dynamics under steady-state conditions. This model can accurately simulate local thermal profiles within a real data center, evaluate the viability of proposed server layouts, and compare the efficacy of various optimization strategies. To demonstrate its validity, this local CFD methodology was used to simulate the target rack across five server layout cases. The resulting inlet and outlet air temperature contour plots for the target rack are presented in Fig. 16, providing a highly vivid, visual representation of the thermal distribution.

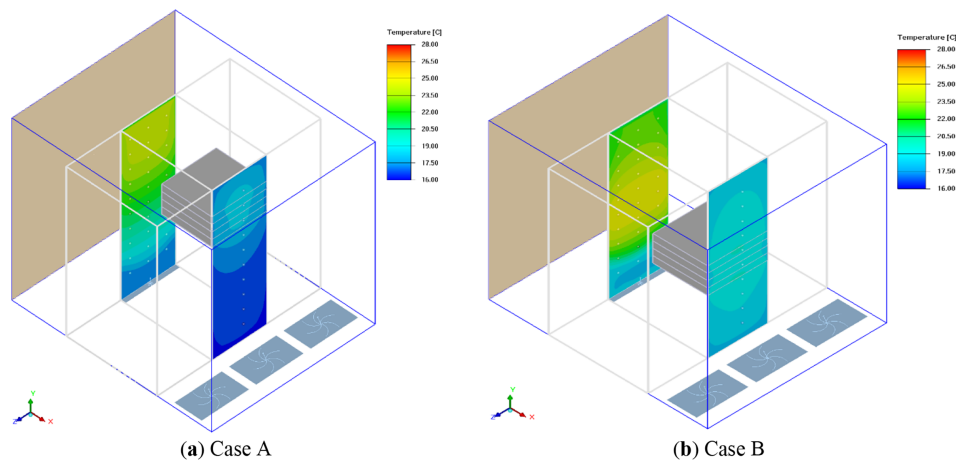


Figure 16: (Continued)

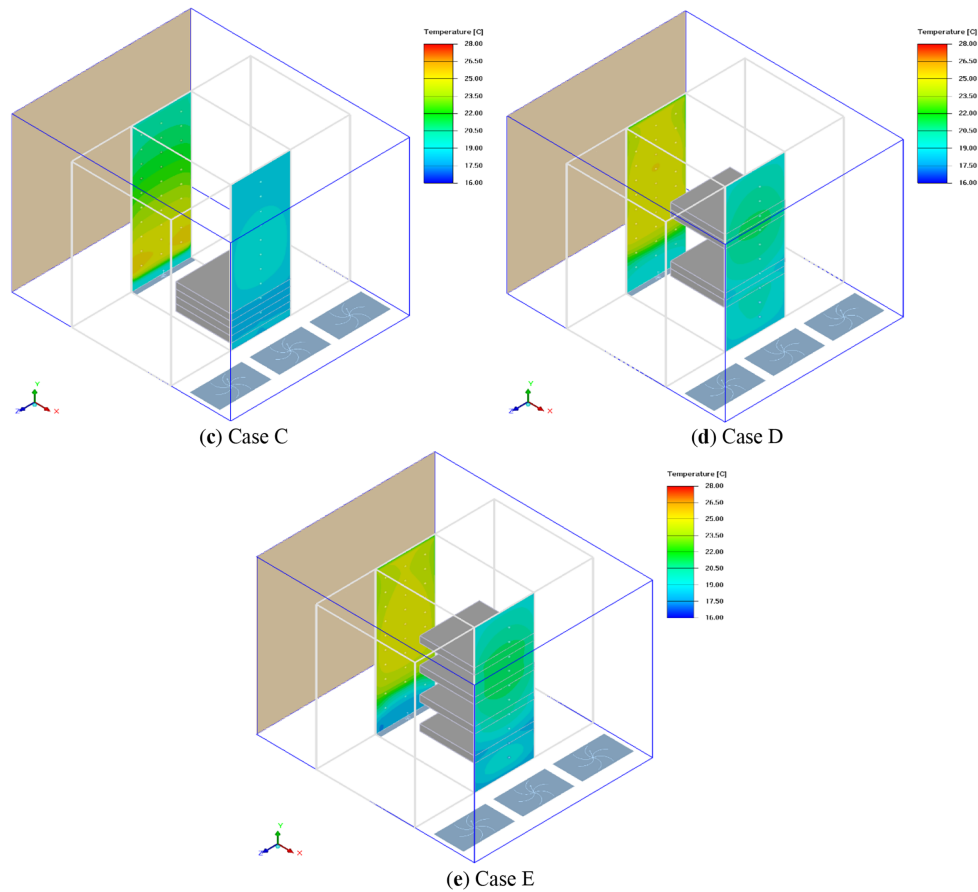


Figure 16: Inlet and outlet air temperature distribution of the target rack. (a) Case A: Rack inlet and outlet air temperatures of top-concentrated server layout; (b) Case B: Rack inlet and outlet air temperatures of middle-concentrated server layout; (c) Case C: Rack inlet and outlet air temperatures of bottom-concentrated server layout; (d) Case D: Rack inlet and outlet air temperatures of grouped dispersed server layout; (e) Case E: Rack inlet and outlet air temperatures of fully dispersed server layout.

In Cases A and B, the rack's inlet air temperature remained uniform, exhibiting only slight elevations in the immediate vicinity of the servers. Due to the buoyancy of the lower-density hot exhaust, the zone of maximum outlet air temperature was physically higher than the actual server locations. In Case C, where servers were concentrated at the bottom, a portion of the cold supply air entered the servers directly. Meanwhile, other regions on the rack's intake side experienced moderate mixing of hot and cold airflows, keeping the local inlet temperatures at the servers relatively low. Conversely, the inlet temperatures for Cases D and E were evenly distributed but slightly elevated overall, averaging 19.61°C and 19.59°C , respectively. Compared to the densely packed server configurations, the outlet air temperature distributions in these distributed layouts (Cases D and E) were significantly more uniform. Additionally, the lower sections of the rack exhibited cooler outlet temperatures due to cold air leaking from the floor gap. Ultimately, these simulated inlet and outlet temperature distributions were highly consistent with the empirical phenomena observed during the physical experiments.

3.3 Local CFD Model Validation

To further validate the feasibility of the local model, the numerical results for the inlet and outlet air temperatures were compared against the experimental data (Fig. 17). For Case A, the simulated and experimental inlet temperatures exhibited excellent agreement in both magnitude and variation trend, with a maximum deviation of only 3.31%. The outlet air temperatures also demonstrated similar overall trajectories; however, the numerical predictions were slightly lower than the experimental measurements, yielding a maximum deviation of 14.49%. This discrepancy can be attributed to the surrounding data center environment: while the target rack operated at a relatively low power density, adjacent high-power racks continuously exhausted high-temperature air. This external hot air mixed within the uncontained hot aisle, artificially elevating the temperature at the rear of the target rack. Because the local CFD model boundary conditions do not account for the thermal influence of these neighboring racks, the simulated outlet temperatures were predictably lower.

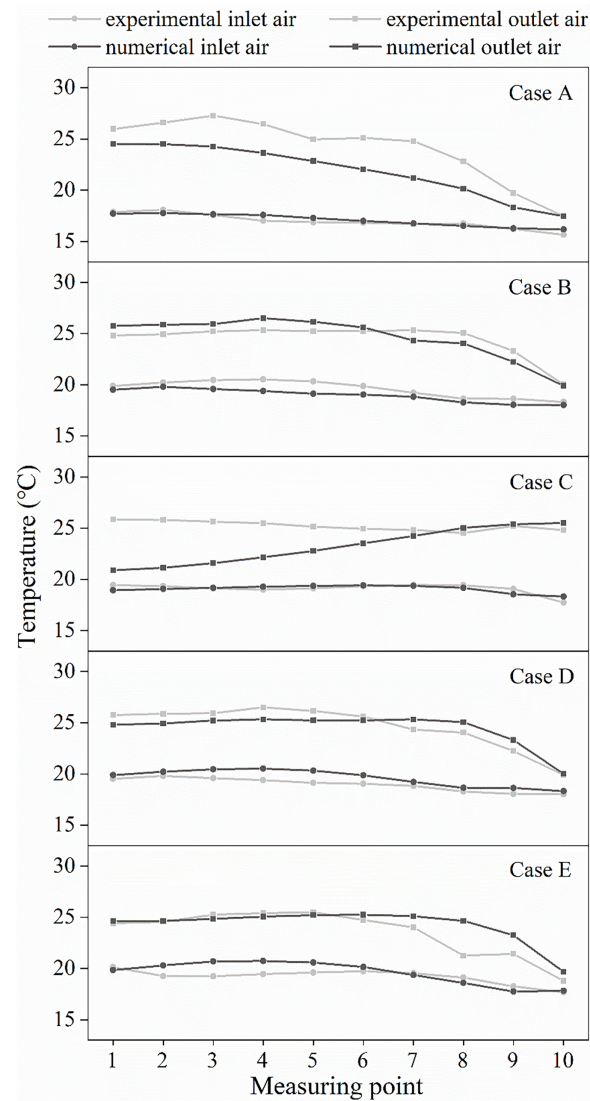


Figure 17: Comparison of numerical and experimental temperature results.

In Case B, the simulated and experimental temperature profiles closely aligned, recording maximum deviations of 6.91% and 8.61% for the inlet and outlet, respectively. Notably, the experimental outlet temperatures exhibited a rapid decline at measurement points 8, 9, and 10 due to cold air leaking from the floor gap. While the numerical results captured this local cooling trend, the magnitude of the temperature drop differed slightly. This variation likely occurred because the physical internal structures of the actual rack partially obstructed and redirected the leakage airflow, creating a complex temperature distribution that the simplified CFD geometry did not perfectly replicate.

For Case C, the inlet air temperatures matched closely, with a maximum deviation of just 3.91%. Conversely, the simulated variation trend for the outlet air temperature was fundamentally opposite to the experimental data, resulting in a maximum deviation of 19.18%. Closer inspection revealed that while the numerical and experimental outlet temperatures at the bottom of the rack matched well, massive discrepancies occurred in the empty upper positions. Similar to Case A, this error stems from the local CFD model's inability to capture thermal intrusion from adjacent high-power racks. Consequently, this study concludes that a highly local CFD boundary model is unsuitable for assessing racks that feature a small number of centralized, low-power servers but are physically surrounded by high-power equipment.

Finally, Cases D and E demonstrated strong agreement between the simulations and experiments. In Case D, the maximum deviations for the inlet and outlet air temperatures were merely 6.18% and 4.80%, respectively. In Case E, the maximum inlet deviation was 7.59%. It is important to note an anomaly in the Case E outlet data at measurement point No. 8, which recorded a sudden, uncharacteristic temperature drop compared to adjacent sensors. This local drop contradicted the broader thermodynamic trends, suggesting the physical thermocouple was likely disturbed during the experiment and failed to reflect the true airflow conditions. After excluding this anomalous data point, the adjusted maximum deviation for the Case E outlet temperature was calculated at 8.51%, which falls well within an acceptable margin of error for CFD validation.

In summary, the experimental and numerical results for the local CFD model demonstrated strong overall consistency. With the notable exception of the outlet temperatures in Case C, the deviations between the simulated and experimental results across all temperature measurement points were strictly maintained below 15%, with the vast majority falling under 10%. Furthermore, the spatial variation trends were fundamentally identical.

To further quantify model accuracy, the root mean square error (RMSE) was calculated, as it is a widely accepted validation criterion for CFD models. Generally, an RMSE of less than 15% indicates that the numerical predictions are reliably consistent with experimental data [3]. For the proposed local CFD model, the RMSE values for the inlet air temperatures across Cases A through E were 1.88%, 3.73%, 1.82%, 3.76%, and 4.25%, respectively. The corresponding RMSE values for the outlet air temperatures were 9.74%, 4.35%, 11.20%, 3.56%, and 3.55%. Therefore, this study confirms that evaluating the local thermal environment by establishing a local CFD model is highly feasible for most scenarios within large and medium-sized data centers. This proposed numerical simulation method is computationally efficient and highly practical, significantly reducing the complexity of rack-level thermal assessments. Ultimately, it provides robust, reliable guidance for the local retrofitting and layout optimization of active, operational data centers.

4 Conclusion

In this paper, the local thermal environment within an operational air-cooled data center was thoroughly investigated. The impact of server layout and the use of blind plates on rack thermal profiles under partial-load conditions was experimentally evaluated utilizing custom server simulators. Furthermore, a steady-state CFD simulation methodology for assessing rack-level thermal environments was proposed and rigorously validated against the empirical data. The primary conclusions are summarized as follows:

- (1) The developed 2U server simulators closely mimic the physical and aerodynamic characteristics of actual commercial servers. Because they can be installed directly into existing target racks and integrated with precision measurement devices, they enable the rapid, *in-situ* evaluation of real-world rack thermal environments.
- (2) Both server layout and the implementation of blind plates profoundly impact the rack's thermal environment. For optimal cooling efficiency, the four 675 W, 2U servers were best positioned in the lower section of the target rack. This specific configuration yielded the lowest IOM value of 0.28, indicating minimized airflow mixing and a superior local thermal environment. Additionally, blind plates are critical for physically preventing the short-circuiting of cold supply air and the recirculation of hot exhaust air; omitting them was shown to increase the target rack's IOM by up to 718%.
- (3) The local CFD model encompassed three racks, adjacent hot and cold aisles, and an uncontained overhead clearance space to accurately capture the complex mixing dynamics of the hot and cold airflows. The internal servers were effectively simplified as porous media using a flow resistance model calibrated with experimental data. Ultimately, the spatial variation trends of the numerical predictions consistently aligned with the experimental observations.
- (4) Across the five evaluated server layout configurations, the vast majority of deviations between the numerical and experimental temperatures at the inlet and outlet measurement points remained strictly below 10%. The maximum RMSE values for the inlet and outlet air temperatures were calculated at 4.25% and 11.20%, respectively. Consequently, establishing this local CFD model is a highly feasible, accurate, and practical approach for evaluating rack-level thermal environments in large and medium-sized data centers.

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