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Optimization on jet-induced ventilation to enhance the uniformity of airflow distribution in data center

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Abstract

Improving the utilization efficiency of cold airflow in data center (DC) has already attracted widespread concern. A broad consensus has been reached that cold/hot-aisle containment technologies can reduce the mixture of cold and hot air to weaken the overheating phenomenon of the rack in a certain extent. However, local hot spots still occur in the front racks due to lower tile flow rate at the entrance of the cold aisle, which means the thermal environment of front racks can be further improved to achieve the more uniform airflow distribution horizontally and vertically in DC. In this paper, an innovative method of airflow optimization applying jet fans in the cold aisles is proposed to make up the lower tile flow rate, adjust the flow path of cold air from the perforated tiles to racks, and balance temperature heterogeneity. In addition, inductive velocity, nozzle height, horizontal position, and attachment distance of jet fans are optimized to explore the optimal parameters. Results show that the incorporation of jet-induced ventilation can effectively improve the cooling performance and overall thermal environment in DC, while the amount of IT equipment that exceeded the ASHRAE recommended supply air temperature (SAT) is reduced by about 38%. The jet fan with optimal parameters has a broad application space in the raised floor DC.

KEYWORDS

air flow distribution, CFD investigations, data centers, jet-induced ventilation system

1 INTRODUCTION

Data centers (DCs) are expensive and energy-intensive computing structures that house an enormous number of information technology (IT) equipment and relevant supporting systems. With the rapid development of computer technology in network information age, the number and scale of DC are growing continuously, the problem of high energy consumption is becoming increasingly prominent.¹ The energy consumption of DC has accounted for 2% of the total generation capacity of the United States and is growing by about 12% every year.² Therefore, it is urgent to take various measures to reduce the energy expenses required for DC. To keep the severs operating safely and prevent them from overheating,³,⁴ cooling system is indispensable and the cooling power takes up about
45% of the total energy consumption in DC.\(^5\) Accordingly, the improvement of cooling efficiency and the balance between cooling energy consumption and safety operation of servers are worth further studying.\(^6,7\)

At present, many strategies have been proposed to improve the efficiency of the cooling system and reduce energy consumption in DCs, including airflow management, free cooling, higher allowable IT temperatures, and cooling management. Among them, free cooling and airflow management are generally considered as effective and promising methods to cut down the energy consumption of cooling system in DC.\(^8,9\) There are three ways of free cooling strategy: air-side free cooling, water-side free cooling, and heat pipe free cooling.\(^9-13\) Zhang et al\(^10\) reported free cooling is an ideal energy-saving strategy which uses natural cooling source to cool the DCs when the outdoor temperature is relatively low. Ma et al\(^11\) also agreed that free cooling with better cooling performance could reduce energy consumption of DCs. However, the application of free cooling strategy required strict environmental conditions has certain restrictions and limitations. Outside air must be sufficiently cool and dry in air-side free cooling strategy.\(^10,12-14\) Water-side free cooling relies on natural cold water which brings potentially harm to IT equipment in DCs.\(^14\) According to T. Ding,\(^15\) the current immature technology limits the wide application of heat pipe free cooling technology. In contrast, air flow management strategy is convenient to implement and operate, hence it is considered as the mainstream method to improve the thermal environment and cooling efficiency of DCs.

As we all know, airflow distribution is an important bridge connecting thermal environment and energy consumption of cooling system in DC. Airflow management aims to maximize the cooling performance with minimum energy consumption. Nowadays, the under-floor air supply system (UASS) is widely used in contemporary DCs because it allows an energy-efficient configuration of above-floor hot and cold aisles.\(^16,17\) Great efforts have been conducted on the airflow distribution optimization of the UASS in the lower or upper spaces separated by the raised floor respectively.

In the lower space, the structure parameters of plenum, such as the plenum height,\(^18-22\) floor perforation rate,\(^18,23,24\) the layout of cables and pipes,\(^23,25,26\) and the adoption of new structures\(^22,27,28\) in the plenum are widely investigated. Schmid\(^23\) and Fakhim\(^26\) studied the effect of the layout of cables and pipes in the plenum on airflow from the perforated tiles, the results indicated that reasonable layout effectively enhanced the uniformity of airflow exiting from perforated tiles and reduced the rack inlet temperatures significantly. Yuan et al\(^27\) in 2017 applied under-floor flexible baffles (UFBs) to transform the under-floor airflow, the result indicated that the application of UFBs reduced the rack hot spots effectively and improved the cooling performance and thermal environment in DC. In 2020, Lu et al\(^28\) indicated that plenum with gradient cross-sections created using inclined partitions significantly improved the temperature uniformity and reduced the hot spots. Besides, Tradat et al\(^22\) in 2021 proposed a novel approach using porous partitions in the plenum to eliminate the presence of vortices in the plenum and thus enabled a more uniform pressure distribution and tile airflow delivery.

In the upper space, some containment technologies are applied such as cold aisle containment system (CACS), hot aisle containment system (HACS), and the blanking panel segregation.\(^21,22,24,28,33\) The research results of Nada et al\(^19,30,33\) showed that CACS improve the cooling performance of servers, especially at high power density by reducing the rack inlet temperature by about 13%-40%. Tatchell-Evans et al\(^31\) investigated the effect of cold air bypassing in DC with CACS by numerical simulation and experimental investigation. It is found that energy consumption of DC can be reduced by up to 16% without cold air bypassing. In addition, the layout of DC such as rack arrangement, computer room air conditioning (CRAC) units layout,\(^33\) and the operating parameters such as air supply temperature, air supply volume\(^28,34\) are also be optimized. Yuan et al\(^35-38\) proposed three innovative airflow optimization methods as follows: In 2018, Yuan et al\(^35,36\) proposed flexible baffles (FBs) setting in air inlet of racks, it was found that the FBs significantly improved the thermal environment in DC and effectively weakened the rack hot spots. In addition, Yuan et al\(^37\) put forward an innovative airflow management method of tilting servers, the results showed that installing servers with a 30° tilted angle weakened the overheating phenomenon significantly minimizing the possibility of local hot spots. In 2020, Yuan et al\(^38\) introduced and analyzed a new concept of an in-rack-cold-aisle (IR-CA) system to replace...
under-floor system with cold aisle containment. Their results illustrated that the optimal thermal distribution was achieved for the IR-CA case with partition plane among the all cases.

However, no matter what energy-saving strategy is used, most of the methods are based on overall passive adjustment methods and focused on the air flow distribution optimization in the horizontal direction. Few research on the locally active optimization of airflow distribution are proposed in DC. It should be noticed that how to evenly distribute the air volume of each rack to reduce the vertical temperature difference is still a problem unsolved, especially for the front racks at the entrance of cold aisle. Almoli et al. attached liquid loop heat exchangers and additional fans at the rear of racks. The numerical results showed that it improved the cooling performance and the airflow distribution of DC. Consuming the lower power by fans to further reduced load on the CRAC units was proved to be a completely feasible method to achieve energy-saving. Song proposed a fan-assisted cooling system setting fans in the plenum in under-floor DC. The results showed that fan-assisted perforations was an effective technique to improve the cooling performance of DC, while the fan efficiency and the cooling performance were related to the height of fan within the plenum. To balance cooling energy consumption and safety operation of the IT equipment in DC, only using the overall adjustment mode is becoming increasingly problematical due to inexorable increases in power densities of servers. Using targeted local adjustment for hot spots is an important omission in DCs.

It has been proved that the cooling performance and thermal environment can be significantly improved by fan-assisted technology in DCs. As we all know, no-duct inductive ventilation system has been widely used in large space such as underground garages, tile tunnels, and vessels. The raised-floor DCs also have the characteristics of large space. Taking a sub-module of DC with the CACS in an array of even-numbered columns as the investigated subject, this paper innovatively proposes a locally active optimization method and build a new concept of the coupling air supply system (CASS) with the incorporation of jet-induced ventilation in the UASS without changing the original air supply structure. The advantage of the jet fans instead of the traditional fans is that the inductive velocity is larger, and the stronger auxiliary air flow effectively weaken the cooling capacity attenuation along the vertical direction to eliminate the hot spots in the top servers. In addition, the virtues of easy installation, flexible adjustment, positive ventilation effect and low operation cost also make the jet fans be applied widely.

In order to make up for lower air volume at the front racks, jet fans are setted at the entrance of the cold aisles where has large air velocity gradient in this paper. It can be found that the overall temperature difference of rack inlet and outlet is reduced obviously in both horizontal and vertical directions, and the rack hot spots is minimized significantly. In addition, numerical simulation and experimental verification are used to investigate the effect of inductive velocity, nozzle height, horizontal distance, and attachment distance on thermal performance of DC, so as to obtain the optimal parameters of jet fan. In general, the application of jet fans improves the utilization rate of cooling capacity, enhances the airflow distribution uniformity and cooling performance in DC. It is hoped that this study can provide some reference for DC thermal performance optimization and energy saving in the future.

2 | METHODOLOGY

2.1 | Data center description

The DC is located in Nanjing, China, which is used to provide information processing, network construction, and communication services for a university. The layout of the DC is shown in Figures 1 and 2. To keep simulation consistent with experiments, all the parameters of the physical model are collected from the field experiment. A raised-floor DC of dimensions 10.9 m x 10.4 m x 3.0 m (L x W x H) is considered as the physical model. A typical raised-floor plenum of depth 0.6 m and perforated tiles of 0.6 m x 0.6 m size with an opening ratio of 30% are used for all the analyses. The DC contains 44 cabinets arranged in four rows with a spacing 1.2 m between the two rows, which are marked as A, B, C, and D, respectively. Racks in each row is marked from 1 to 11 from left to right. A typical rack dimension of 1.1 m x 0.6 m x 2.0 m (L x W x H) is considered.
The average heat density is 2-5 kW/rack in most DCs. According to the actual load operation of the rack in the tested DC, it is assumed that the load of the rack is 2 kW/rack in the numerical simulation. Each rack contains 16 servers (2 U) with dimensions 0.8 m × 0.5 m × 0.08 m (L × W × H) in the experimental field, and the distance between adjacent servers along the height direction is 100 mm (see Figure 6). To reduce the calculation and facilitate simulation, the 16 servers in the rack are divided into four modules (see Figure 8). Each module consists of four servers and has equal heat generation of 500W with dimensions 0.8 m × 0.5 m × 0.35 m (L × W × H). Zhang et al. and Lu et al. used the same simplified method for simulation convenience successively. The two CRAC units are symmetrical with dimensions 1.70 m × 0.87 m × 1.97 m (L × W × H). DC specifications, parameters settings and operating conditions are summarized in Table 1.
2.2 Working principle of jet-induced ventilation

According to Yuan et al., heat accumulation mostly occurs in the middle and top of the rack, and the thermal environment at the bottom of the rack is relatively good in DC. To explain this uneven temperature distribution, Figure 3A illustrates the raised floor DC arrangement with CACS. As can be observed in the image, the CRAC units put cold air into the plenum under the floor and recycle the hot air emitted by servers from the hot aisles. The cold air rises from the plenum into the DC through perforated tiles that are positioned in the cold aisles, subsequently it flows into the racks to cool the servers. Meanwhile, the hot air exhaust is expelled from the rear of the racks, finally it returns to the air inlet vent of CRAC units. It has been proved that CACS can effectively prevent the hot air from flowing back into the rack inlets from the sides and the top. However, the thermal environment of the front racks and the overall airflow uniformity in DC can be further improved by more advanced techniques. For the not enough agreeable cooling performance and airflow distribution in DC with CACS, this phenomenon can be attributed to the following two aspects. On the one hand, due to close distance between the front racks and CRAC units, the lower tile flow rates are obtained at the entrance of the cold aisles, besides the cooling attenuation along the vertical direction causes less cold air flow into the top server at the front racks to take away the heat generated by the server. On the other hand, due to the buoyancy effect, the hot air tends to move upward, then a large amount of high-temperature exhaust steam is accumulated on the top of the front racks resulting in a large static pressure, it overcomes the pressure of the supply air inlet of the racks, so part of the hot air backward flows to the cold aisle from the rear of the racks before it returns to the air inlet vent of CRAC units, resulting in mixing of hot and cold air in the top of racks. In summary, the front racks in the closed cold aisles are still prone to overheating to damage the servers. Therefore, setting the jet fans at the cold aisle entrance with larger velocity gradient is effective method to improve the uniformity of airflow distribution in DC, while the lower air volume is compensated and the reverse flow of hot air is eliminated.

Model specifications for rack and jet fan are depicted in Figure 4. The jet fan is composed of fan box, three round nozzles and air return outlet. The dimension of jet fan (YDF-B-2.5, Shangyu Fans, Inc) is 0.50 m × 0.48 m × 0.23 m (L × W × H). The diameter of the nozzles is 0.07 m and the size of the air return outlet is 0.5 m × 0.23 m. The specific technical scheme is as follows: To make up for the lower inlet air volume of the front racks, the jet fan is positioned at the entrance of cold aisles and close to the racks, the air return outlet and nozzles of jet fan parallel to perforated tiles, such that it interacts with the airflow from the perforated tiles and enhances the vertical flow motion with active momentum. That is, a part of the cold air flow from the perforated tiles directly enters the bottom server to take away the heat generated by servers, and a part of the cold air flow enters the air return outlet of the jet fan to obtain enough momentum, which increases the cold air into the top server at the racks and eliminates the rack hot spots in DC. In summary, as shown in Figure 3B, the application of jet fans alleviates the cooling attenuation along the height of the rack, enhancement of cooling performance is attained for top servers at the racks in the CASS. In addition, the strong auxiliary jet increases the pressure of the rack air inlet to prevent the exhaust flow from returning to the cold aisles, and drives the hot air to CRAC units in time.

To explain the active effect of the jet fan on the airflow in DC, the airflow path and velocity contours of jet fan are shown in Figure 5. The low-speed cold air from the perforated tiles flows into the return outlet of the jet

![Figure 3](image)

**FIGURE 3** Comparison of airflow in DC: A, UASS; B, CASS
fan to obtain the momentum enhancement. When the airflow with the high speed inject from the nozzles, it continuously sucks the surrounding low-speed cold air and induces them into the top sever at the racks. Besides, the axial velocity decays unceasingly and the area of the rack inlet covered by the jet gradually expands because of the entrainment effect. Note that, when the jet fan reaches a higher height or a larger inductive velocity, part of the jet can reach the top of the cold aisle containment, it impinges on the closed roof and subsequently diffuses to both sides or flows down rapidly. The inductive velocity and nozzle height of the jet fan have significant impact on the airflow path to affect the cooling performance of the servers. Hence, it is necessary to optimize the installation position and inductive velocity of jet fan to obtain the optimal parameters for local adjustment.

2.3 | Experiment methods

2.3.1 | Experimental setup

The overall arrangement in the experimental DC is shown in Figure 6. The constructions of the tested DC meet the requirements in the China National Standards. Two CRAC units located at one end of the DC are both turned on and operate all day. The racks are arranged in a “face to face” way. Symmetrical layout and forward air supply are adopted in DC. The cold aisles of DC are sealed by special glass walls, and the door is closed during the tests to avoid the indoor and outdoor air exchange to reduce the experimental error.

In this study, the purpose of the experiment is: (1) to verify the reliability of the numerical simulation results and (2) to investigate the effects of jet fan on the thermal performance and the adaptability of jet fan in DC.

2.3.2 | Measurement setup

As we all know, the thermal performance is affected by airflow fields and temperature distribution. Therefore, the temperature and velocity of air are the main items to be measured. Considering that the rack A, B and rack C, D in the DC are symmetrically distributed with the hot aisle as the symmetry line, in order to reduce the measurement workload, rack A1-A11 and B1-B11 are selected for measurement in the experiment. To effectively investigate the inlet and outlet air temperature distribution in the vertical direction of the racks, three representative sections (Z = 0.4/1.2/1.6 m) were selected to measure the air temperature around the racks. The measuring equipment information is shown in Table 2. In the experiment, six T-type thermocouples were set at the front and back doors of each rack and connected with Agilent data acquisition logger to measure the inlet and outlet air temperatures of racks at the same time. Besides, Agilent data acquisition logger connected with the computer to process the data collected by each sensor to obtain the air flow temperature distribution around the racks. In the process of measurement, the Agilent data acquisition logger recorded the...
inlet and outlet air temperature from the sensor every 10 seconds for 90 minutes, which meant there were ~540 temperature data for each measuring point in each measurement case.

The cold air velocity measurements were performed on the perforated tiles in the cold aisles. Figure 7 illustrates the measurement location and arrangement in the perforated tiles. Nine measuring points located 0.1 m above the raised floor were selected in porous tiles to be measured respectively, and the average value of nine test values was selected as the flow rate of the test tile in each measurement case. To obtain more accurate experimental results, nine experiments were carried in 3 days. Finally, the average value of multiple measurements was taken as the final result to improve the accuracy of the experimental results.

### TABLE 2 Specific parameters of experimental instruments

| Experimental instruments | Type            | Measuring range | Accuracy |
|--------------------------|-----------------|-----------------|----------|
| Thermal anemometer       | TESTO 425       | 0-20 m/s        | ±5%      |
| Temperature sensor       | T-type thermocouples | 273-343 K | 0.01 K   |
| Data acquisition logger  | Agilent 34972A  | N/A             | N/A      |

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### 3 NUMERICAL SOLUTION

#### 3.1 Mathematical formulation and numerical solution techniques

The governing equations, initial and boundary conditions (Table 3) are used to simulate fluid flow and temperature distribution in the physical model. The governing equations for the incompressible fluid are
as shown in Equations (1)–(3), which represents continuity, momentum and energy conservation equation, respectively.

\[ \nabla \cdot \overline{\mu} = 0 \quad (1) \]

\[ \frac{\partial \mu}{\partial t} + \overline{\mu} \cdot \nabla \overline{\mu} = \nabla (v_{eff} \nabla \overline{\mu}) - \frac{1}{\rho} \nabla p + \overline{g} \quad (2) \]

\[ \rho c_p \left[ \frac{\partial T}{\partial t} + (\overline{\mu} \cdot \nabla) T \right] = \nabla \cdot (k_{eff} \nabla T) + S \quad (3) \]

The coupled partial differential equations were solved using numerical solution techniques with finite volume discretization in the three-dimensional DC computational domains. In previous similar studies, standard k-\( \varepsilon \) turbulence was generally used in the DC computational model and obtained good results. Therefore, in this paper, the standard k-\( \varepsilon \) turbulence model is still used with an enhanced wall function.\(^{35,37}\) Besides, grid-generation was based on GAMBIT software (version 2.4.6) and the results were analyzed on ANSYS FLUENT software (version 18).

### 3.2 Basic assumptions

A series of simplification were made in the physical model to facilitate numerical simulation.\(^{28,34}\)

1. The flow was regarded as incompressible fluid with low speed, the heat dissipation caused by fluid viscous force was ignored.
2. The airflow satisfied the Boussinesq hypothesis that the change of fluid density only affected the buoyancy.
3. The heat dissipation of servers was determined by its type and usage situation. In fact, the heat dissipation of each server was different and the heat flux density was not constant. To facilitate the numerical calculation, each rack consists of four server modules with equal heat dissipation 500 W was defined (see Figure 8).
4. The front/back doors of racks were provided with holes with certain opening (rack porosity is 60%), and the resistance of racks inlet/outlet was simplified.
5. The plenum interior was simplified in the model, the plenum chamber obstructions and the disturbance of airflow leakage were not considered.
6. It was assumed that the external walls, raised-floor, rack panels, and inner partitions were adiabatic. Besides, the thermal radiation of surface in the room was ignored, and the effect of external heat transfer in DC was not considered.

### 3.3 Initial and boundary conditions

Turbulence transmission is an indispensable part in the airflow simulation of the DC. Setting the correct boundary conditions for the physical model is the prerequisite...
to ensure the correctness of the simulation results, and reasonable initialization of the flow field can speed up the process of numerical calculation to shorten the calculation time. The boundary conditions and convergence criteria are summarized in Table 3. Note that DC was considered as a confined space enveloped with a defined solid-wall boundary. The outlets and inlets of the airflow were the air supply outlet and the return air inlet of the CRAC units respectively. Hence in the physical model, the air supply outlet of CRAC units was defined as the inlet for the computational domain, and the air supply velocity of CRAC units was set as initial value for simulation. Flow
was assumed to be steady with the air supply temperature of CRAC units at 18°C, and the ambient temperature in DC was assumed to be 28°C. Note that the perforated tiles were set as the porous media boundary condition that only provided with certain perforation rate and resistance coefficient, and the thickness of perforated tiles was not considered.

3.4 | Grid generation

Grid generation plays an important role in the accuracy of simulation results. The tremendous increasing of the grid number not only prolongs the calculation time, but also increases the chance of calculation errors, while too few grids may lead to the inaccuracy of simulation. To generate the better-quality grid for models, the method of structured hexahedral mesh was used in this model. The maximum size of the mesh in X, Y, and Z directions was set as 0.12 m, 0.12 m, and 0.1 m, respectively. After the temperature field was calculated, the grid independence study was conducted to ensure the accuracy of the calculation results. Considering that the temperature was one of the most effective parameters in the problem, thus the maximum temperatures of the room and average temperatures of racks were used to assess the calculation accuracy. The grid independence test of CASS model is shown in Table 4. Grid independence solutions were tested by the mesh number from 935,263 to 10,026,405. It can be found that simulation value changed slightly with the increase of the mesh numbers. The results showed that good grid-independence at 2,477,592 grid cells.

To make sure that the solution is stable, the maximum mesh size in X, Y, and Z directions was reduced to 0.04 m, 0.04 m, and 0.03 m, respectively. Besides, local refinement of grids was deployed to make the mesh number reach 10,036,405, which calculated the same result with 2,477,592 grid cells. Considering a good trade-off between the accuracy of calculation and computation time, 2,477,592 and 3,501,524 were chosen as the mesh numbers of UASS and CASS models respectively in the simulation. It should be noted that the required grid number in CASS model was greatly increased due to the application of jet fans. Therefore, the mesh number of 2,477,592 for UASS was enough to capture the flow and temperature fields with high accuracy. The element number of 3,501,524 was used to run the CASS model to achieve higher accuracy.

3.5 | Model validation

As depicted in Figure 8A, four representative locations were selected to measure the intake air temperature and compared with the simulation values. Four T-type thermocouples as the temperature sensor were set on the front door of rack A1-A11 and connected with Agilent data acquisition logger. Taking the average value of the inlet temperatures of rack A1-A11 at the same measuring point as the temperature of this monitoring point, the variation of the experimental and simulated outcomes with the monitoring points was shown in Figure 8B, C. The results show that the top part of the racks has higher inlet air temperature than the bottom parts in the UASS. Obviously, the inlet temperature uniformity is significantly improved in the CASS. It is noticed that the simulation temperature results are a little higher than experimental results on the whole. The difference may be attributed to the comprehensive effect of field equipment, such as thickness of the perforated tiles, outlet angle of cold air, air leakage, or heat sources like lights, which are not considered in the simplified model. Thermocouple is temperature sensor with extremely high sensitivity. Cold air leakage, light source and tester heat dissipation increase the experimental value of rack inlet temperature. Beside, some racks are not full of servers and blanking panel segregation is applied on the vacant racks. This means some racks did not operate at their full load (2.0 kW) during the measurement, but in the process of numerical simulation, it is assumed that all racks are full of servers. Therefore, the reduction of servers reduces the rack inlet temperatures in the experimental test. The results indicate that the impact of not full racks is greater than that of cold air leakage and heat source. Even so, the agreement between the measurements and calculations is satisfactory (within 5% relative error).

| Element number | $T_{\text{max}}$ of the room | $T_{\text{max}}$ ($Z=0.5$ m) | $T_{\text{max}}$ ($Z=1.5$ m) | $T_{\text{ave}}$ of racks |
|----------------|-----------------------------|-----------------------------|-----------------------------|-----------------------------|
| 935,263        | 39.184°C                    | 33.437°C                    | 38.016°C                    | 25.168°C                    |
| 1,346,826      | 39.249°C                    | 33.401°C                    | 38.035°C                    | 25.714°C                    |
| 1,663,381      | 39.254°C                    | 33.577°C                    | 38.044°C                    | 25.956°C                    |
| 2,477,592      | 39.256°C                    | 33.579°C                    | 38.047°C                    | 26.037°C                    |
| 3,846,746      | 39.256°C                    | 33.579°C                    | 38.047°C                    | 26.037°C                    |
| 10,026,405     | 39.256°C                    | 33.579°C                    | 38.047°C                    | 26.037°C                    |

TABLE 4: Study on grid independence of $T_{\text{max}}$ of room and $T_{\text{ave}}$ of racks variations with mesh number for CASS model.
3.6 Numerical calculation process

The CASS DC with closed cold aisles is adopted in the present investigation. Under the condition of keeping room load, the cooling capacity of CRAC units, layout arrangements of racks and numbers of jet fans unchanged, the main factors affecting air flow temperature distribution of DC includes inductive velocity, nozzle height, horizontal distance, and attachment distance. In addition, the influence of four factors is not independent, but interrelated. Therefore, the numerical calculation process in Figure 9 is proposed according to the above factors. “/0” in the figure represents the result without optimization and “/1” represents the result with optimization.

The whole design process can be divided into the following four steps:

1. Remain the horizontal distance, nozzle height, and attachment distance unchanged, adjust the inductive velocity, then use computational fluid dynamics (CFD) to simulate the temperature fields in DC with inductive velocity of 0.5/2.0/3.0/4.0 m/s respectively, and select the inductive velocity corresponding to the optimal temperature distribution to enter the next step.

2. Take the inductive velocity calculated in the first step as the set value, while keeping the horizontal distance and attachment distance unchanged, adjust the nozzle height of the jet fans, then the airflow distribution in DC with different nozzle height (0.7/0.9/1.2/1.4/1.6/1.8 m) are calculated and analyzed, so as to obtain the best inductive velocity and nozzle height in the optimal temperature filed. Besides if the inductive velocity is consistent with the set value through comparison, the next step of optimization will be carried out; if not, adjust the set value of nozzle height according to the calculation results, then return to (1) for recalculation.

3. Keep the inductive velocity, nozzle height and attachment distance determined in the second step unvaried, change the horizontal distance of the jet fans, and simulate the thermal environment in DC when the jet fan is located at 1#, 1#-2#, 2#, 2#-3# respectively to obtain the suitable horizontal distance for the second step.

4. The influence of four factors is not independent, but interrelated. Input the value of the third step, the optimal attachment distance is studied by CFD simulation, then the combination is taken as the final calculation result.

Note that, the design process of optimization for interrelated factors in this paper is developed by Zhang et al. They studied the influence of four interrelated factors such as the porosity of front door of rack, the porosity of internal partition of rack, the number and location of exhaust fan in the racks and the number of grille diffusers on the thermal performance in the raised floor DC. Using the same numerical calculation process, they obtained the optimal parameter range and achieved the optimization of cabinet structure. Meanwhile, the on-site experiment was carried out to prove the correctness of the method.

4 RESULTS AND DISCUSSIONS

4.1 Inductive velocity

Figures 10 and 11 show the simulated results of UASS and CASS, (a), (b), (c), and (d) indicate that the inductive velocity of 0.5/2.0/3.0/4.0 m/s, respectively. The results are consistent with Zhang et al. and Lu et al. at the similar physical model and boundary conditions, which exhibited a relatively higher non-uniformity degree in the UASS. Figure 10 illustrates that the serious overheating phenomenon occurred in the front racks with the hot spots of 38°C. Besides, the overheating area in the racks expanded gradually along the height of the racks. The thermal environment of the top severs in the rack is relatively hotter than those at lower height of the rack, which may cause damage to the servers. Compared with the temperature fields of UASS and CASS, the overheated area at the front racks is decreased by 90% in the CASS, the maximum temperature of DC is reduced by 6°C, and the uniformity of cabinet surface temperature is significantly improved. Table 5 shows the temperature distribution of Z = 1.2 m section in the CASS. Obviously, the hot spots in the front racks is eliminated, the uniformity of airflow distribution and thermal environment in DC are improved significantly.

In the raised floor DC, the cold air flow from the plenum into cold aisles through perforated tiles, and then enters the racks to take away the heat generated by severs. Meanwhile, the hot air rises in the hot aisle and returns to the CRAC units along the upper space of the room. Considering that the rack A, B and rack C, D are symmetrically distributed in DC, the air flow streamlines of rack A and B are shown in Figure 11. The flow path of jet is greatly affected by inductive velocity. The results show that the airflow on the left and right sides of the jet fan is enhanced effectively to take away more heat of front racks. After the jet is ejected from the nozzles, the jet with lower inductive velocity steady flows into the top severs. When \( V_i \) is 0.5 m/s in Figure 11A, the jet only enters 2# cabinets stably, the highest temperature of rack B1-B4 reaches 31°C and the overheated area exceeds 30%. When \( V_i \) is 2.0 m/s in Figure 11B, the jet covers the first three rows of cabinets averagely, minimizing the overheated area can be seen in the rack B1-B4. As the inductive velocity increases continuously, the jet trajectory in the cold aisles also changes.
The jet with larger inductive velocity is constrained by the closed roof in the cold aisles ($Z = 2$ m), it collides with the closed roof and subsequently diffuses to both sides, or flows down rapidly. The larger the inductive velocity is, the more jet prefer to flow down after impinging. In addition, the exhaust air volume of front racks increases with the increase of supply air volume, and larger exhaust air volume also affect the exhaust efficiency of the racks. When $V_i$ reaches 3.0 m/s in Figure 11C, the jet begins to collide with the closed roof and then diffuses to both sides into the racks. The impinging increases the diffusion range of the jet, which covers the first four rows of
FIGURE 11  Temperature distribution and air flow streamlines with different inductive velocities in DC
cabinets. However, it should be noticed that less jet flows downward after impinging and circulates in the cold aisle without entering the racks. Besides, the hot air accumulation occurs at the exhaust outlet of the rack B1, which means the large air volume of high rack altitude may limit the exhaust of the bottom server. When $V_i$ continues rising to 4.0 m/s in Figure 11D, the airflow of rack inlet and outlet are unstable due to the excessive air volume. Then most of the jet flow downward and circulate in the cold aisles, the jet flowing to top sever decreases instead. Besides, hot air accumulation can be found around the air return outlet of the CRAC units and the rack B11. This can be attributed to excessive air volume of the front racks results in relatively higher pressure at the front end of the room, which limits the hot air exhausted from the rear racks to return to the CRAC units in time, resulting in hot air accumulation around the rear rack.\cite{19,33} Note that, due to the obstruction of the fan box itself and the limitation of nozzle angle, the jet cannot reach the bottom of racks, so the temperature of the bottom server is relatively higher than top server with active effect of jet fan, which needs to be further optimized in future research. In summary, the temperature uniformity and cooling efficiency in DC are both improved with the different inductive velocities of jet fan, while the optimal thermal environment is achieved when the inductive velocity is set to be 2.0-3.0 m/s.

### 4.2 Nozzle height

Remaining the inductive velocity, attachment distance and horizontal distance of the jet fans unvaried, Table 6 shows the temperature distribution of vertical sections $X = 3/6/9$ m, as well as the horizontal sections $Z = 0.4/1.2$ m, where (a), (b), (c), (d), (e), and (f) represent the nozzle height of 0.7/0.9/1.2/1.4/1.6/1.8 m, respectively.

Compared with the temperature fields in Figure 10 and Table 6, it can be observed that six CASS models with different nozzle heights all have less local hot spots and more uniform temperature distribution in DC. By adjusting the nozzle height, the overheated area is further weakened and the airflow distribution uniformity is gradually enhanced in DC. When $H_n \leq 1.4$ m, the temperature has evident increase as the nozzle height decreases, especially for the temperature field of the hot aisle. When $1.4$ m $\leq H_n \leq 1.6$ m, the temperature field of the hot aisle is effectively improved, and the overheated area decreases gradually as nozzle height increases. Moreover, the maximum temperature of cabinets drops by 1-3°C and the ambient temperature in the hot aisle is decreased by 1-2°C. However, when $H_n \geq 1.8$ m, the collision between the jet and the closed roof results in huge energy loss and the diffusion speed is reduced greatly. It causes that the jet entering the bottom servers at the front racks is decreased to take away the less heat generated by severs, hence the cooling effect in the CASS is relatively deteriorated compared with others nozzle heights. In consequence, the more uniform airflow distribution in DC is achieved when $V_i$ is set to be 2.0-3.0 m/s and $H_n$ is set to be 1.4-1.6 m.

### 4.3 Horizontal distance

Figure 12 shows the temperature fields and air flow streamlines with different horizontal location of jet fan in the CASS. It can be found that the position of induced fan and jet direction determine the flow path of the jet. In Figure 1A, the jet fans are placed in front of 1# cabinets ($X_c = 3.2$ m). Because the cold aisle’s closed door prevents the airflow from spreading outward, part of jet reverse direction after colliding with the closed door, then the airflow is bypassed in the cold aisles without flowing to the racks. Hence 60% of the jet flows into 2# cabinets. There is little jet entering 1# cabinets, the maximum surface temperature at the rear of the 1# cabinets is as high as 31°C. In Figure 12B, when jet fans...
TABLE 6 Temperature fields of CASS with different nozzle height in DC

| Nozzle height | Vertical directions | Horizontal directions | Z = 0.6 m | Z = 1.4 m |
|---------------|---------------------|-----------------------|-----------|-----------|
| 0.7 m (a)     | ![Image](image1.png) | ![Image](image2.png)  | ![Image](image3.png) | ![Image](image4.png) |
| 0.9 m (b)     | ![Image](image5.png) | ![Image](image6.png)  | ![Image](image7.png) | ![Image](image8.png) |
| 1.2 m (c)     | ![Image](image9.png) | ![Image](image10.png) | ![Image](image11.png) | ![Image](image12.png) |
| 1.4 m (d)     | ![Image](image13.png) | ![Image](image14.png) | ![Image](image15.png) | ![Image](image16.png) |

(Continues)
are located between 1#-2# cabinets ($X_c = 3.5 \text{ m}$), the jet flows into the first two rows of cabinets restricted by the closed roof in the cold aisles, and the overheated area of 1# cabinets is significantly reduced. In Figure 12C, the jet fans are placed in front of 2# cabinets ($X_c = 3.8 \text{ m}$), the jet ejecting from the nozzles collides with the upper roof, then it diffuses into 1# and 3# cabinets. The local hot spots of 30°C occurred at 2# cabinets with relatively poor cooling performance compared with other cabinets. The main reason is that most of the jet flows vertically and stays in the cold aisles and the few jet enters the cabinets due to the continuous reduction of momentum during diffusion process of jet from the nozzles to the cabinets. When jet fans are located between 2# and 3# cabinets ($X_c = 4.1 \text{ m}$) in Figure 12D, the first three rows of cabinets all have jet entered, the maximum scope of jet fan is obtained in the CASS. There is the smallest overheated area of racks in DC, while the maximum temperature of racks is decreased by 1-2°C compared with other locations. In addition, the position of the jet fan in the cold aisle also has an impact on the exhaust efficiency of the racks. The jet fans take away more heat generated by the servers, while the exhaust air volume of the racks increases with the increase of the air inlet volume. Along the air flow path, when the jet fan is set at the location which is the nearer to the CRAC units, a large amount of hot air accumulates around the CRAC units to block the airflow of other racks to return to the air inlet of CRAC units in time. As the jet fan location moves backward in the cold aisle, the phenomenon of hot air accumulation around the CRAC units is gradually alleviated, and the exhaust air from the racks can return to the CRAC units in time to improve the thermal environment in DC.

As we all know, temperature distribution of each rack is always influenced by the path of air flow, which is the direct embodiment of airflow distribution in DC. Figure 13 shows the variation of the average temperature in the rack B1-B4 in scenario S1-S4 with different horizontal location in the CASS. S1-S4 respectively represent the setting location of jet fan at 1#, 1#-2#, 2#, and 2#-3# in the cold aisle. Since the jet cannot cover all rack surfaces, the hot spots in the racks cannot be eliminated completely. In comparison, 2#-3# ($X_c = 4.1 \text{ m}$) is the optimal horizontal position of jet fan due to better balance of temperature distribution of the racks.

### 4.4 Attachment distance

Adjusting the attachment distance, Figure 14 illustrates the temperature distribution of servers in the CASS with $V_i = 2 \text{ m/s}$, $H_i = 1.6 \text{ m}$, and $X_c = 4.1 \text{ m}$. D0 means that the jet fan is not applied in the cold aisle. D1-D4 respectively represent the attachment distance of 0.05/0.1/0.15/0.2 m in the CASS. Comparing the number of servers in different temperature ranges, it can be found that the temperature uniformity of racks has a poor performance in the UASS.
FIGURE 12  Temperature distribution and flow path of jet for different horizontal distance
That is, 21% of servers are higher than 31°C, 36.4% of servers have relatively good cooling performance with the temperature <25°C. In the CASS, the number of overheated servers is reduced by 20% and the maximum temperature of cabinets is decreased by 6–9°C. 70% of the servers are 25–28°C, only a few servers are higher than 28°C but lower than 31°C, which can be attributed to the application of jet fans in the cold aisle taking away the more heat of servers. It is concluded that the CASS achieves a better cooling performance for average temperature reduction and temperature uniformity enhancement of servers compared with the UASS. In consequence, when $V_l = 2$ m/s, $H_n = 1.6$ m, $X_c = 4.1$ m, and $0.05 \leq D_a \leq 0.1$ m, the more uniform temperature distribution of servers in DC is achieved.

5 | COMPARISON BETWEEN TWO AIR SUPPLY MODES IN DC

5.1 | Inlet and outlet air temperatures

The inlet and outlet air flow are critical factors reflecting cooling performance of cabinets. Comparing the UASS and the CASS with $V_l = 2$ m/s, $H_n = 1.6$ m, $X_c = 4.1$ m, and $D_a = 0.1$ m, the inlet/outlet air temperatures of rack A and B at the height of 0.4/1.2/1.6 m are shown in Figures 15,16. Then the results are summarized in Table 7.

Zhang et al\textsuperscript{21} and Lu et al\textsuperscript{28} performed investigations of airflow distribution with similar physical model and boundary conditions in DC. Their results showed that serious non-uniformity of airflow distribution in both

![Figure 13](image1.png)  
**Figure 13** Variation of average temperature in rack B1-B4 in scenarios S1-S4

![Figure 14](image2.png)  
**Figure 14** Temperature distribution of servers in different DC models. A, UASS. B, CASS

![Figure 15](image3.png)  
**Figure 15** Air flow temperature curve at air inlet side of rack A and B. A, UASS. B, CASS
horizontal and vertical directions in the UASS: horizontal and vertical temperature difference of $T_{\text{in}}$ were 7-12°C and 3-7°C respectively, and temperature difference of $T_{\text{out}}$ were 4-5°C and 2-3°C horizontally and vertically. The maximum $T_{\text{in}}$ of 35°C also be observed in Lu’s measurements. In this study, $T_{\text{in}}$ are more than 25°C at the rack height above 1.2 m for 1#-4# cabinets in the UASS from Figures 15,16. As the distance increases between the racks and the CRAC units, the rear racks have lower $T_{\text{in}}$ than front racks. The reason for this phenomenon is that the increasing of tile flow rate along the cold aisle reduces the $T_{\text{in}}$ at rear racks, and the reverse flow of hot air caused by the accumulation of exhaust steam increases the $T_{\text{in}}$ at front racks. Higher inlet temperature worsens the cooling performance of the racks. Obviously, the CASS significantly improves the uniformity of airflow temperature horizontally and vertically, while the amount of IT equipment that exceeded the ASHRAE recommended SAT is reduced by about 38%.

When the total supply air volume of CRAC units remains unchanged, an decrease of inlet air volume at the rear racks occurs with an increase at the front racks to balance the air volume in the whole aisle. For the outlet airflow, the decrease of $T_{\text{out}}$ with the decrease of $T_{\text{in}}$ can be observed at front racks, while the more heat generated by severs is took out of the room due to an improvement of the cooling capacity utilization. For 5#-11# cabinets, as the result of the increase of $T_{\text{in}}$, the $T_{\text{out}}$ go up slightly but is still in standard temperature range.

In conclusion, the CASS solves the problem of uneven airflow delivery from the plenum to servers by making up lower flow rates at the entrance of cold aisles, reduces the temperature difference of cabinets, improves the utilization efficiency of the cooling capacity, and realizes the airflow uniformity horizontally and vertically in DC.

### 5.2 Airflow properties in DC

#### 5.2.1 Rack flow rates

As mentioned above, the application of jet fans eliminates the hotspots at front racks effectively and improves the airflow distribution uniformity and thermal performance in DC. To better explain this positive effect, the streamlines of air flow from tiles to CRAC units in the UASS and CASS are shown in Figure 17. There are relatively high inlet and outlet air temperatures at front racks in the UASS, which can be attributed to the following two aspects. On the one hand, the lower tile flow rates are occurred at the entrance of the cold aisles and the airflow volume attenuation along

| Table 7 Air flow temperature distribution around cabinets in the DC |
|---|---|---|---|---|
| Air supply methods | $T_{\text{in}}$ ($^\circ$C) | $T_{\text{max}}$ ($^\circ$C) | $\Delta T$ in horizontal direction ($^\circ$C) | $\Delta T$ in vertical direction ($^\circ$C) |
| $T_{\text{in}}$ | UASS | 19 | 36.5 | 8-12 | 3-8 |
| | Cass | 20 | 24 | 1.5-3 | 1-3 |
| $T_{\text{out}}$ | UASS | 24.5 | 35.5 | 4-6 | 2-3 |
| | Cass | 26.5 | 31 | 1.5-2.5 | 1-2 |
FIGURE 17  Comparison of streamlines showing temperature of air flow between UASS and CASS

FIGURE 18  Velocity vector diagram of rack inlet
the vertical direction causes less cold air flow into the top server of the front racks. This reason is further described with more detail in Figures 18, 19. On the other hand, due to the buoyancy effect, the hot air tends to move upward, then a large amount of high-temperature exhaust air is accumulated on the top of the racks resulting in a large static pressure, it overcomes the pressure of the supply air inlet of the racks, so part of the hot air backward flows to the cold aisle from the back of the racks before it enter the inlet vent of the CRAC units. The mixing of cold and hot air greatly increases the inlet temperature of the top server at the racks. In the CASS, it can be found that the airflow distribution and thermal environment of DC are improved significantly. This is because the jet fans significantly increase the air flow rates at the entrance of the cold aisles. As shown in Figure 18, the strong auxiliary jet enters the front racks to prevent the exhaust air from returning to the cold aisles, and drives the hot air to CRAC units in time.

The Figure 18A illustrates the maldistributed cold air inlet of the racks in the UASS. The increased flow rates are observed with the distance increase between CRAC units and tiles. There is a significant angle in the airflow parallel to the resistance plane due to an increase of the pressure drop in the horizontal direction of the cold aisles. According to Song, this phenomenon entails that more of the airflow emitted by the perforation tiles flow past the rack inlets without entering it. The maximum velocity of 0.23 m/s is obtained at the bottom of the racks, while the velocity is only 0.07 m/s at the top. Seriously attenuation of the cooling capacity leads to inadequate cold air for the top servers. As a result of the uneven pressure distribution in the cold aisles, the vortex is observed in area “a” (see Figure 18A). This vortex causes inlet airflow non-uniformity and may give rise to the rack hot spots. Inversely, more uniform rack airflow delivery can be obtained in CASS in Figure 18B. The vortex disappeared and the reorientation inclined air flow makes the air distribution more uniform. In addition, the upward auxiliary airflow produced by jet fans increase the velocity at area “b” (see Figure 18B), and this is helpful in eliminating the hot spots at the front racks. The maximum and minimum velocity are increased to 0.24 and 0.19 m/s, respectively. In summary, the application of jet fans avoids reversed flow of the hot air, takes away more heat generated by the servers and eliminates the hot spots at the front racks.

5.2.2 | Tile exit flow rates

The plenum structure remains unchanged, tile exit flow rates depend on the pressure difference between the static pressure in the plenum and the outlet of the tiles. The flow rate increases with the augmentation of pressure difference. The suitable and uniform pressure difference is the precondition to meet the cold air supply for cabinets. Figure 19 describes the airflow distribution of each porous tiles (1#-11#) in the cold aisles. In the UASS, it can be found that the flow rates of 1#-5# tiles have certain fluctuation, and 6#-11# tiles show a gradual increase along the cold aisles direction, which means the uniformity of the tile flow rates has a poor performance. By comparison, the flow rates of 2#-9# tiles are significantly increased, and 10#-11# are weakened obviously in the CASS, the exit flow rates of each porous tiles tend to be more uniform. It should be noticed that the flow rates have a slight increase for 1# tile. Because too large dynamic pressure exists at the location close to the inlet of plenum, the relay pressure generated by the jet is not enough to resist, which results in the lower flow rate at 1# tile compared with other perforated tiles. Further optimization can be done in the future researcher. In general, jet fans have a peak shifting and valley filling effect on longitudinal flow rates in the cold aisles. It can be seen that the perforated floor outlet uniformity is significantly improved, and the average flow rate is increased by 20% in the CASS. Moreover, improving the uniformity of floor outflow can avoid cold air bypass and hot air circulation to a certain extent in DC.

6 | CONCLUSION

To improve the cooling performance and the uniformity of airflow distribution horizontally and vertically in the under-floor DC, this study has innovatively proposed a locally active optimization method applying the jet fan at the entrance of cold aisles and built the CASS without changing the original air supply structure. The optimization of jet fan with different parameters based on four factors is
studied using the numerical simulation validated with the on-site experimental results. The conclusions can be summarized as follows:

1. The airflow distribution and thermal environment of the DC can be improved significantly by applying the jet fan in the cold aisles. The jet fan makes up for the lower flow rate at the front racks, and enhances the uniformity of the overall thermal environment in DC. Meanwhile, the movement track of the cold air from the perforated tiles to the rack is bettered, the more uniform cold air delivery in the cold aisles is achieved.

2. In the CASS, under the condition that the load of the servers, the total cooling capacity of CRAC units and the layout arrangement of the cabinets are constant, the maximum temperature of the cabinets is reduced by 6-9°C and the overheated area is shrunk by about 90%. Moreover, 83% of the servers is 25-28°C without overheating area, and the security performance of IT equipment is enhanced significantly due to decreased surface temperature heterogeneity.

3. Through the numerical simulation to investigate the effect of different inductive velocity, nozzle height, horizontal distance and attachment distance on thermal performance of the DC, the results recommend that the optimal parameter range of 2.0-3.0 m/s (inductive velocity), 1.4-1.6 m (nozzle height), 2#-3# cabinets (horizontal distance), and 0.05-0.1 m (attachment distance).

4. By locally regulating and controlling the airflow path in the cold aisles in the CASS with the optimal parameters, the inlet temperatures of the front racks are significantly reduced, the temperature difference of the inlet/outlet airflow is controlled within 3°C in both horizontal and vertical directions, which improves the utilization efficiency of cooling capacity and realizes the airflow distribution uniformity in DC.

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NOMENCLATURE
| Symbol | Description |
|--------|-------------|
| DC     | Data center |
| SAT    | Supply air temperature (°C) |
| CRAC   | Computer room air conditioning |
| CACS   | Cold aisle containment system |
| HACS   | Hot aisle containment system |
| UASS   | Under-floor air supply system |
| CASS   | Coupling air supply system |
| $g$    | the gravitational acceleration vector (m/s²) |
| $\mu$  | the velocity vectors (m/s) |

S Volumetric heat generation (W/m³)
ρ Density (kg/m³)
$\kappa_{eff}$ the effective thermal conductivity (W/m K)
$C_p$ constant pressure specific heat (kJ/kg.K)
$\Delta T$ Temperature difference (°C)
$X_c$ X-axis coordinate value of jet fan center (m)
$V_i$ Inductive velocity of jet fan (m/s)
$H_n$ Nozzle height of jet fan (m)
$D_h$ Horizontal distance of jet fan (m)
$D_a$ Attachment distance between the jet fan and the rack (m)
$L$ Length (m)
$W$ Width (m)
$H$ Height (m)
$T$ Temperature (°C)
$T_{max}$ Maximum temperature (°C)
$T_{min}$ Minimum temperature (°C)
$T_{in}$ Inlet temperature (°C)
$T_{out}$ Outlet temperature (°C)
$T_{av}$ Average temperature (°C)

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