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Use CFD to Design The Cold Air Outlet

Abstract: In this paper, the air conditioning design problem of the data center is studied. When determining the cabinet layout, the change process of air flow in the machine room caused by different air outlet positions of air conditioners is analyzed.Based on the theory of fluid mechanics and the conservation idea, a 0-1 dual objective programming model is established to determine the air outlet position of the air conditioner which makes the average temperature in the machine room lowest and reduces local overheating.

First of all, in order to determine the velocity of air outlet in each branch pipe, based on the assumption that the main pipe injected with cool air is a one-dimensional compressible and unstable flow system, this paper introduced the mass energy equation and the equation of state characteristics to describe the pressure fluctuation process, and then analyzed and obtained the velocity, pressure distribution and flow rate of air outlet in each branch pipe.

Secondly, in view of the air distribution within the data center machine room, this paper introduced the turbulence model, meshing room, set up and one variable, analysis on the change of each branch number and location of outlets change influence on room air flow, and then compare the average temperature of the engine rooms, cabinets outlet temperature and hot air into the draught, the average temperature is lower, the smaller the difference in temperature, shows that the smaller the convection of cold air into the room, air conditioning refrigeration efficiency is higher.At the same time, it is concluded that the number of air outlets on each branch pipe is 3, and the coordinate of air outlet of air conditioner is as follows:

		Pipe number		
		1	2	3
Outlet number	1	(0.35,2.43)	(4.2,2.43)	(8.3,2.43)
	2	(0.35,3)	(4.2,3)	(8.3,3)
	3	(0.35,4.2)	(4.2,4.25)	(8.3,4.25)

Finally, 6sigmaRoom software was used to analyze the sensitivity of parameters.The higher the wind speed, the lower the temperature.However, with the increase of wind speed, its cooling effect on air distribution decreases.At the same time, when the space between cabinets is within the range of 1.5 2.5 times the width of cabinets, the airflow distribution in the machine room is relatively uniform, with good air return effect and heat dissipation effect.When the height of the machine room is close to 3m, the heat dissipation effect is the best.

Keywords: data center turbulence model Dual objective programming

Contents

I. Introduction

As the heat flux density of the server increases, the problem of heat dissipation in the equipment room needs to be solved. The traditional data room adopts the method of lowering the room temperature[?], which makes the mixing of hot and cold air serious and the air conditioning efficiency is low. Therefore, the design of the air conditioning system is crucial for the power consumption of the data room. Nowadays, many small computer rooms still use the air supply mode of the upper side, and the cold air enters the machine room through the air supply pipe installed on the ceiling. Since the air temperature is lower than the indoor air, the density is relatively large and deposited. At the bottom of the room, an air lake is formed. At the same time, the IT equipment in the data center is constantly dissipating heat and acts as a heat source in the equipment room. When encountering a heat source such as equipment, the air is heated up to form a hot plume and act as the dominant airflow for indoor air flow, thereby transferring heat. Take the return air to the side of the room.[?]。In order to reduce the mixing of hot and cold air, the incoming cold air is usually separated from the extracted hot air to form a hot and cold passage.

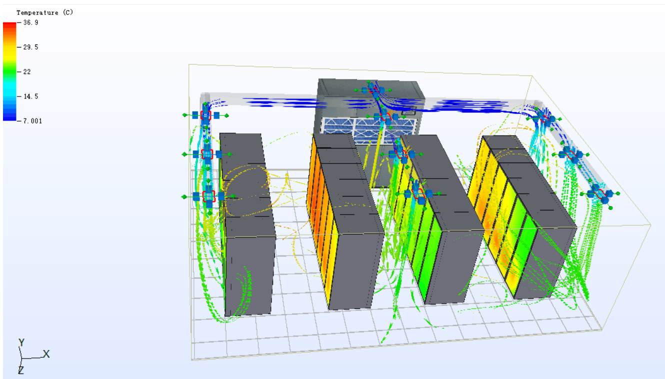

Figure 1 a sketch of Data center airflow

In a confined space, the height of the ceiling and the unit of the machine room and the heat output of the unit have a certain influence on the temperature and speed of the hot and cold air. When these parameters are determined, our goal is to change the vent and layout to reduce convection and reduce the amount of wasted cooling in corridors and crevices to improve cold air utilization, while room temperature meets standards.

II. General Assumptions and Variable Description

2.1 General Assumptions

1. garRedless of the influence of air leakage, the airtightness of the equipment room is considered to be good;
2. Does not consider the influence of pipes, cables and brackets in the equipment room on indoor airflow organization;
3. Air flow in the equipment room is considered in terms of incompressible viscous fluid;

4. Ignore the effects of surface radiation on the cabinet in the equipment room;
5. The air flow in the data center room is turbulent;
6. Ignore the influence of human body heat dissipation and lighting devices on the cooling load of the engine room;

2.2 Variable Description

Symbol	Definition
C	Flow Coefficient
A	Cross-sectional area
Q_{out}^{i}	Flow rate of the i exhaust port of the exhaust pipe
v_{ij}	Flow rate of the j exhaust port of the i exhaust pipe
P_{out}^{i}	Air pressure of the i exhaust port of the exhaust pipe
T_{ij}	Temperature at various points in the equipment room
t_a	Cabinet air outlet temperature
t_b	Heat container inlet temperature
P_0	Indoor atmospheric pressure
ρ_0	Cold air density of exhaust pipe
τ	Viscous stress
Φ	Dissipation coefficient
m	Gas quality
S_h	Air heat source
x_i	Coordinate of the i exhaust pipe in the x direction
y_{ij}	Coordinate of the j exhaust vent of the i exhaust pipe in the Y direction
$Width$	Data center room width
$Heigh$	Data center room heigh
$Length$	Data center room Length

III. Model preparation

IV. Models

4.1 Energy consumption model

Firstly, the energy consumption of the equipment room is analyzed. At a certain moment, in order to maintain a stable temperature and humidity of the room, the amount of cold that needs to be supplied to the room air is called a cold load. The cooling load of the data center is designed in accordance with GB50174 "Design Specification for Electronic Information System Room". According to the regulations, the cold load of the air conditioning system in the computer room in summer should include the heat dissipation of the equipment in the equipment room, the heat transfer of the building envelope structure, the heat dissipation of the human body, the heat dissipation of the lighting device, and the solar radiation heat entering through the outer window.

The cold load formed by heat transfer in the equipment room accounts for about $90 \sim 95\%$of the intercooling load, and the electrical power of the equipment is basically converted into heat dissipation, which is generally97%or more. Knowing the electrical power of the device, then

$$Q_S = X_1 X_2 N_1 \tag{1}$$

Where$Q_S$is the equipment cooling load;$X_1$is the load factor, generally$0.7 \sim 1.0$; X_2 is the simultaneous use coefficient, generally $0.9 \sim 1.0$; N_1 is the electrical power of the equipment.

The air conditioning load of the enclosure structure refers to the air conditioning cooling load caused by indoor and outdoor temperature difference and solar radiation, and the cold load formed by the heat transferred into the room through the building envelope. Consider the effect of heat transfer from the building envelope on the cooling load of the machine room:

$$Q_2 = AK(\Delta t) \tag{2}$$

In the formula, $A$is the outer wall area,$K$is the heat transfer coefficient of the outer wall, and the surrounding structure adopts the more common brick-concrete structure and thermal insulation wall, where the value is$2(W/m^2K)$, Δt is indoor External temperature difference.

Considering that the heat dissipation of the human body and the lighting device have little effect on the cooling load of the machine room, it is neglected in this calculation (may be assumed). With N_2cabinets, the total cooling load of the equipment room is:

$$Q_t = N_2 Q_s + Q_2 \tag{3}$$

Therefore, in order to make the data center room room have stable temperature and humidity at a certain moment, the total cooling capacity of the air conditioner is at least greater than the total cooling load of the equipment room:

$$Q_g \geq Q_t \tag{4}$$

4.2 One dimensional compressible unsteady flow model

4.2.1 *Fundamental equation of fluid mechanics*

When the position of the exhaust pipe is fixed, its capacity is a fixed value. Therefore, according to the theoretical knowledge of fluid mechanics, there are three basic equations[2].

Mass conservation equation: The amount of fluid mass reduction in the control body should be equal to the fluid mass flowing out of the control body.

$$\frac{\partial \rho A}{\partial t} + \frac{\partial uA}{\partial x} = 0 \tag{5}$$

Momentum equation: The increment of the momentum of the control body is equal to the impulse of the external force acting on the control body, that is, the hydrodynamics expression of the momentum theorem.

$$\frac{\partial \rho uA}{\partial t} + \frac{\partial PuA}{\partial x} + \frac{\partial PA}{\partial x} + 2\tau\rho uA = 0 \tag{6}$$

Physical property equation: The amount of fluid mass reduction in the control body should be equal to the fluid mass flowing out of the control body.

$$\frac{\partial \rho A}{\partial t} + \frac{a^2 \partial \rho u A}{\partial x} = 0 \tag{7}$$

4.2.2 *Flow Rate-Density Equation*

When the main pipeline transportation volume per unit time is constant, the number and position of the exhaust ports affect the wind speed of the air outlet, thereby affecting the entire airflow organization of the equipment room. The flow rate$v_{ij}$is equal to the flow rate $q^i_{out}$to the time$t$, and the flow rate $q^i_{out}$of the exhaust port is a function of pressure [1] so it has the following:

$$q^i_{out} = C_b A_b \sqrt{\frac{2(P^i_{out} - P_0)}{\rho}} \tag{8}$$

Where$C_b$is the flow coefficient,$A_b$is the area of the exhaust port,$P^i_{out}$is the air pressure of the$i$exhaust port of the exhaust pipe, and $P_0$is the indoor air pressure.

The amount of change in pressure of the exhaust pipe gas and the amount of change in density are proportional to the ratio ofE/ρ

$$\Delta P = P_{ij} - P_0 = \frac{E}{\rho} * \Delta\rho \tag{9}$$

4.2.3 *Derivation of density versus position*

When the number of exhaust pipes is determined, the volume remains unchanged, and the zero coordinate is used as the mass source. The material of$\Sigma m$is emitted in each unit time Σt, and the material propagates at various directions in the direction of the branch pipe at the wave velocity v. Assuming that the pipe is cut on average by several intercepting faces, the material is just from one section to the next. Differentiating with Σm, since each surface always seeks to have no pressure difference with its neighbors, the surface with more quality will be transferred to the lower part of the mass at the next moment. Therefore, in the $3\Sigma t$ time, the previous three intercepting surfaces S are taken as an example, and the amount of substances contained in each surface is:

$$\begin{cases} \Delta t : f(S_1) = \Delta m; f(S_2) = 0; f(S_3) = 0 \\ 2\Delta t : f(S_1) = \Delta m + \frac{1}{2}\Delta m; f(S_2) = \frac{1}{2}\Delta m; f(S_3) = 0 \\ 3\Delta t : f(S_1) = \Delta m + \frac{3}{4}\Delta m; f(S_2) = \Delta m; f(S_3) = \frac{1}{4}\Delta m \end{cases} \tag{10}$$

By mathematical induction, the mass of the material contained in the closure surface at position x at a certain time t can be obtained as follows:

$$f(x,t) = \begin{cases} 1 + \frac{1}{2} + \frac{1}{2^2} + \cdots + \frac{1}{2^{t-1}}, x\text{Be the first position} \\ \frac{1}{2} f(x-1, t-1) + \frac{1}{4} f(x-1, t-2), x\text{Not the first position} \end{cases} \tag{11}$$

When the cross-sectional area is constant, the formula can be regarded as a derivation formula of the relationship of density with position. Since flow is a function of density, this formula can be thought of as the relationship of flow as a function of position.

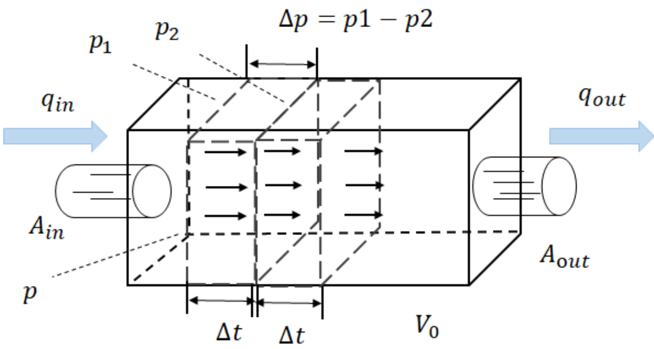

Figure 2 Schematic diagram of liquid flow

4.2.4 *The influence of the variation of the number of exhaust ports on the flow velocity*

ccording to the mass conservation equation, the mass of cold air flowing into the exhaust pipe is equal to the sum of the cold air flowing out of each exhaust port:

$$m_{out} = m_{in} \tag{12}$$

$$\begin{cases} \frac{dm_{in}}{dt} = \rho_a A_a v_{in} \\ \frac{dm_{out}}{dt} = \sum_{i=1}^{3} \rho_b A_b v_{out}^i \end{cases} \tag{13}$$

Where ρ_a a represents the density of the cold air entering the exhaust pipe, and A_a a represents the cross-sectional area of the exhaust pipe.

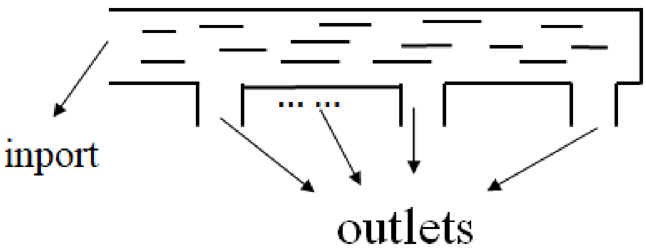

Figure 3 Structure drawing of single air pipeline

4.3 Turbulence model

4.3.1 *Control equation*

Because the airtightness of the equipment room is good, the engine room can be seen as a whole, and the indoor gas movement belongs to the turbulence model. The total mass, total momentum, and total energy of the internal fluid should remain unchanged, that is, the continuity equation, the momentum equation, and the kinetic energy equation are satisfied.

Continuity equation: The rate of increase in fluid quality entering the engine room is equal to the mass of the net inflow. Since the incoming cold gas mass is equal to the hot gas mass, the net inflow is 0, where U is the velocity vector.

$$\frac{\partial \rho}{\partial \mathrm{t}} + div(\rho U) = 0 \tag{14}$$

Momentum conservation equation: The flow of any fluid in the equipment room should follow the law of conservation of momentum, and the momentum increase of the fluid particles should be equal to the sum of all external forces applied to the fluid particles.

$$x\text{分量式} \qquad \frac{\partial(\rho u)}{\partial t} + div(\rho u U) = -\frac{\partial p}{\partial y} + \frac{\partial_{xx}}{\partial x} + \frac{\partial_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + F_x \tag{15}$$

$$y\text{分量式} \qquad \frac{\partial(\rho v)}{\partial t} + div(\rho u U) = -\frac{\partial p}{\partial y} + \frac{\partial_{xy}}{\partial x} + \frac{\partial_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + F_x \tag{16}$$

$$z\text{分量式} \qquad \frac{\partial(\rho w)}{\partial t} + div(\rho u U) = -\frac{\partial p}{\partial y} + \frac{\partial_{xz}}{\partial x} + \frac{\partial_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + F_x \tag{17}$$

Where p is the pressure on the fluid micro-body, F_n is the force on the micro-body, and τ is the viscous stress.

Energy conservation equation: The energy equation is derived from the first law of thermodynamics, defined as: the energy increase rate of a fluid particle is equal to the sum of the heating rate of the fluid particles and the power of the convective particles. The sum of internal energy, kinetic energy and gravitational potential energy is the total energy E.

$$\frac{\partial T}{\partial t} + div(UT) = \mathrm{div}(\frac{\lambda}{\rho c_p} gradT) + \frac{S_h + \Phi}{\rho} \tag{18}$$

Where λ is the thermal conductivity of air,c_p is the specific heat capacity, T is the thermodynamic temperature, S_h is the heat source in the air, and Φ is the dissipation function.

4.3.2 *RNG* $k - \varepsilon$***model***

The airflow in the data center room is a large spatial flow model, and the door, window and wall in the equipment room are well airtight, which belongs to the turbulence calculation model, so the $RNG\ k - \varepsilon$ model is adopted. The model has been widely used in the field of HVAC, such as the simulation analysis of indoor airflow organization. Since the turbulent flow kinetic energy k and the viscous dissipation rate ε can be used to express the turbulent flow viscosity, it is only necessary to calculate the turbulent kinetic energy k,and the flow dissipation rate ε in the $RNG\ k - \varepsilon$ model, namely:

$$\rho \frac{\partial k}{\partial t} + \rho u_i \frac{\partial k}{\partial x_i} = \frac{\partial}{\partial x_j}(\alpha_k \mu_{eff} \frac{\partial k}{\partial x_j}) + G_k + G_b - \rho\varepsilon + S_k \tag{19}$$

$$\rho \frac{\partial \varepsilon}{\partial t} + \rho u_i \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_j}(\alpha_k \mu_{eff} \frac{\partial k}{\partial x_j}) + C_{1\varepsilon} \frac{\varepsilon}{k}(G_k + G_{3\varepsilon} G_b) - C_{2\varepsilon}\rho \frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon \tag{20}$$

$$R_\varepsilon = \frac{G_\mu \rho \eta^3 (1 - \eta/\eta_0)}{1 + \beta\eta^3} \frac{\varepsilon^2}{k} \tag{21}$$

Where G_k is the turbulent flow energy produced by the laminar velocity gradient, G_b is the turbulent flow energy generated by buoyancy, $C_1\varepsilon, C_2\varepsilon, C_3\varepsilon$ is the empirical constant, and u_i is the component of the velocity in the i direction. $\alpha_k, \alpha_\varepsilon$ is the turbulent Prandtl number of the $k - \varepsilon$ equation, S_k, S_ε are user-defined source terms, ,$\eta = S_k/\varepsilon, \eta_0 = 4.38, \beta = 0.012$。

General formula: For the above-mentioned continuity equation, momentum equation, energy equation, and $RNG\ k-\varepsilon$ model, the following equation can be used:

$$\rho div(u\phi) = div(\Gamma grad\phi) + S \tag{22}$$

Where ϕ represents the general dependent variable $(u, v, w, T, \varepsilon, \phi)$,$\Gamma$represents the generalized diffusion coefficient, and S represents the generalized source term.

4.4 Selection of exhaust port position

In a continuous environment, the position of the exhaust port is infinitely varied, but the small displacement of the exhaust port is small for the air flow. To simplify the calculation, we divide the ceiling into M columns $\times N$ rows of mesh. It can be seen that M exhaust pipes are installed on the ceiling. The exhaust pipe number is $i \in [1, 2, \cdots, M]$, we can only choose three or four to discharge cold air; each exhaust pipe There are N exhaust ports, the exhaust port number is $j \in [1, 2, \cdots, N]$. In the selected exhaust pipe, we can only select $1 \sim 3$ exhaust ports to discharge cold air, and close the rest. Unselected exhaust and exhaust ports. We can list the following equations:

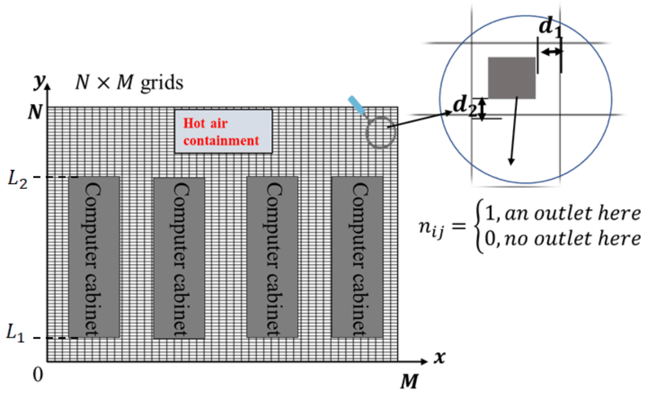

Figure 4 grid

Three or four of the M exhaust pipes are arbitrarily selected and used as a duct for conveying cold air. Its position in the x direction should be within the scope of the machine room.

$$k_i = \begin{cases} 1, \text{Pppe } i \text{ is selected} \\ 0, \text{Pipe } i \text{ is not selected} \end{cases} \tag{23}$$

$$3 \le \sum_{i=1}^{M} k_i \le 4 \tag{24}$$

$$0 \le x_{ij} \le Width \tag{25}$$

No more than three of the N exhaust ports on the exhaust pipe are selected as the outlet for discharging the cold air. In order to reduce the amount of wasted air in the gaps and corridors, the position of the exhaust port in the y direction should be within the length of the unit.

$$n_{ij} = \begin{cases} 1, \text{The outlet } j \text{ on pipe } i \text{ is selected} \\ 0, \text{The outlet } j \text{ on pipe } i \text{ is not selected} \end{cases} \tag{26}$$

$$\sum_{j=1}^{N} n_{ij} \leq 3, i = 1, 2, \&, M \tag{27}$$

$$L_1 \leq y_{ij} \leq L_2 \tag{28}$$

$$y_{i1} = y_{i2} = \cdots = y_{ij}, j \in [0, N] \tag{29}$$

Our goal is to change the position of the exhaust port so that the average temperature in the room is the lowest. In order to reduce the local overheating, we compare the temperature of the air outlet of the cabinet with the temperature of the air inlet of the heat container. The smaller the difference, the hot air and the cold air. The less convection. Double objective equation

In summary, the final model is:

$$\min \overline{T} = \sum_{i=1}^{N} \sum_{j=1}^{M} \frac{T_{ij}}{M \cdot N} \tag{30}$$

$$\min g(v) = |t_a - t_b| \tag{31}$$

$$s.t. \begin{cases} Q_t = N2 \cdot Q_s + Q_2 \\ \Delta P = P_{ij} - P_0 = \frac{E}{\rho} * \Delta\rho \\ q^i_{out} = C_b A_b \sqrt{\frac{2(P^i_{out} - P_0)}{\rho}} \\ m_{out} = m_{in} \\ \frac{dm_{in}}{dt} = \rho_a A_a v_{in} \\ \frac{dm_{out}}{dt} = \sum\limits_{i=1}^{3} \rho_b A_b v^i_{out} \\ \rho div(u\phi) = div(\Gamma grad\phi) + S \\ 3 \leq \sum\limits_{i=1}^{M} k_i \leq 4 \\ \sum\limits_{j=1}^{N} n_{ij} \leq 3 \\ L_1 \leq y_{ij} \leq L_2 \\ y_{i1} = y_{i2} = \cdots = y_{ij}, j \in [0, N] \\ 0 \leq x_i \leq Width \end{cases} \tag{32}$$

V. The optimal layout of exhaust port is solved

5.1 The solution of turbulence model

FiniteVolumeMethod Solving the turbulent model Finite Volume Method,Finite Difference Method，Finite Analytic Method and Finite Element Method.Because the room is airtight container, abide by the three basic equations of fluid mechanics, so we choose the Finite Volume Method to solve the turbulent motion of the gas inside the room. We first divided the machine room space into grids, each grid was represented by a node, and then integrated the control volume with the control equation, and finally derived its discrete equation

Integral equation

$$\oint \rho\phi v \cdot dA = \oint \Gamma_\phi \nabla\phi \cdot dA + \oint S_\phi dV \tag{33}$$

离散方程

$$a_P \phi = \sum_{nb} a_{nb} \phi_{nb} + b \tag{34}$$

Where, nb represents the adjacent grid, a_P and a_{nb} are ϕ,ϕ_{nb} coefficient respectively, and b is the source term

Computer room airflow simulation based on 6sigmaRoom

- Step1:create a room shape. We set the size of the machine room as 6.2m*8.7m*3m.
- Step2:Set up the rack. We set up 4 cabinets in the room, 6 cabinets in each column, and 20 server equipment of 2U600W in each cabinet.

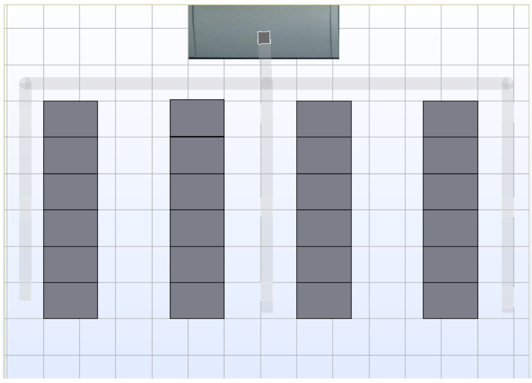

Figure 5 the position of computer cabinet

- Step3:establish the cooling system. The machine room was cooled by installing air conditioning pipes and heat absorbers. We separated the 4 columns of the machine room according to the interval of cold and hot channels. Above the cold channel, we built air conditioning pipes, with a total of three air ducts, each with a different number of air outlets
- Step4: establish other peripheral facilities and buildings. In addition, we have also established UPS and other necessary devices for the data center, which we have not yet shown in this figure
- Step5:simulation of the results. After 220 iterations, the airflow reaches a stable state and the temperature field of the machine room is obtained.

5.2 Narrow down the selection of outlet position

In the model of exhaust port location selection, the grid method is adopted to reduce the calculation amount, but the calculation amount is still huge.If the unit grid area is increased, the accuracy and accuracy of the position of the exhaust port will be reduced. Therefore, we further reduce the calculation amount by designing hot and cold channels and narrowing the range of the exhaust port.For the design of the hot and cold passage in the organic room, there are three layout methods, and we choose the layout method of figure 6 and three exhaust pipes to study the choice of exhaust port location.Therefore, one exhaust pipe corresponds to one cool air channel, and the exhaust pipe can only change its position laterally within the cool air channel.In addition, due to the symmetry of the layout in the direction of x, the number of exhaust outlets in the first and third channels should be equal.

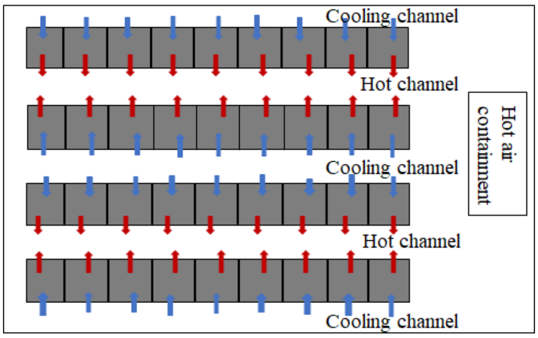

Figure 6 3+2 way

5.3 Matlab analysis of heat diagram

The study of indoor average temperature is also one of our goals, but 6*sigmaRoom* software can only measure the temperature of a certain point through observation point, with low accuracy. Therefore, we used *Matlab* to analyze the thermal map and get the accurate indoor mean temperature.

- step1: read image pixels.
- step2: set a corresponding temperature for each pixel value, and divide the pixel value by the number of pixels to get the average temperature.
- step3:compare the average temperature corresponding to each position and find the optimal position.

5.4 Effect of room height change

The simulated condition is: the air supply temperature must be 7 °C, and the air supply temperature is 2m cubed /s.When the building height is 2.8m,3m,3.2m,3.4m and 3.6m, there are five different working conditions.

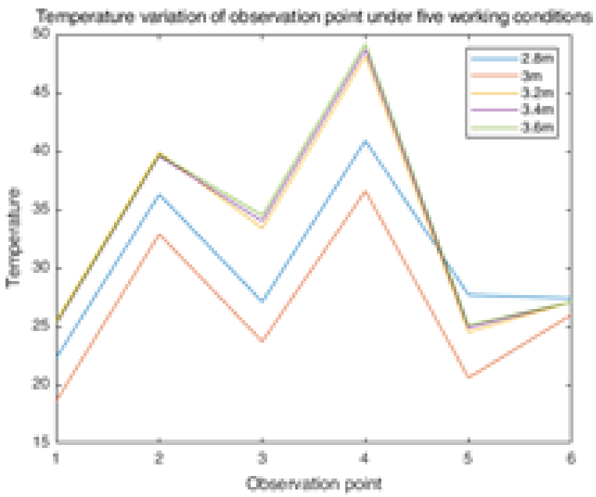

Figure 7 Effect of room height change

The ceiling height works the same way.

5.5 Result

After setting the size of the machine room (6.2m*8.7m*3m) and other parameters, Matlab and 6sigmaRoom were used to calculate that the average temperature and the maximum local overheating temperature could reach the minimum value when three vents were opened on all three exhaust pipes. The average temperature was and the maximum local overheating temperature was.The specific positions of these vents are shown in the table.The thermal map shows that there is an obvious hot and cold air passage with uniform temperature distribution and no obvious local overheating.

		Pipe number		
		1	2	3
Outlet number	1	(0.35,2.43)	(4.2,2.43)	(8.3,2.43)
	2	(0.35,3)	(4.2,3)	(8.3,3)
	3	(0.35,4.2)	(4.2,4.25)	(8.3,4.25)

Figure 8 the optimal choice

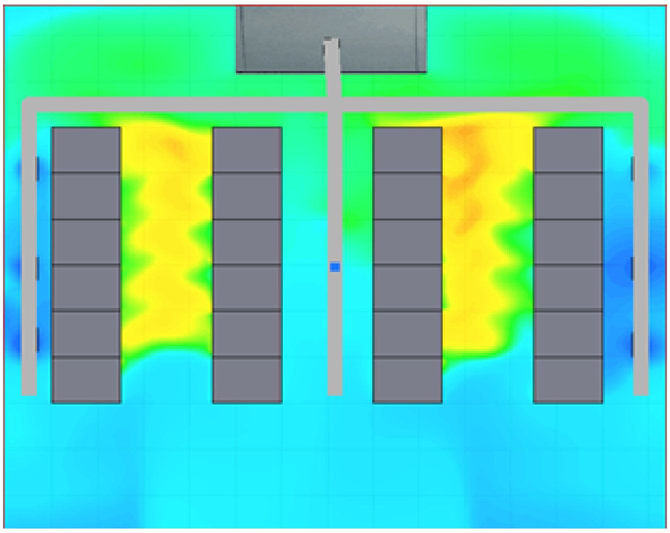

Figure 9 temperature field of optimal choice

VI. Sensitivity analysis

The characteristics of airflow distribution in the computer room of the data center were analyzed, and numerical simulation was carried out through software 6sigmaRoom, boundary conditions were set and network division was calculated. By changing the spacing of the cabinet, air temperature and air volume, the influence on the distribution of air distribution is obtained.

6.1 Machine room layout

We mainly take 6sigmaRoom software as the experimental platform for research. This machine room is mainly composed of cabinet, air conditioning system, air supply pipeline and air return pipeline. The machine room size is 6.2m*8.7m*3m, the size of each column of cabinet is 6*0.6m*90m*2.05m, the spacing of cabinet is 1.2m, and the height of cabinet is 2.05m. At the same time, the air supply is carried out in the way of upper and side return. Meanwhile, the cabinet is of 40kw, and the cooling capacity must be of 120kw.

In the software 6sigmaRoom, we took a plane screenshot with the height of 1m to simulate the temperature field of airflow distribution, and set the positions of 6 observation points as shown in figure to check the temperature of the outlet of the server.

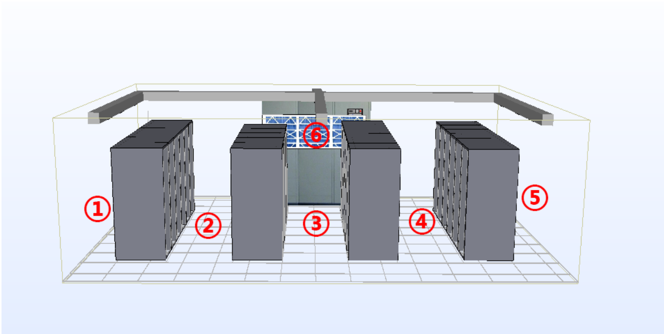

Figure 10 Observation position

6.2 Influence of air supply temperature change on air distribution temperature

In 6sigmaRoom software, we set the air supply capacity as 2m cubed /s, the cooling capacity as 120kw, and the cabinet power as 40kw. When the air supply temperature is changed to 7 °C, 9 °C, 11 °C, 13 °C and 15 °C, respectively, five different working conditions are obtained, as shown in the figure after simulation analysis. We found that the higher the air supply temperature, the higher the air distribution temperature. The temperature field:

6.3 The influence of air supply change on air distribution temperature

Similarly, we set the air supply temperature to be 7 °C, and changed the five different conditions when the air supply was 1m cubed /s, 1.5m cubed /s, 2m cubed /s, 2.5m cubed /s and 3m cubed /s respectively. The figure is obtained after simulation analysis. It is found that the higher the wind speed is, the lower the air distribution temperature is. However, with the increase of the wind speed, its cooling effect on air distribution is less.

6.4 Influence of the change of cabinet spacing on airflow distribution in machine room

The change of the transverse space between the cabinets directly affects the convection and heat transfer and airflow distribution between the two cabinets. The air supply temperature is set

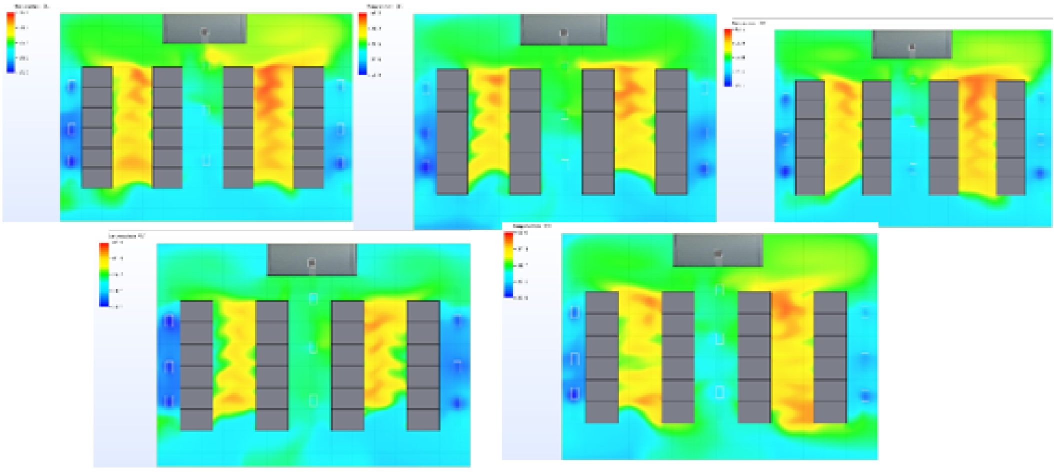

Figure 11 The temperature field

Observation point temperature value under different working conditions （℃）						
Observation point location	1	2	3	4	5	6
Supply air temperature 7℃	15.3	27.2	20.5	31.2	17.2	25
Supply air temperature 9℃	17.3	29.2	22.5	33.2	19.2	27
Supply air temperature 11℃	19.8	31.2	24.5	35.2	21.2	29.7
Supply air temperature 13℃	21.1	34.2	27.2	37.1	22.4	31.7
Supply air temperature 15℃	23.1	37.2	35.2	42.3	25.1	33.9

as 7 °Cand the air supply temperature is 2m cubed /s. When the cabinet spacing was changed to 1.2m,2.05m and 2.9m, the figure was obtained after simulation analysis. It was found that the farther the cabinet spacing was, the smaller the interference between them would be, and the closer it was to the return air port, which was conducive to free return air and better heat dissipation. In the machine room of this experiment, when the space between cabinets is within the range of 1.5 2.5 times of the

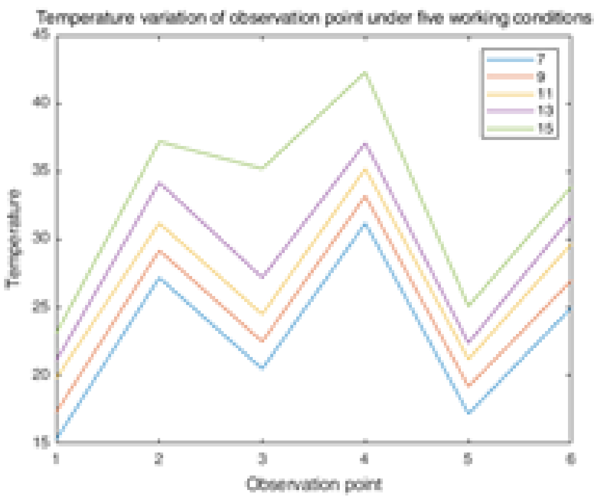

Figure 12

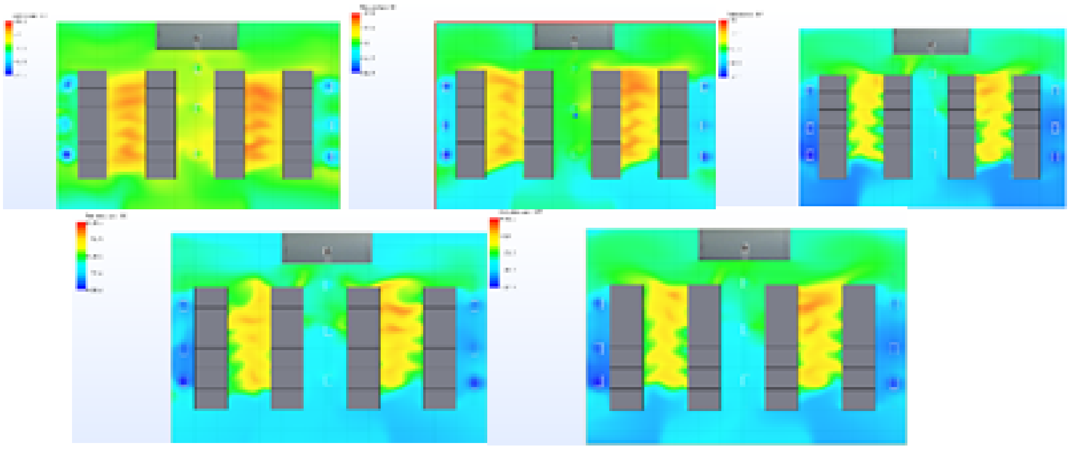

Figure 13 temperature field

Observation point temperature value under different working conditions（℃）						
Observation point location	1	2	3	4	5	6
Air supply 1m³/s	39.5	54.1	47.6	62.3	39.6	46.8
Air supply 1.5m³/s	26.5	41.1	34.1	48.3	26.8	33.8
Air supply 2m³/s	22.3	36.3	27.1	40.9	27.7	27.4
Air supply 2.5m³/s	16.5	30.5	22.4	35.3	17.1	23.1
Air supply 3m³/s	16.8	30.4	21.8	33.9	17.6	22.9

width of cabinets, the airflow distribution in the machine room is relatively uniform, the air return effect is good, and good heat dissipation effect can be achieved.

VII. References

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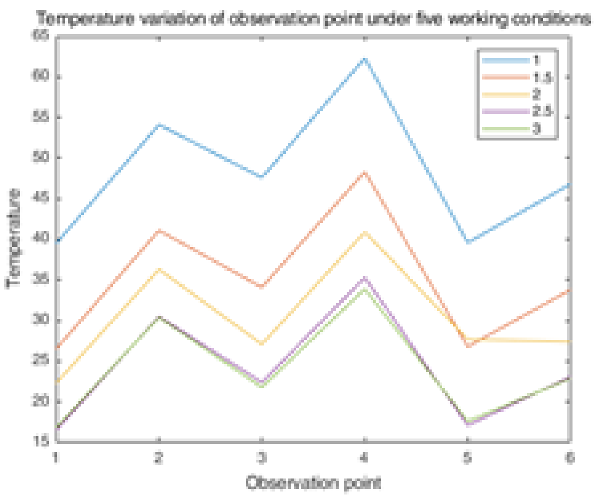

Figure 14 The curve of

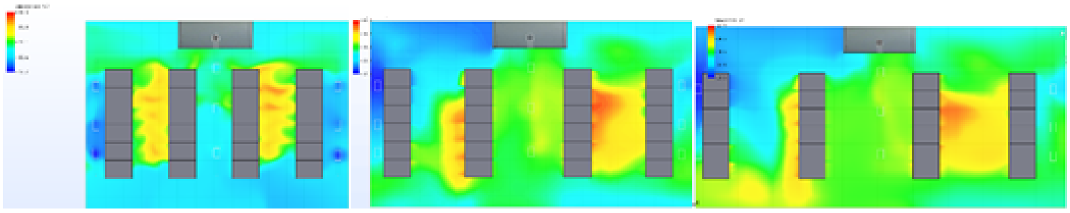

Figure 15

Observation point temperature value under different working conditions (℃)						
Observation point location	1	2	3	4	5	6
distance 1.2m	22.3	36.3	27.1	40.9	27.7	27.4
distance 2.05m	19.4	33.8	32.5	37.3	24.3	27.1
distance 2.9m	18.1	32.9	31.2	46	25.3	27.1

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VIII. Appendix

Listing 1: The matlab Source code of Algorithm

```
clc
clear
m =
    [15.3000000000000,27.2000000000000,20.5000000000000,31.2000000000000,17.2000000000000,25;17.300000000000
plot(m')
legend('7','9','11','13','15')
title('Temperature variation of observation point under five working conditions')
xlabel('Observation point')
ylabel('Temperature')
n =
    [39.5000000000000,54.1000000000000,47.6000000000000,62.3000000000000,39.6000000000000,46.8000000000000;
figure
plot(n')
legend('1','1.5','2','2.5','3')
title('Temperature variation of observation point under five working conditions')
xlabel('Observation point')
ylabel('Temperature')
s =
    [22.3000000000000,36.3000000000000,27.1000000000000,40.9000000000000,27.7000000000000,27.4000000000000;
figure
plot(s')
legend('2.8m','3m','3.2m','3.4m','3.6m')
title('Temperature variation of observation point under five working conditions')
xlabel('Observation point')
ylabel('Temperature')
q =
    [22.3000000000000,36.3000000000000,27.1000000000000,40.9000000000000,27.7000000000000,27.4000000000000;
plot(q')
legend('1.2m','2.05m','2.9m')
title('Temperature variation of observation point under five working conditions')
xlabel('Observation point')
ylabel('Temperature')
p =
    [25.3000000000000,38.5000000000000,26.8000000000000,39.3000000000000,22.7000000000000;24.8000000000000,
plot(p')
legend('2','3','4','5','6','7')
title('Temperature variation of observation point under five working conditions')
xlabel('Observation point')
ylabel('Temperature')
```