INTRODUCTION

The energy demands of data centers are growing rapidly. Currently, less than half of the total power consumption of data centers is consumed by their cooling systems. High integration increases the power consumption of thermal design. The thermal management system becomes the key factor affecting the safety and economic operation of the entire system. Approximately, 55% of the damages in electronic equipment are caused by inadequate thermal management. The continuous demand for improved computing performance has resulted in the increment of system power, which requires a more efficient cooling solution than conventional air cooling. Therefore, indirect liquid cooling which can cool the high-power components or entire computer systems selectively is attracting significant attentions.

A server cabinet can accommodate multiple servers. The cooling loop is generally distributed in parallel. According to the fluid mechanics, the flow is not uniform along the flow direction in the manifold of cabinet, which may lead to insufficient cooling of individual servers and can significantly reduce the performance of the liquid cooling system. Therefore, improving the uniformity distribution of cooling media is essential to increase the cooling efficiency and decrease the operating cost.

The flow distribution of single-phase parallel flow has been studied by numerous researchers. These studies are mainly divided into three categories. The first category is comprised...
of theoretical research. Bajura et al\textsuperscript{5} proposed a momentum equation for U-type and Z-type parallel systems. However, it consisted of several parameters and complicated solutions. Wang et al\textsuperscript{6–8} considered the effects of friction and inertia and achieved the analytic solution for velocity and pressure distribution of U-type and Z-type parallel systems. The second category is the numerical simulations. Huang et al\textsuperscript{9} used the nonlinear least-square method to study the fluid distribution of parallel pipeline under the Z-type piping structure. As a result, the nonuniformity of flow observed in the system can be minimized and the flow rate of tube in the heat exchanger is nearly uniform. Tong et al\textsuperscript{10} achieved the uniform distribution of fluid in the pipe by changing the geometric parameters. In this specific method, the diameter of the main pipe was reduced and the variations in the streamline inside the pipe were observed. Solovitz et al\textsuperscript{11} proposed an analytical model that predicted the optimal manifold shape to produce uniform velocities by examining the pressure drops through each flow path in a multi-channel cooling system. Mohammadi et al,\textsuperscript{12} Manikanda et al,\textsuperscript{13} Xia et al,\textsuperscript{14} and Liu et al\textsuperscript{15} used numerical methods to study the coolant flow distribution of parallel microchannel in similar electronic components and discussed the effects of Reynolds number and structure size on the flow distribution. The third category is consisted of the experimental research. Wang et al\textsuperscript{16,17} investigated the influences of inlet flow condition, tube diameter, header size, area ratio, and flow directions of the single-phase flow in parallel flow heat exchangers experimentally and numerically. Lu et al\textsuperscript{18} proposed a discrete model of the real physical phenomenon to predict the pressure distribution in headers. The model was validated through experimental results under specific conditions.

However, most previous designs have focused on the effects of structure size on the flow distribution. Some design methods of manifold do not rely on mathematical methods. These optimization methods can get a structure of better flow distribution but not the optimal one and less research studied the flow distribution under turbulent conditions. Hence, an optimization method from the mathematical perspective will have substantial significance for designing the ideal uniform flow distribution of the parallel pipe network.

In this paper, an optimization method for designing the manifold to obtain the ideal uniform flow for each branch is proposed. This method is suitable for solving a specific supply manifold structure under a fixed flow rate. Subsequently, the method is validated through numerical simulation. Finally, the factors influencing the accuracy of the optimization method are discussed.

2 PHYSICAL MODEL

The model consists of five-layer standard 2 U servers as shown in Figure 1. U is the unit representing the external height of the server, which is short for unit, and 1 U = 44.45 mm. The amount of cooling water required for each server is 0.017 kg/s and the inlet flow of supply manifold is 0.085 kg/s. The water flows into the rack through the supply manifold, passes through five parallel equidistantly distributed branches, and finally leaves the rack through the return manifold. The distance between the supply and return manifold \( L \) is 450 mm. The distance between adjacent branch pipes \( \delta H \) is fixed as 100 mm (slightly higher than the height of the 2 U server).

The five branch pipes are named channels 1-5 from the bottom to the top. The supply manifold is divided into five independent sections. Each of them has length and diameter denoted by \( H_i \) and \( d_i \) (\( i \) represents the server number from bottom to top, \( i = 1, 2, 3, 4, 5 \)), respectively, where \( H_1 = 150 \text{ mm} \), \( H_2 = H_3 = H_4 = 100 \text{ mm} \), \( H_5 = 53 \text{ mm} \).

3 MATHEMATICAL MODEL

To simplify the problem, the diameters of the branch pipe and return manifold are selected as 6 mm and 15 mm, respectively. The fluid is single phase and incompressible under isothermal steady conditions. For velocity and static pressure, the average values are considered.

Figure 2 presents the sketch of pressure distribution of the pipe network. Considering the inlet of manifold as the reference point, \( P_{i,2i–1} \) and \( P_{i,2i} \) are the pressures at the two junctions of the \( i \)-th branch pipe in the supply manifold, \( P_{h,2i–1} \) and \( P_{h,2i} \) are the pressures at the two junctions of the \( i \)-th branch pipe of the return manifold.

The pressure decrease in the manifold is mainly caused by the friction loss, local loss, and the lateral flow through the branch.

FIGURE 1 Sketch of the flow network of liquid cooled server
The friction loss $\delta P_f$ is caused by the frictional effects, which can be defined as:

$$\delta P_f = \frac{\lambda l}{d} \frac{\rho v^2}{2}$$  \hspace{1cm} (1)

where $\lambda$ is the friction factor along the pipe, $l$ is the flow length, $d$ is the diameter of pipe, $\rho$ is the fluid density, and $v$ is the mean velocity.

The local loss $\delta P_l$ is caused by additional geometric variations such as contractions or expansions. It is defined as:

$$\delta P_l = \zeta \frac{\rho v^2}{2}$$  \hspace{1cm} (2)

where $\zeta$ is the loss coefficient depending on the geometry.

For the sudden expansion,\(^{19}\) the loss coefficient is:

$$\zeta = \left( \frac{A_2}{A_1} - 1 \right)^2$$  \hspace{1cm} (3)

whereas for the sudden contraction,\(^{19}\) the local loss of sudden contraction is related to the ratio of front to back area ratio and flow velocity. To simplify the calculation, this paper uses the engineering empirical formula, which is:

$$\zeta = 0.5 \left( 1 - \frac{A_2}{A_1} \right)$$  \hspace{1cm} (4)

where $A_1$ and $A_2$ are the cross-sectional areas before and after the pipeline contraction, respectively.

The pressure variation by the lateral flow through the branch $\delta P_r$ is defined as:

$$\delta P_r = K \frac{W_a^2 - W_b^2}{2}$$  \hspace{1cm} (5)

where $K$ is the static pressure variation coefficient of manifold, which considers the effects of the losses caused by the lateral flow through the branch pipes, $W_a$ and $W_b$ are the velocities of main flow before and after the branch pipe, respectively.

Subsequently, according to the pressure variation between two adjacent points, the pressure distribution of the manifolds is as follows:

$$P_{f,2i} = P_{f,2i-1} - \delta P_f + \delta P_r + d_{ch} \rho g$$

$$= P_{f,2i-1} - \frac{\rho}{2} \left( \frac{d_{ch}}{d_i} \frac{(Q_i + Q_{i+1})}{2} \right)^2$$

$$+ K_f \frac{\rho}{2} \left( \frac{d_{ch}}{d_i} \frac{(Q_i^2 - Q_{i+1}^2)}{2} + d_{ch} \rho g \right)$$  \hspace{1cm} (6)

where $\delta P_f$ denotes the friction loss in the pipe, $\delta P_r$ denotes the pressure variation through the branch pipe, and $d_{ch} \rho g$ denotes the hydrostatic pressure differences, $d_{ch}$ is the diameter of branch pipe, $Q_i$ is the flow of supply manifold before entering the $i$-th branch pipe and $K_f$ are the static pressure variation coefficient of the supply manifold.

$$P_{f,2i+1} = P_{f,2i} - \delta P_f - \delta P_r + (\delta H - d_{ch}) \rho g$$

$$= P_{f,2i} - \frac{\rho}{2} \left( \frac{\lambda \delta H - d_{ch}}{d_i} \frac{(\zeta + \zeta)}{2} \right) Q_{i+1}^2$$

$$+ \frac{\rho}{2} \left( \frac{\lambda \delta H - d_{ch}}{d_{i+1}} \frac{(\zeta + \zeta)}{2} \right) Q_{i+1}^2 + (\delta H - d_{ch}) \rho g$$  \hspace{1cm} (7)

where $\delta P_f$ denotes the friction loss in the pipe, $\delta P_l$ denotes the pressure variation through the geometric variations, and $(\delta H - d_{ch}) \rho g$ denotes the hydrostatic pressure differences.

$$P_{b,2i} = P_{b,2i-1} + \delta P_f + \delta P_r + d_{ch} \rho g$$

$$= P_{b,2i-1} + \frac{\rho}{2} \left( \frac{d_{ch}}{D} \frac{(Q_i + Q_{i+1})}{2} \right)^2$$

$$+ K_b \frac{\rho}{2} \left( \frac{d_{ch}}{D} \frac{(Q_i^2 - Q_{i+1}^2)}{2} + d_{ch} \rho g \right)$$  \hspace{1cm} (8)

where $\delta P_f$ denotes the friction loss in the pipe, $\delta P_r$ denotes the pressure variation through the branch pipe, $d_{ch} \rho g$ denotes the hydrostatic pressure differences, and $K_b$ is the static pressure variation coefficient of the return manifold.
where \( \delta P_f \) denotes the friction loss in the pipe and \( (\delta H - d_{ch})/\rho g \) denotes the hydrostatic pressure differences.

### 4 | BOUNDARY CONDITIONS AND SOLVING ALGORITHM

#### 4.1 | Boundary conditions

The flow rate across each branch channel is governed by the pressure drop across the channel, which is proportional to the pressure drop. In this paper, the premise is that the fluid flow of each branch is the same. Therefore, the pressure drop across the channels is equal. When the flow resistance is distributed uniformly, the pressure difference between the two ends of each branch pipe is constant, which is described as:

\[
\frac{P_{h,2i} + P_{h,2i-1}}{2} - \frac{P_{h,2i} + P_{h,2i-1}}{2} = \Delta P
\]

where \( (P_{2i+1} + P_{2i-1})/2 \) and \( (P_{h,2i} + P_{h,2i-1})/2 \) are the average pressures at the \( i \)-th branch pipe inlet and outlet, respectively.

The friction factor along the path \( \lambda \) is calculated by the common friction factor expressions shown in Table 1. Moreover, Table 1 shows the common forms for this parameter in different conditions as well as the approximate ranges of validity in terms of Reynolds number. In this paper, the calculation of friction factor in the transitional flow interval is simplified to using the empirical formula of turbulent flow under transitional flow. Here, \( \text{Re} = \rho v_d/\mu \), where \( \mu \) is the fluid viscosity.

The static pressure variation coefficients determine the pressure across the branch. In this case, \( K_f \) is set to 0.8 and \( K_h \) is set to 1.8. \(^{20,21}\)

#### 4.2 | Solving algorithm

According to Equations (6)-(10), the diameter of each section of the supply manifold can be calculated by its recursive relationship under the conditions of known related parameters. As shown in Figure 3, the solution steps are as follows:

1. Assuming the outlet pressure \( P_{h,0} = 0 \), according to Equations (8) and (9), the pressure distribution in the return manifold is obtained through known parameters.
2. According to the pressure drop \( \Delta P \) in a single server, the pressure relationship in the supply manifold \( (P_{f,2i} + P_{f,2i-1})/2 \) is obtained.
3. Assuming \( d_5 = 15 \text{ mm} \), according to \( (P_{f,2i} + P_{f,2i-1})/2 \), Equations (8) and (9), calculate the diameter of each section of the supply manifold step by step.

#### 4.3 | Evaluation criteria

The flow distribution is evaluated by the presentation of the standard deviation \( S \), which is defined as:

\[
S = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} \left( \frac{Q_{i,ch} - Q_{ave}}{Q_{ave}} \right)^2}
\]
where \( N \) is the number of branches, \( Q_{ch,i} \) is the flow of fluid through the \( i \)-th branch, and \( Q_{ave} \) is the average flow through all branches.

5 | RESULTS AND DISCUSSIONS

5.1 | Validation

To validate the proposed method, the numerical model of flow network using the finite element method is developed. The analysis is performed using steady-state solver with incompressible flow. The fluid flow conditions are governed by the conservation laws of mass and momentum. These governing equations are solved using pressure-based solver.

Water is chosen as the working fluid. Simulations were performed under steady-state, incompressible, and isothermal condition without heat transfer. The inlet flow rate is 0.085 kg/s and the velocity is 0.481 m/s. The Reynolds number of the inlet is 7215. The Reynolds number of the channel is 3607.

Renormalization-Group \( k-\varepsilon \) turbulence model is selected and the second-order upwind scheme is adopted for the control equation discrete scheme. Furthermore, the pressure coupling equation semi-implicit method (SIMPLE algorithm) is selected for the flow field calculation. The outlet adopts pressure outlet boundary condition. The residual of convergence is set to \( 10^{-4} \).

Figure 4 shows the grid of computational model. Boundary layer grids are generated near the wall. Unstructured grids are generated the rest. Figure 5 provides the results of mesh independence verification. It shows that the error of the pressure drop from inlet to outlet is less than 1% if the quantity of mesh is more than 1.4 million.

Substituting the boundary conditions in Section 4.1 into Equations (6)-(9), the solving algorithm in Section 4.2 can be used to obtain Case 1 in Table 2. The validity of the method is assessed by comparing the flow distribution between Case 0 and Case 1.

Case 0 and Case 1 were simulated by the finite element method. Figure 6 gives the flow distribution of five branch pipes of Case 0 and Case 1. The minimum flow of branch pipes for Case 1 is greater than that of original Case 0. For a parallel cooling system, the highest temperature is generally observed on the branch with the minimum flow. Therefore,

![FIGURE 4](image_url) The grid of computational model

![FIGURE 5](image_url) Curve of inlet and outlet pressure drop with the grid number (\( \Delta P \) is the pressure drop from inlet to outlet)

![FIGURE 6](image_url) Flow distribution of five branch pipes of Case 0 and Case 1 (the ordinate represents the ratio of branch pipe flow to average flow)

| Case number | The diameter of supply manifold (mm) |
|-------------|-------------------------------------|
| 0           | \( d_1 = d_2 = d_3 = d_4 = d_5 = 15 \) |
| 1           | \( d_1 = 10.144, d_2 = 10.626, d_3 = 11.189, d_4 = 12.012, d_5 = 15 \) |

![TABLE 2](image_url) Model structure
According to the numerical simulation results of the initial model (the diameters of supply and return manifold are the same, 15 mm), Method 3 is obtained by the inverse problem method. Seven working conditions are calculated. The mass flow rate of supply manifold is 0.050 kg/s, 0.060 kg/s, 0.070 kg/s, 0.080 kg/s, 0.085 kg/s, 0.090 kg/s, and 0.100 kg/s. The fitting curve is shown in Figures 8 and 9. The fitting formulation is represented as:

\[
K_f = -2.2652 \times \left(1 - \frac{W_b}{W_a}\right) + 0.8506 \times \left(1 - \frac{W_b}{W_a}\right)^2 + 0.9198, \\
K_h = 17.58 \times \left(1 - \frac{W_b}{W_a}\right)^{-2.8901} \times \left(1 - \frac{W_b}{W_a}\right) + 1.9060
\]

In order to reduce the potential interference of the last branch \((W_b = 0)\) to the fitting curve, the value of \(K\) when

\[
\begin{align*}
K_f & = -2.2652 \times \left(1 - \frac{W_b}{W_a}\right) + 0.8506 \times \left(1 - \frac{W_b}{W_a}\right)^2 + 0.9198, \\
K_h & = 17.58 \times \left(1 - \frac{W_b}{W_a}\right)^{-2.8901} \times \left(1 - \frac{W_b}{W_a}\right) + 1.9060
\end{align*}
\]

5.2 Effects of the static pressure variation coefficient on the model

\(K_f\) and \(K_h\) determine the pressure variation across the branch. The values of \(K_f\) and \(K_h\) are related to the velocity before and after passing through the branch pipe. The model considering \(K_f\) and \(K_h\) as the fixed values in Section 4.1 may cause errors. In the following three methods, the static pressure variation coefficient is regarded as the function of velocity in the manifold.

In Method 1, on basis of the analysis of the momentum conservation equations of microcontrol volume, the formula of local resistance coefficient of the straight pipe in three-way pipe\(^{22,23}\) was obtained. The formulation is

\[
K_f = 1 - \frac{0.35 \left(1 - \frac{W_b}{W_a}\right)}{\left(1 + \frac{W_b}{W_a}\right)}, \quad K_h = 1 + \frac{0.55W_a + W_b}{W_a + W_b} \tag{12}
\]

where \(W_a\) and \(W_b\) are the velocities of main flow before and after the branch pipe, respectively.

Method 2 is based on the experimental data fitting.\(^{24}\) The formulation is

\[
K_f = 2 \times \left(0.57 + 0.15 \frac{W_b}{W_a}\right), \quad K_h = 2 \times \left(0.98 + 0.17 \frac{W_a}{W_b}\right) \tag{13}
\]

Figure 7 provides the standard deviation \(S\) of Case 0 and Case 1. The standard deviation \(S\) of Case 1 is approximately 18.99% as that of Case 0. It can be seen that the flow distribution can be improved by the method presented in this paper. However, the flow distribution of Case 1 is not an ideal flow distribution because of the errors in the calculation process. The forthcoming sections discuss the effect of the static pressure variation coefficient, local loss, and friction factor on the accuracy of the method.
Wb = 0 is calculated separately. The average values of Kf and Kh under five working conditions are taken as the value of Kf and Kh. The average values are represented as:

\[ K_f = 0.5701, \quad K_h = 4.9247 \]  \hspace{1cm} (15)

The fitting steps are as follows: (a) the flow distribution and pressure distribution of the pipe network are obtained by numerical simulation; (b) the above results are substituted into Equations (6)-(9) to calculate the values of Kf and Kh for inlet and outlet of each branch pipe; (c) the value formulas of Kf and Kh are obtained by fitting the value of Kf and Kh of each outlet and inlet and the variables \( 1 - W_b/W_a \)/(1 + W_b/W_a).

Table 3 presents the three new cases calculated by using the above three methods. In order to study the effect of the static pressure variation coefficient on the model, the flow distribution of three new models and the basic model is compared.

Figure 10 provides the flow distributions of five branch pipes of Case 0 to 4. The minimum flow rates of branch pipes of Cases 1, 2, 3, and 4 are greater than those of Case 0. Figure 11 shows the standard deviations of all Cases. The standard deviations of Cases 1, 2, 3, and 4 are smaller than that of Case 0. The standard deviations of Cases 1, 2, 3, and 4 are 18.99%, 46.92%, 59.80%, and 32.08% respectively. It can be seen that the flow distributions of all Cases using the calculation methods of the static pressure variation coefficient are better than that of Case 0.

Further, it can be seen from Figure 11 that the standard deviation of Cases 1, 2, 3, and 4 are significantly different, which shows that the values of Kf and Kh have a substantial influence on the calculation results. The flow distribution of Case 1 is optimal, which is contrary to predictions. Because the static pressure variation coefficients of Case 4 are relatively closer to the actual situation. It indicates that there are calculation errors caused by such other parameters as the local loss and the friction loss in the previous calculation process except for the static pressure variation coefficient.

### 5.3 Effects of the local loss on the model

In order to investigate the influence of local loss on this method, this section presents the comparisons of two cases

| Case number | The diameter of supply manifold (mm) | The value of Kf and Kh |
|-------------|------------------------------------|-----------------------|
| 5           | d1 = 10.253, d2 = 10.741, d3 = 11.325, d4 = 12.210, d5 = 15 | Kf = 0.8, Kh = 1.8    |
| 6           | d1 = 10.851, d2 = 11.269, d3 = 11.765, d4 = 12.531, d5 = 15 | Method 1              |
| 7           | d1 = 11.595, d2 = 11.918, d3 = 12.305, d4 = 12.925, d5 = 15 | Method 2              |
| 8           | d1 = 10.313, d2 = 10.548, d3 = 10.703, d4 = 10.952, d5 = 15 | Method 3              |

**TABLE 3** Model structure

**TABLE 4** Model structure
with and without considering the local loss. Table 4 presents Cases 5-8, which calculated by the same boundary conditions and algorithms in Section 4.1. In this calculation, the local loss in the Equations (6)-(9) is neglected, that is, \( \zeta = 0 \).

Figures 12 and 13 compare the flow distribution and the standard deviation of Case 1 to 8, respectively. The differences in the branch flow of Case 1 and Case 5 are small. For the other calculation methods of \( K_f \) and \( K_h \), the difference is also insignificant. It can be seen from Figure 13 that the difference of standard deviation is negligible between the conditions of with and without considering the local loss. Therefore, local loss has slight effect on the method because that the local loss is significantly smaller than the frictional loss.

### 5.4 Effects of the friction factor on the model

In the calculation process discussed Sections 5.1 to 5.3, the common friction factor expressions in Table 1 were used for the calculation of the friction factor along the path. Owing to the small spaces between the branch pipes, the flow in the

![FIGURE 12](image1.png)

(A) Flow distribution of five branch pipes of Case 1 and Case 5

![FIGURE 12](image2.png)

(B) Flow distribution of five branch pipes of Case 2 and Case 6

![FIGURE 12](image3.png)

(C) Flow distribution of five branch pipes of Case 3 and Case 7

![FIGURE 12](image4.png)

(D) Flow distribution of five branch pipes of Case 4 and Case 8

![FIGURE 13](image5.png)

The standard deviation of Case 1 to 8

| Range of Re | Friction factor |
|-------------|-----------------|
| \( Re < 2000 \) | \( \lambda = \frac{27.948}{Re} + 0.2055 \) |
| \( 2000 \leq Re \) | \( \lambda = \frac{2.8965}{Re} - 0.2689 \) |

**TABLE 5** Fitting formula of friction factor along the supply manifold
manifold is not fully developed. Therefore, employing the common friction factor expressions may cause errors. To investigate the effects of the friction factor on the model, the friction factor along the path under turbulent conditions is fitted according to the simulation results of Case 0. Because of the different flow conditions in supply and return manifold, two fitting formulas are obtained. The transition Reynolds number of the supply manifold is 2000. The transition Reynolds number of the return manifold is 1400. The empirical formula of turbulent flow is used to calculate the friction factor. The following fitting formulas are obtained according to the form of the Blasius formula as shown in Tables 5 and 6. The fitting curves are shown in Figures 14 and 15.

In Section 5.2, the Method 3 is obtained by the inverse problem method. In the calculation process, the empirical formulas of Table 1 are used to calculate the frictional loss. A new Method 4 is obtained by the fitting formulas in Tables 5 and 6. The fitting curves are shown in Figures 16 and 17. The fitting formulations are represented as:

\[
K_f = -3.1423 \left( \frac{1 - \frac{W_b}{W_a}}{1 + \frac{W_b}{W_a}} \right)^2 + 1.3392 \left( \frac{1 - \frac{W_b}{W_a}}{1 + \frac{W_b}{W_a}} \right) + 0.8868, \\
K_h = 19.1150 \left( \frac{1 - \frac{W_b}{W_a}}{1 + \frac{W_b}{W_a}} \right)^2 - 3.7410 \left( \frac{1 - \frac{W_b}{W_a}}{1 + \frac{W_b}{W_a}} \right) + 1.9547
\] (16)

TABLE 6 Fitting formula of friction factor along the return manifold

| Range of Re | Friction factor |
|-------------|----------------|
| Re < 1400   | \( \lambda = \frac{323.1}{Re} - 0.2257 \) |
| 1400 \( \leq \) Re | \( \lambda = \frac{2.1296}{Re^{0.25}} - 0.1643 \) |

FIGURE 14 (A) Linear fitting results of \( \lambda \) under laminar flow in the supply manifold. (B) Linear fitting results of \( \lambda \) under turbulent flow in the supply manifold.

FIGURE 15 (A) Linear fitting results of \( \lambda \) under laminar flow in the return manifold. (B) Linear fitting results of \( \lambda \) under turbulent flow in the return manifold.
Taking five working conditions when \( W_b = 0 \) as examples, the average values of \( K_f \) and \( K_h \) are represented as:

\[
K_f = 0.5843, \quad K_h = 4.9836
\]  

(17)

Table 7 provides the structures of Case 9 and Case 10. In this calculation, the friction loss in the Equations (6)-(9) is calculated by the formula of friction factor in Tables 5 and 6.

Figure 18 compares the flow distribution and the standard deviation of Case 1 and Case 9. Figure 19 compares those of Case 4 and Case 10. As shown in Figures 18 and 19, the flow distribution of Case 10 is improved than that of Case 4. The result is in accordance with the prediction because the calculation process of Case 10 uses the friction factor closer to the actual situation. However, the flow distribution of Case 9 is inferior than that of Case 1. This indicates that some other errors are also incurring in the calculation process of Case 1 and Case 9. Because the local loss has an insignificant effect on the result, the errors are caused by the static pressure variation coefficient. This explains the optimal flow distribution of Case 1 in Section 5.1. There exists a large error in the static pressure variation coefficient and friction loss, which makes the result accidentally optimal.

### Table 7  Model structure

| Case number | The diameter of supply manifold (mm)          | The value of \( K_f \) and \( K_h \) |
|-------------|-----------------------------------------------|-------------------------------------|
| 9           | \( d_1 = 8.659, \ d_2 = 8.655, \ d_3 = 9.072, \ d_4 = 10.442, \ d_5 = 15 \) | \( K_f = 0.8 \), \( K_h = 1.8 \) |
| 10          | \( d_1 = 8.904, \ d_2 = 8.766, \ d_3 = 8.853, \ d_4 = 9.489, \ d_5 = 15 \) | Method 4                           |
Therefore, the friction factor has a significant influence on the results. When the calculation of static pressure variation coefficient is accurate, the model flow distribution calculated by the friction factor of the fitting formula is improved.

6 | CONCLUSIONS

In this paper, an optimization method for designing the manifold to obtain the uniform flow in each branch is proposed. To calculate the diameter distribution of the manifold, the pressure distribution model of the flow network is developed by applying the principle of the equal pressure drop for each branch. Subsequently, the proposed model is validated by finite element method under the turbulence condition. In order to improve the accuracy of the model, the effect of the static pressure variation coefficient, local loss, and frictional loss on the accuracy of the model are studied. Four methods for determining the static pressure variation coefficient are compared. Three of them are based on the existing literature and the rest one is obtained from the numerical simulation.

The values of the static pressure variation coefficient have a significant influence on the calculation results. The study shows that local loss has slight effect on the results. This is mainly because that the local loss is substantially smaller than the frictional loss. The friction loss has a significant influence on the results. The results show that the model calculated with the fitting formula is improved when the calculation of the static pressure variation coefficient is accurate. Consequently, an improved design method of supply manifolds in liquid cooling server cabinet can be achieved to ensure uniform flow distribution when the total flow is constant.

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NOMENCLATURE

- \( A_1 \): cross-sectional area before pipeline contraction
- \( A_2 \): cross-sectional area after pipeline contraction
- \( D \): return manifold diameter
- \( d \): characteristic length
- \( d_i \): diameter of a part of the supply manifold
- \( d_{ch} \): branch pipe diameter
- \( g \): gravitational acceleration
- \( H \): length of the return manifold
- \( H_i \): length of a part of the supply manifold
- \( \delta H \): distance between adjacent branch pipes
- \( i \): the number of the servers
- \( K \): static pressure variation coefficient
- \( K_{f,i} \): static pressure variation coefficient of supply manifold
- \( K_{h} \): static pressure variation coefficient of return manifold
- \( L \): distance between the supply and return manifold
- \( l \): length of flow
- \( P_{f,j} \): static pressure at a point in the supply manifold
- \( P_{h,i} \): static pressure at a point in the return manifold
- \( \delta P_f \): friction loss
- \( \delta P_l \): local loss
- \( \delta P_r \): pressure variation by the lateral flow through the branch
- \( \Delta P \): pressure drop between the inlet and outlet of branch pipe
- \( \delta P \): pressure drop between the inlet and outlet of manifold
- \( Q_i \): flow of supply manifold before entering the \( i \)-th branch pipe
- \( Q_{ch,i} \): flow of the \( i \)-th branch
- \( Q_{ave} \): average flow of all branches
- \( S \): standard deviation
flow velocity

$W_a$ velocity of main flow before the branch pipe

$W_b$ velocity of main flow after the branch pipe

$\lambda$ friction factor

$\rho$ fluid density

$\zeta$ local loss coefficient

$\mu$ fluid viscosity

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