

Article

Research on Design and Optimization of Economic Operation for Indirect Liquid Cooling System in Data Center Servers

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Abstract

With the rapid development of data centers, significant energy consumption challenges have emerged, with cooling system energy consumption accounting for over 30%. Traditional air cooling, limited by airflow organization issues, struggles to meet the cooling demands of high heat flux density chips. Although liquid cooling technology exhibits superior cooling performance, it often leads to high system power consumption due to design and flow matching factors. Therefore, conducting energy-saving optimization of liquid cooling systems holds significant importance. This paper establishes a piping network model for a cabinet-level indirect liquid cooling system, incorporating the heat flow method and piping network fluid dynamics–resistance balance relationships to establish overall system flow and heat transfer constraints. Based on this, optimization analyses are conducted for cabinet liquid cooling systems under centralized and distributed pump configurations. For centralized pump configurations with a constant thermal load, a Lagrangian function is established to minimize system power consumption, and the optimal pump operating frequency is determined using variational principles. When the cooling water temperature rises from 20 °C to 24 °C, the total power consumption increases by 1.55 times. Placing a server with a specific load of 1.2 kW at the bottom of the cabinet rather than the top results in a 34.4% energy savings. With a constant total pump power consumption, a Lagrangian function is established to maximize the system thermal load, and the optimal pump operating frequency is determined. When the cooling water inlet temperature increases by 2 °C, the total thermal load decreases by 4.9%. Servers with higher thermal loads should be placed nearby to make the cooling system more energy-efficient. Comparisons reveal that as the total system thermal load increases from 4.0 kW to 6.0 kW, the distributed pump configuration achieves an average energy savings of 2.5 W, with a maximum savings of 7.09 W, compared to the centralized pump configuration.



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Keywords: indirect liquid cooling system; pump configuration; heat flow method; optimal parameters

1. Introduction

1.1. Background of Data Center Thermal Management

With the rapid development of the digital economy, data centers have become a core component of new infrastructure, and their green development has become a consensus across society. Improving data center energy efficiency and reducing the Power Usage Effectiveness (PUE) have become inevitable trends in industry development. Typical data

center equipment includes IT equipment, temperature control equipment, and power supply and distribution systems. To address the growing demand for computing power and the limited carrying capacity of data centers, increasing the power density of individual cabinets has become a key solution.

Consequently, high power densities have brought unprecedented cooling and energy consumption challenges. Approximately 2% of global electricity is used for data centers, and this figure is growing at an annual rate of 12% [1]. In 2015, China's data centers consumed approximately 100 billion kW·h of electricity annually, accounting for over 1.5% [2] of the country's total electricity consumption, with 43% used for IT equipment cooling, nearly equal to the IT equipment's own energy consumption (45%). Therefore, reducing cooling power consumption is crucial for controlling data center operational costs and achieving green development.

1.1.1. Current Status of Air Cooling in Data Centers

Air cooling systems, as a common thermal management technology in data centers, provide appropriate airflow velocities and temperatures from the chip level to the computer room level [2]. Typical air cooling systems [3] adopt a hot/cold aisle isolation layout, achieving air circulation through raised floor ventilation systems. The main issues with air cooling technology include hot air recirculation, cold air bypass, and uneven airflow temperatures, leading to high server failure rates and unstable operation. Research indicates that optimizing raised floor configurations, hot/cold aisle arrangements, and server-heat exchanger layouts [4] can improve cooling performance by 50% and reduce temperature variability by 60%.

In recent years, Cho et al. [5] proposed an improved cooling system based on rows and computer rooms, verifying the advantages of completely enclosed hot/cold aisles. Hiroshi et al. [6] developed a new air-cooled container data center, utilizing waste heat recovery to reduce overall energy consumption by 20.8%. Song et al. [7] investigated open and raised floor data centers using fan-assisted cooling, verifying that fan placement design significantly affects cooling performance.

Although air cooling technology offers the advantages of low initial investment and simple structure, its cooling effectiveness is limited and struggles to meet the cooling demands of high heat flux servers.

1.1.2. Current Status of Liquid Cooling in Data Centers

Liquid cooling technology uses liquid as a coolant instead of air and is divided into direct and indirect cooling methods. Direct cooling primarily involves immersion liquid cooling, which is further categorized into phase-change and non-phase-change methods based on whether the coolant undergoes a phase change [8–10].

Immersion liquid cooling technology achieves efficient cooling by fully immersing servers in an insulating coolant. Matsuoka et al. [10] proposed a natural convection immersion cooling technology that can reduce PUE to below 1.04. The Sugon Silicon Cube liquid-cooled phase-change cooling computer achieves a cooling power density of 160 kW per cabinet, 4–5 times that of traditional data centers, with a PUE as low as 1.04.

Indirect liquid cooling technology installs cold plate heat exchangers on high heat flux components within servers, connecting internal and external cooling loops through a Coolant Distribution Unit (CDU). This technology has lower requirements for coolant management and higher commercialization levels [11].

Ovaska et al.'s [12] tests on a 632-blade node cluster showed that liquid cooling solutions can reduce energy consumption by up to 14.4%. Iyengar et al.'s [13] experimental results indicated that liquid cooling systems use only 3.5% of their power for cooling, saving over 90% of energy compared to traditional systems.

1.1.3. Current Status of Energy-Saving Optimization in Data Centers

To improve cooling efficiency and reduce energy consumption, researchers have conducted optimization work from multiple perspectives: (1) System-level optimization: Mo et al. [14] used CFD to study three cooling architectures and found that row- and rack-based architectures can reduce cooling power consumption by 29%. Chen et al. [15] proposed a hybrid cooling system combining spray cooling, waste heat reuse, and chilled water storage, achieving 51% energy efficiency savings and 71% operational cost savings. Xiong et al. [16,17] proposed a steam bypass solution for the outdoor heat exchanger of multi-split air-source heat pumps, aiming to enhance heat transfer rates and achieve simultaneous optimization of cooling and heating performance. (2) Operational parameter optimization: Rubenstein et al. [18] proposed a warm water cooling system, reducing cooling infrastructure energy consumption by 50%. (3) Intelligent control strategies: Li et al. [19] proposed the SmartPlace intelligent server placement algorithm, reducing total power consumption by 26.7%.

Although these studies have achieved significant results, most focus on energy consumption economics, cooling equipment layout, control processes, and waste heat reuse [20,21], with relatively few studies directly optimizing cooling system operational parameters.

1.2. System Modeling and Optimization Methods Based on the Heat Flow Method

System modeling is a crucial prerequisite for conducting optimization research. Han et al. [22] studied the impact of data center air conditioning system operational parameters based on the on-demand cooling concept. However, due to complex models, numerous constraint equations, and parameter coupling, the solution process is complex.

Chen et al. [23] derived an explicit expression for heat exchanger thermal resistance from the physical meaning of fire accumulation dissipation, establishing a direct relationship between design objectives and design parameters, and proposed the heat flow method [24]. This method converts complex heat transfer problems into circuit problems using the thermoelectric analogy principle, providing a new approach for system modeling and optimization. Zhao et al. [25] analyzed the impact of optimal heat conductivity of heat exchangers on system performance based on the heat flow method. Wang et al. [26] proposed a collaborative optimization method incorporating overall heat transfer and flow constraints for heat exchange systems. Hao et al. [27] established an overall heat flow model for thermal–electric integrated energy systems, further expanding the application scope of this method.

1.3. The Main Work in This Paper

To alleviate the cooling and high energy consumption issues of data center servers, this paper conducts research on the design and economic operation optimization of indirect liquid cooling systems for data center servers. The main contributions include:

(1) Piping Network Design and Resistance Calculation for Cabinet Liquid Cooling Systems

Establish a physical model for a cabinet-level indirect liquid cooling system, and accurately calculate the resistance coefficients of components such as elbows, sudden expansions, sudden contractions, tees, and valves using empirical formulas and theoretical derivations, analyzing the piping network resistance characteristics. This model provides a theoretical basis for subsequent flow constraint establishment and optimization design.

(2) Modeling of Cabinet Liquid Cooling Systems

Based on the physical model of the system, establish overall heat transfer constraints for the cooling system using the heat flow method, and combine them with fluid dynamics–resistance balance relationships in the flow model to establish overall flow constraints for the system. Construct an optimization model for the liquid cooling system with the goal of minimizing pump power consumption, and obtain optimal system operational parameters through numerical solutions.

(3) Economic Operation Optimization and Analysis of Cabinet Liquid Cooling Systems

Based on overall flow and heat transfer constraints, establish optimization models for liquid cooling systems under centralized and distributed pump configurations, quantitatively comparing the energy-saving effects of the two pump configurations. Study the impact of regulating cabinet liquid cooling system branch thermal loads, cooling characteristic temperatures, and cooling water inlet temperatures on the system's cooling performance.

2. Analysis of Piping Network Resistance Characteristics in Server Cabinet Indirect Liquid Cooling Systems

2.1. Introduction to Cabinet-Level Liquid Cooling Systems and Piping Network Design

2.1.1. Cabinet-Level Liquid Cooling Systems

The primary research object of this paper is liquid cooling systems, as shown in Figure 1, which establishes a framework diagram for a new liquid cooling system.

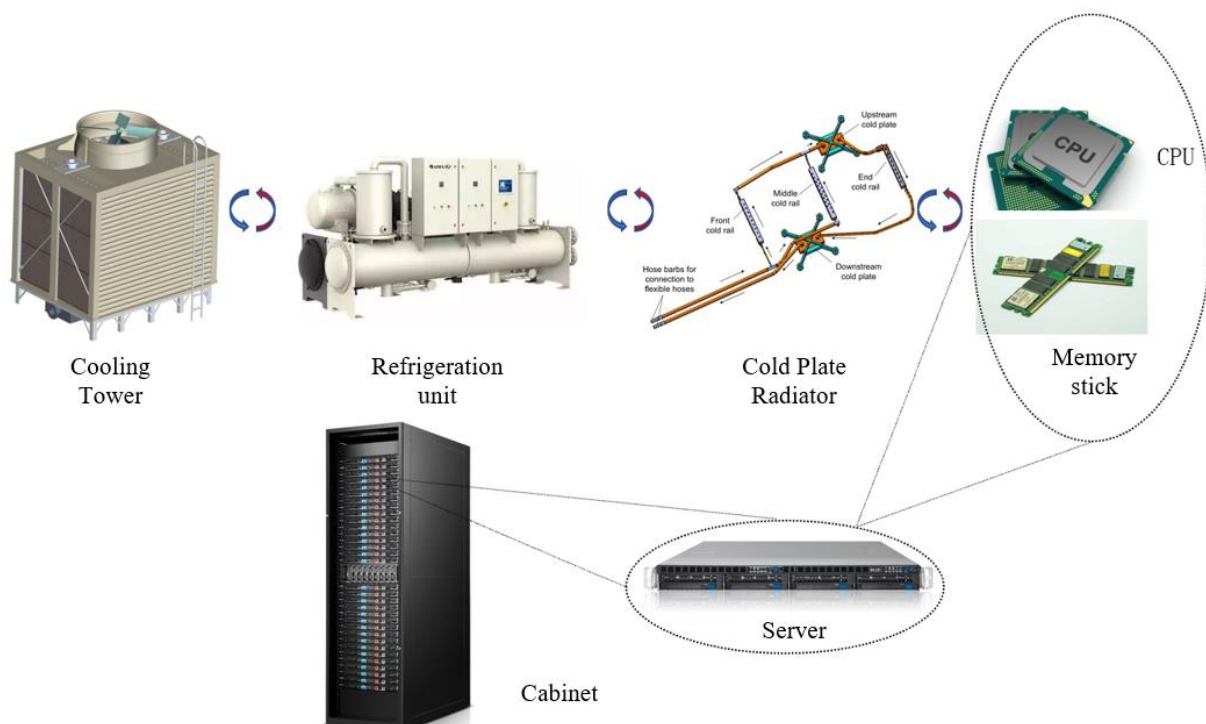


Figure 1. Framework Diagram of Liquid Cooling System.

2.1.2. Piping Network Design for Indirect Liquid Cooling Systems

Taking a single cabinet as an example, establish a physical model and a three-dimensional model of the internal piping for a cabinet-level indirect liquid cooling system, as shown in Figure 2.

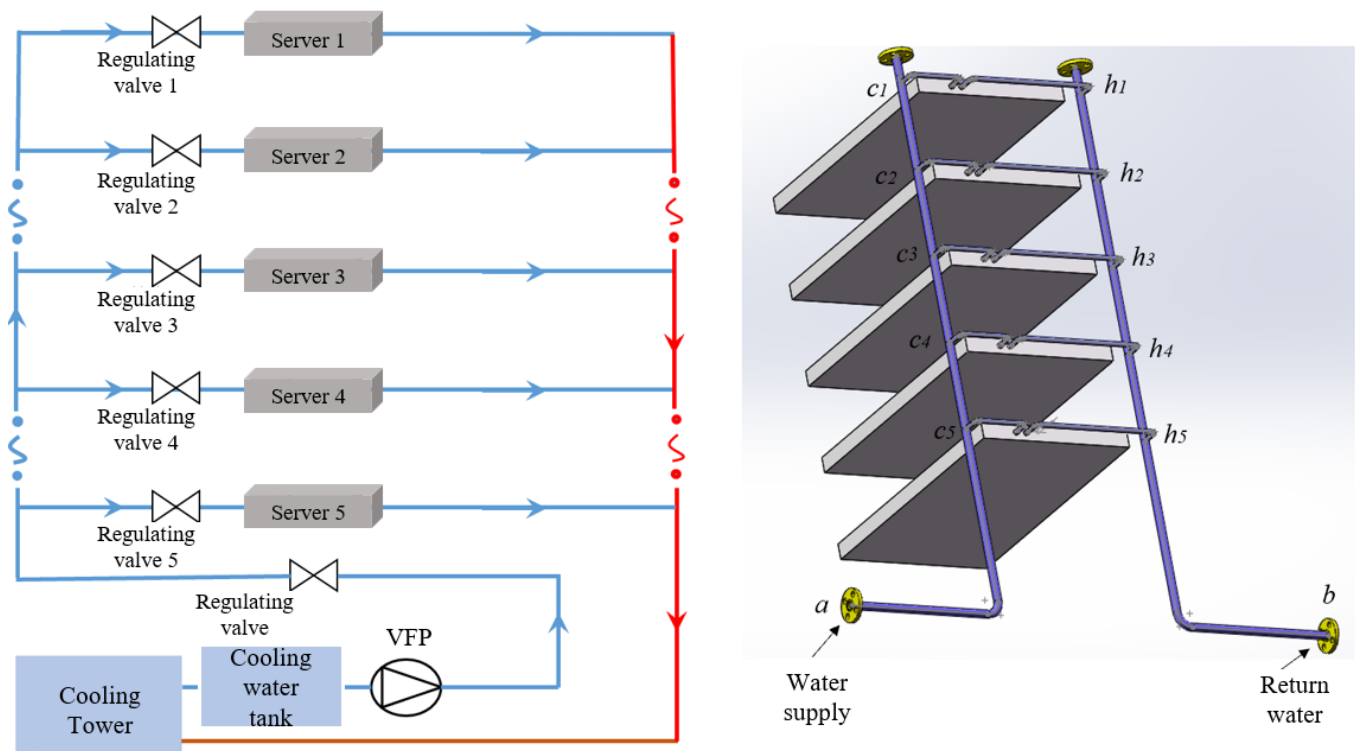


Figure 2. Physical Model and Three-Dimensional Model of Internal Piping for a Cabinet Liquid Cooling System.

The system consists of a cooling water circulation loop and servers. In the figure, five servers within the cabinet operate independently, generating heat as the heat source in the system. In the cooling water circulation loop, the coolant flows from the cold water tank (cold source), through the main piping network into the cabinet, and then sequentially through five branches into the servers, where it removes heat through cold plate heat exchangers before converging into the manifold and exiting the cabinet into the return main piping network, forming a circulation. The cooling water circulation piping network is equipped with variable frequency pumps and control valves on the main line and control valves on the branches. Among them, $a - c_5$, $h_5 - b$ represent the main pipe sections, $c_5 - c_1$, $h_1 - h_5$ represent the branch pipe sections, and $c_i - h_i$ ($i = 1, 2, 3, 4, 5$) represent each branch pipe section.

2.2. Piping Network System Design and Resistance Calculation

2.2.1. Piping Network Model Structure Parameters

Based on Figure 1, the structural parameters of the piping network system are provided according to engineering application design rationality in Tables 1 and 2.

Table 1. Pipe Lengths and Diameters of Different Sections in the Cabinet Cooling System Circulation Piping Network.

Pipe Segment Number	$a - c_5, h_5 - b$	$c_5 - c_1, h_1 - h_5$	$c_i - h_i$
Pipe Length (m)	16.0	2	0.8
Pipe Diameter (mm)	25.4	15.0	6.0

Table 2. Number of Local Loss Structures in the Circulation Piping Network.

Structure	$a-c_5$	Pipe c_5-c_1	Segment h_1-h_5	Number h_5-b	c_i-h_i
Right-Angle Elbow	2	0	0	2	0
Sudden Expansion Structure	0	0	0	1	1
Sudden Contraction Structure	1	0	0	0	1
Tree	0	5	5	0	0
Valve	1	0	0	0	1
Server	0	0	0	0	1

2.2.2. Piping Network Resistance Calculation

(1) Frictional Resistance

For a specific piping network system, with constant pipe diameter and flow rate, the frictional resistance coefficient is the main factor affecting pipeline frictional resistance losses, as shown in Table 3.

Table 3. Empirical Formulas for Calculating Frictional Resistance Coefficients Commonly Used in Industry.

Formula Name	Calculation Formula	Application Scenario
Aritsuri Formula	$\lambda = 0.11 \left(\frac{k}{d} + \frac{68}{Re} \right)^{0.25}$	Comprehensive formula for turbulent flow in piping networks
Blasius Formula	$\lambda = \frac{0.3164}{Re^{0.25}}$	Derived from the aforementioned equation, this empirical formula specifically targets the smooth pipe zone by neglecting the minimal relative roughness (k/d). It is applicable when the condition ($3 \times 10^3 \leq Re \leq 10^5$) is satisfied.

The piping materials in this paper are all stainless steel, and considering the small error between the experimental values of λ and its theoretical values, it can be neglected. Therefore, the frictional resistance coefficient is only related to the Reynolds number, as shown in Equation (1):

$$\lambda = f(Re) \quad (1)$$

Since

$$Re = \frac{vD}{\nu} \quad (2)$$

v : Fluid flow velocity in the piping network, m/s;

D : Pipe segment diameter, m;

ν : Kinematic viscosity of the coolant, m^2/s ;

Re : Reynolds number.

When the mass flow rate of chilled water in the pipeline is $q_v = 0.138 \text{ kg/s}$, and taking the kinematic viscosity of water at 20°C as $\nu = 1.006 \times 10^{-6} m^2/s$, through calculation, it is found that the Reynolds numbers during the fluid flow in each pipe section of the pipe network all satisfy $3 \times 10^3 \leq Re \leq 10^5$, meeting the hydrodynamically smooth zone formula.

Through the Blasius smooth zone formula $\lambda = \frac{0.3164}{Re^{0.25}}$, the resistance coefficients of each pipe section are calculated as shown in Table 4.

Table 4. Frictional Resistance Loss Coefficients for Each Pipe Segment in the Piping Network System.

Pipe Segment Number	Frictional Resistance Coefficient λ
$a - c_5$	0.041276
$c_5 - c_1$	0.028776
$h_1 - h_5$	0.028776
$h_5 - b$	0.041276
$c_i - h_i$	0.034221

(2) Local Resistance

Without considering the impact of pipe fitting relative roughness and Reynolds number on local resistance, the resistance coefficients of piping network components (sudden expansions, sudden contractions, right-angle elbows, tee fittings, and control valves) are calculated using theoretical calculations and empirical formulas, as shown in Table 5.

Table 5. Local Resistance Coefficients for Different Components.

Resistance Component	Pipe Segment	Local Resistance Coefficient
Right-Angle Elbow	$a - c_5, h_5 - b$	1.44
Sudden Expansion	$h_5 - b$	0.89
	$c_i - h_i$	0.70
Sudden Contraction	$a - c_5$	0.38
	$c_i - h_i$	0.42
Tee	$c_5 - c_1, h_1 - h_5$	1.50
Valve	$h_5 - b$	2.70
	$c_i - h_i$	0.10

2.2.3. Analysis of Piping Network Resistance Characteristics

The pressure difference caused by flow resistance in the piping network can be expressed in terms of pressure head:

$$\Delta P = \rho g H = \rho g (H_c + H_d). \quad (3)$$

$$H_d = \sum h_f + \sum h_w = \sum_i \lambda_i \frac{L}{D} \frac{v^2}{2g} + \sum_i \xi_i \frac{v^2}{2g}. \quad (4)$$

H : Pressure head of the working fluid in the pipe, m;

H_c : Static pressure head independent of the working fluid mass, m;

H_d : Dynamic pressure head related to the working fluid mass, m;

h_f : Frictional resistance loss in the pipe, m;

h_w : Local resistance loss in the pipe, m;

λ : Frictional loss coefficient;

L : Pipe length, m;

ξ : Local loss coefficient;

v : Working fluid flow velocity in the pipe, m/s;

g : Gravitational acceleration, m/s²;

i : Represents different pipe segments and components.

In the pipeline,

$$v = \frac{m}{\rho S} \quad (5)$$

where S is the cross-sectional area of the piping network system pipeline.

From the above, the expression for the dynamic pressure head can be derived as:

$$H_d = \sum h_f + \sum h_w = \frac{m^2}{2g\rho^2 S^2} \left(\sum_i \lambda_i \frac{L}{D} + \sum_i \xi_i \right). \quad (6)$$

Let

$$H_d = d_0 m^2 \quad (7)$$

$$d_0 = \frac{1}{2g\rho^2 S^2} \left(\sum_i \lambda_i \frac{L}{D} + \sum_i \xi_i \right) \quad (8)$$

The constant value is called the piping network dynamic pressure head coefficient.

The feature of a centralized pump liquid cooling system for cabinets lies in the fact that the power pump is situated on the main pipe section. Figure 3 presents a schematic diagram of the resistance coefficients within the piping network of a centralized pump cooling system. Due to the identical structure of the five branch paths in the diagram, the dynamic pressure head coefficients d_1, d_2, d_3, d_4, d_5 are the same. The pipe network system is divided into five parts according to the main pipe section and branch pipe sections. Among them, d_{01} represents the supply and return pipe pressure head coefficients of the main pipe section before entering the branch pipe section, with the working fluid flow rate in this section being the total flow rate of the cooling water cycle; d_{02} represents the supply and return pipe section pressure head coefficients after the branch pipe section passes through the No. 5 server branch, and the working fluid flow rate in this section is the total flow rate of the cooling water cycle minus the flow rate of the No. 5 server branch; similarly, d_{03} represents the flow rate on the pipe section as the sum of the flow rates of server branches 1 to 3, while d_{04} represents the sum of the flow rates of server branches 1 to 2 in the pipe section, and d_{05} represents the flow rate entering the No. 1 server in the pipe section.

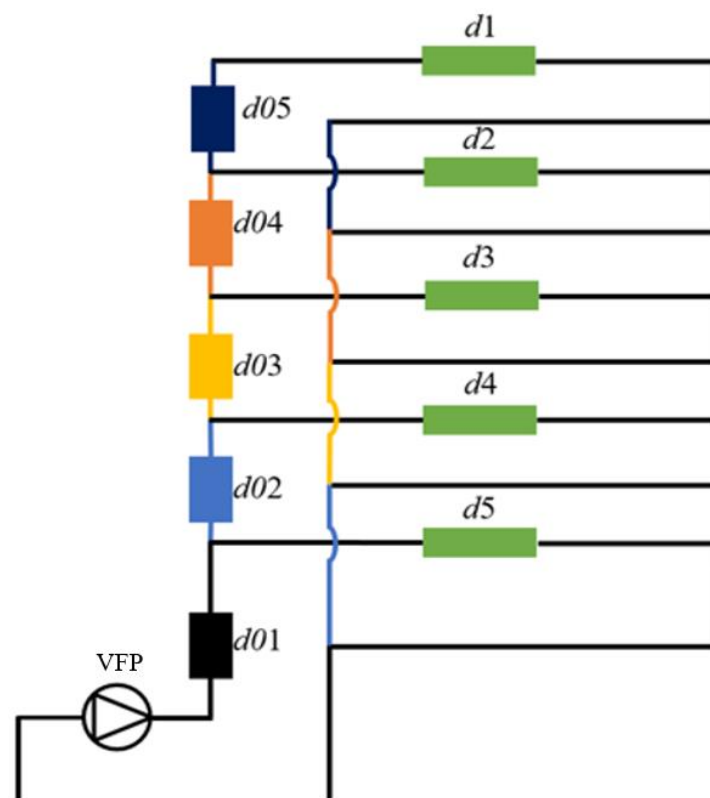


Figure 3. Schematic Diagram of Piping Network Resistance Coefficients.

Based on the above schematic diagram of the piping network resistance system, combined with Tables 2–6, and Equations (1) and (8), the dynamic pressure head coefficients for each pipe segment are calculated as shown in Table 6.

Table 6. Calculation Results of Dynamic Pressure Head Coefficients for Different Pipe Segments.

Pipe Segment Pressure Head Coefficient	$d01$	$d02, d03, d04, d05$	$d1, d2, d3, d4, d5$
Calculation Value ($\text{m} \cdot \text{s}^2/\text{kg}^2$)	10.18	8.06	670.1

3. Heat Flow Model for Cabinet Indirect Liquid Cooling Systems

3.1. Analysis of Heat Transfer and Flow Characteristics in Server Cabinet Cooling Systems

3.1.1. Analysis of Heat Transfer Characteristics in Indirect Liquid Cooling Systems

Based on Figure 4, the liquid cooling system for cabinets comprises a pipe network system, an expansion chilled water tank, and servers. The chilled water exits from the chilled water tank, removes heat as it traverses the interior of the servers, and subsequently returns to the refrigeration unit. In this context, T_{hi} ($i = 1, 2, 3, 4, 5$) signifies the specific heat dissipation temperatures of the servers, while $T_{c,i}$, $T_{c,o}$ represent the inlet and outlet temperatures of the chilled water, respectively. Furthermore, Q_i ($i = 1, 2, 3, 4, 5$) denotes the specific heat dissipation loads of the servers.

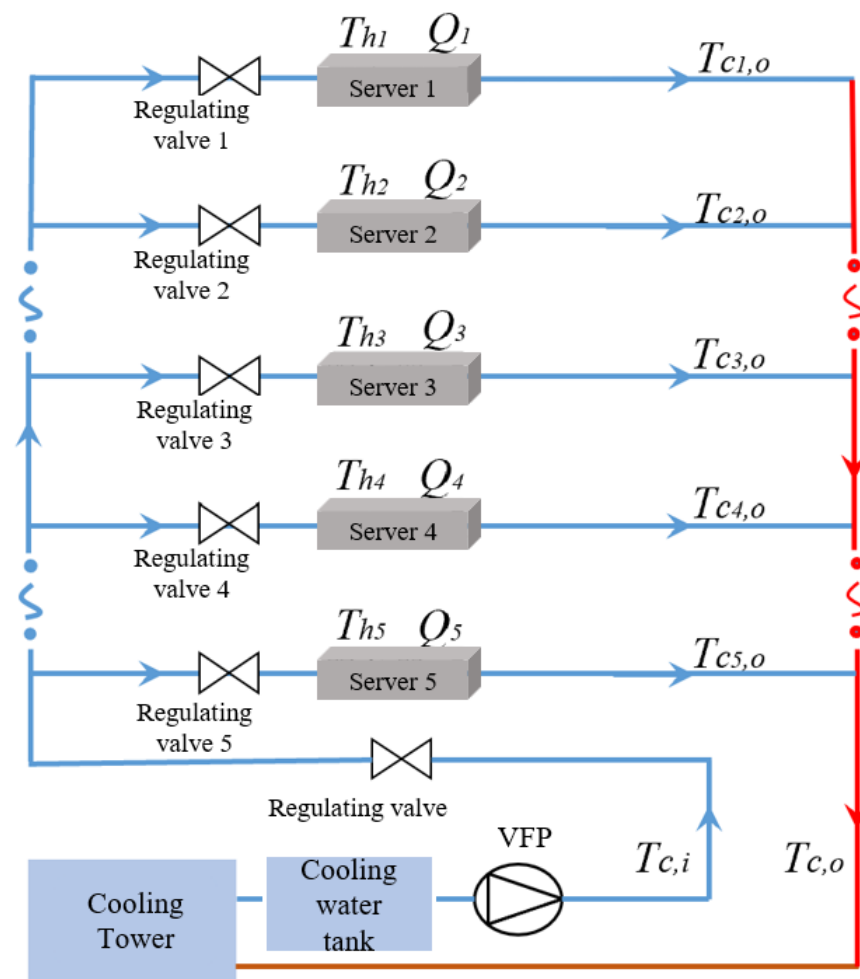


Figure 4. Physical Model of a Cabinet Liquid Cooling System.

The servers in the cabinet are connected in parallel. Based on the heat flow method [24], Figure 5 shows the heat flow model of the cabinet liquid cooling system.

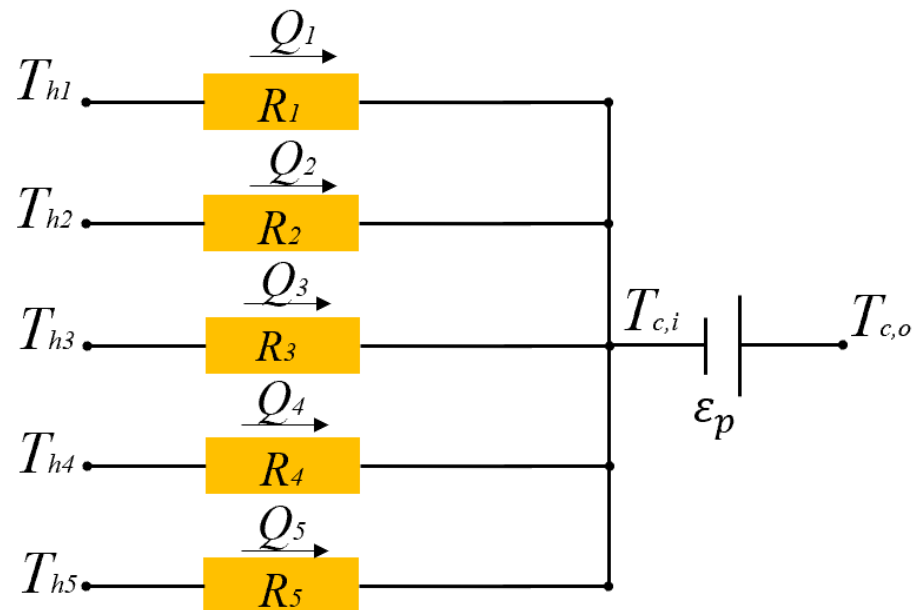


Figure 5. Equivalent Thermal Resistance Network Diagram of the Cabinet Cooling System.

In the figure, R_i ($i = 1, 2, 3, 4, 5$) represents the alternative thermal resistance of the heat transfer process between the cold plate of the liquid-cooled server and the working fluid, ε_p represents the additional electromotive force, and $T_{c,o}$ represents the outlet temperature of the cooling water in the main line. Combining Kirchhoff's laws, the overall heat transfer constraint of the cooling system is obtained:

$$T_{hi} - T_{c,i} = Q_i R_i \quad (9)$$

It is assumed that in the server heat sink, the temperature on the heat dissipation side remains constant, i.e., $T_{hi} = \text{constant}$. In this case, it can be equivalently considered that the heat capacity is infinitely large, i.e., $G_h = m_h c_{p,h}$ tends to be infinitely large. Consequently, the thermal resistance expression becomes

$$R_i = \frac{e^{(kA/m_i c_{p,c})}}{2m_i c_{p,c} (e^{(kA/m_i c_{p,c})} - 1)} \quad (10)$$

kA : Thermal conductivity of the water-cooled plate, $W/(m \cdot K)$

m_i : Mass flow rate of the cooling fluid, kg/s

$c_{p,c}$: Specific heat capacity of the cooling water at constant pressure, $J/(kg \cdot K)$

i : Server numbering

3.1.2. Analysis of Flow Characteristics in Indirect Liquid Cooling Systems

This paper proposes centralized and distributed pump configurations for cabinet liquid cooling systems as research objects. To simplify calculations while ensuring accurate and reliable optimization results, two assumptions are made: (1) The system operates stably. (2) In an established piping network system, the magnitude of the static head generated by the density difference between hot and cold fluids is negligible and can be disregarded [26].

(1) Construction of Flow Constraints for Centralized Pump-Driven Indirect Liquid Cooling Systems

Figure 3 shows a schematic diagram of the piping network resistance coefficients for a cabinet-level centralized pump liquid cooling system, and Section 2.2.3 provides the fluid resistance model in the piping network system, as shown in Equation (7).

The pump model can be derived from Equation (11):

$$H = a_0\omega_c^2 + a_1\omega_c\frac{m_c}{\rho} + a_2\frac{m_c^2}{\rho} \quad (11)$$

H : Dynamic pressure head of the variable frequency pump, m

ω_c : Operating frequency of the variable frequency pump, Hz

ρ : Density of the coolant, kg/m³

m_c : Mass flow rate of the fluid in the pipeline, kg/s

a_0, a_1, a_2 : Characteristic parameter of the variable frequency pump

Combining the flow model and the fluid dynamics–resistance balance relationship in the piping network system, the overall flow constraint for a cabinet-level centralized pump liquid cooling system is established:

$$a_0\omega_c^2 + a_1\omega_c\frac{m_c}{\rho} + a_2\frac{m_c^2}{\rho} = \sum_i^5 \left[d_{0,6-i} \left(\sum_1^i m_i \right)^2 + d_i m_i^2 \right] \quad (12)$$

(2) Construction of Flow Constraints for Distributed Pump-Driven Liquid Cooling Systems

The characteristic of the distributed pump liquid cooling and heat dissipation system for cabinets lies in the fact that the power pumps are arranged on each branch path. Figure 6 presents a schematic diagram of the resistance coefficients in the pipe network of the distributed pump liquid cooling and heat dissipation system for cabinets. Due to their identical structures, the five branch paths in the diagram have the same dynamic pressure head coefficients d_1, d_2, d_3, d_4, d_5 . The pipe network system is divided into five parts according to the main pipe section and branch pipe sections. Among them, d_{01} represents the supply and return pipe pressure head coefficients of the main pipe section before entering the branch pipe section, with the working fluid flow rate in this section being the total flow rate of the cooling water cycle; d_{02} represents the supply and return pipe section pressure head coefficients after the branch pipe section passes through the No. 5 server branch, and the working fluid flow rate in this section is the total flow rate of the cooling water cycle minus the flow rate of the No. 5 server branch. Similarly, d_{03} represents the flow rate on the pipe section as the sum of the flow rates of server branches 1 to 3, while d_{04} represents the sum of the flow rates of server branches 1 to 2 in the pipe section, and d_{05} represents the flow rate entering the No. 1 server in the pipe section.

By combining the flow model and the fluid dynamic-resistance balance relationship within the pipe network system, the overall flow constraints of the distributed pump liquid cooling and heat dissipation system for cabinets are established as follows:

$$a_{i0}\omega_i^2 + a_{i1}\omega_i\frac{m_i}{\rho} + a_{i2}\frac{m_i^2}{\rho^2} = \sum_i^5 \left[d_{0,6-i} \left(\sum_1^i m_i \right)^2 + d_i m_i^2 \right] \quad (13)$$

i : Represents each branch.

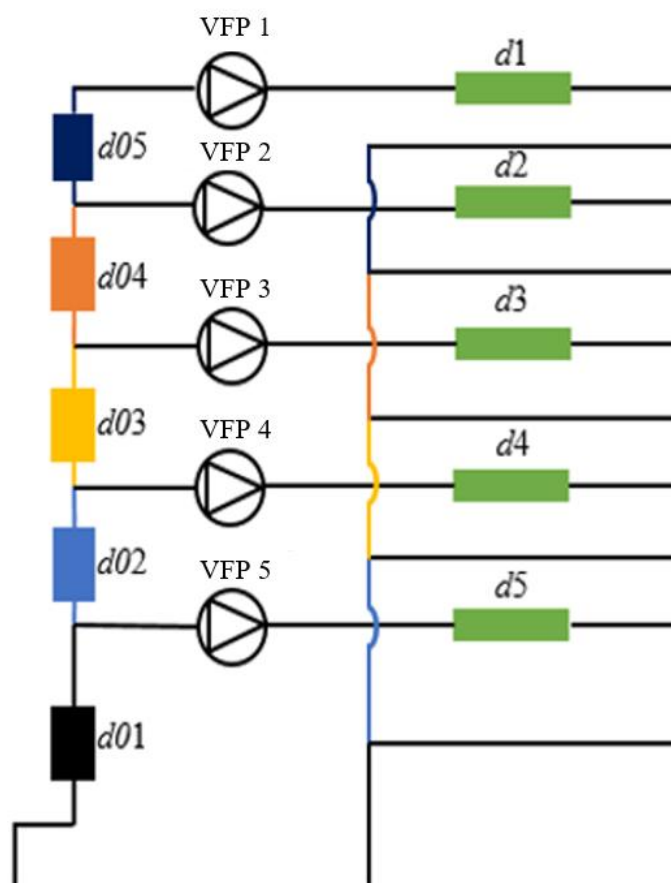


Figure 6. Schematic Diagram of Piping Network Resistance Coefficients for Distributed Pump Cooling System.

3.2. Boundary Conditions for Server Cabinet Cooling Systems

3.2.1. Heat Transfer Boundary Conditions

Based on the heat transfer performance research results of the center-diffusion type water-cooled plate in Section 2.2, the thermal conductivity of the water-cooled plate is set to 64 W/K. The heat transfer boundary conditions for the liquid cooling system model in a parallel structure are shown in Table 7.

Table 7. Heat Transfer Boundary Conditions.

Heat Sink Temperature (°C)	Cooling Water Temperature (°C)	Thermal Conductivity (W/K)	Gravitational Acceleration (m/s ²)
60	20	64	9.86

3.2.2. Flow Boundary Conditions

The flow parameter settings for the cabinet liquid cooling system primarily involve the resistance coefficients of each pipe segment, as shown in Table 8.

Table 8. Resistance Coefficients for Each Pipe Segment.

Pipe Segment Pressure Head Coefficient	d01	d02, d03, d04, d05	d1, d2, d3, d4, d5
Calculation Value(m · s ² /kg ²)	10.18	8.06	670.1

4. Comprehensive Economic Operation Optimization Analysis of Server Indirect Liquid Cooling Systems

The core process of heat exchange system optimization mainly includes the following steps: First, establish overall heat transfer and flow constraints for the system; second, clarify the thermal load, system boundary conditions, and piping network characteristic parameters; finally, determine decision variables and optimization objectives. Given temperature boundaries, heat exchanger thermal conductivity, and piping network resistance characteristics, the dual-objective optimization of enhancing heat transfer and reducing flow resistance can be transformed into two equivalent problems: (1) Minimize total pump power when the system heat exchange rate is constant; (2) Maximize system heat exchange rate when the total pump power is constant.

For cabinet-level indirect liquid cooling systems, this paper primarily conducts the following research: First, under given cooling water inlet temperature and server cooling temperature conditions, achieve the lowest system power consumption by adjusting the pump frequencies under centralized and distributed pump configurations to meet cooling demands; second, under limited total power consumption conditions, analyze the optimal allocation of the maximum system thermal load and compare the energy-saving performance of the two pump configurations; finally, study the impact of regulating branch thermal loads, cooling characteristic temperatures, and cooling water inlet temperatures on the system's cooling performance.

4.1. Economic Operation Optimization of Centralized Pump Indirect Liquid Cooling Systems

4.1.1. Economic Operation Optimization and Analysis Under Given System Total Load

The thermal load and thermal environment of the data center are given conditions. The energy-saving optimization approach for a centralized pump liquid cooling system is to minimize pump power consumption while meeting the thermal environment requirements of server electronic components. This problem can be transformed into a Lagrangian conditional extremum problem that minimizes total system pump power under a given total heat dissipation rate, and the following Lagrangian function is established:

$$L = Pt + \alpha_i [T_{hi} - T_{c,i} - Q_i R_i] + \beta \left[\begin{aligned} & a_0 \omega^2 + a_1 \omega \frac{m_c}{\rho} + a_2 \frac{m_c^2}{\rho^2} \\ & - \sum_i^5 \left[d_{0,6-i} \left(\sum_1^i m_i \right)^2 + d_i m_i^2 \right] \end{aligned} \right] \quad (14)$$

where α_i and β_i are Lagrangian multipliers.

Setting the partial derivatives of the above Lagrangian function with respect to equal to zero yields the following optimization equation group:

$$\frac{\partial L}{\partial X_i} = 0, X_i \in \{m_i, \alpha_i, \beta, \omega\} \quad (15)$$

where $i = 1, 2, 3, 4, 5$.

The equation group contains 12 equations involving 12 unknowns. Solving the equation group yields the variable frequency pump operating parameters that minimize total system pump power under a given cooling system thermal load. Although the operating conditions of servers 1~5 in the cabinet are not identical, resulting in slightly different thermal loads, the thermal loads of servers 1~5 in the cabinet are set to 800 W, and other boundary conditions are given in Tables 7 and 8. By solving, the cooling water flow rate and pump operating frequency that minimize total system pump power can be obtained.

Figure 7 illustrates the changes in system pump power under different total thermal load conditions. Under a specific thermal load condition, the point on the curve represents

the optimal performance of the cabinet liquid cooling system, where the system power consumption is the lowest. Points above the curve represent higher power consumption.

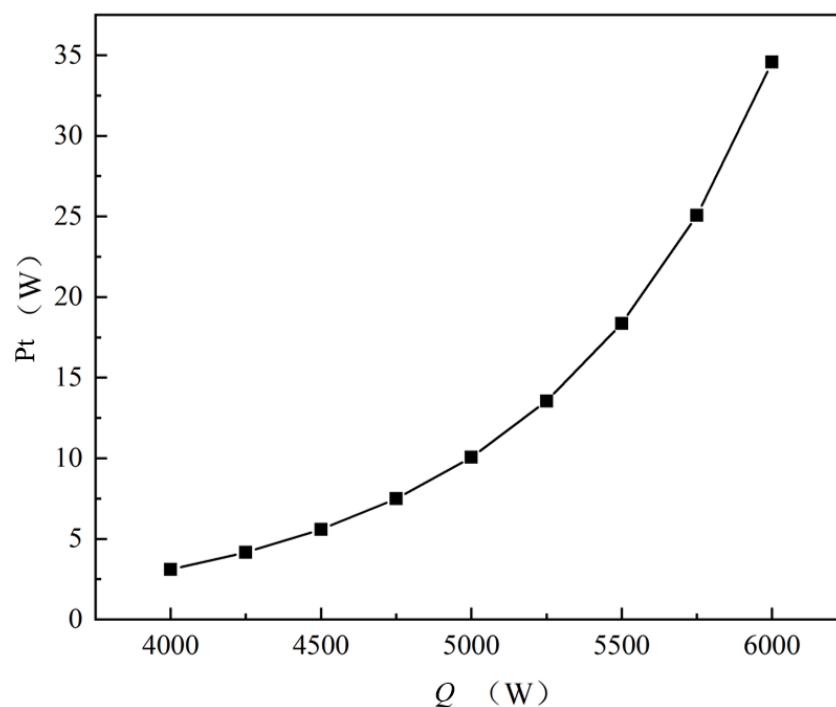


Figure 7. Minimum Power Consumption of the Cooling System under Different Total Heat Load Conditions.

Figure 7 shows that as the heat load increases, the minimum total power consumption rises at an accelerating rate. Figure 8 indicates that total power consumption increases with heat load, and this trend is more pronounced at higher inlet water temperatures. For instance, the power variation at a 24 °C inlet temperature is 1.55 times that at 20 °C.

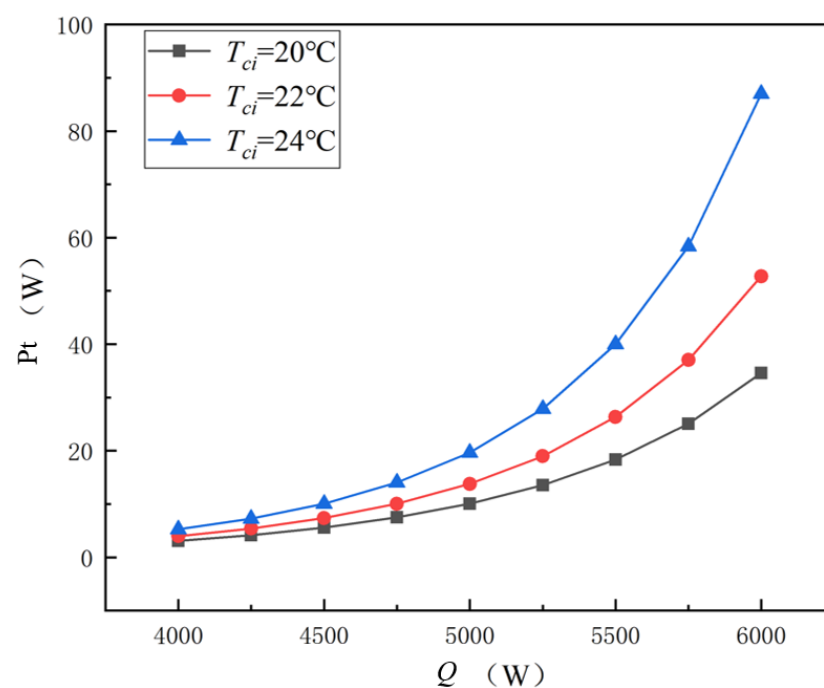


Figure 8. Variation Trend of Total Power Consumption with Chilled Water Temperature under Given Total Load Conditions.

It is necessary to study how fluctuations in individual server conditions affect the overall system. Figure 9 shows the relationship between branch flow requirements and the set dissipation temperature of Server 3. As the temperature of Server 3 increases, its specific branch flow decreases due to the larger heat transfer temperature difference, while other branch flows remain constant.

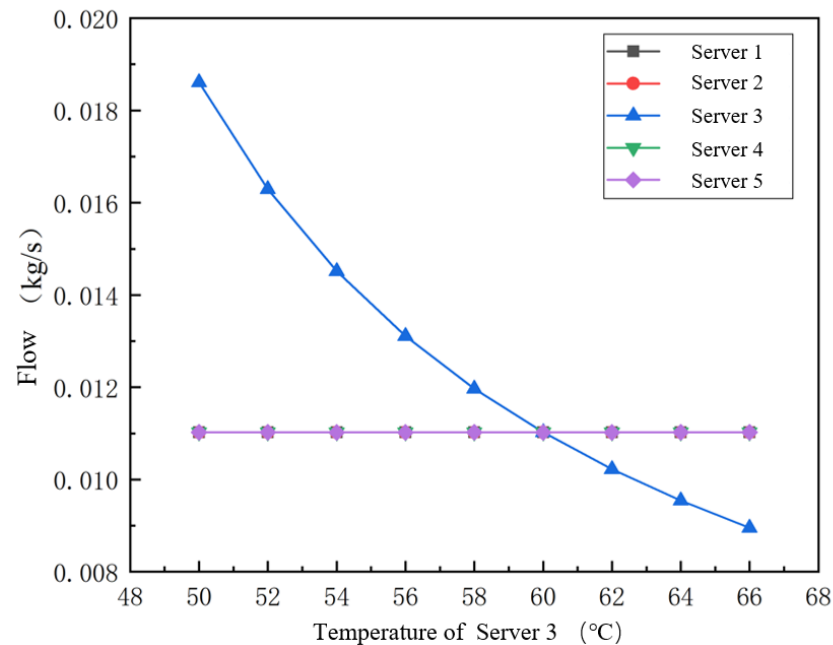


Figure 9. Variation Trend of Branch Flow Rate with the Set Temperature of Server 3.

Figure 10 presents the total power consumption as a function of heat loads for Servers 3 and 4. Power increases with heat load for both, but Server 4 drives a higher total power increase because it is located further from the variable-frequency pump, requiring more power to overcome resistance.

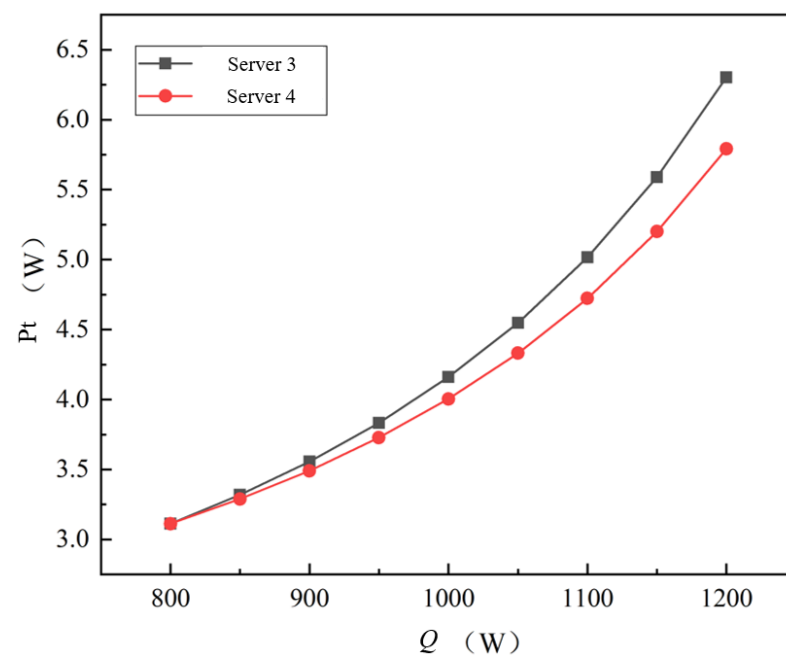


Figure 10. Variation Trend of Total System Power Consumption with the Heat Loads of Servers 3 and 4.

From an energy-saving perspective, placing high-load servers at the bottom of the cabinet is more efficient. Comparing two scenarios (a 1200 W high-load server at the bottom vs. at the top, with others at 800 W), the bottom placement results in 34.2% energy savings.

4.1.2. Optimization and Analysis Under Given Total System Power

Understanding the maximum cooling capacity under specific power constraints is crucial. This is modeled as a Lagrange problem to maximize total heat dissipation for a fixed power:

$$L = Q + \gamma [Pt - (a_0\omega^2 + a_1\omega\frac{m_c}{\rho} + a_2\frac{m_c^2}{\rho^2})m_c g] + \alpha_i [T_{hi} - T_{c,i} - Q_i R_i] + \beta \left[a_0\omega^2 + a_1\omega\frac{m_c}{\rho} + a_2\frac{m_c^2}{\rho^2} - \sum_i^5 \left[d_{0,6-i} \left(\sum_1^i m_i \right)^2 + d_i m_i^2 \right] \right] \quad (16)$$

where α_i, β, γ is a Lagrangian multiplier.

Letting the partial derivatives of the above Lagrangian function with respect to $m_i, \alpha_i, Q_i, \beta, \gamma, \omega$ equal zero yields the following optimization equation group:

$$\frac{\partial L}{\partial X_i} = 0, X_i \in \{m_i, \alpha_i, \beta, \omega\} \quad (17)$$

where $i = 1, 2, 3, 4, 5$.

The equation group contains 18 equations involving 18 unknowns. Solving the equation group yields the variable frequency pump operating parameters that maximize the total heat dissipation for a given total power consumption of the cooling system.

System other boundary conditions are given in Tables 7 and 8. Figure 11 shows the relationship between the total power consumption of the system and the heat dissipation of each branch server. As the total power consumption of the system gradually increases, the heat dissipation of each branch also increases. From the figure, it can be seen that the servers are numbered from top to bottom as 1, 2, 3, 4, and 5, and the difference in heat dissipation between branches gradually decreases.

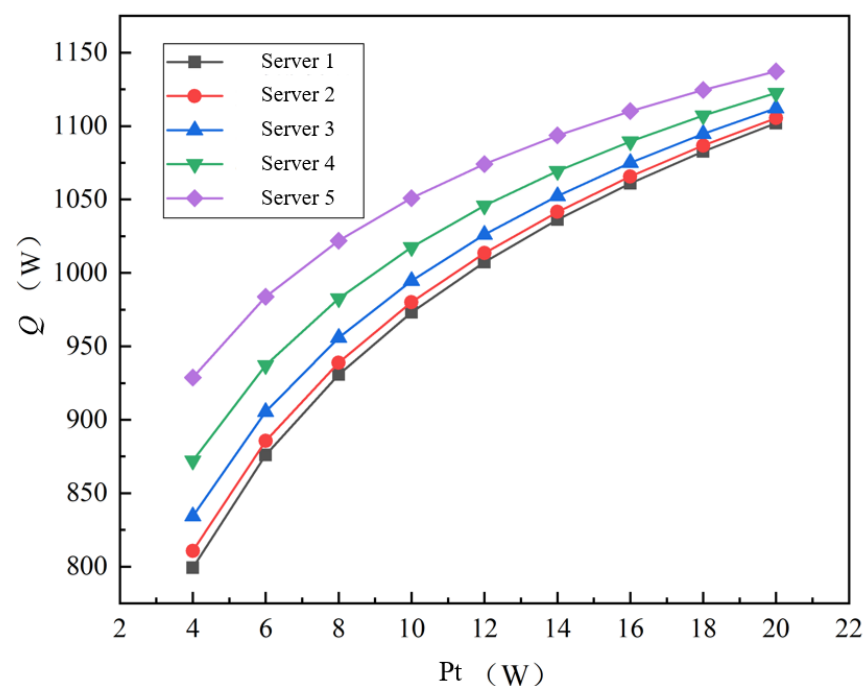


Figure 11. Variation Trend of Heat Dissipation in Each Branch under Given Total Power Consumption Conditions.

From Figure 11, when the maximum total heat dissipation of the system is achieved under a given total power consumption, the thermal load distribution method for each branch increases sequentially from the top to the bottom of the cabinet, which also verifies the conclusion of the previous section. Therefore, placing servers with higher workloads at the bottom of the cabinet is an effective measure for optimizing energy efficiency in indirect liquid cooling systems.

Figure 12 compares the impact of cooling water inlet temperature on the optimized heat dissipation of the cabinet. As the total power consumption of the system increases, the total heat dissipation also increases, but the trend slows down. When the inlet temperature increases, the heat exchange temperature difference decreases, resulting in a smaller total heat dissipation under the same power consumption. From the figure, it can be concluded that when the inlet temperature increases by 2 °C, the total heat dissipation decreases by 4.9%.

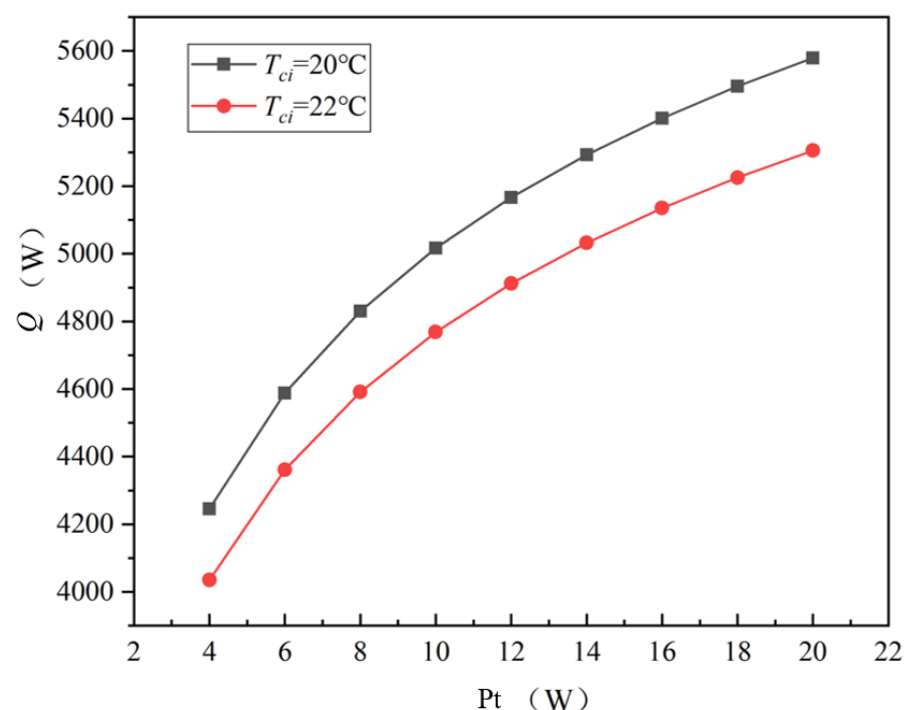


Figure 12. Variation Trend of Total Heat Dissipation with the Inlet Temperature of Cooling Water under Given Total Power Consumption Conditions.

4.2. Economic Operation Optimization of Distributed Pump Indirect Liquid Cooling Systems

The centralized pump arrangement lacks a branch cooling water distribution system, whereas the distributed pump cooling system offers high regulatory flexibility. Figure 6 shows a schematic diagram of the piping network resistance coefficients for the distributed pump cooling system, where variable frequency pumps are arranged on each branch.

Based on the overall heat transfer constraint (Equation (9)) and flow constraint (Equation (13)) of the system, a Lagrangian conditional extremum problem is established to maximize the total heat exchange rate of the system for a given total pump power, and the following Lagrangian function is established:

$$L = Pt + \alpha_i(T_{hi} - T_{ci} - Q_i R_i) + \beta_i \left[\frac{a_{i0}\omega_2 + a_{i1}\omega_i \frac{m_i}{\rho} + a_{i2} \frac{m_i^2}{\rho^2}}{\sum_i [d_{0,6-i}(\sum_1^i m_i)]^2} - d_i m_i^2 \right] \quad (18)$$

where α_i, β_i is a Lagrangian multiplier.

Letting the partial derivatives of the above Lagrangian function with respect to $\alpha_i, \beta_i, m_i, \omega_i$ equal zero yields the following optimization equation group:

$$\frac{\partial L}{\partial X_i} = 0, X_i \in \{m_i, \alpha_i, \beta_i, \omega_i\} \quad (19)$$

where $i = 1, 2, 3, 4, 5$.

The equation group contains 20 equations involving 20 unknowns. Solving the equation group yields the variable frequency pump operating parameters that minimize the total power consumption of the cooling system for a given thermal load.

The thermal loads of servers 1~5 in the cabinet are set to 800 W, and other boundary conditions are given in Tables 7 and 8. From the perspective of energy savings, Figure 13 compares the energy efficiency of the centralized and distributed pump arrangements. As the total thermal load of the system increases from 4000 W to 6000 W, the distributed pump system saves an average of 2.5 W, with a maximum energy savings of 7.09 W.

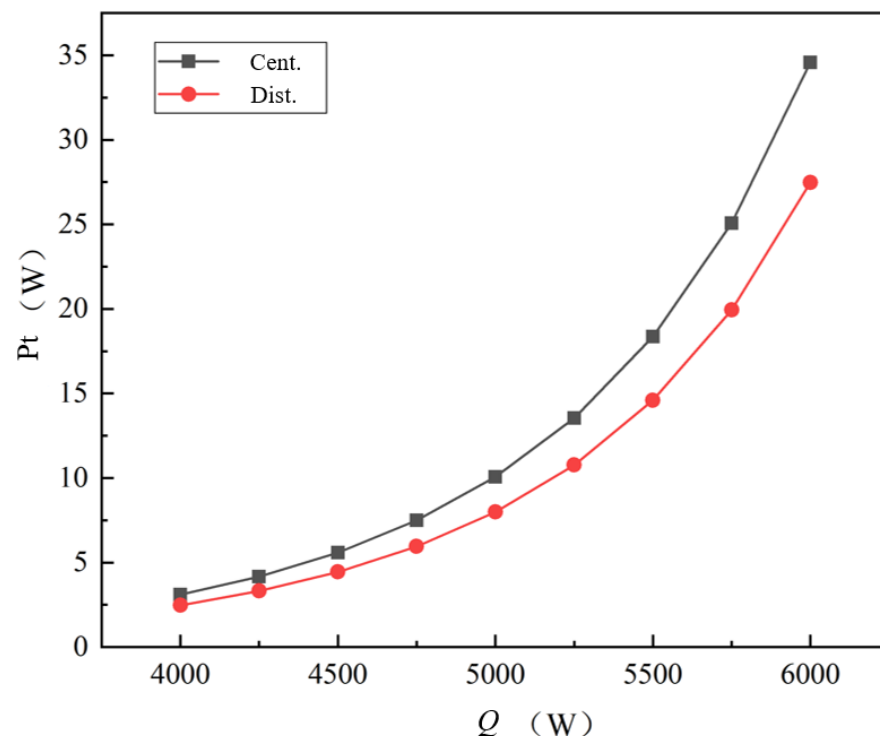


Figure 13. Comparison of Variation Trends of Total System Power Consumption with Total Load for Two Pump Configuration Methods.

Figure 14 shows the trend of the total power consumption of the system with the cooling water inlet temperature under a given total thermal load. As the inlet temperature increases from 20 °C to 28 °C, the maximum power consumption increases by approximately 3.78 times, indicating a significant variation in total power consumption. Therefore, whenever possible, using cooling water with a lower inlet temperature is more energy-efficient.

Figure 15 describes the variation in the frequency of each branch's variable frequency pump with the specific cooling temperature of Server 3. As the temperature of Server 3 increases, the frequency of the variable frequency pumps in all branches decreases, with the frequency of the pump in Branch 3 decreasing the fastest. Due to the increased heat exchange temperature difference between Server 3 and the cooling water, the demand for cooling water in the branch decreases, resulting in a gradual decrease in the flow rate in the main pipe section and related manifold sections of the cabinet's piping network system,

which in turn leads to a decrease in the operating frequency of the variable frequency pumps in each branch.

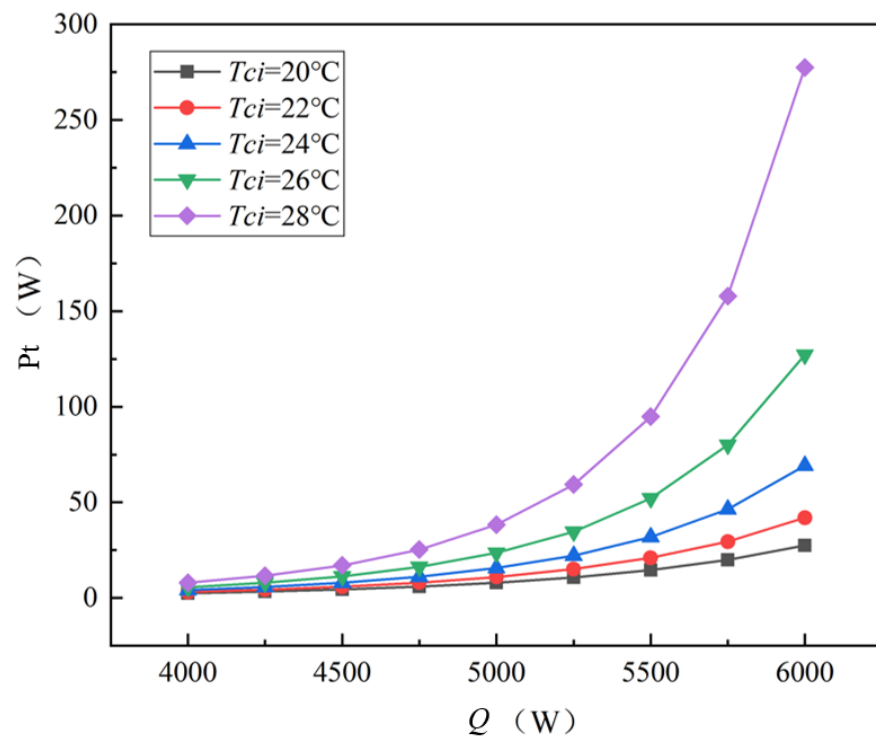


Figure 14. Variation Trend of Total System Power Consumption with Chilled Water Temperature under Given Total Load Conditions.

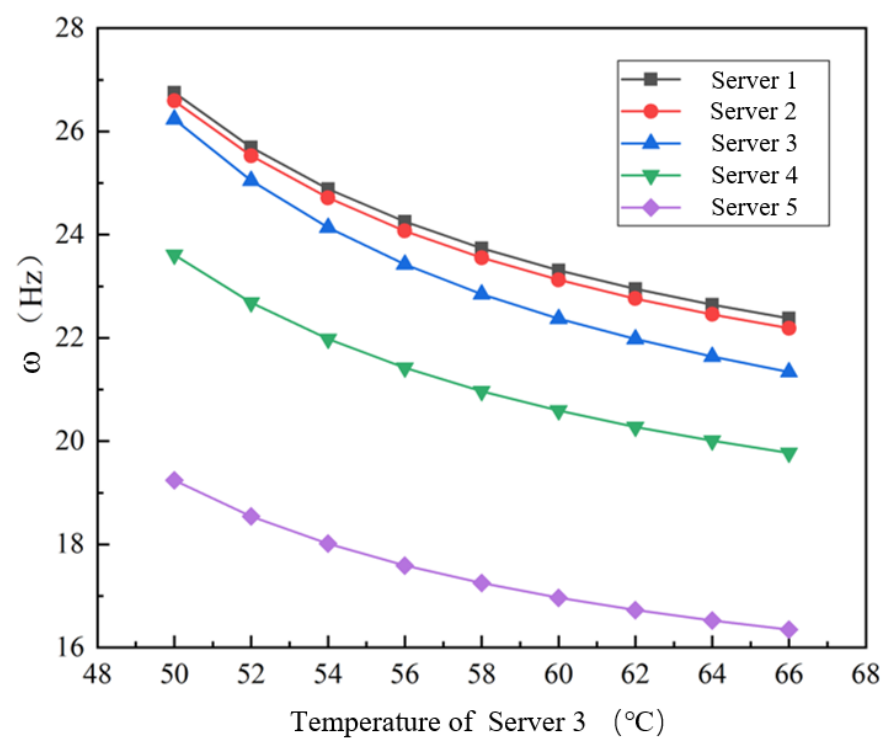


Figure 15. Variation Trend of Variable Frequency Pumps in Each Branch with the Temperature of Server 3.

5. Conclusions

To address the issues of server heat dissipation and cooling system energy consumption in data centers, this paper conducts an economic operation optimization study on the cold plate–rail liquid cooling and heat dissipation system in data centers. The main conclusions are as follows:

1. Under a given thermal load, the total power consumption of the centralized pump liquid cooling system increases by 1.55 times when the cooling water temperature rises from 20 °C to 24 °C. At a given total power consumption, a 2 °C increase in cooling water temperature reduces the thermal load by 4.9%. When a 1.2 kW server is placed at the top and bottom of the cabinet, the total power consumption of the latter is 34.4% lower than that of the former.
2. When the total thermal load increases from 4.0 kW to 6.0 kW, the distributed system saves an average of 2.5 W compared to the centralized system, with a maximum saving of up to 7.09 W.

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Nomenclature

A	Area, m ²
a	Variable—frequency pump characteristic parameter
c_p	Specific heat at constant pressure, J/(kg·K)
D	Diameter, m
d	Pipe network dynamic pressure head coefficient, m·s ² /kg ²
G	Thermal capacity flow, W/K
g	Gravitational acceleration, m/s ²
H	Pressure head, m
h	Resistance loss, m
K	Heat transfer coefficient, W/(m ² ·K)
L	Length, m
m	Mass flow rate, kg/s
P	Pressure, Pa
Q	Heat flow rate, W
q	Heat flux density, W/m ²
R	Thermal resistance, K/W
Re	Reynolds number
S	Pipe cross—sectional area, m ²
T	Temperature, °C
v	Flow velocity, m/s
W	Power, W
Greek Letters	
ε	Thermoelectromotive force, °C
α, β, γ	Lagrange multipliers
ρ	Fluid density, kg/m ³
ω	Variable—frequency pump operating frequency, Hz

λ	Along—route resistance coefficient
ξ	Local resistance coefficient
ν	Fluid kinematic viscosity, m ² /s
Subscripts	
c	Low—temperature side
f	Along—route loss
h	High—temperature side
i	Inlet; i-th branch
j	The j-th
n	The n-th
o	Outlet
w	Local loss
ph	Thermal resistance of parallel—flow heat exchanger
ch	Thermal resistance of counter—flow heat exchanger

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