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Energy and Economic Analysis of Power Generation Using Residual Pressure of a Circulating Cooling Water System

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Abstract: With rising energy prices and the intensification of environmental problems, researchers have paid increasing attention to the recovery of the residual pressure energy of the industrial circulating cooling water system (CCWS) in hydraulic turbines. Taking the existing CCWS as the research object, this study analyzes the feasibility of the transformation of the power generation using residual pressure from the perspectives of energy and economy. The energy flow analysis of the system reveals that the hydraulic optimization of the system should be carried out first to obtain the minimum total energy consumption of the pump and the turbine. Then, combined with the advantages of the traditional hydraulic optimization regulation strategy of the water supply network, a synchronous regulation strategy of the pipeline and the pump station is proposed. On the basis of the synchronous regulation strategy of the pipeline and the pump station, this research proposes a method for a comprehensive feasibility analysis of the CCWS's power generation using residual pressure. Finally, taking a CCWS as an example, the simulation and comparison experiments of four transformations of the power generation using residual pressure are designed. The experiments not only prove the application value of the comprehensive analysis proposed in this research, but also prove the conclusion of the energy flow analysis mentioned above to be correct.

Keywords: circulating cooling water system; residual pressure power generation; energy and economy; feasibility; adjustment strategy

1. Introduction

Amid intensifying global energy and environmental problems, promoting the optimization and upgrading of traditional industries and effectively curbing the blind development of high energy consumption, high emissions, and low energy efficiency projects have become urgent problems for industries worldwide. A circulating cooling water system (CCWS) is an important auxiliary system for industrial production, which has the characteristics of wide application, large water volume, and high energy consumption. According to statistics, the energy consumption of CCWS accounts for about 20–30% of the total energy consumption of industrial production [1,2]. Hence, research on the energy conservation of this system has a certain significance in energy conservation and environmental protection.

In the early stage, Westerlund et al. [3] regarded the process of solving the optimal pump configuration as a mixed integer nonlinear programming (MINLP) problem, in which both series and parallel pumps were considered simultaneously. Then, Pettersson et al. [4] introduced a binary separable program to design a relatively economical and flexible pump network. Their model adopted a technology for frequency conversion drive control to improve the efficiency of the pump system [5]. Recently, Pontes et al. [6] developed a mixed integer non-linear model of cooling water pumping systems. The objective function minimizes the sum of the capital and operating costs of the pumping system, and the decision

variables for the model are the pump and motor models to use, the number of pumps and variable frequency pumps(VFDs), and the operating points of the pumps. In addition, Kim et al. [7] established a mathematical model with the lowest cooling water flow rate as the design goal based on pinch point technology [8]. Feng et al. [9] also proposed a new heat exchange network with an intermediate main pipe based on pinch point technology to reduce flow rate of the cooling water in the system. However, the optimal design of the heat exchange network structure for minimizing the flow rate of the cooling water may increase the network pressure drop [10–12]. Therefore, Kim et al. [13] considered the pressure drop in their subsequent research and further expanded the optimization model. Sun et al. [14] proposed a design method for the two-step sequential optimization of the heat exchanger network and the pump network. The first step is to calculate the best heat exchanger network with a series–parallel configuration according to the thermodynamic model, and then the second step is to obtain the best pump network with auxiliary pumps, according to the hydraulic model. Later, Zhang et al. [15] extended the concept of two-step sequential optimization to systems involving dry type air-cooled heat exchangers and wet type air-cooled heat exchangers. However, the two-step sequential optimization method has some shortcomings. When establishing the best cooler network, the designed pump network may not be the optimal. Then, Ma et al. [16] proposed a synchronous optimization method for a heat exchanger network and a pump network, which considers the internal relationship between the pump network and the cooler network. To verify the effectiveness of the synchronous optimization method, the case of Sun et al. [14] has been used for the current research. Compared with the two-step sequential optimization method, the synchronous optimization method can save 6.4% of the annual total cost. Picón-Núñez et al. [17] proposed a network design with a new heat exchanger as an insert and considered the impact of the new heat exchanger on the performance of the cooling tower and water pump. In addition, some scholars have considered the scaling problem of heat exchangers during optimization [18–20]. Among them, Coletti et al. [18] considered the influence of pressure drop and temperature on scaling. In addition, Oliveira Filho et al. [19] considered the scaling problem of heat exchangers and proposed to adopt appropriate flow rate regulation in some heat exchanger networks, which can fully tap the overall heat exchange performance of the network. Zheng [21] put forward a superstructure comprehensive MINLP model, carried out the overall optimization design of the circulating cooling water system, which simultaneously obtained the optimal location and load of each cooling tower, as well as the optimal configuration of the pump network and cooling water network.

In recent years, many scholars have taken energy recycling as the direction to reduce the energy consumption of CCWS operation. Yang et al. [22] proved that the heat pump can improve the thermal economy by recovering the waste heat of the CCWS. Michele et al. [23] believed that it was economically feasible to replace the pressure-reducing valve in the water distribution network with a pump turbine to recover the energy of the residual pressure. Gao et al. [24,25] and Ma et al. [26] carried out the combined design of pump and turbine for the system with a fixed network structure and operating conditions. They proposed a new structure, with an auxiliary pump and a turbine added to the branch of the heat exchange network. In their research, the modeling analysis was carried out based on the minimum total power and the minimum total cost of the pump and turbine in the system. However, in the research of Gao and Ma, the drastic change that can occur in water consumption of the heat exchanger within a year was not considered. Therefore, the network structure of the designed pump and turbine was too ideal and was not suitable for the transformation and application in the actual system. In addition, Chen et al. [27] recycled the residual pressure of cooling water in the upper tower pipe section of the cooling tower with a hydraulic turbine. The cooling water drives the turbine to rotate first, then the turbine drives the fan above the cooling tower to operate. However, the cooling water in this method is not only the power source of the fan but also the cooling object, which seriously affects the heat dissipation capacity of the cooling tower. The recovery of

residual pressure must not be at the cost of affecting the production and operation of the system; otherwise, it will abandon the basics. Therefore, this method is not suitable for application in practical systems. Through the summary of practical engineering problems and current research results, it is found that research on the recovery of the residual pressure energy of the existing the CCWS is insufficient.

The above introduction indicates that researchers have proposed many effective design methods for the optimization and energy saving of the CCWS. However, in the past, most of the design optimizations of the CCWS were global optimization designs of the equipment or structure of the pump network and heat exchanger network with a certain optimization objectives. Such research is more suitable for the design stage before the system is built. However, for the built system, restrictions are often put in place, such as enterprise planning, determination of system operation indicators, site space, transformation funds, investment return cycle, etc. Thus, a freer design optimization application under ideal conditions is difficult to achieve. As such, carrying out partial transformation for this kind of built inefficient system and obtaining considerable energy-saving benefits in the short term are often more practical.

According to the investigation and research, there is still a lack of theoretical research on the recovery of the residual pressure energy of the built circulating cooling water system by using hydraulic turbines, and there is no feasibility evaluation method for the transformation of the power generation using residual pressure of the system. Therefore, this paper takes the built CCWS as the research object, and conducts an in-depth study on the feasibility of the power generation using residual pressure from the perspective of energy and economy. The innovation of this paper is that it clarifies that the system hydraulic optimization is a prerequisite for the transformation of the power generation using residual pressure, and proposes an improved hydraulic optimization strategy combining the advantages of two traditional hydraulic regulation strategies, and, on this basis, a comprehensive method to evaluate the feasibility of the power generation using residual pressure of the CCWS is proposed.

2. System Energy Flow Analysis

Energy is a measure of the movement of matter. Common energy forms include mechanical energy, electric energy, thermal energy, electromagnetic energy, optical energy [28], chemical energy [28], and nuclear energy [29]. Mechanical energy can be subdivided into kinetic energy and potential energy. The energy flow analysis of the CCWS analyzes the transmission, consumption, and transformation of various energies in the system. The CCWS energy flow analysis is shown in Figure 1.
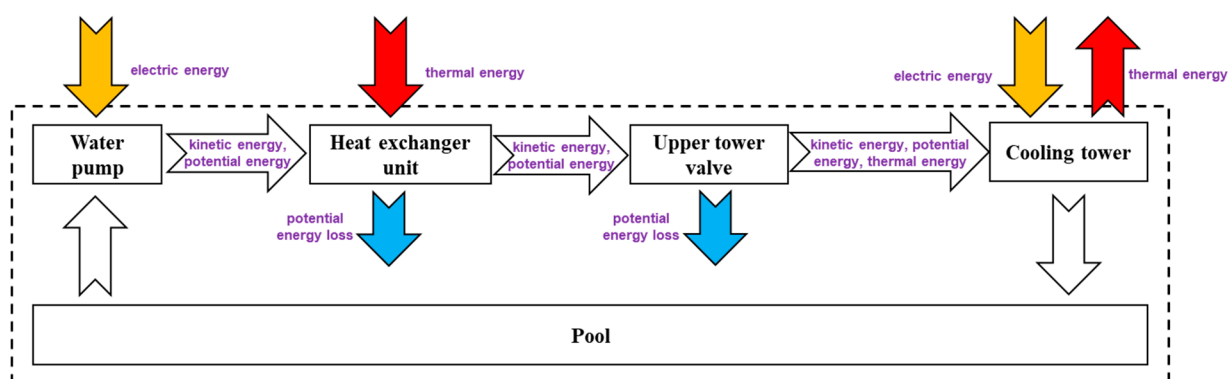

Figure 1. CCWS energy flow.

The diagram for the analysis of the CCWS energy flow shows that the external energy input of the system includes electric energy and thermal energy. The circulating water pump converts part of the electric energy into the kinetic energy and potential energy of the cooling water. When the cooling water flows through the heat exchanger network, it absorbs

the heat energy at the working medium side. Moreover, it is affected by the resistance along the way and the local resistance, and, in the process of flowing through the heat exchanger, pipeline, and branch valve, part of the potential energy is consumed. Afterward, some potential energy is lost when the cooling water flows through the upper tower valve of the cooling tower. After flowing out of the upper tower valve, the remaining potential energy and kinetic energy will send the cooling water to the cooling tower. Inside the cooling tower, the cooling water falls freely into the pool. The fan sucks the air around the cooling tower into the cooling tower, and it flows out from bottom to top. The temperature of the cooling water drops after full contact with the air for heat exchange. Regardless of the weak heat generated by the flow of cooling water in the pipe network and the weak heat dissipated in this process, the electric energy consumed by the fan only participates in the heat exchange process of cooling water, and the electric energy consumed by the pump only participates in the transportation process of cooling water. The two have different functional roles in CCWS.

To reduce pumping power consumption, the hydraulic turbine can be used to replace the decompression effect of the valve on the upper tower of the cooling tower and recover the abundant residual pressure energy of the system. When formulating the energy-saving transformation plan for hydraulic turbines, the goal is to minimize the total energy consumption of the water supply network:

$$\min(E_p - E_t), \tag{1}$$

where E_p and E_t are the power consumption of the pump and the power generation of the turbine, respectively.

The following energy equation can be established from the transmission, consumption, and conversion process of electric energy consumed by the water pump in the CCWS.

$$E_p\eta_{pm} = \Delta E_{net} + E_k + \rho g Q h_{tower} + \frac{E_t}{\eta_{tm}}, \tag{2}$$

where E_k is the kinetic energy of water entering the cooling tower, which accounts for less than 1% of the mechanical energy converted from the electric energy consumed by the water pump and can be ignored. E_t is the generating capacity of the hydraulic turbine. h_{tower} is the height of the cooling tower. ΔE_{net} represents the potential energy loss of the heat exchanger network, which can be expressed as the flow rate Q and the head loss of the heat exchanger network Δh_{net}.

$$\Delta E_{net} = \rho g Q \Delta h_{net}. \tag{3}$$

η_{pm} and η_{tm} are the comprehensive efficiency of water pump and the comprehensive efficiency of water turbine power generation, respectively. Among them,

$$0 < \eta_{pm} < 1, \tag{4}$$

$$0 < \eta_{tm} < 1. \tag{5}$$

The generating capacity of the turbine can be expressed as

$$E_t = \rho g Q \Delta h_t \eta_{tm}, \tag{6}$$

where Δh_t is the head loss of the valve on the cooling tower, which is the available head of power generation using residual pressure in the system.

The transformation of Equation (2) becomes

$$E_p - E_t = \frac{\rho g}{\eta_{pm}} Q \Delta h_{net} + (\frac{1}{\eta_{pm}\eta_{tm}} - 1)\rho g Q \Delta h_t + \frac{\rho g Q h_{tower}}{\eta_{pm}}. \tag{7}$$

That is, the objective function can be expressed as

$$\min(\frac{\rho g}{\eta_{pm}}Q\Delta h_{net}+(\frac{1}{\eta_{pm}\eta_{tm}}-1)\rho g Q\Delta h_t + \frac{\rho g Q h_{tower}}{\eta_{pm}}). \tag{8}$$

Among them,

$$0 < \frac{1}{\eta_{pm}\eta_{tm}}-1. \tag{9}$$

Therefore, for a certain system, to minimize the total energy consumption of the water supply network, the flow rate Q and pressure drop of the heat exchanger network Δh_{net} and the recoverable head of the hydraulic turbine Δh_t must be simultaneously reduced. The operating head of the water pump is determined by Δh_{net}, Δh_t, the height of cooling tower, and the liquid level of the pool. Therefore, at this time, the operating head of the water pump is the minimum head under production conditions.

3. Hydraulic Optimization and Regulation Strategy of Water Supply Network

The analysis of system energy flow reveals that the minimum total power consumption of the water supply network occurs when the flow rate and pressure drop of the heat exchanger network and the operating head of water pump are the minimum. However, due to the complex structure of the CCWS and the large number and variety of heat exchangers, hydraulic imbalance is a common phenomenon in actual operation. Hydraulic imbalance describes the inconsistency between the actual flow rate, pressure state, and production demand of the hydraulic system. However, for the same system, under the same conditions, different hydraulic optimization and regulation strategies will also have different energy performances. Therefore, before the feasibility analysis of the CCWS's power generation using residual pressure can be conducted, the hydraulic optimization and regulation strategy of the system water supply network must be determined.

3.1. Traditional Regulation Strategy

In the production and operation of the traditional regulation strategy, raw material prices and market demand will cause changes in the working medium flow rate. Therefore, the water supply network in the CCWS needs to distribute the cooling water flow rate and the system pressure continuously according to the actual heat exchange demand on the working medium side of the heat exchanger. In other words, the operating point of the pump station must be adjusted according to the demand. The operating point of the parallel pump station is both determined by the Q–H curve of the pump station and the resistance characteristic curve of the current pipe network system. The intersection of the two curves is the actual operating point of the pump station. This operating point is based on the balance of the energy supply and demand between the pump station and the pipeline. As long as one side changes, its operating point will be transferred. The working characteristic curve equation of the parallel pump station can be obtained by the superposition of the working characteristic curve equation of a single pump. The head of the parallel pump station is the same as that of each parallel pump, and the flow rate of the parallel pump station is the sum of the flow of each parallel pump.

The working characteristic curve equation of a single constant speed pump can be expressed as:

$$H = H_b - K_b Q^2. \tag{10}$$

The working characteristic curve equation of a single variable frequency pump [30] can be expressed as:

$$H' = \left(\frac{n'}{n}\right)H_b - K_b Q^2. \tag{11}$$

The resistance characteristic curve equation of the pipe network system can be expressed as:

$$H = \Delta Z + kQ^2. \tag{12}$$

where H_b is the theoretical maximum head of the pump, K_b is the comprehensive resistance coefficient in the pump, k is the comprehensive resistance coefficient of the pipe network, and ΔZ is the height between the height of the cooling tower or the highest level heat exchanger and the water level of the pool.

The traditional hydraulic regulation strategy includes changing the pipeline characteristic curve and changing the pump station characteristic curve, which are called the pipeline regulation strategy and the pump station regulation strategy, respectively. The pipeline regulation strategy is generally implemented to regulate the flow rate dynamically by adjusting the valve opening in the pipeline. Therefore, according to different water demands, the heat exchanger network can be continuously maintained in a hydraulic balance state by reasonably adjusting the valve opening at the end of each branch, as shown in Figure 2.
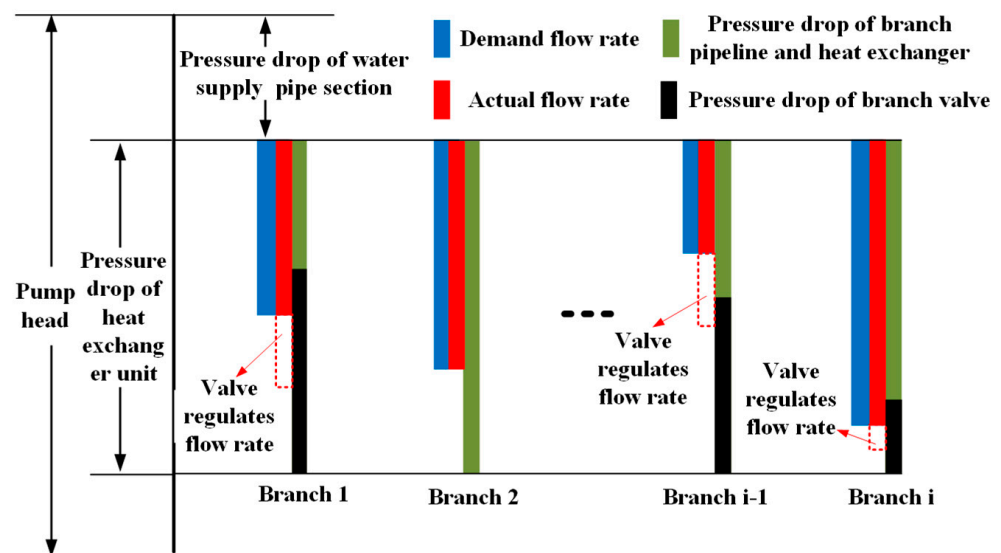

Figure 2. Hydraulic balance adjustment of heat exchanger network.

Figure 3 shows the change of the operating point of the pump station when the pipeline regulation strategy is adopted. At the initial stage, the operating point of the pump station is at the intersection point A (Q_0, H_0) of the pump station characteristic curve H and the pipeline characteristic curve R. As the heat exchange demand of the unit decreases, the total flow rate for the cooling water of the whole heat exchanger network decreases to Q_1. The figure demonstrates that to reduce the flow rate of the heat exchanger network, the outlet valve of the heat exchanger must be turned down. When the opening of some valves in the pipeline decreases and the resistance coefficient of the pipeline system increases, the pipeline characteristic curve becomes steeper and becomes R', and the pump station characteristic curve H remains unchanged. The operating condition point of the pump station moves to the left to point B (Q_1, H_1), and the total flow rate of the whole system decreases. Although the pipeline regulation strategy can continuously regulate the system flow rate, turning down the valve will increase the head loss of the pipe network system, which will lead to the increase of the pump operating head. Hence, this method has a large energy loss. Therefore, the pipeline regulation strategy can obtain the minimum flow rate and the pressure drop of the heat exchanger that meets the production, but cannot obtain the minimum pump operation head.
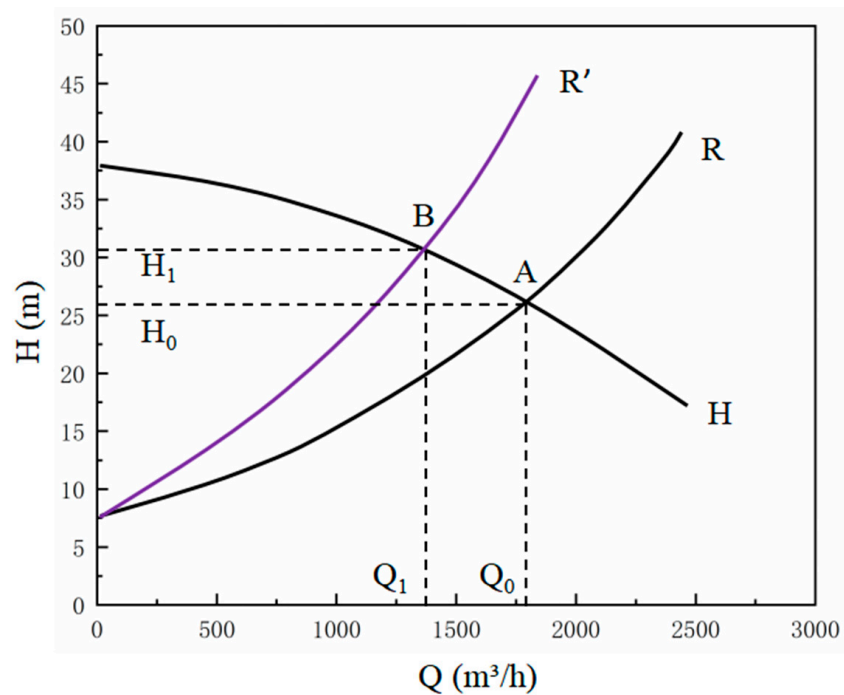
Figure 3. Pipeline regulation strategy.

The pump station characteristic curve can be changed in three ways: changing the pump rotating speed, stopping or starting the pump, and cutting the impeller. Given that the starting and stopping of the pump does not have the ability to adjust continuously and the impeller cutting can only move the pump characteristic curve downward, changing the pump rotating speed is relatively superior way among the three. However, changing the rotating speed of the water pump requires adding frequency conversion or permanent magnet speed regulation devices, and the cost is also the highest among the three. In the actual pump station, only one pump is usually equipped with a frequency conversion device when the system flow fluctuates greatly, combined with the start and stop operations of the parallel pump to ensure continuous regulation ability.

Figure 4 shows the change diagram of the operating point of the pump station by adjusting the rotating speed of the frequency conversion pump. The system flow rate is reduced by adjusting the rotating speed of the pump, and the total water output of the pump station is reduced under the same operating head. Therefore, the pump station characteristic curve moves downward from H to H′, and the pipeline characteristic curve R remains unchanged at this time. Given that the number of heat exchangers in the heat exchanger network is large and their structures are often different, the flow rate reduction of the heat exchanger will not be exactly the same when the heat exchange demand of the device is reduced. After adjusting the characteristic curve of the pump station, hydraulic balance regulation should be established on the basis of the condition that it does not affect the production. The flow rate demand of the heat exchanger for water consumption bottlenecks should be satisfied after adjusting the pump station characteristic curve, which is bound to cause the flow rate of some heat exchangers to be greater than the actual demand. In this case, the operating condition point of the pump station moves to the left to point C (Q_2, H_2). The total flow rate of the whole system decreases, and the operating head of the pump also decreases. Although changing the characteristic curve of the pump station can continuously adjust the system flow and reduce the operating head of the pump, this method has a large flow rate surplus, and the minimum system flow rate and pump head cannot be obtained.
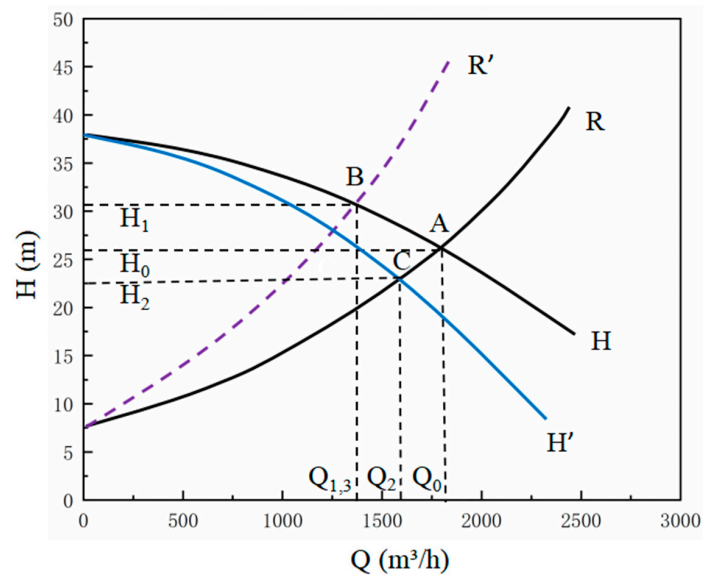
Figure 4. Regulation strategy of pump station.

3.2. Synchronous Regulation Strategy of Pipeline and Pump Station

Combining the advantages of the two traditional strategies, a new synchronous regulation strategy for a pipeline and pump station is proposed in this study. This novel strategy aims to reduce the flow rate of the heat exchanger network, the pressure drop of the heat exchanger network, and the operating head of the water pump as much as possible. Figure 5 shows the change diagram of the operating point of the pump station when the synchronous regulation strategy for the pipeline and the pump station is adopted. The figure also demonstrates that the characteristic curve H of the pump station becomes H″ by adjusting the speed of the variable-frequency pump, the characteristic curve R of the pipeline becomes R″ by adjusting the valve in the pipeline, and the operating condition point of the pump station moves to point D (Q_1, H_3). Compared with the pipeline regulation strategy, the flow rate and pressure drop of the heat exchanger network are the same, but it has a smaller pump operating head. Compared with the pump station regulation strategy, the cooling water flow rate of each branch of the heat exchanger is equal to the demand flow rate, and the total flow rate of the system is smaller. In addition, the head loss and the operating head of the pump are smaller due to the smaller flow rate of the water supply and the return main pipe section.
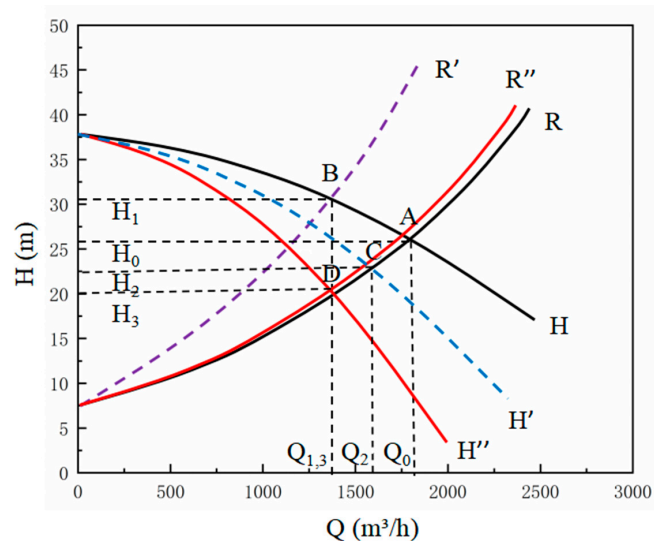

Figure 5. Synchronous regulation of pipeline pump station.

By comparing the three strategies, the synchronous regulation strategy of the pipeline and the pump station is found to have the smallest heat exchanger network flow, heat exchanger network pressure drop, and pump operation head.

4. Comprehensive Analysis Method

This study uses the three-step method to analyze the feasibility of the CCWS power generation using residual pressure. The first step is to analyze quantitatively the flow rate and pressure of the water supply network combined with the synchronous regulation strategy of the pipeline and the pump station. The second step is to analyze quantitatively the residual pressure energy of the system from the perspective of energy based on the results of a quantitative analysis of the flow rate and the pressure of the water supply network. The third step consists of establishing the annual benefit mathematical model of the hydraulic turbine based on the results of the energy analysis, and analyzing the economy of power generation using residual pressure.

4.1. Optimization Analysis of Hydraulic Parameters of the System

The hydraulic imbalance phenomenon in CCWS and the synchronous regulation strategy of the pipeline and the pump station have been discussed previously. On this basis, this section will quantitatively analyze the flow rate and pressure of the water supply network in combination with the equipment structure and installation height of the heat exchanger and the installation height of the cooling tower.

4.1.1. Calculation Model of Minimum Flow Rate and Pressure Drop in Heat Exchanger Pipe Network

The heat exchanger is the service object for cooling water, and its working performance directly affects the performance of the whole CCWS. When the pumping water volume is too large, the system energy consumption will be too high. While the pumping water volume is too small, the expected heat exchange goal cannot be achieved, and problems such as scaling and corrosion can easily occur inside the heat exchanger.

The thermodynamic model of heat exchanger can be expressed as [2]:

$$\begin{cases} f_{Ei} = c_{h_{Ei}} M_{Ei}(T_{Ei,1} - T_{Ei,2}) \\ f_{Ei} = c_w Q_{heat_{Ei}}(t_{Ei,2} - t_{in}) \\ f_{Ei} = K_{Ei}(\Delta t_{max_{Ei}} - \Delta t_{min_{Ei}}) / \ln(\Delta t_{max_{Ei}} / \Delta t_{min_{Ei}}) A_{Ei} \\ K_{Ei} = 1 / (1 / (\overline{K}_{sheel_{Ei}} X_{Ei}^{0.55}) + \overline{r}_{fouling_{Ei}} + 1 / (\overline{K}_{tube_{Ei}} Y_{Ei}^{0.8})) \end{cases}, \qquad (13)$$

where ϕ_{Ei} is the heat exchange for heat exchanger. $c_{h_{Ei}}$, M_{Ei}, $T_{Ei,1}$, and $T_{Ei,2}$ are the specific heat capacity and flow rate of hot fluid and the temperatures at the inlet and outlet of the heat exchanger, respectively. c_w, $Q_{heat_{Ei}}$, $t_{Ei,2}$, $t_{Ei,1}$ are the specific heat capacity and flow rate of cooling water and the temperatures at the inlet and outlet of the heat exchanger, respectively. $\Delta t_{min_{Ei}}$ is a larger temperature difference at both ends of the heat exchanger, while $\Delta t_{min_{Ei}}$ is a smaller temperature difference at both ends of the heat exchanger. A_{Ei} is the total heat transfer area of the heat exchanger. The heat transfer coefficient K_{Ei} can be determined by the convective heat transfer coefficient $\overline{K}_{sheel_{Ei}}$ outside the row of tubes, convective heat transfer coefficient $\overline{K}_{tube_{Ei}}$ in tubes, and fouling thermal resistance coefficient $\overline{r}_{fouling_{Ei}}$. X_{Ei} is the increase rate of the working fluid flow rate, and Y_{Ei} is the increase rate of the cooling water flow rate.

The outlet temperature of the working medium side should be less than the alarm temperature. In the actual system, a certain margin must exist between the outlet temperature of the working medium and the alarm value. In this study, a margin of 2 °C is reserved.

$$T_{Ei,2} = \overset{\leftrightarrow}{T}_{Ei,2} - 2, \qquad (14)$$

where $\overset{\leftrightarrow}{T}_{Ei,2}$ is the alarm temperature at the working medium outlet.

In addition to meeting the demand of heat exchange, the heat exchanger should also avoid the increase in scaling rate in the heat exchanger tubes due to low flow rate. Moreover, it must ensure that the flow rate of the cooling water in the tubes is greater than the safety standard. With reference to the specification for the flow rate requirements of heat exchangers in SINOPEC enterprises, the minimum flow rate in the tubes shall not be less than 0.6 m/s. The minimum anti-fouling flow of heat exchanger $Q_{fouling_{Ei}}$ can be expressed as

$$Q_{fouling_{Ei}} = 0.6 n_{Ei} \pi d_{Ei}^2 / 4 N_{P_{Ei}}, \tag{15}$$

where n is the number of tubes, d_{Ei} is the inner diameter of tubes, and $N_{P_{Ei}}$ is the number of tube passes.

The cooling water demand flow rate Q_{total} of the heat exchanger should be the larger flow rate required for heat exchange Q_{heat} and the minimum flow rate for anti-fouling $Q_{fouling}$.

$$Q_i = \max(Q_{heat_{Ei}}, Q_{fouling_{Ei}}). \tag{16}$$

The sum of the cooling water flow rate of each branch is equal to the total flow rate:

$$\sum Q_i = Q_{total}. \tag{17}$$

The pressure drop of the parallel pipe network shall meet the water demand of the heat exchanger in all branches:

$$\Delta h_{f_{net}} = \max\left(k_{f_{i,line}} Q_i^2 + k_{f_{Ei}} Q_i^2 \right) \tag{18}$$

where $k_{f_{i,line}}$ represents the comprehensive resistance coefficient of branch i pipeline. In addition, $k_{f_{Ei}}$ represents the comprehensive resistance coefficient of the heat exchanger on branch i.

4.1.2. Calculation Model of Minimum Head of Water Pump

In the quantitative calculation of the minimum head of the pump, the pressure distribution characteristics of the water supply network of the CCWS must be analyzed, and the possible pressure bottleneck position of the cooling water in the flow process of the network should be discussed. Then, the water pump head constraint is established according to the pressure at the bottleneck position, and the minimum water pump head is finally determined.

Figure 6 shows the water supply network, which considers the height of equipment. The minimum value of the cooling water pressure may appear in two places in the water supply network. One is behind the outlet valve of each heat exchanger, which is the end point Ci of the water supply section. The other is at the water inlet of the cooling tower, which is point 5 at the end of the water return section.
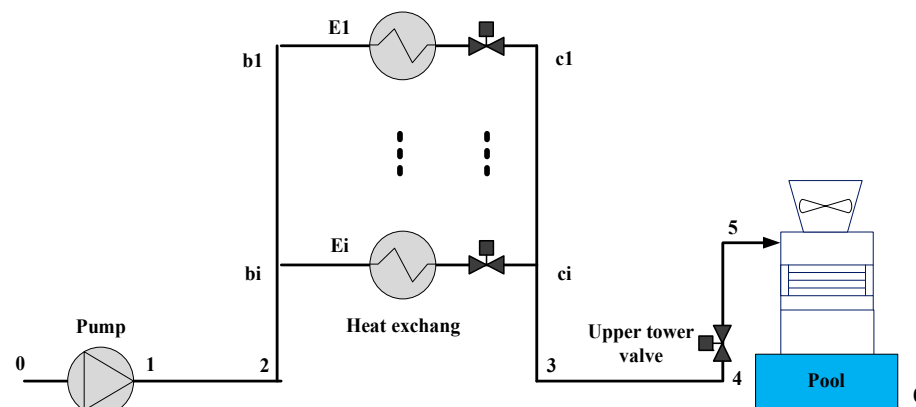

Figure 6. Schematic diagram of water supply network.

According to the analysis of the pressure distribution of the front pipe network, to ensure the normal operation of the system, the pump head H_p should meet the following two pressure constraints:

(1) The pressure value behind the outlet valves of the heat exchangers on each branch is not less than zero.

$$H_p + h_c - \left(h_{f_{0-2}} + h_{b,i} + \Delta h_{f_{net}}\right) \geq 0. \tag{19}$$

(2) The pressure of cooling water entering the cooling tower is not less than zero.

$$H_p + h_c - \left(h_{tower} + h_{f_{0-2}} + \Delta h_{f_{net}} + h_{f_{3-4}}\right) \geq 0, \tag{20}$$

where h_c is the liquid level height of the pool, $h_{b,i}$ is the height of branch i, and h_{tower} is the height of the cooling tower. $h_{f_{0-2}}$ and $h_{f_{3-4}}$ represent the head loss of the main pipe section of the water supply from position 0 to position 2 and the head loss of the return main pipe section from position 3 to position 4, respectively.

According to the above constraints, the minimum head of the water pump can be expressed as:

$$H_p = \max\left(h_{f_{0-2}} + h_{b,i} + \Delta h_{f_{net}} - h_c, h_{tower} + h_{f_{0-2}} + \Delta h_{f_{net}} + h_{f_{3-4}} - h_c\right). \tag{21}$$

When the system is not equipped with a booster pump, the pump head should meet the water demand of the heat exchanger with the highest layout, and the branch height should be taken at maximum $(h_{b,i})$ at this time.

Therefore, when $\max(h_{b,i}) \geq h_{tower} + h_{f_{3-4}}$,

$$H_p = \max(h_{b,i}) - h_c + h_{f_{0-2}} + \Delta h_{f_{net}}. \tag{22}$$

When $\max(h_{b,i}) < h_{tower} + h_{f_{3-4}}$,

$$H_p = h_{tower} - h_c + h_{f_{0-2}} + \Delta h_{f_{net}} + h_{f_{3-4}}. \tag{23}$$

When the system is equipped with a booster pump for the high-level heat exchanger, the pressurization effect of the booster pump on the branch must be considered when calculating the pump head. As the research object in this study does not involve the branch booster pump, the analysis is not carried out.

The head loss of the water supply main pipe and the return main pipe section is squared with the total flow rate of the system:

$$\begin{cases} h_{f_{1-2}} = k_{f_{1-2}} Q_{total}^2 \\ h_{f_{3-4}} = k_{f_{3-4}} Q_{total}^2 \end{cases}, \tag{24}$$

where $k_{f_{1-2}}$ and $k_{f_{3-4}}$ are the comprehensive resistance coefficients of water supply and return main pipe sections, respectively.

4.2. Energy Analysis

From the previous analysis, engineers will reduce the excessive return water pressure by reducing the opening of the valve on the cooling tower. Replacing the valve with a hydraulic turbine can not only reduce the pressure of the valve, but also use the residual pressure of the system to generate electricity and recover part of the pumping power consumption.

The head loss of upper tower valve $H_{v,tower}$ can be expressed as:

$$H_{v,tower} = H_p + h_c - h_{f_{0-2}} - \Delta h_{f_{net}} - h_{f_{3-4}} - h_{tower}. \tag{25}$$

Combined with the calculation Formulas (22)–(23) for the minimum head of the water pump, we can obtain:

When $\max(h_{b,i}) \geq h_{tower} + h_{f_{2-3}}$,

$$H_{v,tower} = \max(h_{b,i}) - h_{tower} - h_{f_{2-3}}. \tag{26}$$

When $\max(h_{b,i}) < h_{tower} + h_{f_{2-3}}$,

$$H_{v,tower} = 0. \tag{27}$$

After analysis, when the maximum height of each branch of the heat exchanger network is greater than the sum of the height of cooling water entering the tower and the head loss of the return water pipe, the head loss of the upper tower valve exists. At this time, the maximum generating capacity of the turbine can be expressed as

$$\max E_t = \rho g Q_{total}(\max(h_{b,i}) - h_{tower} - h_{f_{2-3}})\eta_{tm}. \tag{28}$$

In the actual system application, the transformation of the hydraulic turbine will increase the hydraulic loss of the upper tower pipe, and the available head of the hydraulic turbine should be reduced by the pressure loss caused by the residual pressure transformation. In addition, some CCWS will adopt a hot return water bypass filter, and the available flow rate of hydraulic turbine should be deducted this part of the flow rate. The actual generating capacity of the turbine is

$$E_t = \rho g Q_{total}(1 - \Gamma_{bf})(\max(h_{b,i}) - h_{tower} - h_{f_{2-3}} - h_{add})\eta_{tm}, \tag{29}$$

where Γ_{bf} is the bypass filtration coefficient of the hot return water and h_{add} is the additional flow resistance brought by the transformation of power generation using residual pressure.

4.3. Economic Analysis

The reconstruction cost of the hydraulic turbine power generation system is often high. Thus, whether the hydraulic turbine can recover the residual pressure energy of the system should also be subject to economic analysis, as the CCWS may operate under different working conditions within a given period. Therefore, both the large flow rate condition and the small flow rate condition must be considered in the economic analysis of the hydraulic turbine. The maximum flow rate condition refers to the system operation state, which has the maximum load rate of the system working medium and the upper limit of the water supply temperature control. The minimum flow rate condition corresponds to the system operation state, which has the minimum load rate of the system working medium and the lower limit of the water supply temperature control.

The annual power generation benefit of hydraulic turbine can be expressed as [31]:

$$PGB = \rho g Q_{total}(1 - \Gamma_{bf})(\max(h_{b,i}) - h_{tower} - h_{f_{2-3}} - h_{add})\eta_{tm} \times C_{price}\tilde{t}, \tag{30}$$

where C_{price} is the electricity price. $\tilde{t}$ is the annual operation time of the turbine.

According to references [24,26,31], the investment cost IC of a single turbine can be expressed as a function of turbine head H_t and flow rate Q_t.

$$IC = \beta_1 (Q_t \cdot H_t)^{\beta_2}, \tag{31}$$

$$Q_t = Q_{total}(1 - \Gamma_{bf})/s, \tag{32}$$

$$H_t = \max(h_{b,i}) - h_{tower} - h_{f_{2-3}} - h_{add}, \tag{33}$$

where β_1 and β_2 are the investment factors of the hydraulic turbine, which can be obtained according to the actual material costs and construction costs of different factories.

Therefore, the annual total benefit (TAB) of hydraulic turbine can be expressed as:

$$
\begin{aligned}
\text{TAB} &= \text{PGB} - (A_f + \widetilde{C})\,\text{IC} \\
&= \left[\rho g Q_{\text{total}}(1 - \Gamma_{\text{bf}})(\max(h_{b,i}) - h_{\text{tower}} - h_{f_{2\text{–}3}} - h_{\text{add}})\eta_{\text{tm}} \cdot C_{\text{price}}\widetilde{t} \right] \\
&\quad - (A_f + \widetilde{C}) \cdot \left[\beta_1 \left(\tfrac{Q_{\text{total}}(1-\Gamma_{\text{bf}})}{s} \cdot (\max(h_{b,i}) - h_{\text{tower}} - h_{f_{2\text{–}3}} - h_{\text{add}}) \right)^{\beta_2} \right],
\end{aligned}
\tag{34}
$$

where $\widetilde{C}$ is the interest on bank loans. A_f is the depreciation factor of the hydraulic turbine.

When $\text{TAB} \leq 0$, no economic condition exists for the transformation of the power generation of the system using residual pressure.

5. Results and Discussion

First, the model in this chapter is applied to a small CCWS, which is an experimental system based on the actual CCWS. Only one circulating water pump in the system delivers cooling water to the pipe network, and the cooling objects are five heat exchangers with different heights and structures. The system is equipped with a cooling tower, which is 6 m high, and the liquid level in the pool below the cooling tower is 2 m high. The alarm temperatures of heat exchangers E1, E2, E3, E4, and E5 are 40 °C, 40 °C, 40 °C, 45 °C, and 42 °C, respectively. In the actual control, the outlet temperature of the working medium side should be more than 2 °C lower than the above temperature.

Structural parameters of heat exchanger are shown in Table 1.

Table 1. Structural parameters of heat exchanger.

Heat Exchanger Tag Number	E1	E2	E3	E4	E5
Height (m)	5	15	25	27	30
Inlet pipe diameter (mm)	500	250	300	200	400
Inner diameter of tube (mm)	15	15	15	15	15
Tube length (m)	9	4.5	6	6	6
Number of tubes	2246	568	594	444	1946
Number of tube passes	1	2	2	2	2

The original operation state of the system is shown in Figure 7 (the reference plane of the system height is the height of the water pump, and the elevation corresponding to the pressure in the figure is 0 m). At this time, the water supply temperature of the system is 31.5 °C, and the working medium load rate is 100%.
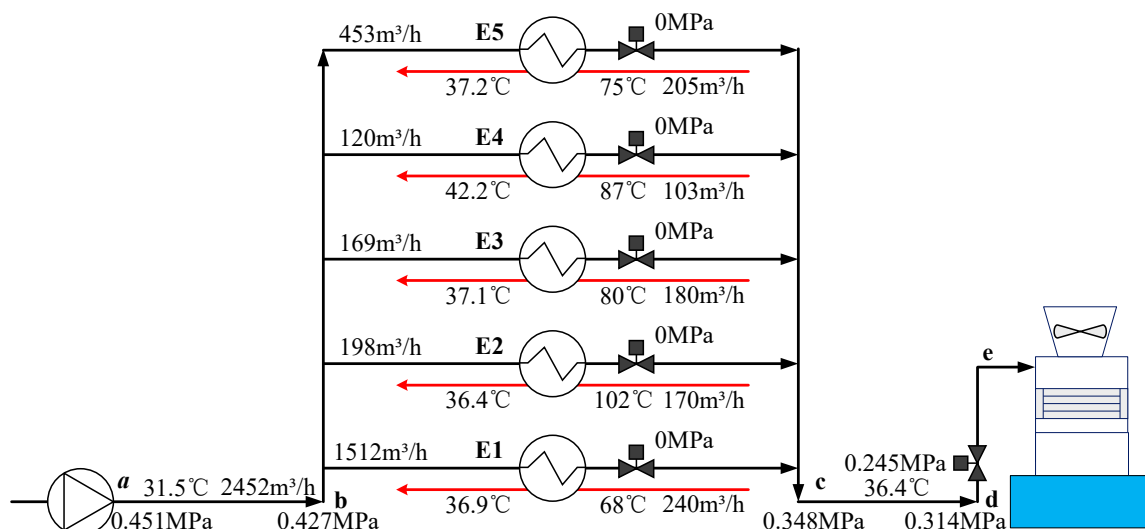

Figure 7. Original operating parameters of the system.

As a method study, we simplified the setting of various cost parameters, set the annual operation time of the system as 8000 h and set the electricity price purchased by the factory as 0.5 CNY/kW·h. The comprehensive efficiency of pumps and motors under different working conditions is 75%, and the comprehensive efficiency of turbines and generators is also 75%. The annual depreciation factor of the device for power generation using residual pressure is 0.2, the interest of bank loans is 15%, and the investment factor of turbines β_1 and β_2 are 117,340 and 0.54, respectively. The bypass filtration coefficient of hot return water is taken as 0.03, and the additional flow resistance caused by the transformation of power generation using residual pressure is taken as 1 m.

To illustrate the effectiveness of the joint regulation strategy of the pipeline and the pump station in analyzing the back pressure energy of the system, we designed a comparative simulation test, which mainly compares and analyzes the operation cost of the water supply network under extreme water consumption conditions after adding the hydraulic turbine generator in four different ways. The control range of water supply temperature is 25–33 °C, and the working medium load rate operation range is 95–105%.

Scheme 1: Leave the original operating parameters of the system unchanged and directly replace the upper tower valve of the cooling tower with a hydraulic turbine. Scheme 1 is adopted to control the system flow rate. Some operating parameters of the system under extreme conditions are shown in Table 2.

Table 2. Partial operating parameters of scheme 1 system.

Project	Minimum Flow Rate Condition	Maximum Flow Rate Condition
System flow rate (m³/h)	2452	2452
Pump head (m)	43.94	43.94
Pressure drop of heat exchanger unit (m)	8.01	8.01
Branch 1 flow rate (m³/h)	1512	1512
Branch 2 flow rate (m³/h)	198	198
Branch 3 flow rate (m³/h)	169	169
Branch 4 flow rate (m³/h)	120	120
Branch 5 flow rate (m³/h)	453	453
Turbine head (m)	24.94	24.94

Scheme 2: Optimize the flow rate by adjusting the valves in the pipe network. Then, transform the optimized system with the hydraulic turbine. The hydraulic parameters of each heat exchanger network can be determined according to Formulas (13)–(18) simultaneously, and then combined with Formulas (10) and (12) to determine the operating point of the pump. The head flow of the turbine is determined according to Formulas (32) and (33). Scheme 2 is used to control the system flow rate. Some operating parameters of the system under extreme conditions are shown in Table 3.

Table 3. Partial operating parameters of scheme 2 system.

Project	Minimum Flow Rate Condition	Maximum Flow Rate Condition
System flow rate (m³/h)	1534	1927
Pump head (m)	49.75	47.74
Pressure drop of heat exchanger unit (m)	5.37	7.61
Branch 1 flow rate (m³/h)	857	1130
Branch 2 flow rate (m³/h)	108	126
Branch 3 flow rate (m³/h)	113	153
Branch 4 flow rate (m³/h)	85	117
Branch 5 flow rate (m³/h)	371	401
Turbine head (m)	37.03	31.43

Scheme 3: Optimize the flow rate and pressure of the system by changing the rotating speed of the pump. Then, transform the optimized system with the hydraulic turbine. In Scheme 3, due to the hydraulic imbalance between heat exchangers, the flow of each branch is not less than the minimum demand flow.

$$Q_i \geq \max(Q_{heat_{Ei}}, Q_{fouling_{Ei}}) \tag{35}$$

The hydraulic parameters of each heat exchanger network can be simultaneously determined according to Formulas (13)–(15), (17)–(18) and (35), combined with Formulas (11) and (12) to determine the operating point of the pump. The head flow of the turbine is determined according to Formulas (32) and (33). Scheme 3 is used to control the system flow rate. Some operating parameters of the system under extreme conditions are shown in Table 4.

Table 4. Partial operating parameters of scheme 3 system.

Project	Minimum Flow Rate Condition	Maximum Flow Rate Condition
System flow rate (m^3/h)	2009	2391
Pump head (m)	35.05	37.98
Pressure drop of heat exchanger unit (m)	5.37	7.61
Branch 1 flow rate (m^3/h)	1239	1474
Branch 2 flow rate (m^3/h)	162	193
Branch 3 flow rate (m^3/h)	138	165
Branch 4 flow rate (m^3/h)	98	117
Branch 5 flow rate (m^3/h)	371	442
Turbine head (m)	20.65	19.67

Scheme 4: Optimize the flow rate and pressure of the system through the joint regulation strategy of the pipeline and the pump station proposed in this study. Then, transform the optimized system with the hydraulic turbine. The hydraulic parameters of each heat exchanger network can be determined according to Formulas (13)–(18) simultaneously, and then combined with Formulas (11) and (12) to determine the operating point of the pump. The head flow of the turbine is determined according to Formulas (32) and (33). Scheme 4 is used to control the system flow rate. Some operating parameters of the system under extreme conditions are shown in Table 5.

Table 5. Partial operating parameters of scheme 4 system.

Project	Minimum Flow Rate Condition	Maximum Flow Rate Condition
System flow rate (m^3/h)	1534	1927
Pump head (m)	34.35	37.15
Pressure drop of heat exchanger unit (m)	5.37	7.61
Branch 1 flow rate (m^3/h)	857	1130
Branch 2 flow rate (m^3/h)	108	126
Branch 3 flow rate (m^3/h)	113	153
Branch 4 flow rate (m^3/h)	85	117
Branch 5 flow rate (m^3/h)	371	401
Turbine head (m)	21.63	20.84

To compare the flow rate of the heat exchanger network, the heat exchanger network pressure drop, and the pump operation head under different regulation strategies, the three parameters are dimensionlessly processed by comparing them with the original system parameters, which are recorded as Q′, Δh′, and H′, respectively. Figure 8 shows the comparison of Q′, Δh′, and H′ in extreme conditions under the four schemes.
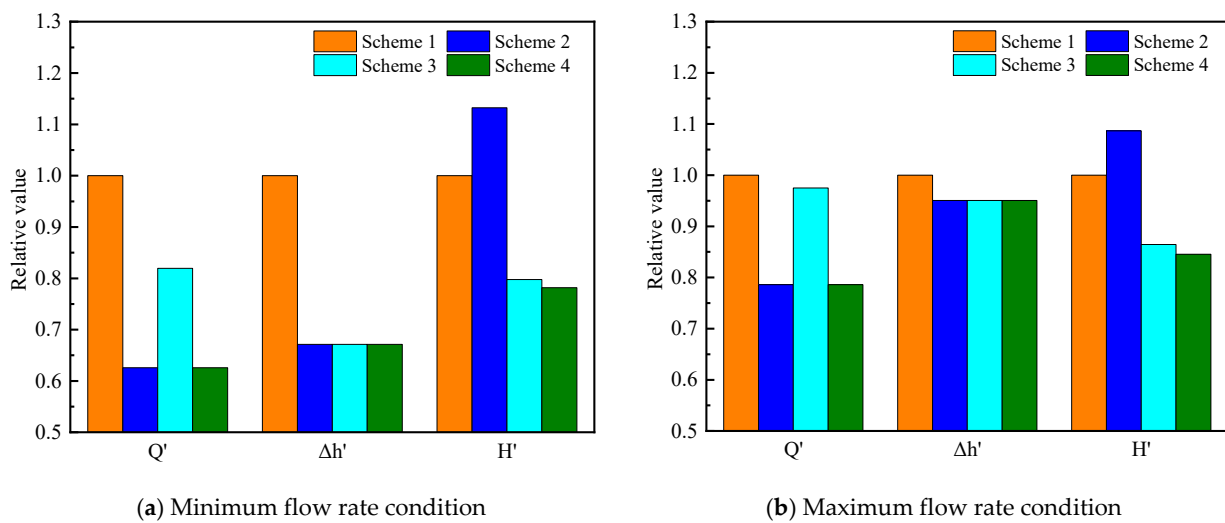
(**a**) Minimum flow rate condition

(**b**) Maximum flow rate condition

Figure 8. The comparison of Q′, Δh′ and H′ in extreme conditions under the four schemes.

In the four schemes, Q′, Δh′, and H′ have the same law under minimum flow rate conditions and maximum flow rate conditions. Under minimum flow rate conditions, the network flow rate of the heat exchanger, the pressure drop of the heat exchanger, and the pump operating head of the system with the adjusting strategy change more than the original parameters. Figure 8a,b show that the pressure drop of the heat exchanger network is equal when different regulation strategies are adopted for the system in schemes 2–4; however, the pressure drop of scheme 1 is much larger than that of schemes 2–4. When schemes 2 and 4 are adopted, the heat exchanger network flow rate remains the same, and the flow rate value is much smaller than that of schemes 1 and 3. Except for the pump operating head of scheme 2, the Q′, Δh′, and H′ of the system are less than 1 when the regulation strategy is adopted. In addition, compared with other methods, the heat exchanger network flow rate, the heat exchanger network pressure drop, and the pump operation head under the synchronous regulation strategy for the pipeline and the pump station proposed in this study are the smallest, thus being consistent with the hydraulic optimization regulation goal of the system.

Figure 9 shows the annual pumping cost, income from turbine power generation, and the total cost comparison of the four schemes under the maximum flow rate and the minimum flow rate. The pumping cost, the turbine power generation income, and the total cost under the four schemes have the same law under the maximum flow rate and the minimum flow rate. Compared with the maximum flow rate condition, the three economic indicators are smaller than the minimum flow rate condition under different regulation strategies.

From the perspective of pumping cost, the pipeline regulation strategy is adopted under the second scheme, which is a common regulation method used by industrial field engineers. Under this scheme, the system flow rate is optimized. Hence, the pumping cost is greatly reduced compared with the initial working condition. However, the operating head of the pump in this method is even higher than the initial parameters. Thus, the optimization of the pumping cost still has much room for improvement. The regulation strategy for the pump station is adopted under the third scheme, and the pumping cost of scheme 3 is further reduced compared with scheme 2. However, the third scheme optimizes the system pressure and flow rate from the upstream supply, while the optimization effect is still limited. The synchronous regulation strategy of the pipeline and the pump station (scheme 4) proposed in this study integrates the advantages of schemes 2 and 3, which can optimize the system flow rate and pressure most effectively. Thus, the pumping cost is the least. From the perspective of power generation income and total operation cost of hydraulic turbines, although the power generation income of hydraulic turbines in

scheme 1 is highest, the total operation cost of the system is much higher than that of the three schemes under the hydraulic optimization regulation strategy due to the high pumping cost. Although the power generation income of the water turbine in scheme 4 is the smallest, the total operating cost of the system has decreased more.
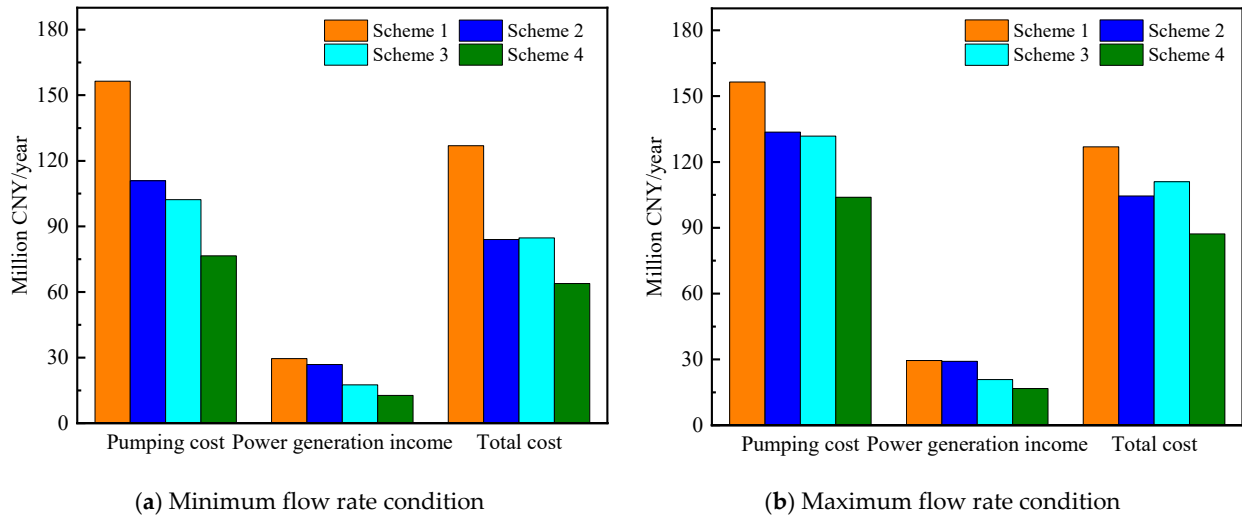

(**a**) Minimum flow rate condition

(**b**) Maximum flow rate condition

Figure 9. Comparison of operating costs.

Therefore, according to Figures 8 and 9, scheme 4 is the best scheme. The minimum heat exchanger network flow rate, the minimum heat exchanger network pressure drop, and the minimum pump operating head can be obtained by hydraulic optimization through the joint regulation strategy of pipeline and pump station, and the minimum pumping cost and the minimum total operation cost can be obtained.

According to the calculation, the annual total cost of the water pump under the initial parameters of the system is 1.564 million yuan. Table 6 shows the reduction rate of the total cost of the original system under the four schemes. It shows that the hydraulic parameters of the water supply network remain unchanged in scheme 1, and the operation and management investment of the system are minimum, but the total cost decline rate is also minimum. The total cost reduction rate under the maximum flow rate condition is much higher than that the minimum flow rate condition in schemes 2–4. When the water supply network is not optimized, the total cost can be reduced by 18.85% by adding hydraulic turbines. However, compared with the original system, the total cost of the water supply network can be decreased by 39.74%, 32.78%, and 47.89%, respectively, when the pipeline regulation strategy, the pump station regulation strategy, and the synchronous regulation strategy of the pipeline and pump station are adopted. This result not only proves the application value of the comprehensive analysis method proposed in this paper, but also proves the conclusion of the energy flow analysis in the previous article to be correct. In other words, when the CCWS is transformed for power generation using residual pressure by the hydraulic turbine, the hydraulic optimization and regulation should be carried out first to obtain the minimum flow rate of the heat exchanger network, the minimum pressure drop of the heat exchanger network, and the minimum operation head of the pump.

Table 6. The reduction rate of the total cost relative to the original system.

Project	Scheme 1	Scheme 2	Scheme 3	Scheme 4
Minimum flow rate condition	18.85%	33.22%	29.01%	44.23%
Maximum flow rate condition	18.85%	46.27%	36.54%	51.56%
Average value	18.85%	39.74%	32.78%	47.89%

6. Conclusions

In conventional CCWS, the residual pressure of the return water in the system will be wasted by the valve on the cooling tower. In this study, the hydraulic turbine is used to replace the pressure reducing effect of the upper tower valve. Combining the advantages of the traditional pipeline regulation strategy and the pump station regulation strategy, this research puts forward the synchronous regulation strategy of the pipeline and the pump station. Then, based on the synchronous regulation strategy of the pipeline and the pump station, a comprehensive feasibility analysis method of the CCWS's power generation using residual pressure is proposed. Finally, taking CCWS as an example, the simulation and comparison experiments of the transformation of residual pressure power generation in four ways of the system are designed, which proves the comprehensive method to be correct. The main conclusions of this study are as follows:

(1) The energy consumption of the water pump should be considered at the same time when the system is transformed into the hydraulic turbine. Through the analysis of system energy flow, it can be known that in order to obtain the minimum total energy consumption of the pump and the turbine, the hydraulic optimization and regulation should be carried out for the system to obtain the minimum flow rate of the heat exchanger network, the minimum pressure drop of the heat exchanger network, and the minimum operation head of the pump.

(2) The pipeline regulation strategy can obtain the minimum flow rate and the pressure drop of the heat exchanger that meets the production. However, it cannot obtain the minimum pump operation head. Although the pump station regulation strategy can continuously regulate the system flow rate and reduce the pump operating head at the same time, the system flow rate margin of this method is large, and the minimum system flow rate and pump head cannot be obtained. The synchronous regulation strategy of the pipeline and the pump station proposed in this study combines the advantages of two traditional regulation strategies. Moreover, it can obtain the lowest flow rate, pressure drop of the heat exchanger network, and the pump operation head at the same time.

(3) For the existing CCWS, the energy condition of the power generation using residual pressure is that the maximum height of each branch of the heat exchanger network is larger than the sum of the cooling tower height and the return header head loss. In the economic analysis of the power generation using residual pressure, the minimum and maximum water supply conditions must be considered at the same time. The case study results show that the comprehensive method proposed in this research can save 29.04% of the annual pump cost compared with the direct transformation of the power generation using residual pressure. Moreover, it can save 8.15% and 15.11% of the annual pump cost compared with the two traditional regulation strategies, respectively.

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