Energy Saving Application of Variable Speed Auxiliary Pump Plus Hydro Turbine in Circulating Cooling Water System

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Abstract. The circulating cooling water system (CCWS) is a common industrial auxiliary system, and water pumps need to consume much energy to transport cooling water to distributed heat exchangers. Previous studies show that a pump network configuration with constant speed auxiliary pump plus hydro turbine (CSAP-T) plays a significant role in energy conservation. However, given the fluctuations in the production load and cooling water supply temperature, the flow demand of heat exchangers for cooling water varies. Under different working conditions, the CSAP-T scheme cannot supply the minimum cooling water flow required by all heat exchangers at the same time, thereby wasting energy. Therefore, this paper proposes a new fluid machinery network configuration called variable speed auxiliary pump plus hydro turbine (VSAP-T) and establishes a mathematical model of the total output power of the fluid machinery network in CCWS to minimize network energy consumption. To illustrate the effectiveness of the VSAP-T scheme, a real system is used as the research object, and both CSAP-T and VSAP-T schemes are used to optimize the system. When the working conditions are changed, the total output power of the system fluid machinery network can be conserved by 30% to 50%.

1. Introduction
The circulating cooling water system (CCWS) is composed of three functional modules: a pump network, a heat exchanger network, and a cooling tower network. The water pump serves as a power source that delivers cooling water flow to the distributed heat exchangers, consuming a large amount of energy in the process. Statistics show that pump systems consume approximately 20% of the global energy supply [1]. Therefore, a proper pump network should be designed to effectively reduce the operating cost of CCWS.

Many scholars view energy recovery and utilization as potential solutions to the increasing consumption of network energy, and using hydro turbines to recover the residual pressure of the pipe network has begun to attract attention [2, 3]. 2017, Gao, W. et al. [4] labelled the network containing auxiliary pumps and turbines as the fluid machinery network in CCWS and introduced the effective height curve of branches and cooling towers, which can be used to guide the settings of pumps and turbines. Ma et al. [5] analyzed two cases and found that hydro turbines have a greater energy-saving potential than auxiliary pumps and that the network structure with auxiliary pumps and turbines is the most energy-efficient configuration.

Researchers in [4, 5] adopted the constant speed auxiliary pump plus hydro turbine (CSAP-T), a fluid machinery network configuration where the flow rate of each heat exchanger can only be controlled by adjusting the system pressure difference. In this way, the cooling water flow rate of each branch changes at relatively the same rate. However, given the influence of several factors, such as the heat
exchanger structure parameters and pipeline conditions, the cooling water flow rate of each branch changes at relatively the different rate.

Under this background, this article considers the flow demand changes under different working conditions of a heat exchanger, proposes the variable speed auxiliary pump plus hydro turbine (VSAP-T) fluid machinery network configuration in CCWS, and established a mathematical model of the total output power of the network under this configuration. When using the VSAP-T scheme, the operator can control the cooling water flow of each branch according to the actual demand of the heat exchanger by adjusting the speed of the auxiliary pump.

2. CCWS superstructure model configured by VSAP-T

This article stipulates that the production load is calibrated according to the working fluid flow and that the heat exchanger design working fluid flow represents 100% of the production load. The CCWS upper structure of the proposed VSAP-T scheme is shown in Figure 1. The upper structure includes all possible situations of the VSAP-T configuration. In the upper structure, \( i \) is the index of the branch in the water supply network, \( j \) is the index of the heat exchanger in each branch, \( M \) represents the working fluid flow of the heat exchanger, and \( t_{in} \) is the temperature of the cooling water supply. When either \( M \) or \( t_{in} \) changes, the flow demand of the heat exchanger for cooling water may also change. \( Q_i \) is the cooling water flow of branch \( i \), \( Q_{total} \) is the cooling water flow of the main pipe, and \( Q_{total} \) is the sum of all branch flows. The upper structure sets a variable-speed auxiliary pump plus hydro turbine for each branch. The variable-speed auxiliary pump plus hydro turbine replace the valves to maintain the hydraulic balance of each branch while simultaneously recovering the pressure energy loss caused by these valves. Given that the throttle valves of the operating equipment are fully open, all valves in the pipe network are eliminated in the superstructure. The cooling water enters the pipe network from position 0 and flows out from position 3. Positions 1 and 2 are the diversion and confluence points of parallel branches, respectively. VSP\( i \) refers to the variable speed auxiliary pump close to position 1 on branch \( i \), and VSP0 refers to the variable speed main pump close to position 0 on the water supply main road. Wi refers to the hydro turbine at position 2 on branch road \( i \), and W0 refers to the hydro turbine on the main return road. Variable-speed auxiliary pumps with the same pump head on the branch can be combined to provide a unified water supply, and those hydro turbines with the same recyclable pressure head on the branch can also be combined to generate electricity with a hydro turbine.

![Figure 1. CCWS superstructure of the VSAP-T configuration](image-url)
3. Mathematical model

3.1. Target function
The VSAP-T scheme aims to minimize the total output power of the fluid machinery network in CCWS. The difference between the total output power of each branch water pump and the total recoverable power of the hydraulic turbine is the total output power \( P^{\text{target}} \) of the entire fluid machinery network.

\[
\text{Obj} = \text{MIN}(P^{\text{target}}) = \text{MIN}(P_P - P_W)
\]

(1)

where

\[
P_P = \sum \rho g Q_i H_{p,i} / \eta_{Mi} \eta_{Pi}
\]

(2)

\[
P_W = \sum \rho g Q_i H_{t,i} / \eta_{Wi} y_i
\]

(3)

In the above equation, \( i \) is a non-negative integer (where \( i = 0 \) indicates that the equipment is on the main pipeline, whereas \( i > 0 \) indicates that the equipment is on each branch pipeline), \( H_{p,i} \) is the operating head of the pump on pipeline \( i \), and \( H_{t,i} \) is the operating head of the hydro turbine on pipeline \( i \). \( \eta_{Pi} \) is the operating efficiency of the variable-frequency water pump on pipeline \( i \), \( \eta_{Mi} \) is the operating efficiency of the motor supporting the pump on pipeline \( i \), and \( \eta_{Wi} \) is the comprehensive power generation efficiency of the hydro turbine on pipeline \( i \). \( y_i \) is a binary variable that determines whether a hydro turbine is available on the pipeline.

3.2. Variables
The variables include the flow rate \( Q_i \) of each pipeline, the head \( H_{p,i} \) and efficiency \( \eta_{Pi} \) of the pump on each pipeline, and the head \( H_{t,i} \) of the hydro turbine on each pipeline. Each parallel branch is equipped with a variable speed auxiliary pump. \( y_i \) takes a value of either 0 or 1, where \( y_i = 0 \) indicates that no hydro turbine is set and \( y_i = 1 \) indicates that a hydro turbine needs to be set.

3.3. Constraints
The pressure head of the variable speed auxiliary pump needs to meet the following constraints:

1. The pressure at position 3 where the cooling water enters the cooling tower should be greater than or equal to 0:

\[
H_{p,i} + h_c - (h_{f0,i} + h_{f1,i} + h_{f2,i}) \geq 0
\]

(4)

where \( h_c \) is the height of the liquid level of the cooling water sump, \( h_{f0,i} \) is the total pressure head loss of the pipeline and heat exchanger of branch \( i \), and \( h_{f1,i} \) is the height of the cooling water entering the cooling tower. \( h_{f0,i} \) and \( h_{f2,i} \) represent the head loss of the main pipe section from positions 0 to 1 and the head loss of the main pipe section from positions 2 to 3, respectively.

2. To prevent pipeline vibration due to negative pressure, the cooling water pressure at the high point of each branch should be greater than or equal to 0:

\[
H_{p,i} + h_c - (h_{f0,i} + h_{f1,i} + h_{f2,i}) \geq 0
\]

(5)

where \( h_{h,i} \) is the branch height.

The sum of the cooling water flow of each branch should be equal to the main flow:

\[
\sum_{i=1}^{n} Q_i = Q_{\text{total}}
\]

(6)

To ensure the normal operation of the heat exchanger, the outlet temperature of the working fluid should not exceed the control index. \( T_{2,ij}^{\text{index}} \) is a fixed temperature determined by the Eij process conditions.

\[
T_{2,ij} \leq T_{2,ij}^{\text{index}}
\]

(7)
To prevent the fouling rate from increasing due to a too low flow rate, the flow rate of the cooling water in the heat exchanger tubes should not be set too small.

$$Q \geq \kappa_{ij} \pi d_{ij}^2 v_{\min} / 4N_{pij}$$ (8)

where $\kappa_{ij}$ is the number of tube in $E_{ij}$, $d_{ij}$ is the inner diameter of $E_{ij}$, and $N_{pij}$ is the number of tube pass in $E_{ij}$. $v_{\min}$ is the minimum flow rate requirement of the cooling water to prevent fouling in the internal tubes of the heat exchanger.

3.4. Branch flow $Q_i$

The heat transfer between the hot and cold fluid inside the heat exchanger satisfies the following heat balance equation:

$$\phi_{ij} = C_{\text{hot}_{ij}} M_{ij} \Delta T_{ij} = C_{\text{cool}_{ij}} Q_{ij} \Delta t_{ij}$$ (9)

where $C_{\text{hot}_{ij}}$, $M_{ij}$, and $\Delta T_{ij}$ are the specific heat capacity, flow, and temperature difference of the working fluid of the heat exchanger, and $C_{\text{cool}_{ij}}$, $Q_{ij}$, and $\Delta t_{ij}$ are the specific heat capacity, flow, and temperature difference of the heat exchanger cooling water.

The heat transfer of the heat exchanger is related to the following design factors:

$$\phi_{ij} = K_{ij} \Delta t_{m_{ij}} A_{ij}$$ (10)

$$\Delta t_{m_{ij}} = (\Delta t_{\text{max}_{ij}} - \Delta t_{\text{min}_{ij}}) / \ln(\Delta t_{\text{max}_{ij}} / \Delta t_{\text{min}_{ij}})$$ (11)

where $K_{ij}$ is the total heat transfer coefficient of the heat exchanger, $A_{ij}$ is the heat transfer area, and $\Delta t_{m_{ij}}$ is the logarithmic average temperature difference. $\Delta t_{\text{max}_{ij}}$ and $\Delta t_{\text{min}_{ij}}$ refers to relatively large and small temperature differences between both ends of the heat exchanger, respectively.

In state $a$, the flow on the working fluid side of $E_{ij}$ is denoted by $M_{a_{ij}}$, whereas that on the cooling water side is denoted by $Q_{a_{ij}}$. At this time, the total heat transfer coefficient of $E_{ij}$ can be expressed as

$$K_{a_{ij}} = 1 / \left( \frac{1}{K_{\text{shell}_{ij}}(Q_{a_{ij}}/Q_{a_{ij}})^m} + r_{\text{fouling}_{ij}} + \frac{1}{K_{\text{tube}_{ij}}(Q_{a_{ij}}/Q_{a_{ij}})^{m'}} \right)$$ (12)

where $M_{a_{ij}}$ and $Q_{a_{ij}}$ are the design flows of the working fluid of the heat exchanger and cooling water, respectively, $K_{\text{shell}_{ij}}$ and $K_{\text{tube}_{ij}}$ are the design convective heat transfer coefficients on the shell and tube sides, respectively, and $r_{\text{fouling}}$ is the fouling resistance of the heat exchanger.

When the heat exchanger is cleaned, the fouling thermal resistance takes the design value. When the heat exchanger has been running for a long time, the fouling thermal resistance can be calculated and determined based on operating data (i.e., four temperatures on the working fluid and cooling water sides, working fluid, or cooling water flow). $m$ and $m'$ are the correction coefficients of the heat transfer coefficient on the working fluid and cooling water sides with respect to flow change. According to [6], $m$ takes a value of 0.55, whereas $m'$ takes a value of 0.8.

All heat exchangers on the same branch have the same flow:

$$Q_i = Q_{ij}$$ (13)

The flow ($Q_i$) of the cooling water on the branch road needs to meet the heat exchange and flow rate requirements of all heat exchangers on the branch road.
3.5. Branch water pump head $H_{p,i}$ and hydro turbine head $H_{t,i}$

3.5.1. Branch water pump head $H_{pump,i}$

$$H_{p,i}=\max(H_{tower,i}, H_{b,i})$$  (14)

where $H_{tower,i} = h_{tower} + h_{f_0-1} + h_{f_1} + h_{f_2-3} - h_c$ and $H_{b,i} = h_{f_0-1} + h_{b,i} + h_{f_1} - h_c$. $h_{fi}$ can be expressed as

$$h_{fi}=k_{f,i,line}Q_i^2 + \sum_{j=1}^{m} k_{fij}Q_j^2$$  (15)

where $k_{f,i,line}$ and $k_{fij}$ are the total flow resistance coefficients of the pipes and heat exchangers, respectively.

3.5.2. Head of branch turbine $H_{t,i}$

Starting from the 0 point of the cooling water inlet, the point pressure of node 2 along the direction of water flow is calculated as

$$P_{2+}=H_{p,i} + h_c - h_f0-1 - h_{fi}$$  (16)

Starting from point 3 of the cooling water outlet, the pressure at node 2 against the water flow direction is calculated as

$$P_{2-}=h_{tower} + h_{f2-3}$$  (17)

The hydro turbine can only be set when $P_{2+} > P_{2-}$. The head of the branch turbine can be expressed as

$$H_{t,i}=P_{2+} - P_{2-}$$  (18)

When $H_{p,i} = H_{tower}$,

$$H_{t,i}=H_{tower} - h_f0-1 + h_c - h_f1 - h_{tower} - h_{f2-3} = 0$$  (19)

When $H_{p,i} = H_{b,i}$,

$$H_{t,i}=H_{b,i} - h_f0-1 + h_c - h_f1 - h_{tower} - h_{f2-3} = h_{b,i} - h_{tower} - h_{f2-3}$$  (20)

The above formula shows that the head of the hydro turbine changes along with the head loss $h_{f2-3}$ of the main return water pipeline. Therefore, the necessary conditions for installing a hydro turbine on the branch road are

$$h_{b,i} > h_{tower} + h_{f2-3}$$  (21)

3.6. Frequency conversion pump efficiency

The efficiency $\eta_N$ of the pump after speed regulation can be solved as [7]

$$H = a_2Q^2 + a_1Q + a_0$$  (22)

$$\frac{H}{Q^2} = \frac{H_N}{Q_N^2}$$  (23)

$$\eta_N = b_2Q^2 + b_1Q + b_0$$  (24)

where $a_0$, $a_1$, and $a_2$ are the $H$-$Q$ hydraulic characteristic curve coefficients of the pump, $b_0$, $b_1$, and $b_2$ are the pump $\eta$-$Q$ hydraulic characteristic curve coefficients, and $Q_N$ and $H_N$ are the running flow and running head of the water pump after speed regulation, respectively.
4. Case study

4.1. System description
This article is based on a real system in Nanjing, China. The operating data of the system in May this year is shown in Figure 2. The working fluid flow of each heat exchanger in Figure 2 is the design value. The cooling water is used by five heat exchangers with different structural parameters as shown in Table 1. The liquid level of the cooling water sump is flushed with the center of the main pipeline water pump, and this level is used as the reference height.

| Ei | d (m) | L (m) | n | Np | A (㎡) |
|----|------|------|---|----|-------|
| E1 | 0.019| 12.9 | 1930 | 1   | 1602  |
| E2 | 0.019| 12.6 | 1750 | 1   | 1413  |
| E3 | 0.019| 11.0 | 1391 | 1   | 980   |
| E4 | 0.019| 10.8 | 923  | 1   | 639   |
| E5 | 0.020| 6.0  | 872  | 1   | 399   |

4.2. Optimization plan
The working fluid flow rate of the heat exchanger and the temperature of the cooling water supply are two important factors that affect the total power of the pump network. First, the fluctuation range of these factors needs to be determined. According to the historical operating data of the system over the past 2 years, the cooling water supply temperature varies from 25 °C to 33 °C throughout the year, whereas the working fluid flow of the heat exchanger varies from 90% to 110% of the design value.

To illustrate the effectiveness of the proposed model, the existing system is optimized by using two optimization schemes of different network configurations. Scheme 1 adopts the CSAP-T pump network configuration. Meanwhile, Scheme 2 adopts the VSAP-T pump network configuration and installs a variable speed auxiliary pump on all branches.
4.3. Comparison of VSAP-T and CSAP-T under variable conditions

4.3.1. Comparison of the system flow

Figure 3 shows the changes in the system flow along with changing water supply temperature when VSAP-T and CSAP-T are used at production loads of 90%, 95%, 100%, 105%, and 110%. Figure 3 shows that the system flow follows the same changing law for both schemes. In a certain lower water temperature section, the cooling water flow required by the heat exchanger for heat exchange is less than the flow corresponding to the lowest flow rate, and an increase in water temperature or production load will not increase the system flow. When the cooling water flow required for heat exchange is greater than the flow corresponding to the lowest flow rate, the system flow gradually increases along with water temperature or production load. By contrast, VSAP-T increases the flow around 28 °C, whereas the CSAP-T scheme only increases the flow when the water supply temperature exceeds 30 °C.

Figure 3. System flow at different production loads and water supply temperatures

Figure 4 shows the system flow and reduction rate when the VSAP-T and CSAP-T schemes are used. In Figure 4(a), the production load is fixed at 110%, whereas in Figure 4(b), the water supply temperature is fixed at 33 °C. Compared with that in CSAP-T, the system traffic in VSAP-T has been significantly reduced. Under different working conditions, VSAP-T achieves a system flow reduction rate of 16% to 30%. A relatively high reduction rate is achieved when the water supply temperature is either low (under a constant production load) or the production load is either high (under a constant water supply temperature).

Figure 4. System flow and reduction rate during the optimization of VSAP-T and CSAP-T
4.3.2. Comparison of the total output power of the fluid machinery network

Comparison of the total output power of the fluid machinery network Figure 5 shows the total output power and reduction rate of the fluid machinery network when the VSAP-T and CSAP-T schemes are adopted. A lower water supply temperature and production load corresponds to a smaller total output power of the fluid machinery network. Comparing the two solutions, under different working conditions, the total output power of the network using VSAP-T is significantly lower than that of the network using CSAP-T. The total output power of the network is reduced by a minimum of 30% and a maximum of 50%.

Figure 5. Total network output power and reduction rate when VSAP-T and CSAP-T are used for optimization

5. Conclusion

This paper proposes VSAP-T, a new configuration of the fluid machinery network in CCWS. Afterward, based on the superstructure method, a mathematical model of the total output power of the fluid machinery network in CCWS is established. The following conclusions are drawn from the case simulation results:

(1) With the increase of water supply temperature and production load, the system flow follows the same changing law for both schemes, and the water-saving effect of the VSAP-T scheme is much better than that of the CSAP-T scheme.;

(2) With the increase of water supply temperature and production load, the total output power of the fluid machinery network follows the same changing law for both schemes. Compared with that in CSAP-T, VSAP-T can reduce the total output power of the fluid machinery network to a lower level compared with CSAP-T;

(3) Given that controlling the water supply temperature has a relatively large impact on the parameters of both CSAP-T and VSAP-T, controlling the appropriate water supply temperature is essential for system energy saving.

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