INTRODUCTION

1.1 Background

In terms of building energy consumption, the heating, ventilation, and air-conditioning (HVAC) system has been the major energy consumer of large public buildings. The HVAC system alone accounts for more than 50% of the total energy consumption of large public buildings. Considering the whole HVAC system, the operating energy consumption of the water system accounts for about 60%-80% of the total energy consumption. The cooling water system is an important part of the water system of HVAC system, accounting for 15%-20% of the total energy consumption of the HVAC system. Thus, it is of great significance to study the optimization method of the cooling water system for the energy saving in HVAC system.
The main components that consume energy in the HVAC water system are the chiller, chilled water pump, cooling water pump, and the cooling tower fan. The chilled water carries heat from the terminals and delivers it to the cooling water via the circulation of refrigerant. Then, the cooling water with high temperature is pumped into the cooling tower to have heat exchange with outdoor air and back to the condenser. Thus, the cooling water system is an indispensable part of the HVAC system in the elimination of the indoor cooling load.

The optimal control in the cooling water system is complicated and interconnected among its components. Thus, it is not reasonable to pursue the optimal control effect of a local component and ignore the whole system's control effect. Moreover, many factors can cause system disturbance and impact the energy consumption of the system, such as the variation of cooling load and the changes in outdoor environment. Commonly, the predictive control method is often adopted to optimize the performance of the system. Thus, it is appropriate to build a simplified gray box model for the whole system to optimize the global performance of the cooling water system by considering practicality and feasibility. The goal of the optimization is to minimize the energy consumption of the system on the premise of satisfying the cooling demand. Meanwhile, the optimization variables should be representative and controllable, such as the cooling water flow and the air volume of the cooling tower. The cooling load and outdoor wet-bulb temperature will be the disturbance variables.

Conventional optimization of cooling water system has mainly focused on a single component and seldom considered the performance of the whole system. A large and growing body of literature has investigated the characteristics of the components of the cooling water system, including cooling towers, chillers, and pumps.

1.2 Literature review

As early as 1925, research on thermal characteristics of cooling towers had been conducted by Merkel, and the developed methods and empirical equations are still used up to now. Later, Sutherland considered the evaporation loss of cooling water and proposed a new model of cooling tower by comparing with the enthalpy difference model developed by Merkel. The experimental results showed that the model was accurate with an acceptable error. Braun conducted a detailed analysis on cooling towers and studied the model of heat exchange efficiency by presuming a linear saturated enthalpy, using saturated specific heat constant \( C_s \) as the corrected effectiveness definition. Jaber and Webb proposed the effectiveness-number of heat transfer unit \( (\varepsilon \cdot \text{NTU}) \) design method to give guidance for designers of cooling towers.

In their research, the evaporation loss of water was neglected, and the Lewis coefficient was assumed as a constant. Poppe and Rögener developed an accurate and comprehensive cooling tower model, which was complicated in calculation and relied on iterative computations. Kloppers and Kröger discussed the difference among the Poppe model, Merkel model, and \( \varepsilon \cdot \text{NTU} \) model. Their study reported that the \( \varepsilon \cdot \text{NTU} \) was the most widely used model due to its simplified principle. They also found that the predicted outlet water temperatures of the three models were consistent, but the outlet air temperatures were not. The authors further exhibited that the Poppe model showed the highest accuracy for the outlet air temperature. Hernández-Calderón et al simplified the differential equations of the Poppe method and solved the heat and mass transfer equations by the orthogonal collocation method. The comparative results in their report showed that the orthogonal collocation method requires less computing time for similar accuracy.

It can be learnt that most studies mainly focused on the research on thermal characteristics of the cooling tower and the mathematical model of heat transfer. Nevertheless, the application of the mathematical model of heat transfer to the optimization of cooling water system is too complicated and not conducive to the online optimization control. Therefore, it is necessary to seek a suitable cooling tower model for the system optimization control.

Research on chiller models has mainly focused on the methodology of modeling and identification since the 1990s. One of the most classical chiller models was the “GN model” proposed by Gordan et al., which is simple in structure and has been verified by both theory and experiment. It is noteworthy that the GN model is still used till now. Allen and Hamilton developed a steady-state model for chiller based on the energy conservation equation, in which the cooling capacity and the power of chiller are expressed as the polynomial equation of chilled and cooling water temperatures. Swider et al adopted a generalized radial basis function neural network to simulate and predict the chiller performance for fault detection and diagnosis purposes. Chen et al combined neural networks and particle swarm optimization algorithm (NNPSO) to simulate and optimize the chiller power consumption. The results showed that the NNPSO method exhibited a considerable power-saving ability compared with the linear regression and equal loading distribution methods. Kim et al developed an analytical model for the adsorption chiller to estimate the mass diffusivities and adsorption capacities of adsorbents. Nasruddin et al selected the parameters of global horizontal radiation and dry-bulb temperature as the predictors to study the prediction performances of three types of artificial neural networks of chiller.

Large contributions on the components of cooling water systems have been made by the literatures above. However, most of the optimization methods mentioned in the literature
have focused on the accuracy of the model and ignored the practicability and feasibility, which rendered the method complicated and difficult to implement in a practical system. The difficulty in achieving optimal control in cooling water systems can be attributed to the following two aspects of limitation.

Firstly, the model of controlled object is complex with many control variables. An accurate model is significant for the optimal research of cooling water system. In all the components of cooling water system, the cooling tower model is the most complex. Recently, the most adopted cooling tower model is the component-based model proposed by Braun in 1989. However, this model is not only complicated in expression, but also inconvenient in calculation, which usually relies on iterated method. The heat exchange between cooling water and surrounding air is not only a simple heat transfer problem, but also involves mass transfer. The performance of cooling tower is closely related to the ambient temperature (wet-bulb temperature). In addition, there are many mathematical models of the chiller, including many empirical models, semi-empirical models, and theoretical models. Selecting, simplifying, and modifying these mathematical models, and matching the models of various components also make system optimization more difficult.

Secondly, since the control of the components in the cooling water system is coupled with each other, there is a balance between local and global optimization. There are many interference factors and operating conditions in controlling cooling water systems. The change in load and outdoor environment will also influence the control effect. Moreover, each restricts and influences and any parameter change among components will affect the performance of the whole system. For example, the frequency conversion of cooling tower fan and cooling water pump will affect the return temperature of cooling water, which is in turn affect the performance of the chiller. In addition, there are interactions between the fan and the pump, which also have an effect. Therefore, it is significant to study how to minimize the total energy consumption of these components on the premise of cooling demand.

Currently, more and more researchers have commenced to study the cooling water system synthesis and to focus on the global performance of the system. Lu et al.18 proposed a modified genetic algorithm for the cooling water system of HVAC system, and the experimental results demonstrated that the optimization strategy can adequately save the operating cost of the cooling water system. Kim et al.19–21 conducted a series of investigations on cooling water system synthesis by presenting systematic methodologies, which were meaningful for the design of cooling tower and cooling water networks. The inadequacy of their research was that they, respectively, studied the cooling tower and cooling water networks, which might lead to suboptimal results.22 Accordingly, a holistic consideration of the cooling water system was presented in Majozi and Moodley’s research by combining the cooling towers and cooling water networks as a whole. A mixed integer nonlinear programming (MINLP) model was presented for targeting and design in the cooling water system with multiple cooling towers. The results showed that the global optimality only existed when the system was studied holistically. Zheng et al.24 presented a MINLP formulation for the cooling water system based on the superstructure description by regarding the pump network, cooling water network, and cooling tower as a whole system; the optimum was obtained in the case study. Sun et al.25 studied the optimization problems of cooling water system by using a novel two-step sequential methodology, in which the thermodynamic model of the cooler network and the hydraulic model of the pump network were established successively. The effectiveness of the proposed method was demonstrated in a case study by saving 23.3% of the cooler network cost and 11% of the pump network cost. Liu et al.22 proposed an integrated methodology of simultaneous optimization of the total annual cost (TAC) of both the cooling water system and the heat exchanger network based on the MINLP model. The results of an industrial case study showed that the proposed methodology had 4.5% reduction in TAC compared with the conventional one. Cheng et al.26 proposed a robust optimal methodology for the design of cooling water system to minimize the TAC based on sequential Monte Carlo simulation. The results of their case study demonstrated that the minimum TAC could be realized in different operating conditions. Similarly, Ma et al.27 presented an optimization MINLP model for multiplants cooling water system by using a novel pump network to minimize the system TAC, and verified the optimal effectiveness of the model in two case studies. Meanwhile, other research from the same group studied the cooling water system by optimizing the water coolers, air coolers, pumping networks, and cooling towers simultaneously, in which a MINLP model was also formulated to solve the problem.

Although all these research studies have led to significant improvements on designing and operating cooling water system, the complicated implemented processes render these methodologies and models challenging for online application in a practical plant. Due to the complexities of cooling system, the optimal control method is rarely applied in practical engineering. Thus, it is significant to study the optimal control method of cooling system by considering the practicability and feasibility.

1.3 | Research objectives

In this paper, we focus on the global optimization issue of cooling water system by regarding the system as a whole. To improve the feasibility and practicality of the control method, firstly, the mathematical models of chiller, power of fan and
pump, and outlet temperature of cooling tower are simplified on the premise of easy implementation, and the model parameters are identified based on real experimental data. Then, the objective function of the whole cooling water system is established on a simulation platform, and the power consumption of the variable flow cooling water system is analyzed with a given cooling load and outdoor wet-bulb temperature. By analyzing the effects of cooling tower airflow and water flow to the power consumption of cooling water system, the water flow optimization control method is proposed. The method aims to find out an optimum cooling water flow to minimize the system power consumption while the fan operates at full speed, which could simplify the coupling relationships among multicontrol loops of cooling water system. Besides the optimization of power consumption, it also suggests the fans operate at full speed and without the installation of variable speed drivers to reduce the initial investment of system. Compared to the constant temperature difference control method and the constant speed control method, the proposed control method exhibits acceptable performance on energy savings of cooling water system and is feasible to be implemented in the actual engineering.

2 | MODELLING OF COOLING WATER SYSTEM

The cooling water system consists of a chiller, a cooling water pump, and a cooling tower. The specifications of the main components of the cooling water system are listed in Table 1.

The sample data used for the identification of the model were obtained from the experiments on the system and collected by the monitoring system of the cooling water system. The controller of the monitoring system is the XL100 of Excel 5000 series provided by Honeywell. The data between the controller and devices (sensors and actuators) were transmitted via the field bus called LON-Bus. The control schematic diagram of the cooling water system is shown in Figure 1. The power information of the cooling water system was collected by a multifunctional data collector called SUN-DAU, which can collect the power information from meters with MODBUS RS485 protocol.

### Table 1: The main components and specifications of the cooling water system

| Objective       | Brand                | Type                        | Specification                                      |
|-----------------|----------------------|-----------------------------|---------------------------------------------------|
| Chiller         | Trane WPWE0805 heat pump | Water source heat pump       | Cooling capacity: 19.0 kW; nominal power for cooling: 4.8 kW Heating capacity: 29.5 kW; nominal power for heating: 7.9 kW |
| Cooling pump    | Wilo MHI 805-1/10/E/3-380-50-2 | Variable frequency pump     | Lift: 59.0 m; nominal power: 1.85 kW; rated flow: 12 m³/h |
| Cooling tower   | Liang Chi LBCM-5     | Variable frequency fan       | Water flow: 6 m³/h; fan power: 0.12 kW             |

2.1 | Chiller

Generally, the chiller models can be classified into three types: theoretical model, empirical model, and semi-empirical model. The establishment of the theoretical model involves the internal parameters of the chiller, and it is necessary to establish the models of the evaporator, compressor, condenser, throttle valve, and other components. In this case, the model is too complicated for online optimal control to save energy.

The empirical model adopts the method, such as the least squares method, to establish the relationship between input and output based on the experimental data. This method is simple and easy to implement with a higher accuracy in fitting results, but it lacks theoretical basis and general applicability.

A semi-empirical model is used to determine the basic mathematical model through theoretical analysis of equipment, and the unknown parameters in the model can be identified through experimental data. This method only needs to identify the corresponding unknown parameters according to the actual operation data of the equipment, which ensures its general applicability. It is also a simplification of the theoretical model and is very suitable for system optimization diagnosis.

The semi-empirical model is adopted by most of the researchers due to its advantages of universality and veracity. In our simulation research, the chiller model was established based on the GN model, which is also a semi-empirical model proposed by J. M. Gordon. The model can be expressed as follows:

\[
\frac{1}{\text{COP}} = -1 + \left( \frac{T_{ci}}{T_{eo}} \right) + \left( \frac{1}{Q_e} \right) \left( q_c T_{ci} - q_e \right) + f_{HX}
\]

where \( T_{ci} \) is the inlet temperature of condenser, °C; \( T_{eo} \) is the outlet temperature of evaporator, °C; \( Q_e \) is the cooling capacity of chiller, kW; \( q_e \) is the heat loss of evaporator, kW; \( q_c \) is the heat loss of condenser, kW; and \( f_{HX} \) is the dimensionless term defined by J. M. Gordon; the coefficient of performance (COP) can be defined as follows:

\[
\text{COP} = \frac{Q_e}{P_{in}}
\]
where $P_{in}$ is the input power of chiller, kW. Then, the relationship among power, cooling capacity, inlet temperature of condenser, and outlet temperature of evaporator can be obtained, which can be expressed as follows:

$$P_{in} = Q_e + q_c \left( \frac{T_{ci}}{T_{eo}} + (f_{HX} - 1) \right) Q_e - q_c$$  \hfill (3)$$

In Equation (3), if we set the heat loss $q_c$ and $q_e$, and $f_{HX}$ as constant, then it can be simplified as follows:

$$P_{in} = (Q_e + A) \frac{T_{cm}}{T_{eo}} + BQ_e + C$$  \hfill (4)$$

where the parameters $A$, $B$, and $C$ are the model coefficients that can be determined by fitting the history operating data.

In cooling season, the evaporating temperature is lower than the outlet temperature of chilled water on the chilled waterside. Similarly, the condensing temperature is higher than the supply and return temperature of cooling water on the cooling side. Herein, in analogy with the calculation of heat exchange amount in exchanger, we proposed the mean temperature model updating method (MT method) and the outlet temperature model updating method (OT method) to correct the chiller power model, respectively. The MT method is based on replacing the inlet temperature of condenser ($T_{ci}$) by the mean temperature of supplied and returned cooling water ($T_{cm}$), and the model can be expressed as Equation (5). On the other hand, the OT method is to replace $T_{ci}$ by the outlet temperature of cooling water ($T_{co}$), and the model can be expressed as Equation (6).

$$P_{in} = (Q_e + A) \frac{T_{cm}}{T_{eo}} + BQ_e + C$$  \hfill (5)$$

$$P_{in} = (Q_e + A) \frac{T_{co}}{T_{eo}} + BQ_e + C$$  \hfill (6)$$

Table 2 shows the fitting results of the three models. It can be learnt that the OT model exhibits the superior performance with the minimum RMSE and maximum $R^2$. Figure 2 shows the scatter diagram of the calculated values against the observed values of the OT model, which provides higher fitting results with the relative error within 5%. Thus, the OT model is used to calculate the chiller power in our research.
2.2 Outlet temperature of cooling tower

Before we establish the outlet temperature model of cooling tower, some assumptions should be made for the heat and moisture exchange of the cooling tower as follows:

a. The heat and mass transfer in the cooling tower only follows the vertical direction of water flow and airflow.
b. The Lewis number is constant.
c. The water evaporation loss in the heat balance equation is ignored.
d. The temperature distribution in the horizontal direction of the cooling tower is consistent, and there is no temperature difference.
e. The enthalpy of saturated wet air is linear to the wet-bulb temperature.

Based on these assumptions, the heat transfer process can be divided into two parts: waterside convection and air side convection. Figure 3 shows the heat and mass transfer process in the cooling tower, where \( G, T, \) and \( i \) represent mass flow, temperature, and enthalpy, respectively; and the subscripts \( w, a, i, \) and \( o \) represent water, air, inlet, and outlet, respectively. Thus, the heat exchange amount can be defined as follows:

\[
Q_{\text{rej}} = \frac{t_{wi} - t_{wbi}}{R_w + R_a} \quad (7)
\]

where \( Q_{\text{rej}} \) is the heat exchange amount between water and wet air in the cooling tower, \( W; t_{wi} \) is the inlet water temperature, \( ^\circ C; t_{wbi} \) is the wet-bulb temperature of inlet air, \( ^\circ C; \) the \( R_w \) is the heat resistance on water side, \( m^2 \cdot ^\circ C/W; \) and \( R_a \) is the heat resistance on air side, \( m^2 \cdot ^\circ C/W. \)

According to the research contribution made by Jin et al.,\(^{31}\) \( Q_{\text{rej}} \) can be simplified as follows:

\[
Q_{\text{rej}} = \frac{c_0 G_w^c_1}{1 + c_1 \left( \frac{c_2}{c_1} \right)^c_2} (t_{wi} - t_{wbi}) \quad (8)
\]

where \( c_0, c_1, \) and \( c_2 \) are constant; \( G_w \) is the mass flow rate of water, kg/s; \( G_a \) is the mass flow rate of air, kg/s.

The heat exchange amount of the cooling tower \( (Q_H, W) \) can be calculated by the inlet and outlet temperatures of the cooling tower, as follows:

\[
Q_H = \rho G_w c_{pw} (t_{wi} - t_{wo}) \quad (9)
\]

where \( \rho \) is the density of water, kg/m\(^3\); and \( c_{pw} \) is the specific heat capacity of water under constant pressure, J/(kg \cdot ^\circ C).

Ignoring the loss of heat transfer, the heat exchange amount of cooling tower \( (Q_H) \) is approximately equal to the heat exchange amount between water and wet air in the cooling tower \( (Q_{\text{rej}}). \)

Therefore, the outlet temperature of the cooling tower can be obtained by combining Equations (8) and (9), and it can be expressed as follows:

\[
t_{wo} = t_{wi} - \frac{c_3 G_w^{c_3-1}}{1 + c_1 \left( \frac{c_2}{c_1} \right)^c_2} (t_{wi} - t_{wbi}) \quad (10)
\]

Based on the operating data, the parameters of \( c_1, c_2, \) and \( c_3 \) are determined as 1.504, 1.198, and 1.278, respectively.

To analyze the reliability and feasibility of the simplified method, we compared the fitting results to the traditional method (NTU method\(^6\)). The fitting comparison results of the two methods are shown in Figure 4. Compared to the traditional method, the results of the simplified method are relatively dispersed and have a relatively lower accuracy. However, the calculation of the outlet water temperature by the NTU method requires more complex formulae, which involves calculations of the moisture content and enthalpy of outdoor wet air, and the iterative calculation is required until the accuracy is met. The simplified method only needs a very simple expression to calculate the outlet water temperature of the cooling tower, which shortens the calculation time. Furthermore, the biggest advantage of the simplified method is that the air volume or cooling water flow in cooling tower can be directly calculated based on the outlet water.
temperature, which renders the model easy to implement online. Therefore, the simplified model is more suitable for online optimization in our research than the NTU method. The relative error of the simplified method is still within 5%, which is acceptable for simulation research.

### 2.3 Cooling water pump

The cooling water pump is another main component that consumes energy in the cooling system. Regulating the frequency of the cooling water pump is difficult because of the characteristics of the system pipe network, the operation of the chiller, and the temperature difference of supply and return water.

Based on the similarity law, the relationship among pump flow, power, and speed can be expressed as follows:

\[
\frac{Q}{Q_n} = \frac{n}{n_n} \tag{11}
\]

\[
\frac{N}{N_n} = \left(\frac{n}{n_n}\right)^3 \tag{12}
\]

where \(Q\) is the real flow rate of pump, \(m^3/h\); \(Q_n\) is the rated flow rate, \(m^3/h\); \(N\) is the real power of pump, \(kW\); \(N_n\) is the rated power, \(kW\); \(n\) is the real speed of pump, \(rpm\); and \(n_n\) is the rated speed, \(rpm\).

Theoretically, the pump power is proportional to the third power of the rotation speed when the pump is operating with frequency regulation. Unlike the chilled water system, the cooling water system is an open system; there exists height difference between the water distributor of the cooling tower and the liquid level of the water tank. The height difference is a certain value that will not change with the flow rate of cooling water (see Figure 5).
Thus, the pump head can be expressed as follows:

$$H = SQ^2 + \Delta H$$  \hspace{1cm} (13)$$

where $H$ is the pump head, m\(H_2O\); $S$ is the resistance of the pipe network of the cooling water system, \((m^3/s)^2\); and $\Delta H$ is the height difference between the water distributor and the liquid level of the water tank, m.

The pump shaft power can be calculated by pump head and flow as shown as follows:

$$P_e = \rho QH$$  \hspace{1cm} (14)$$

Thus, the pump power can be expressed as follows:

$$P = \frac{P_e}{\eta}$$  \hspace{1cm} (15)$$

Based on Equations (13)-(15), the power can be expressed as follows:

$$P = \frac{\rho Q}{\eta} (SQ^2 + \Delta H) = \frac{\rho S}{\eta} Q^3 + \frac{\rho \cdot \Delta H}{\eta} Q$$  \hspace{1cm} (16)$$

In fact, the efficiency of the water pump ($\eta$) is not a constant value and will change while the flow changes. Considering that the water flow will not change much while the frequency of cooling water pump changes in practice, the efficiency of the water pump is considered as a constant value. Then, the power of the cooling water pump can be expressed as a cubic polynomial of the flow rate:

$$P = AQ^3 + BQ$$  \hspace{1cm} (17)$$

The speed of cooling water pump is ranged from 60% to 100%, while the cooling water system operates normally. Figure 6 shows the fitting results between cooling water flow and pump speed. Based on the pump operating data, the parameters $A$ and $B$ are determined as 0.0073 and 0.033, respectively. Figure 7 shows the fitting results between pump power and flow rate.

### 2.4 Cooling tower fan

The direct measurement of airflow is challenging since the fan is installed on the top of the tower in mechanical draft or counter-current induced draft cooling towers. Therefore, it will be relatively hard to establish the model of the fan. In our research, we build the fan power model by referring to the modeling method of the pump and by using the relative rotation speed of the fan to approximately represent the airflow of the cooling tower. The relationship between fan power and fan speed can be expressed as follows:

$$P = a + b \cdot F_{sp} + c \cdot F_{sp}^2 + d \cdot F_{sp}^3$$  \hspace{1cm} (18)$$

$F_{sp}$ is the relative rotation speed of fan, \%; and $a$, $b$, $c$, and $d$ are the model parameters.

Based on the operating data, the parameters $a$, $b$, $c$, and $d$ can be determined as $1.789 \times 10^{-4}$, $2.818 \times 10^{-4}$, $-2.37 \times 10^{-5}$, and $1.83 \times 10^{-7}$, respectively. Figure 8 shows the fitting results between fan power and speed. The RMSE and $R^2$ of the model are 0.00622 and 0.98, respectively, which are acceptable for the simulation research.

### 3 METHOD

#### 3.1 Mathematical description of optimization

##### 3.1.1 Objective function

The cooling water system is an inseparable whole. The solution of the overall optimization should not compromise on pursuing the local optimization of the system. Furthermore, there are numerous interference factors that will affect the energy consumption of the whole system. These factors include the various operating conditions and the changes in the outdoor load and the environment.

In the practical operation, the inlet water temperature of the condenser is closely related to the outdoor wet-bulb temperature, and the cooling water temperature difference varies with the load on the evaporator side. Therefore, there exists an optimum temperature difference setpoint or an optimum cooling water flow to minimize the total energy consumption of the cooling water system. To find the optimum setpoint, the objective function of optimization is built as Equation (19) or (20).

$$f = \min P = \min \left( P_{\text{chiller}} + P_{\text{pump}} + P_{\text{fan}} \right)$$  \hspace{1cm} (19)$$
in which \( P \) is the total power of cooling water system, kW; \( \text{COP}_z \) is the global COP of cooling water system; \( Q_e \) is the cooling capacity of chiller, kW; and \( P_{\text{chiller}}, P_{\text{pump}}, \) and \( P_{\text{fan}} \) are the power of chillers, pumps, and fans, respectively, kW.

\[
f = \max \text{COP}_z = \max \left( \frac{Q_e}{P_{\text{chiller}} + P_{\text{pump}} + P_{\text{fan}}} \right) \tag{20}
\]

3.1.2 Optimizing variables

For the optimization of a multivariable nonlinear system, too many control variables will increase the difficulty of model solving and bring a negative impact on the optimization results. Therefore, it is necessary to take an overall consideration to simplify the control variables.
Theoretically, there are many controllable variables in the cooling water system, such as the number of fans and pumps, the speed of fans and pumps, and the valve position. In a general system, we know that the optimum performance of a set of the same type pumps at the same hydraulic conditions can be obtained when they operate at the same speed. Thus, the optimization of pump speed is prior to pump number. The cooling water system is an open system, so there is no hydraulic imbalance issue. Since changing the water flow by only regulating the valve position can increase the pump energy consumption, it is reasonable to change the water flow by changing the speed of the pump instead of regulating the valve position. Therefore, the ideal control variables of cooling water system are the pump speed and the fan speed. Meanwhile, the cooling water flow and cooling tower airflow are determined as regulating variables, and the cooling load and outdoor air wet-bulb temperature are determined as disturbance variables.

3.1.3 | Constraint conditions

The constraint conditions of the optimization can be expressed as follows:

1. Temperature constraint of the chiller

\[ T_{eo,\min} \leq T_{eo} \leq T_{eo,\max} \]  \hfill (21)

\[ T_{ci,\min} \leq T_{ci} \leq T_{ci,\max} \]  \hfill (22)

2. Water flow constraint

The designed water flow of the chiller in this research ranges from 65% to 135% of nominal value, which comes from the designed instruction of Trane WPWE0805 heat pump used in our experiment, so there are the following constraints.

\[ 0.65G_{e,n} \leq G_e \leq 1.35G_{e,n} \]  \hfill (23)

\[ 0.65G_{c,n} \leq G_c \leq 1.35G_{c,n} \]  \hfill (24)

3. Speed constraint

Considering that the power and flow rate of the fan and pump varies slightly when the speed is under 50%, thus we set the following constraint.

\[ 50\% \leq F_{sp} \leq 100\% \]  \hfill (25)

\[ 50\% \leq P_{sp} \leq 100\% \]  \hfill (26)

3.2 | Optimization method

3.2.1 | Conventional control method

The commonly used method for cooling water system is the constant temperature difference control method.\(^\text{29,32,33}\) This method is based on keeping the cooling approach (the difference between outlet water temperature of cooling tower and outdoor wet-bulb temperature) by adjusting the fan speed and on keeping the cooling water temperature difference by adjusting the pump speed. Figure 9 shows the flowchart of conventional constant temperature difference control method; each step is described in detail as follows:

a. Set the cooling approach (Approach) and cooling water temperature difference (\(\Delta T\)).

b. Determine the supply and return temperature of cooling water based on the approach and temperature difference, as shown in Figure 10.

c. Calculate the chiller power based on the power model, return temperature of cooling water, supply temperature of chilled water, and the cooling load.

d. Calculate the heat exchange amount of cooling tower by the energy conservation law based on the cooling capacity and power of the chiller.

f. Calculate the cooling tower airflow (\(G_a\)) based on Equation (10). If the calculated airflow is larger than maximum airflow of fan, this indicates that the setpoint of cooling approach is too small and the cooling water cannot be cooled to a lower temperature even if the fan operates at full speed. Therefore, the cooling water temperature should be recalculated with fan speed of 100% based on Figure 10. Similarly, if the calculated water flow is lower than the lower limit of chiller (2.76 m\(^3\)/h), then the cooling water temperature should be recalculated with the cooling water flow of 2.76 m\(^3\)/h based on Figure 10.

Besides the constant temperature difference control method, the constant speed control method is also often used in existing cooling water systems, with which the speed of fans and pumps is set as constant and not controllable.

3.2.2 | Water flow optimization control method

In order to find a more ideal control method for cooling water system, we have analyzed the effects of cooling tower inlet airflow and cooling water flow on the total energy consumption of cooling water system based on the system models. Figure 11 shows the effects of fan speed on the power consumption of the
system power when the water flow is 5 m³/h, which indicates that the power consumption of the system decreased as the speed of the fan increased. Thus, the fan should be maintained at full speed to reduce power consumption during the operation of the system. Figure 12 shows the effects of cooling water flow on power consumption of the system when the fan speed is 100%. It can be found that the total power consumption of system decreased first, and then it increased slightly with the raising water flow, which may be attributed to the characteristics of water pipe network. There exists an optimum water flow corresponding to the minimum total power, and then, the water flow optimization control method is proposed. The difference between the proposed method and traditional variable water flow methods is that the proposed method does not refer to the pressure difference or temperature difference to change water flow. Instead, it is trying to determine the optimal water flow corresponding to minimum system power directly.

The flowchart of water flow optimization control method is shown in Figure 13. The outlet water temperature of evaporator is set as 8°C and the fan speed is set as 100% while the cooling water system operates with the water flow optimization control method in our research. The details of the method are described as follows:

a. Set the initial cooling water flow as 2.76 m³/h (the required lower limit of chiller) and calculate the cooling water temperature based on Figure 10.
b. Calculate the chiller power based on the cooling water temperature, cooling load, and supply temperature of chilled water, and the calculated pump power and the total power based on cooling water flow.
c. Increase the water flow by 0.01 m³/h, then calculate the corresponding total power ($P_{z2}$), and compare it with the total power without flow increasement ($P_{z1}$).
d. Repeat step c until $P_{z1} < P_{z2}$, and then output the corresponding water flow and total power.

e. Repeat steps a-d to enter the optimization of next time period operating condition.

4 | RESULTS AND DISCUSSION

4.1 | Calculation of cooling load

Since the cooling load is closely related to the outdoor climate during the actual operation, the hourly cooling load should be obtained based on outdoor climate before the optimization of cooling water flow. In the present research, the cooling load is calculated using the simulation software TRNSYS with a design indoor temperature of 26°C and a design indoor relative humidity of 60%. The cooling season is from June to October, which corresponds to the time sequence from 3254 hours to 7035 hours in TRNSYS, and the cooling load is calculated hourly. Table 3 shows the detailed information of external building envelope.

Figures 14 and 15 present the outdoor dry and wet-bulb temperatures and the indoor cooling load in cooling season, respectively. These figures infer that the variations of the wet-bulb temperature in cooling season are relatively small in comparison with the dry-bulb temperature. Likewise, the variations of the cooling load are relatively small due to the heat resistance of external building envelope.

4.2 | Comparisons of different control methods

Due to the fixed speed of fans and pumps used in the constant speed control method, we know the cooling water flow and the cooling tower inlet airflow. The temperature of cooling water supply and return can be also calculated based on Figure 10. Then, the total power in real time and total energy consumption in cooling season of the system can be obtained based on the known cooling load, temperature of cooling and chilled water temperature, and water flow. Figure 16 shows the system hourly total power in cooling season.

Figure 17 shows the simulation results of real-time total power while the system operates with constant temperature difference control method in different cooling approaches. The temperature difference in cooling water supply and return is set as 5°C. It can be learnt that the lower cooling approach renders the lower power consumption of the cooling water system. Figure 18 shows the hourly optimized cooling water flow and system total power, while the system operates in water flow optimization control method in cooling season.

In view of the long time period and complicated outdoor climates in cooling season, it is indistinct if the comparison results of a whole cooling season were depicted in the same figure. Therefore, we selected the comparison results of 1 day to analyze the performances of different methods. As shown in Figure 19, the comparison results of the system total power of 1 day are depicted while the system operated in the water flow optimization control method (Method A), the constant temperature difference control method (Method B) with different cooling approaches, and the constant speed control method (Method C), respectively. It can be found that the system total power is the minimum when the system operates with the water flow optimization control method. For the constant temperature difference control method, the larger the
cooling approach is, the larger the system total power is. And the total power consumption even exceeded the power consumption in constant speed control method when the cooling approach was 5°C. It can be explained that the chiller power increment caused by inlet cooling water temperature increment is larger than the reduction in the same term caused by fan. Therefore, at the same conditions, the lower the cooling approach is, the less the power consumption of the system has. When the cooling approach raises until the fan operates at full speed, the system total power can be minimized by an optimized cooling water flow, and that is what variable water flow does.

Figure 20 represents the hourly cooling water flow and temperature difference when the system operated in constant temperature difference control method with the cooling approach of 3°C. It can be found that the system operated with the cooling water flow of lower limit (2.76 m³/h) in some conditions, which implies that the fans and pumps operated in variable speed states at these moments. Even though the fan and pump power are saved adequately, the relatively higher inlet cooling water temperature renders a higher power consumption in the chiller. Thus, the constant temperature difference control method only considered the energy savings in fans and pumps. On the contrary, the water flow optimization control method is used to optimize the operation by taking the cooling water system as a whole.

Tables 4 and 5 display the comparison results of global COP and energy consumption among different methods, respectively. It can be noticed that the energy consumption of the constant temperature difference control method (Method B) with approach of 5°C is larger than the same term of the constant speed control method (Method C). This implies that the cooling approach and cooling water temperature difference setpoint should be scientifically selected based on actual cooling load and outdoor climate when the system operates with constant temperature difference control method. By contrast, the water flow optimization control method is much more reasonable for the energy saving of cooling water system, because there exists an optimized cooling water flow corresponding to the minimum power consumption for each operating condition based on the study in Section 2.3.2.

As can be seen from Tables 4 and 5, the water flow optimization control method (Method A) presented the highest system global COP and the minimum energy consumption, and the reduction in energy consumption was 72.34 kWh compared with the constant temperature difference control method with approach of 3°C. This indicates that there still exists some energy-efficient space for the conventional...
FIGURE 13  The flowchart of water flow optimization control method

![Flowchart](image)

TABLE 3  Parameters of building envelope

| Type      | Materials                  | Thermal conductivity [kJ/(h m K)] | Thickness (mm) | Heat transfer coefficient [W/(m² K)] |
|-----------|----------------------------|----------------------------------|----------------|-------------------------------------|
| Exterior wall | Common plaster             | 1.26                             | 20             | 0.466                               |
|           | Brick                      | 0.9                              | 300            |                                     |
|           | Low-density polyurethane  | 0.08                             | 16             |                                     |
| Roof      | Concrete block             | 1.84                             | 100            | 0.389                               |
|           | Low-density polyurethane  | 0.08                             | 49             |                                     |
| Floor     | Common plaster             | 1.26                             | 20             | 0.214                               |
|           | Light concrete floor       | 0.5                              | 100            |                                     |
|           | Clay                       | 5.4                              | 200            |                                     |
|           | Low-density polyurethane  | 0.08                             | 80             |                                     |
| Window    | Double-deck aluminum sash | /                                | /              | 3.200                               |

FIGURE 14  The simulated outdoor climate of cooling season

![Graph](image)
constant temperature difference control method. Compared to the constant speed control method, the reduction in energy consumption of water flow optimization control method was 691.9 kWh in cooling season, which is a considerable amount of savings, accounting for 3.45% of total energy consumption of cooling water system and for 31.11% of energy consumption of pumps.

5 | CONCLUSION

In this paper, the models of cooling water system are firstly introduced and simplified on the premise of online optimization. The simplification of the models of cooling water system renders these models capable and easy to implement for online optimization with acceptable accuracy. Then, the
**FIGURE 18** The system power with water flow optimization control method in cooling season

**FIGURE 19** Comparison of system total power in different optimization methods

**FIGURE 20** The cooling water flow and temperature difference with approach of 3°C

**TABLE 4** Global COP comparisons of cooling water system indifferent methods

| Time     | Method A Approach of 3°C | Method A Approach of 4°C | Method A Approach of 5°C | Method C |
|----------|-------------------------|-------------------------|-------------------------|----------|
| June     | 3.147                   | 3.102                   | 3.052                   | 2.992    | 3.028    |
| July     | 3.217                   | 3.192                   | 3.147                   | 3.091    | 3.131    |
| August   | 3.300                   | 3.279                   | 3.234                   | 3.177    | 3.220    |
| September| 3.379                   | 3.340                   | 3.286                   | 3.221    | 3.266    |
| October  | 2.862                   | 2.803                   | 2.751                   | 2.693    | 2.706    |
objective function of the optimization is determined by considering the cooling water system as a whole. Through the impact analyses of cooling tower airflow and cooling water flow on system total power, we find that the total power gradually decreases as the airflow increases at the same cooling water flow, which implies that the energy saving of the cooling water system would benefit from the full speed operating fans. Besides, the total power initially decreases and subsequently increases as the cooling water flow increases at the same airflow, so there must be an optimum water flow corresponding to the minimum total power for each operating condition.

Based on the above, the water flow optimization control method is proposed, which is trying to optimize the cooling water flow directly to minimize the total power of cooling water system while the fan operates at full speed. The method suggests the fans operate without installation of variable speed drivers to save initial investment, which can also simplify the coupling relationships among the multiple control loops to improve the control stability of the cooling water system.

From the simulated comparison results, the proposed method presents the highest global COP and the least energy consumption among the three methods. The energy consumption reduction in the proposed method is 3.45% compared with the conventional constant speed control method and 0.37% only in comparison with the conventional constant temperature difference control method. The negligible energy savings imply that the conventional constant temperature difference control method with proper cooling approach and temperature difference setpoint is also a considerable control method for designing or retrofitting the cooling water system. Even though it is not the optimum method, it can still be considered as a potential method for energy savings of the system.

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**NOMENCLATURE**

| Symbol | Description |
|--------|-------------|
| $T_{cm}$ | mean temperature of supplied and returned cooling water, °C |
| $Q_c$ | cooling capacity of chiller, kW |
| $q_e$ | heat loss of evaporator, kW |
| $P_{in}$ | input power of chiller, kW |
| $Q_{rej}$ | heat exchange amount between water and wet air in cooling tower, W |
| $t_{wi}$ | inlet water temperature of cooling tower, °C |
| $t_{wbi}$ | wet-bulb temperature of inlet air, °C |
| $R_w$ | heat resistance on water side, m²°C/W |
| $R_a$ | heat resistance on air side, m²°C/W |
| $G_w$ | mass flow rate of cooling water, kg/s |
| $G_a$ | mass flow rate of air, kg/s |
| $Q_H$ | heat exchange amount of cooling tower, W |
| $c_{pw}$ | specific heat capacity of water under constant pressure, J/(kg°C) |
| $Q$ | real flow rate of pump, m³/h |
| $Q_n$ | rated flow rate, m³/h |
| $N$ | real power of pump, kW |
| $N_n$ | rated power, kW |
| $n$ | real speed of pump, rpm |
| $n_n$ | rated speed, rpm |
| $H$ | pump head, mH₂O |
| $S$ | resistance of the pipe network cooling water system, (m³/s)² |
| $F_{sp}$ | relative rotation speed of fan, % |
| $P_{sp}$ | relative rotation speed of pump, % |
| $P$ | total power of cooling water system, kW |
| $\text{COP}_z$ | global COP of cooling water system |
| $T$ | temperature, °C |
| $G$ | water flow, m³/h |

**TABLE 5** Energy consumption comparisons of cooling water system in different methods

| Time    | Method A (kWh) | Method B (kWh) | Method C (kWh) |
|---------|----------------|----------------|----------------|
|         | Approach of 3°C | Approach of 4°C | Approach of 5°C |
| June    | 5040.03        | 5112.37        | 5196.45        | 5299.5 | 5236.63 |
| July    | 3928.28        | 3957.97        | 4014.76        | 4088.09 | 4035.7 |
| August  | 4023.52        | 4048.37        | 4105           | 4179.42 | 4122.66 |
| September | 3380.05      | 3419.83        | 3475.68        | 3545.62 | 3497.47 |
| October | 2969.37        | 3032.57        | 3089.35        | 3155.48 | 3140.73 |
| Total   | 19341.25       | 19571.1        | 19881.24       | 20268.12 | 20033.19 |
Subscripts
ei inlet of evaporator
eo outlet of evaporator
ci inlet of condenser
co outlet of condenser
min lower limit
max upper limit
n nominal value

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