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A Simplified Thermal Modelling of Cooling Tower for Optimization of HVAC Systems to Enhance Energy Efficiency

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Abstract

This paper presents a simplified model to analyze the heat flow in Closed Wet Cooling Tower (CWCT). Based on the existing research results and some suitable assumed conditions, we develop a model to simplify the handling of the heat rejection rates under four different input parameters and variable ambient wet-bulb temperatures. The analytical results of the data from different sources demonstrate that the model with the new parameters is accurate enough for the control and optimization of CWCT operation. After validating the accuracy of the model, we also describe the extension of the model for analyzing the energy flows of a CWCT. When the model in combination with the input power model, it may represent the quantitative relationship between energy flow rates and principal input-output variables of the CWCT.

Keywords

Closed Wet Cooling Tower, Cooling Capacity, Thermal Modeling, Simplified Analysis

1. Introduction

In order to reduce energy consumptions for building and efficient use of energy, continuous energy audit and performance monitoring of the operating parameters in installed systems are essential. The building sector alone represents about 33% of electricity consumption of the total electricity consumption in India, with commercial sectors and residential sectors accounting for 8% and 25% respectively [1]. Studies indicate that air-conditioning is responsible for 10% to 60% of the total building energy consumption, depending on the building type [2]. In developed countries, HVAC systems are the most energy consuming devices, accounting for about 10% - 20% of final energy use [3]. As one of the two basic types of evaporative cooling devices, CWCT is essentially com-

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bining a heat exchanger and direct-contact evaporative cooling tower into one device [4]. With the advantages of better thermal performance and lower energy consumption than traditional cooling towers, CWCTs plays an important role in the heat rejection of residential and commercial buildings. Many literatures which are related to the modeling and experiment of CWCT can be found. The mass and heat process cooling tower has always been received much attention. In order to obtain experimental correlations for mass and heat transfer coefficients of a new CWCT, Facao and Oliveira [5] present different thermal models to predict the performance of CWCT. As is shown in Ref. [6], a detailed analysis of the mass transfer coefficient and spray heat transfer coefficient correlations is made experimentally, including the effects of the errors in their evaluation on tower efficiency. Beside the experimental method, the numerical simulation method is often used for the same purpose. A numerical simulation of a CWCT with novel design is presented for the determination of heat and mass transfer coefficients and the influence of Lewis number on the cooling tower performance [7]. To investigate the thermal performance, a program is coded for the heat and mass transfer in CWCT whether the outlet air is supersaturated or not [8].

As part of the heat and mass transfer characteristics, cooling capacity of CWCT has also got the same attention. The results of a theoretical investigation show that the highest fall of serpentine water temperature and losses of spray water are observed for the lowest value of inlet wet bulb temperature [9]. From the experimental study, cooling capacity and pressure drop of CWCT have been studied with respect to variable air inlet parameters, cooling water inlet temperatures and air to cooling water volume flow rate ratio [10]. The results of other experimental measurements for performance of a prototype CWCT show that tower flow rates and number of tubes and rows are optimized for the required cooling load to achieve a high coefficient of performance (COP) [11]. Based on Number of Transfer Units-effectiveness (NTU-ε) models by simplification of heat and mass balance and transfer equations, a simple model for indirect-contact evaporative cooling-tower behavior has been developed [12].

Although there is a large amount of researches on the heat and mass transfer characteristics of the CWCT, few of them are reported on the simplified analysis of its cooling capacity prediction. As part of a project to analyze the energy flows in a typical cooling system, this paper describes a simplified model to analyze the heat flow in CWCT. Based on some additional assumptions, this model is derived from an existing model. The accuracy of the models with new parameters is validated by using experimental data from different sources. After this, further analysis of the predictions of these models is made under specific operational conditions. In the end, we analyze the model's significance on practice and relative further researches.

2. Mathematical Modeling

Some approaches are adopted for modeling the heat and mass transfer processes of CWCT, such as the Merkel method, the Poppe method and the NTU-ε method [7] [8] [12]. By applying these methods to get the expected results, we generally consume considerable computational time. The CWCT mathematical model to be used for analyzing the energy flows in air-conditioning system must be simplified enough with acceptable accuracy. For obtaining experimental correlations between mass and heat transfer coefficients of a new CWCT, simpler models could give as good, or even better, consequences as models based on finite difference techniques [5]. In this work, the performance model of CWCT is derived from an existing model, as described below, and its main input variables include ambient wet bulb temperature (T_{wb}, ˚C), dry air flow rate (m_a, kg/s), spray water flow rate (m_{sw}, kg/s), and inlet parameters of cooling water such as the temperature (T_w, ˚C) and the mass flow rate (m_w, kg/s). The main assumptions of the modeling process are detailed in Refs. [7]-[9] [12]. Two CWCTs from two different sources are selected for the verification of the model. **Figure 1** and **Figure 2** indicates the schematic diagram of the three fluids in a cross-flow CWCT unit and a counter-flow CWCT unit, respectively.

The model for CWCT behavior adopted in this paper is the one developed by Stabat and Marchio [12]. In this model, two undetermined coefficients could be determined by the analysis of input-output parameters from only two operating conditions. Once parameterized, the model can be used for calculating the heat rejection rate of the CWCT. The total heat rejection rate (Q, kW) is determined under the following conditions $1200 \le Re_a \le 14000$, $50 \le Re_{sw} \le 240$, $12 \le d_{ext} \le 40\text{mm}$, $2500 \le Re_w \le 10000$ and $3 \le Pr \le 8$ ($15 \le T_w \le 60$˚C), expressed as follows (see detailed in Ref. [12]):

$$Q = -c_w m_w \left(T_{w,o} - T_{w,i}\right) = m_a \left(h_{wb,o} - h_{wb,i}\right) \cong m_a c_{psat} \left(T_{wb,o} - T_{wb,i}\right) \tag{1}$$

$$Q = UA\Delta T_{LM} \tag{2}$$

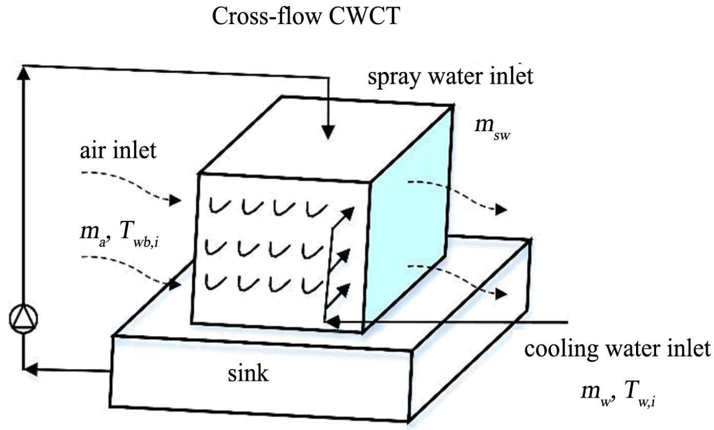

Figure 1. Schematic diagrams of the three fluids in a cross-flow CWCT unit.

with,

$$\frac{1}{UA}=\frac{I}{\beta_{ext}c_{psat}m_a^{0.8}}+\frac{\mu_{sw}^{0.5}}{\beta_{int}m_{sw}^{0.8}} \tag{3}$$

$$\Delta T_{LM}=\frac{\left(T_{w,o}-T_{wb,i}\right)-\left(T_{w,i}-T_{wb,o}\right)}{\ln\dfrac{T_{w,o}-T_{wb,i}}{T_{w,i}-T_{wb,o}}} \tag{4}$$

where, *Re* is Reynolds number; *Pr* is Prandtl number; d_{ext} is tube diameter (m); c_w is specific heat of cooling water at constant pressure ($kJ\cdot kg^{-1}\cdot{}^\circ C^{-1}$); c_{psat} is fictitious specific heat of moist air at constant pressure ($kJ\cdot kg^{-1}\cdot{}^\circ C^{-1}$); *m* is flow rate (kg/s); *T* is temperature (˚C); *h* is enthalpy of moist air ($kJ\cdot kg^{-1}$); *U* is heat transfer coefficient ($kW\cdot m^{-2}\cdot{}^\circ C^{-1}$); *A* is area ($m^2$); μ is dynamic viscosity coefficient of spray water ($kg\cdot m^{-1}\cdot s^{-1}$); ΔT_{LM} is logarithmic mean temperature difference (˚C); β_{int} is a constant which is influenced by the coil's geometry and constant water-properties; β_{ext} is a constant which depends on the thermal properties of air and on the coil's geometry; Subscripts *w*, *sw*, and *a* mean cooling water, spray water and dry air, respectively; Subscript *wb* represents ambient wet-bulb temperature, Subscripts *i*, *o* stand for inlet and outlet.

Under typical operating conditions of air-conditioning systems, the enthalpy of moist air is a single function of wet-bulb temperature and can be approximated by the following relation [13]:

$$h_a\left(T_{wb}\right)=1.01\times T_{wb}+3.99431\times e^{0.06482\times T_{wb}}\times\left(2.5+0.00184\times T_{wb}\right) \tag{5}$$

Based on the regression analysis of Equation (5), liner equation may be ideally established between the enthalpies of moist air and ambient wet-bulb temperatures. When ambient wet bulb temperature is varied from 15.0˚C to 30.0˚C, Equation (5) can be then simplified as follows:

$$h_a\left(T_{wb}\right)\cong 3.9496\times T_{wb}-20.475 \tag{6}$$

When $h_a(T_{wb})$ from Equation (6) is substituted to Equation (1), the fictitious specific heat of moist air (c_{psat}) is given as:

$$c_{psat}\cong\frac{h_{a,o}-h_{a,i}}{T_{wb,o}-T_{wb,i}}\cong 3.9496 \tag{7}$$

When $\Delta T_{max}/\Delta T_{min}<2$, Equation (3) is modified and can be shown that [14] [15]:

$$\Delta T_{LM}=\frac{\Delta T_{max}+\Delta T_{min}}{2}=\frac{\left(T_{w,o}-T_{wb,i}\right)+\left(T_{w,i}-T_{wb,o}\right)}{2} \tag{8}$$

Assuming the dynamic viscosity coefficient of spray water is constant over small ranges of temperature, one simplifies Equation (4) as:

$$\frac{1}{UA}=\frac{a_0}{m_a^{0.8}}+\frac{a_1}{m_{sw}^{0.8}} \tag{9}$$

where, a_0, a_1 indicate two undetermined coefficients.

By substituting Equations (7)-(9) into Equation (2), we obtain the performance model of the CWCT (Equation (10)):

$$Q=\frac{T_{w,i}-T_{wb,i}}{\frac{a_0}{m_a^{0.8}}+\frac{a_1}{m_{sw}^{0.8}}+\frac{1}{2c_{psat}m_a}+\frac{1}{2c_w m_w}} \tag{10}$$

In this section, a simplified model for the heat rejection rate of the CWCT is described based on relevant research results and additional assumptions. The relation between its main input-output variables can be determined using Equation (10).

3. Model Verification

CWCT can be classified by the movement of spray water and air as counter-flow and cross-flow types. In order to confirm the proposed model, these two types of CWCTs have been selected to provide the associated experimental data obtained under quasi-steady state conditions. Levenberg-Marquardt method is implemented to determine undetermined coefficients of the model.

The model is validated experimentally on the studied of heat rejection rates, at different inlet conditions and variable ambient wet-bulb temperature. In addition, the predicted cooling loads using the mathematical model described in Section 2 are obtained based on the well-established sets of test data. In order to visually describe and quantitatively show the heat rejection rates of the model prediction, an error index, Root-Mean-Square of Relative Error (RMSRE), is adopted [16]:

$$\text{RMSRE}=\sqrt{\frac{\sum_{i=1}^{n}\left(\frac{Q_{\text{calculated}}-Q_{\text{predicted}}}{Q_{\text{calculated}}}\right)^2}{n}} \tag{11}$$

3.1. Validation Using a Cross-Flow CWCT

Figure 1 indicates the sketch of a cross-flow CWCT unit [17]. The size of the CWCT is 570 mm (Height) × 310 mm (Width) × 180 mm (Length). As shown in the figure, the spray water flow is in cross flow with the air flow and process water flow. In the tower, fin-tube structure is employed to effectively enhance the heat transfer between the spray water and the process water. The finned tubes are made of stainless steel.

Table 1 lists the designed parameters of the cooling tower. The parameters of CWCT structure and the schematic diagram of the testing configuration are detailed in Ref. [17].

The tests for obtaining relevant data of the cross-flow CWCT unit were performed in an environmental chamber. In order to study the cooling capacity of the CWCT, 11 operating conditions and 46 sets were provided from the perfect experiment platform. As listed in **Table 2**, the main parameters are the inlet parameters of process water and the ambient wet-bulb temperature etc. To verify the accuracy of the model, the experimental data were adopted. The undetermined coefficients of the model is determined as a_0 = 0.350 and a_1 = 0.098 by using the proposed method above. The relatively errors of the calculated and predicted heat rejection rate generally were within ±10% and expressed in **Table 2**. The RMSRE for the model validation of 11 data points was 0.062, which means the model is reliable.

3.2. Validation Using a Counter-Flow CWCT

A counter-flow CWCT is selected from Ref. [18]. The size of the CWCT is 270 mm (Height) × 225 mm (Width) × 550 mm (Length). As shown in **Figure 2**, the spray water flow is in counter flow with the air flow. In this tower, padding upon the un-finned coil is employed to strengthen the heat and mass transfer between the spray water and air. The coil is made of stainless steel to facilitate heat exchange between the spray water and the process water.

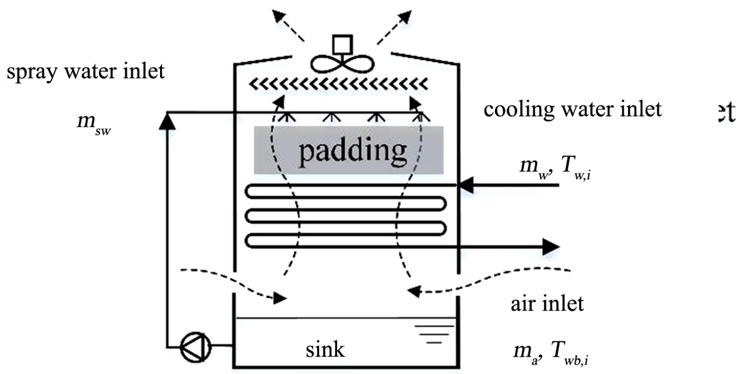

Figure 2. Schematic diagrams of the three fluids in a counter-flow CWCT unit.

Table 1. Designed parameters of the cross-flow CWCT [17].

Description	Value
Mass flow rate of circulating water (t/h)	63,500
Temperature drop of circulating water (˚C)	9.47
Area of water drenching (m^2)	6000
Elevation of the top of the tower (m)	120
Elevation of the top of cooling fill (m)	10.2
Height of cooling fill (m)	1
Cooling heat (MW)	694

Table 2. Comparison between the calculated [17] and predicted cooling capacity of the cross-flow CWCT test.

Test No.	m_a (kg/s)	$T_{wb,i}$ (˚C)	m_{sw} (kg/s)	$m_{w,i}$ (kg/s)	$T_{w,i}$ (˚C)	$T_{w,o}$ (˚C)	$Q_{calculated}$ (kW)	$Q_{predicted}$ (kW)	Err (%)
1	0.19	21.1	0.12	0.32	30.2	27.7	3.36	3.14	6.42
2	0.27	20.6	0.12	0.32	30.8	27.6	4.3	4.3	0.06
3	0.35	20.1	0.13	0.32	30.3	26.7	4.84	4.99	-3.06
4	0.35	20.8	0.12	0.26	30.4	26.6	4.15	4.43	-6.88
5	0.35	20.9	0.12	0.32	32.9	28.8	5.51	5.77	-4.76
6	0.35	21.1	0.12	0.2	30.2	26.1	3.44	3.95	-14.78
7	0.35	22	0.12	0.32	36.6	31.2	7.26	7.02	3.23
8	0.35	22.6	0.06	0.29	30.2	27.6	3.17	3.02	4.51
9	0.35	22.6	0.13	0.3	29.9	27	3.65	3.53	3.51
10	0.35	22.8	0.09	0.3	30.2	27.4	3.53	3.3	6.44
11	0.35	22.9	0.12	0.31	30.2	27.5	3.52	3.49	0.68

Based on the operating processes of the counter-flow CWCT developed by You [18], this mathematical model has been validated with the relevant experimental data. **Table 3** summarizes a set of 24 test data sets. The

RMSRE is estimated as 0.040 (a_0 = 0.321; a_1 = 0.119) for the counter-flow CWCT and these are acceptable for the purposes of control and optimization.

Figure 3 shows the result of a comparison between predicted and calculated results for heat rejection rates of the CWCT. Three lines are showed in the figure to indicate cooling capacities. The middle line means the model exactly predicts the calculation of cooling capacities. If the data points are located in the area under the line, it means that the model under-predicts the cooling capacities. Conversely, the model over-predicts the cooling capacities. The model leads to a maximum relatively error in heat rejection rates of 7.9%, which means the predictive model, is accurate.

4. Results and Discussion

4.1. Effect of Inlet Flow Rates

Two specific operating conditions of the counter-flow CWCT are selected for analyzing the cooling capacities under different flow rates of three fluids, such as cooling water, spray water and dry air. When $T_{w,i}$ = 37.0˚C, $T_{wb,i}$ = 16.6˚C, the predicted cooling load is expressed as follows:

$$Q = \frac{20.4}{\dfrac{0.321}{m_a^{0.8}} + \dfrac{0.119}{m_{sw}^{0.8}} + \dfrac{0.127}{m_a} + \dfrac{0.119}{m_w}} \tag{12}$$

As shown in **Figure 4**, the cooling performance curves of the counter-flow CWCT under constant cooling water flow rater (m_w = 11.11 kg/s) and invariable dry air flow rate (m_a = 24.3 kg/s) can be displayed, respectively. Obviously, a trend of increase in cooling capacity is consistent with the increase of substances mass flow rate at the same operating conditions. Changes of spray water flow rate have minimal impact on cooling capacity,

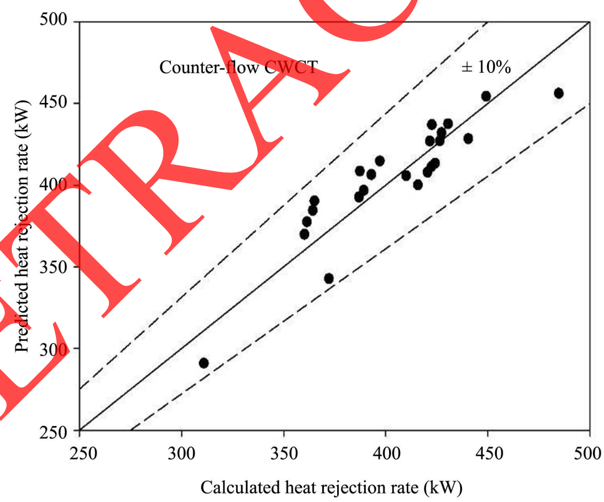

Figure 3. Comparison of the Predicted heat rejection rate against experimental data of the counter-flow CWCT.

Table 3. The experimental data of the counter-flow CWCT test [17].

Test no.	m_a (kg/s)	m_{sw} (kg/s)	$m_{w,i}$ (kg/s)	$T_{w,i}$ (˚C)	$T_{w,o}$ (˚C)	$T_{wb,i}$ (˚C)	$Q_{calculated}$ (kW)
1 - 4	24.33		9.72	37.09 ± 0.01	28.20 ± 0.06	17.34 ± 0.05	362.70 ± 2.55
5 - 12	20.00; 24.33	19.44; 22.78;	11.11	37.20 ± 0.15	28.49 ± 0.28	15.55 ± 1.41	406.70 ± 19.83
13 - 16	24.33	25.56; 28.89	12.5	37.14 ± 0.11	29.20 ± 0.24	16.18 ± 0.46	418.69 ± 8.66
17 - 24	20.00; 24.33		13.89	38.61 ± 1.59	31.21 ± 1.57	18.93 ± 3.50	397.83 ± 86.92

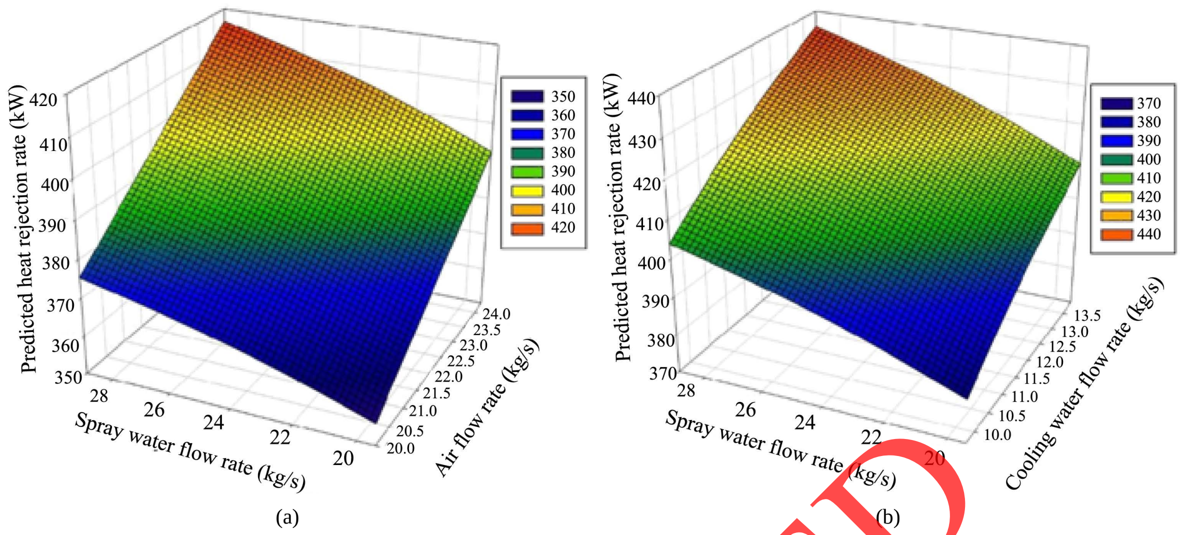

Figure 4. The Predicted heat rejection rate of the counter-flow CWCT under different flow rates of three fluids. (a) Cooling water flow rate of the counter-flow CWCT:11.11 kg/s; (b) Air flow rate of the counter-flow CWCT:24.33 kg/s.

followed the cooling water flow rate.

4.2. Effect of Inlet Temperatures

In this section, the predicted heat rejection rates of the cross-flow CWCT were calculated under different cooling water inlet temperatures and variable ambient wet-bulb temperatures. When m_a = 0.35 kg/s; m_{sw} = 0.12 kg/s and m_w = 0.30 kg/s, the correlation equation for the predicted heat rejection rate of the cooling tower is expressed as follows:

$$Q = 0.475 \times \left(T_{w,i} - T_{wb,i}\right) \tag{13}$$

Figure 5 indicates the predicted cooling loads under constant flow rates and variable inlet temperatures. As shown in this figure, an increase in cooling water inlet temperature can enhance the heat rejection. On the contrary, the higher ambient wet-bulb temperature, the lower heat rejection at the given operating conditions. *RMSRE* for this model validation of eight data points is 0.063 and is approximately equal to the prediction accuracy of the proposed model above (*RMSRE* = 0.062).

4.3. Extended Analysis

The simplified model's significance on practice and relative further researches is discussed as follows. The results of the analysis above show that the relevant performances of CWCT should be obtained by the model for a large domain of operation conditions. Meanwhile, input power model of CWCT is a function of spray water flow rate and air flow rate [19]. In combination with the simplified model, the input power model should indicate the quantitative relationship between energy flow rates and main input-output parameters of the tower. For example, the input power (p, kW) of a CWCT is the objective function for minimizing the sum of the pump input power (p_p, kW) and the fan input power (p_f, kW), expressed as follows:

$$\text{Min} \sum p = p_p + p_f \tag{14}$$

with the constraints:

$$p_{p,\min} \le p_p \le p_{p,\max} \tag{15}$$

$$p_{f,\min} \le p_f \le p_{f,\max} \tag{16}$$

$$Q = Q\left(T_{wb,i}, T_{w,i}, m_w, m_{sw}\left(p_p\right), m_a\left(p_f, p_p\right)\right) \tag{17}$$

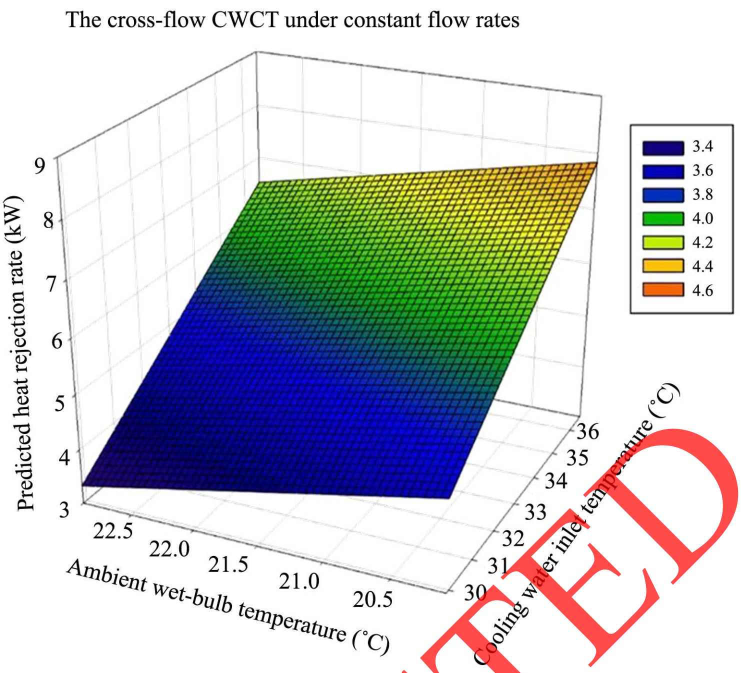

Figure 5. The predicted heat rejection rate of the cross-flow CWCT under constant flow rates of three fluids.

5. Conclusion

The significance of this study is to promote CWCT for water -cooled chillers under the climate change. A simplified model for the cooling load of CWCT operating, adapted from an existing model, has been developed based on the assumed suitable conditions. The model has been validated on data from CWCT tests for some given conditions (cooling water inlet parameters, wet-bulb temperatures, spray water flow rates and air flow rates). The parameterization is obtained by the Levenberg-Marquardt method. Assuming the dynamic viscosity coefficient of spray water is constant over small ranges of temperature, the heat rejection rates of the calculated and predicted results are in good agreement with each other. The results of the model verification show that the models with new parameters are accurate enough (RMSRE < 0.1) for the control and optimization of CWCT operation. As time goes on, the characteristic parameters of the tower should be updated periodically because of the effect of negative factors, such as spray water pollution. The quantitative relationships between power consumption and main input-output variables of CWCT should be determined by the combination of the model and input power model.

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