**A new theoretical method for predicating the part-load performance of natural
draft dry cooling towers**

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1 **Abstract:**

2 A new theoretical model is proposed to predict thermal performance of natural draft

3 dry cooling tower (NDDCT) under various ambient conditions. When evaluating the

4 effect of ambient temperature, heat rejection of the NDDCT is converted to the form

5 that is proportional to the power function of initial temperature difference of the tower,

6 using the theory of similar triangles. Heat rejection of the NDDCT under windy

7 condition is simply calculated based on the outlet air velocity obtained by a proposed

8 algorithm. Although this theoretical model is inappropriate at crosswind velocity higher

9 than the critical wind velocity, it's still meaningful for thermal performance prediction

10 of the NDDCT, because this critical wind velocity is usually large and uncommon in

11 the practical operation of the tower. Testing work for a 600MW_{th} and other two

12 NDDCTs is conducted using this theoretical model flexibly, and the predictions agree

13 well with practical measurement data or results available in the existing literatures. In

14 addition, analysis of the 600MW_{th} NDDCT shows that under a larger initial

15 temperature difference, the tower has larger resistance to crosswind and cooling

16 efficiency declines more slowly. Moreover, heat rejection of the tower increases more

17 rapidly with initial temperature difference under a smaller crosswind velocity.

18 **Key words:** thermal performance; natural draft dry cooling tower; theoretical model;

19 ambient temperature; crosswind; initial temperature difference

1. Introduction

Natural draft dry cooling tower(NDDCT) including heat exchanger which is arranged vertically or horizontally at the bottom of the tower, is an energy-efficient cooling equipment in power plants. Thanks to the water-saving advantage, it has been widely used in the regions with rich coal and lack of water [1, 2]. Air movement in or around the NDDCT is dependent on density difference between entering cold air and internal warm air[3]. When passing across heat exchanger, the entering air will be heated by hot circulating water inside the heat exchanger bundles by convective heat transfer. Then, buoyancy originating from density difference between entering cold air and internal warm air drives upwelling flows inside the tower and draws fresh air in at the same time. As a consequence, thermal performance of the NDDCT is highly sensitive to ambient conditions[3-6].

To figure out performance variation caused by ambient condition, there are two main kinds of methods, one is experimental research, and the other is numerical research, sometimes verified by experiment under certain conditions. As for experimental research, Ardekani et al. [7] experimentally evaluated thermal performance of the cooling tower under actual wind conditions prevailing around the tower. These measurements showed that the tower front cooling sectors experience better airflow distribution compared to sectors parallel to wind direction, which improves their thermal performance by about 20% compared to still-air conditions. Wind tangential velocity at corner sector is four times the velocity at the reference point, decreasing air pressure and tower suction. Wei et al. [8] investigated the

1 unfavorable effect of wind on the performance of drycooling towers with full-scale
2 and wind-tunnel experiments, and then explained three reasons for deterioration of
3 cooling efficiency reduced by wind, i.e., an unfavorable pressure distribution at the
4 tower entrance, breaking of the plume rising at the tower exit, and cold inflow of cool
5 air induced by leading edge separation. There is no doubt that full-scale experiment is
6 the most direct way to figure out thermal performance of the tower. However, the
7 NDDCT is a complex large-scale object involving fluid and heat flows, and it takes a
8 lot of time and resources for one experiment. Moreover, real-world environment is
9 uncontrollable, changing stochastically. All these reduce the reliability and practicality
10 of full-scale experiment. As for wind-tunnel experiment, it can't well predict the real
11 thermohydraulic behavior of the tower, owing to a set of restrictions and doubtful
12 simplifications applied in dimensional analysis and the experience procedure[9].

13 As for numerical research, with the development of turbulence theory and
14 computational technology, it has been more and more applied in the field of
15 NDDCT. Su et al.[10] numerically investigated fluid flow and temperature distribution
16 in and around a dry cooling tower. The study indicated that the wind-caused tangential
17 airflow at external sides of the heat exchanger is the major cause for the
18 cooling efficiency reduction, and the numerical results have a satisfied agreement with
19 full-scale measurement data. Du Preez et al. [11] carried out extensive experimental
20 and numerical research on the influence of the particular heat exchanger arrangement
21 on the performance of the tower in windy conditions. Practical suggestions
22 concerning the arrangement of heat exchanger and wind-break walls were

1 recommended to reduce the adverse effect of crosswind on the dry cooling tower
2 performance. Al-Waked et al. [3, 5] investigated the effects of crosswind and ambient
3 temperature on thermal performance of an NDDCT with a three-dimensional CFD
4 model which was verified by experimental results. To improve thermal performance
5 of the tower under windy conditions, the shape, dimensions, location and material of
6 the windbreak walls were discussed in detail. Moreover, thermal performance is
7 influenced by crosswind in the same manner at different ambient temperatures. Ma et
8 al. [12] also compared the effects of crosswind and ambient temperature on the dry
9 cooling tower performance with numerical simulation coupling the tower and
10 condenser. The pressure, temperature fields and streamlines were presented in their
11 study, and parameters of the system under steady state were obtained. Results
12 indicated that outlet water temperature of the tower is approximately linear with
13 ambient temperature, whereas nonlinear with crosswind, and reasons for this
14 conclusion were explained in detail. In addition, the computational models and
15 coupled method mentioned in their paper were proved reliable by operation data.

16 Although numerical simulation is quite popular in the research of the NDDCT,
17 it's never an easy job owing to the large scale of the tower and different geometric
18 sizes of different towers. Therefore, a simple theoretical tool is much needed for
19 scaling of such buoyancy-induced flows. Hooman [13] simplified heat exchanger as
20 porous media, presented scaling laws for the dry cooling tower, and studied the
21 turbulent free convection through the heat exchanger bundles and along the cooling
22 tower. Then, with scale analysis, he further presented a simple theoretical model to

1 predict the effect of crosswind on the performance of the NDDCT [14]. It's uplifting
2 to see the simple theoretical model established by Hooman has a good agreement with
3 available experimental data, and the maximum relative error is observed to be 15%.

4 However, as for his model, there still exists two main problems. The first one is
5 that his theoretical model can't evaluate heat transfer variation caused by ambient
6 temperature, while ambient temperature plays a crucial role in thermal performance of
7 the NDDCT. The other problem is that no method is presented in his model to
8 calculate the key parameter (i.e. outlet air velocity at tower exit under windless
9 condition, expressed as $(V_{o,n})_{NW}$), based on which can heat transfer variation caused
10 by crosswind be obtained. $(V_{o,n})_{NW}$ is closely related to the structure and size of the
11 tower, and there is no fixed $(V_{o,n})_{NW}$ for different NDDCTs. In addition, $(V_{o,n})_{NW}$
12 varies with initial temperature difference. Under a larger initial temperature difference,
13 the more airflow will be drafted into the tower, which induces the increase of $(V_{o,n})_{NW}$.
14 On the whole, a method is much needed for the calculation of $(V_{o,n})_{NW}$ for different
15 NDDCTs under various initial temperature differences.

16 This study aims to establish a new theoretical model based on some theories from
17 Ma [12] and Hooman [14]. In this new theoretical model, both the impacts of ambient
18 temperature and crosswind on thermal performance of the tower are taken into
19 account, and an algorithm is introduced to get the velocity of air flowing out of the
20 NDDCT. Using this model, heat rejection and outlet water temperature of the tower at
21 any specified ambient condition can be easily generated with the data of reference
22 condition obtained in advance. In addition to the example of a 600MW_{th} NDDCT, the

new theoretical model is also verified by other NDDCTs researched in the existing literatures.

2. Theoretical analysis of heat transfer variation caused by ambient temperature

The driving force in a NDDCT is the air density difference that, following the use of Boussinesq approximation, takes the following form[13]:

$$F_D = \rho_a g H_T \beta_a (t_{a,m} - t_a) \quad (1)$$

Here, under windless condition, flow and temperature fields of air in or around the tower are axially symmetric. In Ref [10], the difference between the highest temperature located near the radiator and the lowest temperature at the centre of the tower is only 1.1°C, and that is 1.2°C in Ref [5]. Thus, the distribution of air temperature inside the tower is nearly uniform, and taking $t_{a,m}$ as the average temperature of internal warm air under windless condition is reasonable.

When air passes through the tower, the resistance of tower to airflow is the sum of the resistance of each part of the tower. The resistance of each part is proportional to the dynamic head of air passing through this part. To simplify the calculation, resistance coefficient of each part is normalized to the heat exchanger inlet velocity and area, and then adding up all these converted resistance coefficients, the overall resistance coefficient is obtained. Therefore, the resistance of the tower is expressed as follows:

$$F_R = K_T \frac{\rho_a V_{i,n}^2}{2} \quad (2)$$

According to the equality between the draft and resistance of the tower, it can be

1 obtained combining Eqns. (1) and (2):

2
$$t_{a,m} - t_a = \frac{K_T V_{i,n}^2}{2\beta_a g H_T} \quad (3)$$

3 As for heat transfer between water and air at the tower entrance, the conductive
4 thermal resistance through the tube wall and the convective thermal resistance of
5 water are minor and can be neglected, compared with the convective thermal
6 resistance of air. Thus, heat transfer at the tower entrance can be regarded as
7 convective heat transfer between tube wall and air. Using lumped parameter radiator
8 model [1, 3, 5, 6], the tube wall temperature can be replaced with mean water
9 temperature, and the amount of heat transferred from hot water to cold air is:

10
$$Q_T = h A_e \left(\frac{t_{w1} + t_{w2}}{2} - t_a \right) \quad (4)$$

11
$$Q_T = \rho_a V_{i,n} A_i c_{p,a} (t_{a,m} - t_a) \quad (5)$$

12
$$Q_T = m_w c_{p,w} (t_{w1} - t_{w2}) \quad (6)$$

13 The air side convective heat transfer coefficient, h ,
14 is a function of normal velocity component [1-3], and can be specified as the following
15 form:

16
$$h = c V_{i,n}^f \quad (7)$$

17 Where c and f , which are performance parameters of heat exchanger, are specified as
18 constants.

19 Combining Eqns. (3), (4), (5) and (7), it can be obtained:

20
$$\frac{t_{w1} + t_{w2}}{2} - t_a = \frac{\rho_a A_i c_{p,a} K_T}{2 c A_e \beta_a g H_T} V_{i,n}^{3-f} \quad (8)$$

21 In Eq. (8), c , A_e , A_i and H_T are parameters of the tower or heat exchanger, and

remain constant for a certain NDDCT. ρ_a , β_a and $c_{p,a}$, which vary little with ambient temperature, are treated as constants. g , which is local gravitational acceleration, is specified. The overall resistance coefficient of the tower, K_T , is assumed as invariable regardless of temperature difference of heat exchanger here, because for most parts of the tower, the resistance coefficient is dependent on structure and size of this part [15, 16]. Then, Eq. (8) can be changed as the following form:

$$V_{i,n} \sim \left(\frac{t_{w1} + t_{w2}}{2} - t_a \right)^{\frac{1}{3-f}} \quad (9)$$

The heat transfer ratio (ε_a), which reflects thermal performance variation of the NDDCT with ambient temperature, is expressed as:

$$\varepsilon_a = \frac{Q_T}{Q_T^0} \quad (10)$$

With Eqns. (4), (7), (9) and (10), ε_a is changed as the following form:

$$\varepsilon_a = \frac{Q_T}{Q_T^0} = \left(\frac{\frac{t_{w1} + t_{w2}}{2} - t_a}{\frac{t_{w1}^0 + t_{w2}^0}{2} - t_a^0} \right)^{\frac{3}{3-f}} \quad (11)$$

In Eq. (11), the variable of t_{w2} is unknown, and will be calculated by Eq. (6) after Q_T is obtained. Therefore, it is necessary to further simplify Eq. (11) to remove the unknown variable. Next, a simplified method is introduced.

As has been well explained in Ref [12], when ambient temperature increases, heat rejection of the tower drops sharply, but will be offset by the overall ascent of cooling water temperature, and the indirect dry cooling system reaches to another steady state. Compared with the initial state, final heat rejection drops tinily. Moreover, in steady state, the inlet and outlet water temperature of the tower increases

linearly with ambient temperature, shown in Figure 1. Physically, lines L_{w1} and L_{w2} means inlet and outlet water temperatures of the tower at different ambient temperatures in steady state respectively. Geometrically, the equations of lines L_{w1} and L_{w2} can be expressed as $y=t_{w1}$ and $y=t_{w2}$ respectively. As mentioned above, after ambient temperature increases, the final cooling capability of the tower can't recover to, and is smaller than that of initial steady state. Therefore, the slope of line L_{w2} is larger than that of line L_{w1} , and lines L_{w1} and L_{w2} intersect at Point A. Then, line L_w with the equation of $y=(t_{w1}+t_{w2})/2$ can be drawn through Point A, at which the dry cooling tower loses heat transfer ability. This is caused by the equality of inlet water temperature of the tower (t_{w1}) and ambient temperature (t_a), i.e. heat transfer temperature difference of the tower is zero in this condition. Therefore, line L_a with the equation of $y=t_a$ passes through Point A as well. Then, two running states of the tower are pointed out in Figure 1, one is under reference condition with ambient temperature of t_a^0 , and the other is under any condition with ambient temperature of t_a . Lengths of segment BE , BD are equal to the values of $t_{w1}^0 - t_a^0$, $\frac{t_{w1}^0 + t_{w2}^0}{2} - t_a^0$ respectively, and lengths of segment FI , FH are equal to the values of $t_{w1} - t_a$, $\frac{t_{w1} + t_{w2}}{2} - t_a$ respectively. According to the theory of similar triangles, it can be obtained:

$$\frac{FI}{BE} = \frac{FH}{BD} = \frac{FA}{BA} \quad (12)$$

That is,

$$\frac{t_{w1} - t_a}{t_{w1}^0 - t_a^0} = \frac{\frac{t_{w1} + t_{w2}}{2} - t_a}{\frac{t_{w1}^0 + t_{w2}^0}{2} - t_a^0} \quad (13)$$

Here, it's important to note that Point A in Figure 1 can't be the actual point of the physical process. Because Point A means no heat transfer in the cold-end system of power plant, while any thermodynamic cycle can hardly do without cold source according to the second law of thermodynamics. In addition, there is a safe range of ambient temperature during the practical operation of the indirect dry cooling system. Ambient temperature affects outlet water temperature of the tower, and then impact backpressure of the condenser. However, too large or too small backpressure will affect the safe operation of the system. Thus, for the safe operation of the system, there exists upper and lower bound of ambient temperature, which can be calculated based on specified system under specified conditions.

$t_{wl} - t_a$ is named initial temperature difference of the NDDCT, expressed as:

$$ITD = t_{wl} - t_a \quad (14)$$

Then, with Eqns. (11), (13) and (14), ε_a is finally simplified as follows:

$$\varepsilon_a = \frac{Q_T}{Q_T^0} = \left(\frac{ITD}{ITD^0} \right)^{\frac{3}{3-f}} \quad (15)$$

During above formula derivation, some properties of air like density, specific heat and thermal expansion coefficient are treated as temperature independent, and then the sensitivity of such assumption is discussed. With Eqns (3) and (5), it can be obtained:

$$Q_T = \frac{\rho_a A_i K_T c_{p,a}}{2\beta_a g H_T} V_{i,n}^3 \quad (16)$$

Treating air as ideal gas, the ideal gas equation can be written as:

$$\frac{p}{\rho_a} = R_a t_a \quad (17)$$

As β_a is equal to $1/t_a$, then Eq. (16) can be changed as:

$$Q_T = \frac{pA_i K_T c_{p,a}}{2R_a g H_T} V_{i,n}^3 \quad (18)$$

Though $c_{p,a}$ which we set as constant should increase slightly with ambient temperature and Q_T will be exaggerated with our method at low ambient temperature, this deviation is tiny and negligible for $c_{p,a}$ varies little with ambient temperature.

3. Theoretical analysis of heat transfer variation caused by crosswind

As has been well documented in literatures [8,10], crosswind plays the role of a wind-cover and leads to a reduction in air exit mass flow rate, which further affects the tower intake and reduces cooling efficiency of the tower. Under windless condition, the generated plume rises vertically. With the increase of crosswind velocity, the plume that rises out of the tower gets deflected. A deflection angle θ is defined as the angle between normal plume rise pathlines (under windless conditions) and the actual flow pathlines (under windy conditions), as shown in Figure 2. To describe the relationship between velocity vectors and plume rise pathlines, a tangent function is defined as:

$$V_W = (V_{o,n})_{NW} \tan(\theta) \quad (19)$$

The tangent curve suggested in Eq. (19) is proved acceptable in Ref [14], and the variation of outlet air velocity compared to windless case is also researched and explained. A thin control volume, only extended to a short height outside the tower from a short distance inside the tower, is assumed around the tower exit, as shown in Figure 2. Due to the effect of crosswind, the rising plume gets deflected, and only normal component of the velocity can rise up and carry heat out of the tower while the

radial component drags apart of the rising plume away creating a circulating region. As

a result, the effective velocity out of the tower under crosswind can be expressed as:

$$(V_{o,n})_W = (V_{o,n})_{NW} \cos(\theta) \quad (20)$$

With Eqns. (19) and (20), it can be obtained:

$$\frac{(V_{o,n})_W}{(V_{o,n})_{NW}} = \frac{1}{\sqrt{1 + \left(\frac{V_W}{(V_{o,n})_{NW}} \right)^2}} \quad (21)$$

Eqns. (19), (20) and (21) are all presented by Hoomon in Ref [14]. Then, considering the same initial temperature difference of the NDDCT due to the same ambient temperature, he regarded the variation of heat transfer coefficient as the main reason for thermal performance variation of the tower under windy conditions, and simplified heat transfer ratio (windy divided by no wind) as:

$$\varepsilon_W = \frac{(Q_T)_W}{(Q_T)_{NW}} = \frac{(h)_W}{(h)_{NW}} \quad (22)$$

Considering mass conservation of airflow in and out of the tower, it can be obtained that:

$$A_i V_{i,n} = A_o V_{o,n} = A_s V_{s,n} \quad (23)$$

Then, Eq. (7) can be changed as the following form:

$$h = c \left(\frac{A_o}{A_i} \right)^f V_{o,n}^f \quad (24)$$

As a consequence, the heat transfer ratio (windy divided by no wind), ε_W , will be only a function of variation in air velocity, combining Eqns. (21), (22) and (24).

$$\varepsilon_W = \frac{(Q_T)_W}{(Q_T)_{NW}} = \frac{(h)_W}{(h)_{NW}} = \left(\frac{(V_{o,n})_W}{(V_{o,n})_{NW}} \right)^f = \left(1 + \left(\frac{V_W}{(V_{o,n})_{NW}} \right)^2 \right)^{-\frac{f}{2}} \quad (25)$$

In Eq. (25), normal outlet air velocity under windless condition, $(V_{o,n})_{NW}$, is unknown, as a result, the basic job for calculation is to get $(V_{o,n})_{NW}$. Here an algorithm is introduced to get $(V_{o,n})_{NW}$, and then obtain $(Q_T)_W$ under any ambient condition with initial temperature difference of ITD and crosswind velocity of V_W . Flow chart of the new theoretical model for thermal calculations of the NDDCT under variable ambient conditions is shown in Figure 3.

(1). First of all, a reference condition should be defined. It is better to take the condition under which thermal performance experiment of the tower is carried out as reference condition, because this thermal performance experiment is required before the operation of a newly built dry cooling tower, and we could get abundant experimental data of performance parameters of the tower, like $(t_a)_W^{ref}$, $(t_{a,m})_W^{ref}$, $(t_{w1})_W^{ref}$, $(t_{w2})_W^{ref}$, $(V_{i,n})_W^{ref}$, V_W^{ref} , m_w and so on. Generally, $(t_{a,m})_W^{ref}$ and $(V_{i,n})_W^{ref}$ are hard to measure, and we just need to know one of them.

(2). Next we need do some calculations. ITD^{ref} is obtained by Eq. (14), and $(Q_T)_W^{ref}$ is obtained by Eq. (6). To get $(Q_T)_{NW}^{ref}$ by Eq. (25), we need to know $(V_{o,n})_{NW}^{ref}$ in advance, and $(V_{o,n})_{NW}^{ref}$ can be calculated by solving Eq. (21), once $(V_{o,n})_W^{ref}$ is obtained.

$$(V_{o,n})_{NW}^{ref} = \sqrt{\frac{\left((V_{o,n})_W^{ref}\right)^2 + (V_{o,n})_W^{ref} \sqrt{\left((V_{o,n})_W^{ref}\right)^2 + 4(V_W^{ref})^2}}{2}} \quad (26)$$

There are two ways to get $(V_{o,n})_W^{ref}$. The first way is to compute $(V_{o,n})_W^{ref}$ directly with $(V_{i,n})_W^{ref}$ by Eq. (23), and the second way is to compute $(V_{o,n})_W^{ref}$ indirectly by Eq. (23) after $(V_{i,n})_W^{ref}$ is obtained by Eq. (5).

(3). After $(V_{o,n})_{NW}^{ref}$ and $(Q_T)_{NW}^{ref}$ are obtained, we can easily calculate performance parameters of the tower under any ambient condition with initial temperature difference of ITD and crosswind velocity of V_W . By Eq. (15), $(Q_T)_{NW}$ is obtained, and it can be deduced with Eqns. (9), (13), (14) and (23):

$$\frac{(V_{o,n})_{NW}}{(V_{o,n})_{NW}^{ref}} = \left(\frac{ITD}{ITD^{ref}} \right)^{\frac{1}{3-f}} \quad (27)$$

Therefore, $(V_{o,n})_{NW}$ is exported by Eq. (27). After that, $(Q_T)_W$ is calculated according to Eq. (25), and $(t_{w2})_W$ is obtained by Eq. (6) as well.

Although the above theoretical model is simple and fast in thermal performance prediction of the NDDCT, it has the restriction, that is, according to Eq. (25) proposed firstly by Hoomon [14], heat transfer efficiency decreases with crosswind, which disagrees with the relationship reported in Refs. [6, 12, 17] under a very high crosswind velocity. For the NDDCTs with heat exchanger arranged vertically, there exists a critical wind velocity, at which hydraulic performance of the dry cooling tower is most deteriorated with mass flow rate of intake air and heat rejection of the tower reaching to their minimum values, and outlet water temperature of the tower reaching to its maximum value at constant inlet water temperature. Moreover, heat rejection of the tower decreases with crosswind velocity lower than the critical wind velocity, whereas increases with crosswind velocity higher than the critical wind velocity. The reason is that under windy conditions, thermal performances of upwind radiators are superior to other radiators, and air mass flow rates of upwind radiators increase with crosswind, while those decrease for other radiators. With the increase of crosswind velocity up to the critical wind velocity, thermal performance of the whole tower is

directly dependent on thermal performances of upwind radiators, because intake air mass flow rates and heat rejections of other radiators vary slightly, and are incomparable with those of upwind radiators. In this case, the NDDCT acts more like a forced draft cooling tower with air flow pushed in from upwind radiators, inducing the increment of heat transfer efficiency, while the above theoretical model for evaluating wind-caused heat transfer variation with crosswind is unavailable any more under this condition. As a result, the following testing work is conducted at crosswind velocity lower than the critical wind velocity. The critical wind velocity is related to geometric parameters of the tower, and is no less than 10m/s for a NDDCT with base diameter larger than 100m, combining Refs. [6, 10, 12, 17].

4. Model validation and results analysis

It is convenient to obtain thermal parameters of the NDDCT under any specified ambient condition with above new theoretical model. A 600MW_{th} Hamon-type NDDCT is taken as an example. The tower is a hyperbolic shell-type tower, at the base of which air-cooled heat exchanger is arranged vertically. Specific parameters of the tower and heat exchanger are shown in Table 1. Thermal performance experiment of the tower was carried out, and measurement point arrangement is shown in Figure

4. Crosswind velocity is measured at the height of 10m with the cup anemometer accurate to $\pm(0.3+0.03V)$ m/s. Four measurement points of crosswind velocity are arranged 40m away outside the tower. Air temperature is measured with pt100 temperature sensor accurate to $\pm(0.15+0.2\%t)$ °C. Ten measurement points of ambient temperature are arranged 1m away around the tower at half of the radiator height, and

three measurement points of inside air temperature are equidistantly set along radius direction. Both the inlet and outlet water temperature are measured at the main pipe with pt100 temperature sensor accurate to $\pm(0.3+0.5\%)^{\circ}\text{C}$, and mass flow rate of water is measured by ultrasonic flowmeter with accuracy of $\pm 1\%$. All the processed experimental data is listed in Table 2.

With above original data of reference condition, operating parameters of the tower under any ambient condition are obtained using the new theoretical model proposed in this paper. Table 3 presents the comparison of calculated results with measurement data of the tower under different ambient conditions. The first ten sets of measurement data taken from control system of the unit is averaged by a period of time, during which power unit is stable running with hardly any fluctuations, and the second ten sets of measurement data is provided by Ref [12], owing to the same NDDCT researched in Ref [12] and this paper. As seen, theoretical approximate calculations are agreeing well with measurement data of the tower, and the maximum error of $(Q_T)_W$ using this new theoretical model is found to be less than 8%. Considering the mutability and complexity of practical operation of the indirect dry cooling system, the error is acceptable in engineering application.

Variation of heat transfer ratio ε_W with crosswind velocity V_W is shown in Figure 5. There is no doubt that ε_W decreases with the increase of V_W . Moreover variation of heat transfer ratio ε_W is closely related to initial temperature difference ITD . Under the larger ITD , ε_W drops more slowly with V_W . The reason is that the larger ITD is, the more air the tower draws in, and the deflection of the rising plume caused by crosswind reduces.

That is, under the larger ITD , the tower has the larger outlet air velocity under windless condition $(V_{o,n})_{NW}$, and then under the same crosswind velocity, the tower has the larger resistance to crosswind, which makes cooling efficiency declines more slowly. Table 4 lists $(V_{o,n})_{NW}$ of different NDDCTs under various conditions. As seen, $(V_{o,n})_{NW}$ is closely related to structure and size of the tower, and for a certain NDDCT, $(V_{o,n})_{NW}$ varies with ITD as well. As a result, it's inappropriate to use any constant $(V_{o,n})_{NW}$ for calculation, and to get accurate variation of heat transfer ratio ε_W with crosswind velocity V_W , $(V_{o,n})_{NW}$ should be clearly obtained in advance.

Variation of $(Q_T)_W$ with ITD is shown in Figure 6, and $(Q_T)_W$ increases almost linearly with ITD . The larger ITD , which means the larger temperature difference of heat exchanger, causes the increase of heat rejection of the tower with other conditions unchanged. The variation of ambient temperature has little impact on heat rejection distribution of the tower among sectors and leads to the uniform variation of heat transfer of all sectors [12]. This brings the comprehensive effect that heat rejection and outlet water temperature of the tower increases and decreases linearly with the drop of ambient temperature at constant inlet water temperature (i.e. linearly with the increase of ITD) respectively, which is in accord with the conclusion of Su [10]. However, the relationships of ITD to $(Q_T)_W$ (i.e. the slopes of these lines in Figure 6) are different at various crosswind velocities. The slope, which means the growth of $(Q_T)_W$ for each degree rise in ITD , is larger under a smaller crosswind velocity. That is ITD has a greater impact on cooling capability of the tower under a smaller crosswind velocity, and heat rejection of the tower increases more

rapidly with *ITD* under a smaller crosswind velocity.

The new theoretical model for thermal performance prediction of the NDDCT can be used following more than the procedure showed in Figure 3. One can change calculation order flexibly, as long as the number of the distinct equations employed is no less than the number of the unknown variables in these equations. Then, the new theoretical model is flexibly applied to thermal calculation for other NDDCTs. Figure 7 shows the results calculated by present new theoretical model versus the data available in Ref [10]. As seen, theoretical calculations agree well with the simulations under various ambient temperatures (Figure 7(a)) and crosswind velocities (Figure 7(b)). Testing work for another NDDCT researched in Refs [17, 19] is conducted as well using this new theoretical model, as shown in Figure 8. When ambient temperature increases from 32°C to 42°C, the increment of outlet water temperature is reported to be 10.5°C, which is close to the increment of 10.0°C obtained with this theoretical model, as shown in Figure 8(a). The variation of outlet water temperature with crosswind coincides well with that in Ref [17] under crosswind velocity less than the critical wind velocity, as shown in Figure 8(b). For a NDDCT with base diameter larger than 100m, the critical wind velocity is usually larger than 10m/s, which is generally large and uncommon during the practical operation of the tower. In consequence, the present theoretical model is still feasible for thermal performance prediction, which has been verified by above testing work.

5. Conclusion

Based on previous theories, a new theoretical model is proposed to calculate heat

rejection of the tower under any specified ambient condition. The theoretical model considers both the effects of ambient temperature and crosswind on thermal performance of the tower. When evaluating wind-caused heat transfer variation, an algorithm is introduced to get the key parameter (i.e. outlet air velocity at tower exit under windless condition), which is related to not only the structure and size of the tower, but also the initial temperature difference.

The new theoretical model is unsuitable for the condition with crosswind velocity higher than the critical wind velocity, which is generally large and uncommon in the practical operation of the tower. As a result, the theoretical model is still meaningful for thermal performance predictions of the NDDCTs, and it is demonstrated effective by the example of a 600MW_{th} NDDCT with the maximum error of heat rejection less than 8%. The example analysis shows that under a larger initial temperature difference, the tower has the larger resistance to crosswind and the cooling efficiency declines more slowly, furthermore, heat rejection of the tower increases more rapidly with initial temperature difference under a smaller crosswind velocity.

Changing calculation order, the new theoretical model can be applied flexibly, as long as the number of the distinct equations employed is no less than the number of the unknown variables in these equations. Testing work for the other two NDDCTs is conducted using this theoretical model, and the predictions have a satisfied agreement with results available in the existing literatures. Testing work re-verifies the effectiveness of the new theoretical model, which is available for thermal performance

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- 1 prediction of the NDDCT and beneficial for operation decision of the tower.

Nomenclature

A	area (m^2)
A_e	total heat dissipation area of heat exchanger (m^2)
c	performance parameter of heat exchanger
c_p	specific heat ($\text{J kg}^{-1} \text{K}^{-1}$)
f	performance parameter of heat exchanger
F_D	the draft of the tower (Pa)
F_R	the resistance of the tower (Pa)
g	gravitational acceleration (m s^{-2})
h	air side convective heat transfer coefficient of heat exchanger ($\text{W m}^{-2} \text{K}^{-1}$)
H	height (m)
ITD	initial temperature difference of the NDDCT (K)
K_T	the overall resistance coefficient of the tower
m	mass flow rate (kg s^{-1})
p	pressure(Pa)
Q	heat rejection (W)
R	gas constant ($\text{J kg}^{-1} \text{K}^{-1}$)
t	temperature (K)
$t_{a,m}$	the average air temperature inside the tower (K)
V	the velocity of air flow (m s^{-1})

Greek symbols

ρ	density (kg m^{-3})
β	thermal expansion coefficient (K^{-1})
ε	heat transfer ratio
θ	deflection angle of plume rise pathlines ($^\circ$)

Superscripts

ref	reference condition
0	reference condition under windless condition

Subscripts

a	ambient air
i	tower inlet(or windward surface of heat exchanger)
n	normal direction
NW	no crosswind
o	tower outlet
r	radial direction

s	any section of the tower
T	tower
w	circulating cooling water
W	crosswind
1	inlet
2	outlet

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Table caption

Table 1. Geometric parameters of the tower and performance parameters of heat exchanger

Table 2. Data collection of measurement points in the experiment of the NDDCT

Table 3. The comparison of results calculated by presented theoretical model with measurement data under the measurement conditions

Table 4. Outlet air velocities of different NDDCTs under windless condition with various initial temperature differences

Figure caption

Figure 1. The relationship of cooling water temperature with ambient temperature

Figure 2. The relationship of outlet air velocity under windless condition with crosswind velocity and the variation of outlet air velocity caused by crosswind

Figure 3. Flow chart of the new theoretical model for thermal calculation of the NDDCT under variable ambient conditions

Figure 4. Measurement points arrangement for thermal performance experiment of the NDDCT

Figure 5. Variations of heat transfer ratio ε_W with crosswind velocity V_W under different $ITDs$

Figure 6. Variations of heat rejection of the tower $(Q_T)_W$ with ITD under different crosswind velocities

Figure 7. Comparison of predictions by present theoretical model and simulations in Su et al. [10], (a) in the condition: $V_W=0\text{m/s}$, $(t_{w1})_W=43^\circ\text{C}$; (b) in the condition: $(t_{w1})_W=43^\circ\text{C}$, $(t_a)_W=15^\circ\text{C}$

Figure 8. Comparison of predictions by present theoretical model and results reported in Zhao et al. [17, 19], (a) in the condition: $(Q_T)_W=887.2\text{MW}$, $V_W=4\text{m/s}$; (b) in the condition: $ITD=29.2^\circ\text{C}$ ($(t_{w1})_W=43.1^\circ\text{C}$, $(t_a)_W=13.9^\circ\text{C}$); $ITD=27.8^\circ\text{C}$ ($(t_{w1})_W=39^\circ\text{C}$, $(t_a)_W=11.2^\circ\text{C}$)

Title: A new theoretical method for predicating the part-load performance of natural draft dry cooling towers

Authors' names: Huan Ma, Fengqi Si *, Yu Kong, Wensheng Yan, Kangping Zhu

Geometric parameters of the tower and performance parameters of heat exchanger	property parameters of air and water	Thermal parameters of the tower under reference condition
A_i, A_o, H_T, f	$\rho_a, c_{p,a}, c_{p,w}$	$(t_a)_W^{ref}, (t_{w1})_W^{ref}, (t_{w2})_W^{ref}, m_w$ $V_W^{ref}, (t_{a,m})_W^{ref}$ or $(V_{i,n})_W^{ref}$

A new theoretical model is proposed for thermal calculation of natural draft dry cooling tower (NDDCT). Both the effects of ambient temperature and crosswind are taken into account in this model, in which an algorithm is introduced to get the velocity of air flowing out of the tower. Despite the ineffectiveness at crosswind speed higher than the critical wind speed, this model is still meaningful for thermal performance predication by the verification of different NDDCTs.

Table 1:

Geometric parameters of the tower and performance parameters of heat exchanger

	Unit	Value
Base diameter of tower	m	155
Height of tower	m	173
Outlet diameter of tower	m	91
Throat diameter of tower	m	88
Height of heat exchanger	m	30
Total heat dissipation area of heat exchanger	m ²	1588700
Windward surface area of heat exchanger	m ²	23396
Performance parameter of heat exchanger (<i>c</i>)		30.86
Performance parameter of heat exchanger (<i>f</i>)		0.41

Table 2:

Data collection of **measurement points** in the experiment of the NDDCT

Measurement point	name	Symbol	Unit	Number	Value
ambient air temperature		$(t_a)_W^{ref}$	°C	1	26.9
				2	26.9
				3	26.7
				4	26.9
				5	26.9
				6	26.7
				7	26.3
				8	26.6
				9	26.8
				10	27.0
				average	26.8
inside air temperature		$(t_{a,m})_W^{ref}$	°C	1	50.0
				2	49.1
				3	48.8
				average	49.3
inlet water temperature		$(t_{w1})_W^{ref}$	°C	1	59.7
				2	59.6
				average	59.7
outlet water temperature		$(t_{w2})_W^{ref}$	°C	1	50.0
				2	50.0
				average	50.0
crosswind velocity		V_W^{ref}	m/s	1	2.8
				2	2.6
				3	2.6
				4	2.5
				average	2.6
water	mass flow rate	m_w	t/h	70300	

The comparison of results calculated by presented theoretical model with

measurement data under the measurement conditions

Figure 1. The relationship of cooling water temperature with ambient temperature

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(a)

(b)

Figure 7. Comparison of predictions by present theoretical model and simulations in Su et al. [10], (a) in the condition: $V_W=0\text{m/s}$, $(t_{w1})_W=43^\circ\text{C}$; (b) in the condition: $(t_{w1})_W=43^\circ\text{C}$, $(t_a)_W=15^\circ\text{C}$

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