**Wind-break walls with optimized setting angles for natural draft dry cooling
tower with vertical radiators**

Huan Ma¹, Fengqi Si^{1,*}, Yu Kong¹, Kangping Zhu², Wensheng Yan²

Abstract:

To get larger cooling performance enhancement for natural draft dry cooling tower with vertical cooling deltas under crosswind, setting angles of wind-break walls were optimized. Considering specific structure of each cooling delta, an efficient numerical model was established and validated by some published results. Aerodynamic fields around cooling deltas under various crosswind speeds were presented, and outlet water temperatures of the two columns of cooling delta were exported as well. It was found that for each cooling delta, there was a difference in cooling performance between the two columns, which is closely related to the characteristic of main airflow outside the tower. Using the present model, air inflow deviation angles at cooling deltas' inlet were calculated, and the effects of air inflow deviation on outlet water temperatures of the two columns for corresponding cooling delta were explained in detail. Subsequently, at cooling deltas' inlet along radial direction of the tower, setting angles of wind-break walls were optimized equal to air inflow deviation angles when no airflow separation appeared outside the tower, while equal to zero when outside airflow separation occurred. In addition, wind-break walls with optimized setting angles were verified to be extremely effective, compared to the previous radial walls.

Key words: dry cooling tower; numerical simulation; cooling performance; crosswind; inflow deviation; wind-break wall

1. Introduction

As key cooling equipment in thermodynamic cycle, the performance of natural draft cooling tower is great important for the economic and safe operation of the power unit. Compared with natural draft wet cooling tower (NDWCT), natural draft dry cooling tower (NDDCT) reduces the loss of steam and water caused by evaporation and blow-down, and has been widely applied in the regions with rich coal and lack of water [1]. Air movement in the NDDCT is dependent on the density difference between the entering cold air and internal warm air. When passing through the tower, the entering cold air absorbs heat from hot water inside the radiator bundles by indirect contact [2]. Then this warm air rises, meanwhile fresh air gets sucked into the tower. As a consequence, cooling performance of the NDDCT is highly sensitive to ambient meteorological conditions, particularly the crosswind [3].

There are two types of the NDDCTs, one is the NDDCT with horizontal A-frame radiators in its inlet cross section (NDDCTH), and the other is the NDDCT with vertical delta radiators around its circumference (NDDCTV).

With regard to NDDCTH, Du Preez et al. [4] illustrated the influence of crosswind on thermal performance of the cooling tower using numerical procedure, and results showed low speed crosswind had little adverse effect on the performance of NDDCTH, but high speed reduced cooling capacity of NDDCTH sharply. Al-Waked et al. [5] reconfirmed the crosswind adverse impact on the cooling performance of NDDCTH, and pointed out the low static pressure at tower laterals reduced the air mass inflow rate there. To lessen the adverse effect of crosswind on

cooling performance of NDDCTH, Du Preez et al [4] proposed the cross wall erecting under the horizontal radiators, and numerical simulation showed the cross wall reduced the distortion of aerodynamic field around the horizontal radiators and then improved thermal performance effectively. As a basis, Al-Waked et al. [6] introduced external wind-break walls installing every 45 degrees along tower circumference, and an improvement of 15% heat rejection rate was reported (compared to no-break wall case) at crosswind speed of 10 m/s. Effects of wind-break walls of various characteristics and combinations on thermal performance of NDDCTH, were further investigated numerically. The best thermal performance was achieved by installing the wind-break walls both inside and outside the tower simultaneously. The cooling performance improvement of NDDCTH by windbreak measures was discussed by Lu et al. [7] as well.

Compared with the equivalent NDDCTH, NDDCTV has the advantage of more cooling capacity, less initial cost, self-supporting ability, simple water distribution and installation [4]. In addition, the inner space freed up can be used as the location arranging flue gas desulfurization and stack to improve the plume dispersion characteristics [8]. As a result, NDDCTVs are applied more widely nowadays. However, NDDCTV is reported to be more susceptible to the crosswind impact than NDDCTH [9]. It is more significant to figure out the effect of crosswind on thermal performance of NDDCTV and propose appropriate improving measures.

Wei et al. [10] experimentally investigated the unfavorable effects of crosswind on the performance of NDDCTV and explained the reasons for the deterioration of

cooling efficiency reduced by crosswind, i.e., an unfavorable pressure distribution at the tower entrance, breaking of the plume rising at the tower exit, and air inflow induced by leading edge separation. Su et al. [11] obtained fluid flow and temperature distribution in and around a dry cooling tower through numerical simulation. Results revealed that air flowing into the tower from upwind radiators shocked with the flow from the opposite side, which formed a large recurrent flow in the bottom of the tower and reduced the discharge of upstream out of the tower. Yang et al. [8] also studied the performance of a dry cooling power plant at various wind speeds and in various wind directions by means of numerical simulations. To enhance cooling performance of NDDCTV under crosswind, Zhai et al. [12] recommended placing two solid wind-break walls symmetrically outside the tower flanks, perpendicular to crosswind direction. Using the same wind-break walls layout, Goodarzi et al. [13] investigated the enhancement in cooling performance with wind-break walls of radiator-type. Numerical results showed that radiator type wind-break walls could substantially more improve the cooling efficiency than the usual solid types do. Zhao et al. [9] suggested external air deflectors installed at interval of two cooling deltas along tower circumference. It was found that the air deflectors improved the aerodynamic field around cooling deltas, and then enhanced cooling performance of the NDDCTV greatly. Throughout the layouts of wind-break walls for NDDCTVs, the general way is installing external wind-break walls in radial directions of the tower regardless of the amount. Though radial walls can gather and induce airflow into the tower, separate vortex will appear at the back of the wall, and then interferes the

aerodynamic field around cooling deltas [12]. Therefore, in some cases, the radial walls can hardly enhance cooling performance, owing to the interference on air inflows.

In this paper, setting angles of wind-break walls are optimized according to air inflow deviations of cooling deltas at corresponding circumference angles. As Fig. 1(a) shows, under low crosswind speeds when no flow separation appears and the inflow air of cooling delta comes from outside main airflow, setting angle of wind-break wall (θ_{wl}) is set equal to air inflow deviation angle (θ_{ad}), which is helpful to reduce flow interference caused by the vortex at the back of the wall. While, under very high crosswind speeds when flow separation occurs and the inflow air of leeward cooling delta comes from the induced secondary flow, θ_{wl} is set equal to 0° to enhance the outside secondary flow, using wind-break walls as spoilers. In this case, the wind-break wall is installed in the radial direction of the tower, as shown in Fig. 1(b).

To get air inflow deviation angles at different cooling deltas' inlet under crosswind, the numerical model will be established. In this model, specific structure of the cooling delta composed of two columns is taken into account, and lumped parameter method is employed to deal with the heat transfer between hot water and cold air in these cooling deltas. Under crosswind, aerodynamic fields around cooling deltas can be presented, and outlet water temperatures of the two columns of cooling deltas at various circumferential angles will be exported as well. The variations of outlet water temperatures of the two columns with circumferential angle, induced by inflow deviation, can be explained by inflow air velocity analysis. Subsequently,

cooling performance improvement induced by wind-break walls with optimized setting angles will be verified, compared to the previous radial walls.

2. Numerical model

2.1 Natural draft dry cooling tower with vertical radiators

The studied 600MW_{th} NDDCTV in Fig. 2(a) has a total height of 173m, a base diameter of 155m, a tower bottom height of 28m, a throat diameter of 88m and an out diameter of 91m. For the symmetrical structure of NDDCTV along crosswind direction, half NDDCTV is taken into consideration. Each cooling delta with finned tube bundles inside consists of cooling Column A and B with an intersection angle about 48° ($\alpha=48^\circ$) at the end. These cooling deltas around the half NDDCTV compose 5 cooling sectors numbered as Sector 1-5 from $\theta_D=0^\circ$ to $\theta_D=180^\circ$, where θ_D is the circumferential angle of cooling delta, shown in Fig. 2(b), and circumferential angle of cooling Column A is smaller than that of Column B.

2.2 Meshing and mesh independence

As the structure of NDDCTV is symmetrical along crosswind direction, only a half computational domain is modeled and simulated to reduce the work of meshing and calculation, as Fig. 3 shows. To get high quality grids, the half computational domain is meshed with software Gambit using multi-block hybrid approach. The meshing job starts from the cooling columns which are meshed with hexahedral structured grids, as Fig. 4(a) shows, and the triangular zones between the two columns are meshed with hexahedral grids. Then, for the two transition zones near cooling columns, successive ratio increasing grids of triangular prism are utilized. The

hexahedral grids are adopted for tower body, shown in Fig. 4(b), and triangular prism grids are employed for the middle annular zone. The outermost annular zone is meshed with successive ratio increasing grids of hexahedral structure.

Three grid systems with mesh cells of 4 736 468, 5 856 218, 7 233 090 were generated, and tested at crosswind speed of 4m/s and ambient temperature of 14.5°C respectively. The test result showed that heat rejection of the whole tower varies less than 0.32% between the two highest grid density solutions, so grid number of 5 856 218 was finally accepted.

2.3 Governing equations

In the NDDCTV, heat transfer drives air to flow, while air flow affects heat transfer in turn. These physical processes occur under atmospheric temperature and pressure, and air static pressure through the tower varies just about 1-2% of the local atmospheric pressure [14]. As a result, air can be assumed as an incompressible ideal gas with boussinesq hypothesis dealing with buoyancy force [15]. Based on the height of cooling column, the Grashof number is near 10^{11} , which is far bigger than the air flow transition critical value from laminar flow to turbulent flow [16]. For the better performance solving flow separation, the realizable k - ε model [3] is employed. As what researched in present paper is under steady conditions, air and heat flows in and around the tower can be described by the steady state Reynolds Averaged Navier-Stokes equations, as follows:

$$\nabla \cdot (\rho \vec{v} \varphi - \Gamma_{\varphi} \nabla \varphi) = S_{\varphi} \quad (1)$$

where, ρ is air density, $\vec{v}$ is air velocity vector, φ is variable. Γ_{φ} is diffusion

coefficient for variable φ , and S_φ is the source term including internal source term of air governing equation and user defined source term describing complex physical problem. For mass, momentum and energy conservation equations, φ stands for 1, air velocity component v_x (or v_y, v_z), and air temperature t_a respectively, and φ stands for turbulent kinetic energy k and turbulent dissipation rate ε as well, for turbulent equations.

Air pressure drop across each column of the cooling delta is assumed to be proportional to the dynamic head, expressed as follows [5]:

$$\Delta p = 0.5k_l \rho v_n^2 \quad (2)$$

As for heat transfer between air and water at the tower entrance, the conductive thermal resistance through the tube wall and the convective thermal resistance of water can be neglected, compared with the convective thermal resistance of air [9]. Then, the lumped parameter method is adopted to characterize each cooling delta. Under windy conditions, the direction of air flowing into the delta can't be the symmetrical direction of this delta, so cooling Column A and B should be modeled respectively. For Column A of the j_{th} delta, the heat rejection can be expressed as [12]:

$$Q_{D_{jA}} = \sum_e h_{D_{jA}}^e \left(\frac{t_{w1} + t_{w2,D_{jA}}}{2} - t_a \right) A_{D_{jA}}^e \quad (3)$$

Where, superscript e denotes the e_{th} cell of the j_{th} Column A zone after meshed, and heat rejection of Column B of the j_{th} delta has the similar expression.

Both the pressure drop coefficient and convective heat transfer coefficient can be expressed as functions of face velocity [6], and by experiment, the coefficients are

computed as follows:

$$k_l = 22.25v_n^{-0.24} \quad (4)$$

$$h = 36.21v_n^{0.45} \quad (5)$$

Before the simulation, a constant inlet water temperature and an assumed outlet water temperature are specified for the two columns of all cooling deltas. For Column A of the j_{th} delta, outlet water temperature is obtained after each iteration [17]:

$$t_{w2,D_jA} = t_{w1} - Q_{D_jA} / (c_w m_{w,D_jA}) \quad (6)$$

and so do the outlet water temperature of Column B. Then, the outlet water temperatures of Column A and B are updated for next iteration.

2.4 Boundary conditions and solution method

Fig.3 shows the boundary conditions for air flowing around the NDDCT. The windward surface, with boundary temperature set as ambient temperature, takes boundary type of pressure inlet and velocity inlet under windless and windy conditions respectively. The crosswind velocity profile at the windward surface obeys the power law [18], defined as follows:

$$v_{cw,z} = v_{cw} \left(\frac{z}{10} \right)^r \quad (7)$$

where z denotes the vertical height above the ground, v_{cw} is referential crosswind speed at the height of 10m. The exponent r is related to the ground roughness and the atmosphere stability, taken as 0.16 [12] in this paper.

The top surface and leeward surface take the boundary type of pressure outlet. At top and leeward surfaces, air static pressure is equal to the atmosphere pressure, backflow air temperature is set as ambient temperature, and backflow air is specified

as that from neighboring cell.

The symmetry boundary is set for the symmetry surface, at which the gradients of all the evaluated variables are set as zero. The external wind-break walls are installed at the positions of $\theta_b=0^\circ, 36^\circ, 72^\circ, 108^\circ, 144^\circ, 180^\circ$ for the 5 sectors, as shown in Fig. 2(b). All solid walls including ground, tower body, cooling column walls and wind-break walls are set as adiabatic and no slip wall boundary, with standard wall function applied for near-wall treatment of turbulent flow.

Turbulence level of ambient air is very low at above far field boundaries, including windward, surface and surface. At these boundaries, turbulence intensity and viscosity ratio can be approximated as 0.1% and 0.1 respectively [9].

The governing equations are discretized using the first-order upwind differencing scheme, and the SIMPLE algorithm is employed in pressure-velocity coupling. The convergence criterion is that the scaled residuals for governing equations drop to less than 10^{-3} except that for energy equation less than 10^{-6} , and the monitored variables (i.e. average outlet water temperature and average air exit velocity from tower top) remain quite stable when iterating. The calculation process is iterated for more than 4000 steps and converged results can be obtained. All the governing equations with associated boundary conditions were numerically solved by FLUENT.

3. Results and discussion

3.1 Air inflow deviation along circumferential angle and its effects on outlet water temperatures of the two columns of corresponding cooling delta

Numerical simulations are conducted under the ambient conditions of design,

and the results are presented in Table 1 in which the calculated outlet water temperatures of the tower are close to the design ones, with the maximum error of tower heat rejection less than 3.5%. Based on Condition 1, cooling performances of the tower under various crosswind speeds are researched, using this credible numerical model.

Under crosswind, air flow around the tower is similar to the flow around a circular cylinder. Velocity of airflow at cooling delta' inlet increases with θ_D from 0° to 90° and then decreases with θ_D from 90° to 180° , and so does the air inflow deviation angle (θ_{ad}), comparing the five subgraphs in Fig. 5. For one cooling delta, more airflow enters into Column B than Column A, which induces cooling performance of Column B better than that of Column A. With the increase of θ_{ad} up to a critical value, the main airflow couldn't directly rush into Column A, which is actually cooled down by the secondary flow induced in the triangular zone, shown in Fig. 5(b). Then, with the further increase of θ_{ad} , the main airflow velocity normal to cooling delta decreases, so does the intensity of the secondary flow induced in the triangular zone. Therefore, cooling performance of Column A continuous to deteriorate at θ_D from 0° to 90° , as shown in Fig. 6, and cooling performance variation of Column A at θ_D from 90° to 180° can be explained conversely. What's more, there are two main valleys on the line of $t_{w2,DB}$ with θ_D , one is at θ_D less than 90° , and the other is at θ_D larger than 90° . This phenomenon appears except at windless condition, seen in Fig. 7. Under windless condition, the two columns of each cooling delta have nearly the same cooling performance, seen in Fig. 7(a). With the

increase of crosswind speed, abscissa values of the two valleys are moving away from 90° gradually, combining Fig. 7(b)-7(d). However, at crosswind speed larger than 10m/s, airflow separation occurs at the back of the tower, which induces the direction variation of air flowing into cooling deltas, and air flows into the leeward cooling delta in the direction of facing Column A, just as Fig. 1(b) shows. In this condition, Column A is cooled down by more airflow than Column B, which induces $t_{w2,DA}$ smaller than $t_{w2,DB}$ for the leeward cooling deltas, seen in Fig. 7(e) and 7(f). This airflow separation greatly affects flow characteristic of leeward cooling delta, but doesn't change that of windward cooling delta, and the abscissa value of Valley 1 is still moving away from 90° , with the increase of crosswind speed. Therefore, taking any cooling delta at θ_D from 0° to 90° as an example, air inflow velocity analysis is shown below.

Velocity vector of air flowing into the cooling delta is $\vec{v}_D$, seen in Fig. 8, and by angle calculation and velocity decomposing, air inflow velocities normal to Column A and B (i.e. $v_{n,DA}$ and $v_{n,DB}$) can be expressed as:

$$v_{n,DA} = v_D \sin\left(\frac{\alpha}{2} - \theta_{ad}\right) \quad (8)$$

$$v_{n,DB} = v_D \sin\left(\frac{\alpha}{2} + \theta_{ad}\right) \quad (9)$$

Comparing Eq. (8) and (9), it's easy to find that in the same condition, $v_{n,DB}$ is larger than $v_{n,DA}$, seen in Fig. 8(a) and 8(b), which confirms that cooling performance of Column B is better than that of Column A. When θ_{ad} increases larger than $\alpha/2$, $v_{n,DA}$ is calculated less than zero by Eq. (8), which means no main airflow rushes into Column A. But that can't be realistic, owing to the draft induced during heating process of air,

Column A is actually cooled down by the secondary flow induced in the triangular zone, as shown in Fig. 8(c). Moreover, at θ_{ad} larger than $\alpha/2$, the main airflow can't rush into the whole radiator surface of Column B, and the secondary flow in the triangular zone will flow into Column B as well. The secondary flow occurred in the triangular zone is hardly described by formulas, and its deviation angle is also unknown. As a result, flow intensity ratio ϕ is defined as:

$$\phi = v_{n,DB}^{sf} / v_{n,DB} \quad (10)$$

where $v_{n,DB}^{sf}$ is the velocity of the secondary flow normal to surface of Column B, seen in Fig. 8(c). Then, for Column B of unit height at θ_{ad} larger than $\alpha/2$, volume flow rate of inflow air can be calculated as:

$$qv_{DB} = v_{n,DB}^{sf} \cdot l \cdot \left(\cos \alpha - \frac{\sin \alpha}{\tan\left(\frac{\alpha}{2} + \theta_{ad}\right)} \right) + v_{n,DB} \cdot l \cdot \left(1 - \left(\cos \alpha - \frac{\sin \alpha}{\tan\left(\frac{\alpha}{2} + \theta_{ad}\right)} \right) \right) \quad (11)$$

With Eq. (9), (10) and (11), qv_{DB} can be expressed as:

$$qv_{DB} = l \cdot v_D \sin\left(\frac{\alpha}{2} + \theta_{ad}\right) \left(1 - (1 - \phi) \left(\cos \alpha - \frac{\sin \alpha}{\tan\left(\frac{\alpha}{2} + \theta_{ad}\right)} \right) \right) \quad (12)$$

Then, combining Eq. (9) and (12), air inflow velocity normal to Column B can be modified as:

$$v_{n,DB} = \begin{cases} v_D \sin\left(\frac{\alpha}{2} + \theta_{ad}\right) & \theta_{ad} < \frac{\alpha}{2} \\ v_D \sin\left(\frac{\alpha}{2} + \theta_{ad}\right) \left(1 - (1 - \phi) \left(\cos \alpha - \frac{\sin \alpha}{\tan\left(\frac{\alpha}{2} + \theta_{ad}\right)} \right) \right) & \theta_{ad} \geq \frac{\alpha}{2} \end{cases} \quad (13)$$

To get the variation of $v_{n,DB}$ with θ_D , $v_{n,DB}$ is dealt with dimensionless method, as

shown in Fig. 9. The non-dimensional inflow velocity normal to Column B (i.e. $v_{n,DB}/v_{D,\theta_D=0^\circ}$) is multiplied by 3 elements (i.e. $ND1$, $ND2$ and $ND3$). With the increase of θ_D , $ND1$ increases, but $ND1$ increases more sharply at a smaller θ_D under a larger crosswind speed. As θ_{ad} increases more sharply with θ_D under larger crosswind speeds shown in Fig. 10, $ND2$ will reach to its max value (i.e. 1) quickly at a smaller θ_D , and then decreases with θ_D . For the same reason, $ND3$ decreases more sharply with θ_D under a larger crosswind speed. Then multiplying the three elements, there is a peak on the line of $v_{n,DB}$ with θ_D , and with the increase of crosswind speed, the abscissa value of the peak moves away from 90° gradually. The abscissa value of Peak 1 in Fig. 9(a) is in accord with that of Valley 1 in Fig. 6, at ϕ equal to 0.25, and the abscissa value of Peak 1 in Fig. 9(b) is close to that of Valley 1 in Fig. 7(d) with error smaller than 3° , still at ϕ equal to 0.25. All these demonstrate air inflow velocity analysis in present paper is credible, and the introduced flow intensity ratio ϕ which is assumed as 0.25 is reasonable as well. Above air inflow velocity analysis is also suitable for the cooling delta at θ_D from 90° to 180° under the condition that no airflow separation appears, but needs to be further improved when airflow separation occurs.

3.2 Improving cooling performance by wind-break walls with optimized setting angles according to air inflow deviation

Fig. 11(a) shows the dimensionless outlet water temperature variation of cooling delta along circumferential angle at crosswind speed of 5m/s. Outlet water temperature of each cooling delta is averaged by those of Column A and B, and

$t_{w2,D}^{\max}$ is the maximum outlet water temperature along θ_D , while $t_{w2,D}^{\min}$ is the minimum. Owing to the different methods employed to model the NDDCTV, the little deviation among the three variation curves is acceptable. To figure out cooling performance differences among the sectors at various crosswind speeds, Fig. 11(b) shows the non-dimensional normal velocities of air flowing into different sectors. For the same sector, three variation curves show a similar trend. That is under windy conditions, more air flows into cooling deltas of windward Sector 1 than other sectors, while the least air flows into sideward Sector 3.

To improve cooling performance of the tower and reduce performance difference among the sectors, an optimized layout of external wind-break walls is proposed. For the leeward or the nearby sideward cooling delta, the inflow air may come from the outside main airflow or the secondary flow induced by airflow separation. But it can be easily distinguished by air inflow deviation angle, as shown in Fig. 10. If deviation angle is between 0° and 90° , which means inflow air of this delta comes from outside main airflow, thus the external wind-break walls should act as flow deflectors to gather more airflows, without interfering outside main airflow. As a result, the setting angles of optimized walls are equal to air inflow deviation angles. If deviation angle is less than 0° , which means the inflow air of this delta comes from the secondary flow at the back of the tower, or even worse, hot air flows out from this delta, then wind-break walls should act as spoilers to enhance the induced secondary flow. Thanks to the radial walls as great obstructions, the intensity of outside secondary flow will get improved. In this condition, the optimized walls are radial walls exactly,

installed in the radial directions of the tower. As cooling deltas are divided into five sectors, the external wind-break walls are installed at the positions of $\theta_D=0^\circ, 36^\circ, 72^\circ, 108^\circ, 144^\circ, 180^\circ$, with a height equal to cooling delta and width equal to its half height.

To obtain setting angles of wind-break walls under various crosswind speeds, air inflow deviation angles should be obtained in advance. Though they are partly shown in Fig. 10, the five curves at v_{cw} smaller than 10m/s (i.e. under the condition with no airflow separation appearing) are nearly symmetric about straight line of $\theta_D=90^\circ$. Therefore, air inflow deviation angles at $\theta_D=0^\circ, 36^\circ, 72^\circ, 108^\circ, 144^\circ, 180^\circ$ can be expressed as the following general computational formula:

$$\theta_{ad,i} = \theta_{wl,i} = C1 \cdot (90 - 36 \cdot i)^2 + C2 \quad (14)$$

where $i=0,1,2,3,4,5$, and if $\theta_{wl,i}$ is calculated larger than 90° , which means the inflow air of corresponding sector doesn't come from the outside main airflow, $\theta_{wl,i}$ is reset equal to 0° , that is, wind-break walls are installed in the radial directions of the tower.

$C1$ and $C2$ are polynomial coefficients, and can be approximately fitted as the functions of the crosswind speed:

$$C1 = -0.0032 \ln(v_{cw}) - 0.0037 \quad (15)$$

$$C2 = 28.8 \ln(v_{cw}) + 26.9 \quad (16)$$

Table. 2 shows the effects of both radial walls and optimized walls on outlet water temperatures of the five sectors at 3 representative crosswind speeds. For each sector, outlet water temperature is averaged by that of each cooling delta in this sector. At low crosswind speed of 4m/s, optimized walls perform better than radial walls.

With optimized walls, the average outlet water temperature is slightly lower than that in radial walls case, while the maximal outlet water temperature difference of the five sectors decreases nearly 20.8% compared to radial walls case, which indicates that with optimized walls, air inflow volume rate differences of the five sectors are decreasing. So compared to radial walls case, the shock of air inflows of windward sectors to inflows of sideward or leeward sectors reduces significantly in optimized walls case. For a specified sector, outlet water temperature of Sector 3 in radial walls case is larger than that in optimized walls case, and so do Sector 4. This is mainly caused by the vortex at the back of the radial wall and the split-flow at the front of the next radial wall, shown in [Fig. 12\(a\)](#). The vortex and split-flow, especially in Sector 3 which has poor cooling performance under crosswind, will weaken the positive effect of the radial wall, and so do Sector 4. For Sector 2, though the vortex and split-flow exist, their adverse effect can be negligible, because Sector 2 is the windward sector which has strong air inflow, and the radial wall is helpful to gather air inflow as well. In addition, the shock from sideward or leeward inflows, which can be regarded as resistance to air flowing into Sector 2, is much smaller than that in optimized walls case. As a result, outlet water temperature of Sector 2 in radial walls case is lower than that in optimized walls case. For windward Sector 1, there is little vortex and split-flow, so the interference on air inflow is slight, and the shock of inflows from other sectors in radial walls case is smaller than that in optimized walls case as well, therefore outlet water temperature of Sector 1 in radial walls case is lower than that in optimized walls case. For aerodynamic field in optimized walls case shown in [Fig.](#)

12(b), a major vortex appears in Sector 5, even though, outlet water temperature of this sector is still smaller than that in radial walls case. With optimized walls, volume rate differences of air flowing into different sectors are decreasing, so the shock of air inflows of other sectors, especially windward sectors, to air inflow of Sector 5 is much smaller than that in radial walls case, which causes outlet water temperature of Sector 5 lower than that in radial walls case.

At crosswind speed of 8m/s, outlet water temperature of windward Sector 1 or 2 in radial walls case is still lower than that in optimized walls case, and is much smaller compared to no wall case. However, cooling performance of Sector 3 in radial walls deteriorates sharply, and outlet water temperature of Sector 3 even increases, compared to no wall case. This is mainly caused by the variation of air inflow ways. Seen in Fig. 10, air inflow deviation angles of cooling deltas in Sector 3 at 8m/s crosswind speed are close to, but smaller than 90°, that is to say, the inflow air of Sector 3 comes from the outside main airflow in no wall case, while it changes in radial walls case. Owing to the radial wall, a great vortex occurs at the inlet of Sector 3, as shown in Fig. 13(a), then the inflow air of Sector 3 comes from this induced vortex. As this vortex is weaker than original outside main airflow in the aspect of intensity, less air flows into Sector 3. In addition, this weak air inflow of Sector 3 can't withstand the shock of strong air inflows from other sectors, especially windward sectors, so some hot air even flows out from Sector 3. All these make cooling performance of Sector 3 with radial wall deteriorated significantly. However, air inflow decrease of Sector 3 gives a chance to Sector 4, and more air flows into

Sector 4, which causes outlet water temperature of Sector 4 in radial walls case lower than that in no wall case. With optimized wall, cooling performance of Sector 3 gets improved conversely, for there is no adverse effect of vortex or split-flow on Sector 3, and air inflow way of Sector 3 doesn't change, seen in [Fig. 13\(b\)](#). In addition, the shock of inflows from other sectors is weaker compared to radial walls case, and coupled with the positive effect of wind-break wall, that is, gathering air, outlet water temperature of Sector 3 in optimized walls case is much lower than that in no wall case. For aerodynamic field no matter in optimized or radial walls case, there is a major vortex in Sector 5, and outlet water temperature of Sector 5 with optimized or radial wall increases, compared to no wall case. Overall, at crosswind speed of 8m/s, the average outlet water temperature of the five sectors with optimized walls decreases nearly 0.83°C than that in no wall case, which is much larger than 0.06°C improved by radial walls. The maximal outlet water temperature difference of the five sectors with optimized walls decreases nearly 17% compared to no wall case, while with radial walls, it increases nearly 21.4% on the contrary. As a consequence, the positive effect on cooling performance of the tower induced by optimized walls is much greater than that brought by radial walls at crosswind speed of 8m/s.

At 12m/s crosswind speed under which flow separation has occurred at the back of the tower, the inflow air of leeward sector comes from the induced secondary flow, and hot air even flows out from sideward sector. In this condition, the optimized wind-break walls are radial walls exactly, which are used as spoilers to enhance the outside secondary flow, shown in [Fig. 14](#). As a result, the overall cooling

performances of sideward or leeward sectors, which relay on this secondary flow, get improved, seen in [Table. 2](#).

[Fig. 15](#) shows the overall effect comparison of radial and optimized walls, on the two aspects of improving cooling performance of the tower and reducing maximal cooling performance difference among the sectors, the optimized walls perform better. At low crosswind speed ($<4\text{m/s}$), the advantage of optimized walls is not obvious, compared to radial walls. At middle crosswind speed ($4\text{-}8\text{m/s}$), the maximal outlet water temperature difference of the five sectors in radial walls case is much larger than that in optimized walls case, and it even increases, compared to no wall case. Moreover, the decrease of outlet water temperature of the tower caused by radial walls is smaller than that induced by optimized walls, so the advantage of optimized walls increases with crosswind speed gradually. At high crosswind speed ($>10\text{m/s}$), the optimized walls, which are radial walls exactly, have a great positive effect on the NDDCTV.

3.3 Generalization of the models and optimized wind-break walls

The present modeling approaches and the introduced optimized wind-break walls apply to all of the natural draft dry cooling tower with vertical radiators. The amount of external optimized wind-break walls is in accord with that of the sectors. At low crosswind speed, each optimized wall is installed with setting angle equal to air inflow deviation angle at corresponding cooling delta's inlet, while at very high crosswind speed, the optimized wall is just installed in the radial direction of the tower. Therefore, the most significant job is to get air inflow deviation angles of all

cooling deltas under any crosswind speed. Firstly, using the numerical model of a specified NDDCT, air inflow deviation angles are exported at several typical crosswind speeds. Then the air inflow deviation angles can be transformed as a general binomial expression of circumferential angle of cooling delta (similar to Eq. (14)), and the polynomial coefficients can be approximately fitted as the functions of crosswind speed (similar to Eq. (15) and (16)). At last, setting angle of each optimized wall can be generated by this binomial expression at any crosswind speed.

4. Conclusion

Considering specific structure of each cooling delta, an efficient numerical model of the NDDCTV is established, and then validated by the design and some published results. To improve cooling performance of the NDDCT, setting angles of external wind-break walls are optimized equal to air inflow deviation angles at low crosswind speeds, while equal to zero when airflow separation occurs at high crosswind speeds. According to air inflow deviation analysis using the present numerical model:

(1) The column with a larger circumferential angle (i.e. Column B) has a better cooling performance than the other for each cooling delta, at windy condition with no airflow separation appearing.

(2) Along half tower circumference, there are two cooling deltas with Column B of best cooling performance, at circumferential angles smaller and larger than 90° respectively, and these two circumferential angles move away from 90° gradually with the increase of crosswind speed.

(3) On the two aspects of improving cooling performance of the tower and reducing maximal cooling performance difference among the sectors, the optimized walls are superior to previous radial walls. Apart from gathering more inflow air at low crosswind speeds, the optimized walls can create stronger secondary flow at very high crosswind speeds when airflow separation occurs.

Nomenclature

A	heat dissipation area (m^2)
c_w	specific heat of water ($\text{J kg}^{-1} \text{K}^{-1}$)
g	gravitational acceleration (m s^{-2})
h	convective heat transfer coefficient of heat exchanger ($\text{W m}^{-2} \text{K}^{-1}$)
l	length of the cooling column (Column A and B have the same length)
k	turbulence kinetic energy ($\text{m}^2 \text{s}^{-2}$)
k_l	non-dimensional pressure loss coefficient
m	mass flow rate (kg s^{-1})
p	pressure (Pa)
qv	volume flow rate of air flowing into cooling column of unit height
Q	heat rejection (W)
S	source term
t	temperature (K)
v	velocity of airflow (m s^{-1})

Greek symbols

α	intersection angle of the two columns of the cooling delta ($^\circ$)
ρ	density (kg m^{-3})
ε	turbulent dissipation rate ($\text{m}^2 \text{s}^{-3}$)
Γ	diffusion coefficient ($\text{kg m}^{-1} \text{s}^{-1}$)
θ	angle ($^\circ$)
ϕ	intensity ratio of secondary flow induced in the triangular zone to main airflow
φ	scalar variable for airflow
Δ	the drop of a variable

Superscripts

<i>e</i>	cell number of a column zone after meshed
<i>sf</i>	secondary flow occurred in the cooling triangular zone
0	the windless condition

Subscripts

<i>a</i>	air
<i>ad</i>	air inflow deviation
<i>cw</i>	crosswind
D	cooling delta
DA	Column A of the cooling delta
DB	Column B of the cooling delta
<i>i</i>	wind-break wall number along tower circumference starting from 0
<i>j</i>	cooling delta number along tower circumference
<i>n</i>	normal direction
S	sector of the tower
<i>w</i>	circulating cooling water
<i>wl</i>	setting angle of wind-break wall
1	inlet
2	outlet

Acknowledgements

The financial support for this research, from the National Science Foundation of China (Grant No. 51176030) and China Power International Development Limited (Grant No. 2013-006-ZDG-KJ-X) is gratefully acknowledged.

Reference

- [1] Hooman K. Dry cooling towers as condensers for geothermal power plants. *International Communications in Heat and Mass Transfer* 2010; **37**(9): 1215-1220.
- [2] Ma H, Si FQ, Kong Y, Zhu KP, Yan WS. A new theoretical method for predicating the part-load performance of natural draft dry cooling towers. *Applied*

Thermal Engineering 2015; **91**: 1106-1115.

- [3] Yang LJ, Wu XP, Du XZ, Yang YP. Dimensional characteristics of wind effects on the performance of indirect dry cooling system with vertically arranged heat exchanger bundles. *International Journal of Heat and Mass Transfer* 2013; **67**: 853-866
- [4] DuPreez AF, Kroger DG. The effect of the heat exchanger arrangement and wind-break walls on the performance of natural draft dry-cooling towers subjected to cross-winds. *Journal of Wind Engineering and Industrial Aerodynamics* 1995; **58**(3): 293-303.
- [5] Al-Waked R., Behnia M. The performance of natural draft dry cooling towers under crosswind: CFD study. *International Journal of Energy Research* 2004; **28**(2): 147-161.
- [6] Al-Waked R, Behnia M. The effect of windbreak walls on the thermal performance of natural draft dry cooling towers. *Heat Transfer Engineering* 2005; **26**(8): 50-62.
- [7] Lu Y, Guan Z, Gurgenci H, Zou Z. Windbreak walls reverse the negative effect of crosswind in short natural draft dry cooling towers into a performance enhancement. *International Journal of Heat and Mass Transfer* 2013; **63**: 162-170.
- [8] Yang LJ, Chen L, Du XZ, Yang YP, Effects of ambient winds on the thermo-flow performances of indirect dry cooling system in a power plant. *International Journal of Thermal Sciences* 2013; **64**: 178–187.

-
- [9] Zhao YB, Long GQ, Sun FZ, Li Y, Zhang CJ, Liu JB. Effect mechanism of air deflectors on the cooling performance of dry cooling tower with vertical delta radiators under crosswind. *Energy Conversion and Management* 2015; **93**: 321-331.
- [10] Wei QD, Zhang BY, Liu KQ, Du XD, Meng XZ. A study of the unfavorable effects of wind on the cooling efficiency of dry cooling-towers. *Journal of Wind Engineering and Industrial Aerodynamics* 1995; **54**: 633-643.
- [11] Su MD, Tang GF, Fu S. Numerical simulation of fluid flow and thermal performance of a dry-cooling tower under cross wind condition. *Journal of Wind Engineering and Industrial Aerodynamics* 1999; **79**(3): 289-306.
- [12] Zhai Z, Fu S. Improving cooling efficiency of dry-cooling towers under cross-wind conditions by using wind-break methods, *Applied Thermal Engineering* 2006; **26**(10):1008-1017.
- [13] Goodarzi M, Keimanesh R. Heat rejection enhancement in natural draft cooling tower using radiator-type windbreakers, *Energy Conversion and Management* 2013; **71**: 120-125.
- [14] Zhao, YB, Long GQ, Sun FZ, Li Y, Zhang CJ. Numerical study on the cooling performance of dry cooling tower with vertical two-pass column radiators under crosswind. *Applied Thermal Engineering* 2015; **75**: 1106-1117.
- [15] Goodarzi M. A proposed stack configuration for dry cooling tower to improve cooling efficiency under crosswind. *Journal of Wind Engineering and Industrial Aerodynamics* 2010; **98**(12): 858-863.

-
- [16]Gebhart B, Jaluria Y, Mahajan RL, Sammakia B, *Buoyancy-induced flows and transport, Textbook Edition*. Hemisphere Publishing Corporation, 1988.
- [17]Ma H, Si FQ, Li L, Yan WS, Zhu KP. Effects of ambient temperature and crosswind on thermo-flow performance of the tower under energy balance of the indirect dry cooling system. *Applied Thermal Engineering* 2015; 78: 90-100.
- [18]Panofsky H, and Dutton J, *Atmospheric Turbulence*. John Wiley & Sons, New York, 1984.

Table caption

Table.1 Simulation and design values under two operation conditions.

Table.2 Effects of wind-break walls on outlet water temperatures of different sectors
under various crosswind speeds.

Figure caption

- Fig.1 Schematic diagrams of air flowing into cooling delta and wind-break walls layout, (a) under low crosswind speeds without flow separation, (b) under high crosswind speeds when flow separation occurs.
- Fig.2 Schematic of the NDDCTV, (a) elevation view with dimension marks, (b) cooling delta distribution and cooling sector division along crosswind direction.
- Fig.3 Schematic of the half computational domain with partial boundary conditions specified.
- Fig.4 Generated grids to solve for the physical domain, (a) grids of the cooling columns and the nearby triangular zones, (b) general view of grids in different zones.
- Fig.5 Velocity vectors of air flowing into cooling deltas of, (a) windward Sector 1, (b) windward Sector 2, (c) sideward Sector 3, (d) leeward Sector 4, (e) leeward Sector 5.
- Fig.6 Variations of outlet water temperatures and θ_{ad} with θ_D .
- Fig.7 Variations of outlet water temperatures of Column A and B with θ_D at various crosswind speeds, (a) $v_{cw} = 0\text{m/s}$, (b) $v_{cw} = 2\text{m/s}$, (c) $v_{cw} = 6\text{m/s}$, (d) $v_{cw} = 8\text{m/s}$, (e) $v_{cw} = 10\text{m/s}$, (f) $v_{cw} = 12\text{m/s}$.
- Fig.8 Velocity vectors of air flowing into, (a) Column A at $\theta_{ad} < \alpha/2$, (b) Column B at $\theta_{ad} < \alpha/2$, (c) Column A and Column B at $\theta_{ad} > \alpha/2$.
- Fig.9 The variation of non-dimensional inflow velocity normal to Column B with

$\theta_D/1^\circ$ at crosswind speed of, (a) 4m/s, (b) 8m/s.

Fig.10 Variations of θ_{ad} with θ_D at various crosswind speeds.

Fig.11 Comparison of results obtained by present paper and previous literatures, (a) dimensionless outlet water temperature variation of cooling delta with θ_D , (b) non-dimensional normal velocities of air flowing into different sectors.

Fig.12 Aerodynamic field at the horizontal cross section of half radiator height under 4m/s crosswind speed, (a) with external wind-break walls of radial direction, (b) with optimized layout of wind-break walls.

Fig.13 Aerodynamic field at the horizontal cross section of half radiator height under 8m/s crosswind speed, (a) with external wind-break walls of radial direction, (b) with optimized layout of wind-break walls.

Fig.14 Aerodynamic field at the horizontal cross section of half radiator height with optimized walls (i.e. radial walls) under 12m/s crosswind speed.

Fig.15 The overall effect comparison of radial and optimized walls on, (a) cooling performance of the tower, (b) maximal cooling performance difference among the sectors.

Table.1 Simulation and design values under two operation conditions

Design operation condition	Condition 1	Condition 2
Atmospheric pressure (kPa)	855.8	854.7
Ambient temperature (°C)	14.5	30.0
Circulating water mass rate (t/h)	69700	69700
Inlet water temperature of the tower (°C)	44.5	64.0
Design outlet water temperature of the tower (°C)	33.4	51.7
Calculated outlet water temperature of the tower (°C)	33.7	51.3
Relative error of heat rejection of the tower (%)	2.70	3.25

Table. 2 Effects of wind-break walls on outlet water temperatures of different sectors under various crosswind speeds

Outlet water temperature of five sectors(°C)	At $v_{cw}=4\text{m/s}$			At $v_{cw}=8\text{m/s}$			At $v_{cw}=12\text{m/s}$		
	No walls	Radial walls	Optimized walls	No walls	Radial walls	Optimized walls	No walls	Radial walls	Optimized walls
No.1	32.54	32.05	32.25	31.48	31.05	31.26	29.91		29.31
No.2	33.39	33.12	33.27	34.61	33.46	34.20	35.51		32.95
No.3	34.82	34.26	34.00	40.85	42.63	39.03	42.78		41.79
No.4	34.09	33.72	33.55	39.51	37.51	36.51	40.35		40.51
No.5	33.44	33.72	33.49	33.14	34.65	34.46	39.67		38.88
Average	33.66	33.37	33.31	35.92	35.86	35.09	37.64		36.69
Maximal difference	2.28	2.21	1.75	9.37	11.38	7.78	12.87		12.48

Figure(s)

Fig.1 Schematic diagrams of air flowing into the cooling delta and wind-break walls layout, (a) under low crosswind speeds without flow separation, (b) under high crosswind speeds when flow separation occurs.

Fig. 2 Schematic of the NDDCTV, (a) elevation view with dimension marks, (b) cooling delta distribution and cooling sector division along crosswind direction.

Fig.3 Schematic of the half computational domain with partial boundary conditions specified.

(a)

(b)

Fig.4 Generated grids to solve for the physical domain, (a) grids of the cooling columns and the nearby triangular zones, (b) general view of grids in different zones.

Fig.5 Velocity vectors of air flowing into cooling deltas of, (a) windward Sector 1, (b) windward Sector 2, (c) sideward Sector 3, (d) leeward Sector 4, (e) leeward Sector 5.

Fig.6 Variations of outlet water temperatures and θ_{ad} with θ_D .

Fig.7 Variations of outlet water temperatures of Column A and B with θ_D at various crosswind speeds, (a) $v_{cw} = 0$ m/s, (b) $v_{cw} = 2$ m/s, (c) $v_{cw} = 6$ m/s, (d) $v_{cw} = 8$ m/s, (e) $v_{cw} = 10$ m/s, (f) $v_{cw} = 12$ m/s.

Fig.8 Velocity vectors of air flowing into, (a) Column A at $\theta_{ad} < \alpha/2$, (b) Column B at $\theta_{ad} < \alpha/2$, (c) Column A and Column B at $\theta_{ad} > \alpha/2$.

(a)

(b)

Fig. 9 The variation of non-dimensional inflow velocity normal to Column B with $\theta_D / 1^\circ$ at crosswind speed of, (a) 4m/s, (b) 8m/s.

Fig.10 Variations of θ_{ad} with θ_D at various crosswind speeds.

Fig.11 Comparison of results obtained by present paper and previous literatures, (a) dimensionless outlet water temperature variation of cooling delta with θ_D , (b) non-dimensional normal velocities of air flowing into different sectors.

Fig. 12 Aerodynamic field at the horizontal cross section of half radiator height under 4m/s crosswind speed, (a) with external wind-break walls of radial direction, (b) with optimized layout of wind-break walls.

Fig. 13 Aerodynamic field at the horizontal cross section of half radiator height under 8m/s crosswind speed, (a) with external wind-break walls of radial direction, (b) with optimized layout of wind-break walls.

Fig. 14 Aerodynamic field at the horizontal cross section of half radiator height with optimized walls (i.e. radial walls) under 12m/s crosswind speed.

(a)

(b)

Fig. 15 The overall effect comparison of radial and optimized walls on, (a) cooling performance of the tower, (b) maximal cooling performance difference among the sectors.