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A Numerical Study on the Improving Techniques of the Cooling Performance of a Natural Draft Dry Cooling Tower under Crosswind

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Crosswind degrades the cooling performance of a natural draft dry cooling tower (NDDCT) by affecting the air flow field at the inlet of the heat exchanger bundle, incurring complex vortexes inside the tower shell, and shifting the flow direction at the outlet of the tower. The pressure distribution change outside the radiators is found to be the main factor to affect the NDDCT's overall cooling performance. An enclosure outside the heat exchanger bundle with an opening at the windward side is proposed to collect the crosswind and increase the pressure level outside the side and back radiator sections, so as to enhance the cooling performance of a NDDCT. As many researchers frequently recommend windbreaks for the same purpose, we combine an enclosure and windbreaks together, and adopted a computational fluid dynamics method to study the cooling performance of a NDDCT at the presence of an enclosure, windbreaks and a combination of the two structures at a wide range of crosswind velocity. Numerical results shows that methods of an enclosure and windbreaks could achieve a similar performance at a crosswind speed less than 16 m/s, and the combination of the two could remain a better performance under all investigated crosswind.

1. Introduction

As one of the most water saving power generation technology, dry cooling technology has developed greatly in the recent years, as regarded as the final solution for the power generation in arid countries and regions. Indirect dry cooling technology is increasingly popular for its characters of no noise, long service life, simple maintenance, energy saving etc.

During the past several decades, much work has been done on the research of natural draft dry cooling tower (NDDCT), which is the primary structure for indirect dry cooling system. In order to reduce the negative effect of crosswind on NDDCT, many ideas were reported in last 70s and 80s, including dry/wet associated cooling, plastic tower shell, periodic dry cooling etc. but further report was rarely seen afterward. Until 1993, wind-breaks was first proposed (Du Preez and Kroger, 1993), and then verified by Al-Waked and Zhai through numerical investigation on different type of NDDCT(Al-Waked and Behnia, 2005) and later in (Al-Waked and Behnia, 2004) with the latest in (Zhai and Fu, 2006). In 2009, Dai reported that guiding channel could promote the cooling performance of a natural draft wet cooling tower (NDWCT) (Dai et al., 2009), and supported by Wang in the same year that guiding channel could improve the ventilation by 5 % to 10 % (Kai, 2009b). Then Zhao (numerical investigation) and Chen (experimental investigation) found that the cross-wall could enhance the cooling efficiency of NDWCT in lower speed of crosswind, but sensitive to wind direction at higher speed (Youliang et al., 2012) and then in (Zhao et al., 2009). In 2010, Goodarzi proposed a new inclined exit configuration, and reported that it could improve the cooling efficiency by 9 % at 10 m/s crosswind (Goodarzi, 2010). Lu (2003) reported that in-tower windbreaks could reverse the negative crosswind effect to be positive in a small NDDCT, but sensitive to the wind attack angles. At the same year, Goodarzi reported that windbreaks constructed as radiators

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could even promote the cooling efficiency. Right in 2014, Goodarzi proposed an elliptical cross section type NDDCT cylinder, and reported that this new geometry could improve the cooling efficiency by 17 % at 10 m/s crosswind speed (Goodarzi and Ramezanpour, 2014).

As the pressure distribution outside the heat exchanger bundle was found to be the main factor to affect the overall cooling performance of a NDDCT under crosswind condition, and an enclosure outside the heat exchanger bundle was found to be effective to increase the pressure level outside the side and back radiator sections to enhance the cooling performance of a NDDCT, this paper presents a comparison among the performances of improving techniques of an enclosure, windbreak, which is frequently recommended by many researchers in previous studies(Goodarzi and Keimanesh, 2013a), and a combination of an enclosure and windbreaks. CFD method was adopted to achieve this work at velocities range from 0 m/s to 20 m/s.

2. Methods

2.1 Problem description

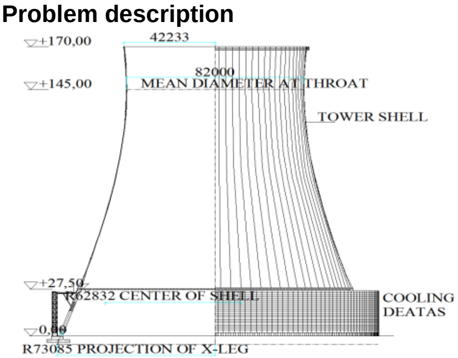

Figure 1: Schematic structure of a NDDCT

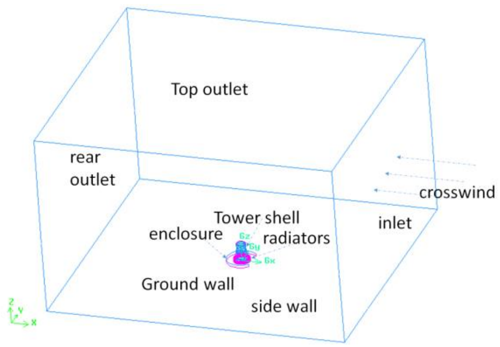

Figure 2: Sketch of the computational field and its boundaries for air flow under crosswind

A sample of a NDDCT overview is shown in Figure 1, which is based on an on service Heller type NDDCT in a 660 MW dry cooling power plant in China. The NDDCT of interest has a total height of 170 m, a tower bottom height of 27.5 m, a base radiator height of 24 m, a radiator support of 2 m, an outlet diameter of 84.466 m, a throat diameter of 82 m, and a base radiator diameter of 146.17 m. 183 deltas and 10 sectors nearly equally distributed at the bottom of the tower, except one part where a flue gas pipe is induced from outside to inside at the up-right part. When crosswind exists, the flow fields around the cooling deltas, inside of the tower, tower outlet etc. are changed, which reduce the ventilation rate, and degrade the cooling effect of the tower.

2.2 Computational field

The computational field in Figure 2 has a dimension of 2,764 m × 2,512 m × 1,700 m, which is large enough compared with the tower size (more than 10 times in directions of x, y and z) to eliminate the unrealistic effect of the domain boundaries on the flow field crosswind outside and inside the tower. The size of the enclosure added to the outside of the cooling deltas is about 220 m in radius and 62 m in height to avoid disturbing the airflow on no wind condition. Hexahedral structured grids were used to generate the computational meshes on a commercial software Gambit. The grid interval size varies from 0.15 m ~ 0.3 m near heat exchanger to about 20 m near the inlet and outlet of the computational field using successive ratio grading scheme.

2.3 Boundary conditions and basic assumptions

In previous works, many researches tend to study the cooling performance of NDDCT by solve the conjugate fluid-solid-fluid heat transfer problems among the air, radiators, and cycling water and fluid dynamics problems in the air side simultaneously (Goodarzi and Ramezanpour, 2014; Al-Waked and Behnia, 2004; Yang et al., 2013a; Yang et al., 2013b; Zhao et al., 2015). Based on the knowledge that the change of water vaporization latent heat is negligible within the common back pressure (decreases less than 1.6 % from 15 kPa to 30 kPa), we assume that the heat release of the condenser is constant, then we get constant heat dissipation from the radiators. We then calculate the ITD based on the energy conservation formula and corresponding boundary conditions (Yang et al. 2013b).

A constant crosswind is assumed at the inlet boundary, and an outflow boundary condition is set at the downstream surface both upward and rearward. Other surfaces like the ground，the inside / outside cooling tower shells and the support and joint faces between adjacent radiators are all set as adiabatic wall conditions with no slip shear condition. The pressure-based solver in Fluent with pressure-velocity coupling SIMPLEC method is used. The governing equations for the momentum, energy, turbulent kinetic energy and dissipation rate are discretized using the second-order upwind differencing scheme. The SIMPLE algorithm is employed in pressure-velocity coupling solution method.

2.4 Governing equations

The flow regime is turbulent, and air density variation in the cooling tower is so small that the flow can be assumed incompressible and Boussinesq approximation can be used in the vertical momentum equation to consider the buoyancy force (Goodarzi and Ramezanpour, 2014). Governing equations for steady, buoyant, and turbulent flow including heat transfer are continuity, momentum, energy, and turbulence modelling equations. The well-known standard $k-\varepsilon$ model has been used to model the turbulent flow.

$$\vec{\nabla}\cdot\vec{V}=0.0 \tag{1}$$

$$\rho(\vec{V}\cdot\vec{\nabla})\vec{V}=-\vec{\nabla}p+\vec{\nabla}\tau-\rho\beta(T-T_{ar})\vec{g}+\vec{S}_h \tag{2}$$

$$\rho(\vec{V}\cdot\vec{\nabla})T=-\vec{\nabla}[(\Gamma+\Gamma_t)\vec{\nabla}T]+Q_h \tag{3}$$

$$(\vec{V}\cdot\vec{\nabla})k=\vec{\nabla}\left[\left(v+v_t/\sigma_k\right)\vec{\nabla}k\right]+P+G-\varepsilon \tag{4}$$

$$(\vec{V}\cdot\vec{\nabla})\varepsilon=\vec{\nabla}\left[\left(v+v_t/\sigma_\varepsilon\right)\vec{\nabla}\varepsilon\right]+C_{1\varepsilon}\varepsilon/k(P+G)-C_{2\varepsilon}\varepsilon^2/k \tag{5}$$

in which

$\tau_{ij}=(\mu+\mu_\tau)S_{ij}$, $P=v_\tau S_{ij}S_{ij}$, $G=-g\beta v_\tau/\sigma_\tau S_{ij}\partial T/\partial Z$, $S_{ij}=0.5\left(\partial V_i/\partial x_j+\partial V_j/\partial x_i\right)$, $v_\tau=u_\tau/\rho=C_\mu k^2/\varepsilon$, $\Gamma=\mu/\Pr$, $\Gamma_t=\mu_t/\Pr_t$

and

$C_\mu=0.09$, $C_{1\varepsilon}=1.44$, $C_{2\varepsilon}=1.92$, $\sigma_k=1.0$, $\sigma_\varepsilon=1.3$, $\sigma_t=1.0$ are constant of standard $k-\varepsilon$ equations. In the above equations $\vec{V}$, P , ρ , μ and μ_t are velocity vector, static pressure, air density, molecular viscosity, and turbulent viscosity. Also, T and T_{ar} are local and reference temperatures, β denotes the air compressibility coefficient, $\vec{g}$ is gravity acceleration vector, and S_{ij} is the tensor of strain rate.

The air-side flow resistances due to the radiators are resolved by solving the flow equations in ANSYS FLUENT, whereas the flow resistance of the radiator is simulated by a porous medium model in the region where the radiator is located, which is merely the addition of a momentum source term to the standard flow Eq(2) in this region (Reuter and Kroger 2011). As the flow regime in the radiator part is laminar (Re < 540), the pressure drop is typically proportionally to velocity and the inertial flow loss in radiator part can be considered to be zero. Ignoring convective acceleration and diffusion, the momentum source term is written as

$$S_i=\nabla p=-\frac{\mu}{a}\vec{v} \tag{6}$$

For an orthotropic fill resistance in the finned-tube radiator cores, the oblique flow entering the fill is forced into the vertical direction by the radiator fins. This change of direction can be modelled by means of the anisotropic porous medium model by making vertical loss coefficients in Eq(6) very large to simulate the impermeability of the fins in that direction (Reuter and Kroger, 2011).

The heat transfer between the radiators and inlet air flow is simplified as the addition of a constant energy source term to the standard energy Eq(3) in this region. In the nominal condition of this paper’s interest,

$$Q_h=258234.7\,W \tag{7}$$

The convective heat transfer coefficient h can be specified as the following form.

$$h = \sum_{n=1}^{3} h_n u^{n-1} \tag{8}$$

where h_n is the polynomial coefficient, calculated as h_1 = 1,451.95, h_2 = 1,156.59, h_3 =-75.615 in terms of the convective heat transfer experimental data through the finned tube bundles(Yang et al., 2013b).

2.5 Validations

The grids consisting of 7,535,281, 8,354,613, 12,268,096 and 13,771,319 cells for a NDDCT were checked at the crosswind speed of 0.5 m/s and 4 m/s are checked. The ventilation rate (mass flow rate) of the NDDCT at the same boundary condition varies slightly as the increase of grid number, by 0.16 % at the most, and only about 0.036 % and 0.016 % between the two grid number of 12,268,096 and 13,371,319. We finally chose mesh with the grid number of 13,771,319 for its better assessment and convergence. In order to further verify the CFD model, we chose two experimental data as the reference cases, and did the simulation on the same conditions. The boundary conditions and calculating results are shown in Table 1. From the Table 1 we can see that the calculated mass flow rates are about 0.8 ~ 0.9 % bigger than the measured data. Considering the assumptions made in CFD modelling, this tolerance well confirms the validity of the CFD numerical model.

Table 1: Validation of the CFD model with experimental data

	$Q_{dissipated}$ (MW)	v_a (m/s)	T_a (℃)	P_a (kPa)	q_{ae} (kg/s)	q_{as} (kg/s)	$error_q$ (%)
Case1	827.7	1.97	27.36	94.02	36209.2	36498.2	0.79
Case2	773.16	3.47	25.22	93.87	37553.7	37902.3	0.9

3. Result and discussions

3.1 Circumferential ventilation properties

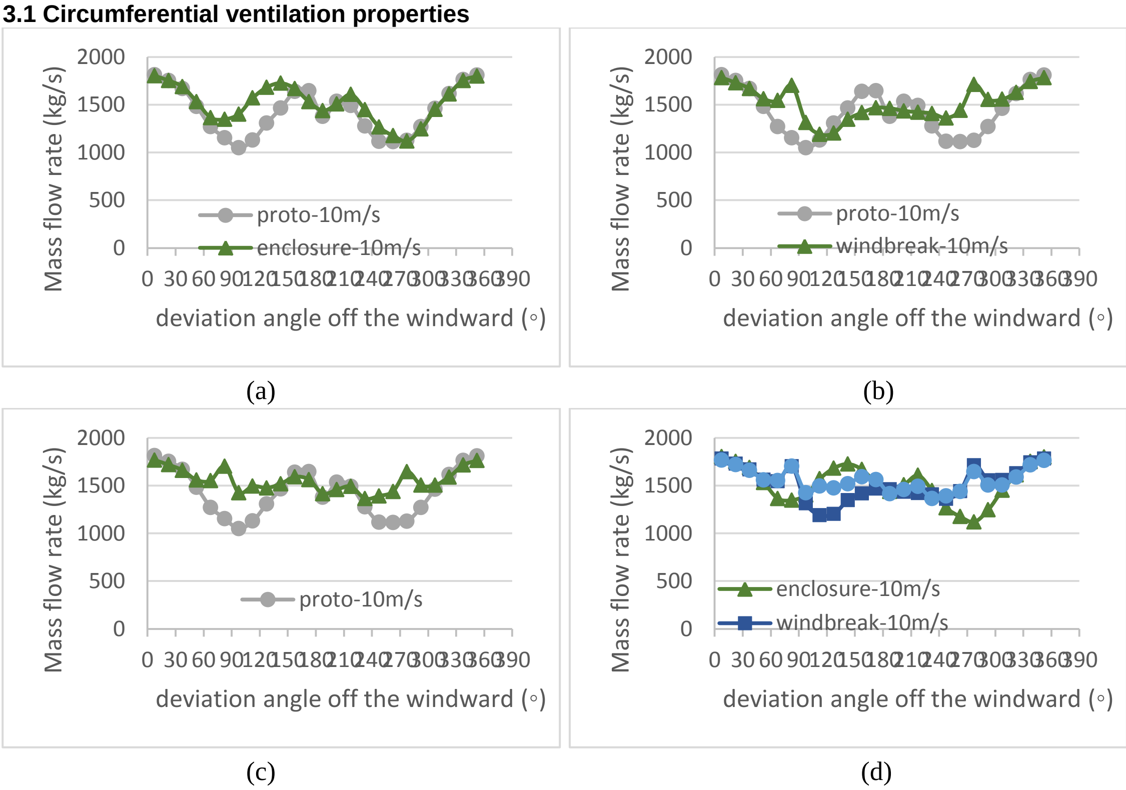

Figure 3: Comparisons of NDDCTs' circumferential mass flow rate distribution at 10m/s: (a) between prototype and with an enclosure, (b) between prototype and with windbreaks, (c) between prototype and with a combination of an enclosure and windbreaks, (d) among prototype, with an enclosure, with windbreaks, and with a combination of an enclosure and windbreaks.

As the improving techniques of a NDDCT investigated in this study are mainly focused on increasing the overall back and side pressure outside the heat exchanger bundle, and 10 m/s is a typical crosswind velocity chosen by many researchers to study the cooling performance of a NDDCT, the circumferential ventilation properties of a NDDCT at crosswind velocity of 10m/s on different conditions were investigated. The comparisons of circumferential mass flow rate distributions on different conditions are illustrated in Figure 3, in which Figure 3(a) ~ (c) show the comparisons of circumferential mass flow rate between a prototype NDDCT and a NDDCT with an enclosure, windbreaks, and a combination of an enclosure and windbreaks, Figure 3(d) shows the comparison of circumferential mass flow rate among a NDDCT with the above three different improving techniques.

From Figure 3 (a) we can find that, compared to the prototype NDDCT, there is an obvious mass flow rate increase of a NDDCT with an enclosure in radiator sections at angle range from around 45° ~ 315°, in which, the increases in radiator sections at angle range from 90°~135° and at angle range from 225° ~ 270° are extremely bigger. That means an enclosure outside the heat exchanger bundle with an opening at the windward side is effective in increasing the general pressure outside the radiators at side and back. Figure 3(b) illustrates an obvious mass flow rate increase of a NDDCT with windbreaks at side radiator sections, however a mass flow rate decrease is also found. The radiator sections with increased mass flow rate mainly locate at the side parts, in which the increases in radiator sections at angle range from 60° ~ 120° and at angle from 240° ~ 300° contribute the most. The decrease area locates mainly at the back. These mean windbreaks are effective in increasing the side pressure outside the heat exchanger bundle, while bad for the pressure distribution outside the back radiator sections. It can be found that in Figure 3(c) with the combination of an enclosure and windbreaks, the mass flow rate of radiator sections at a wilde range of angle is greatly increased, showing the positive features of both structures. Figure 3(d) also shows that, a NDDCT with the combination of an enclosure and windbreaks has the most even distribution of mass flow rate compared to the other two cases.

3.2 The cooling performance of a NDDCT in different cases

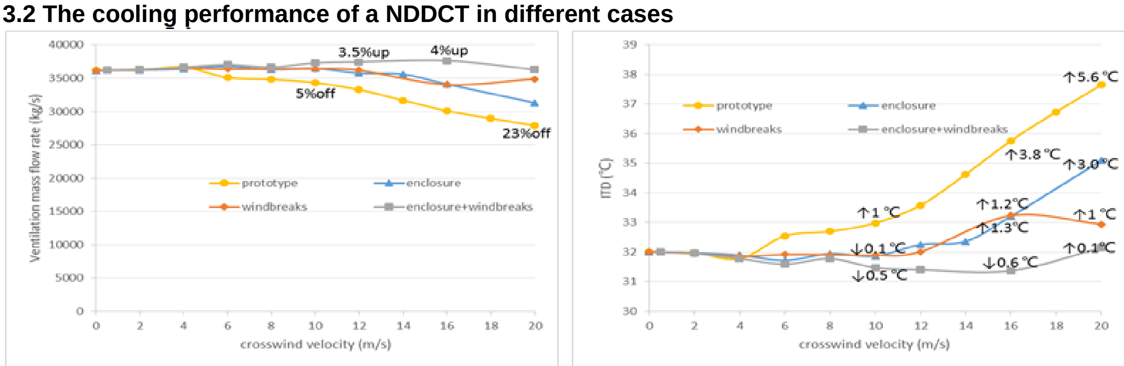

Figure 4: Comparison of the cooling performance at velocity range from 0 m/s to 20 m/s among NDDCTs of prototype, with an enclosure, with windbreaks, and with a combination of an enclosure and windbreaks: (a) the ventilation mass flow rates, (b) the ITD

The cooling performances of a NDDCT on conditions of prototype, with an enclosure, with windbreaks and with a combination of an enclosure and windbreaks were calculated under crosswind velocities range from 0 m/s to 20 m/s, and showed in Figure 4. Figure 4 (a) tells us the change of the overall air mass flow rate with velocity in the above four different cases, and Figure 4(b) illustrates the change of ITD with velocity correspondently. We can find that the ventilation capacity of a prototype NDDCT decreases gradually as the increase of wind velocity at a speed above 4 m/s, and reaches its lowest level of 77 % of the nominal capacity at the highest wind velocity of 20 m/s, resulting in about 5.6 °C temperature increase of ITD. The cooling performance of a NDDCT with an enclosure scarcely change under crosswind at velocity less than 14 m/s, and experiences its performance deterioration at wind velocity above 16m/s. Its lowest mass flow rate also appears at the highest crosswind velocity at 20 m/s, with 12.2 % performance enhancement compared with the prototype NDDCT. A NDDCT with windbreaks shows a similar performance to that with an enclosure at a velocity below 16 m/s. The combination of an enclosure and windbreaks enhance the cooling performance of a NDDCT by 3 ~ 4 % in ventilation rate and 0.5 ~ 0.6 % in ITD under a wide range of the crosswind investigated. Generally the improving techniques of an enclosure and windbreaks have almost the same performance in the enhancement of a NDDCT's cooling performance; a combination of the above two doubles the improving effect and reverses negative effect of cross wind to be enhancement.

4. Conclusions

The cooling performances of a NDDCT at the presence of an enclosure, windbreaks and a combination of the two structures are investigated at a crosswind velocity range from 0 m/s~20 m/s based on an on service indirect dry cooling system of a 660 MW power plant. Numerical results show that an enclosure outside the heat exchanger bundle with an opening at the windward side could increase the pressure level outside the radiator sections at two sides and the back, and then eliminate the deterioration of a NDDCT's cooling performance at a crosswind velocity below 16m/s, showing a similar effect to widely recommended windbreaks, which increase the pressure level outside the radiators at two sides (near the windward). A combination of the two methods could unite their both advantages, and achieve a better performance to reverse the negative effect of crosswind to be enhancement.

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