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CFD simulations on small natural draft dry cooling towers

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Abstract

The natural draft dry cooling tower (NDDCT) is believed to be the only cost-effective option for cooling system of geothermal power plants proposed by Queensland Geothermal Energy Centre of Excellence (QGECE). By reviewing literatures related to the design of NDDCT, the effect of crosswind on the cooling performance of NDDCT is not considered in contemporary design theories for cooling tower system. Practical operations on NDDCT showed that the crosswind larger than 1 m/s will have a profound effect on the performance and that cannot be neglected, especially for small size NDDCT.

A study on the performance of a 15m-high natural draft dry cooling tower under different crosswind conditions is presented in this paper. CFD models (both 2D and 3D) based on porous media have been established for numerical simulations of the air dynamics and heat transfer inside and outside the cooling tower. The CFD results have been validated by comparing the results under no crosswind condition with those obtained in theoretical calculations (1D model). 3D CFD simulations showed that the total cooling capacity of the NDDCT is unfavourably affected by cross wind, and under certain conditions the heat rejection in the cooling tower can be reduced significantly. This trend matches the results of similar studies on large NDDCT found in open literatures well.

Key words: crosswind, natural draft cooling tower, windbreak wall, CFD modelling, numerical simulation

Introduction

Geothermal energy is one of the renewable energies which can supply the base-load electricity with no carbon emissions [4]. Queensland Geothermal Energy Centre of Excellence (QGECE) is researching and developing Enhanced Geothermal System (EGS) power plant in the arid inner land of Australia, where a natural draft dry cooling tower (NDDCT) is the best choice.

In a NDDCT, the air is heated by the heat exchangers which are arranged horizontally at the tower inlet so its density is less than that of the air outside the tower. Consequently, heated air is lift by buoyancy force so that the pressure difference between tower inside and outside occurs which causes continuous air flow passing through the heat exchanger. The air flow is stabilized when two balances are satisfied in the tower: the aerodynamic balance that the sum of all flow resistances should equal to total draft force and the energy balance that heats transferred by water, air and heat exchangers are all same [9,11]. This ideal fundamental of natural draft dry cooling tower is the basis for tower design and sizing, however it does not consider any external influence in real operations such as, the most important one, crosswind.

It has been already proved in both commercial operations and academic researches that the cooling performance of NDDCT is affected by ambient crosswind to certain extents. Systematic studies of the effect of crosswind on NDDCTs with the size larger than 80m in tower height have been carried out by many researchers. The methods used in various studies can be catalogued into two groups: field measurement or laboratory test [3, 5, 13] and numerical analysis [1, 2, 7, 12]. Most of them used the difference of approach temperature (Eq1) as the assessment of wind influence [6], and the similar conclusion was made that approach temperature decreases over 10^{o}C [5] at crosswind velocities of more than 10m/s, which will cause significant loss in power generation.

$$\Delta T_{appr} = \left(T_{wo} - T_{ai}\right) - \left(T_{wo} - T_{ai}\right)_{cw} \qquad (1)$$

Particularly in recent two decades, numerical analysis is much preferred due to the improvement in computation capability of computers. In these numerical studies, both the 2D and 3D CFD model of full scale of large size natural draft dry cooling towers were established and simulated in all kinds of commercial CFD codes. The numerical results generally matched the experimental data well.

For small size NDDCT (less than 30m in height) it is expected that the effect of crosswind will be more significant than that in large NDDCTs. Unfortunately as very few open publications are concerned about this question, to what extent this effect will be is still quite unclear.

QGECE is proposing a small EGS geothermal power plant with the net capacity of 100kW. The total energy transfer efficient is estimated as 15%, so that total around 578kW of heat needs to be dumped. A natural draft cooling tower with horizontally arranged heat exchangers is planned for the cooling system of this plant. The tower size required is 15m in height and 12m in base diameter according to theoretical analysis [10] under the design condition that ambient air and water inlet temperature are 20^{o}C and 40^{o}C respectively in preliminary study. In order to examine the effectiveness of that tower under windy conditions, a numerical study has been carried out, which is presented in this paper.

CFD model

The full scale 3D small NDDCT model is assumed as a cylinder with dimension of 15m in height and 12m in base diameter, while the tower support structure is simplified as a cylinder face with certain pressure resistances. In preliminary CFD studies, it's found that numerical results are influenced if the size of the computational domain is small, therefore the cylindrical domain with height 6 times and diameter 10 times than the corresponding dimension of tower is used to acquire the acceptable accuracy of the simulation results [8]. Figure 1 shows the model dimension and the related boundary conditions.

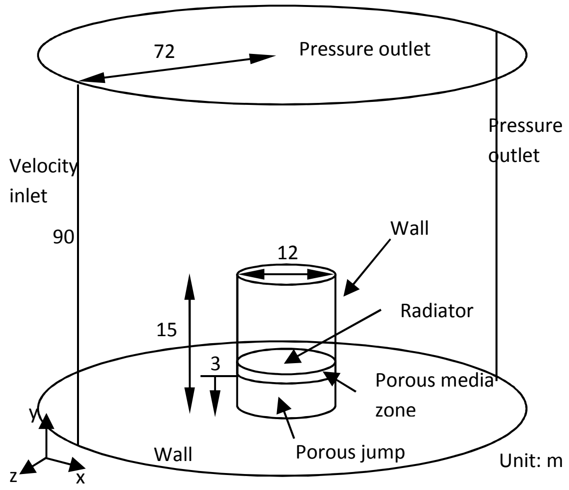

Figure 1. Geometry of 3D model

The heat exchanger bundles are modelled by radiator with porous media boundary condition (as Figure 1). The former simulates the heat transfer at the rate expressed as Eq2 and the latter represents the pressure drop in heat exchangers which is calculated by Eq3.

$$q = h(T_r - T_o) \qquad (2)$$

Where *h*, the convective heat transfer coefficient is a function of air velocity and heat exchanger parameters.

$$F_i = -\left(\frac{\mu_e}{\alpha} v_i + C\frac{1}{2}\rho v_i^2\right) \qquad (3)$$

Where α and *C* are determined by the pressure correlation of heat exchangers. So that both the heat transfer rate (heat flux *q*) and volumetric pressure loss can be related to the air flow velocity and the correlation factor in these equations are determined in theoretical analysis. Subscript *i* in Eq3 denotes any direction of Cartesian coordinate. Very large values of α and *C* are set for two horizontal directions to prevent air flow in these directions, leaving only vertical air flow in porous media zone.

The CFD calculation uses the incompressible air model associated with Boussinesq's approximation to reflect the air density difference caused by heating. The discretization scheme is second order of upwind scheme and the segregate algorithm is set to pressure-based SIMPLE. The air flow turbulence model is assumed as realized k-ε model. The model is simulated by solving a serial of conservation equations of physical quantities, whose general term is expressed as equilibrium of convective term with the sum of diffusive term and source term:

$$div(\rho\vec{v}\phi) = div(\Gamma_\phi grad\phi) + S_\phi \qquad (4)$$

The expressions of ϕ, Γ_ϕ and S_ϕ are shown in table below.

Expressions for governing equation (Eq4)

Equations	ϕ	Γ_ϕ	S_ϕ
continuity	1	0	0
x momentum	u	μ_e	$-\frac{\partial p}{\partial x} + \nabla\cdot\left(\mu_e \frac{\partial}{\partial x}\cdot\vec{v}\right) + F_x$
y momentum	v	μ_e	$-\frac{\partial p}{\partial y} + \nabla\cdot\left(\mu_e \frac{\partial}{\partial y}\cdot\vec{v}\right) - \rho_0\beta(T - T_0)g + F_y$
z momentum	w	μ_e	$-\frac{\partial p}{\partial z} + \nabla\cdot\left(\mu_e \frac{\partial}{\partial z}\cdot\vec{v}\right) + F_z$
energy	T	$\frac{K_e}{C_p}$	$\frac{1}{C_p}\left(\frac{qA_p}{V_p}\right)$
k	k	$\mu + \frac{\mu_t}{\sigma_k}$	$G_k + G_b - \rho\varepsilon$
ε	ε	$\mu + \frac{\mu_t}{\sigma_\varepsilon}$	$\rho C_1 S\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{v\varepsilon}} + C_{1\varepsilon}C_{3\varepsilon}G_b\frac{\varepsilon}{k}$

Where

$\mu_e = \mu + \mu_t$, $\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$,

$C_{1\varepsilon} = 1.44$, $C_2 = 1.92$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.44$, $\Pr = 0.74$, $\Pr_t = 0.85$, $T_0 = 293.15$

Grid-independent test for the model has been done and found that calculation results don't improve further when total cells are more than 3,500,000. The calculation is iterated for more than 15,000 steps and converged when the scaled residuals for all variables (except energy) drop to the order of 10^{-5} and the monitored variables remains constant.

CFD results and discussions

Simulation runs at the condition that the crosswind speed varies from 0m/s to maximum 18m/s at the reference elevation of 10m, which obeys the power law profile:

$$\frac{v_{cw}}{v_{ref}} = \left(\frac{y}{y_{ref}}\right)^a \qquad (5)$$

When there is no crosswind, the temperature contour and velocity vector field are shown in Figures 2 and 3. It is seen that both temperature and velocity distributions displays a symmetric pattern. The CFD results have been validated by the comparison with the analytical ones. Both the results match quite well and the relative error in total heat transferred Q_r at the radiator is about 0.03%.

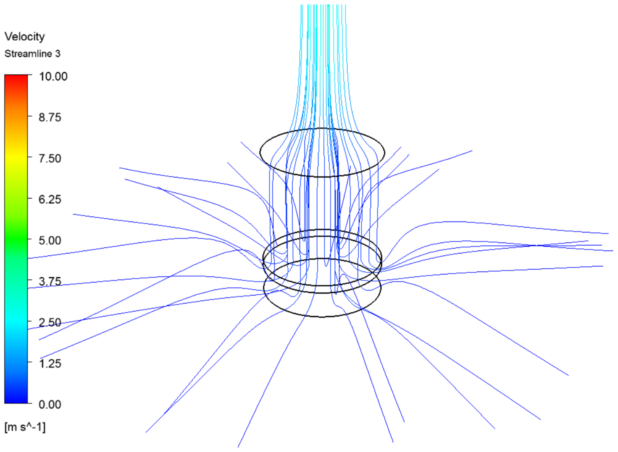

Figure 2. 3D streamlines at no crosswind condition

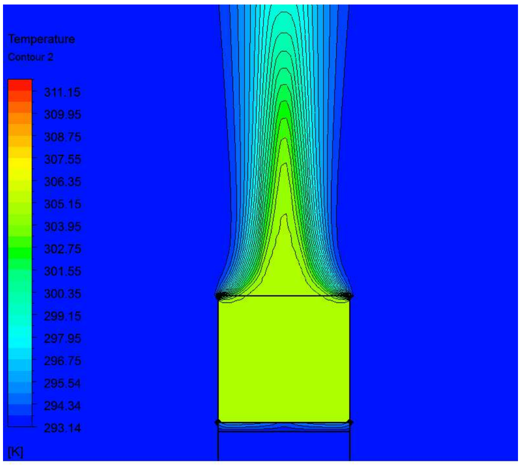

Figure 3. Temperature contour at central vertical cross section at no crosswind condition

With the existence of crosswind, the airflow inside the cooling tower is influenced as expected. Figures 4 and 5 show the airflow 3D streamlines inside and under the cooling tower and the air temperature contour at the central vertical cross section of tower at various crosswinds respectively.

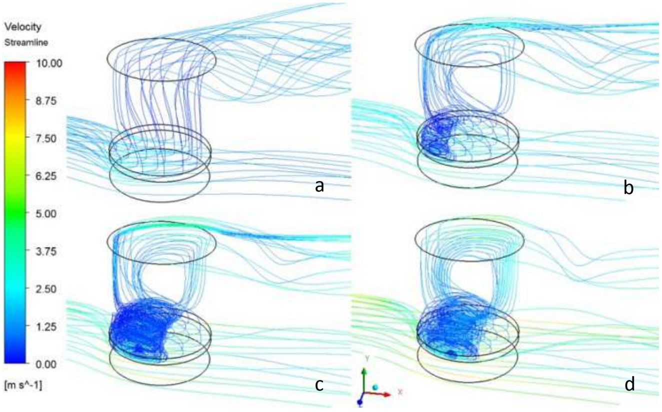

Figure 4. 3D streamlines inside and under cooling tower when crosswind speed is (a) 2m/s, (b) 4m/s, (c) 6m/s and (d) 8m/s.

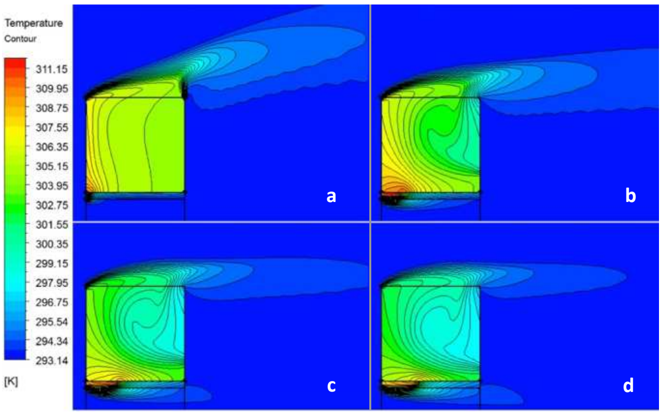

Figure 5. The temperature contour at the central vertical plane when wind speed is (a) 2m/s, (b) 4m/s, (c) 6m/s and (d) 8m/s.

The vertices have been seen inside the cooling tower at high speed crosswind as the fig shows. The vertex at tower upper part, which can be referred to as the 'cold inflow' usually assessed by Froude number [9], occurs because upward-flowing hot airstream is slower than outside wind so that it cannot break through the 'wind lid', but gets cooled immediately near the tower mouth and, thus, some of air sinks back into the cooling tower. While at the tower bottom, hot air at the windward side is sucked down because a negative pressure zone occurs underneath the heat exchangers when crosswind passes by at a high speed, and re-enters into the heat exchanger bundles at leeward side. This process repeats many times for some part of air forming a hot air circulation, which makes heat transfer in this region rather complicated. Further analysis found that the suction effect under the heat exchangers is the dominant factor that causes the vertices.

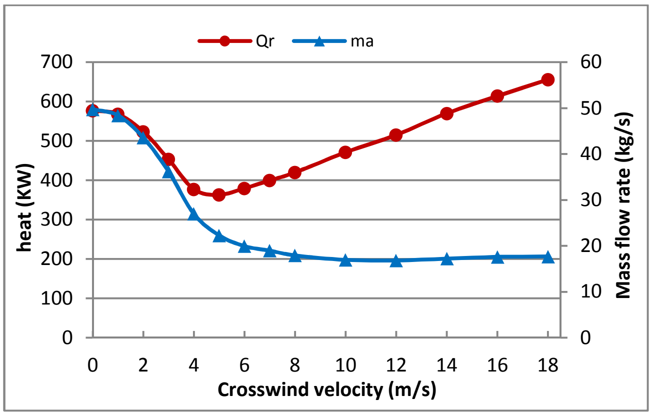

Figure 6. The effects of different crosswind on small NDDCT

Quantitatively, the net air mass flow rate m_a as well as Q_r at the radiator are monitored. Here net air mass flow rate m_a accounts for the net value of it at the radiator face, which equals the upward mass flow rate minus the downward one in the case when inverse air flow occurs at heat exchangers. Figure 6 shows the different m_a, Q_r under different crosswind conditions. The air mass flow rate m_a decreases quickly at crosswind speeds 1-5 m/s and then shows a constant flow rate when crosswind is faster than 10m/s. And the heat dissipated Q_r declines along with the rise of crosswind speed at first and reaches its lowest point at crosswind speed around 5m/s, then it increases with the increase of the crosswind speed.

The unexpected rally of Q_r indicates the heat transfer of heat exchanger bundles have been improved under high-speed crosswind conditions. In fact downward air flow exists at heat exchanger face, as a result a part of radiator heat is diffused in the passing-through air flow underneath the heat exchangers. Hence, the total heat Q_r actually is the sum of heat transferred by the air flow through the heat exchanger caused by the draft force of the tower and the air flow under the heat exchanger caused by the crosswind. In low or no-crosswind cases, heat dissipation through the tower inlet is negligible, but when wind speed is 10m/s or more, that heat dissipation becomes so significant that accounts for larger part of total heat transferred. This phenomenon is seldom seen in large NDDCTs because large tower provides relatively large draft force for hot air, so that normal crosswind cannot cause inverse flow at heat exchangers.

Conclusions

The crosswind effect on cooling performance of small size NDDCT is examined in this paper. The heat exchanger bundles arranged horizontally at tower bottom are simulated with a combined model of radiator and porous media zone. Simulations under different crosswind speed indicate that the heat transfer in cooling tower has been affected by crosswind significantly:

1. the air flow filed inside tower is disturbed by horizontally-flowing crosswind forming two major vortices and inverse air flow. And further investigation shows the main reason of this is suction effect underneath the heat exchanger in tower bottom.

2. when there exists inverse air flow, the total heat transfer between heat exchanger and air *Qr* is not uni-directional, instead it can dissipate through tower outlet and tower inlet at the same time. So it is more important to examine *Qr* in this case.

3. Along with the increase of crosswind speed, the total heat transfer decreases first, then turns to rise and finally exceeds its original value at no-crosswind case.

4. total transferred heat *Qr* could decrease by 37% compared with no-crosswind condition at crosswind speed of 5m/s, which leads a significant drop in net power generation under this cross wind condition.

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