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Natural Draft Dry Cooling Tower Modelling

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Abstract

Predictions based on scale analysis of a natural draft dry cooling tower (NDDCT) are compared with those obtained numerically and experimentally. Experiments were conducted in a lab-scale model built for our NDDCT where CFD-ACE commercially available software was used to simulate the flow in and around the heat exchangers modelled as a porous medium. Both vertical and horizontal arrangements of the finned-tube bundles are examined. The three independent approaches lead to very close predictions for the air velocity and temperature at the exit from the cooling tower. The results of this study will be useful for future work on the development of air-cooled condensers for geothermal power plants in Australia.

Nomenclature

A	cross section area
B	constant
C_F	form drag coefficient
c_p	specific heat
d	diameter
D	tower base diameter
f	friction factor
g	gravitational acceleration
H	tower height
K	permeability
L	heat exchanger height
P	pressure
Q	generated heat
Re	Reynolds number
t	heat exchanger bundle thickness
T	temperature
U	average velocity
Greek Symbols	
ρ	density
β	thermal expansion coefficient
κ	constant
φ	porosity
μ	viscosity
Ω	dimensionless group
Subscripts	
f	fluid

Introduction

Geothermal power plants are the major candidates of next generation renewable and emission-free power generation systems in Australia. This technology is not currently at an advanced stage of commercialization in Australia compared to other renewable counterparts such as wind or solar. However, industry and renewable energy associations, together with government agencies, are now including geothermal energy within their calculations of available generation capacity by 2020 [1]. Australia's renewable energy sector is still in its infancy when compared to most of the other developed countries. Australia's current policy targets an annual renewable energy generation of 45,000 GWh by 2020. Geothermal energy systems have the potential to produce a base load generation capacity capable of replacing existing coal-fired plants. However, there are some technical challenges to be overcome first. One of the major technical difficulties is the cooling system. Although wet cooling is more efficient than dry cooling [2], water shortages and harsh environmental conditions in areas such as the Australian desert have forced designers to consider less efficient and more expensive air-cooled systems, or dry-cooling as it is often termed. Air-cooled plants offer potential economic advantages due to plant sitting flexibility. Both natural draft and mechanical draft dry-cooling towers, equipped with air-cooled heat exchangers (extended airside surface area), are used. Air cooling can be done by using fans (mechanical driven) or by using natural draft through a cooling tower. The fan-driven systems can be built quickly and at relatively low cost but their operating costs are higher due to their higher maintenance requirements and the parasitic losses associated with running the fans [3]. The cooling system is a significant cost item in the power plant and affects the performance of the entire power cycle. If the cooling system does not provide adequate cooling, the overall plant efficiency decreases with serious economic consequences (e.g. decreased electricity production), i.e. a cost-performance trade-off exists. For instance, the selection of a waste heat rejection system for steam-electric power plants involves a trade-off among environmental, energy and water conservation, and economic factors, while achieving the required cooling rate. It has been reported that approximately 0.3 GWh per year of electrical generation in the United States has been lost because of cooling towers operating below their design efficiency. This corresponds to an economic penalty of around 20 million US dollars per year [4]. It is therefore, very important to design and analyse highly efficient dry cooling systems for power plants. Hence, in order to improve the performance of cooling systems, numerical or theoretical investigations have been reported by many researchers (e.g. [5-8]).

On the other hand, few experimental investigations have been reported for the fan-assisted NDDCTs (e.g. [9]). Partly because of huge size of the cooling tower, it is very difficult to conduct on-site experiments to collect data.

The Queensland Geothermal Energy Centre of Excellence has tried to develop an efficient cooling system including a new type of heat exchanger [10]. Experiments will be run in a laboratory scale cooling tower built by QGECE with the main goal of improving the cooling system for a geothermal power plant. However, what is yet missing in the literature is a thorough knowledge about the scaling of cooling towers, e.g. one does not know the functional relation between the height of the cooling tower and the heat that can be dumped by different heat exchangers. In our case, one does not know how the 2m height model in our lab relates to a real prototype which can be a couple of 100 meters height.

Theoretical Analysis

In a NDDCT the driving force is the air density difference [11-13] that, following the use of Boussinesq approximation, leads to the following pressure difference

$$\Delta p = \rho g H \beta \Delta T \tag{1}$$

Porous medium modelling of the heat exchanger leads to the following pressure drop for flow of air across a finned-tube bundle, of thickness *t*, reads

$$\Delta p = t\left(\frac{\mu U}{K} + \frac{C_F \rho U^2}{\sqrt{K}}\right) \tag{2}$$

As shown by [14] for cases when $C_F U\sqrt{K} > O(10^{-5})$ the form drag is the dominant term. Besides, according to [12] both the form drag coefficient and the permeability change with the internal flow structure as well as the porosity of the porous medium; however, the average value of the form drag reported there is $O(0.1)$ while that of the permeability is $O(10^{-5})$ for a commercial finned-tube bundle. Hence, the criterion for a form drag dominant flow through the bundle is $U > O(10^{-2})$.

The tower frictional pressure drop, for fluid velocity U_f, is the sum of distributed and local (changes in cross-sectional area, recirculation, and other imperfections) losses

$$\Delta p = 0.5\rho U_f^2\left(f\frac{4H}{D_h} + \kappa\right) \tag{3}$$

where κ, as given by Table 1.1 of [15], puts on higher values than those given by $f \cong 0.08Re^{-0.25}$. The tower height and hydraulic diameter are, in most of the practical designs, comparable so that one can simply neglect the distributed losses. Then, the pressure drop through the tower scales with local losses

$$\Delta p \sim 0.5\kappa\rho U_f^2 \tag{4}$$

The dimensionless total pressure drop should scale with

$$\frac{\Delta p}{0.5\kappa\rho U_f^2} \sim 1, \Omega \tag{5}$$

The two pressure drop terms can be comparable when the dimensionless group $\Omega = \frac{2tC_F\varphi^2}{\kappa\sqrt{K}} \sim O(1)$. For very high/low values of Ω, the tower/bundle pressure drop is the dominant one. For a specific problem of finned-tube bundle considered in [12], $\Omega \sim O(10)$ thus the heat exchanger pressure drop becomes the dominant one. However, for the experiments conducted in a tower without heat exchangers, as will be shown in the forthcoming discussion, the pressure drop scales with the tower frictional losses.

It is easy to show that the heat transferred to the fluid flowing through the porous medium increases the enthalpy of the fluid, i.e. $Q = \rho A U c_p \Delta T$, to get the volume-averaged velocity as

$$U = \frac{Q}{\rho A c_p \Delta T} \tag{6}$$

Equation (6) is general enough to cover all heat exchanger configurations but for comparison purpose, horizontal and vertical bundle arrangements are further examined in this paper. For the case of vertical arrangement, the cross-sectional area *A* in equation (6) is given by

$$A = \pi D L \tag{7}$$

where for a horizontal tube arrangement, to a good approximation, the area is given by

$$A = \frac{\pi}{4} D^2 \quad (8)$$

with D and L being the tower base diameter and heat exchanger height, respectively. Equations (5-6) should be combined with (7-8) to give the fluid velocity (and thus the mass flow rate) depending on the tower and heat exchanger design.

Numerical Analysis

Assuming no wind conditions, the cooling tower is modelled as an axisymmetric body to reduce the computational time and cost.

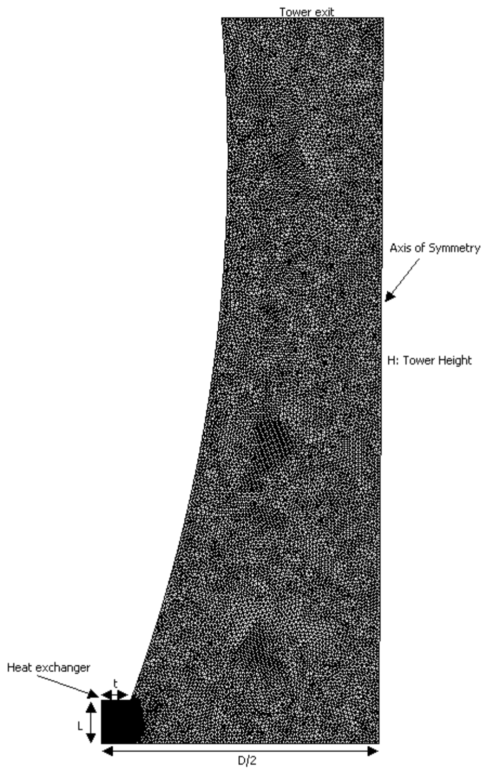

Figure 1. Generated grids; the tower and the heat exchangers around the tower base for the case of vertical heat exchangers.

The computational model is chosen to be bigger than the physical counterpart, as illustrated by figure 1, to eliminate the entrance and exit effects. The governing equations can be written in the form of a generic equation and Table 1

$$\frac{\partial(u\phi)}{\partial x} + \frac{\partial(\mathrm{v}\phi)}{\partial y} = \frac{\partial}{\partial x}(\Gamma_\phi \frac{\partial \phi}{\partial x}) + \frac{\partial}{\partial y}(\Gamma_\phi \frac{\partial \phi}{\partial y}) + S_\phi \quad (9)$$

Commercially available software CFD-ACE (ESI Software) is used to solve full set of turbulent governing equations. The computational domain was generated with triangular grids for this 2-D geometry using the commercial package CFD-GEOM (ESI Software) that is typically used in conjunction with the commercially available finite volume flow solver CFD-ACE. Grids were controlled in CFD-GEOM using curvature criterion, transition factor, and maximum and minimum cell sizes. These values were 30 degrees, 1.1, 0.0025, and 0.00003, respectively. The results were found to be accurate when the total number of nodes is 29579. Grid-independence was tested by control runs on a finer grid with 41580 nodes that produced consistent results (with a maximum error being less than 3%). Hence, finer grids were not used in reporting the results. It should be noted that the convergence criterion (maximum relative error in the values of the dependent variables between two successive iterations) in all runs was set at 10^{-5}.

The heat exchanger is modelled as a heat generating porous medium similar to [12] dumping 283MWth to produce 50MWe [16]. The total heat is divided by the volume occupied by the heat exchanger for a tower of 200m height.

Experimental Analysis

Figure 2 shows the small scale cooling tower and heating element that are used in this study. The cooling tower consists of the tower shell that is made of polycarbonate, the tower support, and the electric heating element. Because of the manufacturing difficulties the cooling tower is made into a pyramidal-square shape.

Figure 2 Experimental setups

The cooling tower shell is supported by four bars 30 cm high. The base area is 2 m^2 with the tower exit area being 0.84 m^2. Four copper heating bars, 10 mm diameter, are arranged horizontally in parallel to form the heating element. Static temperature measurements are introduced as a shortcut to predict the velocity which is very hard to measure directly.

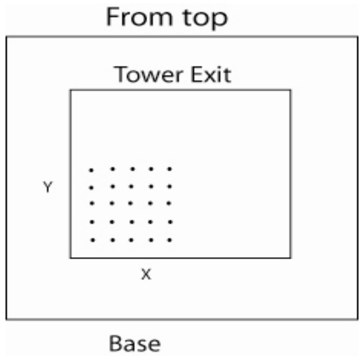

Figure 3 Thermocouple locations to measure the tower exit temperature

The power consumed by the heating element is measured by using "Nanovip power meter". During the experiment, the room temperature is measured using a "tech" thermometer (NO.CN-306, K-type thermocouples). The surface temperature on the copper tube is measured by a k-type thermocouple and is controlled by an in-house manufactured temperature control box. The room temperature and temperatures at the tower inlet and outlet are measured using four "Go! Temp" temperature probes (see figure 3) so that the tower exit velocity is estimated following the use of equation (1).

Results and Discussions

Horizontal bundle

Temperature measurements are conducted using a small scale cooling tower at QGECE labs for an empty tower (with no heat exchanger resistance on top of the heating elements). For this case, the tower pressure drop is the main factor so that the tower exit velocity, predicted by scale analysis, reads

$$U = B\sqrt{\frac{2g\beta\Delta TH}{\kappa}} \quad (10)$$

The result from the temperature measurements involves the uncertainty of measurements which is estimated by using a basic uncertainty analysis [17]. In the scale analysis, the temperature difference of 1.8^{o}C was used (from experiments) with H=2m, κ=0.08 [15] and B = 0.3 to observe that the exit velocity is close to 0.515 m/s in close agreements with the measurements (5% error for 0.543±0.064 m/s) and CFD (14% error for 0.465 m/s).

Vertical bundle

Details of the local velocity and temperature distribution inside the tower are skipped over for the sake of brevity and as a sample of our results table 2 is presented to shows a comparison between numerical and theoretical predictions of the air velocity through a vertical bundle for a fixed porosity, permeability, and form drag coefficient characterized by a typical air-cooled heat exchanger [12].

Table 2. A comparison between theoretical and numerical results

Scale Analysis (m/s)	CFD (m/s)	H (m)	Error
2.48	2.54	62.5	2.3%
2.27	2.22	50	2.25%
2.05	2.08	37.5	1.44%
1.78	1.77	25	0.5%

As seen, numerical results are in close agreement to those predicted by scale analysis and the error is slightly increasing with the tower height (the maximum error is still less than 3% which is quite reasonable).

Conclusions

Theoretical results to predict the performance of a NDDCT are validated with both experimental and numerical observations. Numerical simulations of a small scale NDDCT indicate that the porous medium modelling is suitable for predicting the flow in and around the heat exchanger bundle. In addition, the (k-ε) turbulence model is found to be a reliable model to predict the complex nature of the flow in and around a cooling tower despite the fact that the model is believed to deviate from experimental results for low fluid velocities. The method presented in this paper makes it possible to conduct research on the design and performance of NDDCTs without the need to complex engineering details or iterative solution to the draft equation. Future research at QGECE will aim at examining different heat exchangers in the cooling tower and study the effects of wind on the heat and fluid flow from the air-cooled heat exchangers.

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Table 1. Summary of the governing equations with $\nu_T = 0.09k^2/\varepsilon$ (in porous layer: $k = 1.5\,\varphi^2(u^2+v^2)^{1/2}/10^4$ and $\varepsilon = 1.643k^{3/2}$).

Equations	ϕ	Γ_ϕ	S_ϕ
Continuity	1	0	0
x-momentum	u/φ^2	$(\nu+\nu_T)/\varphi$	$-\frac{1}{\rho}\frac{\partial p}{\partial x}-\frac{(\nu+\nu_T)u}{K}+\frac{C_F u\sqrt{u^2+v^2}}{\sqrt{K}}+\frac{\partial}{\partial x}\left((\nu+\nu_T)\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left((\nu+\nu_T)\frac{\partial v}{\partial y}\right)$
y-momentum	v/φ^2	$(\nu+\nu_T)/\varphi$	$-\frac{1}{\rho}\frac{\partial p}{\partial y}-\frac{(\nu+\nu_T)v}{K}+\frac{C_F v\sqrt{u^2+v^2}}{\sqrt{K}}+\frac{\partial}{\partial x}\left((\nu+\nu_T)\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial y}\left((\nu+\nu_T)\frac{\partial v}{\partial y}\right)+g\beta\Delta T$
Energy	T	$\alpha+\nu_T/Pr_T$	$q_b/(\rho c_p)$
Turbulent energy*	k	$\nu+\nu_T$	$\nu_T\left(2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial u}{\partial y}+\frac{\partial v)}{\partial x}\right)^2\right)-\varepsilon$
Turbulent dissipation*	ε	$\nu+0.77\nu_T$	$\frac{\varepsilon}{k}\left(1.44\nu_T\left(\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial u}{\partial y}+\frac{\partial v)}{\partial x}\right)^2\right)\right)-1.92\varepsilon$