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Dry cooling towers as condensers for geothermal power plants ☆

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ABSTRACT

The aim of this paper is to present scaling laws for a dry natural draft cooling tower by modelling the heat exchanger and the tower supports as a porous medium. Porous medium modelling of the tube bundles that allows a vigorous theoretical analysis of the problem is adopted. Scale analysis is used as the theoretical tool to study the problem of turbulent free convection through the heat exchanger bundles and along the cooling tower chimney. Results are then compared with full numerical simulation of the problem to observe splendid agreement.

1. Introduction

The need for renewable energy has pushed the world towards the use of more difficult or less economical options (at least at the short term) which are environmentally friendly. For instance, Australia has recently focused on the use of solar and geothermal energy on top of the already existing solar and hydropower sources. This is in spite of the fact that Australia has huge coal mines that can feed the coal-fired power plants which are the main power generation systems in the country. Such coal-fired power plants are the most economically feasible practices in Australia should there be no environmental issues. However, there has been great concern about the carbon emission from such coal-fired power plants in the past couple of decades. A large number of research projects aim at carbon capture from the emission sources leading to practical yet expensive ways of (almost) zero-emission power generation. While one research avenue is to reduce the costs of carbon capturing, the other one is to concentrate on renewable and green energy sources like geothermal to produce the electricity for the nation.

Studies show that the earth crustal temperature can be close to 300 °C at a depth of approximately 5 km. Nonetheless, these resources are mostly located at the arid areas where there is no water for evaporative cooling of the power plant. Besides, water scarcity during the past couple of years makes wet cooling even a less popular idea. Hence, air-cooled heat exchangers attract more attention to replace wet cooling systems. Mechanical draft systems lead to high parasitic losses depending on the fans which are in turn affected by the ambient air temperature. Hence, the use of natural draft cooling towers seems to become an immediate alternative to avoid parasitic losses.

There are a number of studies, some of them dating back to early 1960's, addressing the fairly complex problem of heat and fluid flow in such cooling towers [1–15]. Such specific issues as the effects of cross-wind on the performance of cooling towers have been recently addressed [16–24]. Liu [25] compares numerical predictions with field data. Williamson et al. [26] present an interesting one-dimensional model that compares well with the two-dimensional numerical simulations for a wet cooling tower. Lees [27] reports on the economy of wet and dry cooling towers. Kloppers and Kroger [28] investigate the performance evaluation of cooling towers by different methods. In a notable study, the world's tallest cooling tower in Germany has been simulated by Busch et al. [29].

Kroger [30] offers an excellent review of the literature and provides detailed empirical correlation to predict the pressure drop and heat transfer through the cooling tower. However, the use of the draft equation can be impossible without an iterative procedure. Hence, a reliable rough and ready estimate to correlate the tower geometry to its heat transfer performance is yet missing in the literature.

The aim of this paper is to fill this gap in the literature. Besides, running the experiments to try different heat exchangers in a cooling tower is almost impossible. Hence, a small scale cooling tower of 2 m height with square cross-section (1.4 m × 1.4 m base and 1 m × 1 m throat) is built in our QGECE lab [31] to study the scaling of the cooling towers. How the heat transfer and pressure drop of a prototype cooling tower relate to a small scale model is a question to be answered during the course of this paper.

2. Theoretical analysis

The driving force in a cooling tower is the air density difference that, following the use of Boussinesq approximation, takes the following form

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

This pressure difference should be higher than or at least equal to the tower hydrodynamic resistance. There are heat exchanger and

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tower resistances. The tower resistance can be approximated by the tower wall friction when the tower is looked at as a duct or a converging–diverging nozzle. The heat exchanger, on the other hand, can be modelled as a porous medium. In a previous study [32], a typical finned-tube bundle has been modelled as a porous medium with the following permeability and form drag coefficient

$$K = \frac{d^2\varphi^3}{100(1-\varphi)^2} \tag{2}$$

$$C_F = 0.55(9.887(1-\varphi)(\varphi-0.323)-0.8443) \tag{3}$$

where d is the tube diameter, and φ is the porosity of the tube bundle.

Hence, the heat exchanger pressure drop for flow of air across a finned-tube bundle of thickness t reads

$$\Delta p = t\left(\frac{\mu U}{K} + \frac{C_F\rho U^2}{\sqrt{K}}\right) \tag{4}$$

Normalizing the drags with the viscous (Darcy drag), the order of magnitude for the form/viscous drag ratio is $O\left(\frac{C_F\rho U\sqrt{K}}{\mu}\right)$. Making use of atmospheric air properties at 25 °C, the ratio scales well with $O\left(10^5 C_F U\sqrt{K}\right)$. This hints that for cases when $O\left(C_F U\sqrt{K}\right) > O(10^{-5})$ the form drag is the dominant pressure drop term. Besides, according to [32] 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. Making use of that finding, the criteria for a form drag-dominant flow through a finned-tube bundle is $U > O(10^{-2})$. Generally speaking, for high velocity and low viscosity fluids, the flow is very likely to be form drag-dominant (depending on the form drag coefficient and the permeability).

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 [33]

$$\Delta p = 0.5\rho U_f^2\left(f\frac{4H}{D_h} + \kappa\right) \tag{5}$$

where κ, as given by Table 1.1 of [33], puts on higher values than those given by $f \cong 0.08\mathrm{Re}^{-0.25}$. 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

$$\Delta p \sim 0.5\kappa\rho U_f^2 \tag{6}$$

One verifies to see that the order of magnitude of the svelteness [33], defined as the ratio of external flow length scale divided by that of internal flow, of the most of the cooling towers is much less than $O(10)$. This means that one would expect local losses to be comparable with distributed counterparts or even be the dominant pressure losses.

The total pressure drop should scale with

$$\Delta p \sim 0.5\kappa\rho U_f^2, t\frac{C_F\rho U^2}{\sqrt{K}} \tag{7}$$

Mass continuity can be used to relate the fluid velocity through the heat exchanger (volume-averaged velocity) to that in the cooling tower as $U = \varphi U_f$. This leads to

$$\frac{\Delta p}{0.5\kappa\rho U_f^2} \sim 1, \Omega \tag{8}$$

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 [32], we have $\Omega \sim O(10)$ and the heat exchanger pressure drop becomes the dominant one. However, one is in uncharted water if one takes this as a general conclusion that the heat exchanger pressure drop is always the dominant one. It cannot be known which term is the dominant one *a priori*.

In view of the above, the heat exchanger pressure drop should be balanced by the buoyant effects (when $\Omega >> 1$)

$$\rho g H\beta\Delta T \sim t\frac{C_F\rho U^2}{\sqrt{K}} \tag{9}$$

Rearranging the above equation, the volume-averaged velocity is given by

$$U \sim \sqrt{\frac{\sqrt{K}gH\beta\Delta T}{tC_F}} \tag{10}$$

According to the first law of thermodynamics, 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. $Q = \rho AUc_p\Delta T$ to get the volume-averaged velocity as

$$U = \frac{Q}{\rho Ac_p\Delta T} \tag{11}$$

In most of the industrial applications the main goal is to dissipate a certain amount of heat (a known parameter) with a cooling tower of a certain height. This height is to be determined. To answer this question, one equates the right-side of Eqs. (10) and (11) to eliminate the velocity. Then, one can find the minimum required height for a tower that dissipates a certain amount of heat (Q) as follows

$$H \sim t\frac{C_F}{Bo\sqrt{Da}}\left(\frac{Qt}{kA\Delta T}\right)^2 \tag{12}$$

with $Da = \frac{K}{t^2}$ being the Darcy number, $Bo = RaPr$ being the Boussinesq number, $Pr = \frac{\upsilon}{\alpha}$ being the effective Prandtl number, and $Ra = \sqrt{\frac{g\beta\Delta Tt^3}{\upsilon\alpha}}$ being the Rayleigh number. The term in the parenthesis, conduction imperfection, is a measure of heat dissipation by conduction. The total generated heat can be dissipated by conduction through a porous layer of thickness t and effective thermal conductivity k. Only perfect conduction could have dissipated the heat generated by the hot fluid flowing in the condenser tubes. Hence, the numerical value of the term in the parenthesis, conduction imperfection, is always less than one. Poor conduction leads to higher tower and this increases the cost whereas a more conductive material for the heat exchanger leads to a shorter tower and dissipates heat more efficiently. The tower height is linearly proportional to the form drag coefficient and the heat exchanger thickness, as one would expect. An increase in the form drag and/or the heat exchanger thickness increases the pressure drop and this asks for a taller cooling tower to provide the required driving buoyancy force. This observation favours a more compact heat exchanger that introduces less flow resistance. Finally, higher Boussinesq number leads to better heat transfer and thus the cooling tower height is inverse-linearly proportional to it.

Eq. (12) 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 Eq. (12) is given by

$$A = \pi DL \tag{13}$$

where D is the tower base diameter and L is the heat exchanger height.