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Numerical Investigation on Performance of Natural Draft Dry-Cooling Towers with Different Cooling Delta Angles

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The effects of different cooling delta angles on power plant natural draught dry cooling towers are investigated by means of Computational Fluid Dynamics (CFD) calculations. The heat exchangers are arranged horizontally in the inlet cross-section of the tower in the form of A-frames, which is also called cooling delta. A cooling delta consists of two cooling columns (heat exchanger tube banks), the angle between which is called cooling delta angle. This angle causes a non-uniform flow and temperature distribution within the heat exchangers, and thereby influences the thermal performance of the cooling tower. Since it is unpractical to model individual fins and tubes of a heat exchanger, the performance of the finned tube heat exchangers in the tower are modeled using a porous media approach, which is accurate enough instead of pore-scale simulations as long as the macroscopic properties are determined correctly. After a careful validation of the porous media approach, the 2D turbulent flow and pressure losses behind a cooling delta with different angles as well as the 3D natural convection heat transfer in natural draft dry cooling towers are studied. The results agree with the engineering reality and can be used to further find an optimal cooling delta angle for different projects.

1. Introduction

There are many types of power plant cooling systems, including fresh water (once-through) cooling, evaporative cooling and dry cooling systems. The later exclude the direct contact between the cooling water and the air, thus significant water losses at evaporative cooling are avoided. So the plant can be located where rich in mine but with scarce water sources in North China. In indirect dry cooling towers, the finned-tube air cooled heat exchangers are usually arranged horizontally in the inlet cross-section of the tower in "delta" formations, to save space. However, severe non-uniform inlet air flow and temperature maldistribution will occur due to the inclination of the heat exchanger bundles, which may reduce the thermal performance of the heat exchangers and therefore the whole natural draft dry cooling tower.

In the past, a number of efforts have been taken to study the flow losses due to flow through inclined finned tube heat exchanger bundles as well as the deteriorated thermal performance. Moore and Ristorcelli (1979) analyzed the flow field downstream of a V-bundle by using a differential self-similar solution that took into account the variable

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velocity distribution through the bundles. Aarde and Kroger (1993) determined the flow losses through an A-frame finned tube heat exchanger experimentally and derived some correlations describing these losses. Beiler and Kroger (1996) addressed the effect of non-uniform inlet air flow and temperature on the thermal performance of air cooled heat exchangers. They found that the deterioration in heat transfer performance due to maldistribution was negligible. Lately, Kapas (2005) numerically investigated the effects of wind on a Heller–type natural draught dry cooling tower where heat exchangers are arranged vertically around the perimeter of the tower.

As most experiments acquire data merely ahead or behind a heat exchanger bundle through wind tunnel testing instead of a full-scale heat exchanger installation measurement, it is impossible to take into consideration all the factors downstream and upstream. In addition, a very rapid development in the field of CFD makes simulation of the very complex fins and tubes geometry in the air-cooled heat exchangers feasible. Thus, we will first carefully validate the models in the well-known CFD code FLUENT, and then numerically study the 2D turbulent flow and pressure losses behind a cooling delta as well as the 3D natural convection heat transfer of a cooling delta.

2. Validation of Semi-Empirical Heat Exchanger Model

It is impractical to model individual tubes and fins in an air cooled heat exchanger. In this study, the heat exchanger is treated as a fluid zone with momentum and heat transfer. Pressure loss is modeled as a momentum sink in the momentum equation, and heat transfer is modeled as a heat source in the energy equation. For the pressure drop calculation, the porous media approach is employed; while for the heat transfer calculation, the internally contained user-defined heat exchanger model in FLUENT is applied. As we don't know if they are suitable to our problem, the validation of the code must be done firstly.

2.1 Porous Media Approach for Pressure Drop Calculation

As mentioned above, the porous media is nothing more than an added momentum source term to the standard governing momentum equations. The source term is composed of two parts, a viscous loss term (the Darcy term) and an inertial loss term (the Forchheimer term), as follows,

$$S_i = -\left(\sum_{j=1}^{3} D_{ij}\mu v_j + \sum_{j=1}^{3} C_{ij}\frac{1}{2}\rho|v|v_j\right) \tag{1}$$

where D and C are prescribed matrices.

For simple homogeneous porous media,

$$S_i = -\left(\frac{\mu}{\alpha} v_i + C_2 \frac{1}{2}\rho|v| v_i\right) \tag{2}$$

where α is the permeability and C_2 is the inertial resistance factor.

From the early wind tunnel testing of a two tube passes air-cooled heat exchanger (shown in Figure 1), we obtained the correlation between pressure drop and frontal velocity

$$\Delta P = 6.78V_{fr}^{\ 2} + 10.5V_{fr} \tag{3}$$

Comparing Equations (2) and (3), we can get α and C_2 for porous media inputs.

2.2 User-Defined Heat Exchanger Model Theory

Heat exchanger model is a lumped-parameter model in nature. The model can be used to compute auxiliary fluid inlet temperature for a fixed heat transfer rate or total heat transfer rate for a fixed auxiliary fluid inlet temperature. In FLUENT, the fluid flow and temperature distribution in tube side is treated as one dimensional. For heat transfer calculation, users need to provide heat transfer data obtained from experiments or empirical correlations, such as velocity vs. effectiveness curve of heat exchangers. Specifically, the effectiveness of the studied heat exchanger can be taken as

$$\varepsilon = \frac{1}{C^*}\left[1 - e^{-C^*(1-e^{-NTU})}\right] \tag{4}$$

where $NTU = \dfrac{UA}{C_{\min}}$, $C^* = \dfrac{C_{\min}}{C_{\max}}$, U is the heat transfer coefficient, A is the air side heat transfer area and C is the thermal capacity. The correlation between U and V_{fr} is obtained from experiments

$$U = 34.7776V_{fr}^{\ 0.2157} \tag{5}$$

2.3 Solution Procedure

The used computational domain for validation is a rectangular wind tunnel model with a cross section area of 600×320 mm. Four experimental conditions are chosen to carry out the validation, typicallyV_{fr}=1.09 m/s, 1.76 m/s, 2.42 m/s, and 3.27 m/s. The flow is turbulence due to the relatively large hydraulic diameter of the rectangular channel. Since accurate turbulent flow computations are not critical in this study, we choose the modified Spalart-Allmaras one equation eddy-viscosity turbulence model to calculate turbulence. Grid independence has been examined first prior to next step calculation. Finally, a grid number of 33120 is found to be fine enough for this calculation.

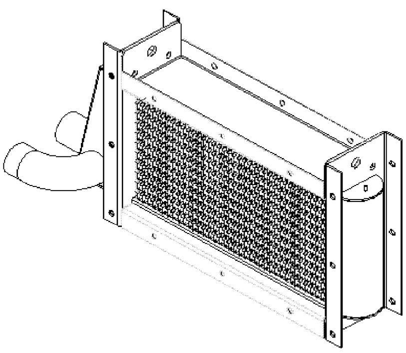

Figure 1: Schematic of the wind tunnel tested heat exchanger.

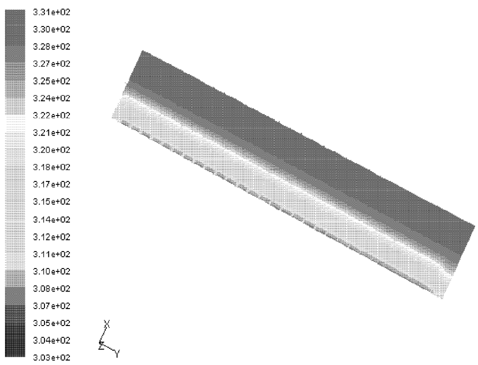

Figure 2: Predicted temperature (K) field of the heat exchanger core.

2.4 Results and Discussion

Figure 2 shows the predicted temperature field of air in the heat exchanger core, which is heated by the tube side hot water as it passes the heat exchanger bundle. The calculated static pressure drop through the heat exchanger bundle and total heat transfer rate compared with experimental data are shown respectively in Figure 3 and Figure 4. We observe that the static pressure drop through the heat exchanger acquired from the porous media approach agrees well with the experimental data. It is because the porous media model directly uses the flow resistance determined by experiment. However, we notice that predicted heat transfer rate is much greater than the experimental data, especially under high velocity conditions. For example, the relative error of heat transfer rate is about 30% when the frontal velocity is between 1.5 m/s and 2.2 m/s which are considered to be the normal velocity in engineering reality. This large error may be caused by a higher effectiveness for each computational cell which is considered to be equal to the effectiveness for the complete heat exchanger.

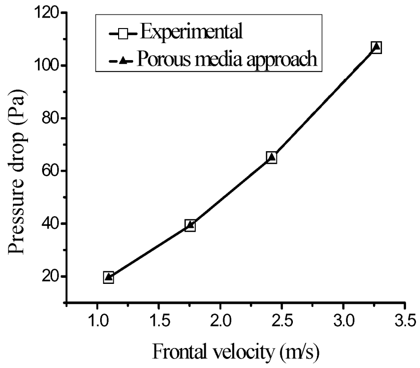

Figure 3: Static pressure drop.

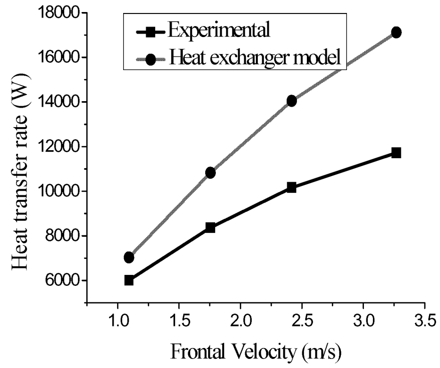

Figure 4: Heat transfer rate.

3. Pressure Losses and Heat Transfer Rate of a Cooling Delta with Different Angles

3.1 Solution Procedure

The 2D forced convection and 3D natural convection of cooling delta with different delta angles are numerically studied respectively here. For turbulent modeling, the behavior of the k-ε model for free shear turbulence is well known, so the k-ε turbulence model is adopted. The above developed semi-empirical heat exchanger model is used to model the air-cooled heat exchangers, which have fine performance for pressure drop calculation as observed in the second part. Though significant discrepancy is observed in predicting the thermal performance of the heat exchanger, we can deduce some qualitative results. Since the geometry is symmetric, only half of the domain is selected for calculation, thus symmetry boundary condition is employed. Besides, ahead of the heat exchanger a 5 m length extended domain is constructed while a 50 m length extended domain behind is applied to predict the long jet flow downstream.

3.2 Results and Discussion

Figure 5 shows the predicted flow field through a cooling delta with a delta angle of 60. As illustrated, streamlines emerge from the heat exchanger in a direction normal to the bundle surface, and then converge into a jet-like flow pattern downstream, making the velocity in the center of the V-bundle surprisingly large. In addition, severe flow separation from the downstream corners occurs due to the rest flows. This distorted downstream profile generates flow losses and hence will lead to larger heat exchanger pressure drops especially under narrower delta angle conditions as observed in Figure 6.

The pressure losses through and behind the cooling delta with different delta angles are shown in Figure 6. As is shown, the static pressure drop increases with the delta angle decreasing from 90° to 40°. This is mainly ascribable to the distorted downstream velocity profile when the delta angle is small. In fact, downstream flow distortion will cause large mixing losses far away from the bundle. Specifically, when the delta angle is less than 60°, the pressure drop soars in a sudden, which explains

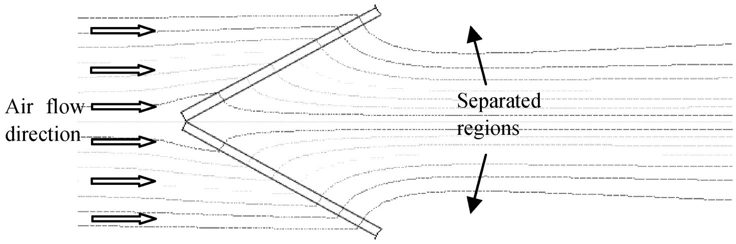

Figure 5: Predicted jet-like flow field downstream of a cooling delta.

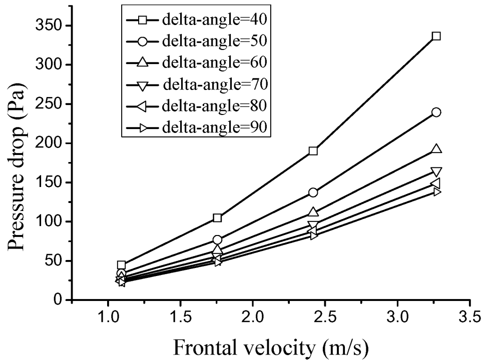

Figure 6: Static pressure drops between flow inlet and outlet of the computational domain.

why the cooling delta is usually arranged at 60° angle in most cooling towers.

The predicted natural convection heat transfer rate of half a cooling delta in dry cooling towers is show in Figure 7. As we can see, for a single cooling delta, the wider the angle, the better the heat transfer performance. On the other hand, for a fixed tower basal area, the heat transfer rate of the whole cooling tower increases as the individual cooling delta angle narrows from 80° to 40°. However, excessively small delta angles may cause other engineering problems such as thermal stresses.

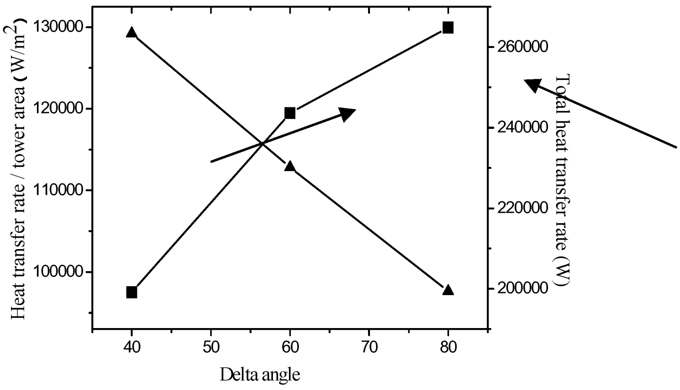

Figure 7: Total heat transfer rate and heat transfer rate per unit tower area.

4. Conclusions

1. The developed semi-empirical heat exchanger model behaves well when predicting the pressure drop through the air-cooled heat exchanger, while it calculates a higher heat transfer rate. Hence more efforts should be taken to improve the heat transfer model.
2. The pressure drop through and behind a cooling delta balloons when the delta angle is smaller than 60°.
3. The performance of the whole dry cooling tower increases with the decreasing delta angle, although the individual cooling delta heat transfer performance deteriorates.

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