Numerical study of the thermo-flow performances of novel finned tubes for air-cooled condensers in power plant

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Abstract. Air-cooled condenser is the main equipment of the direct dry cooling system in a power plant, which rejects heat of the exhaust steam with the finned tube bundles. Therefore, the thermo-flow performances of the finned tubes have an important effect on the optimal operation of the direct dry cooling system. In this paper, the flow and heat transfer characteristics of the single row finned tubes with the conventional flat fins and novel jagged fins are investigated by numerical method. The flow and temperature fields of cooling air for the finned tubes are obtained. Moreover, the variations of the flow resistance and average convection heat transfer coefficient under different frontal velocity of air and jag number are presented. Finally, the correlating equations of the friction factor and Nusselt number versus the Reynolds number are fitted. The results show that with increasing the frontal velocity of air, the heat transfer performances of the finned tubes are enhanced but the pressure drop will increase accordingly, resulting in the average convection heat transfer coefficient and friction factor increasing. Meanwhile, with increasing the number of fin jag, the heat transfer performance is intensified. The present studies provide a reference in optimal designing for the air-cooled condenser of direct air cooling system.

1. Introduction

Direct dry cooling systems (DDCS) are widely used in coal-fired power plants located in the inland regions of China for the superiority of water conservation. DDCS usually comprises dozens of air-cooled condenser cells, commonly arranged in a rectangular array. An air-cooled condenser cell mainly consists of the A-frame finned tube bundles and the axial flow fan below. Since the finned tubes are the major component to reject the heat of the exhaust steam, their thermos-flow performances are directly associated with the economical and effective operation of the power plant. Hence, it is of great value to study the flow and heat transfer characteristics of the finned tubes.

The researches on the heat transfer enhancement of finned tubes are widely conducted at home and aboard. Chi-Wen Lu and Jeng-Min Huang [1] numerically studied the influences of the geometry parameters on the thermo-flow performances for the two-rows finned tubes, including the fin spacing, base tube spacing, fin thickness and base tube diameters. The result shows that the heat transfer performance of finned tubes increases with increasing the base tube spacing and diameter but decreasing the fin thickness. Harun Bilirgen [2] carried out a numerical research about the flow and heat transfer characteristics of annular finned tube with different fin length, spacing, thickness and material. In conclusion, the heat transfer ability is enhanced with increasing the fin length and decreasing the fin spacing. Cathal TO Cleirigh and William J Smith [3] confirmed the reliability of
numerical method applied to the research of the flow and heat transfer characteristics of three-rows spiral finned tubes. The numerical results of air-side pressure drop and Nusselt number matches well with the published empirical equations. Du [4] conducted an experimental study on the air-side thermo-flow performances of flat finned tube for direct dry cooling systems. What’s more, the enhancement of thermo-flow performances by small triangular fins located on the surface of the fins was investigated as well. The results show that the arrangement of small triangular fins is able to improve the heat transfer characteristics of the finned tubes remarkably. By means of numerical method, W.M. Song [5] studied the performances of single row flat finned tubes with crossed ribs in different structures, intervals and attack angles. The results show that Nusselt number and friction factor increase with increasing the rib height and interval. Besides, there is an optimal distance between the entrance and first rib. Yang [6] proposed a novel fin arrangement by changing the established angle of the fins to fit the A-frame air-cooled condenser cell for restraining the ash deposition. The performances of the novel finned tubes were obtained by numerical simulation. The characteristics of various types of finned tubes for the indirect dry cooling systems under different air face velocities were also investigated in Ref. [7] by numerical simulation. The results show that with increasing the wind speed, the increase range of the heat transfer coefficient is less than that of the pressure drop. Weifeng He and Yiping Dai [8] numerically studied the flow performance of single-row and two-row finned tubes under various wind speeds. The results show that the increasing range of the flow resistance with increasing the wind speed for the two-row finned tubes is larger than that of the single one. Lingdong Gu et al. [9] took the air property variability into consideration for the numerical simulation of the finned tubes, the results show that the ignorance of air property variation causes overestimation of the thermos-floe performances of the finned tubes.

Literature review found that analysis on finned tubes is mainly concentrated on the basic geometric parameters, the tubes with jagged fins are not considered in detail. Based on numerical simulation, the effect of jagged fins and the number of jagged fins on the thermo-flow performances of finned tubes are investigated further in this paper.

2. Methodologies

2.1. Physical models

The configurations of commonly used finned tubes for direct dry cooling system are flat base tubes with wavy rectangular fins. In this paper, the normal finned tube is named as type A. The finned tube with two separated jagged fins is named as type B, the finned tube with five separated jagged fins is named as type C and the finned tube with ten separated jagged fins is named as type D. Figure 1 shows the schematic configuration of different type of finned tubes. Since the repeated periodic configuration of fins in the longitudinal direction of the tubes and the symmetric structure of fins in the transverse direction of the tubes, only one fin on half of the tube is taken into consideration.

The structural parameters and mesh generation of the finned tubes are showed in Figure 2 and Figure 3 respectively. The fin pitch is 2.3mm, the fin thickness is 0.26mm, the thickness of flat tubes is 1.3mm, the fin length is 200mm and the fin width is 19mm.
Figure 1. Schematics of finned tubes models.

Figure 2. Structural parameters of finned tubes.

Figure 3. Schematic of grids for finned tube.
2.2. Mathematical models and numerical methods

With the same mathematical model predicted in in Ref. [7], the thermos-flow performances of the finned tubes are investigated in this paper, which is proved to be reliable. The mathematical models are built based on the air-side physical model of the finned tubes for air-cooled condensers. In order to describe the flow and heat transfer processes of the airflow between the fins, the realizable $k$-$ε$ model is adopted. The governing equations are shown as follow:

$$\frac{\partial \rho u_j \varphi}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \Gamma \frac{\partial \varphi}{\partial x_j} \right) + S_\varphi \quad (j = 1, 2, 3)$$

where $\rho$ is air density, $u_j$ is the velocity in the $x_j$ direction and $\varphi$, $\Gamma$, $S_\varphi$ represent the variable, its diffusivity and the source term respectively, which are listed in Table 1.

| Equations       | $\varphi$ | $\Gamma_\varphi$ | $S_\varphi$ |
|-----------------|-----------|------------------|-------------|
| Continuity      | 1         | 0                | 0           |
| x-momentum      | $u_i$     | $\mu_e$          | $-\rho \frac{\partial \varphi}{\partial x_i} + \frac{1}{\rho} \left[ \frac{\partial}{\partial x_i} \left( \mu_e \frac{\partial \varphi}{\partial x_i} \right) + \frac{\partial}{\partial x_j} \left( \mu_e \frac{\partial \varphi}{\partial x_j} \right) \right]$ |
| y-momentum      | $u_j$     | $\mu_e$          | $-\rho \frac{\partial \varphi}{\partial x_i} + \frac{1}{\rho} \left[ \frac{\partial}{\partial x_i} \left( \mu_e \frac{\partial \varphi}{\partial x_i} \right) + \frac{\partial}{\partial x_j} \left( \mu_e \frac{\partial \varphi}{\partial x_j} \right) \right]$ |
| z-momentum      | $u_k$     | $\mu_e$          | $-\rho \frac{\partial \varphi}{\partial x_i} + \frac{1}{\rho} \left[ \frac{\partial}{\partial x_i} \left( \mu_e \frac{\partial \varphi}{\partial x_i} \right) + \frac{\partial}{\partial x_j} \left( \mu_e \frac{\partial \varphi}{\partial x_j} \right) \right] + \rho g$ |
| Energy          | $c_p\rho$ | $\mu_e/\sigma_T$ | 0           |
| Turbulence kinetic energy | $k$     | $\rho \mu + \mu_T/\sigma_k$ | $G_K + G_b - \rho \varepsilon$ |
| Turbulence dissipation rate | $\varepsilon$ | $\rho \mu_T/\sigma_k$ | $\rho c_s^2 \varepsilon - \rho c_s^2 \varepsilon^2 / (1 + \varepsilon) - \rho c_s^2 \varepsilon - \rho c_s^2 \varepsilon^2 / (1 + \varepsilon)$ |

The geometric modelling and computational meshes generating are completed with the commercial software GAMBIT. The computational domain is divided into three parts including the inlet part, middle part and outlet part, as shown in Figure 4. The boundary conditions are presented in Figure 4 as well. The inlet air velocity varies from 0.25 m/s to 2.75 m/s, the temperatures of the inlet air and the tube wall are set as 288.15 K and 324.19 K respectively. The governing equations are discretized and then solved by the commercial software ANSYS Fluent. The SIMPLE algorithm and the double precision are employed. When the residual of energy equation is lower than $10^{-6}$ and the residuals of the other equations are lower than $10^{-4}$, the calculation results are regard as convergent.

![Figure 4. Computational domain and boundary conditions.](image-url)
3. Results and discussion

3.1. Data processing
The average inlet and outlet pressures and temperatures of the cooling air under different inlet air velocities should be obtained by the numerical simulation. And then the pressure drop and heat transfer coefficient under different conditions will be calculated using equation (3) and equation (5). Finally, the correlations of Nusselt number and friction factor versus the Reynolds number will be fitted subsequently. The detailed approaches are described as follows.

The definition of Reynolds number is:

$$Re = \frac{\rho u D}{\mu}$$  \hspace{1cm} (2)

where $\rho$ is air density, $\mu$ is air dynamic viscosity, $u$ is air velocity of the minimum cross-section between fins and $D$ is the hydraulic diameter of the corresponding cross-section.

The pressure drop of the airflow is defined as follow:

$$\Delta p = p_1 - p_2$$  \hspace{1cm} (3)

where $p_1$ (Pa) and $p_2$ (Pa) is the average inlet and outlet air pressure respectively.

The definition of friction factor is:

$$f = \frac{2\Delta p D}{\rho u^2 L}$$  \hspace{1cm} (4)

where $L$ (m) is the flow path of air.

The convective heat transfer coefficient is defined as follow:

$$h = \frac{Q}{A\Delta t_m}$$  \hspace{1cm} (5)

where $Q$ (W) is convective heat transfer rate, $A$ (m$^2$) is heat area of finned tubes and $\Delta t_m$ (K) is logarithmic mean temperature difference.

Nusselt number is calculate as:

$$Nu = \frac{hD}{\lambda}$$  \hspace{1cm} (6)

where $\lambda$ is air thermal conductivity. The arithmetic mean of inlet and outlet air temperature is set as the reference temperature.

3.2. Results and analysis
Figure 5 and Figure 6 respectively show the temperature distributions at the fin surface and central cross-section along the airflow direction. It can be concluded that the temperature gradient is relatively high at the entrance, and along the direction of flow, the air temperature increases and the convective heat transfer recedes, hence the temperature of fins rises gradually and it reaches the maximum value at the exit. Compared to the intact fin, the continuous development of boundary layer near the fin is broken by the jags of the fin. Accordingly, heat boundary layer generates periodically along the length direction of fins, which leading to a thinner heat boundary layer. Consequently, the thermal resistance becomes smaller and the convective heat transfer is enhanced. As a result, the performance of heat transfer gets better with increasing the number of jags.
The variations of air pressure drop under different frontal velocities are presented in Figure 7. It shows that the changes of the air pressure drop for different type of finned tubes have the same trend with the existing researches. With increasing the frontal wind speed, the pressure drop increases for all the finned tubes. However, the pressure drop increases with more jags, which leads to a higher flow resistance, especially at high frontal velocities.

The air-side average convective heat transfer coefficient $h$ for different type of finned tubes under various frontal air velocities $u_f$ are shown in Figure 8. It can be seen that with increasing the frontal velocity of air, the heat transfer coefficient rises gradually. For the fined tubes with more fin jags, the
convective heat transfer coefficient gets higher. That is to say, the finned tube with more fin jags has a better heat transfer performance compared to the original type A.

![Figure 8. Heat transfer coefficient under different frontal air velocities.](image)

As shown in Figure 9, the variations of the friction factor $f$ versus the Reynold number for different type of finned tubes are depicted. Generally, the friction factor $f$ is the function of the Reynold number, which means changing law of the pressure drop with the flow velocity for the cooling air. With increasing the Reynold number, the friction factor decreases correspondingly. The difference of friction factor between two types of finned tubes is basically the same. Moreover, it increases with increasing the number of fin jags. Therefore, the more fin jags, the higher friction factor.

![Figure 9. Friction factor for different Reynolds number.](image)
The Nusselt number varies with the Reynolds number for different type of finned tubes are presented in Figure 10. It can be seen that the changing law of the Nusselt numbers for different finned tube types are essentially consistent. The Nusselt number increases with increasing the Reynolds number for all the finned tubes. The original finned tube type A has a lowest Nusselt number at all the conditions, while the Nusselt numbers for finned tubes type B, C and D are much higher. For the finned tube type D with ten fin jags, it has the highest Nusselt number, which means that the finned tube type D is the most outstanding one to enhance the convective heat transfer performances of the finned tubes.

![Figure 10. Nusselt number for different Reynolds number.](image)

3.3. Characteristic correlations

By means of numerical simulation, the correlations of convective heat transfer coefficient as the frontal velocity of air, the friction factor and Nusselt number as the Reynolds number could be obtained. As listed in Table 2, the correlations in power function form are fitted using least square method. The correlations can be provided as references for the performance calculation and design of air-cooled condensers in direct dry cooling systems. It is of worth to point out that the range of frontal air velocity $u_f$ varies from 0.5 m/s to 2.75 m/s and the range of $Re$ varies from 550 to 3000.

| Type   | Heat Transfer Coefficient Correlation | Friction Factor Correlation | Nusselt Number Correlation |
|--------|---------------------------------------|----------------------------|----------------------------|
| Type A | $h = 29.17u_f^{0.2099}$               | $f = 71.01Re^{-0.5099}$    | $Nu = 18.89Re^{0.2099}$    |
| Type B | $h = 31.08u_f^{0.2098}$               | $f = 74.81Re^{-0.499}$     | $Nu = 20.12Re^{0.2098}$    |
| Type C | $h = 31.66u_f^{0.2078}$               | $f = 79.14Re^{-0.492}$     | $Nu = 20.46Re^{0.2078}$    |
| Type D | $h = 32.67u_f^{0.2198}$               | $f = 83.33Re^{-0.447}$     | $Nu = 21.15Re^{0.2198}$    |
4. Conclusions
The flow and heat transfer characteristics of commonly used finned tubes in air-cooled condensers for the direct dry cooling power plant are numerically investigated. In detail, the thermos-flow performances of finned tubes with different number of jags are simulated. The air-side pressure drop and convective heat transfer coefficient under different frontal velocities of cooling air, as well as the friction factor and Nusselt number under various Reynolds number are calculated and analyzed. Finally, the correlation between convective heat transfer coefficient and frontal air velocity and the correlation between friction factor and Nusselt number and Reynolds number are fitted in order to estimate the flow and heat transfer performances of finned tubes with different quantity of jags. The results show that with increasing the frontal velocity of cooling air, the air-side pressure drop increases corresponding to an improvement of air-side convective heat transfer performance. At small Reynolds number, the change of heat transfer capacity is significant. On the other hand, for the finned tube types B, C and D with jagged fin, the thermal performances are obviously enhanced, especially the type D. The jagged fins have an obvious disturbance effect on the boundary layer, which could raise the heat transfer coefficient of fin surface effectively.

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References
[1] Chi Wen Lu, Jeng-Min Huang 2011 A numerical investigation of the geometric effects on the performance of plate finned-tube heat exchanger [J] Energy Conversion and Management 52 1638-1643
[2] Harun bilirgen, Stephen Dunba, K Levy 2013 Numerical modeling of finned heat exchangers[J] Applied Thermal Engineering 61 278-288
[3] Cathal T Ó Cleirigh, William J Smith 2014 Can CFD accurately predict the heat-transfer and pressure-drop performance of finned-tube bundles[J] Applied Thermal Engineering 73 679-688
[4] Xiaoze Du, Lili Feng 2013 Experimental study on heat transfer enhancement of wavy finned flat tube with longitudinal vortex generators[J] Applied Thermal Engineering 50 55-62
[5] WM Song 2010 Numerical study of air-side performance of a finned flat tube heat exchanger with crossed discrete double inclined ribs[J] Applied Thermal Engineering 30 1797-1804
[6] Lijun Yang 2012 Thermal-flow characteristics of the new wave-finned flat tube bundles in air-cooled condensers[J] International Journal of Thermal Sciences 53 166-174
[7] Lijun Yang, Sining Jia 2012 Numerical study on flow and heat transfer characteristics of finned tube bundles for air-cooled heat exchangers of indirect dry cooling systems in power plants[J] Proceedings of the CSEE 32 50-57
[8] Weifeng He, Yiping Dai 2012 Numerical study on the resistance characteristics of finned tube heat exchangers in an air-cooled steam condenser[J] Thermal Turbine 41 196-200
[9] Lingdong Gu, Jingchun Min, Xiaomin Wu, Lijun Yang 2017 Airside heat transfer and pressure loss characteristics of bare and finned tube heat exchangers used for aero engine cooling considering variable air properties[J] International Journal of Heat and Mass Transfer 108 1839-1849