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Steam Condensation Cooled by Piezoelectric Driven Oscillating Air Flow

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Abstract
The piezoelectric (PE) fan is first put forward to use in the air-cooled condenser, and to enhance the exhaust steam condensation of large thermal power plants. The cooling performance of such a method is evaluated through numerical simulations. The study shows that the surface heat transfer coefficient with PE fan at the air side is higher than that of a conventional axial flow (AF) fan. While at the steam side of the tube, the surface heat transfer coefficient is also enhanced owing to the obvious fluctuation of liquid film caused by the vibration of the piezoelectric fan.

Key words: Piezoelectric fan; Air-cooling condenser; Numerical simulation; Surface heat transfer coefficient

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INTRODUCTION

The air-cooled condenser (ACC) in a power plant consists of an array of the condenser cells. For each condenser cell, the finned tube bundles are arranged in the form of an A-frame fitted with an axial flow fan below. The ambient air is impelled by the fans to flow through the finned tube bundles, removing the thermal duty of the exhaust steam from a turbine. The application and popularity of air-cooled condenser is increasing in the power plants throughout the countries that are short of water resources during the last decade[1, 2].

Strengthening of heat-transfer, reduction of flow resistance as well as enhancement of compactness has been considered crucial for air-cooler due to the minor air heat-transfer coefficient. Existing proposals for enhancing the heat transfer performance can be broadly classified as either active (e.g. piezoelectric fans, surface vibration, electric or magnetic fields, and so on) or passive (e.g. extended surfaces, swirl flow devices, vortex generators, and so on)[3]. Piezoelectric (PE) fan is an innovative design which consists of a patch of piezoelectric material attached to a blade of various flexible materials. The blade vibrates of piezoelectric fan at the resonate frequency of cantilever fan under the actuating voltage, and creates a flow field which is more complex than that produced by natural convection. As a result, a significant improvement in the heat transfer performance is obtained. This oscillatory motion drives air flow which can be exploited for cooling. However, the researches on PE fans mostly focus on the cooling of portable electronic devices without consideration of large-scale applications[4-7]. In this paper, the PE fan is first put forward to be applied in the air-cooling condenser of thermal power plant. The cooling performance of this method is evaluated by numerical calculation. Its effect of strengthening the heat transfer has been analyzed and to be compared to conventional axial fans.

1. COMPUTATIONAL FLUID DYNAMIC METHODOLOGY

1.1 Computational Domain Generation

To study the strengthening of cooling effect by the PE fan, the temperature field and velocity field beside the tube is investigated by computational fluid dynamics (CFD) mathematical modeling and simulation. Without considering the complicated steam and fluid flow inside

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the condensation tube for simple, it takes one side of the tube surface and a PE fan plate in the model and the model sets as a two-dimensional model (Figure 1(a)). In the numerical model, the length of the PE fan is 0.5 m and the distance between the tip of the fan and the out surface of the tube is 0.1 m, with both consideration of the cooling performance and setup requirement. The thickness of the tube is 1.6 mm. Generally the PE fans for electronic cooling are much smaller, which are from millimeter to micrometer. In this work, PE fan should be a larger one to meet the size of the tube. The mesh grid field is shown in Figure 1(b); the main size of calculation field is 1 m×1 m to ensure the fully developed turbulent flow. In order to eliminate the effect of mesh quality and size on the results, the grid at the margin of the fan and the tube surface is modified denser and set more computing node.

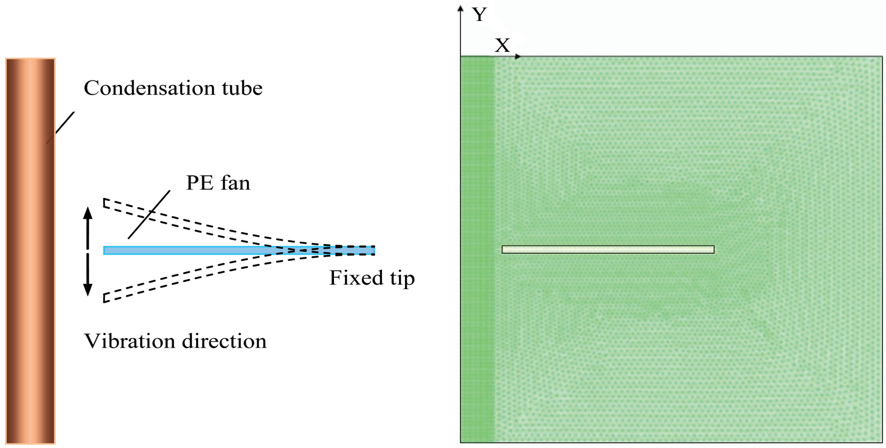

Figure 1
(a) Distribution of a Single PE Fan and the Heat Exchange Tube; (b) The Grid of the Calculation Area

1.2 Governing Equations

The condensation tube is cooling down by the air oscillation caused by periodic vibration of PE fan. The air flow could be incompressible flow. The hydrodynamics of the fluid flow condition can be discretized by the solution of the governing equations. The governing Equations (1)-(6) shown below were used to solve incompressible fluid flow for two-dimension.

Continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

Momentum conservation equation:

$$\rho(\frac{\partial u}{\partial \tau}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y})=F_x-\frac{\partial p}{\partial x}+\eta(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}) \tag{2}$$

Energy conservation equation:

$$\rho(\frac{\partial v}{\partial \tau}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y})=F_y-\frac{\partial p}{\partial y}+\eta(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}) \tag{3}$$

Energy conservation equation:

$$\frac{\partial t}{\partial \tau}+u\frac{\partial t}{\partial x}+v\frac{\partial t}{\partial y}=\frac{\lambda}{\rho c_p}(\frac{\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}) \tag{4}$$

where ρ, λ, and c_p are the density, thermal conductivity and specific heat. u, v, F_x and F_y are the velocity and force on the micro-unit in x and y direction respectively. For the fluid in the field is in the state of turbulence, the Standard k-ε two-equation model is adopted.

$$\rho\frac{\partial K}{\partial t}+\rho u_j\frac{\partial K}{\partial x_j}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial K}{\partial x_j}\right]+\mu_t\frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\rho\varepsilon \tag{5}$$

$$\rho\frac{\partial \varepsilon}{\partial t}+\rho u_k\frac{\partial \varepsilon}{\partial x_k}=\frac{\partial}{\partial x_k}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_k}\right]+\frac{c_1\varepsilon}{K}\mu_t\frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-c_2\rho\frac{\varepsilon^2}{K} \tag{6}$$

where the turbulence dissipation rate is calculated as: $\varepsilon=\frac{u}{\rho}\overline{(\frac{\partial u_i'}{\partial x_k})(\frac{\partial u_i'}{\partial x_k})}$, turbulent viscosity is calculated as: $\mu_t=c_\mu\rho K^2/\varepsilon$.

1.3 Simulation Conditions

In the simulation model, 2-D SIMPLE algorithm and implicit pressure-based separation solver are chosen. The commercial finite-volume based solver Fluent is used to solve Equations (1)-(6) along with the boundary conditions .Model coefficients are taken as $c_\mu=0.09$, $c_1=1.44$, $c_2=1.92$, $\sigma_k=1.0$, $\sigma_\varepsilon=1.3$. A divergence-free criterion of 10^{-4} based on the scaled residual is prescribed for the computations.

1.4 Boundary Conditions

The top and bottom boundary of the calculation field are treated as thermal insulated as well as the PE fan boundary. The right boundary is set as a constant (283.15K) in accord with the temperature of the environment. The left boundary is the inside surface of the tube, which involves phase transition. And the heat transfer coefficient of the phase transition oscillates with

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the position, the time and the surface roughness. As this situation, the left boundary condition is given out in the third kind of boundary condition for simplicity.

1.5 User Defined Function (UDF) for Vibration

Piezoelectric fan is a cyclical movement in the flow field, so it is necessary to reconstruct the grid. Study the internal flow field pressure fan movement, the key is User-Defined Function (UDF) independent definition of the piezoelectric fan movement form. The motion of the angular velocity equation is Navier-Stokes equation. The frequency of the PE fan sets 5 Hz as it gained in experiment, which sets in the model. UDF is used in the model to iterate the fan moving. Basically, UDF is written in ‘C’ programming language. It can access FLUENT solver data using predefined macros or function supplied by the simulator. The UDF would be interpreted and hooked to the solver during simulation. This UDF would access the solver data, perform calculation on the data and update the solver data repeatedly for every iteration. In the model, it chooses Rigid Body type to define the moving grid and define CG motion macro to define the moving.

2. RESULTS AND DISCUSSION

2.1 Heat Transfer Coefficient of the Air Side Surface of the Cooling Tube

While the frequency of the PE fan is 5 Hz, the whole flow field fully developed need over 20 s; the computational domain shown in Figure 2 is the distribution of the density, pressure, x-direction velocity, air flow line and temperature in the flow field at 20.012 s. In the wide range of the area near the fan, density is all the same; the density at the front of the fan is smaller and at the end of the fan is bigger. Along the movement direction, the pressure in front fan is opposite pressure. As to the velocity field, there is two eddies current beside the fan. The numerical result is in agreement with actual situation. In Figure 3 it is the temperature of the field, there is a thin layer near the left boundary which is the tube wall.

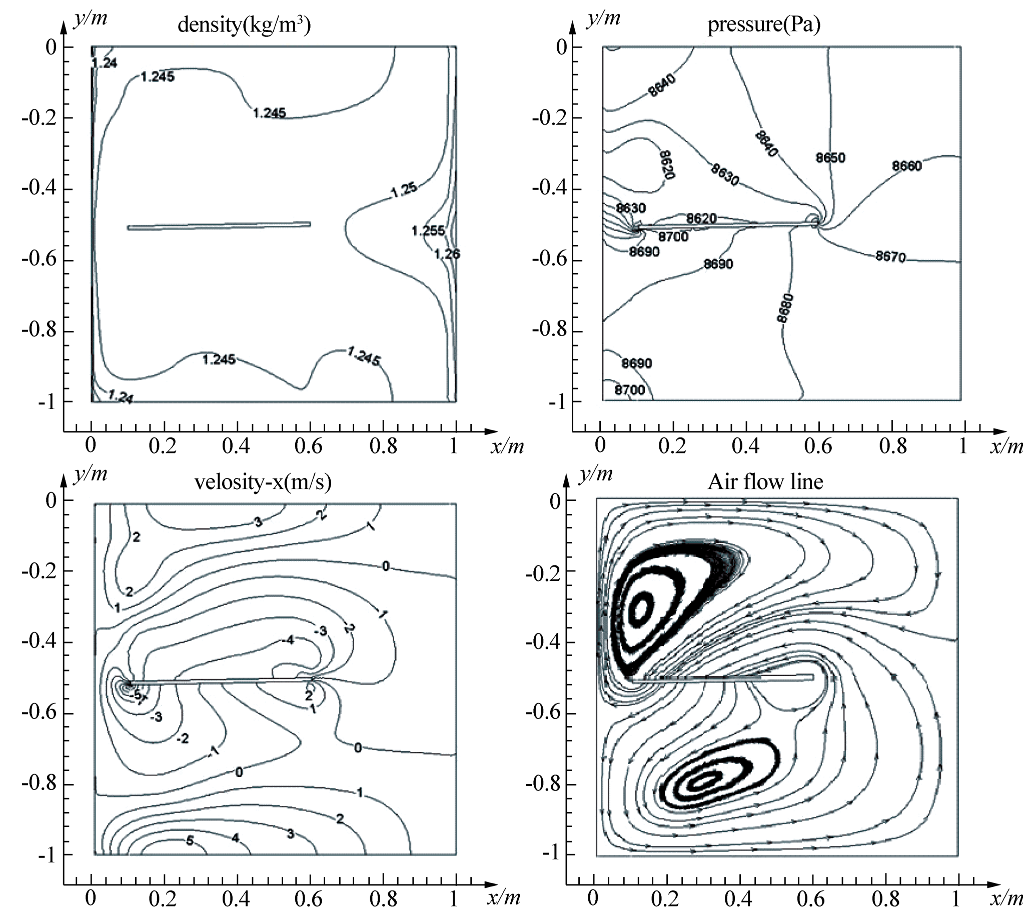

Figure 2
Distribution of the Density (a), Pressure (b), Velocity(c) and Air Flow Line(d) in Air-Side Outside the Tube

The cooling performance of the PE fan is depended on the surface heat transfer coefficient (h) at the left boundary in this model. The surface heat transfer coefficient in the air side is calculated as

$$h = -\frac{\lambda}{\Delta t}\frac{\partial t}{\partial y}\Big|_{y=0} \tag{7}$$

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where λ is the thermal conductivity of the air , $\triangle t$ is the temperature difference between the air and the tube surface and $\frac{\partial t}{\partial y}\Big|_{y=0}$ is the temperature gradient at the outside surface of the tube which could be obtained from the simulation data. To obtain the surface heat transfer coefficient of the boundary with equation (7), we take average of five points in the result. And the surface heat transfer coefficient of each point is calculation by the temperature data on each line (Line in Y= -0.2 m,Y= -0.4 m, Y= -0.5 m, Y= -0.6 m, Y= -0.8 m). The temperature data and the surface heat transfer coefficients are shown in Table 1. Then, the average $h_{average}$ is obtained as 36.2 W/(m^2·K). According to the operation parameters of the same power plant, the h of an AF fan cooling method calculated as $h=Nu\lambda/d$, is 33.51 W/(m^2·K), when the static velocity of the wind is 6.06 m/s. It comes out that the cooling performance of PE fan is better than the AF fan.

Table 1
The Temperature and Surface Heat Transfer Coefficient (h) of Five Chosen Points Along the Tube

Position	Temperature (K)	h (W/(m^2·K))
Y=-0.2m	328.11	36.223
Y=-0.4m	328.09	36.200
Y=-0.5m	328.08	36.195
Y=-0.6m	328.10	36.201
Y=-0.8 m	328.11	36.246

2.2 Heat Transfer Coefficient on the Steam Side Surface of the Cooling Tube

In the analytic solution of laminar film condensation of clean steam inside vertical tubes by Nusselt in 1916, there are the analytic solution of the surface heat transfer coefficient and the film thickness:

$$h_x = \left[\frac{g\gamma\rho_1^{\,2}\lambda_1^{\,3}}{4\mu_l(t_s - t_w)x} \right]^{1/4} \tag{8}$$

$$\delta = \left[\frac{4\mu_l\lambda_1(t_s - t_w)x}{g\gamma\rho_1^{\,2}} \right]^{1/4} \tag{9}$$

where ρ, λ, γ and g are the density, thermal conductivity, latent heat of vaporization and acceleration due to gravity, respectively. The subscript l and s indicate the liquid and steam. m_l, t_s and t_w are the dynamic viscosity of the liquid, liquid saturation temperature and surface temperature of the tube, respectively. Nusselt model is established on the condition of a steady laminar flow in a smooth condensation film and considers about only heat conduction through the film. The heat transfer depends on the surface temperature gradient and the film thickness.

With the simulation result of temperature on the left boundary, the surface heat transfer coefficient could obtain in equation (8). The result with a PE fan h is 20571.36 W/(m^2·K), which is obviously higher than that with an AF fan (12788.24 W/(m^2·K)) when the wind velocity is 6.06 m/s. According to the above, at the laminar film side the cooling effect is obviously higher by PE fan.

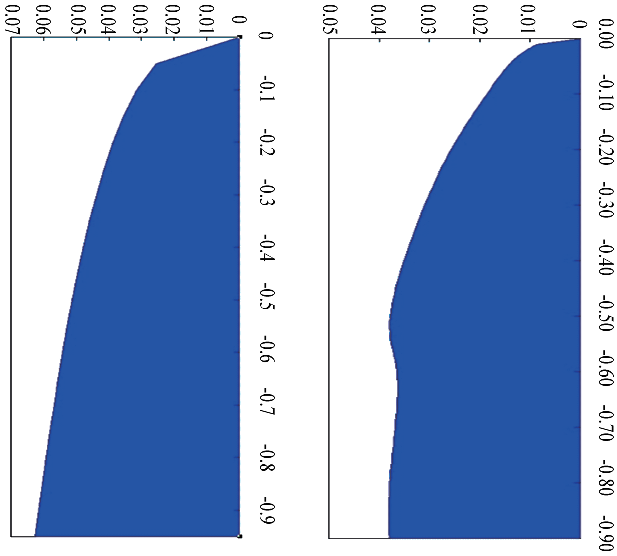

Figure 3
(a) Film Formed at the Surface of the TUBE with an AF Fan, the Wind Velocity is 6.06 m/s According to a Practical Unit; (b) Film Formed at the Surface of the Tube with a PE Fan

The laminar film flow on the steam side is not stable, and there are waves formed as the Reynolds number is higher than a critical value[8-11]. While in this situation with PE fan, Reynolds value of the film is calculated to be about 0.15 ($\mathrm{Re} = u\delta / \nu$, u is the flow velocity, taken as 4.2×10^{-3} m/s; δ is the thickness of the film, taken as 0.0395mm; v is the kinematical viscosity, taken as 1×10^{-6} m^2/s, both u and δ take the value of the bottom point of the calculation field.), which is far lower than the critical value. The film formed in front of an AF fan at the same temperature range is put on in Figure 3(a) with equation (9). As shown, the film increases smoothly on account of the homogeneous temperature distribution. In Figure 3(b) it presents the film formed at 20.013 s with a PE fan, which is obtained by the above numerical data with equation (9). Comparing with Figure 3(a), an obvious wavy film interface is observed. These further indicate that the waves are induced by the non-isothermal vertical plate.

Figure 4 shows the film vibration with the time at five different positions. The film thickness vibration is periodically, and it is caused by the PE fan periodically vibration. The recirculation caused by the waves could enhance the convection heat transfer between the vertical wall and the film, then the overall heat transfer coefficient is higher[12-15]. Therefore the performance of the tube cooled by a PF fan would be further enhanced.

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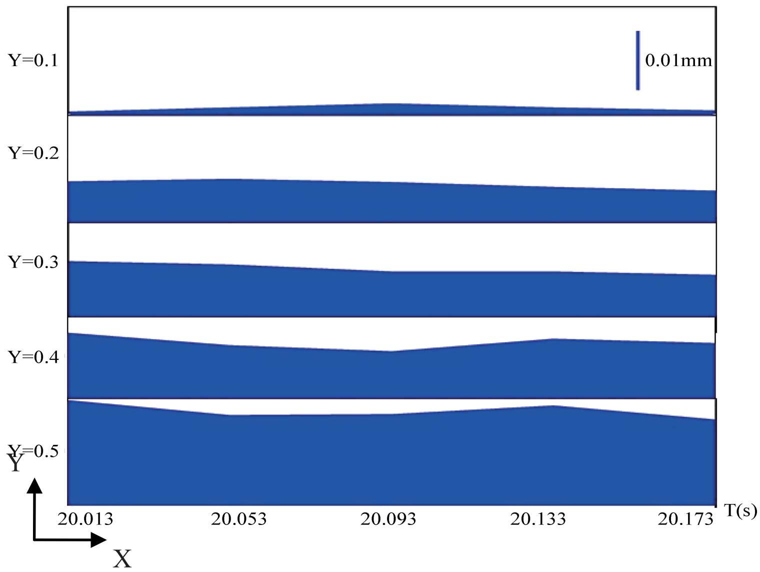

Figure 4
Oscillation of the Liquid Film in a Period is Shown. X-Axis is the Time Step, Y –Axis Presents the Position on the Vertical Surface and the Height of the Blue Region is the Thickness of the Film

CONCLUSIONS

The piezoelectric fan is introduced into the air-cooled condenser of the thermal power plant. Its cooling performance has been taken out by numerical simulation. Firstly, the surface heat transfer coefficient with PE fans is higher than that of conventional AF fans in the air side. Furthermore, the periodic vibration of PE fans can lead to a temperature oscillation on the condensation tube wall, and thus it formed an obvious wavy film in the steam side. Due to the wavy film it could enhance the heat transfer of the film condensation.

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