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Zhi Wang, Yali Liu, and Ruiming Yan

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Zhi Wang, a) Yali Liu, and Ruiming Yan

AFFILIATIONS
Department of Power Engineering, North China Electric Power University, Baoding 071003, China

a)Author to whom correspondence should be addressed: wangzhi@ncepu.edu.cn

ABSTRACT

With increasing ambient wind speed, eddy currents appear near condenser cells, affecting the thermal-flow performance of a direct air-cooled condenser. To examine the impact of ambient wind on the eddy-current distribution of an air-cooling island, a 2 × 600 MW direct air-cooling unit in North China was studied. To compare a series of different inlet wind speeds, the temperature and pressure distribution and the mechanism of and variation in eddy currents were studied. Moreover, the effects of ambient wind speed on the inlet flow rate of an axial-flow fan cluster were examined. Numerical results show that with increasing ambient speed, the chance of hot-air plume generation increases, resulting in a reduction in heat-transfer volume and efficiency. Additionally, more negative-pressure areas are created near the condenser cells. Vortices appear in the condenser-cell flow field, gradually becoming larger. The mass-flow rate of the axial flow fan cluster exhibits a downward trend with increasing ambient speed.

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I. INTRODUCTION

With rapid industrial development and increasing population size, the discharge of large quantities of sewage occurs, along with industrial wastewater violations and noncompliance emissions in the utilization of water resources. Water-resource shortages have hindered the sustainable development of many countries; therefore, it is imperative to protect existing water resources, efficiently use new water resources, and reduce water pollution. Because of their excellent water-saving properties, air-cooled condensers (ACCs) are being increasingly used in coal-rich but water-scarce areas, especially in North China.

Dozens of condenser cells are arranged in a rectangular array, and each condenser cell consists of a steam duct, finned tube bundles, and an axial flow fan. Many finned tube bundles are arranged in an A-frame to enlarge the heat-transfer area, and an axial-flow fan exists below the A-frame. After exhaust steam flows into the steam duct, it travels to the finned tube bundles, while ambient wind is drawn by the axial-flow fan through the A-frame, removing the heat of exhaust steam. Because of its exposure to ambient wind, uncontrollable and inconstant ambient conditions considerably influence the thermoflow performances of an ACC.

The effects of ambient wind on ACCs are complex, involving eddy currents and circumfluence. Over the past several years, much work has been focused on the adverse effects of ambient winds on ACCs. It is well established that adverse effects may result in poor fan performance, manifested, for example, in a low inlet-flow rate and high inlet temperature. Moreover, ambient wind might also lead to hot-air reflow and deterioration of the thermoflow characteristics of the finned tube bundles. Some researchers have focused on physical and mathematical models of air-cooled condensers under various ambient conditions, seeking solutions to challenges inherent to the air-side, vapor side, and fans. Based on historical operating data, numerous numerical studies have also been conducted to construct a semiempirical model to design and thermally assess ACCs.

To reduce the adverse effects of ambient wind on a traditional ACC and improve its cooling performance under ambient wind conditions, many scholars have optimized and reformed the fans on the steam-side and air-side. For the air-side, Butler and Grimes suggested building a model using the historical wind condition distribution to predict an optimal condenser configuration and to optimize performance. Huang et al. studied the impact of air deflectors with various geometric parameters on the thermoflow performance of
ACCs and found that the width, pitch, inclination angle, and number of air deflectors have a significant influence on heat-flow performance. Chen et al.\(^6\) proposed a novel ACC, combining V-frame condenser cells with induced axial-flow fans and found that the new construction greatly improved thermoflow performance. Jin et al.\(^7\) proposed a square array of ACCs to weaken the impact of ambient conditions; the total heat rejection of ACCs increased markedly under wind conditions compared with a conventional array. Kong et al.\(^8\) proposed a combined natural draft ACC, where the finned tube bundles were arranged horizontally in a line inside the dry-cooling tower and configured vertically in a line outside the tower. This ACC performed better than a traditional ACC at a wind speed lower than 9 m/s. Zhang and Chen\(^9\) suggested installing a wind-break mesh, which could protect the peripheral fans to mitigate the adverse impact of ambient conditions. For the steam-side, Deng et al.\(^{10}\) investigated the condensation performance of core tubes in an ACC by numerical simulation, showing that three isosectional tubes with varying patterns, from flat, to oval, to round, differently affected the heat-flow characteristics of the ACC. Deng et al.\(^{11}\) studied the sensitivity of condensation length, condensate thickness, and interfacial shear under various boundary conditions, optimizing the operation of finned tube bundles. Owen and Kröger\(^{12}\) studied the vapor flow distribution in the primary condensers and found that the definite trend in inlet-loss coefficient distribution could be applied to improve vapor-side performance.

Most studies on ACCs only analyze the influence of ambient conditions on thermal-flow performance. However, the specific mechanism of heat transfer deterioration in finned tube bundles is still not clear. Therefore, the effects of ambient conditions on the temperature, pressure, and eddy currents on an air-cooled island are studied in depth in this paper. A 2 × 600 MW direct dry cooling power plant was used as the research object. Based on a simplified structure of the ACC, a physical model of the air-side fluid and heat flow in the ACC was constructed using Fluent software. Through simulation, a variation law of the temperature, pressure, and eddy currents inside the air-cooled island with ambient wind speed was obtained. This work aids the design optimization and energy-efficient operation of an ACC in a power plant under windy conditions.

II. METHODS

A. Physical modeling

A typical 2 × 600 MW direct air-cooling unit, located in Togtoh County, Inner Mongolia Autonomous Region, North China, was investigated. The structural parameters of the unit are shown in Table I, and the performance parameters of the designed air-cooled system are shown in Table II. Figure 1 shows the average temperature and precipitation in the area where the power station was located during 1981–2018.\(^{27–29}\)

Using the power-plant data given in Table I, a geometric model of a row of condenser cells is constructed using Gambit software. Because the impact of ambient wind on the eddy-current distribution on an air-cooling island is to be studied, it is necessary to take a large physical domain to restrain the unrealistic effects of the domain boundaries on the flow field of ambient winds entering the ACC.\(^25\) Therefore, a region of x(500 m) × y(160 m) × z(200 m) is established to simulate the natural flow field, which is shown in Fig. 2. Owing to the unchanging speed and temperature profiles, the outflow boundary condition is set at the domain outlet. The symmetry conditions are set at other boundary surfaces. Under wind conditions, however, one of the vertical surrounding surfaces is set as the speed inlet, and six inlet wind speeds of 0 m/s, 3 m/s, 6 m/s, 9 m/s, 12 m/s, and 15 m/s are assigned to study wind-speed effects. Under windless conditions, the vertical surrounding surfaces of the computation domain are set as the pressure inlet, and the top surface is set as the

| Table I. Structural data of the air-cooled condenser. |
|-----------------------|-----------------------|
| Item                  | Parallel-flow | Counter-flow |
| Number of finned tubes | 608           | 64          |
| Size of the finned tube| 10 × 2.22 × 0.4 | 10 × 2.22 × 0.4 |
| Finned tube frontal area | 13 495       | 1 420       |
| Total area of the finned tube | 1 663 150 | 175 068 |

| Table II. Performance parameters of the air-cooled system under design conditions. |
|-----------------------------------------------|
| Item                                      | Symbol | Value  |
| Environmental pressure (N/m\(^2\))          | Pa     | 86 900 |
| Outdoor dry-bulb temperature (K)            | \(T_a\) | 287.15 |
| Main steam temperature (K)                  | \(T_v\) | 322    |
| Main pipe temperature (K)                   | T      | 320    |
| Air flow of the fan (kg/s)                  | G      | 428    |
| Total pressure of the fan (Pa)              | P      | 66.1   |
| Rotation speed of the fan (r/min)           | n      | 63     |
| Air mass velocity on the windward side (kg/m\(^2\) s) | \(V_n\) | 1.6    |

**FIG. 1.** Average temperature and precipitation in Togtoh County during 1981–2018.
pressure outlet. Figure 2 shows the computational domain and boundary conditions.

The complex structure of the air-cooled condenser is simplified in this paper—the steel-frame structure which plays a support role in the ACC is ignored. Moreover, the finned tube bundles are dealt with as a heat-dissipation surface, and the axial flow fan is dealt with as a pressure jump surface. A simplified schematic of the A-frame condenser cell is shown in Fig. 3(a). A row of condenser cells is arranged, as in Fig. 3(b), which shows the specification of the condenser cell serial number.

Considering available computational power, a tetrahedral unstructured mesh division is adopted to guarantee fewer meshes and higher mesh quality. The grid-interval size of the domain, far from the ACC, is taken to be 10 m, while for the condenser cells and wind-break walls, the grid interval size is set to 1 m. According to the simulation results, local encryption is applied to the condenser cells and wind-break wall to satisfy the computational demands. After a series of local refinements for the condenser cells and wind-break wall, the final total numbers of meshes are about 723,200, 938,400, and 1,562,800. The three grids' density solutions are tested for ACC thermoflow performance at a wind speed of 6 m/s. After grid-independence verification, the total-pressure ratio varies by only about 0.7% between the two highest grid numbers, which meets the calculation requirement of less than 1%. Therefore, the final grid number is about 938,400. Figure 4 shows the grid generation of the physical model.

According to the test method given in Ref. 30, numerical-result validation is conducted by comparing the experimental data with field measurements. Six temperature measuring points at the inlet of the condenser cells were randomly taken. The temperature of these six points was measured at a steady power output. Figure 5 shows that the experimental inlet air temperature agrees well with the simulated inlet air temperature. Therefore, the simplifications associated with the air-cooled condensers and the computational methods are reliable enough for the purposes of this study.
B. Mathematic models

The fluid flow around the condenser cells is three-dimensional, unsteady, and turbulent. The steady state governing equations of the air-side fluid and heat flows are as follows:

\[
\frac{\partial p}{\partial t} + \frac{\partial \rho v_i}{\partial x_i} = 0,
\]

\[
\rho \frac{dv_i}{dt} = -\frac{\partial P}{\partial x_i} - \frac{2}{3} \frac{\partial}{\partial x_i} \left( \rho \frac{\partial v_k}{\partial x_k} \right) + \rho f_i + S_i
\]

\( i, j, k = 1, 2, 3, \text{ and } i \neq j),

\[
\rho \frac{d}{dt} \left( c + \frac{1}{2} v_i v_i \right) = \rho f_i v_i + \frac{\partial}{\partial x_i} \left( \tau_{ij} v_j \right) + \frac{\partial}{\partial x_i} \left( \frac{k}{\rho} \frac{\partial T}{\partial x_i} \right) + S_i,
\]

where \( v_i \) is the speed in the \( x_i \) direction, \( \rho \) is the density, \( P \) is the pressure, \( f_i \) is the mass force in the \( x_i \) direction, \( \mu \) is the effective dynamic viscosity, \( \epsilon \) is the thermodynamic energy, \( k \) is the thermal conductivity, \( T \) is the temperature of the fluid, \( q \) is the amount of heat transmitted to a fluid of unit mass per unit time due to radiation, and \( \tau_{ij} \) is the stress tensor in the \( x_i \) and \( x_j \) direction.

The \( k-\epsilon \) model was first proposed by Launder and Spalding, \( ^{32} \) after which, considerable research was devoted to enhancing the model's accuracy and robustness. Today, the \( k-\epsilon \) model is used widely in industrial-flow and heat-transfer simulations, and the standard \( k-\epsilon \) model is applied to simulate and calculate turbulence in this paper.

\[
\frac{\partial \epsilon}{\partial t} + \nabla \cdot (\epsilon \mathbf{v}) = \nabla \cdot \left( \frac{\tau_{ij}}{\rho} \right) + \frac{\epsilon}{\rho} \left( c_v + \frac{\tau_{ij}}{k} \frac{\partial T}{\partial x_j} \right) - C_{\epsilon 1} \frac{\epsilon^2}{k} + C_{\epsilon 2} \frac{\tau_{ij}}{k} \frac{\partial T}{\partial x_j} - C_{\epsilon 3} \frac{k^{\prime 2}}{\epsilon^{\prime 2}} + S_{\epsilon},
\]

where \( k \) is the turbulent kinetic energy, \( \epsilon \) is its rate of dissipation, and \( C_{\epsilon 1} \) and \( C_{\epsilon 2} \) are empirical coefficients.

C. Boundary condition

The windward surface of the computational domain is set as the boundary condition of the speed inlet, and the natural wind speed, \( u_i \), is described by the following power-law equation:

\[
u_i = u_0 \times \left( \frac{z_i}{z_0} \right)^{m},
\]

where \( u_0 \) is the average wind speed at the height of the uniform flow in m/s, \( u_i \) is the average wind speed at a height of \( z_i \) in m/s, \( z_0 \) is the height of point \( i \) in meters, \( z_0 \) is the height of uniform flow in meters, and the exponent \( m \) is the ground roughness. In addition, six wind speeds of 0 m/s, 3 m/s, 6 m/s, 9 m/s, 12 m/s, and 15 m/s are assigned to \( u_0 \) to study wind speed effects. By referring to a relevant standard, \( ^{33} \) the exponent \( m \) is taken as 0.16 in this paper. Within the studied altitude range, variation in wind temperature with altitude can be neglected, so the average wind temperature is taken as 287.15 K.

The fan is simplified as a pressure jumping surface. Using Fluent software, the air flow at the entrance of the fan is simulated by setting the pressure head of the fan to vary with the flow rate. The pressure rise is expressed as follows:

\[
\Delta \rho = \sum_{n=1}^{N} f_n \times v_z^{n-1},
\]

where \( \Delta \rho \) is the pressure rise in Pa; \( v_z \) is the axial speed in m/s; and \( f_n \) is a polynomial coefficient. According to the performance curve of the fan, the polynomial coefficients are \( f_{\text{fan 1}} = 145.81, \text{fan 2} = -5.76, \) and \( f_1 = -0.82. \)

III. RESULTS AND DISCUSSION

The designed ambient temperature in China is commonly taken to be about 287.15 K, so this is the temperature at which the thermal performance of the ACCs is investigated; wind speed ranges from 0 m/s to 15 m/s in this paper.

A. Temperature fields

Figure 6 shows that under the impact of differences in density caused by temperature differences, hot-air plumes discharge from the finned tube bundles, flowing into the ambient air. According to the temperature contours of the ACC, the air-temperature...
field around the condenser cells is uniform under windless conditions. Figures 6(b)–6(f) show that the direction of the hot-air plume shifts to the ambient wind direction. The larger the ambient speed, the greater the inclination of the hot-air plume, which greatly hinders heat dissipation by the finned tubes. In addition, the high-temperature field area decreases gradually and is concentrated around the condenser cells, also indicating that fin-heat dissipation is adversely affected. Moreover, compared with the temperature distribution of a1 shown in Fig. 7, the temperature in the A-frame increases rapidly when the ambient speed is greater than 6 m/s.
At low wind speed, the wind effect is weak, and the hot plume can be easily discharged. Therefore, the finned tube bundles are cooled efficiently by air, and the temperature distributions of different condenser cells are very similar. However, at high wind speed, the hot-plume discharge from the finned tube bundles is critically restrained by the ambient wind. Cooling air cannot flow through the finned tube bundles, especially at condenser cell a1, so condenser-cell temperature is very high.

B. Pressure fields

Figure 8 shows that when the ambient speed reaches 3 m/s, a negative-pressure area appears at condenser a1. As wind speed increases, the negative pressure area increases, and the number of negative-pressure condenser cells increases. Figure 9 shows that the pressure distribution in the inner region of the A-frame becomes more and more uneven, with negative pressure on the left side and positive pressure on the right side. When the wind speed reaches 15 m/s, almost all condenser cells, a1s, are occupied by a negative-pressure region. The uneven pressure distribution in the A-frame affects inhalation by the fan, which makes it difficult to dissipate heat from the finned tube; the higher the air speed, the lower the pressure. At high wind speed, the pressure on both sides is lower, and, therefore, the region near the condenser cells has a negative pressure, which increases with increasing wind speed.

C. Flow fields

Figure 10(a) shows that, under windless conditions, ambient wind is sucked perpendicularly by the axial flow fan cluster. With increasing wind speed, the inlet streamlines become more tilted, and the inlet-flow condition of the fans deteriorates. At low ambient speed, a streamline is very regular, showing that the hot plume is very easily discharged. When the wind speed reaches 6 m/s, the streamline becomes increasingly unsmooth, and the inhomogeneity of the flow field increases, while as shown in Fig. 11(b), the dead zone of the flow appears in the A-frame of a1. The thermoflow performance of the condenser cell critically deteriorates. When the ambient speed reaches 9 m/s, a vortex appears in the inner part of the A-frame, which critically affects the inhalation capacity of the fan; vortices gradually expand with increasing wind speed. When the wind speed reaches 15 m/s, vortices appear in several A-frames, and the distribution of the flow field becomes very complex. At high wind speed, the hot-air plume is tilted critically by the ambient wind. Therefore, the streamline inclined to the direction of the ambient wind and the angle of incline increase with increasing wind speed. Owing to the negative pressure in the A-frame, it is very difficult for the axial fan to suck ambient wind. The negative region is concentrated on the left side of the A-frame, so the streamlines flow toward the right side of the A-frame; moreover, eddy currents are present at the left side of the A-frame.

When vortices appear in the A frame, the hot plume is not easily discharged, and the suction ability of the fan deteriorates. More importantly, the hot air that has participated in heat transfer will likely flow back to the fan inlet and reparticipate in heat transfer. Consequently, the fan inlet temperature increases. The cooling ability of the condenser cells is now critically reduced. According to hydrodynamic studies, air flow is usually turbulent, and when passing obstacles, vortices are created, forming a negative pressure center. The pressure difference between the top and bottom of the

![Pressure contours vs wind speed.](image-url)
condenser cell is intensified by strong suction of the fan. When wind speed is high, hot air, which has participated in heat transfer, is sucked into the fan again due to the inlet inertia force. Hot-plume recirculation flow results in a significant increase in the inlet temperature of the fan, thus deteriorating heat transfer.

The rotation flow, boundary-layer flow under strong adverse pressure gradients, separation, and recirculation flow can be predicted accurately by the k-ε model. A three-dimensional vortex structure is shown in Fig. 12, in which the pressure distribution is uneven in the A-frame. Under the influence of uneven pressure, air...
Flow rotates around its instantaneous axis in the process of motion, forming eddy currents. The larger the pressure difference and the more uneven the pressure distribution, the higher the likelihood of vortex formation, creating the streamline distribution, as shown in Fig. 10.

D. Fan performance

To better understand the impact of wind on fan performance, Fig. 13 shows the mass-flow rate at wind speeds of 0 m/s, 3 m/s, 6 m/s, 9 m/s, 12 m/s, and 15 m/s. With increasing wind speed, the fan mass-flow rate shows an overall downward trend, and the mass-flow rate of fan 1 clearly decreases; however, the mass-flow rates of other fans decrease only slightly.

To analyze the air inflow of a fan under different ambient velocities, the mass-flow rate of fan 1 was studied separately. Figure 14 shows that the mass flow rate of fan 1 decreases by 8.05% at a wind speed of 3 m/s, by 9.00% at 6 m/s, by 14.84% at 9 m/s, by 21.94% at 12 m/s, and by 31.90% at 15 m/s. When the ambient speed is 0 m/s, the mass-flow rate of fan 1 is 480.59 kg/s, and when the ambient speed is 15 m/s, the mass-flow rate of fan 1 is 182.05 kg/s; the mass flow rate of fan 1 decreases overall by 62.12%. In contrast, the mass-flow rate of fan 2 decreases by 3.64% at a wind speed of 3 m/s, by 0.55% at 6 m/s, by 0.18% at 9 m/s, by 1.49% at 12 m/s, and by 0.58% at 15 m/s. When the ambient speed is 0 m/s, the mass-flow rate of fan 2 is 477.13 kg/s, and when the ambient speed is 15 m/s, the mass-flow rate of fan 2 is 446.98 kg/s; the mass-flow rate of fan 2 decreases overall by 6.32%. Comparing the mass-flow rates of fan 1 and fan 2 shows that with increasing ambient speed, the mass-flow rate of fan 1 decreases rapidly, critically negatively affecting the performance of fan 1. Moreover, the cooling capability of the ambient wind seriously deteriorates owing to the sharp decrease in fan performance; the finned tube bundles now cannot be cooled by the cooling air. Therefore, the small mass-flow rate and the high inlet air temperature are the main reasons for the decrease in heat rejection. In order to mitigate the adverse effects of wind and to optimize the operation of the air-cooled island, the upwind axial-flow fan should be equipped with a relatively large capacity motor.

FIG. 11. Streamlines of condenser cell a vs wind speed.

FIG. 12. 3-dimensional vortex structure of the condenser cells.

FIG. 13. Mass flow rate of the axial flow fan vs wind speed.
IV. CONCLUSIONS

Natural wind speed has an important influence on the temperature, pressure, and flow field of an air-cooled island. Fluent software was used to construct a mathematical model of an air-cooled island flow field and simulate its temperature field, pressure field, and streamline chart. Through a comparative analysis, the following conclusions are drawn:

(1) When ambient speed increases, the hot-air plume in the upper part of the condenser cells deflects to the wind direction, and heat transfer from the surrounding fluid decreases, which is not conducive to heat dissipation by the fins.

(2) The pressure field on the air-cooled island is closely related to ambient speed. The pressure distribution inside the A-frame becomes increasingly uneven with increasing wind speed, which seriously affects the operating performance of the axial-flow fan.

(3) The lower the ambient speed, the more uniform the airflow field around the air-cooled unit. With increasing natural wind speed, vortices appear in the interior of several A-frames, expanding gradually.

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