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# Space characteristics of the thermal performance for air-cooled condensers at ambient winds

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ABSTRACT

Ambient winds may lead to poor fan performance, exhaust air recirculation and mal-distribution of the air across the tube bundles of the air-cooled condensers in a power plant. Investigations of the impacts of the ambient winds on the air-cooled condensers are key area of focus. Based on a representative $2 \times 600$ MW direct dry cooling power plant, the physical and mathematical models of the air-side fluid and heat flow in the air-cooled condensers at various ambient wind speeds and directions are set up by introducing the radiator model to the fin-tube bundles. The volumetric flow rate, inlet air temperature and heat rejection for different air-cooled condensers as a whole, condenser cells and fin-tube bundles are obtained by using CFD simulation. The results show that the thermo-flow performances for the air-cooled condenser as a whole, condenser cells and heat exchanger bundles vary widely in space. The thermal performances of the air-cooled condensers, condenser cells and fin-tube bundles at the downstream are generally superior to those at the upwind. It is of use for the upwind fan regulations and the A-frame condenser cell geometric optimization to investigate the space characteristics of the thermal performance for the air-cooled condensers in a power plant.



## 1. Introduction

The application and popularity of air-cooled condensers (ACCs) is increasing in the power plants throughout the countries that are short of water resources during the last decade [1,2]. Due to the direct utilization of ambient winds as the cooling medium of the exhaust steam from turbine, the environment conditions such as the ambient temperature, wind speeds and directions will play important roles in the thermal performances of air-cooled condensers. Many researches have found that air-cooled condensers work fairly bad in a wide range of specific climate, especially with large wind speeds and adverse wind directions.

van Rooyen and Kroger [3] numerically studied the air flow field about and through a particular air-cooled condenser, in which the performance of the fan is modeled with the aid of the numerical approach – actuator disc model. Hotchkiss et al. [4] studied the effects of the cross flow on the performance of the axial flow fans in air-cooled condensers by using the actuator disk fan model. Duvenhage and Kroger [5] studied the influence of wind on the fan performance and exhaust plume recirculation in air-cooled condensers by using the CFD simulation and found that cross winds significantly reduce the air flow rate in the upwind condenser cells and the winds along the longitudinal axis cause increased hot plume recirculation. Wang et al. [6] investigated the overall velocity and temperature fields of the air in a power plant by numerical simulation and discovered that the plume recirculation always occurs due to the wind effect and the suction of the fan. Installing a side board below or above the fan platform was suggested to avoid the plume recirculation. Gu et al. [7,8] investigated the efficiency of the air-cooled condensers at different wind speeds and directions for a certain power plant by using the wind tunnel simulation and discussed the criteria as well as the methods and measurements of wind tunnel simulation for wind effects. Meyer and Kroger [9] revealed the effects of the fan and heat exchanger characteristics as well as the plenum chamber geometry on the flow losses of air-cooled condensers by experimental investigation. They [10] also investigated the air-cooled heat exchanger plenum chamber aerodynamic behavior for different fan performances by using CFD simulation. Duvenhage et al. [11] studied the fan performance in air-cooled condensers due to the inlet flow distortions numerically and experimentally. Meyer [12] also numerically investigated the effect of inlet flow distortions and found that the inlet flow losses of the periphery fan are dominated by the flow separation occurring around the inlet lip of the fan inlet section. These flow losses can be reduced by the installation of a walkway at the edge of the fan platform or by the removal of the

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**Nomenclature**

| | |
|---|---|
| $C$ | $k$–$\varepsilon$ model constant |
| $e$ | specific internal energy (J/kg) |
| $f_n$ | polynomial coefficient of the pressure rise of the fan |
| $g$ | gravitational acceleration (m/s$^2$) |
| $G$ | turbulence kinetic energy generation (m$^2$/s$^2$) |
| $g_n$ | polynomial coefficient of the tangential velocity of the fan |
| $h$ | air-side convection heat transfer coefficient (W/(m$^2$ K)) |
| $h_n$ | polynomial coefficients of the convection heat transfer coefficient |
| $h_e$ | specific enthalpy (J/kg) |
| $k$ | turbulence kinetic energy (m$^2$/s$^2$) |
| $k_L$ | non-dimensional loss coefficient |
| $m$ | exponent of the wind speed in the power-law equation |
| $p$ | pressure (Pa) |
| $q$ | heat flux (W/m$^2$) |
| $Q_v$ | volumetric flow rate (m$^3$/s) |
| $r$ | radial distance from the fan center (m) |
| $r_n$ | polynomial coefficient of non-dimensional loss coefficient |
| $s_h$ | heat source (W/m$^3$) |
| $s_i$ | momentum source (N/m$^3$) |
| $T$ | temperature (K) |
| $u_i$ | air velocity (m/s) |
| $v$ | component of velocity (m/s) |
| $x_i$ | Cartesian coordinate (m) |
| $z$ | height above the ground (m) |

*Greek symbols*

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| $\rho$ | air density (kg/m$^3$) |
| $\mu$ | air dynamic viscosity (kg/(m s)) |
| $\lambda$ | air thermal conductivity (W/(m K)) |
| $\tau$ | stress tensor (J/m$^3$) |
| $\sigma$ | turbulent Prandtl number |
| $\varepsilon$ | turbulence kinetic energy dissipation rate (m$^2$/s$^3$) |
| $\Phi$ | heat rejection (W) |

*Subscripts*

| | |
|---|---|
| $w$ | wind |
| $a$ | air |

periphery fan inlet section. Bredell et al. [13] investigated the effect of the inlet flow distortions on the flow rate through the fans numerically. The volumetric effectiveness of two different types of axial flow fans at different platform heights was considered and the results showed that the addition of a walkway can significantly increase the flow rate through the fans located near the edge of the fan platform. Meyer and Kroger [14] studied the influence of the air-cooled heat exchanger geometry on the inlet air flow losses by experimental investigation. An equation based on the experimental results was formulated to calculate the heat exchanger inlet air flow losses.

The impacts of ambient winds upon the ACC performance have already been thoroughly investigated by the above mentioned research. But these works mainly focus on the whole ACC performance and more emphasis is placed on the flow characteristics and wind-induced hot air recirculation. In the large-scale modern power plant, the tremendous thermal duty, that is the amount of heat to be rejected by air-cooled condensers should be matched or exceeded by the ACC's designing heat capacity. Investigation on the heat rejection as well as the flow rate of the ACC as a whole, the condenser cell, and even the heat exchanger fin-tube bundles at various wind speeds and directions is very important to clarify the reduced heat capacity due to the ambient winds. In this paper, particular attentions are paid on the thermal performance distributions for the ACC as a whole, condenser cells and tube bundles. It can be as a basis for the optimal design and operation improvement of the direct dry cooling system in a power plant.

## 2. Computational models

To better investigate the space characteristics of the thermal performance for air-cooled condensers at various ambient wind speeds and directions by CFD simulation, the physical domain should be large enough to eliminate the near-wall effect of the main buildings on the flow field of ambient winds entering the ACC. In doing so, the size from several kilometers to ten kilometers is commonly taken for the computational domain. However, in the fin-tube bundles of the air-cooled condenser, the minimum size is only about 2 mm for the fin space. The model of interest covers a multi-scale range from millimeters to kilometers with the order of $10^6$ spans. Simplifications should be made to deal with this kind of multi-scale phenomenon.

In this paper, the fin-tube banks in the air-cooled condenser are dealt with a lumped-parameter radiator model, in which the fin-tube bundles are considered to be infinitesimally thin. The fan is simplified as a pressure jump surface. Furthermore, only the turbine house, boiler house and chimney that are relatively near to the air-cooled condensers are taken into account, other buildings are neglected. The layout of the air-cooled condensers and main buildings in a typical $2 \times 600$ MW direct dry cooling power plant is schematically shown in Fig. 1. An air-cooled condenser consists of an array of A-frame condenser cells each fitted with an axial flow fan as shown in Fig. 2(b). There are two ACCs in this power plant, with each ACC having 56 ($7 \times 8$) condenser cells. Along the $x$ direction, the left ACC is designated as No. 1 ACC, and on the right is No. 2. The specification for the condenser cell serial number is also shown in Fig. 2(a).

Fig. 3 shows the computational domain with the size of $2240 \times 2240 \times 720$ m ($x \times y \times z$). On account of the structure symmetry of the ACC and the main buildings in the power plant, only half of the wind directions are considered. Five wind direction angles, $-90°$ ($y$ direction), $-45°$, $0°$ ($-x$ direction), $45°$, $90°$ ($-y$ direction), are schematically shown in Fig. 3 to investigate the influence of the wind direction upon the ACC performance. The computational meshes are generated with the commercial software Gambit using the multi-block hybrid approach. For the central domain with the air-cooled condensers and main buildings, the tetrahedral unstructured grid is used. For other zones, hexahedral structured grid is adopted. Grids consisting of about 1,200,000, 2,125,000, and 3,100,000 cells are tested for the ACC performance at the wind speed of 9 m/s and direction angle of $0°$. The overall volumetric flow rate and mass-weight average inlet air temperature of the No. 1 and No. 2 ACC are observed to vary by only about 0.9% between the two highest grid density solutions. So the final grids number used is about 2,125,000.

For the radiator model, the pressure loss through the fin-tube bank is assumed to be proportional to the dynamic head of the