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Optimal design of large scale dry cooling tower with consideration of off-design operation

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Abstract

Numerous factors can affect the operating performances and the design of the indirect air cooling system of power plant. The present study developes physico-mathematical model to describe the thermo-flow characteristics of air cooling tower for indirect air cooling system. Based on the model, a comprehensive analysis on optimization of air cooling tower is conducted for 600MW indirect air-cooled power generating unit. By using the software VC++, the indirect air-cooled tower optimization program is developed. With the help of optimization of tower structure, a tower with better structure is used to conduct thermal analysis of the influences of ambient temperature, wind speed, and saturated exhaust flow rate on back pressure of turbine. The present study may be of great value on optimization design and safe operation of large-scale indirect air-cooled power plant.

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Keywords: dry cooling tower, indirect air cooling technology, off-design operation, flow anf heat transfer, design optimization

1. Introduction

Due to lower coal consumption and more stable operation than that of direct air cooling system, indirect air cooling system is widely used in the coal-fired power plant most recently [1]. Air cooling units use ambient air as cooling medium, thus the external environment has a significant effect on its performance. Wei and Al-Waked et al. [2, 3] studied the influences of natural wind on the cooling efficiency of dry-cooling tower. Yang et al. [4] presented the flow and temperature fields of cooling air by CFD simulation. Zhang et al. [5, 6] analyzed the factors that affect the running performance of condenser pressure. Bu et al. [7] set up the calculation model and numerical method on variable condition for direct

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Nomenclature

D_2	outlet diameter of tower	ΣP_a	resistance of air flowing through air cooling tower (kPa)
G	circulating water flow rate (t h^{-1})	ΔP_H	pumping force of air cooling tower(kPa)
H_e	effective height of cooling tower(m)	P_s	condenser backpressure(kPa)
K	the overall heat transfer coefficient (W m^{-2} ℃$^{-1}$)	t_a	ambient temperature(℃)
m_v	saturated exhaust flow rate (t h^{-1})	t_s	condenser exhaust temperature(℃)
n	radiator cooling delta number	t_{w2}	outlet temperature of cooling water (℃)
ΔP_1	resistance of air flowing through radiators and shutters (kPa)	v	wind speed(m s^{-1})
		v_{a2}	tower outlet air velocity (m s^{-1})
ΔP_2	resistance of air flowing through tower outlet (kPa)	v_f	windward velocity (m s^{-1})
		η	efficiency of air-cooled radiator

and indirect air cooling unit. Zhou and Jiang et al. [8, 9] conducted large number of calculation and analysis on variable condition features for air-cooled condenser.

However, the indirect air cooling system is more complex and has more factors influencing the operation of power generating unit. Hence, it requires more considerations in the system design. Based on a 600MW indirect air cooling unit, this paper analyzed the influence of tower change on safety and economy of the unit, and studied the effect on back pressure generated by m_v, v and T_a, etc.

2. Physical and mathematical model

In the indirect air cooling system , the heat exchange between exhaust steam and air is divided into two parts: the first one for steam and cooling water is conducted in the condenser; the second one for cooling water and air is conducted in the cooling tower.

The thermodynamic characteristics model is shown as follows [10]:

$$K = \frac{a_a a_w}{a_a + a_w} \tag{1}$$

$$\eta = \frac{1-\exp[NTU(\frac{W_1}{W_2}-1)]}{1-\frac{W_1}{W_2}\exp[NTU(\frac{W_1}{W_2}-1)]} \tag{2}$$

where K *is* the overall heat transfer coefficient, a_a is air-side heat transfer coefficient, a_w is water-side heat transfer coefficient; η *is* efficiency of air-cooled radiator, *NTU* is number of heat transfer unit, W_1 is water equivalent of air, W_2 is water equivalent of circulating cooling water.

The resistance characteristics model is shown as follows [10]:

$$\Delta P_H = H_e(\rho_{a1} - \rho_{a2})g \tag{3}$$

$$\sum \Delta P_a = \Delta P_1 + \Delta P_2 \tag{4}$$

whereΔP_H is pumping force of tower, He is effective height of tower, ρ_{a1} is inlet air density , $\rho_{a2}$is oulet air density,ΣP_a is resistance of air flowing through towerΔP_1 isresistance of air flowing through radiators and shutters, ΔP_2 is resistance of air flowing through tower outlet.

In engineering, if $\left(\Delta P_H - \sum \Delta P_a\right)/\sum \Delta P_a > 0.005$, the selected tower is qualified.

For a particular unit, if design Q, design t_a, ITD and radiator type are determined, according to the thermodynamic characteristics model, we can obtain a qualified n. Then according to the resistance

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