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Research Paper

Entransy dissipation based optimization of a large-scale dry cooling system

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HIGHLIGHTS

- Correlation between tower loss and geometric parameters of dry cooling tower is investigated.
- Entransy dissipation is applied to dry cooling system as constraints of optimization.
- The minimization of annual cost is taken as the optimal objective with Lagrange multiplier method.
- Total annual cost of dry cooling system can be deduced significantly by optimization.

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ABSTRACT

The natural draft dry cooling tower (NDDCT) with heat exchangers accounts for about 10 percent of the total investment of a power plant, so it is meaningful to optimize the geometric parameters to explore the potential that matching the cooling load of a power generating unit in the most cost effective way. The entransy dissipation equations of two typical irreversible heat transfer processes, including that the ambient air heated by heat exchangers and circulating water heated by the exhausted steam of turbine, are derived in this paper. Combined with the force balance equation of NDDCT, all parameters that may affect the performance and annual cost of a cooling system are took into consideration. Such constituted constraints are applied to the Lagrange multiplier optimization of such a system. Based on the mathematical relation and conditional extremum method, an optimization equation group aiming to obtain the minimum of total annual cost is constructed. Finally, a practical 600 MW power generating unit is taken as an example to illustrate the applications of entransy dissipation based optimization principle. The result indicates that total annual cost of the dry cooling system can drop from $6 420 643 to $5 665 660.

1. Introduction

Dry cooling that depends on natural convection and uses air as the cooling medium is preferable under certain conditions, especially for the insufficient water supplies in arid areas [1]. The high capital and operating costs of cooling system make it necessary to optimize the cost-performance trade-off.

Natural draft dry cooling tower (NDDCT) plays the most important role in the indirect dry cooling system. Since airflow in dry cooling systems is accomplished by natural draft, numerous investigations have concentrated on the influences of geometric parameters of (NDDCT) [2–5]. Huang et al. [2] investigated the cooling performance of NDDCT with variations of geometric parameters via theoretical prediction and experiments. Du Preez et al. [3] made use of scale model tests to study the flow characteristics inside NDDCT and regarded the overall loss dependent on structure. The 17th IAHR international conference on cooling tower [4] gave a brief review of the latest developments in cooling technologies, including the influence of geometric parameters. Jacques du Plessis [5] conducted experimental work and CFD models to develop novel collection trough and basin system design, and empirical loss coefficient relations were presented and discussed.

A lager tower brings more cooling air passing through the heat exchangers and improves the cooling performance with an increasing investment cost yet. Hence the optimal objective is to match the cooling load of a power plant in the most cost effective way. Some researchers have addressed this kind of problem with a detail economic analysis [6–12]. Cui et al. [6] gave a concrete analysis on the relation of total annual cost and different combination

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Nomenclature

A	area (m^2)
C	constant in turbulence model
C_{annual}	annual cost ($)
C_{ec}	electricity cost ($)
C_{he}	heat exchanger cost ($)
C_m	maintenance cost ($)
C_o	operating cost ($)
C_p	pump cost ($)
C_t	tower cost ($)
C_{uhe}	unit cost of heat exchanger ($\$ m^{-2}$)
C_{up}	unit cost of pump ($)
c_p	specific heat ($J kg^{-1} K^{-1}$)
d_5	outlet diameter of tower (m)
d_t	throat diameter of tower (m)
d_0	base diameter of tower (m)
g	gravitational acceleration ($m s^{-2}$)
H_{pump}	pumping head (m)
h_{he}	heat transfer coefficient of the heat exchanger ($W m^{-2} K^{-1}$)
h_c	heat transfer coefficient of the condenser ($W m^{-2} K^{-1}$)
h_5	tower height (m)
h_t	throat height of tower (m)
h_4	inlet height of tower (m)
k	flow loss coefficient
M	molecular
m	mass flow rate ($kg s^{-1}$)
$m_a c_{pa}$	heat capacity rate of air ($J s^{-1} K^{-1}$)
$m_w c_{pw}$	heat capacity rate of water ($J s^{-1} K^{-1}$)
n	number
p	pressure (Pa)
P_{wi}	electricity power for single pump (W)
Q	heat flux ($W m^{-2}$)
r_n	polynomial coefficient of non-dimensional loss coefficient
s_h	heat source ($W m^{-3}$)
S	source term
t	temperature (K)
t_{out}	downstream air temperature of the heat exchangers (K)
T_r	temperature of radiator model (K)
T_s	condensation temperature (K)
v_j	component of velocity ($m s^{-1}$)
v	velocity magnitude ($m s^{-1}$)
n_{pump}	number of pump used, represents

Greek symbols

ε	turbulence dissipation rate ($m^2 s^{-3}$)
μ	dynamic viscosity ($kg m^{-1} s^{-1}$)
ν	kinetic viscosity ($m^2 s^{-1}$)
θ	dimensionless water temperature difference
ρ	density ($kg m^{-3}$)
Γ	diffusion coefficient ($kg m^{-1} s^{-1}$)
τ	number of operating hours (h)
Φ	heat rejection (W)
η_p	efficiency of pump
φ	scalar variable

Subscripts

a	air
ave	average
b	base
c	condenser
fr	front
he	heat exchanger
il	inlet louver
o	outer
op	operating pressure
r	radiator
t	tower
ts	tower supports
to	tower outlet
T	turbulence
w	water

of an indirect cooling system. In their analysis, invest costs and operating costs were converted in a fixed number of service years. Conradie et al. [7] applied sequential quadratic programming (SQP) method to obtain the optimum cost in performance of a dry cooling system. Zou et al. [8] developed a cost model for the solar enhanced natural draft dry cooling tower, and the optimum selection of cooling tower height, heat exchanger areas and sunroof diameter were obtained. Then, different sunroof and tower shape design, namely horizontal sunroof and parabolic tower or titled sunroof plus cylindrical tower were explored further [9]. Xu et al. [10] conducted a comparison about the annual cost of a typical lignite pre-drying power plant and a novel lignite pre-drying system. In the research of Kloppers et al. [11], Wet-Cooling Tower Performance Evaluation software in conjunction with the Leap-frog Optimization Program was developed, and the geometric optimization of a natural draft wet-cooling tower was investigated by the software. Doodman et al. [12] applied global sensitivity analysis and harmony search algorithm to obtain the minimum total annual cost.

Literature review found that the previous investigations usually listed several possible combinations of the structural and operating parameters, then estimated their influences on the cost, and finally chose a better, but not the best values. On the other hand, a new optimization method that combines the Lagrange multiplier method and a new physical quantity, entransy, was proposed [13]. Taking entransy dissipation as the optimization criteria, the method can be applied in various heat transfer modes, including that of heat conduction [14,15], heat convection [16,17], thermal radiation [18,19] and specially, heat exchangers similar to air cooling systems in power plants [20–22]. It can be found that the combination can reduce the number of variables and equations, moreover, simplify the Lagrange multiplier method calculations [23,24].

Based on the above-mentioned method, a global optimal design approach for the dry cooling system of the power generating unit is proposed in this paper. A modified draft equation of NDDCT and entransy balance equations of an indirect cooling system are firstly obtained. Combined the structural requirements, a mathematical model is established by the conditional extremum method, which takes all the parameters that may affect capital and operating costs into consideration. By solving all the equations, the minimum annual cost of mathematical meaning is obtained comparing with the "try-and-error" method.

2. Physical model and control equations for indirect air cooling tower design

According to a typical dry cooling power generating unit, the physical model of dry cooling system including a hyperbolic tower with vertical heat exchanger bundles is shown in Fig. 1.

The stagnant ambient air at the surrounding, 1, driven by the suction force generated by the temperature difference of the inlet