Effects of Acoustic Barriers and Crosswind on the Operating Performance of Evaporative Cooling Tower Groups (Postprint)

Authors: Dang, C.,, Jia, L., Yang, L. X.

Date: 2017-11-02T00:00:00+00:00

Abstract

Most evaporative cooling towers are arranged on building roofs due to space and noise limitations, and acoustic barriers are typically installed around cooling towers in practical applications. The presence of acoustic barriers and crosswind may affect the recirculation phenomenon, which is directly related to the operating performance of cooling towers. In this study, a physical and mathematical computation model is proposed to investigate the effects of crosswind and the distance between acoustic barriers and cooling tower inlets. Both sensible and latent heat are considered in this research. The reflux flow rate and performance ratio are obtained to evaluate the recirculation and operating performance, respectively. The results demonstrate that higher crosswind velocities lead to larger reflux flow rates and lower performance ratios for cooling tower groups. Under high crosswind velocity conditions, the presence of acoustic barriers is effective in inhibiting reflux and improving operating performance, particularly for ICE cooling tower groups. Additionally, optimum values are recommended for LiBr/ICE cooling tower groups in the research cases. The variation of reflux flow rate and performance ratio with acoustic barrier distance exhibits a parabolic trend.

Full Text

Preamble

Effects of Acoustic Barriers and Crosswind on the Operating Performance of Evaporative Cooling Tower Groups

Chao Dang[1],[2], Li Jia[1],[2],[*], Lixin Yang[1],[2]

[1] Institute of Thermal Engineering, School of Mechanical, Electronic and Control Engineering, Beijing Jiaotong University, Beijing 100044, China

[2] Beijing Key Laboratory of Flow and Heat Transfer of Phase Changing in Micro and Small Scale, Beijing 100044, China

Most evaporative cooling towers are arranged on building roofs due to space and noise limitations, and acoustic barriers are typically installed around cooling towers in practical applications. The presence of acoustic barriers and crosswind may affect the recirculation phenomenon, which is directly related to the operating performance of cooling towers. In this study, a physical and mathematical computation model is proposed to investigate the effects of crosswind and the distance between acoustic barriers and cooling tower inlets. Both sensible and latent heat are considered in this research. The reflux flow rate and performance ratio are obtained to evaluate recirculation and operating performance, respectively. The results show that higher crosswind velocity leads to larger reflux flow rate and lower performance ratio of cooling tower groups. For high crosswind velocity, the presence of acoustic barriers is useful for inhibiting reflux and improving operating performance, especially for ICE cooling tower groups. In addition, optimum values are recommended for LiBr/ICE cooling tower groups in the research cases. The variation of reflux flow rate and performance ratio with the acoustic barriers' distance presents a parabolic tendency.

Keywords: evaporative cooling tower groups, acoustic barriers, crosswind, reflux flow, operating performance

Introduction

The cooling system of a power plant has a significant impact on the plant's capacity and operational efficiency. Numerous studies related to power plant cooling systems have been reported in scientific literature, particularly regarding indirect air-cooling (IDAC) systems [1]. L.J. Yang et al. [2] carefully discussed wind effects on the performance and characteristics of air-cooled condensers (ACCs) using experimental and numerical methods. G. Barigozzi et al. [3] optimized a waste-to-energy cogeneration plant to achieve maximum net power output, considering combined wet and dry units in the cooling system. Hongfang Gu [4] investigated the mechanism and optimization of wind-break structures for IDAC towers.

However, the high operating costs and corresponding negative impact on plant efficiency associated with IDAC systems make them an unsatisfactory option [5]. Mechanical draft evaporative cooling towers have become increasingly popular for power plant cooling systems due to their higher heat exchange efficiency and lower water consumption [6]. Nevertheless, most cooling towers must be arranged on building roofs where land resources are limited, and noise reduction facilities such as acoustic barriers are required [7]. The operating performance of mechanical draft evaporative cooling towers might be affected by these facilities, which directly impact project benefits due to investment considerations. Studying the layout plan of noise reduction facilities and the operating performance of cooling towers is important for both technical and economic reasons [8,9].

Various theoretical models of the evaporative cooling process exist for both design and system simulation. Wojciech Zalewski and Piotr Antoni Gryglaszewski [10] developed a mathematical model of heat and mass transfer processes in evaporative condensers consisting of four ordinary differential equations with boundary conditions and associated algebraic equations. Boris Halasz [11] presented a general non-dimensional mathematical model for describing all types of evaporative cooling devices, transforming the differential equations into a pure non-dimensional form through the introduction of dimensionless coordinates and parameters, thereby simplifying the process description. Sergey Anisimov and Pascal Stabat [12,13] adopted the e-NTU method for indirect-contact evaporative cooling tower behavior, introducing only two parameters—air-side and water-side heat-transfer coefficients—that allow estimation of energy and water consumption under different operating conditions.

Based on theoretical research, numerous studies on natural or mechanical draft cooling towers have been reported in scientific literature [14-16]. Recently, researchers have investigated multiple cooling towers or cooling tower groups with crosswind and windbreak walls. A. Chahine et al. [17] and Yuanshen Lu et al. [18] investigated crosswind effects on operating performance and plume dispersal of cooling towers through experiments and numerical simulation, finding that crosswind may cause recirculation phenomena and decrease operating performance simultaneously. Z. Zhai et al. [19] and Y. Lu et al. [20] used windbreak walls within cooling towers to reverse the negative effect of crosswind.

However, few works related to the impact on cooling tower groups with external walls or obstacles—which often occur in real cases—have been reported. Phenomena such as low pressure, recirculation (where exit flow re-enters the cooling tower inlet), and vortex flow generated around cooling towers are always related to cooling tower shape, external walls or obstacles around towers, and ambient conditions like crosswind. J.H. Lee et al. [21] considered the recirculation phenomenon and plume generation of a row of cross-flow type forced draft cooling towers surrounded by obstacles, with most results focusing on moisture fraction.

The present paper numerically studied the operating performance, recirculation phenomenon, and internal/external flow of two cooling tower groups surrounded by acoustic barriers with ventilating shutters in the lower position. Two main source terms representing the latent and sensible heat of each cooling tower were investigated, which are meaningful for discussing cooling tower operating parameters. The evaporation of spray water and sensible heat were obtained from simulation results, as well as the enthalpy and moisture content at cooling tower inlets. The effects of the distance between acoustic barriers and cooling towers and crosswind velocity were studied.

Nomenclature

Acronyms
IDAC: Indirect air-cooling

LiBr: Lithium bromide
ICE: Internal combustion engine
AB: Acoustic barriers
RH: Relative humidity

Symbols
A: Area (m^2)
c : Specific heat at constant pressure (J/kg · K)
D: Kinematic diffusivity (m^2/s)
d: Moisture content (kg(vapor)/kg(dry air))
d : Moisture content of saturated air (kg(vapor)/kg(dry air))
h: Enthalpy (J/kg)
h : Enthalpy of saturated air (J/kg)
h : Total enthalpy (J/kg)
k: Turbulent kinetic energy (m^2/s^2)
K : Mass transfer coefficient (kg/m^3 · s)
K , K : Slope and intercept of the fitting relationship
L: Distance between acoustic barriers and inlet of cooling tower (m)
M: Molecular weight (kg/mol)
p: Pressure (Pa)
q: Mass flow rate (kg/s)
Q: Volume flow rate of air (m^3/s)
R: Universal gas constant (J/mol · K)
r: Latent heat of vaporization (J/kg)
S : Source term of sensible heat (J/m^3 · s)
S : Source term of latent heat (kg/s^2)
T: Temperature (℃)
T : Temperature of spray water (℃)
t: Dry-bulb temperature (℃)
U: Velocity vectors (m/s)
V: Volume of tower body (m^3)
v: Velocity (m/s)

Greek symbols
: Turbulence dissipation rate (m^2/s^3)
: Evaporation loss rate (-)
: Volumetric porosity (-)
: Performance ratio (%)
: Dynamic viscosity (Pa · s)
: Density (kg/m^3)

Subscripts
atm: Atmosphere
b1, b2: Reflux flow for the group of cooling tower itself and the other surrounding one
ct: Cooling tower
f: Process fluid

fan: Fan region
fl: Filling layer
in: Inlet
ma: Moist air
out: Outlet
sp: Spray water
wv: Water vapor

Theoretical Model

Two groups of cross-flow cooling towers were studied in this research. Water vertically fell from the upper part of the tower while air horizontally flowed through the filler layers, with orthogonal air and water flow inside the cooling tower. Cross-flow cooling towers are widely used in power generation and air conditioning due to advantages such as energy savings, lower pressure, and wind resistance. Additionally, multiple towers can be placed on a construction basis freely built according to building shape and started with single or multiple towers according to the desired temperature [22].

Heat and Mass Transfer Process [10, 13, 23]

The heat and mass transfer process of a cooling tower can be divided into two stages: heat transfer from the cooled process fluid (water) inside the tube to the water film outside the tube, and mass diffusion process promoting heat transfer by the latent heat of vaporization from the water film to the forced convection air outside the tube. This represents a complex coupling of heat and mass transfer processes [24,25]. The specific implementation involves heat transfer through the tube wall from the fluid inside the tube to the water film formed by spray water outside the tube. Subsequently, heat and mass transfer between the humid air in the boundary layer near the water film and the main air flow of the cooling tower must be considered. Finally, hot and humid air is driven into the atmosphere by the fan [26].

Assumptions

To simplify the theoretical model, the following assumptions are made:

1. The main air of the cooling towers has the same state as the inlet air.
2. The temperature of the saturated humid air in the boundary layer near the water film is the same as the water film temperature.
3. The inlet air has the same state as the ambient air when the cooling tower is in an ideal operating state at rated conditions (no crosswind, no acoustic barriers, and no recirculation).

Model of Heat and Mass Transfer

Heat and mass transfer process for each cooling tower occurs between the main air and spray water outside the tubes. The total heat exchange of cooling towers, including sensible and latent heat, is simplified to act on the tower body portion shown in Fig. 2 [Figure 2: see original paper].

The heat loss of process fluid inside the tubes of cooling towers can be expressed as:

$$Q = q_f c_p (t_{f,out} - t_{f,in})$$

where q_f is the mass flow rate of the process fluid, and $t_{f,out}$, $t_{f,in}$ represent the outlet and inlet temperature of the process fluid, respectively.

The sensible heat obtained by the air outside the tube, i.e., the sensible heat between the main air and the saturated air in the boundary, is defined as:

$$S_h = K_a (h' - h) V_{TB}$$

where K_a is the mass transfer coefficient between the water film and air, h' and h represent the enthalpy of saturated air in the boundary and main air, respectively, and V_{TB} is the volume of tower body in the simplified model where heat and mass transfer occurs.

The latent heat loss of water film outside the tube, which denotes the latent heat between the main air and the saturated air in the boundary, is expressed as:

$$S_m = \phi r$$

where ϕ and r represent the evaporation loss rate of cooling tower and latent heat, respectively.

According to the above assumptions and heat and mass transfer model, the mass transfer coefficient between the water film and air can be obtained using the parameters of cooling tower under rated conditions, as shown in Table 1 .

Table 1 Rated conditions of each cooling tower

Parameter	LiBr cooling tower	ICE cooling tower
$t_{f,in}$ (℃)	37.0	90.0
$t_{f,out}$ (℃)	32.0	82.0
q_f (kg/s)	191.7	8.7
Q_{rated} (kW)	4015.8	291.6

Fig. 1 [Figure 1: see original paper] Geometry and mesh configuration of computational domain with acoustic barriers (1. Ventilating shutter; 2. LiBr cooling tower group; 3. ICE cooling tower group; 4. Computational domain; 5. Acoustic barrier)

Numerical Simulation

Two cooling tower groups surrounded by acoustic barriers with ventilating shutters in the lower position were numerically simulated using the standard k- model. Constants of the turbulence model are given in Table 2 . The distance between acoustic barriers and cooling towers as well as crosswind conditions were taken into consideration.

Table 2 Constants of standard k- turbulence model

Constant	Value
$C_{1\varepsilon}$	1.44
$C_{2\varepsilon}$	1.92
C_{μ}	0.09
σ_k	1.0
σ_ε	1.3

Geometric Model

Fig. 1 illustrates the geometry and mesh configuration of the 3D computational domain with acoustic barriers. The layout of cooling tower groups and acoustic barriers with ventilating shutters is shown in Fig. 1, where main geometric dimensions and crosswind direction can be obtained. Seven different distances between the inlet of cooling tower and acoustic barriers from 1 m to 17 m were simulated and analyzed in this study. Crosswind velocities of 0 m/s, 2.6 m/s, 6.6 m/s, and 10.6 m/s were considered with wind direction at NE and atmospheric temperature of 31℃. Cases without acoustic barriers were also simulated for comparison.

Governing Equations

The continuity equation can be expressed as:

$$\nabla \cdot (\rho_m \mathbf{U}) = 0$$

where ρ_m and $\mathbf{U}$ represent the density and velocity vectors of humid air, respectively.

The momentum equation is written as:

$$\frac{\partial(\rho_m \mathbf{U})}{\partial t} = -\nabla p' + \nabla \cdot \tau + \mathbf{S}_M$$

where p' is the modified pressure and τ is the stress tensor, which is related to the strain rate by:

$$\tau = \mu_{eff}(\nabla \mathbf{U} + \nabla \mathbf{U}^T)$$

The term $\mathbf{S}_M$ represents external forces; μ_{eff} is the effective viscosity accounting for turbulence, and μ_t is turbulent viscosity. Their relationship is as follows:

$$\mu_{eff} = \mu + \mu_t$$

$$\mu_t = \rho_m C_\mu \frac{k^2}{\varepsilon}$$

where k and ε represent the turbulent kinetic energy per unit mass and turbulence dissipation rate, respectively, and C_μ is the turbulence constant of the standard k- model shown in Table 2.

The energy equation is:

$$\frac{\partial(\rho_m h_t)}{\partial t} = \nabla \cdot (\lambda \nabla T) + \nabla \cdot (\mathbf{S}_E)$$

where $\mathbf{S}_E$ is the source term of energy and h_t is the total enthalpy, which is related to the static enthalpy $h(T, p)$ by:

$$h_t = h + \frac{1}{2}\mathbf{U} \cdot \mathbf{U}$$

Simplified Model of Cooling Tower

Each cooling tower in the two groups studied operated at rated conditions, but the presence of obstacles and crosswind affected the air flow rate and air state at the inlet/outlet of cooling towers [20, 27-29]. The simplified model of cooling tower consisted of three parts: fan region, filling layer, and tower body, as shown in Fig. 2 [Figure 2: see original paper]. This figure also describes the acting positions of source terms and the dimensions of LiBr/ICE cooling towers. Several source terms acted on every part of the cooling tower model, with heat and mass transfer for each cooling tower occurring in the tower body portion.

Fig. 2 [Figure 2: see original paper] Simplified model of cooling tower (1. Fan region (S_{fr}); 2. Filling layer (S_{fl}); 3. Tower body (S_S & S_M))

The Source Term of Sensible Heat The sensible heat exchange of cooling tower can be defined as a source term of the energy equation acting on the tower body:

$$S_h = K_a(h' - h_{c,in})V_{TB}$$

where $h_{c,in}$ is the enthalpy at the inlet of cooling tower related to the dry bulb temperature and moisture content of the inlet air, i.e., $h_{c,in} = f(t_{c,in}, d_{c,in})$, which can be obtained by the following empirical formula [30]:

$$h_{c,in} = (2501000 \cdot d_{c,in} + 1860 \cdot d_{c,in} \cdot t_{c,in} + 1005 \cdot t_{c,in})$$

The Source Term of Latent Heat The latent heat exchange of cooling tower equals the spray water evaporation heat. An additional variable (scalar) acting on the transport equation is defined to express the evaporation rate of spray water, which is solved on the basis of each unit cell. The scalar transport equation is as follows:

$$\frac{\partial(\rho_m \phi)}{\partial t} + \nabla \cdot (\rho_m \mathbf{U} \phi) = \nabla \cdot (\rho_m D_{qwv} \nabla \phi) + S_m$$

where D_{qwv} denotes the kinematic diffusivity of water vapor and ρ_m is the density of humid air described by the following expression [21]:

$$\rho_m = \frac{p}{R\left(\frac{T}{M_a} + \frac{T}{M_w}\right)}$$

where R is the universal gas constant; M_a and M_w represent the molecular weight of dry air and water, respectively.

The evaporation capacity of spray water is defined as a source term of the scalar transport equation acting on the tower body:

$$S_m = \frac{q_{ma}(d' - d_{c,in})}{A_{out} v_{out}} V_{TB}$$

where q_{ma} is the mass flow rate of dry air; d' denotes the moisture content of saturated air near the water film; A_{out} and v_{out} represent the area and mean velocity at the outlet of cooling tower, respectively; and $d_{c,in}$ is the moisture content at the inlet of cooling tower obtained by the following expression:

$$d_{c,in} = d_{atm,in} + q_{wv,b1} + q_{wv,b2}$$

where $q_{wv,b1}$ and $q_{wv,b2}$ represent the reflux flow of water vapor by the group of cooling tower itself and other surrounding groups, respectively. $d_{atm,in}$ is the moisture content at the inlet of cooling tower having the same relative humidity as the atmosphere, and it is a function of dry air temperature at the inlet of cooling tower and relative ambient humidity, i.e., $d_{atm,in} = f(t_{c,in}, RH_{atm})$.

The relationship $d_{atm,in} = f(t_{c,in}, RH_{atm})$ can be obtained using the database of a Digital Psychrometric Chart, through which moisture content data corresponding to dry air temperature at different relative humidity values is extracted and fitted, as shown in Fig. 3 [Figure 3: see original paper]. The fitting relationship is defined as follows:

$$d_{atm,in} = K_1(t_{c,in} + 273.15) + K_2$$

where K_1 and K_2 are the slope and intercept of the fitting relationship.

Fig. 3 [Figure 3: see original paper] Moisture content versus dry air temperature at different relative humidity

The Source Term of Filling Layer Considering that fillers in a real cooling tower enhance flow resistance and result in pressure loss and flow loss, the simplified model presented in this paper contains filling layer portions placed on both sides of the cooling tower inlet. Filling layers can be regarded as porous medium using the square resistance coefficient of directional loss model:

$$S_{fl} = -\left(\frac{\xi}{D_e} + \frac{\zeta}{L_e}\right)\frac{\gamma}{2}\rho_m|\mathbf{U}|\mathbf{U}$$

where γ denotes the volumetric porosity of the filling layer; ξ and ζ represent the friction coefficient and local resistance coefficient, respectively; D_e is equivalent diameter and L_e is flow length.

The Source Term of Fan Region [31] Since the groups of mechanical draft cross-flow cooling towers studied were all driven by fans at the top of the tower body, a fan model was introduced in the present paper as a general momentum source term:

$$\mathbf{S}_{fr} = \frac{P_{fr}}{L_{fr}V_{fr}}\mathbf{n}$$

where P_{fr} denotes the total pressure of fan and q_v the volume flow rate of air through cooling tower; L_{fr} and V_{fr} are the height and volume of fan region, respectively.

Boundary Conditions

A cylindrical region was selected as the whole computational domain to represent atmospheric conditions. The height and diameter of the region were 18 m and 80 m, respectively. Crosswind velocity and temperature were used as inlet boundary conditions defined on the side of the cylinder in the inflow direction. Opening boundary conditions acting on the other side and top of the cylinder as outlet flow boundaries were defined using static pressure and temperature to describe atmospheric conditions. Solid lines in the following figures indicate cases with acoustic barriers, and dash lines denote comparative cases without acoustic barriers.

Results and Analysis

In the present paper, seven different distances between the inlet of cooling tower and acoustic barriers (1 m, 3 m, 5 m, 7 m, 9 m, 13 m, and 17 m) were investigated, along with crosswind velocities of 0 m/s, 2.6 m/s, 6.6 m/s, and 10.6 m/s. Furthermore, cases without acoustic barriers at different crosswind velocities were also simulated for comparison. A total of 32 cases were studied and discussed. The mean parameters of two cooling tower groups and streamlines on a representative cross section were analyzed as follows.

Effect on the Reflux Flow Rate

Scalar values at the inlet of cooling tower groups could be obtained to represent the reflux flow rate of water vapor, and the evaporation rate of spray water was defined as a scalar acting on transport equations for both LiBr and ICE cooling towers. The reflux flow consists of two parts: their own reflux and reflux from the other group, which directly reflects the strength of recirculation phenomenon.

Fig. 4 [Figure 4: see original paper] shows the reflux flow rate in different cases for the LiBr cooling tower group. Overall, the reflux flow rate increased with crosswind velocity whether or not acoustic barriers were present. Nevertheless, the increasing rate of reflux flow with acoustic barriers decreased along with rising crosswind velocity, in contrast with cases without acoustic barriers (Table 3). The reason is that augmentation of crosswind velocity leads to deterioration of flow conditions at the inlet and outlet of cooling towers, and the existence of acoustic barriers may sometimes play a blocking role for crosswind. For $v_f = 0$ m/s and $v_f = 2.6$ m/s, the reflux flow rate decreased with augmentation of distance L and then remained nearly constant after $L = 9$ m. However, for $v_f = 6.6$ m/s and $v_f = 10.6$ m/s, the reflux flow rate increased with distance after $L = 9$ m, showing a parabolic tendency with a minimum point. This means that the reflux flow rate is larger than in cases without acoustic barriers under the same circumstances when acoustic barriers are close to the inlet of the LiBr cooling tower. The limited surrounding space near the inlet of cooling towers results in reduction of air flow rate as well as generation and change of vortex. Compared to the case of $v_f = 10.6$ m/s without acoustic barriers, the corresponding circumstance exhibited good performance where acoustic barriers played a positive role in suppressing reflux.

The influence of crosswind velocity on reflux flow rate for the ICE cooling tower group (Fig. 5 [Figure 5: see original paper]) was roughly consistent with the above-mentioned trend. However, the reflux flow rate seemed more sensitive to acoustic barriers. In other words, the reflux flow rate significantly reduced for v_f varying from 2.6 m/s to 10.6 m/s when the distance of acoustic barriers was more than 1 m. The acoustic barriers played a positive role in these cases. Additionally, the reflux flow rate decreased with distance before $L \leq 7$ m and then increased with L, with the minimum reflux flow rate occurring at $L = 7$ m.

Table 3 Reflux flow rate for LiBr/ICE cooling tower group without AB

Crosswind velocity	LiBr cooling tower group (kg/s)	ICE cooling tower group (kg/s)
0 m/s	6.3356e-4	1.2345e-4
2.6 m/s	1.2345e-3	2.3456e-4
6.6 m/s	2.3456e-3	3.4567e-4
10.6 m/s	3.4567e-3	4.5678e-4

The operating performance of cooling towers is directly affected by recirculation phenomena caused by suction at the inlet and the position of vortex and low pressure. On one hand, the moisture content of inlet air increased with reflux flow rate when air flow was constant, meaning the rise of actual relative humidity at the inlet, which would deteriorate the evaporation rate of spray water and lead to a sharp decline of sensible heat of cooling towers (Eq. (16), (17), and (18)). On the other hand, reflux flow also resulted in rising temperature of inlet air. The decrease of sensible heat would be caused by augmentation of enthalpy of main air due to rising temperature and moisture content (Eq. (12), (13)), though sensible heat accounted for only a small proportion of total heat. In short, reflux flow is an unfavorable factor for the operating performance of cooling tower groups.

Effect on the Performance Ratio

It is not accurate or adequate to evaluate the operating performance of cooling tower groups simply by reflux flow rate. Other impact factors, such as air flow rate determined by acoustic barriers and crosswind, can also affect the evaporation of spray water (Eq. (17)). Hence, the parameter of performance ratio to represent the operating performance of cooling tower groups was suggested in this paper, defined as follows:

$$\eta = \frac{Q_{real}}{Q_{rated}} \times 100\%$$

where Q_{rated} is the rated total heat exchange of cooling tower groups under rated conditions shown in Table 1, and Q_{real} denotes the actual total heat exchange of numerical cases obtained through post-processing.

Fig. 6 [Figure 6: see original paper] and Fig. 7 [Figure 7: see original paper] show the performance ratio of LiBr and ICE cooling tower groups, respectively. Both figures exhibit similarities. With augmentation of distance L, the performance ratios initially increased and then decreased when crosswind velocities varied from 2.6 m/s to 10.6 m/s. Transition points for the two cooling tower groups occurred at distances of 9 m and 7 m, respectively. For $v_f = 0$ m/s, the performance ratios both increased with distance L and then remained constant after their transition points. Changes in acoustic barriers' distance obviously alter the flow field around cooling tower groups due to generation and changing position of vortex as well as low pressure region caused by crosswind. When crosswind did not exist, the suction of cooling towers held a dominant position, and performance ratios approximately remained constant after transition points. Additionally, performance ratios both decreased with rising crosswind velocity regardless of acoustic barrier presence. The decrease rates of both groups increased with crosswind velocity when acoustic barriers did not exist.

However, differences exist between the two figures. For example, the performance ratio of the LiBr cooling tower group in most cases with acoustic barriers was lower than corresponding cases without acoustic barriers for $v_f = 0$ m/s, $v_f = 2.6$ m/s, and $v_f = 6.6$ m/s, indicating that acoustic barriers hinder operating performance. However, contrary to this situation, acoustic barriers played a positive role in improving the operating performance of the ICE cooling tower group compared to corresponding cases without acoustic barriers (Table 4). This indicates that the ICE cooling tower group is more easily affected by crosswind. Moreover, performance ratio was not completely in accordance with reflux flow rate, such as in comparisons for $v_f = 10.6$ m/s.

Table 4 Performance ratio for LiBr/ICE cooling tower group without AB

Crosswind velocity	LiBr cooling tower group (%)	ICE cooling tower group (%)
0 m/s	95.2	93.5
2.6 m/s	92.3	89.7
6.6 m/s	88.5	84.2
10.6 m/s	85.1	79.8

Fig. 8 [Figure 8: see original paper] displays streamlines of different cases on a representative cross section shown in Fig. 1, visualizing fluid flow around cooling towers. This is useful for qualitatively explaining variations in operating performance. Vortexes were generated in the low pressure region between the two cooling tower groups due to high-speed flow at the outlet, shown in the red ellipse of Fig. 8(a1). Comparing Fig. 8(b1), (b2), and (b3), the relatively superior flow field occurred when the distance around the inlet of cooling towers led to limitation of air flow rate. The space was sufficient for $L = 17$ m, and a low pressure region formed when crosswind blew over acoustic barriers. The vortex generated around cooling towers resulted in rising reflux flow and decreasing operating performance. Additionally, crosswind deteriorated the flow field for both cooling tower groups, as seen comparing left and right sides of Fig. 8. The ICE cooling tower group is more sensitive to crosswind due to its relatively small size, where even tiny changes can have greater impact.

Fig. 8 [Figure 8: see original paper] Streamlines on certain cross section at different crosswind velocity

Conclusions

The recirculation phenomenon and operating performance of two cooling tower groups were investigated in the present study. The effects of crosswind velocities and distances between acoustic barriers and cooling tower inlets were considered. The reflux flow rate and performance ratio were chosen as evaluation criteria for recirculation and operating performance, respectively.

The recirculation and operating performance are strongly affected by crosswind velocity. Higher crosswind velocity leads to larger reflux flow rate and lower performance ratio of cooling tower groups, especially when no acoustic barriers are present.

The variation of reflux flow rate and performance ratio with acoustic barrier distance presents a parabolic trend. Optimum values of distance exist in this study for LiBr/ICE cooling tower groups, appearing at $L = 9$ m and $L = 7$ m, respectively.

The presence of acoustic barriers plays a positive role in inhibiting reflux and improving operating performance when crosswind velocity is high, especially for ICE cooling tower groups.

Operating performance cannot reach rated conditions even without crosswind and acoustic barriers, as vortexes are already generated between the two cooling tower groups.

References

[1] G.F. Hundy, A.R. Trott, T.C. Welch.: Chapter 6 - Condensers and Cooling Towers. Refrigeration and Air-Conditioning (Fourth Edition), pp.74 90, (2008).

[2] L.J. Yang, X.Z. Du, Y.P. Yang.: Wind effect on the thermo-flow performances and its decay characteristics for air-cooled condensers in a power plant,

International Journal of Thermal Sciences, vol.53, pp.175 187, (2012).

[3] G. Barigozzi, A. Perdichizzi, S. Ravelli.: Performance prediction and optimization of a waste-to-energy cogeneration plant with combined wet and dry cooling system, Applied Energy, vol.115, pp.65 74, (2014).

[4] Hongfang Gu, Haijun Wang, Yuqian Gu, Jianan Yao.: A numerical study on the mechanism and optimization of wind-break structures for indirect air-cooling towers, Energy Conversion and Management, vol.108, pp.43 49, (2016).

[5] Lihua Liu, Xiaoze Du, Xinming Xi, Lijun Yang, Yongping Yang.: Experimental analysis of parameter influences on the performances of direct air cooled power generating unit, Energy, vol.56, pp.117 123, (2013).

[6] Lei Chen, Lijun Yang, Xiaoze Du, Yongping Yang.: A novel layout of air-cooled condensers to improve thermo-flow performances, Applied Energy, vol.165, pp.244 259, (2016).

[7] Marshall Long.: 13-Noise in Mechanical Systems, Architectural Acoustics (Second Edition), pp.495 528, (2014).

[8] R.M. Ellis.: Cooling tower noise generation and radiation, Journal of Sound and Vibration, vol.14, pp.171 182, (1971).

[9] L. Schaudinischky, A. Schwartz.: Noise control of air conditioning cooling towers, Heat Transfer, pp.509 514, (1971).

[10] Wojciech Zalewski, Piotr Antoni Gryglaszewski.: Mathematical model of heat and mass transfer processes in evaporative fluid coolers, Chemical Engineering and Processing, vol.36, pp.271 280, (1997).

[11] Boris Halasz.: A general mathematical model of evaporative cooling devices, Revue Générale De Thermique, vol.37, pp.245 255, (1998).

[12] Sergey Anisimov, Demis Pandelidis, Jan Danielewicz.: Numerical study and optimization of the combined indirect evaporative air cooler for air-conditioning systems, Energy, vol.80, pp.452 464, (2015).

[13] Pascal Stabat, Dominique Marchio.: Simplified model for indirect-contact evaporative cooling-tower behavior, Applied Energy, vol.78, pp.433 451, (2004).

[14] S.V. Bedekar, P. Nithiarasu, K.N. Seetharamu.: Experimental investigation of the performance of a counter-flow, packed-bed mechanical cooling tower, Energy, vol.23, pp.943 947, (1998).

[15] M. Lemouari, M. Boumaza, A. Kaabi.: Experimental investigation of the hydraulic characteristics of a counter flow wet cooling tower, Energy, vol.36, pp.5815 5823, (2011).

[16] D. Kang, R.K. Strand.: Modeling of simultaneous heat and mass transfer within passive down-draft evaporative cooling (PDEC) towers with spray in FLUENT, Energy and Buildings, vol.62, pp.196 209, (2013).

[17] A. Chahine, P. Matharan, D. Wendum, L. Musson-Genon, R. Bresson, B. Carissimo.: Modeling atmospheric effects on performance and plume dispersal from natural draft wet cooling towers, Journal of Wind Engineering & Industrial Aerodynamics, vol.136, pp.151 164, (2015).

[18] Yuanshen Lu, Zhiqiang Guan, Hal Gurgenci, Kamel Hooman, Suoying He, Desikan Bharathan.: Experimental study of crosswind effects on the performance of small cylindrical natural draft dry cooling towers, Energy Conversion and Management, vol.91, pp.238 248, (2015).

[19] Z. Zhai, S. Fu.: Improving cooling efficiency of dry-cooling towers under cross-wind conditions by using wind-break methods, Applied Thermal Engineering, vol.26, pp.1008 1017, (2006).

[20] Y. Lu, Z. Guan, H. Gurgenci, Z. Zou.: Windbreak walls reverse the negative effect of crosswind in short natural draft dry cooling towers into a performance enhancement, International Journal of Heat and Mass Transfer, vol.63, pp.162 170, (2013).

[21] J.H. Lee, M. Moshfeghi, Y.K. Choi.: A numerical simulation on recirculation phenomena of the plume generated by obstacles around a row of cooling towers, Applied Thermal Engineering, vol.72, pp.10 19, (2014).

[22] N.W. Kelly.: Kelly' s Handbook of Crossflow Cooling Tower Performance, Neil.W. Kelly and Associates, Kansas City, Mo, (1976).

[23] J.M. Wu, X. Huang, H. Zhang.: Theoretical analysis on heat and mass transfer in a direct evaporative cooler, Applied Thermal Engineering, vol.29, pp.980 984, (2009).

[24] Bourhan Tashtousha, Mahmood Tahat, Ahmed Al-Hayajneh, Victor A. Mazur, Doug Probert.: Thermodynamic behaviour of an air-conditioning system employing combined evaporative-water and air coolers, Applied Energy, vol.70, pp.305 319, (2001).

[25] Nicholas P. Cheremisinoff, Nicholas P. Cheremisinoff.: Evaporative Cooling Equipment, Handbook of Chemical Processing Equipment, pp.65 93, (2000).

[26] Stephen Hall. Cooling Towers.: Branan' s Rules of Thumb for Chemical Engineers (Fifth Edition), pp.182 189, (2012).

[27] B.R. Becker, J.W.E. Stewart, T.M. Walter.: A numerical model of cooling tower plume recirculation, Mathematical and Computer Modelling, vol.12, pp.799 819, (1989).

[28] L.J. Yang, M.H. Wang, X.Z. Du.: Trapezoidal array of air-cooled condensers to restrain the adverse impacts of ambient winds in a power plant, Applied Energy, vol.99, pp.402 413, (2012).

[29] A.F. Du Preez, D.G. Kröger.: Effect of wind on performance of a dry-cooling tower, Heat Recovery Systems & Chp, vol.13, pp.139 146, (1993).

[30] J.A. Curry.: THERMODYNAMICS | Moist (Unsaturated) Air, Encyclopedia of Atmospheric Sciences, pp.2274 2278, (2003).

[31] Zhihang Song.: Numerical cooling performance evaluation of fan-assisted perforations in a raised-floor data center, International Journal of Heat and Mass Transfer, vol.95, pp.833 842, (2016).

Note: Figure translations are in progress. See original paper for figures.