Effect of Inlet Air Turbulence on the Cooling Performance of Solar Enhanced Dry Cooling Towers

Rui Wang, Suoying He, Ming Gao, Yuetao Shi and Fengzhong Sun

Abstract Solar enhanced natural draft dry cooling towers (SENDDCTs) use solar energy to reheat the air coming from vertically arranged heat exchangers for tower performance enhancement. The SENDDCT produces large differences in air density between the inside and outside of the tower (i.e., the tower driving force is intensified), and therefore enhances the ventilation inside the tower for better cooling. How to efficiently introduce solar energy for reheating the air is important for a SENDDCT. This paper is to study different air turbulence regimes so as to enhance the heat transfer in the solar reheating section. The different air turbulences are achieved by changing the endothermic ground from flat to rectangular ribs. A 3-D model is developed using FLUENT 18.0 to simulate the operation of the above-mentioned SENDDCTs. The model will be validated by comparing with literature. According to the simulation, the rectangular-rib SENDDCT enhancement can go up to 13.1% by increasing the heat rejection rate of the flat SENDDCT from 133 to 149 MW, which proves that the cooling performance of the SENDDCT can be improved by intensifying the inlet air turbulence.

Keywords Solar enhanced natural draft dry cooling tower · Air turbulence · Simulation

---

R. Wang · S. He (✉) · M. Gao · Y. Shi · F. Sun
School of Energy and Power Engineering, Shandong University, Jinan 250061, China
e-mail: suoyinghe@hotmail.com

R. Wang
e-mail: wangrui199588@163.com

M. Gao
e-mail: gm@sdu.edu.cn

Y. Shi
e-mail: shieddie@sdu.edu.cn

F. Sun
e-mail: sfzh@sdu.edu.cn

Z. Wang et al. (eds.), *Proceedings of the 11th International Symposium on Heating, Ventilation and Air Conditioning (ISHVAC 2019)*, Environmental Science and Engineering, https://doi.org/10.1007/978-981-13-9528-4_10

1 Introduction

Cooling towers are widely used in thermal power plants as the heat dissipation devices to cool the working fluid, such as thermal power plants and nuclear power plants [1]. Cooling towers can be divided into dry cooling and wet cooling in terms of cooling medium. The working principle of wet cooling tower is evaporative cooling. In wet cooling towers, water is distributed to the tower fills through nozzles and forms counter-flow with the air to transfer heat and mass [2]. However, wet cooling consumes a lot of water and is not suitable in dry areas while dry cooling can fill this gap. Dry cooling towers rely on sensible heat transfer between the air and the working fluid [3, 4]. Natural draft dry cooling tower (NDDCT) generates continuous airflow through the heat exchanger by the air pressure difference between the inside and outside of the tower, which depends on the ambient air temperature. NDDCT has low cooling efficiency during hot periods, like summer when compared with wet cooling towers [5]. Hot periods are usually the peak of power demand [6]. To improve the cooling performance of the NDDCT, Solar enhanced natural draft dry cooling tower (SENDDCT) was proposed by Zou [7].

The design of a SENDDCT is shown in Fig. 1. According to Zou [8], a SENDDCT consists of three major components, i.e., heat exchangers, a solar collector (sunroof and ground) and a tower. The solar collector further heats the air from the heat exchangers, and thus the temperature difference between the inside and outside of tower increases, which increases the airflow rate through the heat exchangers and more heat will be rejected. The air temperature increased here is the air inside the solar collector. This will not affect the air temperature that cools the hot water.

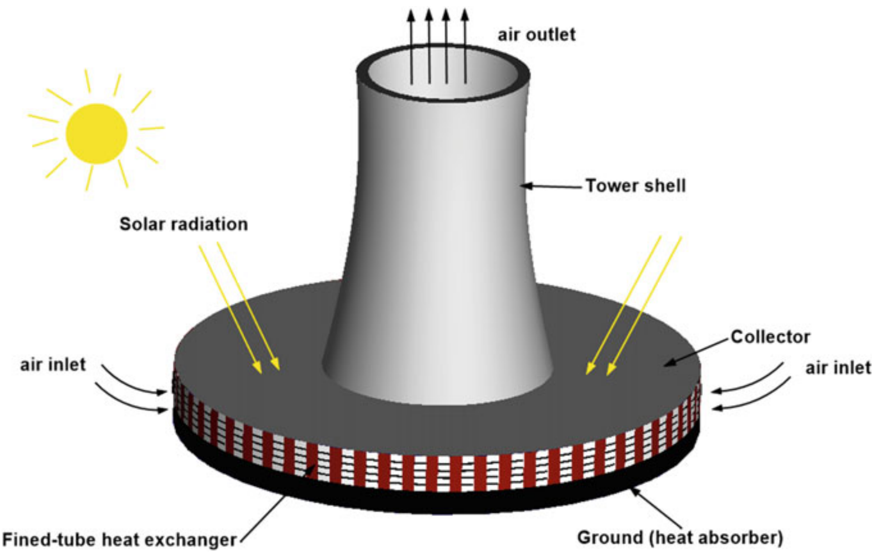

Fig. 1 Concept of a solar enhanced natural draft dry cooling tower

The temperature of the air transferring heat with the hot water is always the ambient temperature. In SENDDCT's structural design, the heat exchanger was placed vertically along the outer edge of the solar collector. SENDDCTs are designed in parabolic shape for reducing the pressure loss from the collector to the transition zone of the tower [9, 10]. Generally, the larger the size of the solar collector, more heat is achieved at the ground and the building cost is increased [11].

Although studies about the SENDDCT have been conducted by some authors [3, 4], their studies focused on the optimization of the collector or the arrangement of the heat exchangers. The effect of inlet air turbulence on the cooling performance of the SENDDCT is scarce. Theoretically, intensified inlet air turbulence improves the heat transfer, but this will bring an extra air-side flow resistance since the turbulence is intensified by changing the heat transfer ground from flat to rectangular ribs. To this end, the current paper takes into account both the heat transfer enhancement and the extra air-side flow resistance to investigate the cooling performance of a 140 m height SENDDCT. The specific objectives are to investigate the effect of inlet air turbulence on the cooling performance of the proposed SENDDCT.

2 Mathematical Model

2.1 Geometric Model

The design of the SENDDCT is shown in Fig. 1. The parameters of the SENDDCT used in simulation are listed in Table 1. This study uses a geometric model of an SENDDCT with three-row finned tube heat exchangers which are placed vertically along the outer edge of the solar collector.

Table 1 Design parameters of the SENDDCT

Description	Symbols	Values
Tower height (m)	H	140
Tower base diameter (m)	D	101
Collector diameter (m)	D_c	195
Heat exchanger height (m)	H_h	15
Water inlet temperature (K)	T_{wi}	333.15
Ambient temperature (K)	T_a	303.15
Water flow rate (kg/s)	m_w	2375
Solar radiation (W/m^2)	q	1000
Ambient pressure (Pa)	P_a	101,325
Gravitational acceleration (m/s^2)	g	9.8

2.2 Numerical Model

The operation of the SENDDCT is coupled by the energy and draft equations. The airflow state can be described by the mass, momentum and energy equations. The mass conservation equation is

$$\nabla \cdot (\rho_a v_a) = S_m \tag{1}$$

where ρ_a is the wet air density, kg/m^3, v_a is the air velocity, m/s, S_m is volumetric mass transfer rate (kg/(m^{3}s)). The momentum conservation equation is

$$\nabla \cdot (\rho_a v_a v_a) = -\nabla p_a + \nabla \cdot (\mu_1 + \mu_2) \cdot \left[(\nabla v_a + \nabla v_a^T) - \frac{2}{3} \nabla \cdot v_a I \right] + \rho_a g + F \tag{2}$$

where μ_l and μ_t are the laminar and turbulent viscosity coefficients, respectively, kg/(m s), I is the unit tensor of air, g is the gravitational acceleration, m/s^2, F is the volumetric resistance for air, N/m^3. The energy conservation equation is

$$\nabla \cdot (\rho_a v_a \int_{t_{ref}}^{t_a} c_a dt) = \nabla \cdot \left((k_1 + k_2) \nabla t_a - \sum_n h_n J_n \right) + S_{ae} \tag{3}$$

where t_a is the air temperature, °C, $t_{ref} = 0$ °C is the reference temperature, °C, k_l and k_t are the laminar and turbulent thermal conductivity coefficients, respectively, W/(m °C), h_n is the sensible enthalpy corresponding to component n, J/kg, J_n is the diffusion flux of component n, kg/m^2 s, $\sum_n h_n J_n$ is the sensible enthalpy caused by the diffusion of component n and S_{ae} is the volumetric energy transfer rate of air, W.

Thermal Analysis

In FLUENT 18.0, the radiator boundary condition can be used to model the heat exchangers [9, 10]. The heat transfer coefficient can be expressed as a function of air velocity [12, 13].

$$h = 2038 - 3380 V_a + 3433 V_a^2 - 1608 V_a^3 + 282.4 V_a^4 \tag{4}$$

The radiator is considered to be an infinitely thin wall, and the heat transfer through the radiator can be calculated by

$$q_h = A_{fr} h (T_{wi} - T_{ao}) \tag{5}$$

The heat balance between air-side and water-side can be demonstrated as

$$Q = M_a c_{pa} (T_{aho} - T_{ai}) = M_w c_{pw} (T_{wi} - T_{wo}) \tag{6}$$

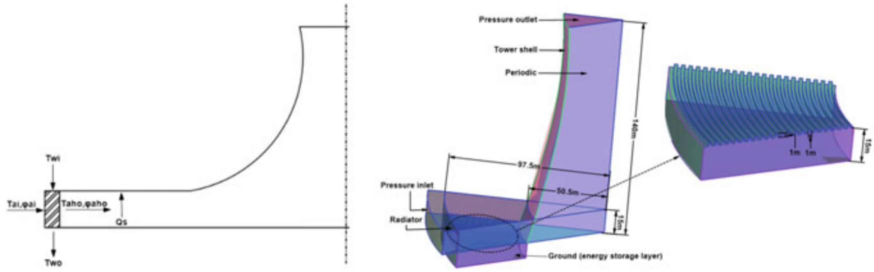

Fig. 2 Heat flux (left) and 3-D model of a SENDDCT (right)

where A_{fr} is the frontal area of heat exchangers, m^2; C_{pa} is specific heat of air, J/kg K; C_{pw} is specific heat of water, J/kg K; M_a is air mass flow rate, kg/s; M_w is water mass flow rate, kg/s, T_{ai} is the temperature of the inlet air, K; T_{wi} and T_{wo} is the temperature of water inlet and outlet, K; T_{aho} is the temperature of the air at heat exchanger exit, K. The coefficient of heat transfer by convection between the sunroof and the ambient is specified as 14 W/(m^2 K). A schematic to show temperatures and heat flux is shown in Fig. 2, φ_{ai} is the air humidity. Inside the solar collector, heat radiation is neglected when compared with the heat convection, and the heat transferred to the air can be calculated by Eq. (7) [11].

$$Q_s = hA(T_g - T_{ac}) = M_a c_{pa}(T_{ac} - T_a) \tag{7}$$

where T_g is the temperature of the collector ground and T_{ac} is the temperature under the collector in K.

Pressure Drop Analysis

The pressure drop through the radiator can be calculated by

$$\Delta P = k\frac{1}{2}\rho_{ai}V_a^2 \tag{8}$$

The pressure drop of the heat exchanger can be expressed as Eq. (9) [12, 13].

$$k = 74.19 - 100.9V_a + 91.44V_a^2 - 38.97V_a^3 + 6.218V_a^4 \tag{9}$$

The loss coefficient of the rectangular ribs is evaluated by Eq. (10) [14].

$$f = \Delta P \Big/ \left(\frac{1}{2}\rho u_{max}^2 \cdot N\right) \tag{10}$$

where the u_{max} is the average of the maximum velocity in m/s and N is the rib rows.

Table 2 Detailed boundary settings

Place	Type	Value
Sunroof	Wall	$T_{ext} = 303.15$ K (30 °C), $h_s = 14$ W/m^2 K, $\varepsilon_{ext} = 0.84$
Heat exchangers	Radiator	$T_{wi} = 333.15$ K (60 °C), Eqs. (4) and (9)
Inlet faces	Pressure inlet	$T_{ext} = 303.15$ K (30 °C), P_i=0 Pa
Tower exit	Pressure outlet	$P_o = 0$ Pa
Heat storage layer	Wall	0.0001 m, 7,560,000 W/m^3
Tower shell	Wall	Adiabatic free-slip
The bottom of ground	Wall	$T_g = 300$ K
Both sides of sector	Periodic	[–]

2.3 Boundary Conditions and Solution Method

A 30-degree sector of the SENDDCT was modeled in this study (Fig. 2). Only hexahedral elements were used. Detailed boundary settings are illustrated in Table 2. The 3-D steady-heat transfer model, the pressure-based solver with SIMPLE segregated algorithms, a standard k-ε model and second-order discretization are used in simulation.

2.4 Model Validation

The grid independence was studied at grid numbers of 0.63×10^6, 1.03×10^6 and 3×10^6. It was found that grid number of 1.03×10^6 is suitable for our case where both computational accuracy and time is acceptable.

The model was validated by comparing with literature [8]. The calculated heat transfer rate and air mass flow rate are 132.9 MW and 15,485 kg/s, respectively. The corresponding values obtained in Zou's model were 135.9 MW and 15,358 kg/s [8]. The differences in heat transfer rate and air mass flow rate are 2.1% and 0.82%, respectively. The comparison between calculated values and literature found good agreement. This validated model was then adapted to simulate the operation of the SENDDCT with different inlet air turbulences.

3 Result and Discussion

Figure 3 mainly illustrates the distribution of the air temperature in the SENDDCT. The air is further heated when it leaves from the heat exchangers to the solar collector. It can be seen from Fig. 3 that the SENDDCT with rectangular ribs performs

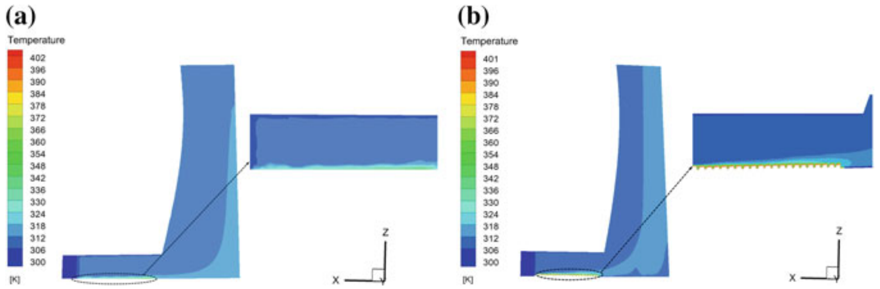

Fig. 3 Temperature distribution graph of **a** SENDDCT with flat ground, **b** SENDDCT with rectangular ribs in CFD

better in heating the air inside the solar collector than the one with flat ground. Giving the design parameters of the SENDDCT in Table 1 and the boundary conditions in Table 2, the overall heat rejection performance of the SENDDCT can be obtained based on simulation results. The result indicates that the heat rejection rate of the SENDDCT increases from 133 to 149 MW by 13.1% increment when the endothermic ground is changed from flat to rectangular ribs.

The air temperature distribution predicted by the 3-D model at different heights inside the solar collector is reported in Fig. 4. It can be seen from Fig. 4 that the air temperature inside the solar collector of the rectangular ground is higher than that for flat ground, which will then enlarge the air density difference of the SENDDCT, improving the tower buoyancy effect and the heat rejection performance. This proves that rectangular ground intensifies the inlet air turbulence and provides better cooling compared with flat ground. Figure 4 also indicates that the air temperature experiences a drop trend from the ground to the sunroof inside the solar collector.

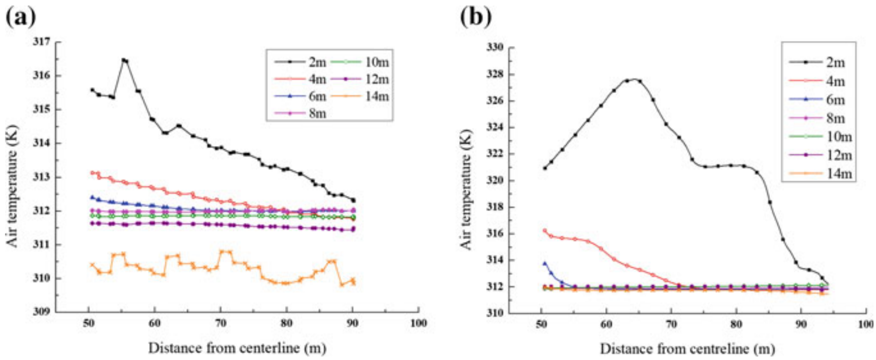

Fig. 4 Air temperature of **a** SENDDCT with flat ground, **b** SENDDCT with rectangular ribs distributions at different heights in the radial direction

4 Conclusion

A 3-D model was established using FLUENT 18.0 to simulate the operation of a SENDDCT. The inlet air turbulence of the SENDDCT was changed by modifying the endothermic ground from flat to rectangular ribs. The effect of inlet air turbulence on the performance of a SENDDCT was studied. The result indicates that the heat rejection rate of the SENDDCT increases from 133 to 149 MW by 13.1% increment when the endothermic ground is changed from flat to rectangular ribs. The study proves that the cooling performance of the SENDDCT can be improved by intensifying the inlet air turbulence.

Acknowledgements This work was supported by "Young Scholars Program of Shandong University" (YSPSDU, No. 2018WLJH73). The financial supports from Shandong Natural Science Foundation (Grant No. ZR2017QEE010) and National Natural Science Foundation of China (Grant No. 51776111) are gratefully acknowledged.

References

1. Asvapoositkul, W., Kuansathan, M.: Comparative evaluation of hybrid (dry/wet) cooling tower performance. Appl. Therm. Eng. **71**, 83–93 (2014)
2. Lucas, M., Ruiz, J., Matinez, P.J., Kaiser, A.S., Viedma, A., Zamora, B.: Experimental study on the performance of a mechanical cooling tower fitted with different types of water distribution systems and drift eliminators. Appl. Therm. Eng. **50**, 282–292 (2013)
3. He, S., Guan, Z.Q., Gurgenci, H., Jahn, I., Lu, Y.S., Alkhedhair, A.M.: Influence of ambient conditions and water flow on the performance of pre-cooled natural draft dry cooling towers. Appl. Therm. Eng. **66**, 621–631 (2014)
4. He, S., Gurgenci, H., Guan, Z.Q., Alkhedhair, A.M.: Pre-cooling with Munters media to improve the performance of natural draft dry cooling towers. Appl. Therm. Eng. **53**, 67–77 (2013)
5. Kanoglu, M., Cengel, Y.A.: Improving the performance of an existing air-cooled binary geothermal power plant: a case study. J. Energy Resour. Technol. **121**, 196–202 (1999)
6. Hooman, K.: Dry cooling towers as condensers for geothermal power plants. Int. Commun. Heat Mass Transfer **37**, 1215–1220 (2010)
7. Zou, Z., Guan, Z., Gurgenci, H., et al.: Solar enhanced natural draft dry cooling tower for geothermal power applications. Sol. Energy **86**, 2686–2694 (2012)
8. Zou, Z., Guan, Z., Gurgenci, H.: Numerical simulation of solar enhanced natural draft dry cooling tower. Sol. Energy **101**, 8–18 (2014)
9. Al-Waked, R., Behnia, M.: The performance of natural draft dry cooling towers under crosswind: CFD study. Int. J. Energy Res. **28**, 147–161 (2004)
10. Al-Waked, R., Behnia, M.: The effect of windbreak walls on the thermal performance of natural draft dry cooling towers. Heat Transf. Eng. **26**, 50–62 (2005)
11. Yuan, L., Zhu, J., Li, T., Fan, H., Lu, X.: Numerical simulation of flow and heat transfer characteristics in Solar Enhanced Natural Draft Dry Cooling Tower. Appl. Therm. Eng. **87**, 98–105 (2015)
12. Briggs, D.E., Young, E.H.: Convective heat transfer and pressure drop of air flowing across triangular pitch banks of finned tubes. Chem. Eng. Prog. Symp. Ser. **59**, 1–10 (1963)

13. Robinson, K., Briggs, D.E.: Pressure drop of air flowing across triangular pitch banks of finned tubes. Chem. Eng. Prog. Symp. Ser. **62**, 177–184 (1966)
14. Guo, P., Li, J., Wang, Y.: Numerical simulations of solar chimney power plant with radiation model. Renew. Energy. **62**, 24–30 (2014)