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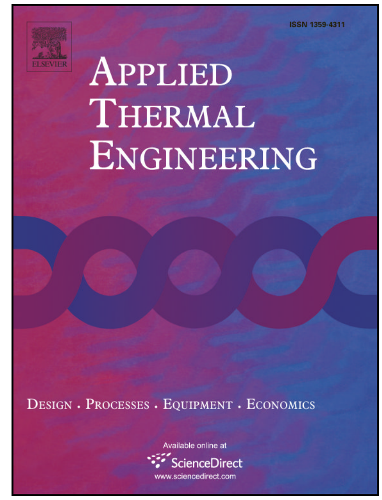

Research Paper

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PII: S1359-4311(17)33977-7
DOI: http://dx.doi.org/10.1016/j.applthermaleng.2017.09.071
Reference: ATE 11128

To appear in: *Applied Thermal Engineering*

Received Date: 12 June 2017
Revised Date: 28 July 2017
Accepted Date: 13 September 2017

Please cite this article as: X. Li, H. Gurgenci, Z. Guan, Y. Sun, Experimental Study of Cold Inflow Effect on a Small Natural Draft Dry Cooling Tower, *Applied Thermal Engineering* (2017), doi: http://dx.doi.org/10.1016/j.applthermaleng.2017.09.071

Experimental Study of Cold Inflow Effect on a Small Natural Draft Dry Cooling Tower

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Highlights

- Detailed experimental data of cold inflow behaviours are presented.
- The mechanism of cold flow and its effect on cooling tower performance is discussed.
- A solution is proposed to deal with the problem.

ABSTRACT: The heat rejection rate of natural draft dry cooling tower, as well as the operating performance of a power plant, can be affected by numerous ambient factors. The cold inflow is an unfavourable air turbulence at the top of the cooling tower and has a significant negative effect on the performance of natural draft cooling towers. In the present research, results are given for a 20 m high natural draft dry cooling tower experimental system tested at different ambient conditions. Several events of cold air incursion into the top of the cooling tower are identified and the detailed experimental data are presented. The experimental data show that this effect could seriously impair the thermal performance of the cooling tower. The water outlet temperature of the cooling tower has increased by as much as to 3°C in these tests because of the cold inflow effect. The mechanism and the solution are discussed based on the experimental data. The findings in this paper can lay an important foundation for future small natural draft cooling tower design and operation.

Keywords: Cold inflow, Natural draft dry cooling tower, Experimental study

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Nomenclature

A	Area (m^2)
C_p	Specific heat (J kg^{-1} K^{-1})
d	Diameter (m)
Fr	Froude number
f	Friction factor
h	Heat transfer coefficient (kW/m^2 K)
H	Height, elevation (m)
K	Flow resistance
L	Length (m)
$\dot{m}$	Mass flow rate (kg/s)
n	Number
Nu	Nusselt number
p	Pressure (Pa)
Pr	Prandtl Number
Q	Heat transfer rate (kW)
q	Heat flux (kW m^{-2})
Re	Reynolds Number
T	Temperature (°C)
U	Overall heat transfer coefficient (kW/m^2 K)
v	Velocity (m s^{-1})

Greek letters

ρ	Density, mean density (kg m^{-3})
Δ	Property difference

Subscripts

a, w	Air side, water side
bot	bottom measurement level
e	Effective
i, o	Inside or inlet, outside or outlet
mid	middle measurement level
t	Tube
tow	Tower
top	top measurement level

1. Introduction

Cooling towers are a core component of thermal power plants [1]. While the wet cooling towers use the water evaporation to discharge the heat, dry cooling towers transfer the heat to air. The extended surfaces or finned tubes in the water-to-air heat exchangers offer large air contact areas. In a natural draft dry cooling tower (NDDCT), the airflow through the heat exchanger is created by the density difference between the hot air inside the tower and the ambient air outside the tower. This cooling technology can effectively discharge the heat without consuming water and virtually with no parasitic power consumption. NDDCTs are therefore believed to be the cost effective option for the power plants located in the arid area [2, 3].

The performance of all air-cooled heat exchangers and cooling towers are affected by the changes in ambient conditions [2, 4, 5]. Changes in air temperature, air humidity, crosswinds, rain, snow, hail and the solar radiation all affect the performance of NDDCT to a greater and less extent. Two ambient factors, the hot ambient temperature and the crosswind, are considered most significant for NDDCTs and received a lot of attention in recent years [6-9]. He *et al* [7] investigated the performance of the NDDCT at different ambient conditions and proposed the wetted-medium pre-cooling technology to cool the air when the ambient air is hot and dry. Fahmy and Nabih [10] investigated the impact of ambient air temperature and the heat load variation on the performance of air-cooled heat exchangers in a LNG plant. Li *et al* [11, 12] tested the performance of a 20-m tall NDDCT under different ambient conditions and then validated with the numerical model. Kroger *et al* [2, 13] summarized the performance of several industrial cooling towers under windy conditions and discussed the effect of heat exchanger arrangement and wind-break walls on the performance of NDDCT subjected crosswind. Yang *et al* [8, 14, 15] discussed the dimensional characteristics of wind effects on the performance of indirect dry cooling system with both vertically and horizontally arranged heat exchanger bundles. Zhao *et al* [16, 17] simulated the cooling performance of a dry cooling tower with vertical two-pass column radiators under crosswind.

Phenomena such as low pressure and vortex flow often occur around cooling towers, affecting the performance of the cooling tower. Re-entry of hot air back into the tower were reported for mechanical cooling towers [18-20]. This strange airflow behaviour is usually due to the limited space around the cooling tower. The unfavourable flow interaction occurs and result in the recirculation of the hot exhaust air. On the other hand, for natural draft cooling towers and chimneys having a relatively slow airflow inside, instability could exist at the top of the tower. The instability manifests itself with external cold air intruding into the tower. This phenomenon has been reported in chimneys and cooling towers where the air velocity is not sufficiently high. Jörg and Scorer [21] demonstrated this phenomenon by simulating the cold inflow in a water tank and with some supplementary investigations in air. They developed a correlation to predict the cold inflow to a tube based on their experimental result. Sparrow *et al* [22] reported the cold inflow during natural convection in a one-side heated open ended vertical channel. The cold air reversals were observed near the top of the channel. Modi and Torrance [23] investigated the cold inflow at the exit of buoyant channel flows. They discussed the influence of the Reynolds number and Froude number on the structure of cold inflow at moderate Reynolds number. According to their research, the cold inflow is associated with the premature separation of the wall boundary layer in a buoyant channel flow. Fisher and Torrance [24] quantified the cold inflow effect on the chimney–enhanced free convection experiments. Their results indicate that the overall heat transfer is approximately decreased by 4 percent because of the cold inflow effect. Chu et al. [25-27] studied the effect of cold inflow on chimney height of natural draft cooling towers. They also proposed the wire mesh to prevent this cold air from sinking into the chimney duct. However, all the above researches on the cold inflow effect were based on the lab-scale or numerical models. No detailed full-scale experimental data was reported and the cooling tower performance hasn't been connected with the effect of the cold inflow. In fact, very few people paid attention to this phenomenon in the recent cooling tower research and it hasn't been mentioned in the recent cooling tower experimental research [20, 28-35].

The authors' research group has been developing small NDDCT technology for small-scale (1-30 MW) concentrating solar thermal (CST) power generation. Existing cooling tower designs optimised for large steam power cycles are not optimal for such relatively small plants. The NDDCTs for small CST power plant would be shorter and would have slower moving air. Therefore, they are more vulnerable to cold inflow. In this research, a real renewable power plant size of 20-m NDDCT test system was developed and built at the University of Queensland. The performance of this cooling tower was tested at different ambient conditions and cold air incursion was observed in some of the tests. The experimental data of cold air incursion from the top of the cooling tower are presented in this paper. The mechanism of the cold inflow effect is explained by considering the negative effect of the cold air inflow on natural draft process and the air momentum exchange. A possible solution of this effect is proposed which can provide further assistance for future small cooling tower design.

2. Experimental System

2.1 Tower structure and the air-cooled heat exchanger

Due to the relatively simple and low cost of the construction, this experimental cooling tower is constructed with a lightweight PVC membrane supported by a steel truss. This tower is 20 m high overall and 12.52 m in diameter at both the heat exchanger level and tower outlet level. The tower inlet height is 5 m and the waist diameter is about 9.7 m. 18 air-cooled heat exchanger bundles are horizontally installed at the inlet of the cooling tower which covered 70% of the inlet area of the cooling tower. Fig. 1 shows the overall size of the experimental cooling tower and the heat exchanger layout. More details of the heat exchanger bundle can be found in literature [12].

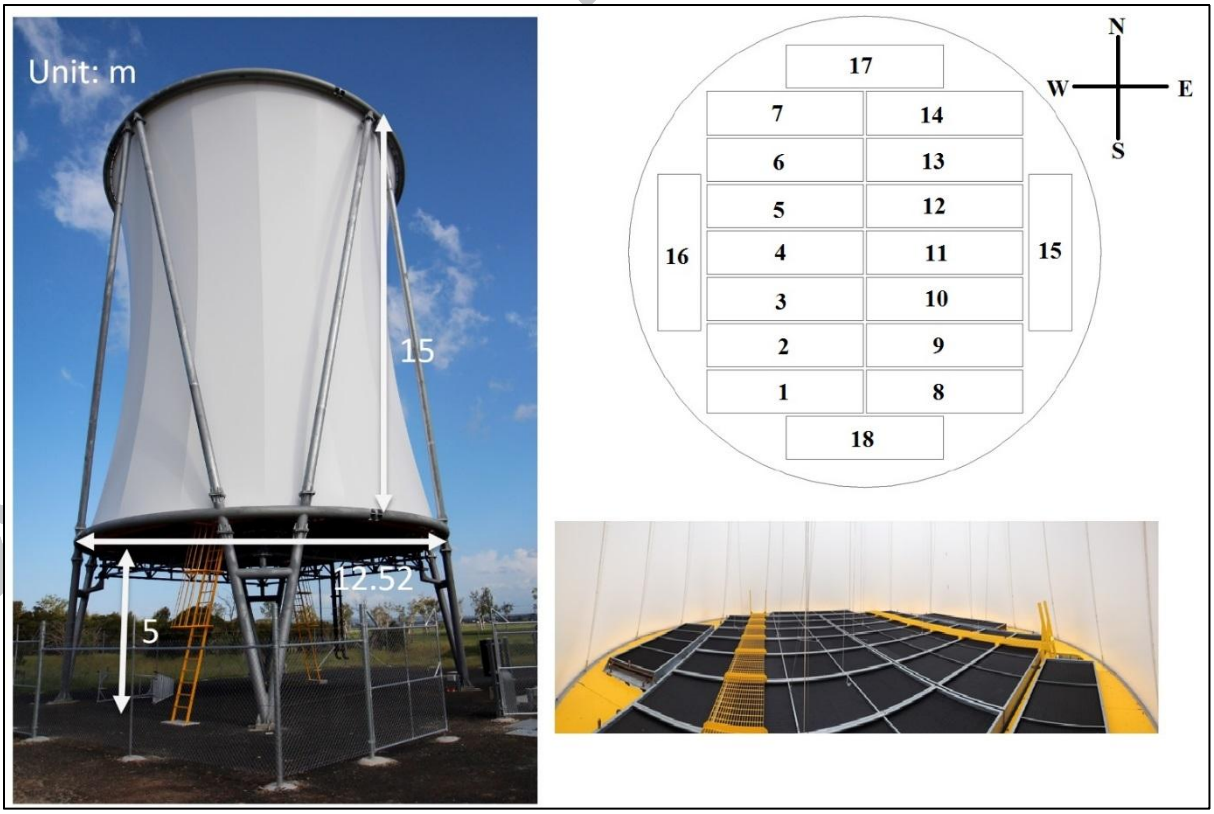


Fig. 1. Cooling tower configuration and heat exchanger layout

2.2 Heater unit and instrumentation

The heat input of the experimental tower is supported by a diesel-fired non-condensing boiler. By controlling the mass flow rate of the diesel and the pressure of the combustion room, the heat output of the heater can be controlled to be 400 kW, 600 kW and 840 kW.

In order to get the performance of the cooling tower in real conditions, an extensive instrumentation and data acquisition system were installed for the experimental system. For the air side properties, 4 layers of testing sensors are installed at different locations of the cooling tower, as shown in Fig. 2. Each layer owns 9 temperature and humidity sensors. For the waterside measurement, each heat exchanger bundle is installed with one temperature sensor at the bundle inlet and another at the bundle outlet. The overall mass flow rate of the hot water is controlled by a variable speed pump and monitored by a mass flowmeter. The overall water inlet and outlet temperatures are measured by two temperature sensors at the inlet and outlet of the cooling tower. Environmental factors such as the wind speed, wind direction, solar irradiation are collected by a separate weather station located nearby the cooling tower.

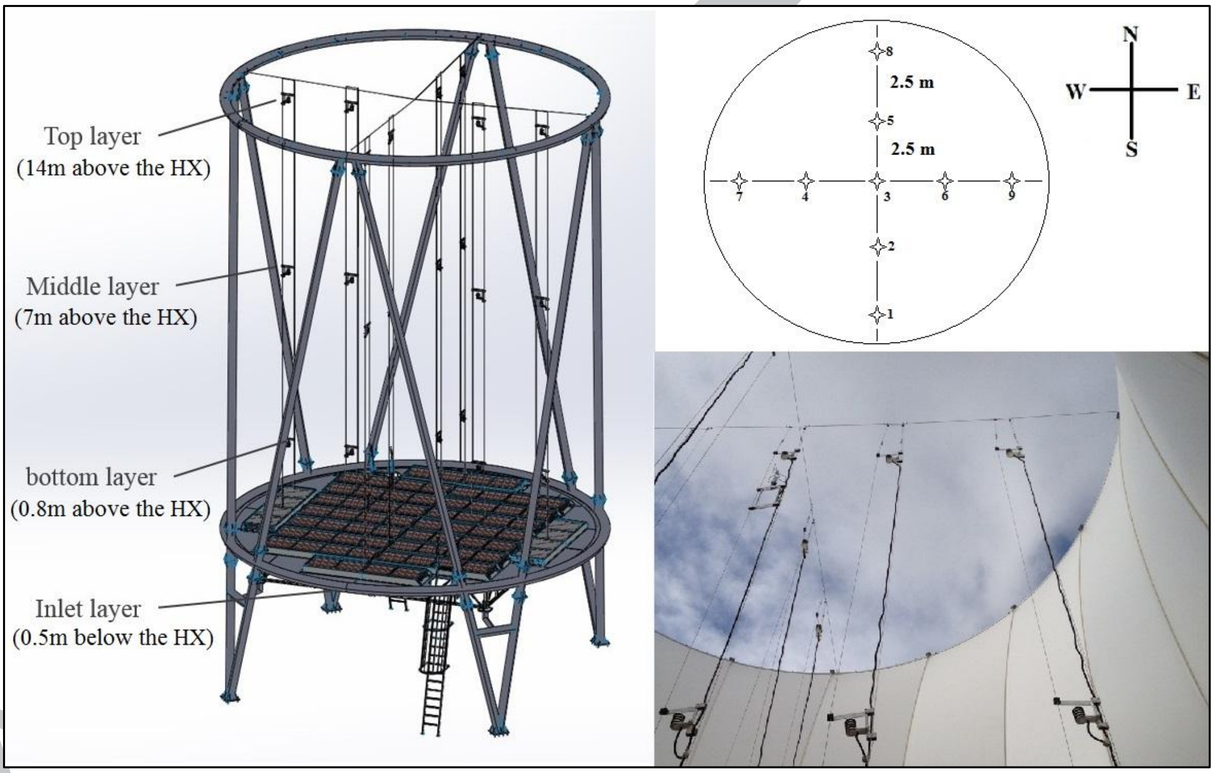


Fig. 2. Air temperature sensor layout

Table 1 provides a list of the sensors used in the experimental system. All the sensors used in this study were calibrated before the test was started. The uncertainty analysis of the measurements is carried out based on the Type A evaluation of standard uncertainty [36]. The data acquisition system was designed using the National Instrument CRIO real time data logging and analysis system. This system uses 3 remote base stations that communicate via fiber optic with a dedicated PC. All the sensor data are recorded once every 1 s.

Table 1: The sensor specs of the experimental system

Sensors/instruments	Supplier	Measuring range	Uncertainty/ accuracy	Quantities of the sensor
Air temperature	Thermistor	0-150°C	±0.2°C	36
Air humidity	Vaisala	0-100% RH	±5%	36
Water temperature	TC Direct	0-90°C	±0.5°C	38
Water mass flow	Krohne	0-20 kg/s	±0.46 kg/s	1
Crosswind velocity	Vaisala	0-60 m/s	±3%	2
Wind direction	Vaisala	-	±3%	2

3. Experimental data of the cold air inflow

3.1 General performance of the cooling tower

For a NDDCT, the driving force of the heat exchange process is the temperature difference between the hot water and the ambient air; the cold air through the heat exchanger is driven by the density difference between the hot air inside the tower and the cold air outside the cooling tower. Thus the ambient temperature has a great influence on the performance of the cooling tower. Fig. 3 shows the general performance of this cooling tower at different ambient temperature. The heat rejection rate in this test is controlled to be 840 kW and the total water mass flow rate is 15 kg/s. As shown in Fig. 3, in general, the overall inlet and outlet water temperature increase with the increase of the ambient temperature. The average water temperature difference is unchanged because the cooling tower is running with constant heat load. The overall cooling tower water outlet temperature increased from 25°C to 37°C when the ambient temperature ranges from 11°C to 28°C.

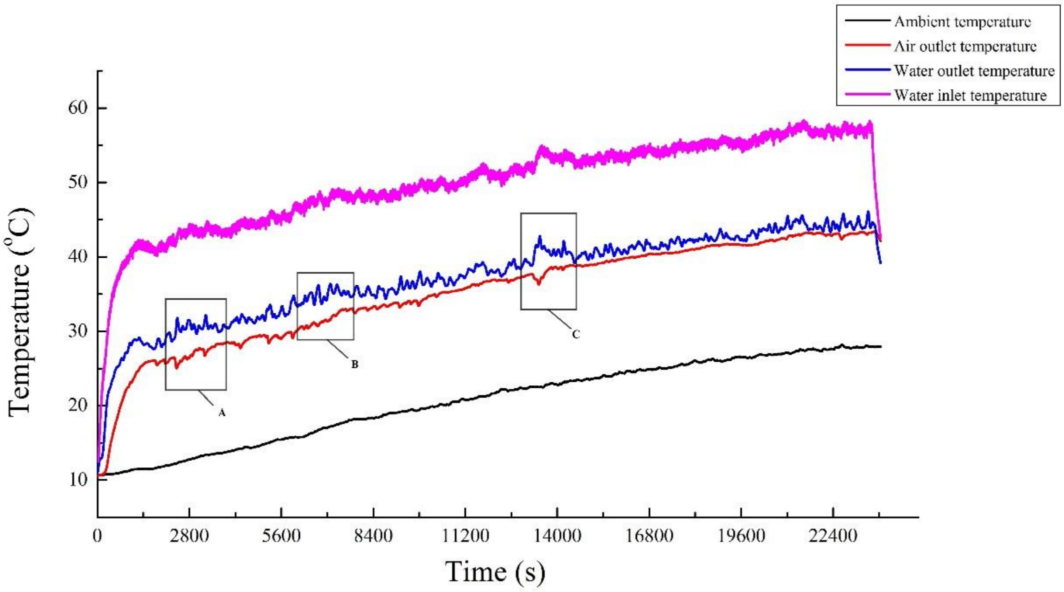


Fig. 3. General performance of the cooling tower at different ambient temperatures

3.2 Effect of the cold air inflow

For the NDDCT worked in the above test ambient conditions, the performance of the tower is influenced by several factors simultaneously. Thus the water inlet and the outlet temperatures experienced several oscillations during the test. In phase A, B and C shown in Fig. 3, the performance of the cooling tower suffered several large turbulent disruption. The hot water outlet temperature increases up to 3°C compared with the steady state performance.

Figs. 4-6 presents the detailed air temperature distribution inside the cooling tower during phase A defined in Fig. 3. The timing of these figures are relative to the start of Phase A. The numbers for each line present the location of the air temperature sensors as defined in Fig. 2. According to the experimental data, the air temperature inside the cooling tower suffered several obvious turbulent during the test time. At about 140 s, 400 s, 700 s, and 1600 s, the air temperature of all the three levels inside the cooling tower experienced several significant decreases. The air temperature suffered the greatest decrease between 600 s to 800 s. As shown in Figs. 4-6, from 600 s to 700 s, the air temperature of the top and the middle levels slightly increased, while the air temperature at bottom layer remained relatively stable. At about 700 s, most of the air temperature of the top level started to decrease, then followed the middle layer at about 729s and the bottom level at about 740 s. The timing of the air temperature variation indicated the cold air was coming from the top of the cooling tower and then penetrated to the bottom layer of the cooling tower.

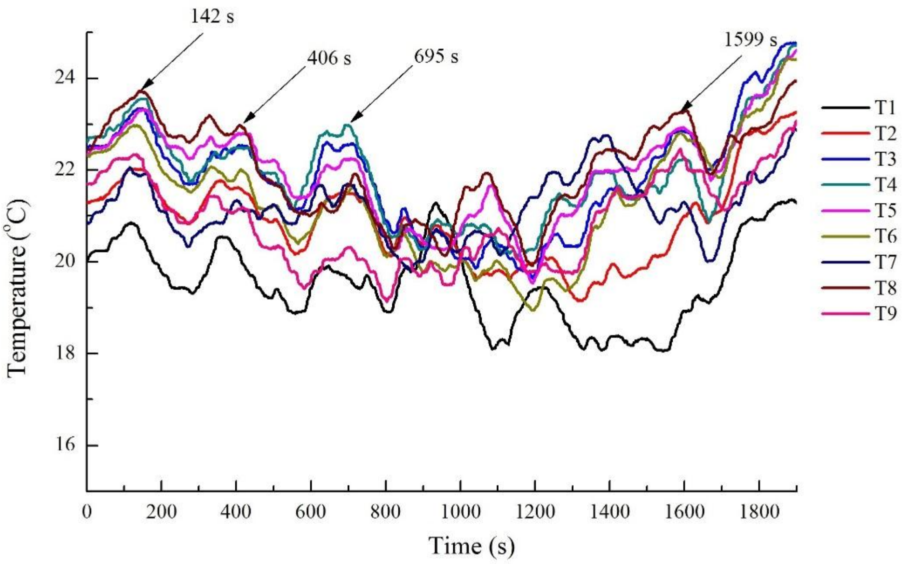


Fig. 4. Air temperature distribution at the top level: Phase A

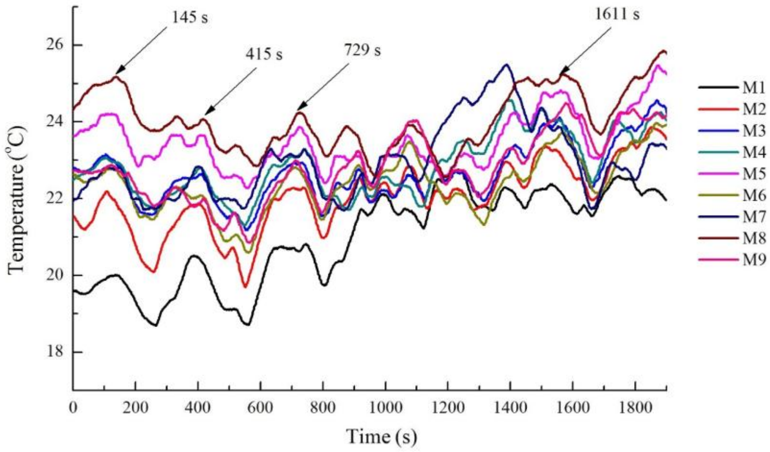


Fig. 5. Air temperature distribution at the middle level: Phase A

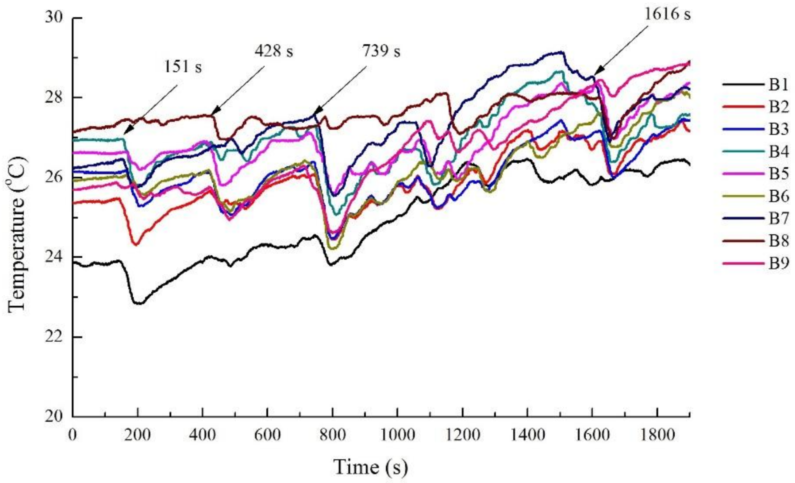


Fig. 6. Air temperature distribution at the bottom level: Phase A

Fig. 7 presents the detailed tower performance during the Phase A in Fig. 3. Fig. 7 (a) shows general performance of the cooling tower and the average air temperature of the three measurement levels inside the cooling tower. Fig. 7 (b) gives the detailed information of the crosswind. As can be seen in this figure, water temperature and the air temperatures at all 3 levels are relatively stable at the beginning of the test. The average air temperature at the top level and the middle level are almost the same. However, the air temperatures of all three layers inside the cooling tower slightly decreased at about 140 s. After that, there is a slight temperature difference between the top and the middle. At about 400 s and 700 s, the average air temperatures of all three levels significantly decreased again and caused an increase of the temperature gap between the middle and the top layers, which lasted until the end of the test. During the test time, the water outlet temperature increased at 140 s, 400 s, 700 s, and 1600 s, which can be perfectly matched against the timing of the cold air incursion. Thus we conclude that the cold inflow from the top is the main reason why the water outlet temperature increased periodically during the test. The similar phenomenon can also be found in Phases B and C in Fig. 3.

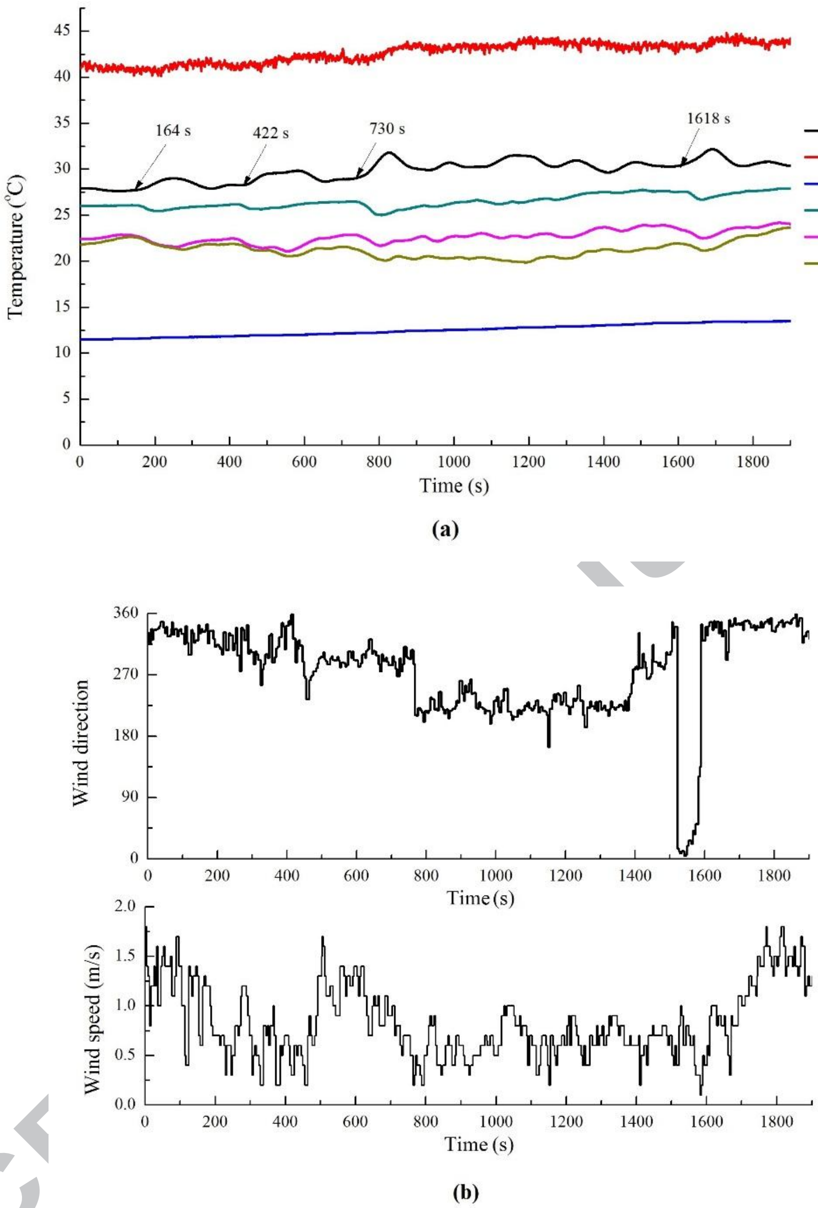
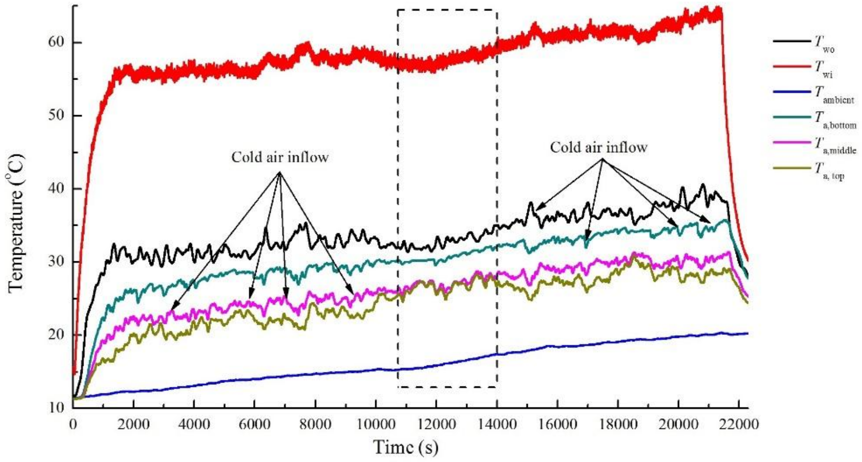


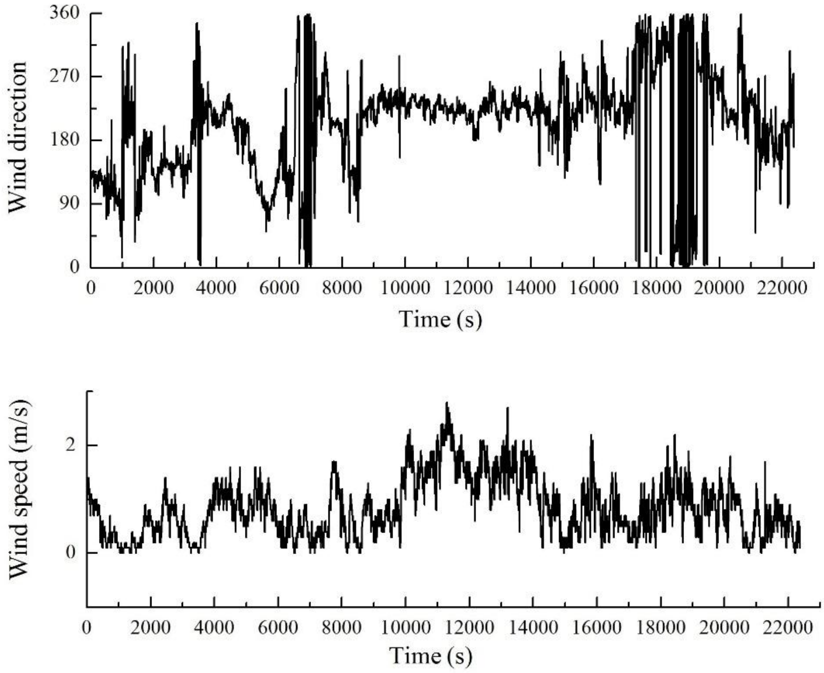
Fig. 7. The detailed performance of the cooling tower suffering cold air inflow:

(a) Tower performance (b) Wind condition

Fig. 8 gives another long time test data of the general performance of the cooling tower. In this test, the water mass flow rate was 7.25 kg/s with 840 kW heat input into the system. The cooling tower suffered a number of cold air incursions in this period as marked in Fig. 8. As in Fig. 7, the cold air incursion led to a reduction of the air temperature inside the cooling tower and a significant temperature gap between the middle and the top levels of the cooling tower. To keep the heat rejection constant, the hot water temperature had to be increased by the control system to compensate for the negative effect of the cold air inflow.



(a)



(b)

Fig. 8. The performance of the cooling tower suffering cold inflow:

(a) Tower performance (b) Wind condition

An interesting observation about the experimental data plotted in Fig. 7 and 8 is that the effect of cold inflow becomes less significant with the increase of the crosswind. As shown in Fig. 7, at about 600 s and 1800 s, the wind speed is significantly higher than the rest of the interval. The air temperature inside the tower is the highest in this period. Similarly, in the 11000 s to 14000 s in Fig. 8, where the crosswind speed is the highest during the test, the effect of cold air incursion is reduced. This region is marked by the shaded rectangle in Fig. 8. The air temperature in the middle of the cooling tower equals to the air temperature of the top and the performance of the cooling tower is the best during this test interval. That's why water temperatures decreased a little even with the increase of the ambient temperature.

Table 2 gives the experimental data of the cold air inflow effect on the performance of the cooling tower. ΔT_{air} is the average air temperature change of the 27 temperature sensors inside the cooling tower during cold air inflow while ΔT_{water} is the average water outlet temperature change compared with the steady state performance. Before the cold incursion occurs, there are intervals in which the general performance of the cooling tower is stable in short time (around 2-3 minutes) with the water temperatures staying unchanged. The average value calculated based on these intervals is defined as the steady state performance. The numerical values were computed for cold inflow events observed in the experimental traces given in Fig. 3. The approach temperature is selected to evaluate the effect of the cold inflow. The approach temperature is defined as

$$T_{\text{approach}} = T_{\text{wo}} - T_{\text{ai}} \tag{1}$$

The lower approach temperature means the hot fluid temperature can be better matched with the cold fluid and the thermal performance is better. As can be seen in the table, the approach temperature is increased with the increase of ΔT_{air}. This indicates the performance of the cooling tower is worse with more cold air getting into the cooling tower from the top.

Table 2 Cold air incursion effect on the performance of the cooling tower

Measure points	1	2	3	4	5	6
$\Delta T_{,\text{air}}$ (°C)	0	0.71	1.01	1.25	1.52	2.13
$\Delta T_{,\text{water}}$ (°C)	0	1.81	2.27	2.30	2.59	2.99
Approach (°C)	15.10	16.81	17.37	17.40	17.69	18.09

4. Discussion

4.1 Mechanism of the cold air incursion on tower performance

In a NDDCT, the air is forced through the heat exchanger by the density difference between the ambient air and the hot air inside the cooling tower. With the cold inflow at the top of the cooling tower, the air density difference is decreased making the driving force of the airflow smaller. So the performance of the cooling tower is negatively affected by the cold air inflow.

Eq. (2) is the draft equation of NDDCT, the right side of this equation presents the driving force of the air flow while the left side represents the total flow resistance [2].

$$-\rho_{\text{a}} v_{\text{a}}^2 K_{\text{tow}}/2 \approx \left(\rho_{\text{a,tow}} - \rho_{\text{a,amb}}\right) \text{g} (H_T - H_{in}) \tag{2}$$

In this equation, K_{tow} is the total loss coefficient of the cooling tower, v_{a} is the average air velocity inside the cooling tower, ρ_{a} is the mean air density before and after the air pass through heat exchanger, $\rho_{\mathrm{a,amb}}$ is the density of the ambient air and $\rho_{\mathrm{a,\,tow}}$ is the average air density inside the cooling tower. The average air densities are calculated by the air temperatures, which are computed using the average value measured by each temperature sensor.

The air mass flow rate inside the cooling tower can be expressed by the following equation:

$$\dot{m}_a = v_a \rho_a A \tag{3}$$

and the air density can be calculated by

$$\rho = \frac{P}{RT} \tag{4}$$

Assume K_{tow} is constant, and by substituting Eq. (3) and Eq. (4) to Eq. (2), find

$$\frac{\dot{m}_a'}{\dot{m}_a} = \sqrt{\frac{\left(\rho_{\mathrm{a,amb}}^2 - \rho_{\mathrm{a,\,tow}}'^{\,2}\right)}{\left(\rho_{\mathrm{a,amb}}^2 - \rho_{\mathrm{a,\,tow}}^2\right)}} = \sqrt{\frac{1 - \frac{T_{\mathrm{a,amb}}^2}{T_{\mathrm{a,tow}}'^{\,2}}}{1 - \frac{T_{\mathrm{a,amb}}^2}{T_{\mathrm{a,tow}}^2}}} \tag{5}$$

where $\dot{m}_a'$, $\rho_{\mathrm{a,\,tow}}'$ and $T_{\mathrm{a,tow}}'$ are the air mass flow rate, air density inside the cooling tower and the average air temperature inside the cooling tower under the cold air incursion condition.

For the air-cooled heat exchanger, the energy balance equation is given by

$$Q = m_{\mathrm{a}} c_{\mathrm{pa}} (T_{\mathrm{ao}} - T_{\mathrm{ai}}) = m_{\mathrm{w}} c_{\mathrm{pw}} (T_{\mathrm{wi}} - T_{\mathrm{wo}}) \tag{6}$$

and

$$Q = UAF_{\mathrm{T}} \frac{(T_{\mathrm{wi}} - T_{\mathrm{ao}}) - (T_{\mathrm{wo}} - T_{\mathrm{ai}})}{\ln\left[\frac{(T_{\mathrm{wi}} - T_{\mathrm{ao}})}{(T_{\mathrm{wo}} - T_{\mathrm{ai}})}\right]} \tag{7}$$

The overall heat transfer coefficient of the air-cooled heat exchanger can be calculated by Eq. (8)

$$UA = \frac{1}{\frac{1}{h_{\mathrm{a}} A_{\mathrm{a}}} + \frac{\ln\,(d_0/d_{\mathrm{i}})}{2\pi k_{\mathrm{t}} L_{\mathrm{t}}} + \frac{1}{h_{\mathrm{w}} A_{\mathrm{w}}}} \tag{8}$$

where h_{a} is the air side heat transfer coefficient, A_{a} is the total air side heat transfer area, k_{t} is the thermal conductivity of the tube, L_{t} is the length of the tube, h_{w} is the water side heat transfer coefficient, A_{w} is the water side heat transfer area

The following equations were proposed by Kroger [2] to calculate the waterside heat transfer coefficient in the heat exchanger:

$$Nu = \frac{\left(\frac{f_D}{8}\right)(Re_w - 1000)Pr\,[1 + \left(\frac{d}{L}\right)^{0.67}]}{1 + 12.7\left(\frac{f_D}{8}\right)^{0.5}(Pr^{0.67} - 1)}$$

(9)

$$h_w = \frac{Nuk}{d}$$

(10)

where f_D is the friction factor inside the tube and can be expressed by the following equation:

$$f_D = 0.3086[\log_{10}\{\frac{6.9}{Re_w} + (\frac{\varepsilon/d}{3.7})^{1.11}\}]^{-2}$$

(11)

The air side heat transfer coefficient is provided by the heat exchanger manufacture and refined by the experiment data [12],

$$h_a A_a = 0.0143{Re_a}^2 + 83.2Re_a + 22210 \quad (12)$$

where Re_a is the Reynolds number of air. The characteristic length of Re_a is the equivalent circular diameter of the airflow passage, which is 0.017 m for this particular heat exchanger

Based on above equations, the effect of the cold air inflow on the natural draft process can be calculated using the following process;

1) Input the ambient condition, get the air temperature change from the experimental data.
2) Use the equation 2-5 to get the mass flow rate ratio $\dot{m}_a'/\dot{m}_a$ and combine $\dot{m}_a'/\dot{m}_a$ with equations 6-12 to get the performance of the heat exchanger performance under the cold air inflow
3) Output T_{wo}'

Fig. 9 gives the comparison between the experimental data and the above analyse. The blue line in the figure is the linear correlation produced from the experimental data represented by the discrete points while the red line is the modelling result using the above simulation method. As can be seen in this figure, there is an obvious gap between the experimental data and the modelling prediction. The above numerical modelling underestimated the effect of cold air inflow, which indicates there might be another reason which could negatively affect the heat exchange process.

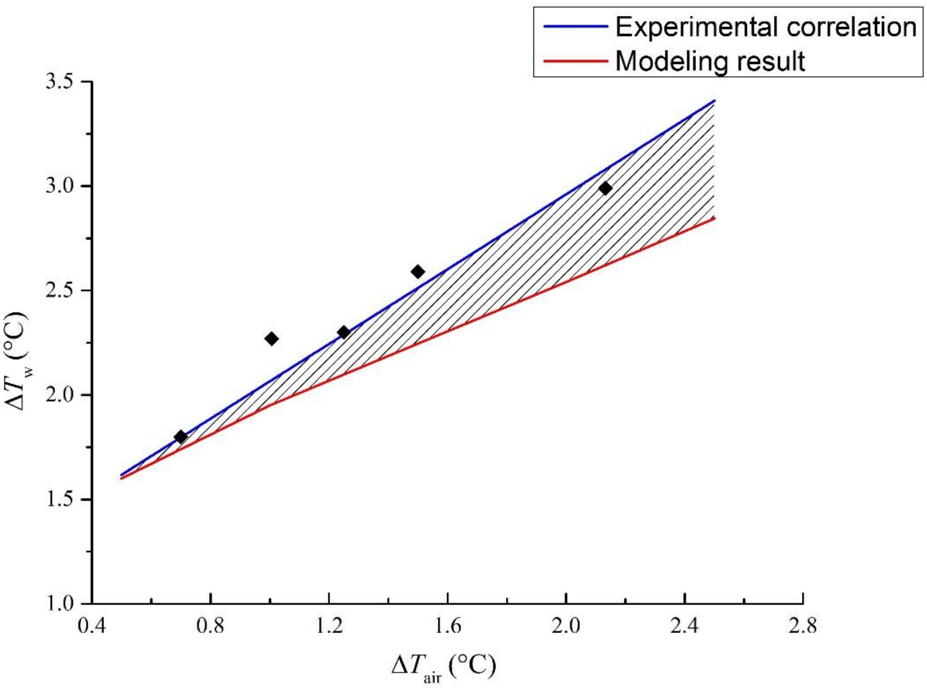


Fig. 9. Cold incursion effect comparison

According to the experimental data of section 3, the cold air comes in at the top and then penetrates into the bottom layers. While doing so, it meets the hot air rising to the top. When two fluid streams collide, a flow resistance is formed due to the momentum exchange between them. This further decreases the air mass flow rate through the heat exchanger, consequently the thermal performance of the cooling tower. In Fig. 9, the large ΔT_{air} indicates a large cold air mass flow rate while the small ΔT_{air} does the opposite. When ΔT_{air} is small, the mass flow rate of the cold air from the top is small. The effect of the momentum exchange on the air mass flow rate through the heat exchanger is small. Increase of the water temperature is mainly due to the decreased natural draft effect. That's why the blue line and the red line are very close when ΔT_{air} is small. With the increase of the cold air mass flow rate from the top of the cooling tower, a large ΔT_{air} is formed as well as a greater flow resistance for the air through the heat exchanger. Thus in high ΔT_{air} condition, the cold air inflow decreased the natural draft effect and at the same time formed another flow resistance for the hot air. These two factors work together to influence the performance of the cooling tower.

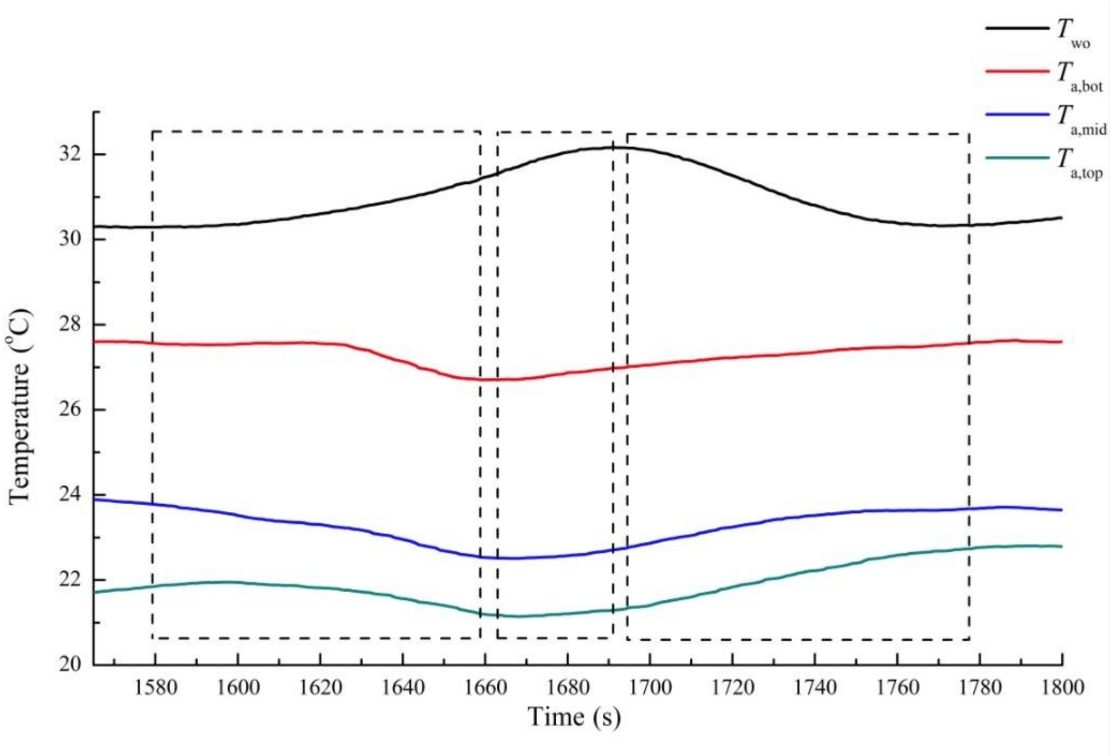


Fig. 10. Single process of the cold air incursion

Fig. 10 presents a single process of the cold air incursion. As shown in this figure, this effect can be divided into 3 stages. At the first stage, the cold air begins to get in the cooling tower from the top of the tower. The average air temperatures inside the cooling tower start to decrease, so as the natural draft effect. Due to the decrease of the driving force for the air flow and an extra flow resistance caused by the air momentum exchange, less air is sucked into the cooling tower, which makes the situation worse. Thus the air temperature inside the cooling tower keeps decreasing while hot water temperature keeps increasing in this stage. This is however a self-correcting process. The increased hot water temperature improves the driving force. The second stage in Fig. 10 represents this reversal, when the air temperature inside the cooling tower reaches the minimum. The cooling tower suffers the minimum natural draft effect and the air mass flow rate is the smallest during this interval. In the third stage, because of the increased hot water temperature, the average air temperature inside the cooling tower starts to increase again and improves the natural draft effect. The performance of the cooling tower is becoming better in this stage. With more moving air go through the heat exchanger, the water temperature decreases and finally the steady state is reached again.

4.2 Reason and solution

As demonstrated by Jörg and Scorer [21], if the buoyant fluid in an open topped vessel is in a uniform environment, the surrounding fluid will flow in and replace the buoyant fluid unless the upward velocity is large enough. That's why the cold inflow was observed at the top of the cooling towers and chimneys with relatively slow moving air. The driving force of the cold inflow is the density gradient between the heated air and the unheated ambient air. For cooling tower running in the steady state, the total pressure inside the top of the cooling tower is equal to the total pressure outside the cooling tower if the upward velocity inside the cooling tower is uniform. However, as shown in Fig. 2, the flow passage at the top of the Gatton tower is divergent. According to Sparrow's [37] research about flow separation, when fluids flow through in a diverging passage, the flow regime may change and the fluids may not be able to follow the contour of the bounding walls. This effect may form a low speed zone near the divergent wall boundary layer at the upper part of Gatton tower and makes the inside total pressure smaller than the outside, result in the cold inflow.

In continuum mechanics, the Froude number (Fr) is a dimensionless number defined as the ratio of the flow inertia to the external field which can be expressed as

$$Fr = v^2/gl \tag{13}$$

where v is a characteristic flow velocity, g is in general a characteristic external field, and l is a characteristic length.

In cooling tower area, the previous research has proposed a densimetric Froude number based on the tower outlet diameter [2].

$$\frac{1}{Fr_D} = (\rho_{\mathrm{a}} - \rho)gd/\rho v^2 \tag{14}$$

where ρ_{a} is the ambient air density at the elevation of the tower outlet and ρ is the density of the air leaving the tower

According to the research by Lucas and Richter [38] and Richter [39], the air flow at the tower outlet tends to become increasingly more unstable with the increase of the $1/Fr_{\mathrm{D}}$. Cold air inflow is entrained by plume for $3.05 < 1/Fr_{\mathrm{D}} < 6$ while the cold air penetrated to the heat exchanger level occurs when $1/Fr_{\mathrm{D}} > 7$. For this experimental cooling tower, the air velocity at the tower outlet is about 0.7 m/s when the ambient temperature is 30°C and the hot water inlet temperature is 55°C. Under this operation condition, the value of $1/Fr_{\mathrm{D}}$ for Gatton tower is about 10. For a large industrial-scale cooling tower (58 m x 83 m x 120 m, top diameter x bottom diameter x height) operated at the same condition [2], the air velocity at the tower outlet is around 3 m/s. The value of $1/Fr_{\mathrm{D}}$ in big tower is 3.8 times smaller than Gatton tower, result in a better resistance on the cold air incursion. That is probably why this behaviour didn't get enough attention in the past because past cooling practice and analysis were limited to towers much taller.

For conventional big cooling towers at heights of up to more than 100 m and with a relatively great wind load at the top structure, the reinforced concrete columns are used to support the tower. The structural strength and stability of the tower shell are the first concern for the geometry design of the cooling tower [40]. On the other hand, for a small cooling tower contains a mass of slowly moving air that is slightly buoyant relative to the surrounding air. The tower performance is susceptible to the turbulence at the top of the cooling tower. So the influence of the cold air inflow should be given extra attention in the small cooling tower design. The outlet diameter of the cooling tower has a significant influence on the behaviour of cold air inflow. Decrease of the tower outlet diameter could accelerate the air outlet velocity and therefore decrease the value of $1/Fr_D$ exponentially. A converging tower outlet is recommended for small cooling tower since this shape can overcome the problem of the cold air inflow by accelerate the air speed at the upper part of the cooling tower and avoid the effect of flow separation. However, the convergence shouldn't be too excessive because the dynamic loss increases with the decrease of the tower outlet diameter.

5. Conclusion

In this research, the performance of a 20-m NDDCT (real size cooling tower for small renewable power plants) was tested in different ambient conditions and the phenomenon of the cold air incursion from the top of the cooling tower was observed. The detailed experimental data of the ambient condition, air temperature distribution inside the cooling tower and the variation of the hot water temperature are presented.

Repeated cold air incursion events were observed that cause a significant decrease of the air temperature inside the cooling tower. The water outlet temperature can be increased up to 3°C as a result of these events. Further analysis of the cold inflow mechanism shows that this process operates by decreasing the driving force and also forming an extra flow resistance for the airflow through the heat exchanger. The performance depression of the cooling tower is inversely correlated by a densimetric Froude Number based on the tower outlet diameter. This phenomenon has been observed even in tall towers in the past but the effect was not significant and did not receive much attention. Our results show that cold inflow phenomenon should be paid extra attention in small cooling tower design. One measure identified in the paper to mitigate this effect is to use a converging tower outlet for small NDDCTs.

ACKNOWLEDGEMENTS

This research was performed as part of the Australian Solar Thermal Research Initiative (ASTRI), a project supported by Australian Government, through the Australian Renewable Energy Agency (ARENA). The first author, Xiaoxiao Li, would also like to thank China Scholarship Council (CSC) for their financial support.

References

[1] G.B. Hill, E.J. Pring, P.D. Osborn, Cooling towers: principles and practice, Butterworth-Heinemann, London ; Boston, 1990.
[2] D.G. Kröger, Air-cooled heat exchangers and cooling towers, Pennwell Corp, Tulsa, Okl, 2004.
[3] H. Gurgenci, Z. Guan, Dirigible natural draft cooling tower in geothermal and solar power plant applications, in: Proceedings of 14th IAHR International Conference on Cooling Tower and Air-Cooled Heat Exchangers, Stellenbosch, 2009.
[4] K. Hooman, Dry cooling towers as condensers for geothermal power plants, Int. Commun. Heat Mass Transfer, 37 (2010) 1215-1220.

[5] Y. Lu, Small natural draft dry cooling towers for renewable power plants, in, Vol. Dissertation/Thesis, The University of Queensland, School of Mechanical and Mining Engineering, 2015.
[6] Z. Zou, Z. Guan, H. Gurgenci, Y. Lu, Solar enhanced natural draft dry cooling tower for geothermal power applications, Sol. Energy, 86 (2012) 2686-2694.
[7] S. He, H. Gurgenci, Z. Guan, A.M. Alkhedhair, Pre-cooling with Munters media to improve the performance of Natural Draft Dry Cooling Towers, Appl. Therm. Eng., 53 (2013) 67-77.
[8] L.J. Yang, X.P. Wu, X.Z. Du, Y.P. Yang, Dimensional characteristics of wind effects on the performance of indirect dry cooling system with vertically arranged heat exchanger bundles, Int. J. Heat Mass Transfer, 67 (2013) 853-866.
[9] Y. Lu, H. Gurgenci, Z. Guan, S. He, The influence of windbreak wall orientation on the cooling performance of small natural draft dry cooling towers, Int. J. Heat Mass Transfer, 79 (2014) 1059-1069.
[10] M.F.M. Fahmy, H.I. Nabih, Impact of ambient air temperature and heat load variation on the performance of air-cooled heat exchangers in propane cycles in LNG plants – Analytical approach, Energy Convers. Manage., 121 (2016) 22-35.
[11] X. Li, Z. Guan, H. Gurgenci, Y. Lu, S. He, Simulation of the UQ Gatton natural draft dry cooling tower, Appl. Therm. Eng., 105 (2016) 1013-1020.
[12] X. Li, S. Duniam, H. Gurgenci, Z. Guan, A. Veeraragavan, Full scale experimental study of a small natural draft dry cooling tower for concentrating solar thermal power plant, Appl. Energy, 193 (2017) 15-27.
[13] A.F. du Preez, D.G. Kröger, The effect of the heat exchanger arrangement and wind-break walls on the performance of natural draft dry-cooling towers subjected to cross-winds, J. Wind Eng. Ind. Aerodyn., 58 (1995) 293-303.
[14] L.J. Yang, L. Chen, X.Z. Du, Y.P. Yang, Effects of ambient winds on the thermo-flow performances of indirect dry cooling system in a power plant, Int. J. Therm. Sci., 64 (2013) 178-187.
[15] H.T. Liao, L.J. Yang, X.Z. Du, Y.P. Yang, Influences of height to diameter ratios of dry-cooling tower upon thermo-flow characteristics of indirect dry cooling system, Int. J. Therm. Sci., 94 (2015) 178-192.
[16] Y. Zhao, G. Long, F. Sun, Y. Li, C. Zhang, J. Liu, Effect mechanism of air deflectors on the cooling performance of dry cooling tower with vertical delta radiators under crosswind, Energy Convers. Manage., 93 (2015) 321-331.
[17] Y.B. Zhao, G. Long, F. Sun, Y. Li, C. Zhang, Numerical study on the cooling performance of dry cooling tower with vertical two-pass column radiators under crosswind, Appl. Therm. Eng., 75 (2015) 1106-1117.
[18] G. Ge, F. Xiao, S. Wang, L. Pu, Effects of discharge recirculation in cooling towers on energy efficiency and visible plume potential of chilling plants, Appl. Therm. Eng., 39 (2012) 37-44.
[19] J.H. Lee, M. Moshfeghi, Y.K. Choi, N. Hur, A numerical simulation on recirculation phenomena of the plume generated by obstacles around a row of cooling towers, Appl. Therm. Eng., 72 (2014) 10-19.
[20] W.L. Zhao, P.Q. Liu, Experimental researches of the effect of environmental wind on thermal recirculation under the tower of direct air cooled system, Dongli Gongcheng/Power Engineering, 28 (2008) 390-394.
[21] O. Jörg, R. Scorer, An experimental study of cold inflow into chimneys, Atmospheric Environment (1967), 1 (1967) 645IN3647-3646IN5654.

[22] E.M. Sparrow, G.M. Chrysler, L.F. Azevedo, Observed Flow Reversals and Measured-Predicted Nusselt Numbers for Natural Convection in a One-Sided Heated Vertical Channel, J. Heat Transfer, 106 (1984) 325-332.
[23] V. Modi, K. Torrance, Experimental and numerical studies of cold inflow at the exit of buoyant channel flows, ASME J. Heat Transfer, 109 (1987) 392-399.
[24] T.S. Fisher, K.E. Torrance, Experiments on Chimney-Enhanced Free Convection, J. Heat Transfer, 121 (1999) 603-609.
[25] C.-M. Chu, M.M. Rahman, S. Kumaresan, Effect of cold inflow on chimney height of natural draft cooling towers, Nucl. Eng. Des., 249 (2012) 125-131.
[26] C.C.-M. Chu, R.K.-H. Chu, M.M. Rahman, Experimental study of cold inflow and its effect on draft of a chimney, Advanced Computational Methods and Experiments in Heat Transfer XII, WIT Transactions on Engineering Sciences, 75 (2012) 73-82.
[27] C.C.M. Chu, M.M. Rahman, A Method to Achieve Robust Aerodynamics and Enhancement of Updraft in Natural Draft Dry Cooling Towers, (2009) 817-823.
[28] Q.-d. Wei, B.-y. Zhang, K.-q. Liu, X.-d. Du, X.-z. Meng, A study of the unfavorable effects of wind on the cooling efficiency of dry cooling towers, J. Wind Eng. Ind. Aerodyn., 54–55 (1995) 633-643.
[29] G.M. Amur, Experimental study of wind effects on the airflow of natural draft wet cooling towers, in: 7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Antalya, Turkey, 2010.
[30] K. Wang, F.-z. Sun, Y.-b. Zhao, M. Gao, L. Ruan, Experimental research of the guiding channels effect on the thermal performance of wet cooling towers subjected to crosswinds – Air guiding effect on cooling tower, Appl. Therm. Eng., 30 (2010) 533-538.
[31] Y. Lu, Z. Guan, H. Gurgenci, K. Hooman, S. He, D. Bharathan, Experimental study of crosswind effects on the performance of small cylindrical natural draft dry cooling towers, Energy Convers. Manage., 91 (2015) 238-248.
[32] M. Gao, F.-z. Sun, A. Turan, Experimental study regarding the evolution of temperature profiles inside wet cooling tower under crosswind conditions, Int. J. Therm. Sci., 86 (2014) 284-291.
[33] T. Ning, D. Chong, M. Jia, J. Wang, J. Yan, Experimental investigation on the performance of wet cooling towers with defects in power plants, Appl. Therm. Eng., 78 (2015) 228-235.
[34] M.T. Chaibi, K. Bourouni, M.M. Bassem, Experimental Analysis of the Performance of a Mechanical Geothermal Water Cooling Tower in South Tunisia, American Journal of Energy Research, 1 (2013) 1-6.
[35] M. Lemouari, M. Boumaza, A. Kaabi, Experimental investigation of the hydraulic characteristics of a counter flow wet cooling tower, Energy, 36 (2011) 5815-5823.
[36] S.L. Ellison, M. Rosslein, A. Williams, Quantifying uncertainty in analytical measurement, in: Quantifying uncertainty in analytical measurement, Eurachem, 2000.
[37] E.M. Sparrow, J.P. Abraham, W.J. Minkowycz, Flow separation in a diverging conical duct: Effect of Reynolds number and divergence angle, Int. J. Heat Mass Transfer, 52 (2009) 3079-3083.
[38] M.J. Lucas, J.M. Buchlin, A Study of Cold Air Inflow into a Vertical Heated Tube, in: Fifth International Association for Hydraulic Research Cooling Tower Workshop, 1986, 1986.
[39] E. Richter, Untersuchung der Strömungsvorgänge am Austritt von Naturzugkühltürmen und deren Einfluss auf die Kühlwirkung, in, Vol. Doctoral thesis, Technische Universität, Dresden, 1969.
[40] P. Gould, W. Krätzig, Cooling Tower Structures, in: Handbook of Structural Engineering, Second Edition, CRC Press, 2005.