19th Australasian Fluid Mechanics Conference
Melbourne, Australia
8-11 December 2014

Experimental Study on Inlet Air Cooling by Water Spray for Natural Draft Dry Cooling Towers Enhancement

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Abstract

This paper deals with an experimental investigation of inlet air pre-cooling with water sprays aimed to enhance the performance of Natural Draft Dry Cooling Towers during high ambient temperature periods. An open-circuit wind tunnel with a test 1×1 m^2 test cross section and length of 5.2 m was employed as an approximation to the inlet flow in a Natural Draft Cooling Tower. Experimental measurements of droplet evaporation and air cooling are presented. Nine high pressure, hollow cone nozzles were tested at various droplet sizes, air velocities (1, 2, 3 m/s), and injection rates under different ambient conditions. The water spray was characterized using a Phase Doppler Particle Analyser (PDPA). The effects of drop size distribution and air velocity on droplet evaporation and cooling effectiveness were investigated. The data shows clear trends of cooling with low air velocity or small droplet size distribution. It was found that the spray cooling efficiency, to a large extent, is dependent on spray coverage area. The experimental findings will benefit optimizing spray cooling performance in Natural Draft Dry Cooling Towers and nozzle arrangement.

Introduction

This paper presents results of a study on the effect of droplet size and air velocity on droplet evaporation, and the cooling performance of inlet air spraying to enhance the performance of natural draft dry cooling towers (NDDCTs). This knowledge is crucial for designing spray cooling systems for NDDCTs. NDDCTs has become a necessity for many power plants despite the lower performance (particularly during high ambient temperate periods), owing to water consumption restrictions, environmental regulations and flexibility of plant site selection [1]. Dry cooling towers performance can be enhanced during these periods by various techniques [2].

Water spray cooling is one of the promising technologies where water is sprayed into the inlet air in order to reduce the inlet air temperature. Reducing the cooling media temperature, leads to an increase in overall cycle efficiency, thus allowing some of the performance reduction to be recovered. However, this requires careful consideration of water spray generation to avoid issues related to non-uniform cooling distribution and incomplete evaporation of droplets. Therefore, there is a need to optimize this kind of system under the target conditions. Although a considerable amount of literature has been published about spray cooling on different applications mainly gas turbine fogging and air conditioning, no study exists on analysing spray cooling under conditions typical of NDDCTs. Applications of spray cooling in gas turbine fogging and air conditioning generally have different operating conditions to those for natural draft cooling towers.

Over the past decades, spray cooling has become more popular due to its simplicity, low capital cost, and ease of operation and maintenance [3].

The knowledge of the optimum droplet size is crucial for designing spray cooling systems. According to Wells [4], spray droplet size distribution is a major parameter that impacts droplet movement and evaporation efficiency. Spray cooling performance is also strongly influenced by air velocity [5]. Some of the other important parameters are: (a) cone angle, (b) injection rate, (c) droplet velocity, (d) injection direction, (e) metrological condition [5-8].

Numerical and experimental studies have been carried out on spray cooling performance [1, 9, 10]. In particular, a numerical investigation conducted by Tissot [8] on a small channel has shown that sprays with small droplet size distributions may result in a reverse effect on the spray cooling efficiency due to the compromise effect of momentum exchange and evaporation rate. Moreover, a numerical model was developed by the present authors [3] to investigate spray cooling performance using a uniform drop and velocity distributions at different air velocities. It was found that cooling performance relies mainly on droplet size distribution and air velocity. This work has shown also that there is a trade-off between droplet size and air velocity and the resulting spray dispersion. Based on the numerical results on the previous paper [3], a wind tunnel approximating the inlet flow in NDDCTs was adapted to carry out experimental tests for this study.

In present work, the droplet evaporation, and the resulting cooling of the air are investigated experimentally for a range of inlet air conditions and a number of spray nozzle characteristics. Droplet evaporation and cooling effectiveness are quantified in terms of air temperature and humidity. The effects of drop size distribution and air velocity on droplet evaporation and cooling effectiveness are discussed.

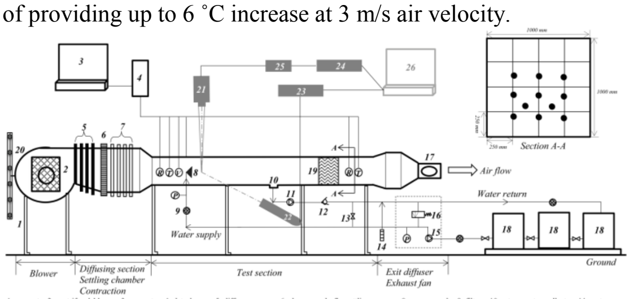
Materials and Methods

Test Section

An open-circuit wind tunnel located at the University of Queensland (Gatton Campus) was employed as an approximation to the inlet flow environment in a NDDCT to perform our spray cooling tests. The wind tunnel setup and associated instruments utilized in this experiment are illustrated in Fig.1. The tunnel overall length is 10 m with test section dimensions of 1 m height, 1 m width, and 5.2 m in length. The experimental rig mainly consists of an air system, a water system, a test section and measurement systems. During the experiment, the air is drawn through a centrifugal blower fan then passes through a diffuser followed by a honeycomb to eliminate flow eddies to approximately provide a uniform air velocity profile throughout the working section. One nozzle is installed in the centre of the working section and directed horizontally in a co-flow direction (air and water moving in the same direction) at a height of 0.6 m and a 0.55 m distance from the contraction cone. The test section side walls are made of transparent acrylic to allow visualization of the water spray.

Air System

Air entering the wind tunnel is drawn directly from the atmosphere. The air is provided via a centrifugal blower fan powered by a 75 kW electric motor. The temperature is controlled utilizing a 24 kW electric air heater which is capable of providing up to 6 ˚C increase at 3 m/s air velocity.



1=supports, 2=centrifugal blower, 3=computer, 4=data logger, 5=diffuser screens, 6= honeycomb, 7= settling screens, 8=spray nozzle, 9=filters, 10=return water collector, 11=water pump, 12=check valve, 13=flow control, 14=direction control, 15=high pressure pump, 16=pressure relief, 17=exhaust fan, 18=water tanks, 19= drift eliminator, 20= Heater, 21= transmitter, 22= receiver, 23= signal processer, 24=Laser, 25= beam splitter, 26=Computer with FLOWSIZER software. R=humidity sensor, T=temperature sensor, V=air velocity sensor, P=pressure sensor.

Figure 1 Schematic diagram of the wind tunnel with PDPA system

Water System

During the experiment, tap water is pumped from the water tank to the spray nozzles using the high pressure water pump. The flow rate delivered to the nozzle is controlled by a bypass valve. Nonevaporated water that accumulates on the tunnel floor is drained out to two drains located at the bottom of the tunnel.

Nozzles

Experiments were conducted using different spray nozzles. Nozzle selection is one of the key elements of the experiment. Main factors that need to be considered in selecting a spray nozzle are: droplet sizes distribution, flow rate and spray cone angle. The selections of appropriate droplet size, liquid flow rate and spray angle are determined by the application and the ambient conditions. All of the nozzles used were hollow cone type as it is the widely used for humidifying applications.

Before selecting the nozzles, the water flow rate required to fully saturate the inlet air was calculated based on the saturation of the area covered by the nozzle rather than the entire tunnel cross section. Depending on the cross-sectional area covered by the nozzle, the required flow rate can be assigned to obtain full saturation of air (theoretically) assuming the cooling process is adiabatic. Knowing the air mass flow rate along with the inlet air ambient conditions, the water mass flow rate required to fully saturate the air can be calculated using the water vapor balance between air and water spray.

Unfortunately, there are not many products that can supply this flow rate at droplet sizes of $D_{v90} \leq 50\mu m$ and with a wide spray angle. The current study uses 9 commercially available nozzles that have a small drop size distribution, have a wide spray angle, and that deliver the required liquid flow rate: (10N, 16N and 22N from H. Ikeuchi & Co. Ltd.), (LNN0.6, LNN1.5, and M6 from Spraying system Co. Ltd.), (UM150 and UM200 from Bete Ltd.) and (A300 from Mist-jet nozzle from Steinen Ltd.).

Instrumentations

Air velocity were measured upstream of the nozzle with two velocity/temperature transmitters (FMA1001R-V2, OMEGA Engineering). The uncertainty in air velocity measurement is 1.5 % of full scale with a working range of 0-5.08 m/s. The inlet and outlet air temperatures were measured with 13 transition joint-style thermistors probes (TJ36-44004-1/8-xx, OMEGA Engineering) with accuracy of ±0.2 ºC. All thermistors were calibrated over a temperature range of 10 ˚C to 50 ˚C. Two thermistors were used for inlet air temperature measurement. For outlet temperature, a grid of 11 thermistors was used. However, due to the large cross section of the tunnel, the measurement grid was positioned on the expected cooling area as shown in Fig.1, (section A-A). A PVC S-shaped drift eliminator was employed upstream of the thermistors to avoid direct contact of the thermistors by unevaporated droplets. Three humidity transmitters (EE-21-FT6-B51, Elektronik GmbH) with the range of 0-100 % and accuracy of 2% RH were used. One humidity sensor was used to measure the humidity of the inlet air. Water pressure was measured using a pressure sensor (P1600-3000, Pace Scientific Inc.) with a working range of 0-20.6 MPA and accuracy of 1% of full scale. Water flow rate was measured using the "bucket and stopwatch" technique in separate calibration runs. All signals from the above sensors are transmitted to a computer by a controlled data acquisition system (UEI DNA-PPC8-1G, United Electronic Industries, Inc., Massachusetts, USA) at a sample rate of 1 Hz for period of approximately 10 mins after attaining steady state condition (10 mins settling time were allowed).

A Phase Doppler Particle Analyser (PDPA) was used for droplet size and velocity measurements. PDPA system is a point measurement technique where it measures droplet size and velocity simultaneously in a small volume which is the interference of the two laser beams. The PDPA uses the light refracted by a droplet moves through the intersection volume to determine the droplet size and velocity [11]. The PDPA used in this study was an Aerometric two-dimensional laser system from TSI, Inc. It consists of 600mW argon-ion laser (532, 561 nm wavelength) transmitter probes, an optical receiver, flow size electronic signal processor and a 2-D traverse system. The focal lengths of the laser transmitter and the receiver unit are 750 mm and 1000 mm, respectively. The laser beams has a beam separation distance of 50 mm and fringe spacing of 8.55 µm. Frequency shifting by the Bragg cells is set to 40MHz. Fig.1 is a schematic illustration of the experimental set-up with the 2D PDPA system. A TSI 2-Axis Traverse System was used to move the PDPA system during the experiment.

Spray Characterisation

Using the two-dimensional phase Doppler particle analyser (PDPA) and the high speed photography system, the detailed information on sprays characteristics including droplet size distribution, droplet velocity distribution, spray breakup length and spray angle were determined over all the experimental conditions.

Droplet Size Distribution

Droplet size distribution is not uniform and droplets ranging in sizes from a few microns to several hundred microns are present. The spray plumes can be described by empirical mathematical functions using the statistical properties of the droplet size distribution. During spray characterisation, results are obtained in terms of D_{v90}. The D_{v90} expresses diameter of droplets in which 90% of droplets diameters are smaller. It is of importance to the determination of droplet transport and evaporation.

Spray characterisation measurements were taken at the breakup length. Break up lengths and cone angles for all combinations of experimental parameters were extracted using the still images recorded by a high speed camera equipped with a 1W continuous laser system to provide a laser light sheet at a wavelength of 532 nm for illuminating the spray. Break-up length defined as the distance between the nozzle exit and the location in which atomization starts. The spray characterisation measurements were recorded across the entire spray plume by several measurement points. At each measurement point, 10000 samples are collected at least or a maximum measuring time of 120 s.

Experimental Conditions

In total, 30 experiments were carried out. Table.1 lists the experimental conditions.

Case	V_a (m/s)	Inlet air: $T_{db,i}$ (°C)	Inlet air: RH (%)	Inlet water: P_w (MPa)	Inlet water: $\dot{m}_w$ (kg/min)	Case	V_a (m/s)	Inlet air: $T_{db,i}$ (°C)	Inlet air: RH (%)	Inlet water: P_w (MPa)	Inlet water: $\dot{m}_w$ (l/min)
1	1	34	34.6	0.3	0.16	16	2	37.2	26.5	0.4	0.42
2	1	35	36	5.6	0.17	17	2	26.7	48	0.4	0.22
3	1	35.3	30	9.1	0.19	18	3	30.2	38	9.5	0.44
4	1	37	26	1.4	0.21	19	3	35.5	31.5	6.9	0.63
5	1	34.5	35	1.4	0.17	20	3	30.6	36.3	6.7	0.46
6	1	36	28	1.3	0.2	21	3	34.3	33.5	0.55	0.52
7	2	33	31	4	0.36	22	3	26.3	52	3	0.30
8	2	33.2	31	6.3	0.36	23	3	25	50.2	4.6	0.31
9	2	32.8	34.5	10	0.33	24	3	29	59	8.2	0.3
10	2	31	34.5	3.4	0.32	25	3	26.5	46	10	0.35
11	2	32	33	0.9	0.34	26	3	35.5	31.5	6.9	0.63
12	2	25.5	44	1.7	0.23	27	3	26.8	43	4.2	0.36
13	2	33.5	37	2.1	0.32	28	3	29	45	0.3	0.37
14	2	26	47.5	1.7	0.22	29	3	26	47	1.1	0.37
15	2	26	44	0.2	0.25	30	3	29.5	41	1.3	0.41

Table 1 Experimental conditions

Experimental Results

Spray Characteristics

The nozzles initial drop size and velocity distribution were obtained for all tests. The drop size and velocity spectra were obtained at the breakup length using the PDPA system. As an example, the droplet number distribution histograms of the nozzle (22N, H. Ikeuchi & Co. Ltd.) at different pressures (0.6, 1.4, 4.6 MPA) at the centre point of the spray are shown in Fig.2. These data have been used to obtain the area and volume flux weighted average Dv_{90}.

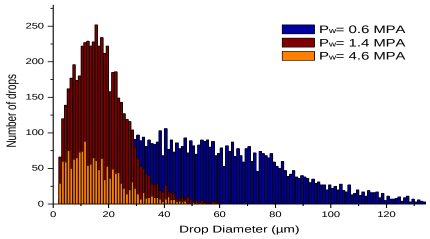


Figure 2 Droplet number distribution histograms of 22N nozzle at different pressures at the centre point

Inlet Air Cooling

A spray cooling system in NDDCTs is mainly designed to humidify the inlet air, therefore, enhancing the performance of the cooling tower. In spray cooling system analyses, cooling efficiency is generally considered as a good indicator in evaluating the performance of evaporative cooling systems. It represents how close the exiting air is cooled compared to the maximum possible temperature reduction (wet bulb depression). The cooling efficiency of a spray is defined by ASHRAE [12] as the ratio of the actual air temperature drop to the maximum possible temperature drop. Consequently, it can be expressed as:

$$\eta_c = \frac{T_{db,i} - T_{db,o}}{T_{db,i} - T_{wb}} \qquad (1)$$

where $T_{db,i}$, $T_{db,o}$ and T_{wb} are the dry-bulb temperatures of inlet and outlet, and wet-bulb temperature of the air, respectively. The cooling efficiency defined as above was used to investigate the performance of spray cooling system. Inlet, outlet and wet bulb temperatures measured in the experiments were used to estimate the spray cooling efficiency. However, the temperature distribution measured for the outlet air temperatures was not uniform which affects the average outlet air temperature. This is due to the nozzle spray cannot cover the whole cross section area of the tunnel.

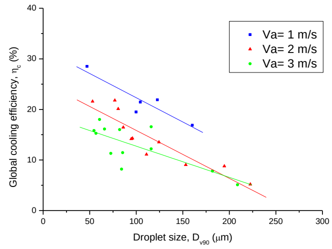


Figure 3 Spray cooling efficiency at the outlet section at different velocity (1, 2, 3 m/s) and different droplet size distributions

Fig.3 shows the cooling efficiency trend lines for various spray nozzles with different droplet sizes. Three air velocities were tested and the global average outlet temperature of the whole cross section was used for the cooling efficiency calculation. It is clear from Fig.3 that there are some data overlaps for the air speeds of 2 m/s and 3 m/s. Although the residence time is shorter and droplet size is larger, the cooling efficiency of 3 m/s air velocity was higher than or the same as 2 m/s for different tests. This can be explained based on the spray coverage area that the coverage area of 1 m/s is larger than 2 m/s and the coverage area of 2 m/s is slightly larger than 3 m/s. For the same droplet size, as the air velocity increases, the response time for droplets to lose momentum and follow the air stream decreases [13]. This is due to the fact that higher air velocity means higher air momentum causing the droplets losing their momentums quicker and penetrates less into the air stream. This conclusion is in agreement with [3, 8] which showed that spray dispersion is an important parameter affecting spray cooling efficiency as well as evaporation rate due to the momentum exchange between the air stream and droplets.

In order to distinct the effect of spray dispersion on the spray cooling efficiency, a modified spray cooling efficiency is defined as in Eq.2. In this calculation, only the areas which can cause at least 5% of cooling effectiveness are used:

$$\eta_m = \frac{T_{db,i} - T_{db,ao}}{T_{db,i} - T_{wb}} \qquad (2)$$

where $T_{db,i}$ and T_{wb} are the dry, and wet bulb temperatures of the inlet air, respectively. $T_{db,ao}$ is the average outlet temperature of the area with more than 5% cooling efficiency. Using this modified efficiency, influence of spray coverage area on spray cooling efficiency is isolated from droplet size and air velocity effects. Hence, the number of nozzles required to achieve uniform cooling distribution at a certain cross section could be identified.

Effect of Air Velocity on Spray Cooling Efficiency

Air velocity has a large influence on spray cooling efficiency and droplet transport. It affects droplet residence time. Furthermore, it affects the droplet dispersion which determines the coverage area. Fig.4 shows the changes of the modified cooling efficiency against the droplet sizes at three different velocities for all experimental conditions at 4.6 m downstream of the injection point. Two trends can be observed from Fig.4. Firstly, as expected, decreasing air velocity increased the modified spray cooling efficiency. Secondly, due to residence time influence, the difference between 1 m/s and 2 m/s cooling efficiencies is higher than the difference between 2 m/s and 3 m/s cooling efficiencies.

Fig.4 demonstrates that air velocity has a significant influence on spray cooling. The lower the air velocity is, the more cooling is achieved. Lower air velocity means longer droplet travelling time for the droplets. Furthermore, lower air velocity means larger coverage area because the time for droplets to lose momentum and follow the air stream is longer, which results in a better coverage area. These trends are in agreement with previous studies [3, 8].

In addition, Fig.4 shows that the spray cooling efficiency difference between 1 m/s and 2 m/s is more than double that between the 2 m/s and 3 m/s. The difference is due to the residence time difference. The droplet travelling time for air velocity of 1 m/s is double that of 2 m/s whereas the residence time difference between 2 m/s and 3 m/s are small, assuming that droplets follow the air flow immediately.

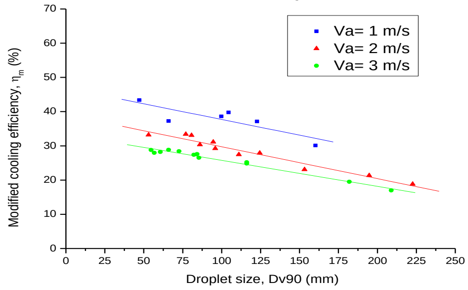


Figure 4 Modified spray cooling efficiency at the outlet section, 4.6 m downstream of the injection point for different velocities (1, 2, 3 m/s) at different droplet size distributions

Effect of Drop Size Distribution on Spray Cooling Efficiency

Another important factor affecting spray cooling efficiency is droplet size. Smaller droplets provide more surface area per unit volume than larger droplets and evaporation only occurs at the water/air interface. Evaporation rate per unit volume of droplets in gaseous media is related to the square of the droplet diameter and increases rapidly when droplet diameter is decreased [14]. The effect of droplet size on modified cooling efficiency can also be deduced from Fig. 4.

Fig.4 shows the modified spray cooling efficiency at different D_{v90}. It is clear from this figure that at the same air velocity, the smaller the droplet diameter is, the more evaporation and consequently higher cooling is observed. For instance, at 2 m/s of air velocity, the modified cooling efficiency at droplet size of D_{v90}=53 µm is approximately 33% while it is 19% at D_{v90}=222 µm. This is because the total exposed water surface area between water and air flow is increased. Therefore, the evaporation rate is higher.

Conclusions

In this study, the droplet evaporation, and the resulting cooling of the air were investigated experimentally at a range of inlet air conditions and a number of spray nozzle characteristics. Spray characteristics were determined with a PDPA system. A modified cooling efficiency term was used to distinct the effect of spray dispersion. The impacts of droplet size and air velocity on evaporation and cooling effectiveness are studied. The main conclusions from this study are as follows:

1- The results indicate that spray dispersion is a major factor for the evaluation of a spray cooling system in NDDCTs. A practical spray cooling system need to achieve uniform cooling distribution across the whole inlet section which to a large extent, depend on spray dispersion. Spray dispersion is mainly determined by droplet size and air velocity.

2- Spray cooling efficiency is higher at lower air velocities. This is a direct result of residence time influence. Lower air velocity means longer travelling time for droplets before air reaches its destination, i.e. the heat exchangers in a cooling tower.

3- It is clear that spray coverage area has a significant effect on the global spray cooling efficiency. In comparing the effect of different conditions, the spray coverage must also be considered. Otherwise, conflicting results can be obtained. Ignoring the effect of coverage area resulted in a calculated cooling efficiency of 3 m/s air velocity better than the 2 m/s cooling efficiency in some conditions although the residence time is shorter for 3 m/s.

4- The effect of spray coverage area on cooling efficiency emphasises the importance of nozzle arrangement and appropriate flow rate to get effective cooling and at the same time to avoid local oversaturation.

Acknowledgements

This work was financially supported by the Saudi Arabian Government (King Abdulaziz City for Science and Technology (KACST)) and the Queensland Geothermal Energy Centre of Excellence.

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