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

Experimental Study of Heat Transfer Coefficient and Pressure Drop of Cellulose Corrugated Media

Suoying He, Zhiqiang Guan, Hal Gurgenci, Kamel Hooman and Abdullah M. Alkhedhair

School of Mechanical and Mining Engineering
The University of Queensland, Brisbane 4072, Australia

Abstract

Wetted-medium evaporative cooling is presently applied in many fields, where the wetted media are to provide large water-air contact surface area and more contact time. The present authors are trying to pre-cool the entering air of Natural Draft Dry Cooling Towers (NDDCTs) for their performance improvement during hot seasons. However, the medium introduces extra pressure drop which reduces the air flow of a NDDCT, and as a result, impairs the tower performance. There is a trade-off between the cooling potential and pressure drop. In this study, a type of wetted medium, which has corrugated structure, is selected as part of a broader investigation on pre-cooling the entering air of NDDCTs. An open-circuit low-speed wind tunnel was used to study the performance of evaporative cooling with cellulose corrugated medium. The heat transfer coefficient, cooling efficiency and pressure drop across the medium with various thicknesses were experimentally studied in the wind tunnel. The test data were non-dimensionalized and curve fitted to yield a set of correlations. The effects of air and water flows on cooling efficiency and pressure drop were investigated. It was found that greater pressure drop occurs at larger medium thickness and higher air velocity. The pressure drop increases with the increase in water flow rate, however, the largest impacts on pressure drop are due to the changes in air flow and medium thickness. Higher cooling efficiency is associated with thicker medium at lower air velocity. The effect of water flow on cooling efficiency is negligible as long as the water is properly distributed and the media are fully wetted.

Introduction

Rising energy costs, together with water scarcity, urge the use of evaporative cooling systems that are economical and highly water and energy efficient. Wetted-medium evaporative cooling is presently applied in many fields and is proven to be effective [1, 2].

Natural Draft Dry Cooling Tower (NDDCT) is an alternative cooling method when large quantities of water are not available. Examples of this are the enhanced geothermal and concentrated solar thermal plants in Australia and the rest of the world, most of which are expected to be constructed in dry climates. A NDDCT creates the air flow through the heat exchanger bundles by means of buoyancy effects due to the difference in air density between the inside and outside of the tower. Essentially, the density difference is due to the difference in air temperature. The performance of a NDDCT is particularly reduced when the ambient air temperature is high. Reduced cooling tower performance lowers the efficiency of the thermal power stations they are serving. The present authors proposed to improve the tower performance using wetted-medium evaporative pre-cooling which limits the water consumption only to the periods when the ambient temperatures are too high [1, 2].

The wetted media are the critical components of wetted-medium evaporative cooling systems [3]. Choosing a suitable wetted medium requires knowledge of different working parameters. Some researchers suggested that the factors influencing the selection of media are cooling performance, pressure drop, cost and durability [4, 5]. The selection of media is also determined by the type of process to be cooled, environmental conditions, water quality, space availability, location and economic requirements. Previous simulations by the present authors [1, 2] found that the extra pressure drop introduced by the media is significantly important because it reduces the air flow passing through the NDDCT. This means, in this pre-cooling application, the pressure drop should be a major criterion during the selection of wetted media. From literature survey, we found that cellulose corrugated medium is engineering to provide maximum cooling, low pressure drop and long life of reliable service [6]. The low pressure drop is of critical importance to NDDCT operation. To better understand the performance of cellulose corrugated media, one type of cellulose corrugated medium is experimentally studied in this paper.

Materials and Methods

The wetted medium used in this study was made of corrugated cellulose paper, CELdek7060 as in the market, which will be referred to as cellulose medium in the paper. Three thicknesses were tested, i.e., 100, 200 and 300 mm. The medium samples were assembled to fit the 1000 mm × 1000 mm cross-section of the wind tunnel. This medium height is more than the minimum height of 600 mm as recommended by ASABE Standard [7]. Air passed through the media to form a cross flow with the water. The media must be installed in a proper orientation to the air flow, i.e., the 45 degree flute must be aligned upwards in the direction of the air flow and the 15 degree flute must be aligned downwards with the air flow (as shown in figure 1) [6]. The specific surface area of the medium was found to be 364.7 m^2/m^3.

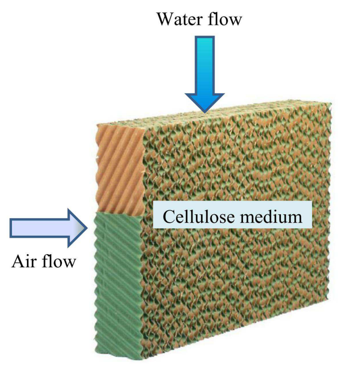

Figure 1. Cellulose corrugated medium.

A low-speed open-circuit wind tunnel (Gatton Campus, The University of Queensland, Australia), was used to achieve uniform and stable air flow for evaporative cooling tests. The schematic of the wind tunnel is depicted in figure 2. The apparatus consisted of air inlet and flow stabilization section, test section and exit section. A uniform and stable air flow was achieved after the air flow stabilizer. The test section had a cross-section of 1000 mm × 1000 mm with around 6000 mm length. The test section was designed to accommodate wetted-medium evaporative cooling with different medium thicknesses; it had transparent glass sides to observe water distribution. The exit section had a fan which was capable to drive air flow rate of 21 m^3/s. The fan was used during tests to control the air velocity by changing the RPM of the fan motor.

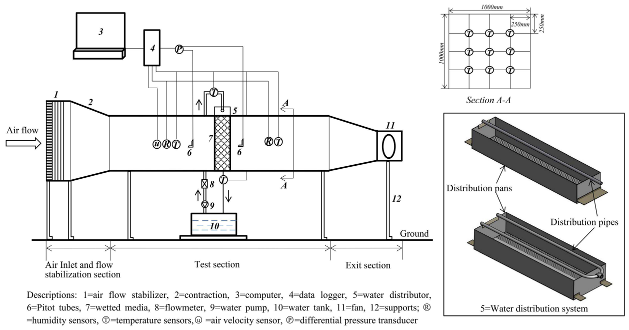

Figure 2. Schematic of the wind tunnel incorporated with wetted media (not to scale).

The water distribution system shown in figure 2 was used by J.G. Mannix [8] as well. The distribution pan with perforations at the bottom was located at the top of the media to feed water to the media more uniformly by gravity. During the tests, water in the distribution pan kept at a constant level by cotton cloth filters put at the bottom of the distribution pan. The cotton cloth filters slowed down the water flow and made the distribution more uniform. The water was fed to the distribution pan through distribution pipes, which were constructed of 19 mm diameter steel pipes with 2.5 mm holes 45 mm apart (one-pipe system was for 100 and 200 mm-thick media; two-pipe system was for 300 mm-thick media). The water from the distribution pan dripped down by gravity and capillarity to wet the media uniformly. Excess water was collected at the bottom of the media and stored in the water tank. The water in the water tank was recirculated by a 350 W water pump (PBC–350 Submersible Barrel Pump, Commercial Electric, Victoria, Australia). Water flow rate was controlled by a valve and monitored by the turbine flowmeter (DigiFlow 6710M–44, Savant Electronic Inc., Taiwan) with the range of 1.5–25 l/min and accuracy of ±5%. To calibrate the flowmeter before tests, a stopwatch was used to measure the time required for the distribution pipes to fill a container of known volume held at the level where the distribution pipe normally ran.

A uniform and stable air flow was achieved after air flow stabilizer. Generally, the flow out of a contraction often takes a distance equivalent to about 0.5 diameters before the non-uniformities are reduced below an acceptable level [9]. Thus, all the measurements were started at 500 mm downstream from the contraction end. The temperature and humidity of the inlet air were recorded during the tests but not controlled.

The sensors used in the tests are given in table 1. Two air velocity transmitters were used to measure the air velocity to increase the reliability in readings, and the average value was taken as the air velocity. Three thermistors were used to measure the air inlet dry-bulb temperature, and nine thermistors were used to measure the air outlet dry-bulb temperature as shown in figure 2 Section A–A. Two thermistors were used to measure the water inlet and outlet temperature respectively. All the thermistor probes were calibrated by Fluke Field Metrology Wells (Series 9142, Fluke Corporation, Washington, USA) before the tests. One humidity transmitter was used to measure the relative humidity of the inlet air while the other three were to measure the relative humidity of the outlet air. All the humidity transmitters were cross checked and calibrated by Fluke 1620A Digital Thermometer-Hygrometer (Series 1620A, Fluke Corporation, Washington, USA).

Sensor type and quantity	Model and manufacturer	Range Accuracy
2 Pitot tubes	Pitot tubes: Series 160–24, Dwyer Instruments, Inc. Differential pressure transmitter: MS321, Dwyer Instruments, Inc.	Transmitter: Range: 0–100 Pa Accuracy: ±1 %
2 Velocity transmitters	FMA1001R–V2, OMEGA Engineering, INC.	Range: 0–5.08 m/s Accuracy: 1.5 % F.S.
14 Thermistor probes	TJ36–44004–1/8–xx, OMEGA Engineering, INC.	Range: Max. 150 °C Accuracy: ±0.2 °C
4 Humidity transmitters	EE–21–FT6–B51, E+E Elektronik Ges.m.b.H.	Range: 0–100 % Accuracy: ±2 %RH

Table 1. Sensors used in the tests.

All data were recorded on a computer controlled data-acquisition system (UEI DNA–PPC8–1G, United Electronic Industries, Inc., Massachusetts, USA) and monitored once every 1 s. The media were wetted before testing to ensure saturation. At the beginning of each test, the water flow was fixed. The initial air velocity of 0.5 m/s was maintained for 30 min, then increasing this air velocity by 0.5 m/s at each step up to maximum 3.0 m/s during tests. For each air velocity increment, at least 10 min waiting period was maintained to ensure equilibrium between the media and the new air and water conditions. At each air velocity, 120 data points were recorded at equilibrium condition by all the sensors at 2 min intervals and the average values were used in data analysis. The water flow rates used in the tests were 62 and 31 l/min per m^2 horizontal exposed surface area.

Results and Discussions

<u>Pressure Drop</u>

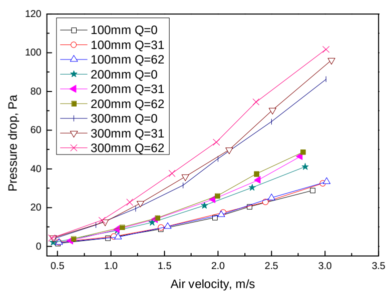

Figure 3. Measured pressure drop across three medium thicknesses at three different water flow rates (water flow rate Q is in the unit of $l/min/m^2$).

The measured pressure drop across the medium is presented in figure 3. In general, greater pressure drop is obtained with thicker media, which is in agreement with the previous studies [3, 4]. The pressure drop increases with the increase in air velocity and, to some extent, with increasing water flow rate. This is in accordance with the literature [3]. The increase in water flow increases the films of water retained on the transfer surface of the media and therefore decreases the volume for air flow in the media, and as a result increases the pressure drop. However, the largest impacts are due to the changes in air flow and medium thickness.

The measured pressure drop can be correlated as equation (1),

$$\Delta p = 0.124\left(\frac{l_e}{l}\right)^{-1.038}\left(1+1825\frac{q_w}{q_a}\right)\frac{\rho_a u_a^2}{2} \tag{1}$$

where R^2=0.979; l_e is the medium geometric length defined by the ratio of the volume occupied by the medium to the medium surface area (m); l is medium thickness (m); q_w and q_a are the volumetric flow rates of water and air flow respectively (m^3/s).

Cooling Efficiency

The definition of cooling efficiency was given by J.R. Watt [10] as,

$$\eta = \frac{T_{a1}-T_{a2}}{T_{a1}-T_{wb}} \tag{2}$$

where T_{a1} and T_{a2} are the air dry-bulb temperature before and after cooling; T_{wb} is the air wet-bulb temperature.

Figure 4a reports the cooling efficiency of the medium at difference air velocities and medium thicknesses. Figure 4a shows that the cooling efficiency decreases with increasing air velocity and increases with increasing medium thickness. This is in accordance with the literatures [3, 4].

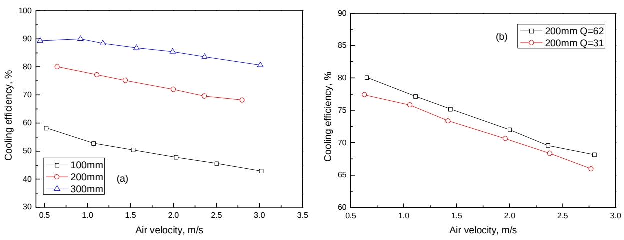

Figure 4. The effects of air velocity and (a) medium thickness at Q=62, (b) water flow rate at 200 mm medium thickness on cooling efficiency (water flow rate Q is in the unit of l/min/m^2).

Figure 4b investigates the effect of water flow on cooling efficiency at the medium thickness of 200 mm. Figure 4b reveals that the effect of the studied water flow rates on cooling efficiency is weak as the water was properly distributed and the media were fully wetted during tests (fully wetted means there is no streaking and dry area in the media). Similar results were obtained by A. Franco et al. [3]. This is of great importance, as we can reduce the water supplied to the media with providing its flow rates enough to fully wet the media, while the cooling performance remains unaltered. This makes it possible to reduce the designing pump power for wetting the media and to use less water, and thereby decreases the pressure drop across the media as shown in figure 3. However, T. Wang [11] suggested that in real applications, the water flow is chosen to be 10–30 times of the evaporation rate to not only obtain a better wetted surface but also to wash away debris deposited on the surface during operation.

Heat Transfer Coefficient

Heat transfer coefficient is a function of a large number of dimensional parameters (e.g., medium characteristics, air velocity, viscosity, conductivity, etc.). It is convenient to use the dimensionless quantity (the Nusselt number) to reduce the number of independent variables required to describe the dimensionless problem. To correlate our test data in terms of appropriate non-dimensional parameters, the Buckingham Pi theory was applied. The heat transfer coefficient correlation obtained from the dimensional analysis was given by,

$$Nu = 0.192\left(\frac{l_e}{l}\right)^{-0.191}\left(\frac{T_{a1}-T_{wb}}{T_{w1}}\right)^{-0.03}\mathrm{Re}^{0.682}\,\mathrm{Pr}^{1/3} \tag{3}$$

where R^2=0.983; the characteristic length in non-dimensional groups is the medium thickness l; T_{w1} is the water inlet temperature (°C); l_e, T_{a1} and T_{wb} represent the same parameters as in equations (1) and (2); the air properties in equation (3) are those of dry air at the average dry-bulb temperature through the wetted media.

Figure 5a is the experimental data presented using empirical correlation in the reference [12] while figure 5b is experimental data plotted using the new developed correlation, i.e., equation (3). Figure 5 shows that with the addition of (l_e/l) and (T_{a1}-T_{wb})/T_{w1}, the correlation fits the data well.

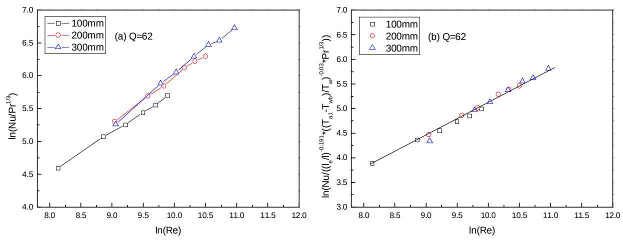

Figure 5. Non-dimensional correlation for heat transfer coefficient (a) non-calibrated (b) calibrated (water flow rate Q is in the unit of l/min/m^2).

Uncertainty Analysis

The uncertainty was analyzed according to the ISO Guide [13]. The resulting uncertainties of the measurement are given in table 2.

Parameter	Test range	Sensor accuracy	Max. standard deviation
Air inlet Temperature, T_{a1}	22.5–30.6 °C	±0.2 °C	0.2 °C
Air outlet temperature, T_{a2}	17.7–30.3 °C	±0.2 °C	0.2 °C
Air inlet humidity, RH_1	20.5–64.9 %RH	±2 %RH	0.5 %RH
Air outlet humidity, RH_2	25.4–92.1 %RH	±2 %RH	0.8 %RH
Air velocity, u_a	0.41–3.15 m/s	1.5 %F.S.	0.04 m/s
Water inlet temperature, T_{w1}	15.8–22.2 °C	±0.2 °C	0.04 °C
Water outlet temperature, T_{w2}	15.2–21.0 °C	±0.2 °C	0.05 °C
Pressure drop, Δp	0.8–101.7 Pa	±1 %	2.2 Pa

Table 2. Uncertainty analysis of test measurement.

Conclusions

Heat transfer coefficient and pressure drop correlations for the studied medium were developed by experimental studies. The new developed correlations represent the test data well and can be used for the performance predictions of evaporative cooling using such medium.

Generally, higher pressure drops are obtained with thicker media. The pressure drop across the media also increases with the increase in air velocity and, to a certain extent, with increasing water flow rate. The largest impacts are due to the changes in air velocity and medium thickness. The pressure drop range of the cellulose medium is 1.5 Pa to 101.7 Pa, depending on the medium thickness, air velocity and water flow rate. The cooling efficiency decreases with the air velocity increased while it increases with the medium thickness increased. The effect of water flow on cooling efficiency in this study is negligible as long as the water is properly distributed and the media are fully wetted. The cooling efficiencies of the cellulose medium vary from 43 % to 90 %, depending on the medium thickness and air velocity.

Acknowledgments

This research was performed as part of the Australian Solar Thermal Research Initiative (ASTRI), a project supported by Australian Government. The author, Suoying He, would also like to thank China Scholarship Council (CSC) for their financial support.

References

[1] S. He, Z. Guan, H. Gurgenci, I. Jahn, Y. Lu, A.M. Alkhedhair, Influence of ambient conditions and water flow on the performance of pre-cooled natural draft dry cooling towers, *Appl. Therm. Eng.*, **66**, 2014, 621-631.

[2] 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.

[3] A. Franco, D.L. Valera, A. Madueno, A. Pena, Influence of water and air flow on the performance of cellulose evaporative cooling pads used in Mediterranean greenhouse, *Trans. ASABE*, **53**, 2010, 565-576.

[4] R.W. Koca, W.C. Hughes, L.L. Christianson, Evaporative cooling pads: test procedure and evaluation, *Appl. Eng. Agric.*, **7**, 1991, 485-490.

[5] A.K.M. Mohiuddin, K. Kant, Knowledge base for the systematic design of wet cooling towers. Part II: Fill and other design parameters, *Int. J. Refrig.*, **19**, 1996, 52-60.

[6] Munters Corp., *Munters Products Handbook*, viewed: 08th July 2014, Retrieved from: www.munters.com/en/Global/.

[7] ASABE Standard (ANSI/ASAE EP406.4), *Heating, ventilating and cooling greenhouses*, American Society of Agricultural and Biological Engineers, 2008.

[8] J.G. Mannix, *Evaluation of the convective heat transfer coefficient of an evaporative cooling pad (Mphil thesis)*, Virginia Polytechnic Institute and State University, 1981.

[9] R.D. Mehta, The aerodynamic design of blower tunnels with wide-angle diffusers, *Prog. Aerosp. Sci.*, **18**, 1977, 59-120.

[10] J.R. Watt, W.K. Brown, *Evaporative Air Conditioning Handbook*, Fairmont Press, Inc., 1997.

[11] T. Wang, *Heat and mass transfer characteristics of indirect evaporate cooling system (PhD thesis)*, Marquette University, 1994.

[12] F.P. Incropera, D.P. Dewitt, T.L. Bergman, A.S. Lavine, *Fundamentals of Heat and Mass Transfer*, John Wiley & Sons, 2007.

[13] JCGM/WG 1, *Evaluation of measurement data—Guide to the expression of uncertainty in measurement*, Joint Committee for Guides in Metrology, 2008.