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Evaluation of thermal performance for natural and forced draft wet cooling tower

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ABSTRACT

This paper presents an experimental and numerical investigation of the thermal performance of natural draft wet cooling tower (NDWCT). The experimental investigation is carried out under natural draft condition and forced draft condition created by an axial fan. The operational parameters considered in this study are the thickness of the fill (10 and 20 cm), inlet water temperature (40, 45, and 50 °C) and inlet water volume flow rate (5.68, 7.75, and 9.46 L/min). The experimental results showed that the thermal performance is improved when the fans are used with the NDWCT. The temperature difference between inlet and outlet and effectiveness increase by 35% and 37.2%, respectively at fill thickness of 20 cm and water volume flow rate of 11.35 L/min. The temperature distribution of the air and the relative humidity were numerically simulated for both cases of natural and forced draft by employing the commercial CFD software ANSYS Fluent 15. The experimental and numerical results were validated with results from a previous work and showed a good agreement. The experimental results showed that the effectiveness increase by 22% and 30% for NDWCT and FDWCT respectively when in case of fill thickness 20 cm.

Keywords: Cooling tower; experimental measurements; natural draft, forced draft; CFD modelling; spray nozzles.

INTRODUCTION

Cooling tower is a heat exchanger to reduce water temperature when small amount is evaporated to the ambient air when they become in direct contact with each other. Cooling tower is complement to the power generation units, nuclear reactor, chemical and petrochemical industry, and air conditioning plants [1]. To improve the efficiency of these systems the researchers and designers focus on developing the thermal performance of the cooling tower because of their great influence to increase the efficiency of these mentioned systems [2]. The hot water inside the tower is cooled by spraying the water by nozzles within the internal zone opposite to ambient air stream moving upward inside the tower by fans or by nature draft. Air and water movement inside the tower leads to evaporate part of the water which increases the humidity of the outlet air and cooling the fallen water [3],[4]. One of the methods to improve the efficiency of cooling towers is to use fans assisted natural draft. It is similar to a natural draft design but with less height for the chimney than the natural draft one, with the fans at the base of the chimney. This type is likely to be used for hot weather with high relative humidity condition, for a large cooling capacity and when there is a limitation of available height.

Lemouari et al [5] carried out an experimental analysis to investigate the heat and mass transfer in a cooling tower between water and air by direct contact. The tower is filled with a vertical zig zag packing. They investigated the influence of water and air flow rate on heat and mass transfer coefficients and the evaporation rate of water in the air flow. Two flow patterns were inside the tower, a Pellicular Regime (PR) and Bubble Dispersion Regime (BDR). These two flow regimes help to investigate the best way to improve the performance of the tower. The main conclusions of this study are that there are two operational hydrodynamic methods of the cooling tower were observed; a Pellicular method at low water flow rates, and a bubble dispersion method at large water flow rates. According to Alok [6], all thermodynamics properties change after rain zone either increase or decrease, temperature reaches its higher value at centre line or lower values near the wall, the pressure decreases to the value from 7 Pa to zero at fill zone then increases with height.

Qasim et al [7] investigated experimentally the performance of natural draft wet cooling tower under the effect of the cross wind for a trickle fill of two thicknesses of (5 and 10) cm. The volume flow rate of water was varied from (0.8 to 2.4) gpm, and cross wind velocity from (0 to 1) m/s. The obtained results showed that cooling capacity, heat rejection and air enthalpy variation will increase whenever fill thickness or even water flow rate are increased. Yet, increasing cross wind shows that a knee point is found at critical cross wind velocity at bottom of the tower equals to (0.6) m/s. Alavi et al [8] investigated the performance of heat transfer in counter-flow NDWCT, under cross-wind and windless conditions by wind-creator setup to implement actual characteristics of the natural wind speed profile. This research studied the influence of water volume flow rate, cross-wind velocity, fill thickness and inlet water temperature on the water temperature variation and effectiveness. Saad et al. [9] investigated experimentally the performance of natural wet cooling tower. The effects of fill type, nozzle size and water flow rate were examined. The model dimensions were: top outlet diameter (370 mm), bottom diameter (680 mm) and height (850 mm). Three cases were investigated for film and splash fills. For film fill type, the thickness was 60, 90 and 120mm while for splash fill, the thickness was 30, 45 and 60 mm. The performance parameters of the tower such as temperature range, approach, effectiveness and Merkel number were investigated. Zhigang et al [10] conducted a 3D numerical simulation to investigate the thermal performance of NDWCT of a large scale for different fan diameter and speed. The results showed that the larger fan diameter leads to higher cooling efficiency and higher uniformity of air flow. They concluded that the using of fan enhances the performance of the cooling tower. N. Williamson et al [11] conducted a numerical simulation of heat and volume transfer inside a NDWCT using ANSYS- Fluent. A 2D axisymmetric model developed to present a more detailed model of cooling tower fill.

The aim of present study is to produce experimental data for the hyperbolic Fans assisted natural draft cooling tower which is not experimentally studied before in Iraq. These experimental data are also used to validate a CFD model developed to simulate the cooling process. Then, the CFD model is used to extend the range of the variables controlling the performance of the cooling tower to identify the optimum operating conditions suitable for wide range of temperature (0 ^oC in winter to 50 ^oC in summer). The contours from the CFD provide detailed visualised results which help in better understanding of the process at different zones within the shell. The thickness of the honey section, inlet water temperature and inlet water volume flow rate are the variables investigated in this study.

PHYSICAL MODEL

In wet cooling towers, the inlet water is cooled until its temperature becomes close to the wet ambient air temperature. The cooling process is achieved by spraying and distributing hot water from the top of the tower as droplets that falling down, and these droplets will be covered with uniform spherical saturated air film, the temperature of the water droplet surface supposed to be equal to the temperature of the section. The process of water cooling is based on the heat and mass transfer phenomena inside the cooling tower through direct interface between water and air while a part of the water evaporates into the air

Merkel theory [12] states that the heat transfer rate at any position of the tower is proportional to the difference between the air enthalpy and the saturated air enthalpy at the water temperature at the same point in the tower. The heat transfer rate can be calculated as [13,14].

$$q'' = h_{de}(i_{as} - i_a) \tag{1}$$

Where h_{de} is an empirical mass transfer coefficient which can be obtained from the experimental work and ($i_{as} - i_a$) is the difference between enthalpy of saturated air and dry air. The effectiveness of the cooling towers the ratio between the actual transferred energy to the optimum energy transfer. The effectiveness equation (ϵ) is:

$$Effectiveness\ (\epsilon) = \frac{(T_w)_{in}-(T_w)_{out}}{(T_w)_{in}-(T_{wet})_{in}} \tag{2}$$

EXPERIMENTAL SETUP

The experiments reported in this paper were conducted with experimental setup shown in schematic diagram in Figure 1. The tower consists of the tower shell, the tower base, nozzles system, packing, water heating system, axial fans to assist air draught and the measurement equipment. The tower can be operated in two modes, a natural draft wet cooling tower (NDWCT) and forced draft wet cooling tower (FDWCT). The tower shell dimensions are: bottom base diameter = 98 cm, tower height = 131 cm and the upper base diameter = 58 cm. The base is fabricated to be compatible with the tower shell as a circular shape (diameter = 100 cm). Eight axial fans, of power 33 W and 20 cm in diameter are attached to the base and spaced equally. The speed of the fans was controlled by a voltage regulator. The nozzle system consists of five nozzles. The nozzles are fixed on a circular pattern and spaced equally to ensure uniform distribution of water over the fill. Plastic honey cell packing is used as a fill and this type of fills is the most common used in cooling towers. The effect of fill thickness was tested by using two fills of 10 cm and 20 cm. Two electrical heaters, of 3000 W, fixed inside two tanks have been used to create the heating load and to ensure providing hot water in different weather conditions. To ensure a continuous flow of the hot water at constant temperature during the experiments with values of 40, 45 and 50 °C. Three centrifugal pumps, of flow rate 40 L/min, have been used. Two pumps for pumping the cold water from cold water tank to the Small hot tank and the third pump for pumping hot water from large water tank to the nozzles inside cooling tower. Four valves were used to control the flow of the hot and cold water. A flow meter was used to measure the flow rate of the hot water entering the tower through the nozzles system. Four thermocouples were used to measure the dry and wet bulb temperature of the air at the inlet and the outlet of the tower. Two thermocouples were used to measure the temperature of the water at the inlet and outlet. The thermocouples were calibrated with reference thermometer as shown in Figure 2.

T-1	Inlet air temperature	E-1	Small tank
T-2	Outlet air temperature	E-2	Large tank
T-3	Inlet water temperature	A	Axial fans
T-4	Outlet water temperature	H	Electric heater
P-1	Hot water pump	N	Nozzles system
P-2	Cold water pump	F	Fill
M	Flow meter	V-1	Small tank valve
V-2	Large tank valve	V-3	Small tank valve
V-4	Drain Valve	D	Drain basin

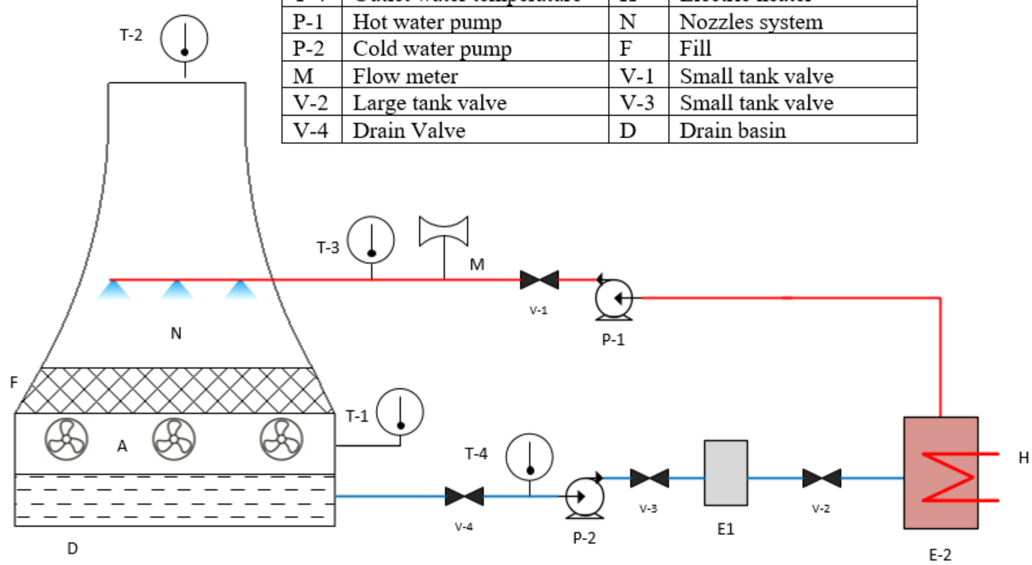

Figure 1. Schematic diagram of the experimental setup.

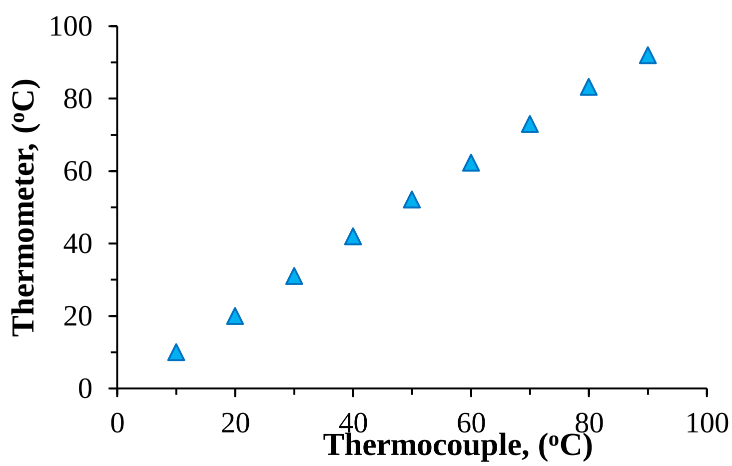

Figure 2. Calibration curve of thermocouple.

NUMERICAL MODEL

Computation Fluid Dynamic (CFD) was employed investigate the thermal performance of NDWCT and FDWCT tower. The commercial software ANSYS Fluent 15 was employed to develop a three dimensional steady state numerical simulation of NDWCT and FDWCT. The simulation was two phase flow. The primary phase was the gas phase. Air represented the gas phase. The secondary phase was water. the secondary phase was modelled by using discrete phase model. All the physical properties of the air and water were represented as functions of temperature [15].

Table 1. Properties of air.

Description	Equations
Density	$\rho = 345.57\,(T - 2.6884)^{-1}$ (*kg/m*^{3})
Viscosity	$\mu = 2.5914 \times 10^{-15}T^3 - 1.4346 \times 10^{-11}T^2 + 5.0523 \times 10^{-8}T + 4.1130 \times 10^{-6}$ (*Ns/m*^{2})
Specific heat	$C_p = 1.3864 \times 10^{-13}T^4 - 6.4747 \times 10^{-10}T^3 + 1.0234 \times 10^{-6}T^2 - 4.3282 \times 10^{-4}T + 1.0613$ (*kJ/kg.K*)
Thermal conductivity	$k = 1.5797 \times 10^{-17}T^5 + 9.46 \times 10^{-14}T^4 + 2.2012 \times 10^{-10}T^2 - 2.3758 \times 10^{-7}T^2 + 1.7082 \times 10^{-4}T + 7.488 \times 10^{-3}$ (*W/m.K*)
Prandtl number	$Pr = 1.0677 \times 10^{-23}T^7 - 7.6511 \times 10^{-20}T^6 + 1.0395 \times 10^{-16}T^5 + 4.6851 \times 10^{-13}T^4 - 1.7698 \times 10^{-9}T^3 + 2.226 \times 10^{-6}T^2 - 1.1262 \times 10^{-3}T + 0.88353$

Creation of Geometry

The three-dimensional model was created and designed by SolidWorks software. The geometry has variable diameter, so it is divided into a number of two dimension planes at different heights. Then, the command loft was used to create the 3D geometry. Then, the geometry is imported to the ANSYS workbench design modeller. The geometry consists of three zones, the tower, the fill and the entrance as shown in Figure. 3. After the geometry was imported into ANSYS design modeller, the model bodies were assigned as a fluid.

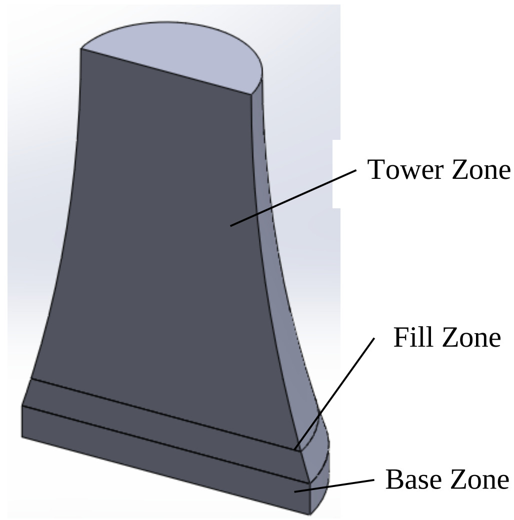

Figure 3. Geometry of the domain.

Mesh Independency

The mesh of the model is created with ANSYS meshing tool. The flow domain was discretized into a finite set of control volumes or elements, then solving the governing equations for these elements. The accuracy of ANSYS- Fluent increases when the element size is smaller. The influence of cells number of the effectiveness is shown in Figure.4 for different volume flow rate of water. Three numbers of element cells were tested. The results obtained from the 5425061 cells show the best agreement with those from experimental. The meshed domain is shown in Figure 5.

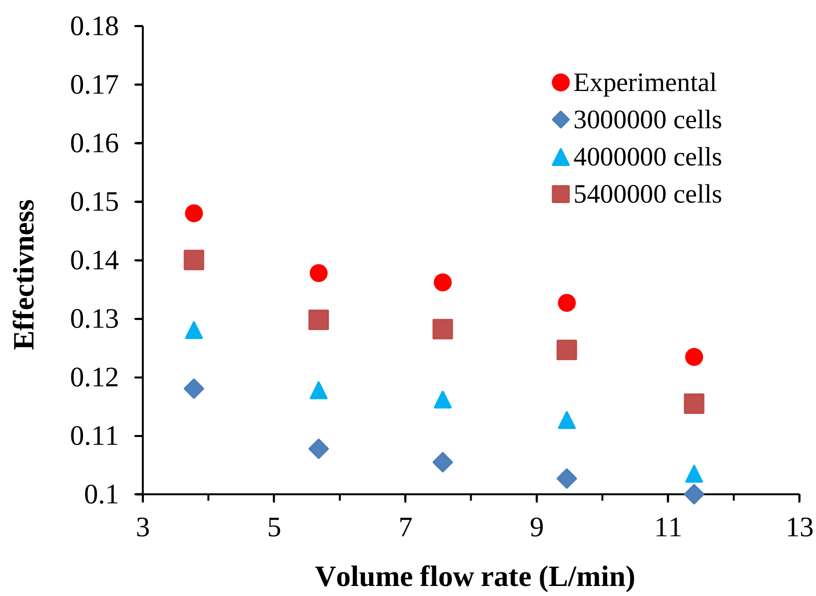

Figure 4. Mesh independency.

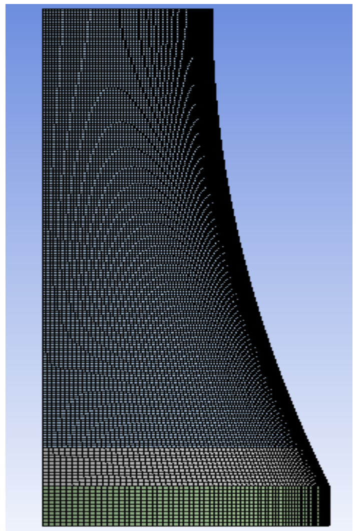

Figure 5. The domain after meshing.

Governing Equations

The conservation equations of mass, momentum and energy are used [16,17]. The model formulated was steady state and turbulent.

Continuity Equation:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{3}$$

where, u_i is the instantaneous fluid velocity.

Momentum Equation:

$$\rho \frac{\partial u_i u_j}{x_j} = -\frac{\partial P}{\partial x_i} + \mu \frac{\partial}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) + \rho \frac{\partial}{\partial x_j}\left(-\overline{u_i' u_j'}\right) \tag{4}$$

Energy Equation:

$$\rho c_p \frac{\partial u_i T}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\lambda \frac{\partial T}{\partial x_i} - \rho \overline{u_i' T'}\right) \tag{5}$$

where, λ is the thermal conductivity

Turbulence Model:

Realizable k-ε turbulence model is applied in the present study by considering the work of Klimanek et al [18]. They found that realizable k-ε is the most suitable model for the simulation of the flow inside the wet cooling tower.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k \tag{6}$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \rho C_1 S_\varepsilon + C_{1\varepsilon}\frac{\varepsilon}{k} C_{3\varepsilon} G_b + S_\varepsilon \tag{7}$$

RESULTS

In order to assess the accuracy of the present work, comparison was made with data from literature. Figure 6 compares the experimental and numerical data with those obtained by Avari [8]. Regardless the effectiveness value for volume flow rate less than 5 L/min. The average discrepancy between of experimental and numerical data with literature is 13% and 18 % respectively. This discrepancy can be considered as acceptable due to the complex nature of the flow inside the cooling tower. The complexity of the flow comes from the evaporation and the high level of turbulence. The compared data are of the same operational conditions.

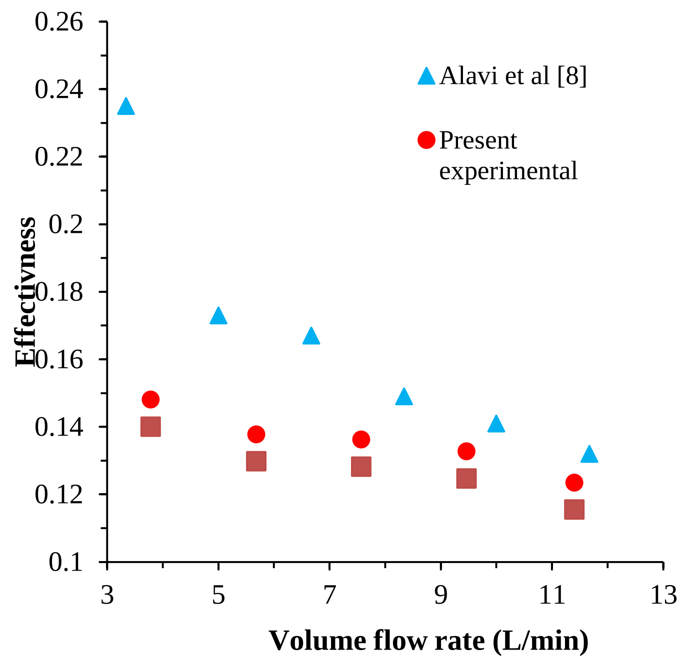

Figure 6. A comparison with data from [8].

Experimental Results

The influence of using assisted fans in NDWCT, different fill thicknesses, different inlet water temperatures and water volume flow rate on cooling tower effectiveness were investigated. Figure 7 and 8 show the effectiveness variation with water ass flow rate for different air wet bulb temperatures in the case of NDWCT, at fill thickness of 10 and 20 cm, respectively. When water flow rate increases the effectiveness decreases for each thicknesses.

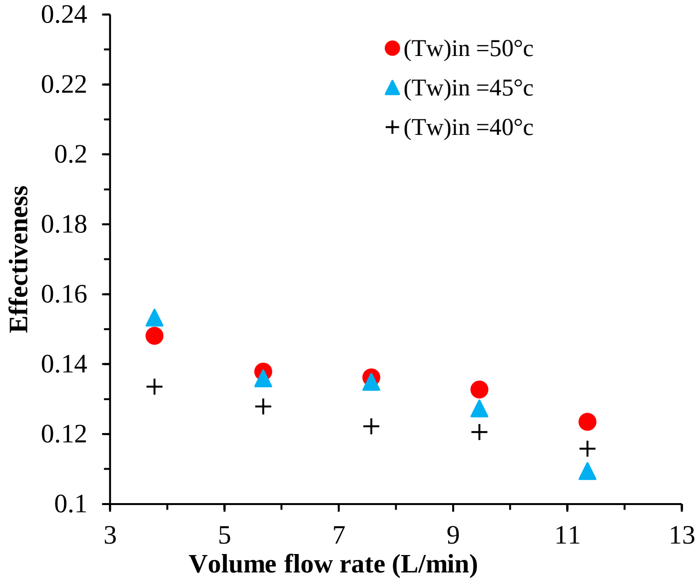

Figure 7. Effectiveness with water flow rate at (10 cm) fill thickness, NDWCT.

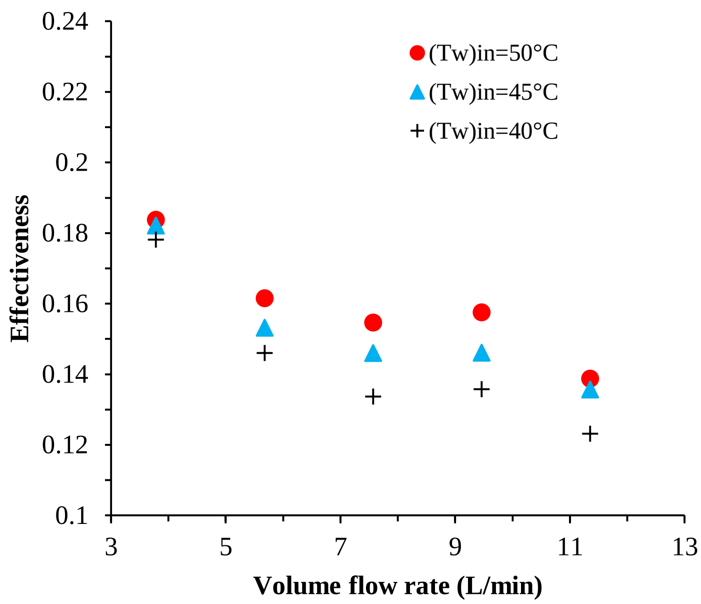

Figure 8. Effectiveness with water flow rate at (20 cm) fill thickness, NDWCT.

This may be attributed to the higher heating load for the same contact surface area. When fill thickness increases the effectiveness increases too. For instance, at the inlet temperature, (T_w) in = 50 °C, the effectiveness enhanced by 14.3% at water flow rate 5.68 L/min. This can be attributed to the increase contact area and water residence time inside the tower.

Figures 9 and 10 represent the effectiveness variance in the FDWCT at fill thickness of 10 and 20 cm, respectively. It can be observed that, when water flow rate increases the effectiveness increases. The reason of that is effectiveness depends on the cooling range and wet bulb air temperature. The range increases with the increase of the water flow rate at constant wet bulb air temperature to 15.5°C.

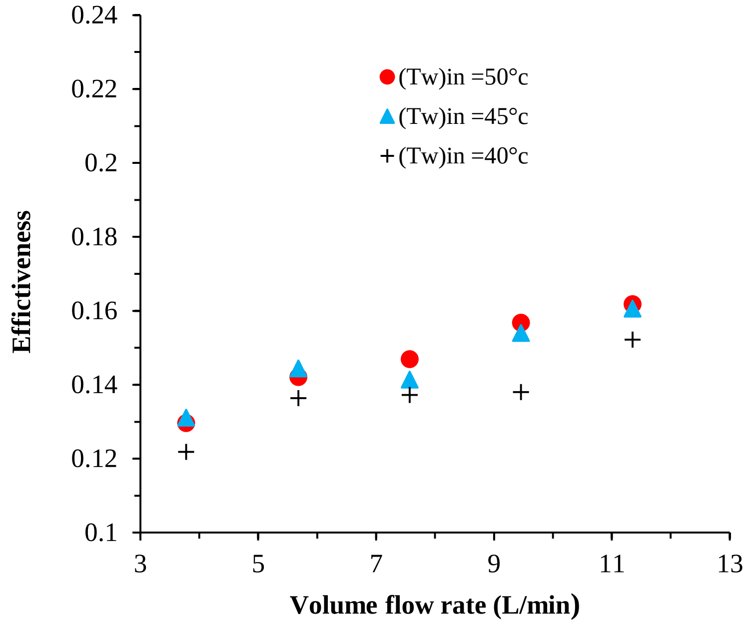

Figure 9. Effectiveness with water flow rate at (10 cm) fill thickness, FDWCT.

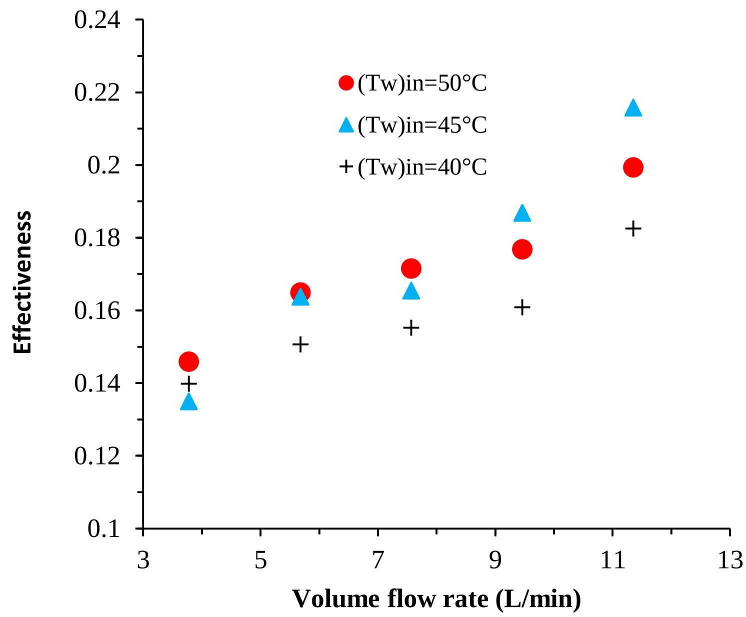

Figure 10. Effectiveness with water flow rate at (20 cm) fill thickness, FDWCT.

Numerical Results

Figures 11 and 12 represent the temperature contours at different planes for both case of FDWCT and NDWCT. It can be noticed that the air temperature is higher around the spraying nozzles due to heat exchange between water and air. Also, it is noticed that the temperature distribution is non uniform in case FDWCT due the increased turbulence level of incoming air.

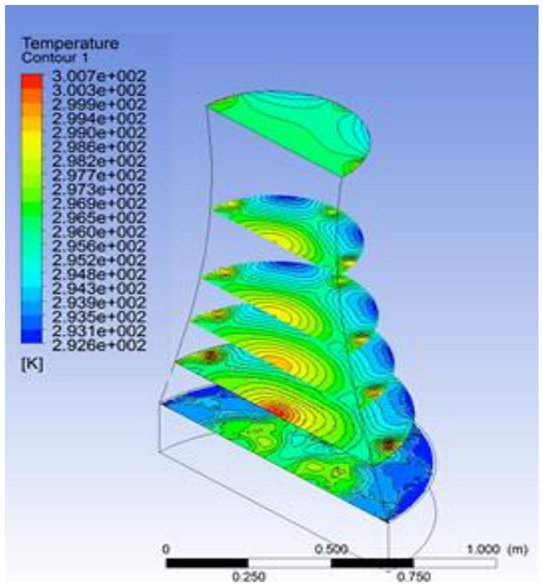

Figure 11. Air temperature contour, NDWCT, fill thickness (20cm).
(Tw)in = 313K, (Ta)in= 292.6K, and (42%) inlet air relative humidity.

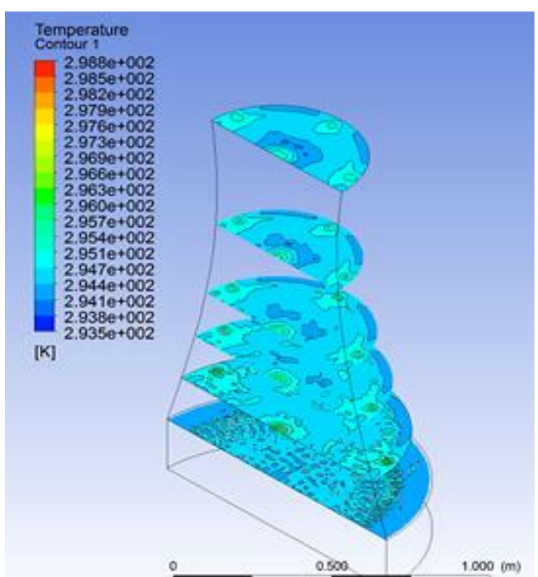

Figure 12. Air temperature contour, FDWCT, thickness (20cm).
(Tw)in= 313K, (Ta)in= 292.6K, and (42%) inlet air relative humidity.

Figure 13 represents the relative humidity contour at different planes in case NDWCT. The relative humidity tends to decrease gradually as the air moves upward. The base cross-section shows three air zones, the first one has a very high RH of 75%, the second one has less RH of 62%, the third one has the lower RH value of 45%. Moving up to the zone between the fill and the spray zone, the cross-section shows a uniform distribution for the air RH, and this uniformity remains for the upper cross-sections till the exit section due to the lower air velocity through the cooling tower. Figure 14 represents the relative humidity contour at different planes in case of FDWCT. The base cross-section has two air zones, the first one around the centre, its relative humidity about 77%, the second zone surrounded and partially overlaps with the first one, its average value is 48 % with an increase near the nozzles. This contour seems to be of non-uniform distribution because of the use of the assisted fans leads to increases the turbulence level of incoming air.

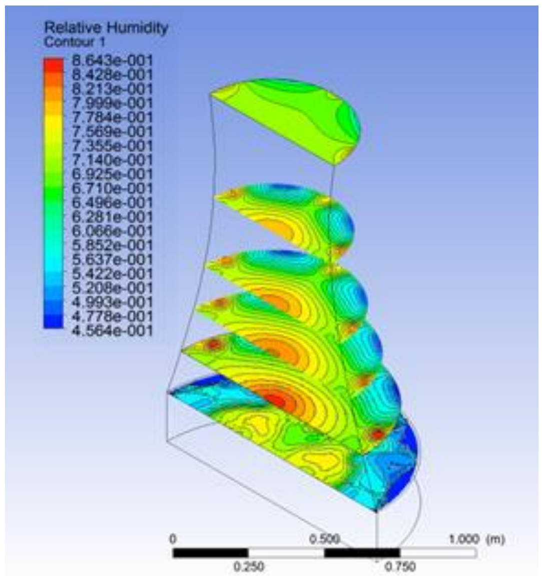

Figure 13. Relative humidity contours, without assisted fans fill thickness (20cm).
(Tw)in= 313K , (Ta)in= 292.6K, and (42%) inlet air relative humidity.

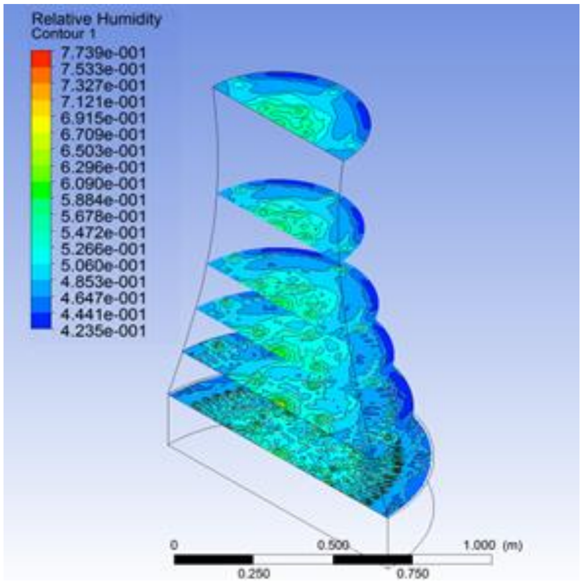

Figure 14. Relative humidity contours, with assisted fans fill thickness (20cm). (Tw)in= 313K, (Ta)in= 292.6K, and (42%) inlet air relative humidity.

CONCLUSIONS

Experimental and numerical results of NDWCT and FDWCT are presented. Effects of the fill thickness, water volume flow rate and water inlet temperature on the effectiveness of the tower were investigated. The main conclusions of this study can be summarized as:

- For NDWCT, the effectiveness at different inlet water temperatures decreases gradually as the water flow rate increases from 3.78 L/min to 11.35 L/min.
- For FDWCT, the effectiveness at different inlet water temperatures increases gradually as the water flow rate increases from 3.78 L/min to 11.35 L/min.
- For FDWCT, the effectiveness increases as water volume flow rate increases from 3.78 L/min to 11.35 L/min
- For NDWCT and FDWCT, the effectives increase for higher water inlet temperatures.
- For NDWCT and FDWCT, the effectiveness is increased by 17% and 15% as the fill thickness increase from 10 cm to 20 cm.
- The relative humidity near the nozzles system is higher than other locations due to the higher exchange rate between air and water in this location.
- The experimental and numerical results are in good agreement with data from literature.

REFERENCES

[1] Mahdi QS, Jaffal HM. Experimental study with using anfis to evaluate the performance of a modified closed wet cooling tower. The Journal of the University of Duhok. 2017; 20(1): 416-32.

[2] Bilal AQ, Zubair SM. A complete model of wet cooling towers with fouling in fills. Applied Thermal Engineering. 2006; 26(16): 1982-989.

[3] ASHRAE Handbook: Heating, ventilating, and air-conditioning systems and equipment. Atlanta, GA: ASHRAE, 2008.

[4] Hamon. Fan assisted natural cooling tower. Retrieved from https//www.hamon.com/wet-cooling–towers/fan-assisted-naturaldraft/; March, 2019.

[5] Lemouari M, Boumaza M, Kaabi A. Experimental analysis of heat and mass transfer phenomena in a direct contact evaporative cooling tower. Energy Conversion and Management. 2009; 50(6): 1610-617.

[6] Alok S, Rajput SPS. Application of CFD in natural draft wet cooling tower flow. International Journal of Engineering Research. 2012; 2(1): 1050-056.

[7] Mahdi QS, Al-Hachami MR. Experimental analyses for NDWCT performance using trickle fill under the effect of cross wind. International Journal of Scientific Research and Education. 2015;3(3): 2969-2977.

[8] Alavi SR, Rahmati M. Experimental investigation on thermal performance of natural draft wet cooling towers employing an innovative wind-creator setup. Energy Conversion and Management. 2016; 122: 504-514.

[9] Saad M. Saleh, Qasim S. Mahdi and Basima S. Khalaf. Investigation of natural draft cooling tower performance using neural network. International Congress on Energy Efficiency and Energy Related Materials. 2014; 155: 113.

[10] Dang Z, Zhang Z, Gao M, He S. Numerical simulation of thermal performance for super large-scale wet cooling tower equipped with an axial fan. International Journal of Heat and Mass Transfer. 2019; 135:220–234.

[11] Williamson N, Behnia M, Armfield S. Numerical simulation of heat and mass transfer in a natural draft wet cooling tower. In: 15th Australasian Fluid Mechanics Conference, Sydney, Australia; 13-17 December, 2004.

[12] Merkel, F, Verdunstungskühlung, VDI-Zeitschrift. 1925; 70(70):123 – 128.

[13] Gudmundsson Y. Performance evaluation of wet-cooling tower fills with computational fluid dynamics. Master's Thesis, Stellenbosch University, 2012.

[14] Kareem FA, Al-Dulaimi MJ, Lafta NS. Investigation the exergy performance of a forced draft wet cooling tower. International Journal of Engineering & Technology. 2018;7(4):2575-2580

[15] Zografos AI, Martin WA, Sunderland JE. Equations of properties as a function of temperature for seven fluids. Computer Methods in Applied Mechanics and Engineering. 1987;61:177-187

[16] Too JHY, Azwadi CSN. Numerical analysis for optimizing solar updraft tower design using computational fluid dynamics (CFD). Journal of Advanced Research in Fluid Mechanics and Thermal Sciences. 2016; 229 (1):8-36.

[17] Al-Dulaimi MJ, Abdul Rasool AA, Hamad FA. Investigation of impingement heat transfer for air-sand mixture flow. The Canadian Journal of Chemical Engineering. 2016;94:134-141.
[18] Klimanek A, Cedzich M, Białeck R. 3D CFD modeling of natural draft wet-cooling tower with flue gas injection. Applied Thermal Engineering. 2015;91:824-833.