Al-Nahrain University, College of Engineering Journal (NUCEJ) Vol.19 No.2, 2016 pp.310 - 326

Experimental Investigation for the Thermal performance of Modified Closed Wet Cooling Tower

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Abstract

Researchers and designers keenly seek to improve the performance of cooling towers because of the extensive impact on the work and efficiency of the systems concerned to these towers. For this purpose, a Closed Wet Cooling Tower (CWCT) modified with added packing was designed, manufactured and tested for cooling capacity of 9 kW in Iraq. A series of experiments was carried out at different operational and conformational parameters. Operational parameters demonstrate: air flow rate, spray water flow rate, cooling water flow rate, inlet cooling water temperature and inlet air wet bulb temperature.

Conformational parameters indicate: height of packing used and location of packing. The results showed a significant performance improvement when using packing with CWCT under the heat exchanger and above the heat exchanger as compared to CWCT. Empirical correlations are obtained to predict water film heat transfer coefficient and air-water mass transfer coefficient considering the influences of operational parameters.

Key words: Closed Wet Cooling Tower, heat exchanger, packing, thermal performance

1. Introduction

There are two types of wet cooling towers: open and closed cooling towers. In the open cooling tower, the water is in direct contact with the air at surface of the packing. In conventional CWCTs recirculated water is sprayed over a horizontal tube bundle, while air is drawn over the bundle and cooling water is circulated in tubes and never contacts the outside air. Because of the advantages of the close cooling tower that limited contamination risks with airborne and corrosion, it has a wide range of applications in the fields of electrical power, chemical industry and building air conditioning. With more and more closed cooling tower applications, the study also received increasing attention **[1].**

Much attention has been paid to issues on CWCTs relating to experimental studies and developed correlations of heat and mass transfer coefficients as a function of operating conditions. **Oliveira & Facao [2]**, designed a new CWCT in order to examined effects of the operating parameters on the saturation efficiency for a CWCT modified for use with chilled ceilings in buildings. Design circumstances were a cooling limit of 10 kW, for temperature of water inlet 21°C, a flow rate of cooling water 0.8 kg/s and an inlet Air Wet Bulb Temperature (AWBT) of 16°C .The area tower is 600 × 1200 mm and a height of 1550 mm. That tube package has 288 tubes of 10 mm external diameter in staggered array, for an aggregate transfer area of 8.6 m^2. Their results indicated that the efficiency increments with the increment in flow rates of air and spray water while; efficiency diminishes with the increment flow rate of cooling water. On the other side of argument, efficiency increases somewhat with increasing AWBT. Also, they developed test correlations for coefficients of mass and heat transfer as seen in Eqs. (1) & (2), respectively, these correlations used well to predict cooling tower performance when applied to all possible operating conditions.

$$\alpha_m = 0.1703\left({}^{m}/_{m_{max}}\right)_{air}^{0.8099} \quad \text{... (1)}$$

$$\alpha_s = 700.3\left({}^{m}/_{m_{max}}\right)_{spray}^{0.6584} \quad \text{... (2)}$$

Thermal performance of tow evaporative cooled heat exchangers, Investigated by **Hasan & Sirén [3]**.They studies two heat exchangers; plane and plat-finned circular tube types occupy the exact volume and the ratio of total area (finned tubes /plate tubes) is four. The dimensions of the test section of heat exchanger is 88 mm×130 mm, with 250 mm length. The outer diameter of copper tubes having is 10 mm and tube even length is 88 mm. The finned shaped is developed by presenting 0.5 mm thick copper plates in the middle of the tubes. The tubes are in staggered array with 2.8D equilateral pitch in 8 rows and 4 columns. They developed correlations for coefficient of mass transfer as a function only of air flow rate for both plane and finned tubes:

For the plane tubes

$$\alpha_m = 3.36(m)_{air}^{0.812} \quad \text{... (3)}$$

For the finned tubes

$$\alpha_m = 3.29(m)_{air}^{0.874} \quad \text{... (4)}$$

They showed that an increase from 92% to 140% in heat transfer happens in the case for utilizing plate-finned tube heat exchanger in comparison with the plane tube for the same scope of working conditions. **Shim et al. [4 & 5]** investigated experimentally the thermal performance of two heat exchangers in closed-wet cooling tower having a rated capacity of 2TR. Both heat exchangers have multi path that is consumed as the entrance of cooling water and are consisting of bare-type copper tubes of 15.88 mm and 19.05 mm. It may have been revealed that the cooling range of CWCT utilizing double ways is higher by around 20% than the single way. For working both tubes with two ways, cooling capacity per unit volume with 15.88 mm tube is nearly 27.5% and 41.01% higher than those with 19.05 mm when the inlet AWBT of CWCT were 24℃ and 28℃ individually.

Heyns & Kroger [6] investigated the thermal performance characteristics of an evaporative cooler, which consist of 15 tube rows and 8 columns with 38.1 mm outer diameter, galvanized steel tubes arranged in a triangular pattern of 76.2 mm. From the test results, coefficients of mass and heat transfer were correlated. Those test outcomes demonstrated that the flow rate of spray water has the best impact on the coefficient of heat transfer however this coefficient is likewise a stream's element of air and the temperature of spray water. They had also been found that the coefficients of mass transfer are functions of the air and spray water mass velocities as demonstrated over Eqs. (5) & (6) respectively:

$$\alpha_m = 0.038(G_{air})^{0.73}\left(\frac{\Gamma}{D}\right)^{0.2} \quad \text{... (5)}$$

$$\alpha_s = 470(G_{air})^{0.1}\left(\frac{\Gamma}{D}\right)^{0.35}\left(T_{sw,ave}\right)^{0.3} \quad \text{... (6)}$$

Al-Tayyar [7] Modified an available open circuit cooling tower (WL 320 Demo cooling tower, constructed by GUNT company in Germany) to make utilized likewise closed circuit cooling tower by designing furthermore manufacturing a heat exchanger located under packing. The heat exchanger consisting of 8 mm outside tube diameter arranged in 6 rows and 12 columns in an inline arrangement. The capacity of cooling tower is 1 kW. He investigated different operating parameters such as velocity of air, ratio of spray water to air flow rate, flow rate of spray water and inlet AWBT on the performance of cooling tower. He saw that these parameters influence the tower performance and the utilization of packing materials is a decent way to deal with improves the performance for various working conditions. Correlations for coefficients of mass and heat transfer were developed as follows:

$$\alpha_m = 0.063(G_{air})^{0.81}\left(\frac{\Gamma}{D}\right)^{0.1} \quad \text{... (7)}$$

$$\alpha_s = 555(G_{air})^{0.075}\left(\frac{\Gamma}{D}\right)^{0.48}\left(T_{sw,ave}\right)^{0.1} \quad \text{... (8)}$$

An important conclusion of this work is that the performance of the closed circuit cooling tower using packing shows a relatively good enhancement as compared to the tower operating without packing. The main recommendations of his work that carrying out the experiments on an entire large closed circuit cooling tower and investigated factors related to the heat exchanger such as its location inside the cooling column, using staggered tubes arrangement therefore, studying the effect of cooling water flow rate. **Zheng et al [8]** investigated thermal performance of an oval tube CWCT based on heat and mass transfer under different operating conditions. Induced draft CWCT has an external dimension of 1420 mm×1140 mm×3398 mm tested in this study. The tube group consists of 8 rows of remotely excited steel oval tubes with 37 oval tubes per tube rows organized in stagger array. The oval tube are 1200 mm long, the real hub of the tube is 31.8 mm, and minor pivot 21.16 mm with tube divider thickness 1.5 mm. The analysis was directed with varieties of flow rate of cooling water, inlet temperature of cooling water, flow rate of spray water, inlet AWBT and velocity of inlet air. Empirical correlations are presented for the coefficients of mass and heat transfer as follows:

$$\alpha_m = 0.034(G_{air})^{0.977} \quad \text{... (9)}$$

$$\alpha_s = 350.3(1 + 0.0169T_w)(G_{air})^{0.694}(G_{sw})^{0.512} \quad \text{... (10)}$$

In the relevant literature, no results have been reported so far involving the CWCT with packing. The aim of this research is to evaluate thermal performance of modified CWCT with added packing in hot and arid environmental of Iraq.

2. Experimental Apparatus and Procedure

2.1. Description of Test Rig

A new CWCT was designed and constructed in which different operating parameters could be varied and tested in the laboratories of Environmental Engineering Department of Al-Mustansiriya University College of Engineering. The general arrangement of the equipment is shown in Figure (1). In general, the apparatus consists essentially of cooling column and three major systems; spray water, cooling water and air blowing.

The tower fabricated from galvanized steel sheet to provide protection from rusting and corrosion, each sheet of 1.5 mm thickness, connected together by screws and nuts as a rectangular box of external dimensions (700 mm × 400 mm × 2300 mm), mounted rigidly on a frame which is welded construction with a channel section at the base welded together from the rectangle. As exists in every forced cooling, the test section consists of three zones: spray, fill (cooling zone) and rain zone. Spray zone is at a height of 180 mm suitable to ensure water distribution uniformly to all points in the fill section. Fill zone at 1000 mm height and characterized as consisting of three places for sliding removable drawer rectangular boxes at the same dimensions, manufacturing for packing and heat exchangers to ensure change the locations and types of heat exchangers and height of packing to study the influence of all these additions on the performance of the tower. The rectangular drawer made of galvanized steel with dimensions of 420 mm in width, 760 mm in depth and 280 mm in height. Six holes along the side of each (drawer) box were done to measure the water temperature, air dry bulb temperature and air relative humidity. The rain zone at a height of 450 mm in the case of three boxes and it will be variable when lifting one or two packing's and increases as decreases the packing height.

Air from the atmosphere, enters the single stage centrifugal blower at a rate which is controlled by the butterfly valve. The fan discharges into the PVC pipe and the entrance duct before entering the packed column. As the air flows through the packing and heat exchanger, its moisture content increases and the water in the heat exchanger are cooled. Hot water is pumped from the load tank through the control valve and a water flow meter to the heat exchanger placed inside the test section of tower. Plain tube heat exchanger was designed and manufactured for the present work. The tubes were fixed horizontally in test section inside supported frame of rectangular drawer. Cooling water moves through the tubes while the spray water and air move over the tubes in perpendicular direction. The tubes are arrayed in staggered arrangement with (equilateral) tube pitch of $3D_o$ (pitch over diameter of 3).The specification of heat exchanger shown in Table (1).

The water distribution system in the cooling tower should distribute the water uniformly over the tube bundle and packing inside the tower, to be the most coefficient method of uniformly water distribution in counter flow wet-cooling tower a pressurized spray system used. The spray water passes through the spray nozzles and constantly distributed at the upper part of the test section, controlled by means of flow control valve globe type located downstream of the spray water pump. In the spray frame a header distributes or divides the deluge water into several conduits or lateral branches. Spray water nozzles were fitted the end of each lateral branch.

2.2. Test Procedure

In order to evaluate the thermal performance of cooling tower, a series of experiments was carried out at different operational and conformational parameters. Operational parameters demonstrate: air flow rate of (0.12-0.3) kg/s, spray water flow rate of (20,25,30,35,40,45) l/min, cooling water flow rate of (10,15,20,25,30,35,40,45,50) l/min, inlet cooling water temperature of (35,40,45,50,55)$^\circ$C and inlet air wet bulb temperature of (7-24)$^\circ$C. Conformational parameters indicate: height of packing used (280 & 560) mm, location of packing (under and above the heat exchanger).

Thermocouples type K inserted before and after the cooler coil to measured cooling water temperature. To measure the spray water temperatures at intermediate locations inside test section, specially channels have been manufacturing to insert thermocouples type K through holes. These holes are closed by rubber stoppers through which thermocouples are inserted to measure the temperature profile. The variations of air dry bulb temperature and relative humidity along the test section as well as the inlet and outlet of the tower were measured by humidity meter, which combined temperature/humidity sensor. The humidity meter model TH-305 has a temperature and relative humidity measurement range from 0 to 60°C and 20 to 95% respectively. The sensor probe handle is placed directly in the air stream and connected to display.

2.3. Performance Parameters

In viewpoint of energy analysis, the parameters used to determine the performance of cooling tower are:

1-Cooling range: is the temperature difference between the water inlet and exit states. Range can be measured by the temperature difference between the inlet and outlet from cooling tower:

$$CR = T_{cw,in} - T_{cw,out} \quad \text{... (11)}$$

2- Thermal efficiency: The most important parameter of cooling tower performance is the thermal efficiency, which can be defined as the ratio of actual released of heat to the maximum theoretical heat from cooling tower. The thermal efficiency for the closed circuit cooling towers was defined as [2&9]:

$$\eta = \frac{T_{cw,in} - T_{cw,out}}{T_{cw,in} - T_{awb,in}} \quad \text{... (12)}$$

3-Cooling capacity is the heat rejected or heat dissipation, given product of mass flow rate of water, specific heat and temperature difference.

$$q = \dot{m}_{cw} C_{p,cw}\, CR \quad \text{... (13)}$$

4-Mass transfer coefficient

The mass transfer coefficient obtained using enthalpy balance for an elementary transfer surface [2].

$$m_a dh_a \alpha_m \, (h_i - h_a) \; dA \qquad \dots (14)$$

Which is known as the Merkel equation and integrated for the whole heat exchanger in tower gives:

$$\frac{\alpha_m A}{\dot{m}_a} = ln \frac{h_i - h_{a,in}}{h_i - h_{a,out}} \qquad \dots \; (15)$$

where, α_m is the mass transfer coefficient for water vapor between spray water film an air, A is the surface area of the heat exchanger and h_i is the specific enthalpy of the saturated air at the mean spray water temperature .

The average of spray water temperatures was taken as the interface temperature according to [8] while the inlet and outlet air enthalpies were calculated from Psychometric chart according to the measured data. Outlet air enthalpy could be also calculated considering that all the heat goes from water to air [10]

$$\dot{m}_a \left(h_{a,out} - h_{a,in}\right) = \dot{m}_{cw} C_{p,cw} \left(T_{cw,in} - T_{cw,out} \;\right) \qquad \dots (16)$$

Then the outlet air enthalpy calculates as:

$$i_{a,out} = i_{a,in} + \frac{\dot{m}_{cw} C_{p,cw} \; (T_{cw,in} - T_{cw,out} \;)}{\dot{m}_a} \qquad \dots \; (17)$$

5-Heat transfer coefficient

Heat transfer from cooling water inside tubes to spray water and air through a water film .the rate of heat transfer from cooling water dq_c is given by [11]:

$$dq_c = \dot{m}_{cw} C_{\mathrm{p},cw} dT_{cw} \\ = -U_o \; (T_{cw} - T_{sw}) dA \qquad \dots (18)$$

Integrated Eq.18 from the inlet to outlet of cooling water, with constant spray water T_{sw}, gives.

$$\frac{U_o A_c}{C_{p,cw} m_{cw}} = ln \frac{T_{cw,in} - T_{sw,m}}{T_{cw,out} - T_{sw,m}} \qquad \dots \; (19)$$

where, U_o is the overall heat transfer coefficient between cooling water inside the tubes, tube wall and spray water on the outside .It is calculated by the following formula [4]:

$$\mathrm{U_o} = \left[\frac{R_o}{\mathrm{R}_i} \frac{1}{\alpha_c} + \frac{R_o}{\mathrm{k_t}} \ln \frac{R_o}{R_i} + \frac{1}{\alpha_s} \right]^{-1} \qquad \dots (20)$$

After the overall heat transfer coefficient was calculated from Eq.(19), it used to calculate, αs, tube to water film heat transfer coefficient (W/m^2 C).

$$\alpha_s = \left[\frac{1}{U_o} - \frac{R_o}{\mathrm{R}_i} \frac{1}{\alpha_c} - \frac{R_o}{\mathrm{k_{tube}}} \ln \frac{R_o}{R_i} \right]^{-1} \qquad \dots (21)$$

Where, α_c is the convection heat transfer coefficient of cooling water inside the tubes, it was calculated by the "**Dittuse-Boelter**" relation [12]:

$$\alpha_c = 0.023 \; \frac{k_{cw}}{D_i} Re^{0.8} \; Pr^{0.3} \qquad \dots \; (22)$$

Where, Reynolds number and Prandtl number were taken for the cooling water inside the tubes.

A MATLAB program was written to calculate the following parameters: water cooling range, tower approach, thermal efficiency, cooling capacity, heat transfer coefficient and mass transfer coefficient. The input data to this program is the measured parameters taken from the experimental runs.

3. Results and Discussion

3.1. Verification of the Experimental Apparatus

To verify the reliability of the experimental apparatus, energy balance of the air and cooling water was adopted using eq. (18). As shown in figure (2), the unbalance of the heat gained by the ambient air and the heat lost by the cooling water are within ±10%. The heat balance of the apparatus could be claimed to be satisfactory.

3.2. Influence of Air and Spray Water Flow Rates

The effect of air flow rate on the cooling capacity for different values of the spray water rate is illustrated in figure (3).For each value of spray flow rate, as the air flow rate increases; the cooling water range is increases therefore, cooling capacity increased. This can be explained by as the air flow rate increases, evaporated water per unit of air increases too. On the other hand, cooling capacity is increasing exponentially while the spray water flow rate is increasing as a result of increasing in the amount of water exposed to air during the unit time and providing a largest contact surface for the heat and mass transfer between water and air. The effect of air flow rate on thermal efficiency for different spray water flow rates illustrated in figure (4).The cooling tower thermal efficiency increases with the increase of air flow rate and spray water flow rate due to the increase in cooling range and the decrease in tower approach as its calculation from Eq. (12). This behavior was observed by **Yoo et. al. (2010), [13]**. Therefore, the best cooling tower thermal efficiency is achieved at the highest flow rates of air and flow rate of spray water as shown in figure (4). The variation of air-water mass transfer coefficient with air and spray water flow rates represented in figure (5). It can be seen that the mass transfer coefficient is in proportional relation with both air and spray water flow rates were increased. This is mainly because when spray water increases, the number of droplets increases so the air-water interfacial area increases with any increasing in air flow rate. Therefore, the mass transfer coefficient enhanced. The spray water heat transfer coefficient versus air flow rate with different spray water flow rates is shown in figure (6). It can be stated that the heat transfer coefficient is increasing with the increasing air and spray water flow rates. This can be explained by Eq. (19), when spray and air flow rate increase, the outlet cooling water temperature decreases, then the overall heat transfer coefficient increases. Also, spray heat transfer coefficient will be increases too. This behavior is determined by different experiments of authors such as **Shim et. al. (2010), [5]** and **Yoo et. al. (2010), [13]**.

3.3 .Influence of Cooling Water Flow Rate

The cooling capacity of tower versus cooling water flow rate with different spray water flow rates is shown in figure (7). It can be noticed that the cooling capacity is proportional with cooling and spray water flow rates. When the spray water flow rate remains constant, cooling capacity increases significantly to increase in cooling water flow rate in spite of decreasing in cooling water range for this case according to Eq. (13). This confirms the experimental results of **Shim et. al. (2008) [4].** The effect of cooling water flow rate on thermal efficiency for different spray water flow rates illustrated in figure (8).As expected, the thermal efficiency depends on the cooling and spray water flow rates. Thermal efficiency inversely proportional with cooling water flow rate which is due to decrease in in cooling range .When the cooling water is low and spray water flow rate is high, the higher value of thermal efficiency is achieved. The effect of cooling water flow rate on mass transfer coefficient for different spray water flow rates are illustrated in figure (9). Tt is indicated that the mass transfer coefficient depends on the cooling and spray water flow rates .As cooling water flow rate increases, mass transfer coefficient increases for all spray water flow rates. The effect of cooling water flow rate on spray water heat transfer coefficient for different spray water flow rates illustrated in figure (10). The results indicated that the heat transfer coefficient increases with increasing cooling and spray water flow rates. This is simply because of the increasing in the cooling capacity. As cooling water flow rate increases, it will be leads to an increase in the overall heat transfer coefficient and thus the heat transfer coefficient increases.

3.4. Influence of Inlet Cooling Water Temperature

Cooling capacity with respect to variable inlet cooling water temperature and spray water flow rate has been shown in figure (11).It is shown that if the spray water flow rate remains constant, cooling capacity increases rapidly with the increase of inlet cooling water temperature due to increase in rate of heat and mass transfer. This behaviour is determined by different experiments of authors **Shim et. al. (2008), [4]** and **Yoo et. al. (2010), [13]**. Figure (12) indicate the effect of variable inlet cooling water temperature upon the tower thermal efficiency for different values of spray water flow rates. The thermal efficiency increases almost exponentially as the inlet cooling water temperature increases for all values of spray water flow rates. The thermal efficiency is high at higher inlet cooling water temperature and spray water flow rate. Small increment at low water temperature will gradually increases with an increase in water temperature. The effect of inlet cooling water temperature on the mass transfer coefficient for different values of the spray water flow rate is illustrated in figure (13). The mass transfer coefficient increases almost exponentially as the inlet cooling water temperature increases for all values of spray water flow rates. The spray water temperature is proportional to the cooling water temperature. The vapour pressure driving force was increased at higher spray water temperature caused an increase in mass transfer rate as well as mass transfer coefficient. Also, it can be observed from this figure that the decrement of the rate of mass transfer coefficient at high inlet cooling water temperature is increased because of the increasing in rate water evaporated. Spray water heat transfer coefficient with respect to variable inlet cooling water temperature and spray water flow rate has been shown in figure (14). For each value of inlet cooling water temperature, as the spray water flow rate is increasing, the spray water heat transfer coefficient is increasing too .This is mostly because at constant cooling water flow rate, the increasing in cooling capacity, as inlet cooling water temperature increases, lead to an increase in the overall heat transfer coefficient and thus the heat transfer coefficient.

3.5 .Influence of Inlet Air Wet Bulb Temperature (AWBT)

The change in cooling capacity versus inlet AWBT for different inlet cooling water temperature presented in figure (15). It is clear that the cooling capacity in inversely proportional with the inlet AWBT for both inlet cooling water temperatures. It is believed because any increase in inlet AWBT reflected to decreases the enthalpy potential between saturated vapour mixture (film surrounding the water droplet) and surrounding air.

The effect of inlet AWBT on tower thermal efficiency for different inlet cooling water temperatures is investigated in figure (16). It can be seen for both inlet cooling water temperatures that the thermal efficiency decreased as inlet AWBT increased which is brought about by the temperature fall at outlet of the heat exchanger. This behavior was observed by **Sarker (2007), [14]**. Also, it can be apparent that higher tower thermal efficiency achieved at higher inlet cooling water temperature. Effect of inlet AWBT on mass transfer coefficient for different inlet water temperatures illustrated in figure (17).The mass transfer coefficient increases almost linearly with the increase of inlet AWBT for both inlet water temperatures. This is because the rate of heat and mass transfer influenced greatly with the difference between the inlet spray water and air temperatures. As temperatures difference increases, the enthalpy change between outlet and inlet air increase too. Higher temperature difference achieved at lower AWBT causing a decrease in rate of mass transfer coefficient according to Eq. (15). In figure (18), the spray water heat transfer coefficient for different inlet cooling water temperature has been shown against inlet AWBT. The result shows that the heat transfer coefficient decreases with the increasing inlet AWBT for constant inlet water temperature, while heat transfer coefficient increases with the increasing inlet cooling water temperature. This is mainly because when water is warmer than air, there is a tendency for the air to cool the water. The air then gets hotter as it gains the sensible heat of the water and the water is cooled as its sensible heat is transfer to the air.

3.6 .Influence of Added Packing

As can be seen from figure (19), there is a significant variation in the cooling capacity of cooling tower with added packing on CWCT. The result indicated that the cooling capacity for added packing height of (280 mm) & (560 mm) approximately (6%) & (28%) higher than that CWCT respectively. Figure (20) gives an indication to the thermal efficiency enhancement of added packing on conventional CWCT. It can be observed that the thermal efficiency for added packing height of (280 mm) & (560 mm) approximately (12%) & (40%) higher than that CWCT respectively.

3.7. Influence of Packing Location

Figure (21) shows the cooling capacity comparing for different positions of packing. The result indicated that the cooling capacity for CWCT with packing lower under the heat exchanger and CWCT with packing above the heat exchanger approximately (28%) & (16%) higher than that CWCT respectively. In figure (22), the thermal efficiency enhancement for different positions of packing is illustrated. It can be observed that the thermal efficiency for CWCT with packing under the heat exchanger and CWCT with packing above the heat exchanger approximately (40%) & (25%) higher than that CWCT respectively.

3.8. Empirical Correlations

According to the results of the experiments of this work, for different operational parameters, correlations for heat and mass transfer coefficients were developed for cooling tower operates without packing. These correlations are:

a-Mass transfer coefficient

$$\alpha_m = 0.000001(G_{air})^{0.5038}(G_{sw})^{0.7456}(T_{cw})^{2.4478} \quad \ldots \ (23)$$

b-Heat transfer coefficient

$$\alpha_s = 0.1349(G_{sw})^{0.3758}(G_{cw})^{0.2051}(T_{cw})^{1.7749} \quad \ldots \ (24)$$

The average roots square mean error between correlations and experimental data for mass and heat transfer was (0.9666), (0.9424) respectively.

4. Conclusions

The CWCT with packing has a better performance than without packing. Furthermore, it is noticed that the height of packing (560 mm) has a significant effect on tower performance in comparison with (280 mm) packing height. The result indicated that the cooling capacity for added packing height of (280 mm) & (560 mm) approximately (6%) and (28%) higher than that conventional CWCT respectively. Comparing CWCT with packing for both locations under and above the heat exchanger, it has been observed that the best performance for the CWCT with packing under the heat exchanger. The result indicated that cooling capacity for CWCT with packing under the heat exchanger and CWCT with packing above the heat exchanger approximately (28%) and (16%) higher than that CWCT without packing respectively.

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Nomenclature

A=total heat transfer area, m^2
Cp=specific heat at constant pressure, kJ/kg ^oC
CR=cooling range, ^oC
D=tube diameter, m
G=mass flux , $kg/m^2.s$
h=specific enthalpy, kJ/kg
k=thermal conductivity, W/m ^oC
$\dot{m}$=mass flow rate, kg/s
Q=volume flow rate, l/min
q=cooling capacity, W
Pr=Prandtl number
R=tube radius, m
Re=Reynolds number
T=temperature , ^oC
U_o=overall heat transfer coefficient, $W/m^2\ ^oC$

Greek Symbols

α_m= mass transfer coefficient for water vapour, between spray water film and air, $kg/m^2\ s$
α_s =heat transfer coefficient between tube external surface and spray water film, $W/m^2\ ^oC$
α_c =heat transfer coefficient for water inside the tubes, $W/m^2\ ^oC$
η = thermal efficiency,(%)
Γ= Spray water mass flow rate per length of tube, kg/m s
Φ=relative humidity, %
ω= humidity ratio, $kg/kg_{dry\ air}$

Subscripts

a=air
cw=cooling water
i=Interface between spray water film & air in=inlet
m=mean
out=outlet
sw=spray water
t=tube

Appendix A
Sample of Calculations

Sample calculation for parameters of thermal performance cooling tower for measured conditions of operation and configuration that shown in tables (A.1) & (A.2)

Air flow rate:

From psychometric chart at (To=31.2 ˚C & Φ_o=26.8 %) w_o=0.00757239 $kg/kg_{\ dry\ air}$
To=31. 2˚C =31.2+273.15=304.35 K
From Bernoulli equation and manometer reading:

$$Q_{orifice} = 0.01423366\sqrt{\Delta h \times T_0 \times (1 + w_o)} \quad (A.1)$$

$$Q_{orifice} = 0.0142337\sqrt{0.46 \times 304.35 \times (1 + 0.0076)} = 0.169\,\frac{m^3}{s}$$

From calibration of orifice plate by Pitot tube:

$$Q_a = Q_{orifice} \times 1.375 - 0.035 \qquad (A.2)$$

$Q_a = 0.169 \times 1.375 - 0.035 = 0.1974 \frac{m^3}{s}$

$\rho_o = \frac{P_o}{RT_0} = \frac{101.325}{0.287 \times 304.35} = 1.16 \frac{kg}{m^3}$

To determine air mass flow rate:

$\overset{\circ}{m}_a = \frac{\rho_o \times Q_a}{(1+w_o)} = \frac{1.16 \times 0.1974}{(1+.00757239)} = 0.2272 \frac{kg}{s}$

Spray water:

Tsw,m=33˚C=33+273.15=306.15 K

Density of water can be determined [15]:

$$\rho_w = [1.49343 \times 10^{-3} - 3.7164 \times 10^{-6} \times (T) + 7.09782 \times 10^{-9} \times (T)^2 - 1.90321 \times 10^{-20} \times (T)^6]^{-1} \qquad (A.3)$$

For spray water:

$\rho_{sw} = [1.49343 \times 10^{-3} - 3.7164 \times 10^{-6} \times (306.15) + 7.09782 \times 10^{-9} \times (306.15)^2 - 1.90321 \times 10^{-20} \times (306.15)^6]^{-1} = 994.7809 \frac{kg}{m^3}$

To determine spray water mass flow rate:

$\overset{\circ}{m}_{sw} = \rho_{sw} \times Q_{sw} = 994.7801 \frac{45}{60000} = 0.7461 \frac{kg}{s}$

Cooling water:

$T_{cw,bulk}=(T_{cw,i}+T_{cw,o})/2$=(46+39.7)/2=42.85˚C=42.85+273.15=316K

From Eq. (A.3), density of cooling water can be determined:

$\rho_{cw} = [1.49343 \times 10^{-3} - 3.7164 \times 10^{-6} \times (316) + 7.09782 \times 10^{-9} \times (316)^2 - 1.90321 \times 10^{-20} \times (316)^6]^{-1} = 991.22 \frac{kg}{m^3}$

To determine cooling water mass flow rate:

$\overset{\circ}{m}_{cw} = \rho_{cw} \times Q_{cw} = 991.22 \frac{10}{60000} = 0.1652 \frac{kg}{s}$

1-Cooling range:

Cooling range can be determined by Eq. (11):

$CR = 46 - 39.7 = 6.3\ ℃$

2- Thermal efficiency:

From psychometric chart at (Tadb,in=35.62 ˚C & Φ_1=20.79 %) $Tawb_{in}$=19.315℃

Thermal efficiency can be determined by Eq. (12):

$$\eta = \frac{46 - 39.7}{46 - 19.315} = 23.61\ \%$$

3- Cooling capacity

Cooling capacity of water can be determine [15]:

$C_{pw} = 8.15599 \times 10^3 - 2.80627 \times 10 \times (T) + 5.11283 \times 10^{-2} \times (T)^2 - 2.17582 \times 10^{-13} \times (T)^6 \qquad (A.4)$

From Eq. (A. 4), at $T_{cw,bulk}$=316˚C

$C_{pcw} = 8.15599 \times 10^3 - 2.80627 \times 10 \times (316) + 5.11283 \times 10^{-2} \times (316)^2 - 2.17582 \times 10^{-13} \times (316)^6 = 4177 \frac{J}{kg.K}$

Cooling capacity can be determined using Eq.(13):

$$q = 0.1652 \times 4177 \times 6.3 = 4347.25\ W = 4.347 kW$$

4- Mass transfer coefficient

From psychometric chart at (Tadbi=31.4˚C & Φ_i=77 %) $h_{a,in}$= 89185.3 J/kg

From psychometric chart at (Tadb2=32.5˚C & Φ_2=97 %) $h_{a,out}$=110942.97 J/kg

$T_{sw,av}=(T_{sw,m}/2)+273$=(33/2)+273.15=289.65K

i_{fgwo} = 2.5016×10^6 J/kg (evaluated at 273.15K)

air specific heat can be determine [15]:

$$C_{pa} = 1.045356 \times 10^3 - 3.161783 \times 10^{-1} \times (T) + 7.083814 \times 10^{-4} \times (T)^2 - 2.705209 \times 10^{-7} \times (T)^3 \qquad (A.5)$$

From Eq.(A.5) at $T_{sw,av}$

$C_{pa} = 1.045356 \times 10^3 - 3.161783 \times 10^{-1} \times (289.65) + 7.083814 \times 10^{-4} \times (289.65)^2 - 2.705209 \times 10^{-7} \times (289.65)^3 = 1006.63 \frac{J}{kg.K}$

Saturated vapor specific heat can be determined [15]:

$C_{pv} = 1.3605 \times 10^3 + 2.31334 \times (T) - 2.46784 \times 10^{-10} \times (T)^5 + 5.91332 \times 10^{-13} \times (T)^6 \qquad (A.6)$

From Eq.(A.6):

$C_{pv} = 1.3605 \times 10^3 + 2.31334 \times (289.65) - 2.46784 \times 10^{-10} \times (289.65)^5 + 5.91332 \times 10^{-13} \times (289.65)^6 = 1876.6224 \frac{J}{kg.K}$

The vapor pressure can be determined [15]:

$p_{vwm} = 10^Z \qquad (A.7)$

$Z = 10.79586 \times \left(1 - \frac{273.16}{Tw}\right) + 5.02808 \times \log_{(10)}\left[\frac{273.16}{Tw}\right] + 0.000150474 \times \left[1 - 10^{-8.29692 \times \left(\frac{Tw}{273.16-1}\right)}\right] + 0.00042873\left[10^{+4.76955 \times \left(\frac{1-273.16}{Tw}\right)} - 1\right] + 2.786118312 \qquad (A.8)$

The vapor pressure can be determined at the average spray water temperature from Eqs.(A.7) & (A.8)

$Z = 10.79586 \times \left(1 - \frac{273.16}{306.15}\right) + 5.02808 \times \log_{(10)}\left[\frac{273.16}{306.15}\right] + 0.000150474 \times \left[1 - 10^{-8.29692 \times \left(\frac{306.15}{273.16-1}\right)}\right] + 0.00042873\left[10^{+4.76955 \times \left(\frac{1-273.16}{306.15}\right)} - 1\right] + 2.786118312 = 3.7016$

$p_{vwm} = 10^{3.7016} = 5030.4\ Pa$

The corresponding humidity ratio can be determined by **Heyns** [6]:

$\omega_{wm} = \left[\frac{0.62509 P_{vwm}}{P_a - 1.005 P_{vwm}}\right] = \left[\frac{0.62509 \times 5030.4}{101325 - 1.005 \times 5030.4}\right] = 0.032663 \frac{kg}{kg_{dry\ air}}$

The enthalpy of saturated air at mean spray water temperature can be calculated from:

$h_i = C_{pa} \times T_{sw,m} + \omega_{wm}\left(i_{fgwo} + C_{pv} \times T_{sw,m}\right) = 1006.63 \times 33 + 0.032663(2.5016 \times 10^6 + 1876.6224 \times 33) = 116950\ J/kg$

Finally, mass transfer coefficient can be calculated from Eq.(15)

$\alpha_m = \frac{0.2272}{1.03269} \ln \frac{(116950 - 89185.3)}{(116950 - 110942.97)} = 0.3367 \frac{kg}{m^2.s}$

5- Heat transfer coefficient

Cooling water mass flow rate for one tube is

$\overset{\circ}{m}_{cw} = \frac{0.1652}{5} = 0.0275 \frac{kg}{s}$

Velocity of cooling water inside tube:

$$V_c = \frac{\mathring{m}_{cw}}{\rho_{cw} A_c} = \frac{0.0275}{991.22 \times \frac{\pi}{4}(0.015054)^2} = 0.18727\,\frac{m}{s}$$

Dynamic viscosity of water can be determined [15]:

$$\mu_{cw} = 2.414 \times 10^{-5} \times 10^{\left(\frac{247.8}{T-140}\right)} \qquad (A.9)$$

Dynamic viscosity of cooling water from Eq. (A.9):

$$\mu_{cw} = 2.414 \times 10^{-5} \times 10^{\left(\frac{247.8}{316-140}\right)} = 0.00061578\,\frac{kg}{m.s}$$

$$Re_{cw} = \left(\frac{\rho_{cw} V_c D_i}{\mu_{cw}}\right) = \frac{991.22 \times 0.18727 \times 0.015054}{0.00061578} = 4538$$

Thermal conductivity of water can be determined [15]:

$$k_{cw} = -6.14255 \times 10^{-1} + 6.9962 \times 10^{-3} \times (316) - 1.01075 \times 10^{-5}(316)^2 + 4.74737 \times 10^{-12}(316)^4 \qquad (A.10)$$

Thermal conductivity for cooling water calculated from Eq.(A.10)

$$k_{cw} = -6.14255 \times 10^{-1} + 6.9962 \times 10^{-3} \times (316) - 1.01075 \times 10^{-5}(316)^2 + 4.74737 \times 10^{-12}(316)^4 = 0.634586\,\frac{W}{m.K}$$

Prandtl number for cooling water calculated as:

$$Pr_{cw} = \left(\frac{\mu_{cw} C_{pcw}}{k_{cw}}\right) = \frac{0.00061578 \times 4177}{0.634586} = 4.0532$$

Convection heat transfer inside tube from Eq.(22)

$$\propto_c = 0.023 \times (4538)^{0.8} (4.0532)^{0.3} \times \frac{0.634586}{0.015054} = 1242.78\,\frac{W}{m^2.K}$$

From Eq.(20) , overall heat transfer coefficient can calculated:

$$U_o = \frac{0.1652 \times 4177}{1.03269} \ln\left(\frac{46-33}{39.7-33}\right) = 442.908\,\frac{W}{m^2.K}$$

Thermal conductivity of cooper tube is:

k_{tube}=398 W/m.K

Film heat transfer coefficient calculated from Eq.(21)

$$\propto_s = \left[\frac{1}{442.908} - \frac{0.01588}{0.015054}\,\frac{1}{1242.78} - \frac{0.01588}{2 \times 398} \ln\frac{0.01588}{0.015054}\right]^{-1} = 710.256$$

Table 1: Physical dimension of heat exchanger

Heat exchanger configuration	Value	Unit
Length	690	mm
Height	166	mm
Width	381	mm
Tubes for coil	30	-
Vertical tube spacing	24	mm
Horizontal tube spacing	80	mm
Tube per row	5	-
Outside tube diameter	15.88	mm
Tube thickness	0.81	mm
Total heat transfer area	1032691.77	mm^2
Minimum free flow area	209148	mm^2

Table A.1: Measured operating conditions

	Items	Value
1	Ambient air dry-bulb temperature	To=31.2 ˚C
2	Ambient relative humidity	Φ_o=26.8 %
3	Atmospheric pressure	Pa=101325 a
4	Inlet cooling water temperature	$T_{cw,i}$=46˚C
5	Outlet cooling water temperature	$T_{cw,o}$=39.7˚C
6	Mean temperature of spray water	$T_{sw,m}$=33˚C
7	Inlet spray water temperature (from upper)	T_{sw1}=32.2˚C
8	Outlet spray water temperature	T_{sw2}=31.8˚C
9	Manometer reading	Δh=45.9 cm
10	Spray water flow rate	Q_{sw}=45 l/min
11	Cooling water flow rate	Q_{cw}=10 l/min
12	Inlet air dry-bulb temperature to tower (from bottom)	T_{adb1}=35.62˚C
13	Inlet relative humidity to tower	Φ_1=20.79 %
14	Inlet air dry-bulb temperature to heat exchanger (from bottom)	T_{adbi}=31.4˚C
15	Inlet relative humidity to tower	Φ_i=77 %
16	Outlet air dry-bulb temperature to tower (from upper)	T_{adb2}=32.5˚C
17	Outlet relative humidity to tower	Φ_2= 97 %
18	Temperature of makeup water	T_{mak}=29.9˚C

Table A.2: Configuration conditions

	Items	Value
1	Type of heat exchanger tubes arrangement	Staggered
2	Type of spray nozzle	Jet nozzle
3	Length of packing used	560 mm
4	Location of packing	Under the heat exchanger
5	Outer diameter of pipe	15.88 mm
6	Inner diameter of pipe	15.054 mm
7	Number of rows	5
8	Number of the columns	6
9	Length of the heat exchanger	690 mm
10	Width of the heat exchanger	381mm

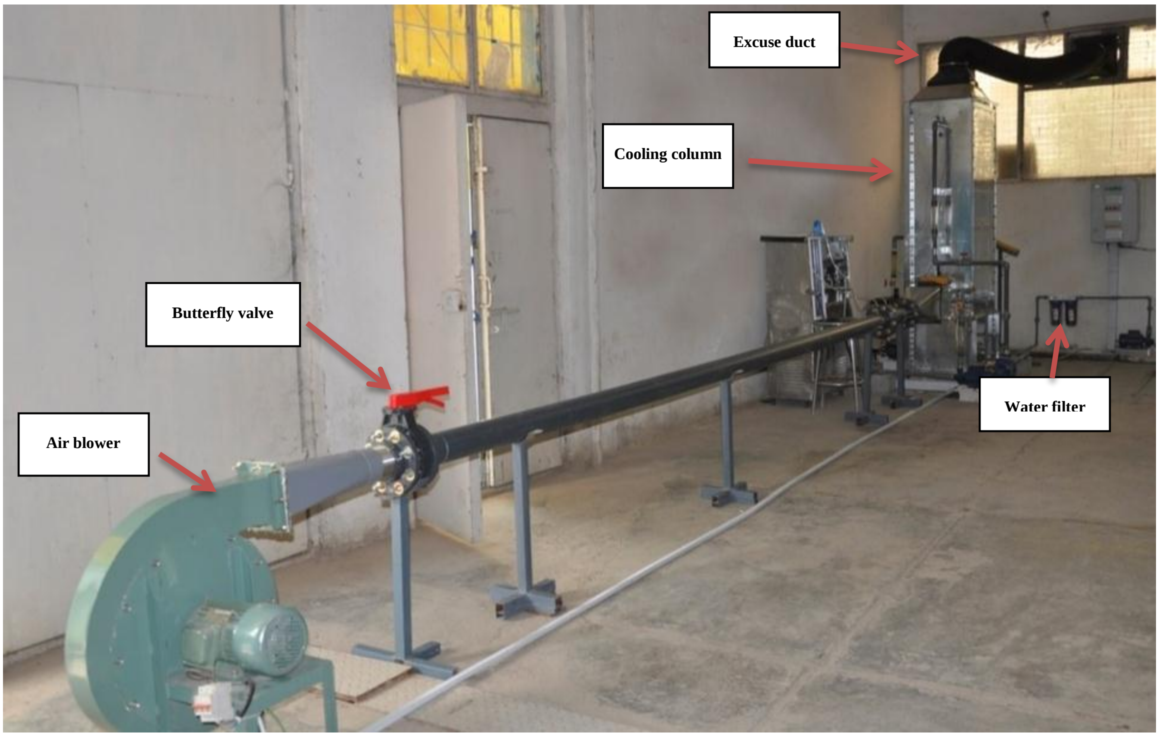

Figure 1a: photographic picture for experimental apparatus (lateral view)

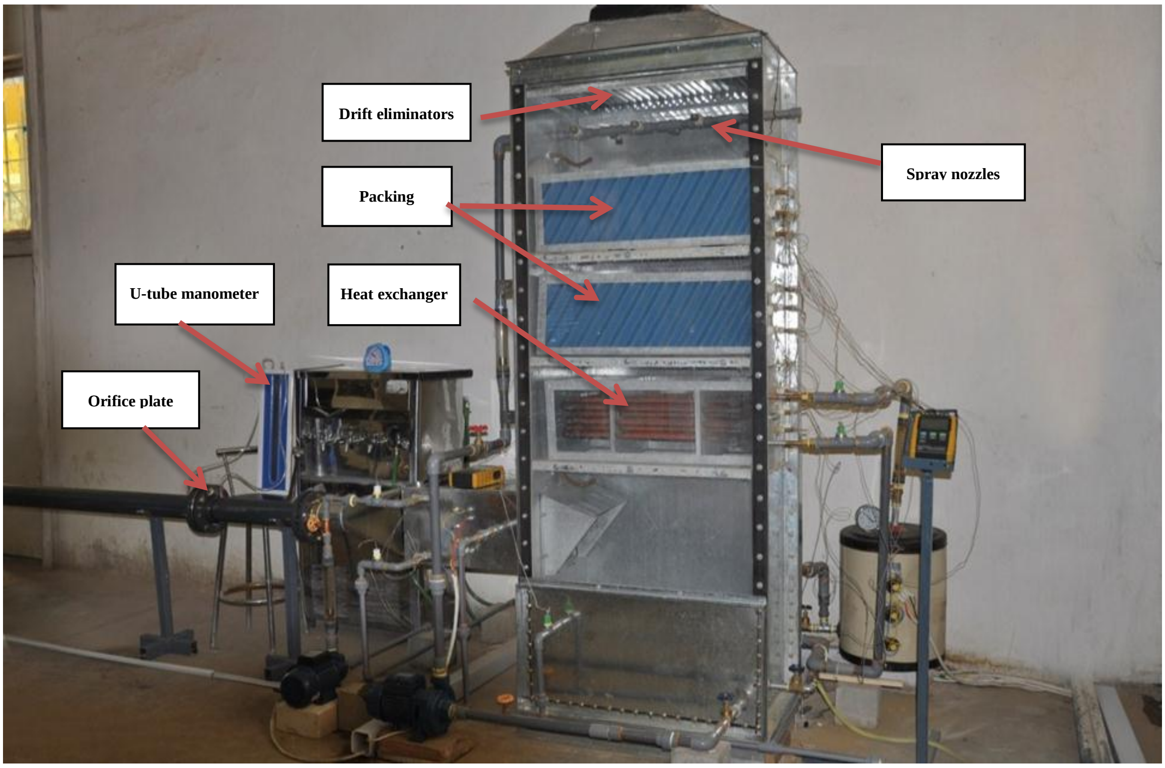

Figure 1b: Photographic picture for experimental apparatus (front view)

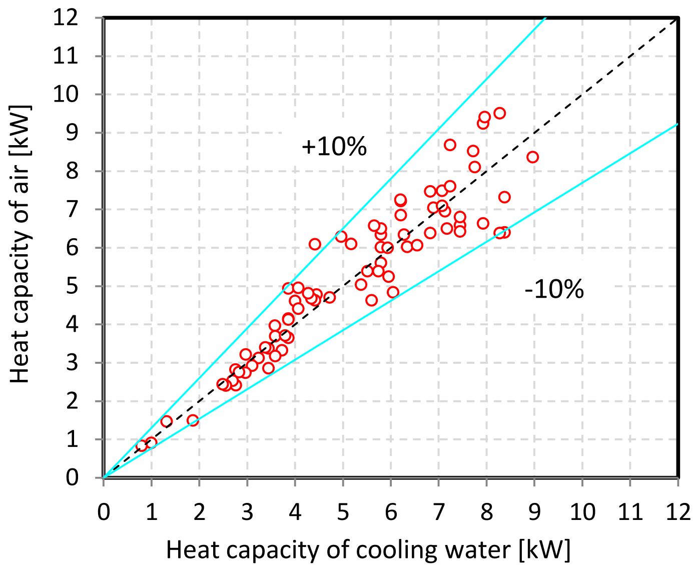

Figure 2: Energy balance of the experimental apparatus

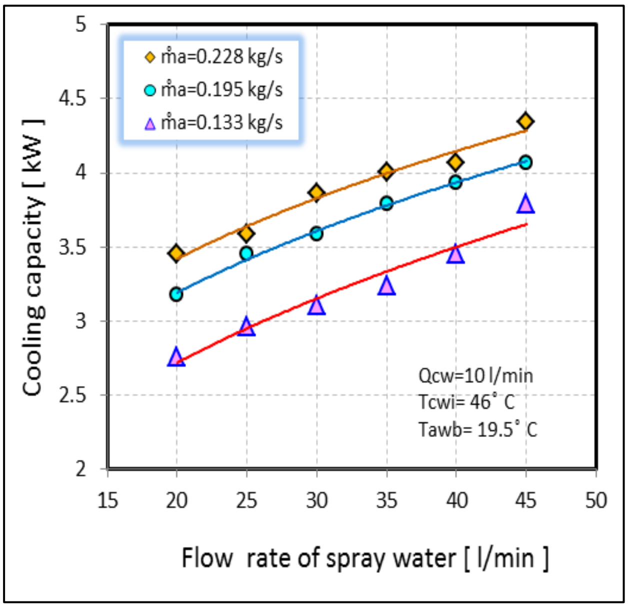

Figure 3: variation of cooling capacity with spray water flow rate for different air flow rates

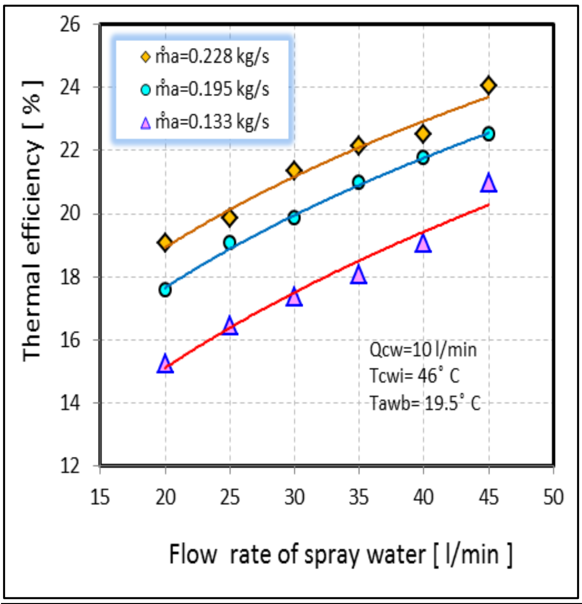

Figure 4: Variation of thermal efficiency with spray water flow rate for different air flow rates

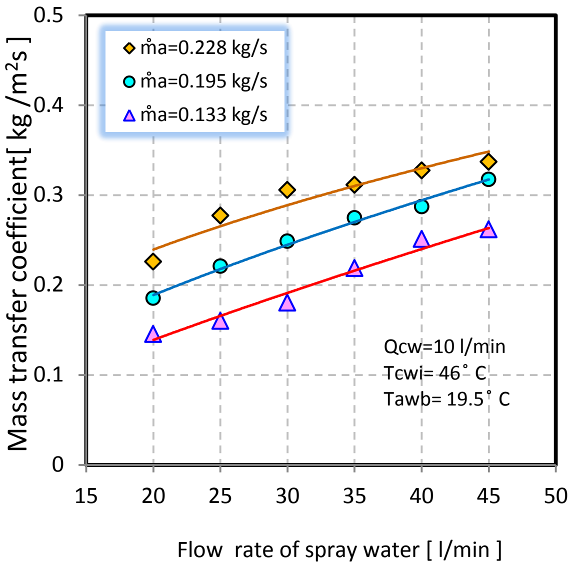

Figure 5: Variation of mass transfer coefficient with spray water flow rate for different air flow rates

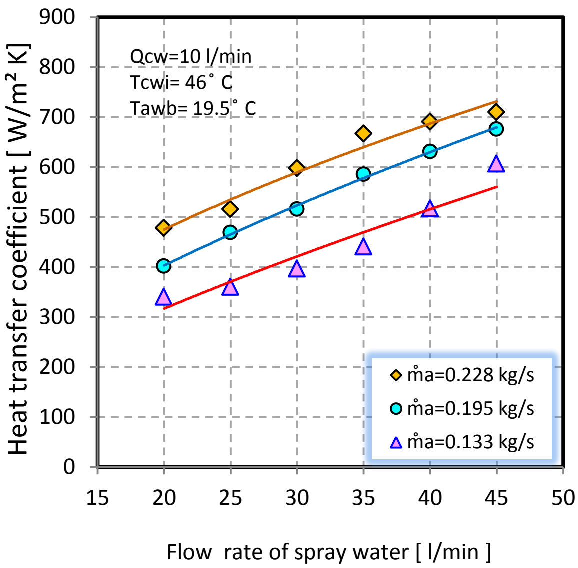

Figure 6: variation of heat transfer coefficient with spray water flow rate for different air flow rates

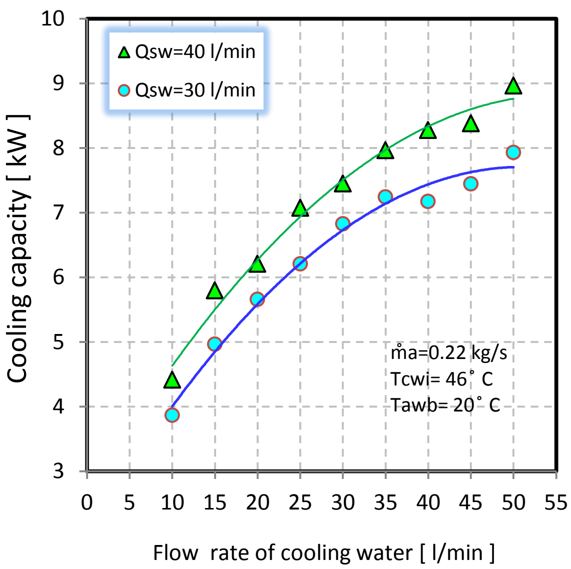

Figure 7: variation of cooling capacity with cooling water flow rate for different spray water flow rates

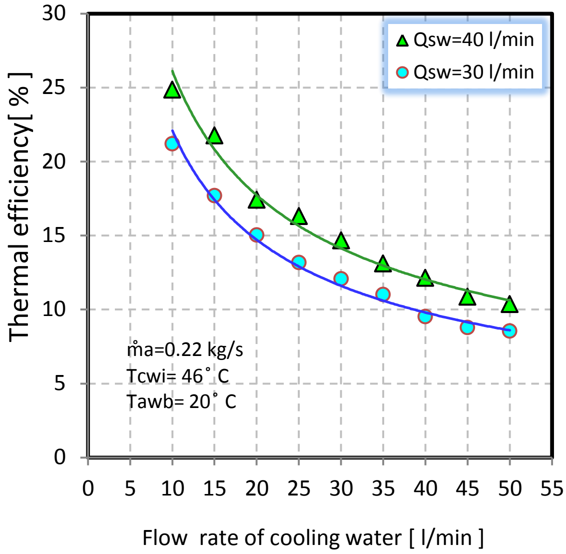

Figure 8: Variation of thermal efficiency with cooling water flow rate for different spray water flow rates

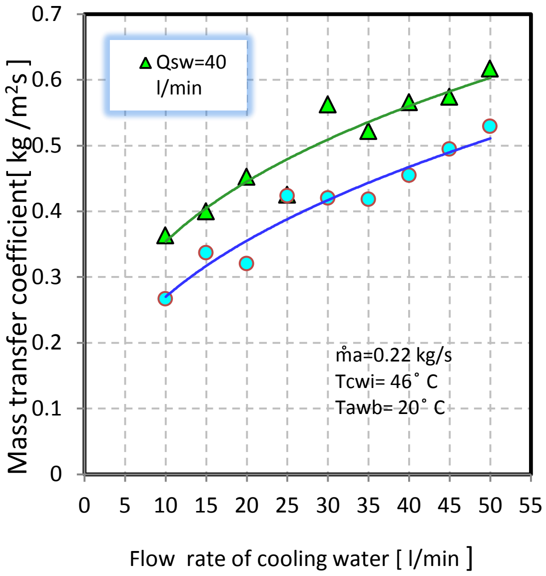

Figure 9: Variation of mass transfer coefficient with cooling water flow rate for different spray water flow rates

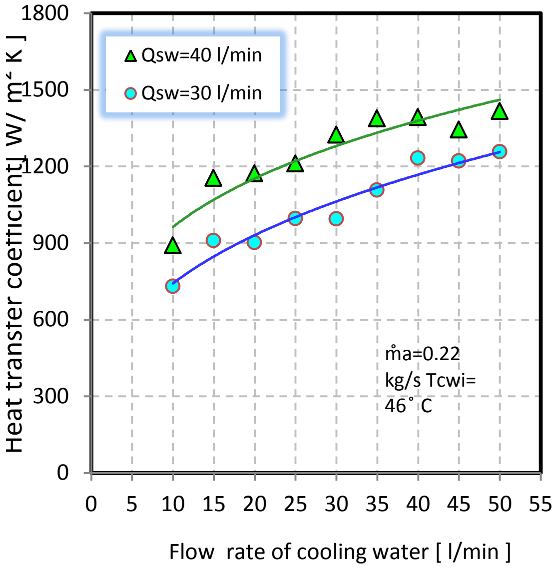

Figure 10: Variation of heat transfer coefficient with cooling water flow rate for different spray water flow rates

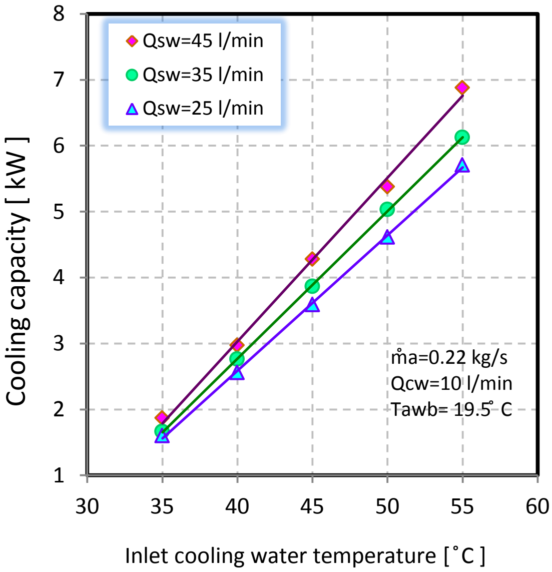

Figure 11: Variation of cooling capacity with inlet cooling water temperature for different spray water flow rates

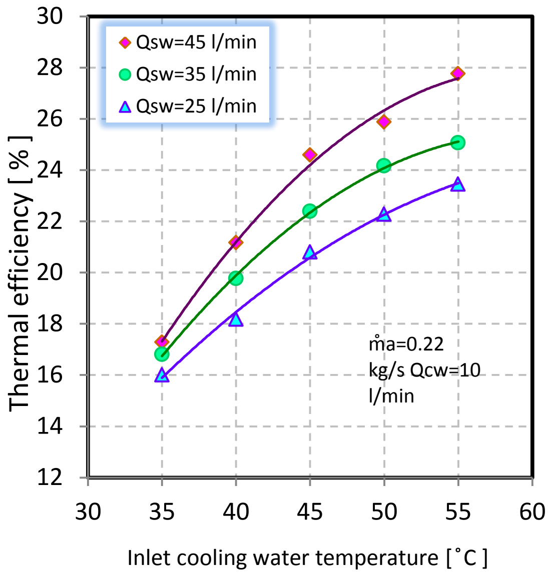

Figure 12: Variation of thermal efficiency with inlet cooling water temperature for different spray water flow rates

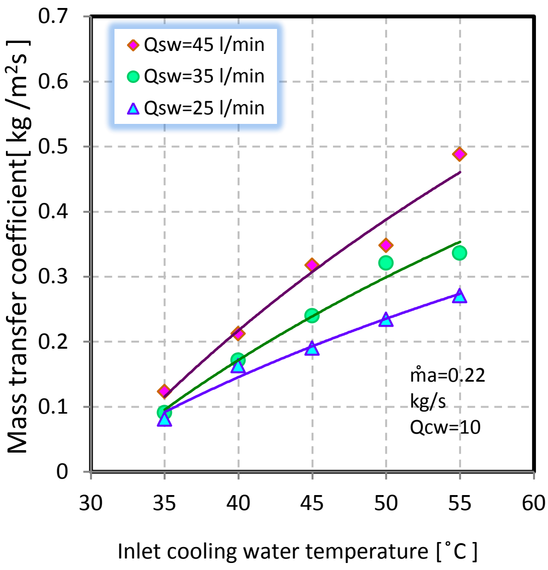

Figure 13: Variation of mass transfer coefficient with inlet cooling water temperature for different inlet water temperatures

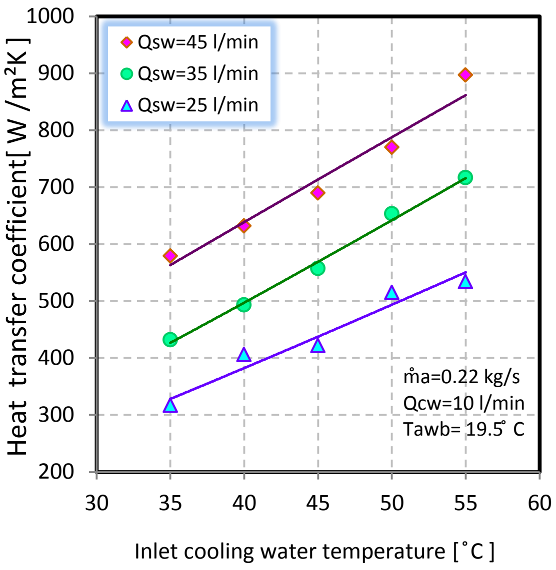

Figure 14: Variation of heat transfer coefficient with inlet cooling water temperature for different spray water flow rates

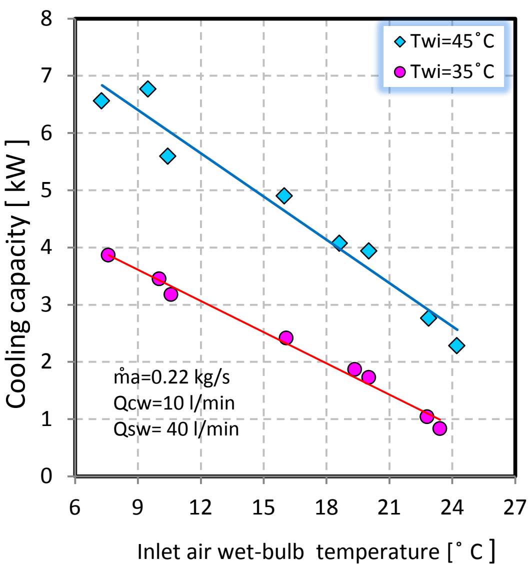

Figure 15: Variation of cooling capacity with inlet AWBT for different inlet water

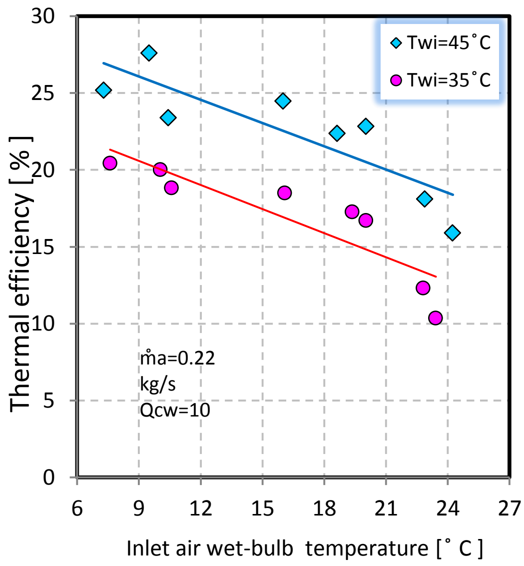

Figure 16: Variation of thermal efficiency with inlet AWBT for different inlet water temperatures

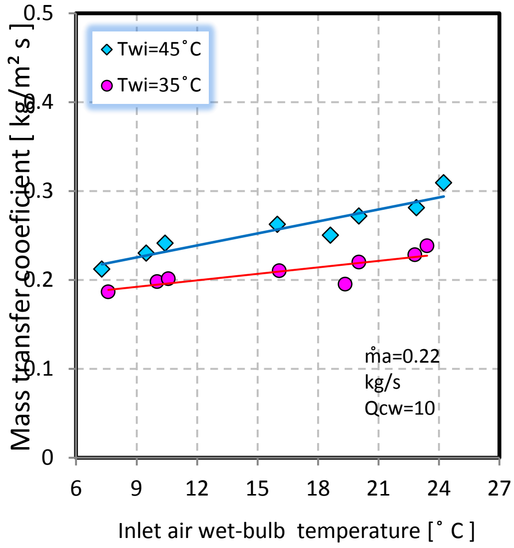

Figure 17: Variation of mass transfer coefficient with inlet AWBT for different inlet water temperatures

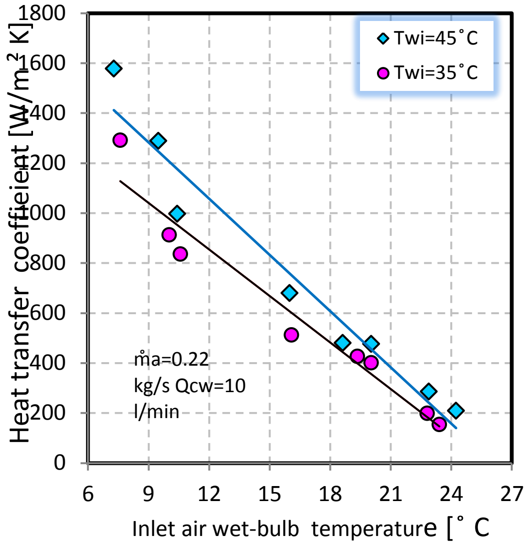

Figure 18: Variation of heat transfer coefficient with inlet AWBT for different inlet water temperatures

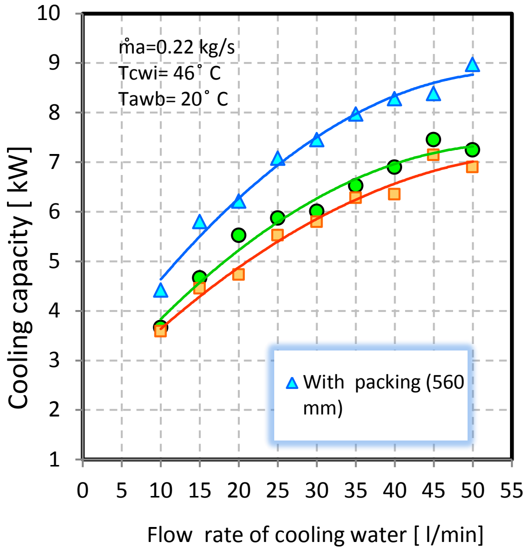

Figure 19: Variation of cooling capacity with cooling water flow rate for different heights of packing

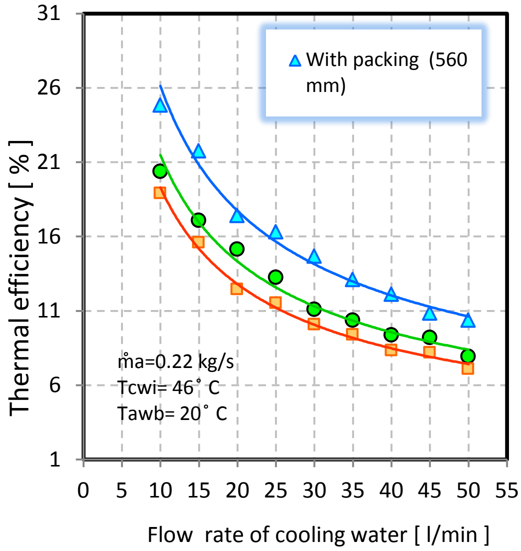

Figure 20: Variation of thermal efficiency with cooling water flow rate for different heights of packing

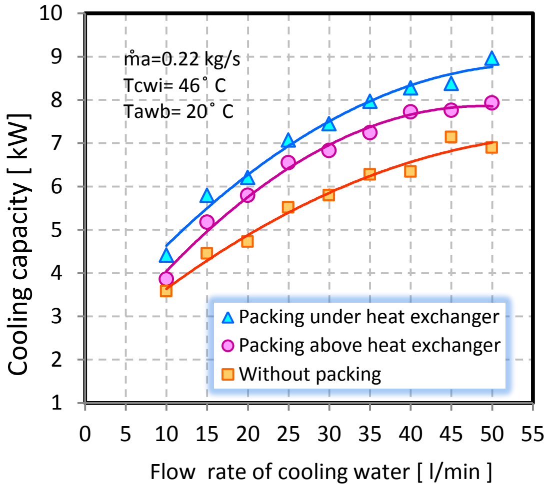

Figure 21: Variation of cooling capacity with cooling water flow rate for different locations of packing

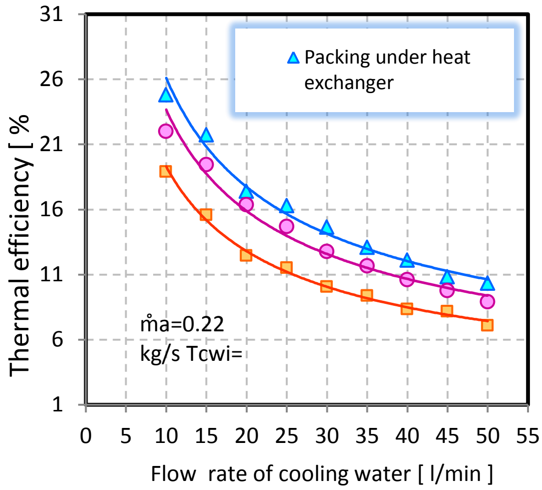

Figure 22: Variation of thermal efficiency with cooling water flow rate for different locations of packing

التحقيق العملي للأداء الحراري على برج تبريد رطب مغلق مطوّر

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الخلاصة

يحرص الباحثون والمصممون على رفع كفاءة أبراج التبريد لما لذلك من تأثير كبير على عمل وأداء المنظومات المرتبطة بهذه المنظومات. لهذا السبب تم تصميم وتصنيع واختبار برج تبريد رطب مغلق مطوّر بإضافة الحشوات بقدرة تبريد (9kW) في العراق . سلسة من التجارب تم اعتمادها لمتغيرات تشغيلية وتصميمة مختلفة . المتغيرات التشغيلية شملت: معدل تدفق الهواء ,معدل تدفقات ماء الرش و ماء التبريد , درجة حرارة الدخول لماء التبريد ودرجة حرارة الهواء الرطبة.أما المتغيرات التصميمة فهي استخدام أطوال مختلفة للحشوة لمواقع متغيرة . وضحت النتائج تحسين ملموس للأداء الحراري عند إضاقة الحشوة إلى البرج المغلق اسفل واعلى المبادل الحراري بالمقارنة مع برج التبريد المغلق بدون استخدام الحشوات. تم استنباط عدد من العلاقات التجريبية للتنبؤ بمعاملي انتقال الحرارة والكتلة بدلالة للمتغيرات التشغيلية.