Thermal Characteristics of Closed Wet Cooling Tower Using Different Heat Exchanger Tubes Arrangement

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ABSTRACT

This paper presents thermal characteristics analysis of a modified Closed Wet Cooling Tower (CWCT) based on heat and mass transfer principles to improve the performance of this tower in Iraq. A prototype of CWCT optimized by added packing was designed, manufactured and tested for cooling capacity of 9 kW. Experiments are conducted to explore the effects of various operational and conformational parameters on the thermal performance. In the test section, spray water temperature and both dry bulb temperature and relative humidity of the air measured at intermediate points of the heat exchanger and packing. Heat exchangers consist of four rows and eight columns for an inline tubes arrangement and six rows and five columns for staggered tubes arrangement. According to experimental data, the inline tubes arrangement has a better thermal performance than the staggered tubes arrangement. The inline tubes arrangement enhanced thermal efficiency more than (7%) compared to the staggered tubes arrangement. Furthermore the effect of added packing to CWCT on thermal performance was significant compared to CWCT without packing. Comparing CWCT with packing, it has been observed that the best performance for the CWCT with packing under heat exchanger. It can be watched that the thermal efficiency for CWCT with packing under heat exchanger and CWCT with packing above heat exchanger approximately (28%) and (16%) higher than that CWCT without packing respectively. This study provides correlations to predict heat and mass transfer considering the influences of operational parameters for both inline and staggered heat exchanger tubes arrangement.

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

الخصائص الحرارية لبرج تبريد رطب مغلق باستخدام ترتيب مختلف لانابيب المبادل الحراري

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الخلاصة

يتناول هذا البحث تحليل الخصائص الحرارية لبرج تبريد مغلق مطوّر وفقاً لمبادئ انتقال الحرارة والكتلة لتحسين أداء هذا البرج في العراق. تم تصميم وتصنيع واختبار نموذج لبرج تبريد رطب مغلق مطوّر بإضافة الحشوات لسعة تبريد (9kW) . أجريت التجارب لبيان تأثيرات المعاملات التشغيلية والتصميمة على الأداء الحراري للبرج. في مقطع الاختبار تم قياس درجة

حرارة ماء الرش وكلاً من درجة الحرارة الجافة والرطوبة النسبية للهواء في نقاط متوسطة للمبادل الحراري والحشوة . المبادل الحراري يتكون من أربعة صفوف وثمانية أعمدة للترتيب المتتالي للأنابيب و ستة صفوف وخمسة أعمدة للترتيب المتعاقب للأنابيب.أوضحت النتائج المستحصلة إن الأداء الأفضل للمبادل الحراري هو عند ترتيب الأنابيب المتتالي مقارنة بالترتيب المتعاقب .الترتيب المتتالي للأنابيب يحسن الكفاءة الحرارية بمقدار يزيد على (7%) مقارنة مع الترتيب المتعاقب. علاوة على ذلك فان إضافة الحشوة إلى البرج المغلق له تأثير فعّال على الأداء الحراري للبرج .مقارنة مع برج التبريد المغلق بوجود الحشوة لوحظ أن افضل أداء في حالة إضافة الحشوة اسفل المبادل الحراري .لوحظ إن الكفاءة الحرارية عند إضافة الحشوة اسفل واعلى المبادل الحراري بأفضية تصل إلى نسبة (40%) و(25%) مقارنة مع برج التبريد المغلق بدون إضافة الحشوة على التوالي . زودت هذه الدراسة عدد من العلاقات التجريبية للتنبؤ بمعاملي انتقال الحرارة والكتلة بدلالة للمتغيرات التشغيلية للترتيب المتتالي والمتعاقب لأنابيب المبادل الحراري .

الكلمات الرئيسية: برج تبريد مغلق رطب ,مبادل حراري, حشوة ,أداء حراري

1. INTRODUCTION

Cooling towers are heat reject equipment used in much industrial process such as power generation units, refrigeration and air conditioning industries, **Stabat, 2004**. With respect to design of heat exchanger surface, there are two types of cooling towers: open and closed cooling towers. A direct, open circuit cooling tower is an enclosed structure that distributes warm water over a labyrinth-like packing, or fill, which provides an expansion of the interface between the air and water for heating of the air and evaporation to take place. An indirect, or closed circuit cooling tower, does not involve any contact between the air and the fluid being cooled. This tower has two separate fluid circuits, one in which circulated liquid on the outer surface of the second circuit, which is a bundle of tubes (closed coils) through which the hot water is flowing.

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, and Facao, 2000**, designed a new CWCT in order to examine the effects of operating parameters on the saturation efficiency for a CWCT modified for use with chilled ceilings in buildings. Thermal performance of two evaporative cooled heat exchangers, Investigated by **Hasan, and Sirén, 2003**. 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. **Shim, et al., 2008 and 2010** investigated experimentally the thermal performance of two heat exchangers in closed-wet cooling tower having a rated capacity of 2RT. 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. **Heyns, and Kroger, 2010** investigated the thermal performance characteristics of an evaporative cooler, which consist of 15 tube rows with 38.1 mm outer diameter galvanized steel tubes arranged in a triangular pattern of 76.2 mm. **Al-Tayyar, 2011,** 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. **Zheng, et al., 2012,** investigated thermal performance of an oval tube CWCT based on heat and mass transfer under different operating conditions.

Most of above studies involved thermal performance of CWCT with heat exchanger either inline or staggered tubes arrangement. In this study, thermal performance of modified counter flow forced draft CWCT in hot and arid environmental Iraqi weather conditions will be evaluated in order to use for the chilled ceiling of the building in Iraq. The modification based on addition packing to the conventional CWCT. To clarify the effect of heat exchanger tubes arrangements on tower performance, two heat exchangers with different tubes arrangement (inline and staggered) are used in this study.

2. EXPERIMENTAL APPARATUAS AND PRCEDURE

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 photographically in **Fig.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 1.5mm 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 zones. Spray zone is at 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 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 height of 450 mm in the case of three boxes and it will be variable when lifting one ore tow 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 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 water flow meter to the heat exchanger placed inside 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 moves over the tubes in perpendicular direction. The tubes are arrays in inline and staggered arrangement with (equilateral) tube pitch of $3D_o$ (pitch over diameter of 3) as shown in **Fig.2**.The specification of heat exchangers shows 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 with different types of spray nozzles. The spray water passes through the spray nozzles and constantly distributed at 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)˚C and inlet air wet bulb temperature of (7-24)˚C. Conformational parameters indicate: height of packing used (280 and 560) mm, location of packing (under Heat exchanger and above Heat exchanger) and arrangement of heat exchanger (staggered and inline).

Thermocouples type K insert 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:

$$CR = T_{cw,in} - T_{cw,out} \tag{1}$$

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 **Oliveira, and Facao, 2000 and Yingghan,et al., 2011**:

$$\eta = \frac{T_{cw,in} - T_{cw,out}}{T_{cw,in} - T_{awb,in}} \tag{2}$$

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

$$q = \dot{m}_{cw} C_{p,cw}\, CR \tag{3}$$

4-Mass transfer coefficient

The mass transfer coefficient obtained using enthalpy balance for an elementary transfer surface **Oliveira, and Facao, 2000**.

$$m_{air} dh_a = \alpha_m \, (i_i - i_{air}) \; dA \tag{4}$$

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

$$\frac{\alpha_m A}{\dot{m}_a} = ln \frac{i_{masw} - i_{a,in}}{i_{masw} - i_{a,out}} \tag{5}$$

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 i_{masw} 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 **Zheng, et al., 2012** 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 **Oliveira, and Facao, 2008:**

$$\dot{m}_a \left(i_{a,out} - i_{a,in} \right) = \dot{m}_{cw} C_{p,cw} \, (T_{cw,in} - T_{cw,out} \;) \tag{6}$$

Then the outlet air enthalpy calculates as:

$$i_{a,out} = i_{a,in} + \frac{\dot{m}_{cw} C_{p,cw} \, (T_{cw,in} - T_{cw,out} \;)}{\dot{m}_a} \tag{7}$$

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 **Hasan, and Sirén, 2002**:

$$dq_c = \dot{m}_{cw} C_{\mathrm{p},cw} dT_{cw} = -U_o \, (T_{cw} - T_{sw}) dA \tag{8}$$

Integrated Eq.8 from the inlet to outlet of cooling water, with constant spray water T_{sw}, gives.

$$\frac{U_o A_c}{C_{p,cw} m_{cw}} = ln \frac{T_{cw,in} - T_{sw,m}}{T_{cw,out} - T_{sw,m}} \tag{9}$$

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 **Shim, et al., 2008 :**

$$\mathrm{U_o} = \left[\frac{R_o}{\mathrm{R}_i} \, \frac{1}{\alpha_c} + \frac{R_o}{\mathrm{k_t}} \ln \frac{R_o}{R_i} + \frac{1}{\alpha_s} \right]^{-1} \tag{10}$$

After the overall heat transfer coefficient was calculated from Eq.(9), it used to calculate, αs, tube to water film heat transfer coefficient (W/m^2 C).

$$\alpha_s = \left[\frac{1}{U_o} - \frac{R_o}{\mathrm{R}_i}\frac{1}{\alpha_c} - \frac{R_o}{\mathrm{k_{tube}}}\ln\frac{R_o}{R_i}\right]^{-1} \tag{11}$$

Where, α_c is the convection heat transfer coefficient of cooling water inside the tubes, it was calculated by the "**Dittuse-Boelter**" relation and **Incropera,et al.,2011**:

$$\alpha_c = 0.023\ \frac{k_{cw}}{D_i} Re^{0.8}\ Pr^{0.3} \tag{12}$$

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 DESCUSSION

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. (8).As shown in **Fig.3**, 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. Effects of Operational Parameters

Figs. 4 to 6 indicate the effects of air flow rate, spray water flow rate, cooling water flow rate, inlet cooling water temperature and inlet AWBT (due to an atmospheric conditions) on the cooling water range for CWCT with (560 mm) packing height that located under heat exchanger with staggered tubes arrangement.

The effect of spray water flow rate on the cooling water rang for different values of the air flow rate is illustrated in **Fig.4**. For each value of spray flow rate , as the air flow rate increases , the cooling water range is increases. This can be explained by as the air flow rate increases, evaporated water per unit of air increases too. On the other hand, cooling water range is increasing exponentially while the spray water flow rate is increasing. The most important reason for increasing cooling rang with spray water flow rate is 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 relationship between the cooling water range and cooling water flow rate with different spray water flow rates are illustrated in **Fig.5**. It can be noted that the cooling water range is inversely proportional to the cooling water flow rate when both air and spray water flow rates are constant. For constant heat load, at low flow rate of circulation cooling water inside the heat exchanger tube, the opportunity to be the largest in completion of heat exchange with spray water and air through the tube surface within the tower test section for same air flow rate caused an increasing in temperature difference of cooling water. If maximum cooling range is desired, the low flow rate of cooling water should be used. For the same manner in **Fig.4**, cooling water range for 40 l/min spray water flow rate approximately 14% higher than that 30 l/min.

The variation of cooling water range with inlet cooling water temperature for different values of spray water flow rate is illustrated in **Fig.6**. For each value of inlet cooling water temperature, as the spray water flow rate increased, the cooling water range is increased**.** It is apparent. that cooling water range increases .Which is due to the increasing in the air hold up as a result of a decreased viscosity of spray water that was caused by increased an inlet spray water temperature at the first stage of the tower. Therefore, at a higher inlet spray water temperature, the vapour pressure driving force is increased by operating cooling tower at a given inlet air condition, this conforms well to **Shim, et al., 2008.** What was observed from this figure is that the decrement of the cooling range at high inlet cooling water temperature is increased because of the increased in rate of heat and mass transfer.

Cooling range with respect to variable inlet AWBT for different inlet cooling water temperature has been shown in **Fig.7**.For each inlet cooling water temperature, cooling range decreases almost linearly with the increase of inlet AWBT and vice versa. This is because when the inlet AWBT increases, the amount of heat exchange between air and water by convection and evaporation decreases due to a decrease in temperature difference between the inlet air and cooling water temperatures. Also, it can be known for the same reason that when inlet cooling water increases for the same inlet AWBT, the cooling range increases.

3.3. Effect of Added Packing

The effect of added packing to CWCT on the cooling range is shown in **Fig.8**. Results clearly demonstrated that the water cooling range increases with an increase in packing height. This can be attributed to the decrease of spray water temperature due to the increasing in the mass transfer generated by intense adding packing that substantially increases the air-water contact area and the water resident time in the tower. It can be observed that the cooling range for added packing height of (280 mm) and (560 mm) approximately (6%) and (28%) higher than that conventional CWCT respectively. As can be seen from **Fig.9**, 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) and (560 mm) approximately (6%) and (28%) higher than that conventional CWCT respectively.

3.4. Effect of Packing Location

From **Fig.10** it is observed that added packing to CWCT displays a higher cooling range when located the heat exchanger at the top of the tower than the lower location. It is believed due to the rate of evaporation will be at a maximum value at the top of the tower; therefore, a maximum rate of mass transfer will be found at this stage. The rate of mass transfer is decreased gradually from the top of tower to the bottom. If heat exchanger located at the top of the tower, spray water in the form of small droplets easily evaporate at the surface of heat exchanger .Whereas, if the heat exchanger located at the bottom of the tower, spray water outlet from packing will be big drops that may not cover all surface of heat exchanger. On the other hand, contact between warm spray water and cold air gives better heat and mass transfer for packing when located at the bottom It may be observed that the cooling range for CWCT with packing under heat exchanger and CWCT with packing above heat exchanger approximately (28%) and (16%) higher than that CWCT without packing respectively. **Fig.11** shows the cooling capacity comparing for different positions of packing. The result indicated that the cooling capacity for CWCT with packing under heat exchanger and CWCT with packing above heat exchanger approximately (28%) and (16%) higher than that CWCT without packing respectively.

3.5. Effect of Heat Exchanger Tubes Arrangement

To clarify the effect of heat exchanger tubes arrangements on tower performance, the performance analysis has been illustrated for CWCT with packing of 560 mm height located under heat exchanger for different types of tubes arrangement. **Figs .12 to 15** represent the comparison between cooling range, thermal efficiency and cooling capacity as a function of spray water flow rate for different tubes arrangement: staggered and inline.

Fig.12 presents the variation of cooling range with spray water flow rate for different air flow rate and different tubes arrangements. From this figure, in both arrangements, it can be observed that the cooling range increases with an increase in both air and spray water flow rates. However, the inline tubes arrangement had a higher cooling range than the staggered tubes arrangement. This could be due to air velocity through the minimum flow area; this area depends on the geometric of tube arrangement. Higher number of tubes per row in inline arrangement decreases minimum flow area normal to the flow then increase in air velocity. On the other hand, in spite of raised in the wakes of upstream tubes beyond the first row in inline arrangement, that causes a decrease in heat and mass transfer, the inline arrangement performs much better in case of widely space tube. The inline arrangement increasing cooling range more than (7.5%) compared to the staggered arrangement.

Fig .13 illustrates the variation of thermal efficiency with spray water flow rate for different air flow rates and different heat exchanger tubes arrangements. Results show that in both arrangements, it is indicated that the thermal efficiency increases with an increase in both air and spray water flow rates. However, the inline tubes arrangement had a higher thermal efficiency than the staggered tubes arrangement. The inline arrangement enhanced thermal efficiency more than (7%) compared to the staggered arrangement.

Variation of cooling capacity with spray water flow rate for different air flow rates and different heat exchanger tubes arrangements has been shown in **Fig .14**. It is stated that in both arrangements, it is indicated that the cooling capacity increases with an increase in both air and spray water flow rates. However, the inline tubes arrangement had a higher cooling capacity than the staggered tubes arrangement. The inline arrangement increases cooling capacity more than (5%) compared to the staggered arrangement.

Another e comparison between heat exchanger tubes arrangements are shown in **Fig.15**. In **Fig.15a**, the variation of cooling capacity with spray water flow rate for inline and staggered tubes arrangement are presented. As mentioned in **Fig.14**, the cooling capacity for inline tubes arrangement greater than the staggered tubes arrangement. On the other hand, the variation of cooling capacity per unit (heat exchanger) volume with spray water flow rate for inline and staggered tubes arrangement are presented in **Fig.15b**.Two heat exchangers do not have the same volumetric size, so it is expressed that cooling capacity which was divided by the volume of heat exchanger in this figure. Cooling capacity per unit volume for using staggered tubes arrangement has highest value than the inline tubes arrangement. This is because the staggered tubes arrangement has more compact comparing to the inline arrangement for similar tube dimensions. Generally, it is indicated that the inline arrangement gives higher cooling capacity in kW, whereas the staggered perform better for cooling capacity in kW/m^3.

Fig.16 shows a comparison of the impact of inlet cooling water temperature on the thermal efficiency of CWCT with packing under heat exchanger between the present work and test results of **Al-Tayyar, 2011**. Al-Tayyar studied the outline of a heat exchanger in a CWCT of 1 kW cooling capacity and an inline tubes arrangement of 8 mm outside tube diameter arranged in 6 rows and 12 columns .The thermal efficiency was increased with the increasing inlet cooling water temperature. The qualitative agreement in results between two studies is observed. The difference in the thermal efficiency between two studies is due to the distinction between the two experimental apparatus.

3.6. Correlations of Heat and Mass Transfer Coefficients

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 for both inline and staggered arrangements. These correlations are:

1-Inline tubes arrangement

a-Mass transfer coefficient

$$\alpha_m = 0.00003\left(\dot{G}_{air}\right)^{0.4925}\left(\dot{G}_{sw}\right)^{0.527}(T_{cw})^{1.839} \tag{13}$$

b-Heat transfer coefficient

$$\alpha_s = 0.1872\left(\dot{G}_{sw}\right)^{0.6406}\left(\dot{G}_{cw}\right)^{0.4}(T_{cw})^{1.5089} \tag{14}$$

1-Staggared tubes arrangement

a-Mass transfer coefficient

$$\alpha_m = 0.000001\left(\dot{G}_{air}\right)^{0.5038}\left(\dot{G}_{sw}\right)^{0.7456}(T_{cw})^{2.4478} \tag{15}$$

b-Heat transfer coefficient

$$\alpha_s = 0.1349\left(\dot{G}_{sw}\right)^{0.3758}\left(\dot{G}_{cw}\right)^{0.2051}(T_{cw})^{1.7749} \tag{16}$$

The average roots square mean error between correlations and experimental data for mass and heat transfer was (0.9702), (0.9722) for inline tubes arrangement and (0.9666), (0.9424) for staggered tubes arrangement, respectively.

4. CONCLUSION

The inline tubes arrangement has a better thermal performance (higher cooling rang, thermal efficiency, cooling capacity heat and mass transfer coefficients and lower tower approach) than the staggered tubes arrangement. On the other hand, the inline arrangement gives higher cooling capacity in (kW), whereas the staggered perform better for cooling capacity per unit volume (kW/m$^{3)}$. 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. I t is found that the cooling capacity for added packing height of (280 mm) and (560 mm) approximately (6%) and (28%) higher than that CWCT respectively. Comparing CWCT with packing for both locations under and above heat exchanger, it has been observed that the best performance for the CWCT with packing under heat exchanger. Also, the result indicated that the cooling capacity for CWCT with packing under heat exchanger and CWCT with packing above 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
D=tube diameter, m
CR=cooling range, ^oC
G=mass flux , kg/m^2.s
i=specific enthalpy, kJ/kg
k=thermal conductivity, W/m ^oC
$\dot{m}$=mass flow rate, kg/s
q=cooling capacity, W
Q=volume flow rate, l/min
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 ^{o}C
α_c =heat transfer coefficient for water inside the tubes, W/m^2 ^{o}C
η = thermal efficiency,(%)

SUBSCRIPTS

a=air
cw=cooling water
in=inlet
m=mean
out=outlet
sw=spray water
t=tube

Table 1. Physical dimension of heat exchangers.

Heat exchanger configuration	value		Unit
	Inline	staggered	
Length	680	690	mm
Height	190	166	mm
Width	381	381	mm
Tubes for coil	32	30	-
Vertical tube spacing	47.64	24	mm
Horizontal tube spacing	47.64	80	mm
Tube per row	8	5	-
Outside tube diameter	15.88	15.88	mm
Tube thickness	0.81	0.81	mm
Total eat transfer area	1085573.57	1032691.77	mm^2
Minimum free flow area	175310	209148	mm^2

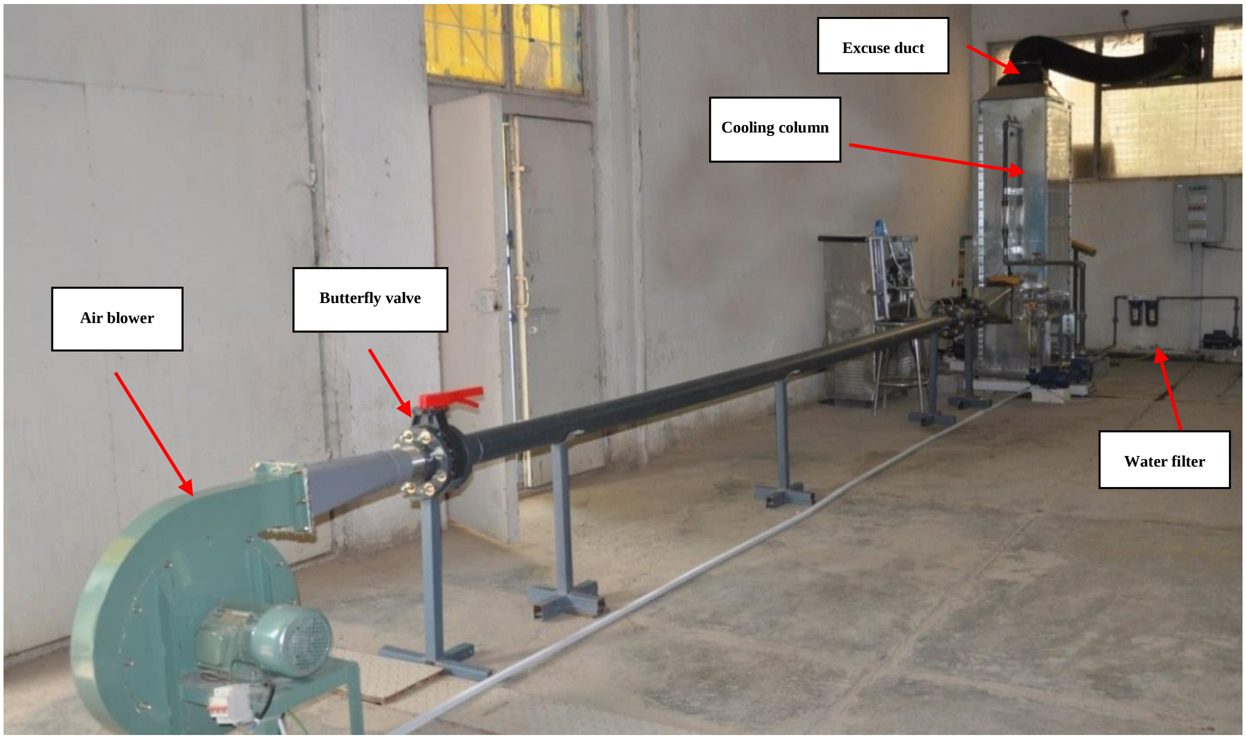


Figure 1a. Photographic picture for experimental apparatus (lateral view).

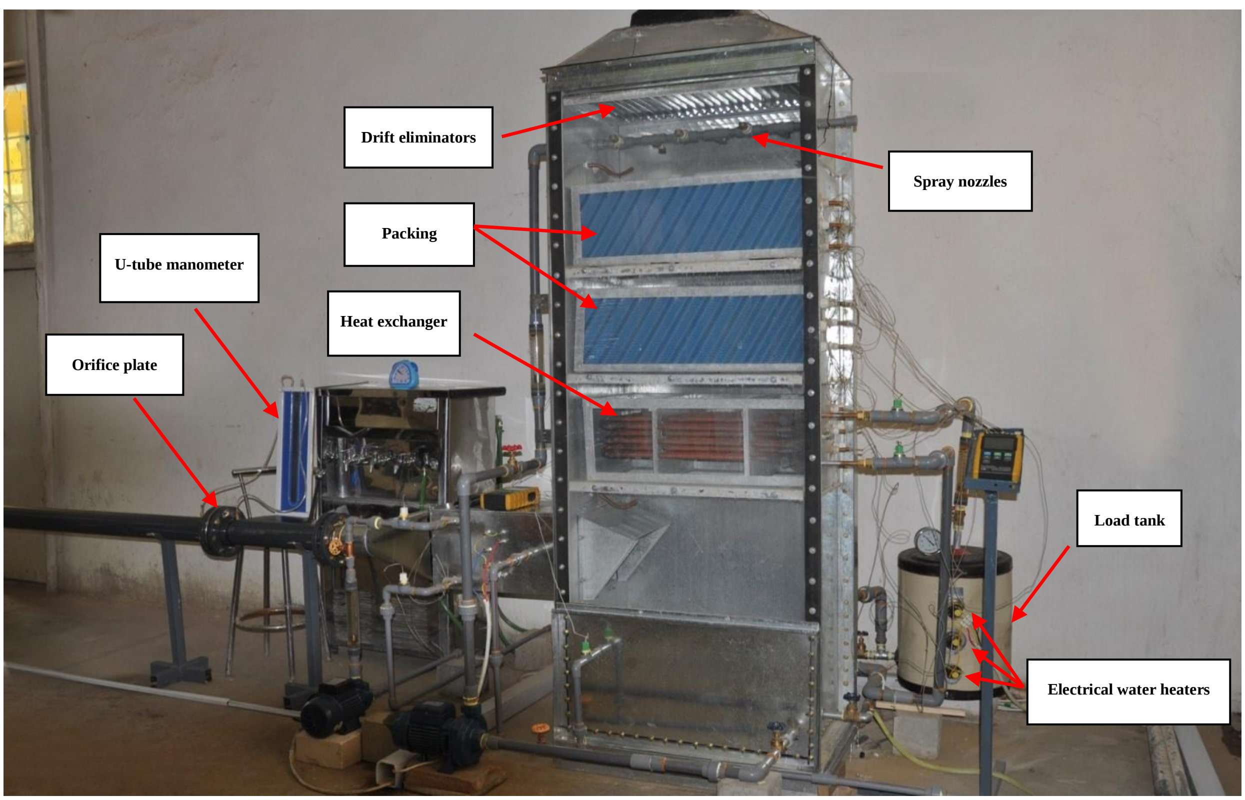
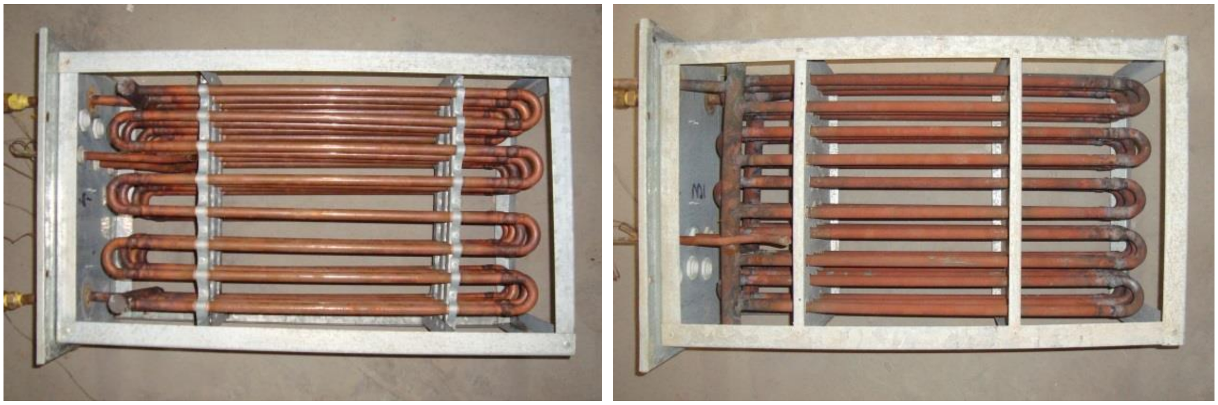


Figure 1a. Photographic picture for experimental apparatus (lateral view).



(a) (b)

Figure 2. Arrangement of tubes ;(a) inline arrangement, (b) staggered arrangement.

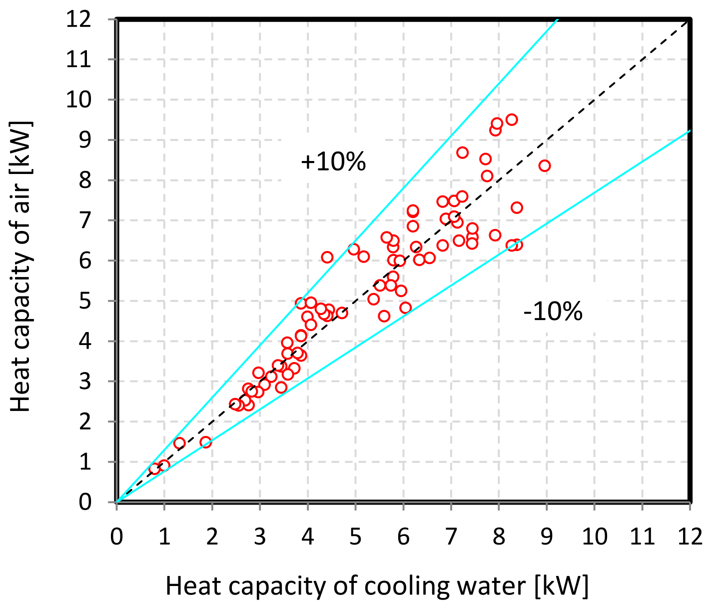


Figure 3. Energy balance of the experimental apparatus.

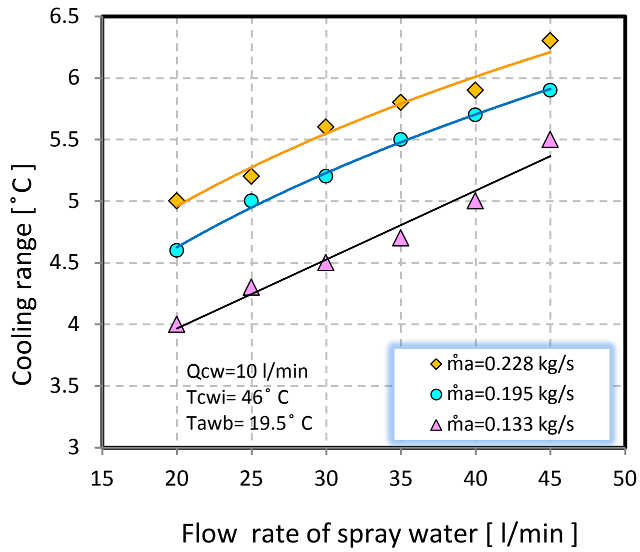


Figure 4. Variation of cooling range with spray water flow rate for different air flow rates.

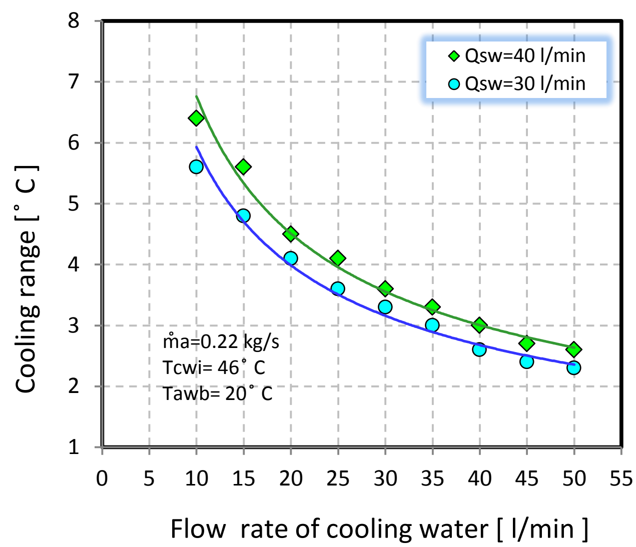


Figure 5. Variation of cooling range with cooling water flow rate for different spray water flow rates.

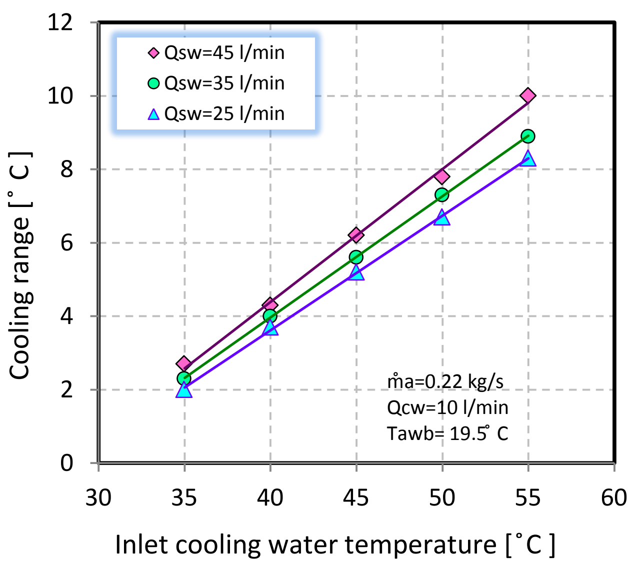


Figure 6. Variation of cooling range with inlet cooling water temperature for different spray water flow rates.

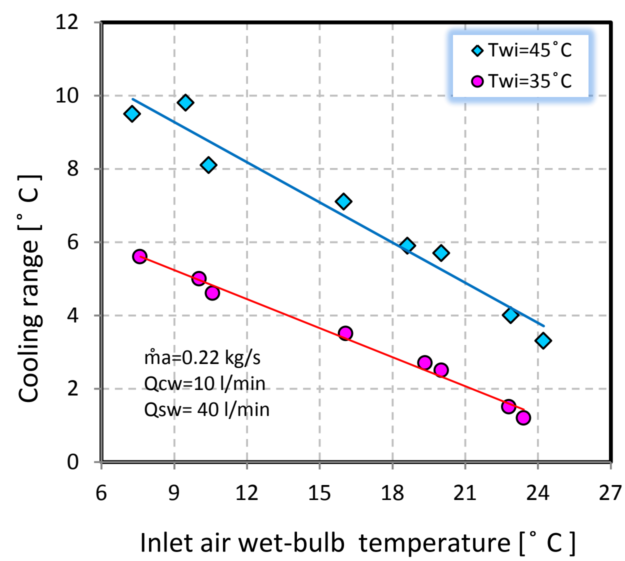


Figure 7. Variation of cooling range with inlet AWBT for different inlet water temperatures.

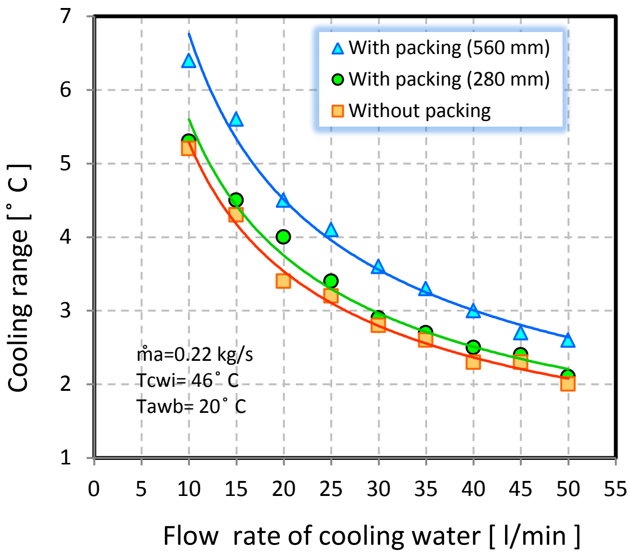


Figure 8. Variation of cooling range with cooling water flow rate for different heights of packing.

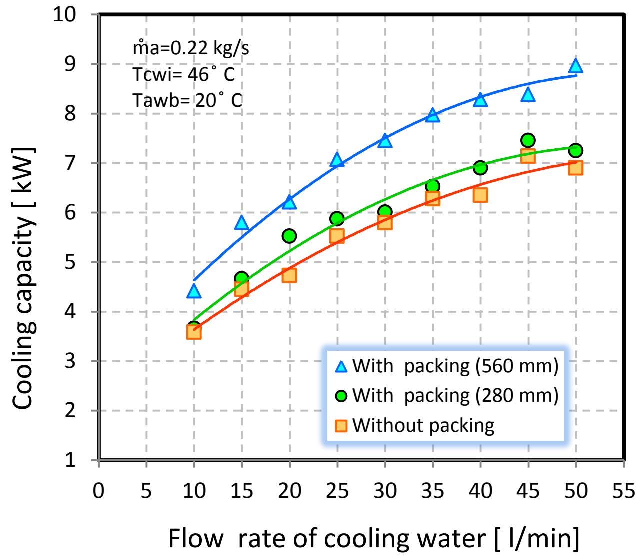


Figure 9. Variation of cooling capacity with cooling water flow rate for different heights of packing.

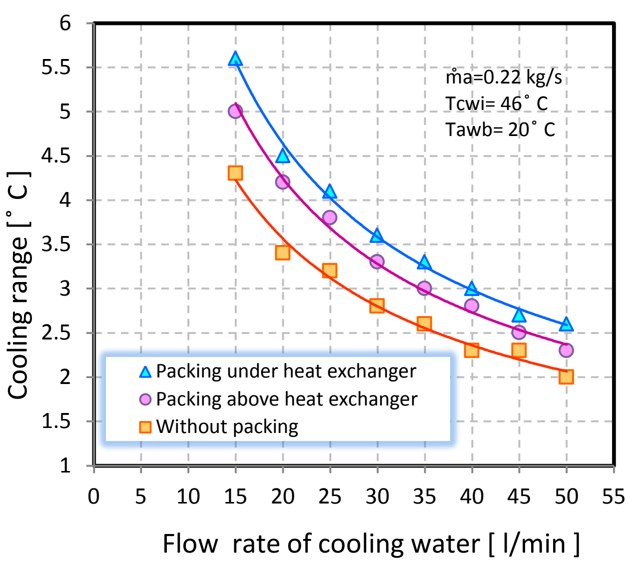


Figure 10. Variation of cooling range with cooling water flow rate for different locations of packing.

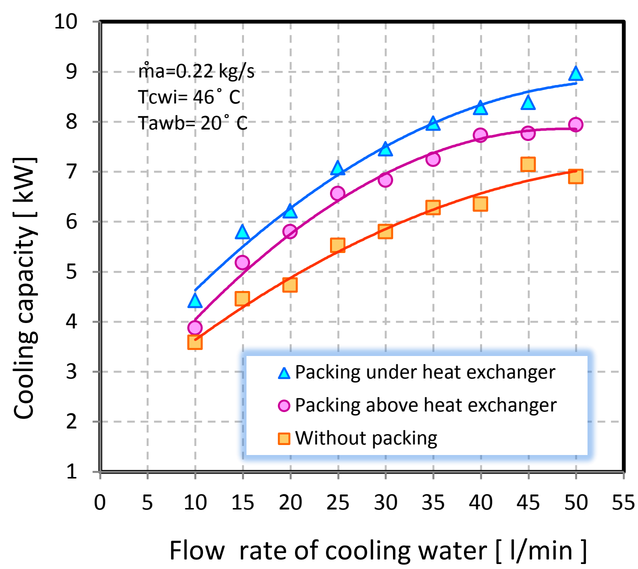
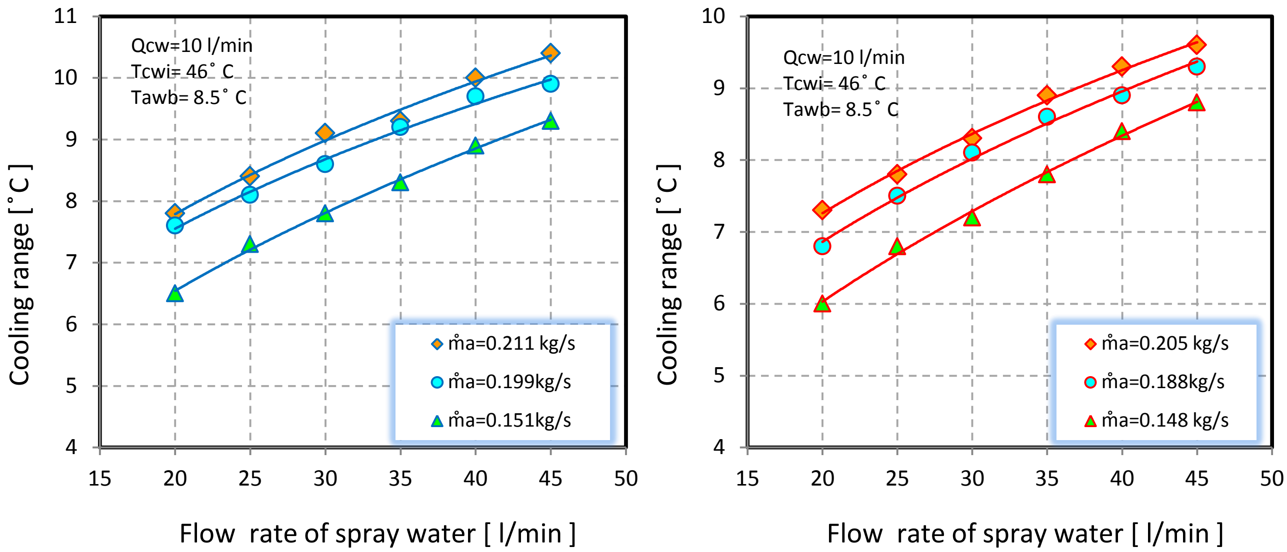
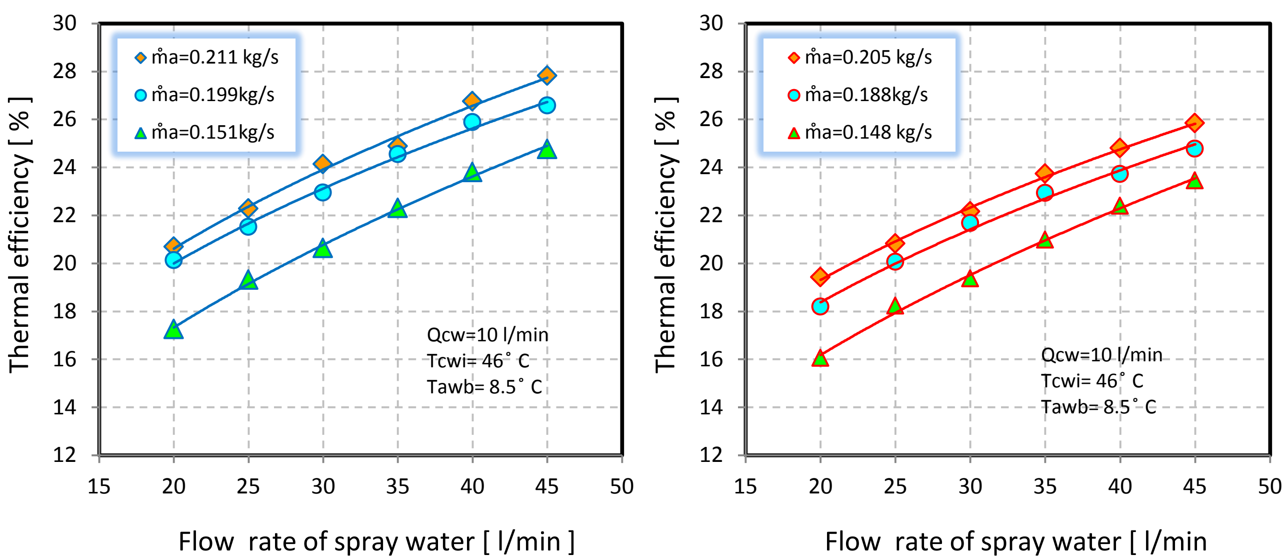


Figure 11. Variation of cooling capacity with cooling water flow rate for different locations of packing.



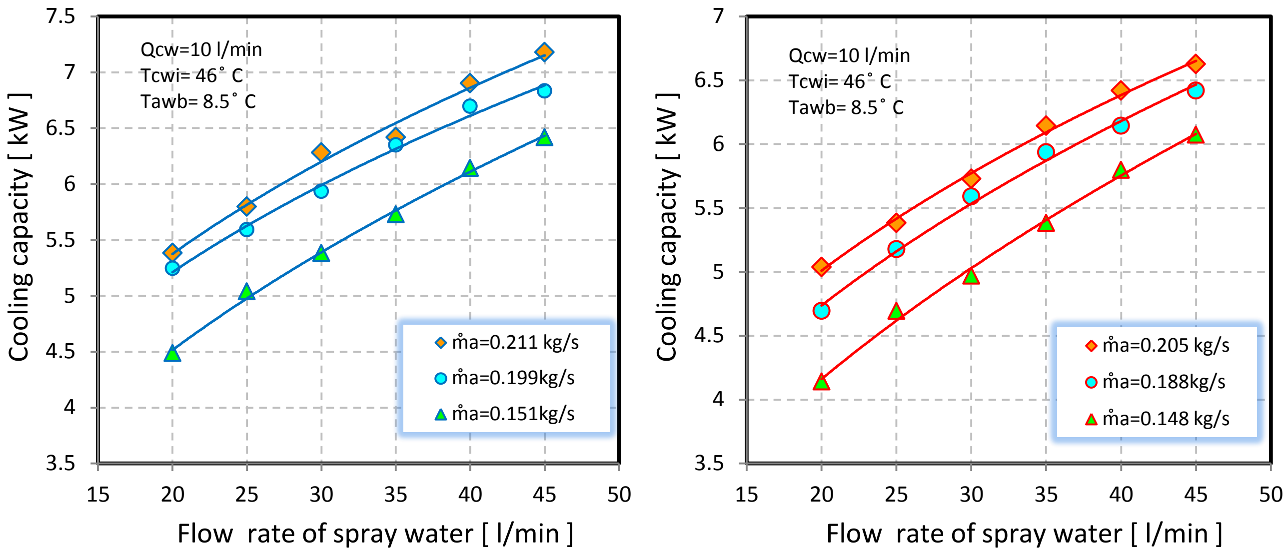
(a) (b)

Figure 12. Variation of cooling range with spray water flow rate for different air flow rates: (a) Inline arrangement, (b) Staggered arrangement .



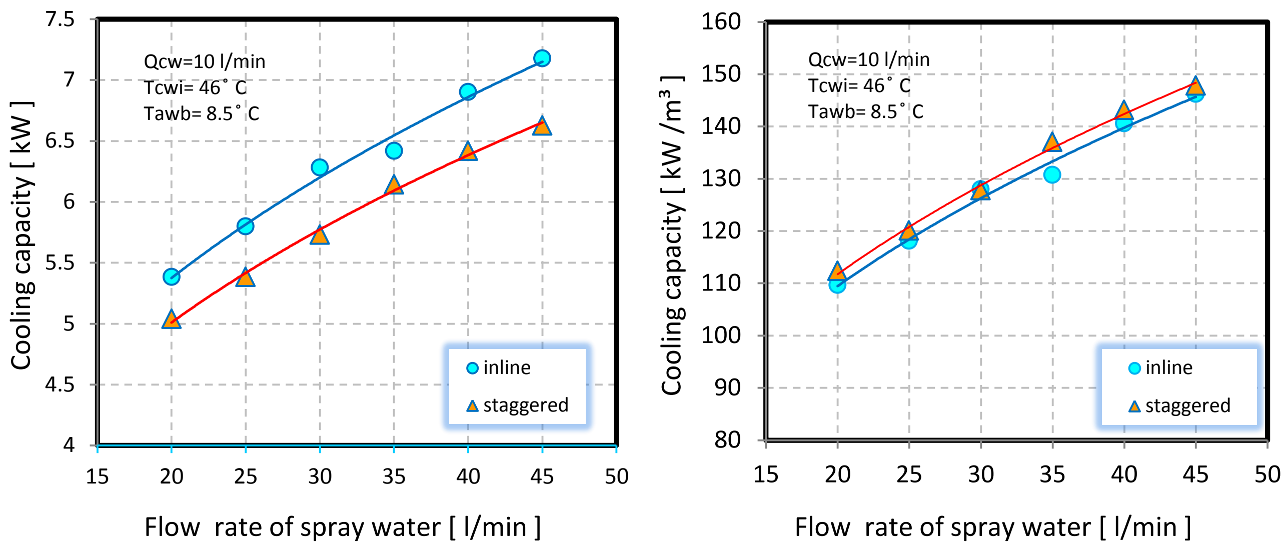
(a) (b)

Figure 13. Variation of thermal efficiency with spray water flow rate for different air flow rates: (a) Inline arrangement, (b) Staggered arrangement.



(a) (b)

Figure 14. Variation of cooling capacity with spray water flow rate for different air flow rates: (a) Inline arrangement, (b) Staggered arrangement.



(a) (b)

Figure 15. Variation of cooling capacity with spray water flow rate for different tubes arrangements: (a) Cooling capacity in kW, (b) Cooling capacity per unit volume kW/m^3.

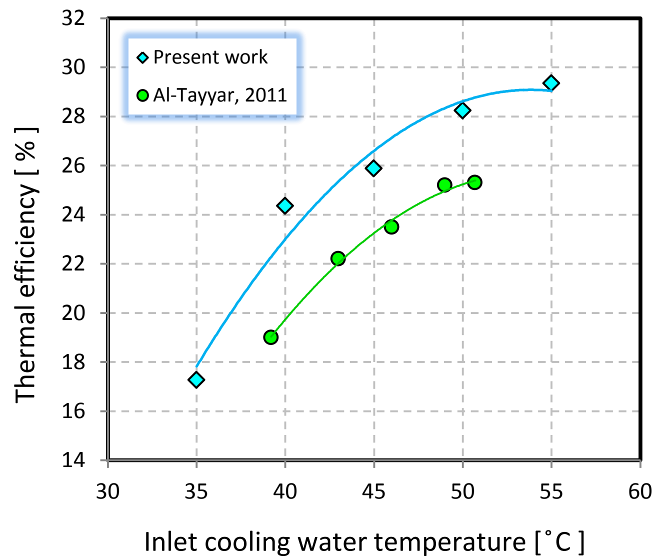


Figure 16. Comparison of the concluded effect of inlet cooling water temperature on thermal efficiency with other works.