An Investigation into Thermal Performance of Closed Wet Cooling Tower for Use with Chilled Ceilings in Buildings

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ABSTRACT

Chilled ceilings systems offer potential for overall capital savings. The main aim of the present research is to investigate the thermal performance of the indirect contact closed circuit cooling tower, ICCCCT used with chilled ceiling, to gain a deeper knowledge in this important field of engineering which has been traditionally used in various industrial & HVAC systems. To achieve this study, experimental work were implemented for the ICCCCT use with chilled ceiling. In this study the thermal performances of closed wet cooling tower use with chilled ceiling is experimentally and theoretically investigated. Different experimental tests were conducted by varying the controlling parameters to investigate their effects on the ICCCCT characteristics such as tower cooling capacity, chilled ceiling cooling capacity, tower saturation efficiency, mass transfer coefficient and heat transfer coefficient. The following controlling parameters are varied during experiments: spray water flow rate (90 to 150 kg/hr), ambient air wet bulb temperature (12 to 18 ^{o}C), and also changing chilled ceiling flow rate (2 to 6 l/min).

KEY WORDS: Cooling tower, chilled ceiling, Droplet trajectory, Two-phase flow, Computational fluid dynamics

دراسة الأداء الحراري لبرج تبريد من النوع المغلق ذي الإتصال غير المباشر مع السقوف المبردة

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الخلاصة

توفر انظمة السقوف المبردة الطاقة بكفاءة كبيرة.إنَ الهدف الرئيسي من هذا البحث هو دراسة الأداء الحراري لبرج تبريد ذي النوع المغلق والإتصال غير المباشرالمستخدم مع انظمة السقوف المبردة, للحصول على معرفة اعمق في هذا المجال الهندسي المهم الذي يستخدم بكثرة في كثير من التطبيقات الصناعية وفي تكييف الهواء. لتحقيق هذه الدراسة، فلقد تم إجراء إختباراتٍ عملية لبرج التبريد المذكور.في هذه الدراسة قد تم التحقق من الأداء الحراري ابراج لتبريد الرطبة المغلقة المستخدمة مع السقوف المبردة نظريا وعمليا.تم إجراء عدد من الإختبارات العملية عن طريق تغيير العوامل الحاكمة في أداء برج التبريد لتحري تأثيراتها على خصائص البرج مثل السعة الحرارية للبرج،السعة الحرارية للسقف المبرد, معامل الأداء، كفاءة الإشباع، معامل إنتقال الكتلة، معامل إنتقال الحرارة. تم إجراء الإختبارات العملية بتغيير الظروف الآتية:معدل تدفق ماء الرش بِمَدى(90 إلى 150 كلغ/ساعة)، درجة حرارة البصلة الرطبة للهواء الخارجي بِمَدى(12 إلى 18 م^{o})، وكذلك تم تغيير معدل تدفق السقف المبرد(2 إلى 6 لتر/ثانية) من أجل تحري تأثير ذلك على هبوط ضغط الهواء.

1. INTRODUCTION

Chilled beam means a cooled element or cooling coil situated in, above or under a ceiling which cools convectively using natural or induced air flows. The cooling medium is usually water. Chilled beam systems are used mostly in the nonresidential buildings. These are commercial buildings, offices, hotels, banks, universities, schools and hospitals.

Inside a CWCT, three fluids flow: cooling water, spray water and air. The cooling water is the fluid to be cooled, which flows inside tubes arranged in rows inside the tower. Spray water, injected on the tubes surface, re-circulates in a closed circuit, which makes it an intermediate fluid in the heat transfer process. Heat carried away from the building by the cooling water transfers to the spray water through the tube walls. From the spray water, heat transfers to air by both sensible heat and latent heat. The latter makes the major contribution and is caused by the evaporation of a small amount of the spray water into the air stream. The use of the closed type of cooling towers, which is indirect contact equipment, permits high level of cleanliness of the piping resulting in effective internal heat transfer surface, reduced maintenance costs and longer operational life.

A system consisted of a CWCT and chilled ceilings when used for cooling of buildings will result in a low cost, CFC free and environmentally clean system. The initial and running costs of the system are low when compared with traditional vapor-compression cooling systems. The low energy consumption results in higher coefficient of performance and lower CO_2 emissions.

The cooling rate is affected by the heat transfer from the water inside tubes to the spray water and then to the air. Hence, the optimum performance of a cooling tower depends on the number and arrangement of tube coils and on the uniformity and effectiveness of the water distribution as well as the cooling capacity of ambient air and the distribution of air velocity through the coils. An unsatisfactory distribution of water over coils would lead to a reduced evaporative cooling surface and increased air flow and flow resistance in areas devoid of water flow. A non-uniform distribution of air would increase flow resistance in areas of high local velocities and decrease heat transfer in areas of low velocities. Simultaneous occurrence of non- uniform distributions of air and water would further degrade the performance of the tower.

Chilled beam systems are based on dry cooling principle. Therefore these systems are applied in the premises where humidity of air is controlled by means of dehumidification of primary air and limitation or control of infiltration through the enclosing structures. In other words the building should meet standards of air tightness. Condensation on surface of chilled beam exchanger should be prevented via control of water temperature.

Research dealing with the application of cooling towers as the sole free cooling source in air-water systems, i.e. systems comprising chilled ceilings or chilled beams for cooling, is however limited. The earliest found is by **,Niu and Kooi 1993. and ,Niu et al. 1995**. Their work consisted of measurements of chilled ceilings in a laboratory and development of a simulation program, ACCURACY, with the aim of studying the performance of chilled ceilings. In their work, they also studied the performance of chilled ceilings in connection with an evaporative cooling tower.

Koschenz 1995. presented an analytical model for CWCT to be used with chilled ceilings. He made two assumptions: first, constant spray water temperature along the tower and second that the spray water temperature was equal to the outlet cooling water temperature. However, spray water temperature is variable inside the tower and if, the heat loss from the spray water piping outside the CWCT casing is ignored, spray water temperature will be equal in the tower's inlet & outlet.

Almadari et al 1998. reported the test performed by BRE (Building Research Establishment, London UK) where an office room was equipped with a chilled ceiling connected to a cooling tower. At a sensible heat gain of 60 W/m^2, a maximum indoor air temperature of 25.2°C was registered.

Gan and Riffat 1999. used computational fluid dynamic CFD calculations to predict the thermal performance of a CWCT for chilled ceiling system. They used CFD package, FLUENT, to simulate the two-phase flow of gas and water droplets in a 2D simulation. The Eulerian approach was used for the gas phase flow using standard k-ε model to represent the effect of turbulence while the Lagrangian approach was used for the water droplet phase flow, with two-way coupling between the two phases with the assumption of uniformly distributed volumetric heat generation from the tubes of the heat exchanger. The gas phase equations were solved numerically using a control volume technique embodied in the SIMPLE algorithm for the coupling between the pressure and the velocity fields. The computational results showed an increase of cooling water temperature for the lower tube rows in the tower. This behavior was attributed to the assumption of uniform heat flux generation. In reality, heat flux is higher for the upper rows because of the higher inlet cooling water temperature. The CFD technique was validated by comparing the predicted cooling performance with measurement. The predicted maximum decrease in the coil temperature is also in good agreement with the measured result for the water temperature.

Sprecher and Tillenkamp 2003. published an investigation on a system with a closed evaporative cooling tower and a water circuit embedded in the concrete slab. They used the simulation program TRNSYS for investigating such a system. Sprecher and Tillenkamp processed dynamic simulations of a large shopping centre, 10 000 m^2, in Lucerne, Switzerland, with internal heat gain from people and lighting at 42 W/m^2. During the warmest weak of the year, the indoor air temperature was kept below 26°C.

Hasan et al. 2007. uses a simulation tool (IDA-ICE) to find out the best performance of a cooling tower combined with chilled ceiling, serving a four storey residential building. The highest yearly mean COP achieved was 8,3.

It can be concluded that there are few published data concerning the resulting indoor climate in buildings equipped with chilled ceilings connected to an evaporative cooling tower, especially where the cooling tower is the sole provider of chilled water. The available data indicate the possibility to achieve an indoor thermal. In this work, a new application of already established cooling techniques is investigated under Iraqi climate conditions. The application comprises the use of a cooling tower, as the sole provider of chilled water, in conjunction with a hydronic cooling system with chilled ceiling.

2. EXPERIMENTAL WORK

The system that used in the experimental tests is a (WL 320 Demo cooling tower, made by Gunt Company in Germany). It was an open circuit direct contact counter flow forced draft cooling tower. This cooling tower was modulated to be used as an indirect contact closed type cooling tower by adding several components such as a bare-tube heat exchanger & the cooling water circuit The heat exchanger was designed and then manufactured according to the procedure that presented by (Kern in 1978). It was consisted of 8 mm outside copper tube diameter with 6 rows and 12 columns in an inline arrangement. The experimental apparatus consists generally of the cooling column, cooling water circuit, spray water circuit, the air circuit& test room. A schematic diagram & a photograph of the experimental apparatus are shown in Figures (1) & (2), respectively.

Typical experiments were conducted with the following procedure:

1) Experiment of Changing Chilled Ceiling & Tower Flow Rate:

After all the measured parameters were stabilized at the desired level, the tower let to operate with 2 l/min chilled ceiling & tower flow rate. It was noticed that after about 30 minutes all the measured values were stabilized. At this moment all the measured values were collected and recorded and then chilled ceiling & tower flow rate was increased 1 l/min, for each test, and so on until chilled ceiling & tower flow rate reached 6 l/min recording the measured values at each test.

2) Experiment of Changing Spray Water Flow Rate:

After all the measured parameters were stabilized at the desired level, the tower let to operate with 90 l/hr spray water flow rate. It was noticed that after about 30 minutes all the measured values were stabilized. At this moment all the measured values were collected and recorded and then spray water flow was increased 15 l/hr, for each test, and so on until spray flow rate reached 150 l/hr recording the measured values at each test.

3) Experiments of changing The Inlet Air Wet Bulb Temperature:

After turning on the fan and pumps and heating the cooling water, the dry bulb temperature and the relative humidity of the inlet air were changed by the use of the humidifier and the dehumidifier. The wet bulb temperature was taken then from a computer program for psychometric chart according to the measured values of dry bulb temperature and relative humidity. After few minutes the air stabilized at the desired wet bulb temperature and then measurements were recorded. This experiment was started with 12 ^{o}C wet bulb temperature increasing it 2 ^{o}C at each time until it reached 18 ^{o}C.

3. Analysis of the Experimental Data

The cooling capacity of the cooling tower was calculated according to the following equation by **Hasan & Serin (2002)**:

$$Q_{C.T}=\dot{m}_{cw}C_{p,\ cw}dT_{cw} \qquad (1)$$

Where dT_{cw} is the cooling water range defined as:

$$dT_{cw}=T_{cw,in}-T_{cw,out} \qquad (2)$$

The coefficient of performance of the tower was calculated from the following equation by **Hasan & Serin (2002)**:

$$COP_{C.T}=\frac{Q_{C.T}}{W_{tot}} \qquad (3)$$

Where, W_{tot} is the total power consumption by the air fan and spray water pump defined as:

$$W_{tot}=W_f+W_{sp} \qquad (4)$$

Where, W_f is the fan power (W) while W_{sp} is the spray water pump power (W).

Another important variable in a cooling tower performance is the saturation efficiency which refers to how much the discharged air is saturated with water vapor. The saturation efficiency, ε, for the closed circuit cooling towers was calculated from the following equation by **Yeon Yoo et al. 2010.**

$$\varepsilon=\frac{T_{cw,in}-T_{cw,out}}{T_{cw,in}-WBT_{in}} \qquad (5)$$

The calculation of the experimental mass transfer coefficient of water vapor between spray water film and air, and the calculation of the experimental heat transfer coefficient between tube external surface and spray water film are important part of the presented work because it permits to comparison with the other works.

Olivera & Facao (2002) presented a procedure to determine the mass transfer coefficient after experimental measurements by using an enthalpy balance by the following equation:

$$G\left(h_{air,out}-h_{air,in}\right)=\alpha_m \, A \, LMhD \qquad (6)$$

where, α_m is the mass transfer coefficient for water vapor between spray water film and air (kg/m^2 s), A is the surface area of the heat exchanger equal to 0.226 m^2 , and LMhD: logarithmic mean enthalpy difference (kJ/kg) defined as:

$$LMhD=\frac{h_{air,out}-h_{air,in}}{\ln\frac{h_{sat,Ti}-h_{air,in}}{h_{sat,Ti}-h_{air,out}}} \qquad (7)$$

Where $h_{sat,Ti}$ is the specific enthalpy of the saturated air at the interface temperature (kJ/kg).

The average of spray water temperatures was taken as the interface temperature according to **Olivera & Facao (2002)** as well as **Stabat & Marchio (2004)** while the inlet and outlet air enthalpies were taken from the psychrometric chart according to the measured data.

The overall heat transfer coefficient, U_o, between water inside tubes and the interface based on the outer area of the tube. U_o is calculated as follows:

$$Q_{C.T}=\dot{m}_{cw}C_{p,cw}dT_{cw}=U_o \, A \, LMTD \qquad (8)$$

Where LMTD: is the logarithmic mean temperature difference (^{o}C) defined as:

$$LMTD=\frac{T_{cw,out}-T_{cw,in}}{\ln\frac{T_{cw,out}-T_{sp,ave}}{T_{wc,in}-T_{sp,ave}}} \qquad (9)$$

Where $T_{sp,ave}$ is the average spray water temperature (^{o}C) according to **Olivera & Facao (2002)** as well as **Stabat & Marchio 2004**.

After the overall heat transfer coefficient was calculated from the two equations above, it used to calculate spray heat transfer coefficient between the tubes external surface & spray water film:

$$\frac{1}{U_o}=\frac{1}{\alpha_w}\frac{D}{d}+\frac{D}{2k_{tube}}\ln\frac{D}{d}+\frac{1}{\alpha_s} \quad (10)$$

The coefficient of spray heat transfer which takes place between tubes external surface and spray water film was calculated as follows which was presented by **Olivera & Facao (2002)**.

$$\alpha_s=\left[\frac{1}{U_o}-\frac{1}{\alpha_w}\frac{D}{d}-\frac{D}{2k_{tube}}\ln\frac{D}{d}\right]^{-1} \quad (11)$$

Where α_w is the heat transfer coefficient for water inside the tubes (W/m^2 ^{o}C) and it was calculated according to **Stabat & Marchio 2004.** by the following equation:

$$\alpha_w=0.023\frac{k_{cw}}{d}Re^{0.8}Pr^{0.3} \quad (12)$$

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

The cooling capacity of the chilled ceiling was calculated according to the following equation:

$$Q_{ceiling}=\dot{m}_{ceiling}C_{p,\ cw}dT_{ceiling} \quad (13)$$

Where dT_{cw} is the cooling water range of the chilled ceiling defined as:

$$dT_{ceiling}=T_{ceiling,out}-T_{ceiling,in} \quad (14)$$

4. RESULT AND DISCUSSION

Figures (3) to (8) indicate the effect of inlet air wet bulb temperature and chilled ceiling flow rate on the cooling tower performance and test room represented by cooling capacity, coefficient of performance, saturation efficiency, spray heat transfer coefficient, mass transfer coefficient, and chilled ceiling cooling capacity respectively.

As it seen in **Fig. 3**, wet bulb temperature affects the tower cooling capacity greatly. This figure shows that as the wet bulb temperature increases, cooling capacity decreases significantly. The increasing in inlet air wet bulb temperature means increasing in both the dry bulb temperature & the humidity of the inlet air and this decreases the transfer of heat both by convection & evaporation. It is also decrease when coil flow rate decrease with constant spray water flow rate.

When air wet bulb temperature reduces the tower cooling capacity, it reduces the tower coefficient of performance by the same rate as for cooling capacity because the wet bulb temperature has no effects on the power consumption of the equipment, this clear in **Fig. 4** and it decreases when chilled ceiling flow rate decreases too because the tower coefficient of performance depended on cooling capacity.

The wet bulb temperature of air has also influence on the tower efficiency as shown in **Fig. 5**. The efficiency increases slightly with the wet bulb temperature. If the wet bulb temperature increases, the difference between the inlet cooling water temperature & the inlet wet bulb temperature in the denominator of Eq. (5) is decreased with a rate more than the rate of decreasing cooling water range so that the saturation efficiency is increased with increasing the wet bulb temperature. Also the efficiency is increased when coil flow rate decreases due to the increasing of the difference between the inlet cooling water temperature & outlet cooling water temperature.

When the wet bulb temperature increases, the spray heat transfer coefficient decreases, as expected, and as shown in **Fig. 6**. This is simply because of that the decreasing in the cooling capacity, as wet bulb increases, leads to decrease the overall heat transfer coefficient and thus the spray heat transfer coefficient decreases. It's also decrease with decreasing the chilled ceiling flow rate due to the decreasing in the cooling capacity, as wet bulb increases.

When the inlet air wet bulb temperature increases, mass transfer coefficient increases slightly as shown in **Fig. 7**. It is clear as the wet bulb temperature increases the difference between the outlet and inlet air enthalpy, Δh_{air}, is decreased with a rate less than that in the logarithmic mean enthalpy difference, LMhD, causing an increasing in mass transfer coefficient. Also it's decrease with decreasing the chilled ceiling flow rate due to the increase in cooling water range that is lead to decrease the difference between the outlet and inlet air enthalpy, Δh_{air},.

As it seen in **Fig. 3**, the wet bulb temperature has effect on the tower cooling capacity consequently on the chilled ceiling cooling capacity as it seen in **Fig. 8**. This figure shows that as the wet bulb temperature increases, cooling capacity of the chilled ceiling decreases significantly and also its decrease when chilled ceiling flow rate decrease.

Generally, air wet temperature has effect on the performance of cooling towers as it dictates the physical phenomena that take place inside it. It decreases the heat transfer; on the other hand it increases the efficiency of the tower & mass transfer.

Figures (9) to (13) indicate the effect of the spray water flow rate and chilled ceiling flow rate on the cooling tower performance represented by cooling capacity, coefficient of performance, saturation efficiency, spray heat transfer coefficient and mass transfer coefficient, respectively and **Fig. 14** indicates the effect of spray water flow rate and chilled ceiling flow rate on the test room performance represented by chilled ceiling cooling capacity.

Fig. 9 indicates that the cooling capacity of the cooling tower depends greatly on the spray water flow rate and it increases approximately linearly as the spray water flow rate increases. This is mainly because that when spray water flow rate increases, the rate of evaporation is augmented causing more heat transferred from the cooling water. For a spray water flow rate more than 135 kg/hr it is seen that the increasing rate in cooling capacity becomes rather less and this is partially attributable to the fact that this flow rate is sufficient to achieve completely wetting of the outer surface of the heat exchanger tubes. The evaporation of the water droplet at this flow rate can’t continue that smoothly. When the mass flow rate of the spray water increased, the heat transfer increased and thus the heat absorbed by the air both by convection and evaporation was increased causing in increasing the cooling capacity. This means that the air flow rate becomes sufficient to absorb moisture at the interface between the air and the spray water film by the evaporative cooling, but as the spray water flow is constant so the absorption of moisture will not still to increase in the same rate when the air flow rate becomes high. Also the cooling capacity of chilled ceiling increase when the chilled ceiling flow rate increase.

The variation of the coefficient of performance of the experimental apparatus with respect to the variation of spray water flow rate and constant chilled ceiling flow rate is shown in **Fig. 10**. It is clear that the coefficient of performance increases as spray water flow rate increases. When the flow rate of spray water increases both the cooling capacity & the power of the spray water pump, which are shown in table (1), increase but the increasing in the cooling capacity is much larger than that in pump power and this makes COP to increase with the increasing in spray water flow rate. The coefficient of performance for a cooling tower is the ratio of the cooling capacity to the total power consumption of the fan and the spray water pump and it is calculated according to Eq. (4). For example, when spray water mass flow rate increases, for the case with chilled ceiling flow rate equal to 6 l/min, from 90 to 150 kg/hr, then the cooling capacity increases with a ratio of 1.55 from 378 to 588 Watt while the spray water pump power increases with a ratio of 1.05 from 448 to 426 Watt and the fan power constant equal to 134 Watt because no changing in air fan power and this causing in increasing COP from 0.65 to 1.03 and it's also increase when chilled ceiling flow rate increase.

The effect of the spray water mass flow rate on tower efficiency can be seen in **Fig. 11**. An increase in spray flow rate increases cooling range and this increases the tower efficiency up to a certain level. Above a rate of about 135 kg/hr, an increase in spray flow rate does not significantly improve tower performance because the tube surface is almost completely wet as discussed in **Fig. 9** and this conforms well to the results of Riffat et al.

We studied the effects of the operating parameters on the thermal efficiency of the tower defined as Eq. (5) we observations were that: $T_{cw,in}$ had a very little influence on tower efficiency , increase of spray water flow rate resulted in an increase in tower efficiency and increased with the increase of air mass flow rate and decreased with the increase of coil mass flow rate. And it's also increase when chilled ceiling flow rate increase.

Fig. 12 shows spray water heat transfer coefficient, $\alpha_{s,}$ as a function of spray water flow rate. It indicates that the heat transfer coefficient increases with increasing spray water flow rate. This can be explained by Eq. (8). When spray water increases, the cooling capacity increases too leads to increase the overall heat transfer coefficient, U_o, and then α_s increases according to Eq. (10). From the other side, when spray water flow rate increases then cooling water range increases leading to increase the logarithmic mean temperature difference, LMTD in Eq. (8), although α_s increases and this is because that the increasing rate in LMTD is less than that of cooling capacity and this needs U_o as well as α_s to increase. For example, when spray water flow rate increases, for the case with chilled ceiling flow rate equal to 6l/min, from 90 to 150 kg/hr, then the cooling capacity increases from 378 to 588 Watt with a ratio of 1.55 while LMTD increases from 2.6 to 3.05 ^oC with a ratio of 1.17 and this makes both U_o & α_s increase.

Fig. 13 indicates that the mass transfer coefficient is influenced by spray water flow rate. As spray flow increases mass transfer coefficient increases too. This mainly because that the increasing in spray flow means there is a large amount of water droplet could be evaporated and transferred to the air stream.

When spray water flow rate reduces the tower cooling capacity, that lead to reduces the cooling capacity of the test room by the same rate as for cooling capacity because the tower cooling capacity effects on the test room cooling capacity, this clear in **Fig. 14**.

Generally, spray water flow rate affects ICCCCT characteristics and it leads to enhance the performance of the tower. The heat transfer coefficient improved better with spray flow.

3.COMPARISON BETWEEN THE PRESENT WORK AND OTHER WORKS

Some of the present experimental results, for the common case when the ICCCCT operates with chilled ceiling, are compared in this section with other previous experimental works. The comparison is a qualitative comparison and it is not a quantitive comparison, i.e. the agreement was attained in the behavior of the characteristics and the tendency of the curves but there is no agreement in the values with the previous works and this was shown in Figures. (15) to (19).

Fig. 15 shows a comparison between the present experimental results including cooling capacity with respect to inlet air wet bulb temperature for different cooling water flow rate (the right panel of figure) and the experimental results (the left panel of figure) presented by **Sarker 2007**. Sarker studied an investigation on the optimum design of a heat exchanger in a hybrid closed circuit cooling tower of 1RT rated capacity and staggered arrangement heat exchanger of the relevant design parameters were selected based on the typical East Asian meteorological constrains for the year-round smooth operation of the cooling tower. Sarker is evident from figure, the cooling capacity increases with increasing of the cooling water flow rate but decreases with the increase of WBT as well as by the decrease of the cooling water flow rate. The difference in the amount of cooling capacity between the two studies belongs to the difference in dimensions of the tower & the heat exchanger between the two experimental apparatus but both results having the same behavior.

Fig. 16 shows compares the presented experimental results of the range of cooling water with respect to variable inlet wet-bulb temperature (WBT) and results represented by **Sarker el at. 2009.** From this figure it is clear that the temperature drop of the cooling water decreases with increasing of inlet wet-bulb temperature. This is because when the inlet WBT increases, the temperature difference between inlet cooling water and air decreases. Therefore, the rate of evaporation of spray water flowing at the outer surface of the pipes decreases so that the falling of temperature of cooling water flowing inside the tubes deceases.

Fig. 17 compares the presented experimental results of the temperature drop and cooling capacity and presented by **Sarker el at. 2009.** with respect to spray water volume flow rate are shown at the left and right axes, respectively. As it is clear in this figure, that both tower's cooling capacity & temperature drop increase linearly as spray flow rate increases. This result coincides well to the experimental results of the presented work.

In **Fig. 18** the experimental results of the presented work for the influence of the difference of inlet and the outlet temperatures of the cooling water with respect to the ratio of cooling water to air mass flow rate (on the right panel of figure) are compared with that presented by **Sarker et al. 2005.** (on the left panel of figure). This figure shows a qualitative agreement between the presented work and that presented by **Sarker et al. (2005)**in the experimental behavior of cooling water range with respect to the ratio of cooling water to air mass flow rate shows the same behavior between results presented by **Sarker et al. (2005**) and present results.

Fig. 19 shows the experimental effect of spray water flow rate on both cooling capacity & saturation efficiency, (on the right panel of figure). It is compared with the results presented by **Yeon Yoo et al. 2010.**, (on the left panel of figure). **Yeon Yoo et al. 2010** xpressed, in the experimental behavior of both cooling capacity & saturation efficiency with respect to the spray water flow rate shows the same behavior between results presented by **Yeon Yoo et al. 2010**. and present results.

4. CONCLUSIONS

The following conclusions are valid only for the given parameters of the heat exchanger, configuration and the other operational conditions of the present work:

1. All the characteristics of the performance of the ICCCCT increases with respect to chilled ceiling flow rate, spray water flow rate, inlet air wet bulb temperature and provided

heat load but performance of chilled ceiling increases with respect to chilled ceiling flow rate, spray water flow rate, and decreases with respect to inlet air wet bulb temperature and provided heat load.

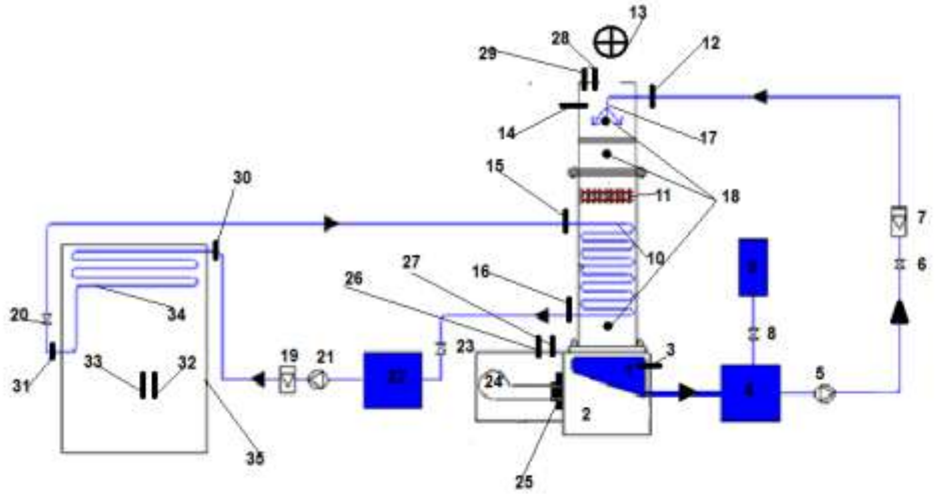
Tower cooling capacity increases with a ratio of 1.55 for the case with chilled ceiling flow rate equal to 6 l/min and spray water flow rate varying from 90 to 150 kg/hr, COP increase from 0.65 to 1.03, while LMTD increases from 2.6 to 3.05 ^{o}C with a ratio of 1.17 and this makes both U_o & α_s increase. But higher tower cooling capacity, higher coefficient of performance, higher saturation efficiency and higher chilled ceiling cooling capacity can be obtained with low gas to liquid flow ratio. For example, when G/L increases from 1.4179 to 2.3633, with constant chilled ceiling flow rate equal to 6l/min, the cooling capacity decreases from 588 to 378 Watt, coefficient of performance decreases from 1.0103 to 0.6494, the tower efficiency decreases from 11.3821 % to 9.375 % and chilled ceiling cooling capacity decreases from 462 to 252 Watt, respectively. This mostly because when G/L increases for constant mass air flow rate, then the spray water flow rate decreases and becomes insufficient to accomplish the same transfer of heat especially by evaporation and this causing reducing of the cooling water range leading to decrease these characteristic keeping G constant and varying L.

2. The tower coefficient of performance is influenced greatly by the cooling capacity and little by the total power consumption of air fan and spray water pump, it increases as the cooling capacity increases. From another perspective, the better COP can be obtained at about 6l/min chilled ceiling flow rate & 150 kg/hr spray flow rate as the increasing rate in cooling capacity above these values becomes less while the power consumption increases approximately linearly.

3. Spray flow rate has the greatest influence on spray heat transfer coefficient, α_s, but it is also a function of the air flow rate and the spray water temperature.

4. Both air and spray water flow rate affect the mass transfer coefficient, α_m, but the great effect belongs to the air flow rate.

5. Inlet air wet bulb temperature has a great influence on the all characteristics of the tower. Tower Cooling capacity, COP, chilled ceiling cooling capacity and spray heat transfer coefficient are decreased while saturation efficiency and mass transfer coefficient are increased with wet bulb temperature varying from 12 to 18 ^{o}C.



1	Drain	19	Cooling Water Flow Meter
2	Air Chamber	20	By-Pass Valve
3	Outlet Spray Water Temp. Sensor	21	Cooling Water Pump
4	Spray Water Tank	22	Cooling water Tank
5	Spray Water Pump	23	Controlling Valve
6	Controlling Valve	24	Fan
7	Spray Water Flow Sensor	25	Butterfly Valve (Damper)
8	Controlling Valve	26	Inlet Air Temp. Sensor
9	Make-Up Water Tank	27	Inlet Air Moisture Sensor
10	Heat Exchanger	28	Outlet Air Temp. Sensor
11	Fill Packings	29	Outlet Air Moisture Sensor
12	Inlet Spray Water Temp. Sensor	30	Inlet Cooling Water Temp. Sensor
13	Anemometer	31	Outlet Cooling Water Temp. Sensor
14	Combined Temp. /Moisture Sensor	32	Inlet Room Temp. Sensor
15	Inlet Cooling Water Temp. Sensor	33	inlet Room Moisture Sensor
16	Outlet Cooling Water Temp. Sensor	34	Chilled Ceiling
17	Spray Water Nozzle	35	Test Room
18	Holes for Sensing Pressure Drop		

Figure.1 system components.

Figure. 2 Photograph of the experimental apparatus.

Table.1 the measured electrical power consumptions of the spray water pump and air fan at a voltage of 220 volt.

Spray water pump power consumption

Spray water flow rate	kg/hr	90	105	120	135	150
The measured current	Amp.	1.936	1.97	1.97	2.01	2.03
The power consumption	Watt	426	433	437	442	448

Air fan power consumption

Air Velocity*	m/s	10.5
The measured current	Amp.	0.6
The power consumption	Watt	134

* The values of air velocity are measured at the outlet air section.

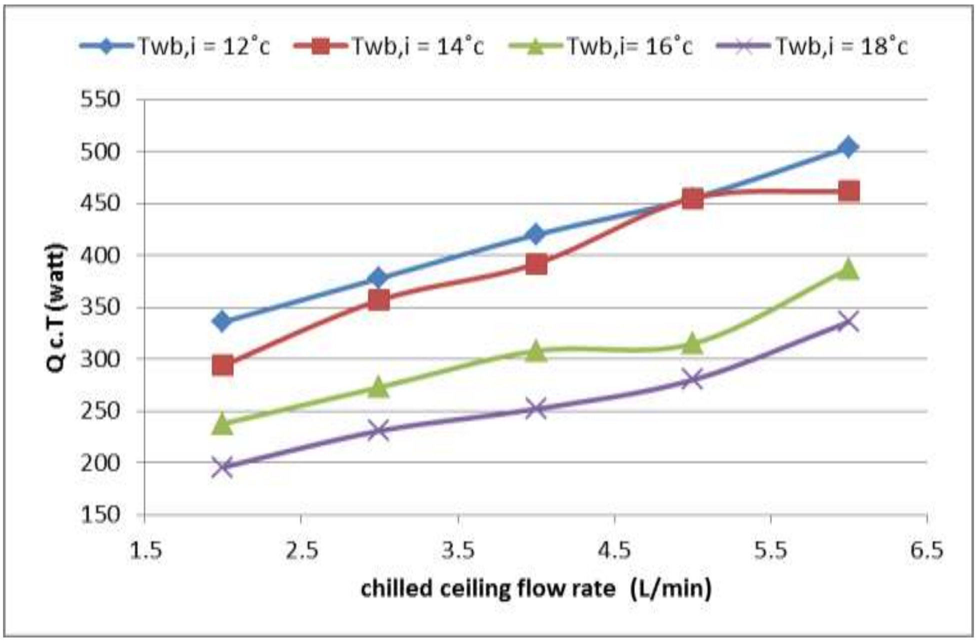


Figure. 3 Influence of the chilled ceiling flow rate on cooling capacity for different inlet air wet bulb temperature

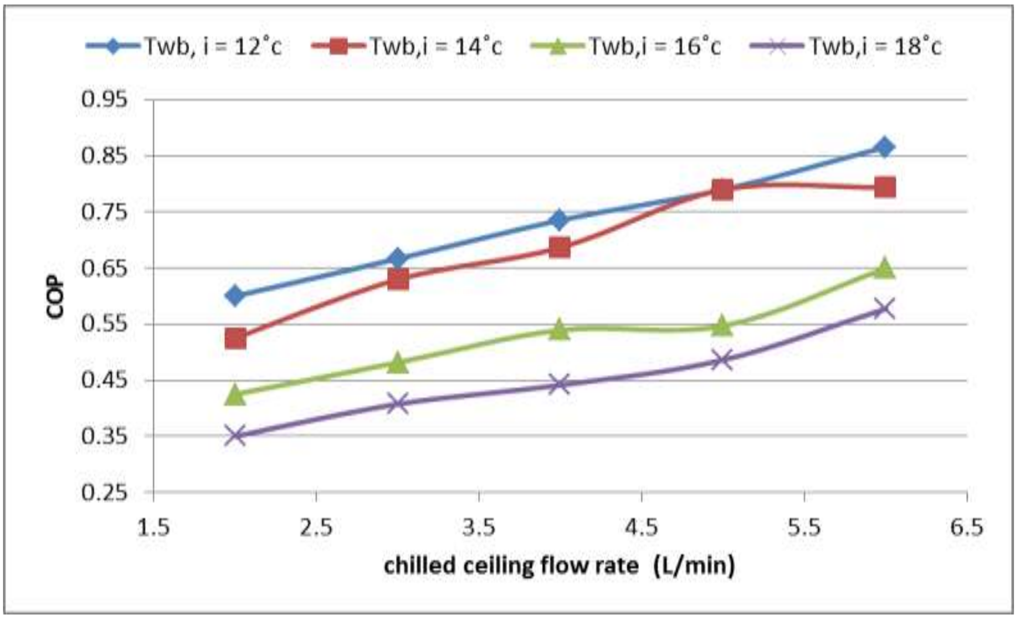


Figure. 4 Influence of the chilled ceiling flow rate on the coefficient of performance for different of the inlet air wet bulb temperature

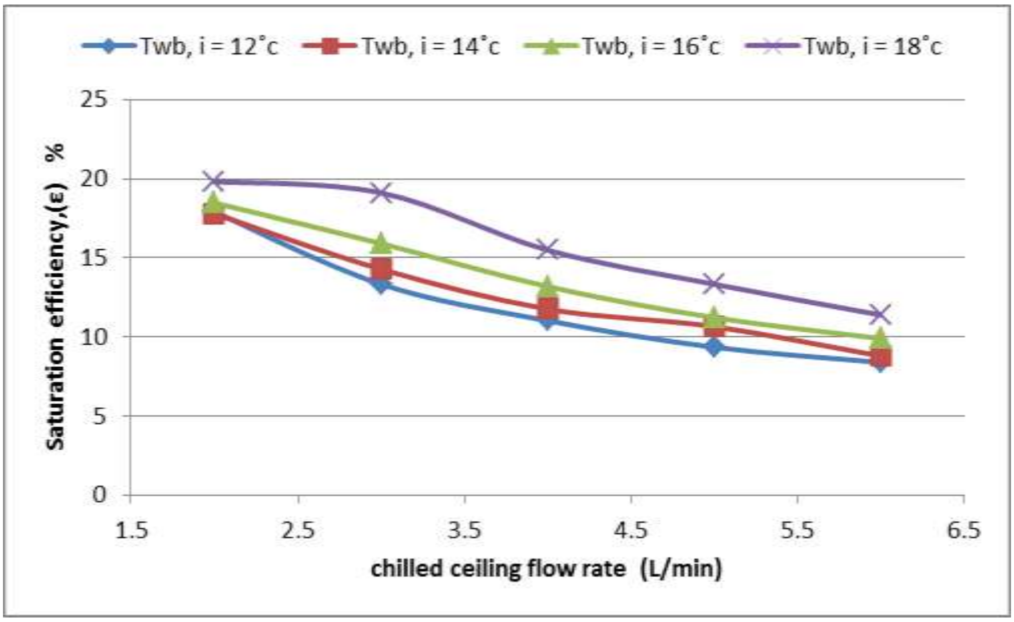


Figure. 5 Influence of the chilled ceiling flow rate on saturation efficiency for different of the inlet air wet bulb temperature

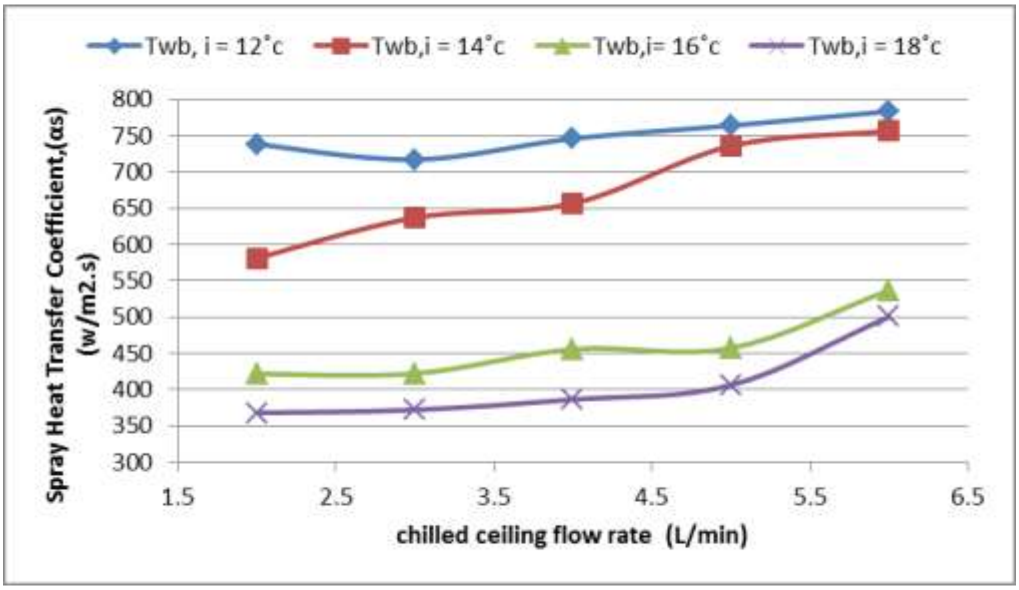


Figure. 6 Influence of the chilled ceiling flow rate on spray heat transfer coefficient for different of the inlet air wet bulb temperature

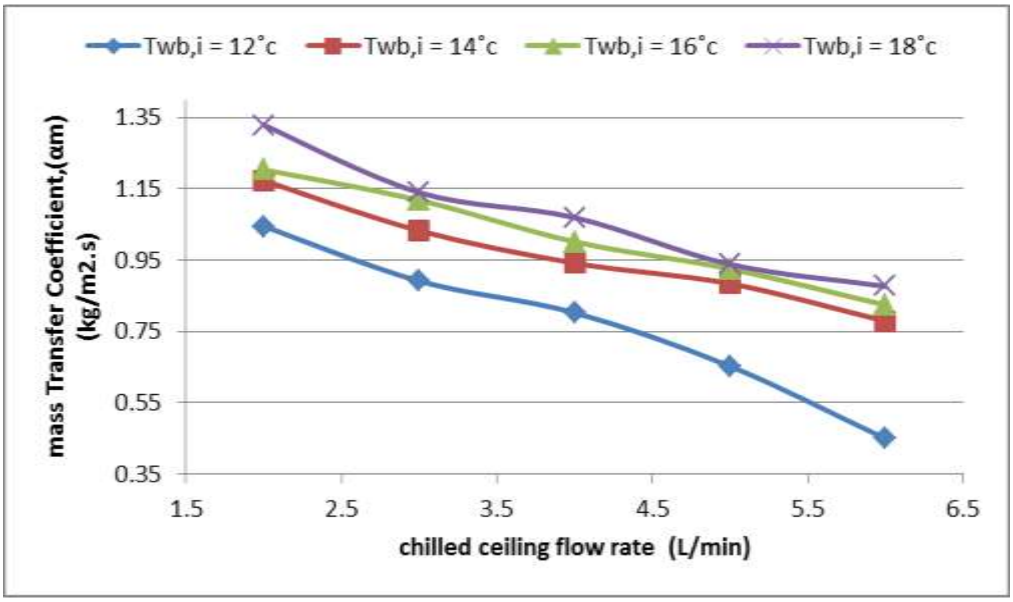


Figure. 7 Influence of the chilled ceiling flow rate on mass transfer coefficient for different of the inlet air wet bulb temperature

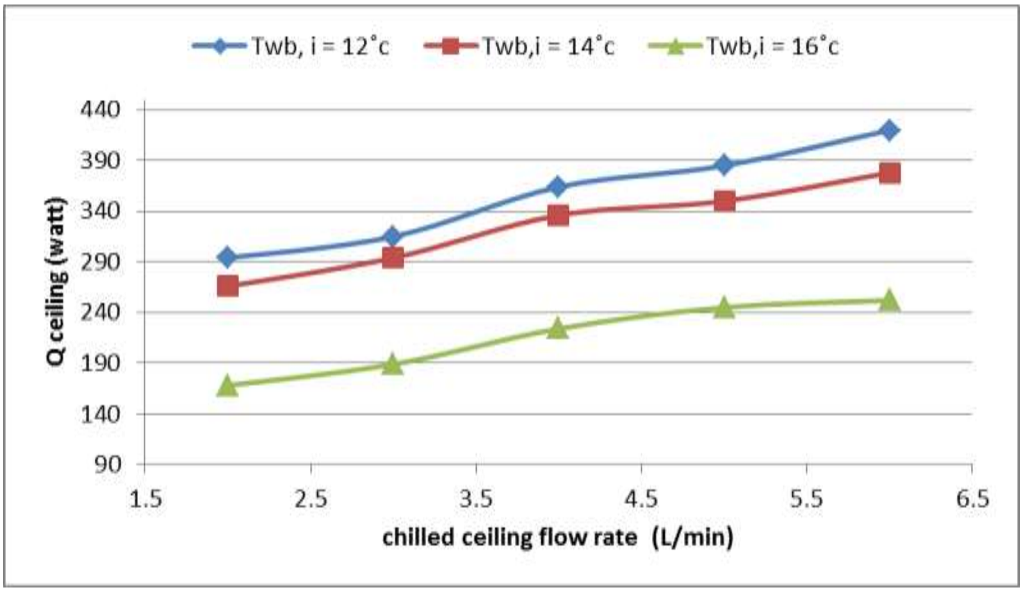


Figure. 8 Influence of the chilled ceiling flow rate on chilled ceiling cooling capacity for different of the inlet air wet bulb temperature

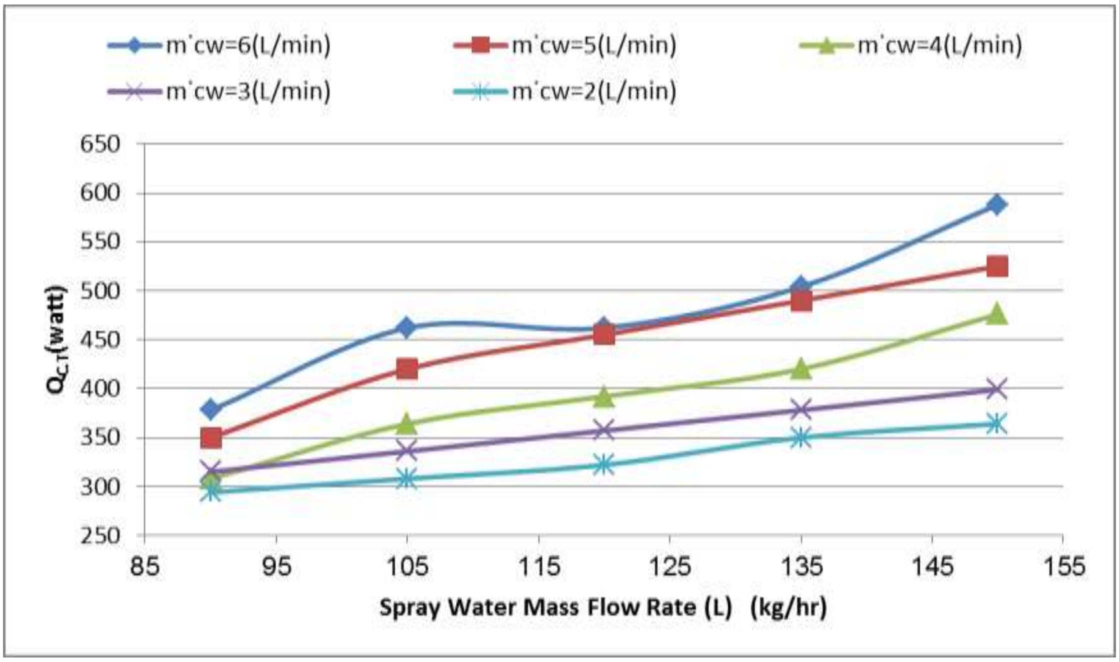


Figure. 9 Influence of the spray water mass flow rate on tower cooling capacity

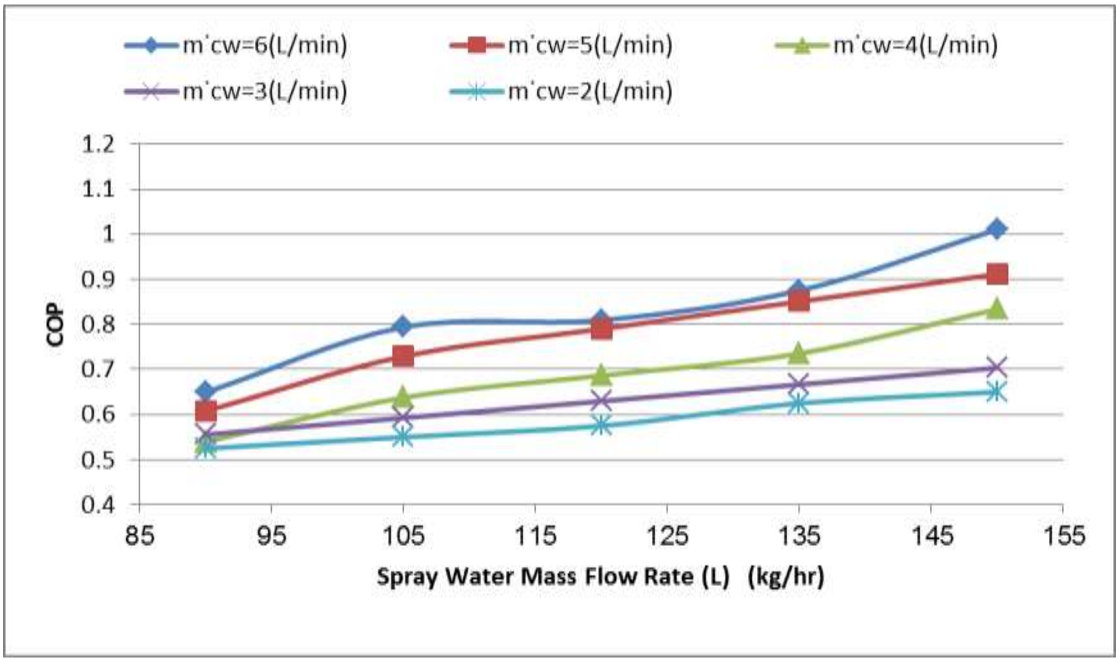


Figure. 10 Influence of the spray water mass flow rate on the coefficient of performance

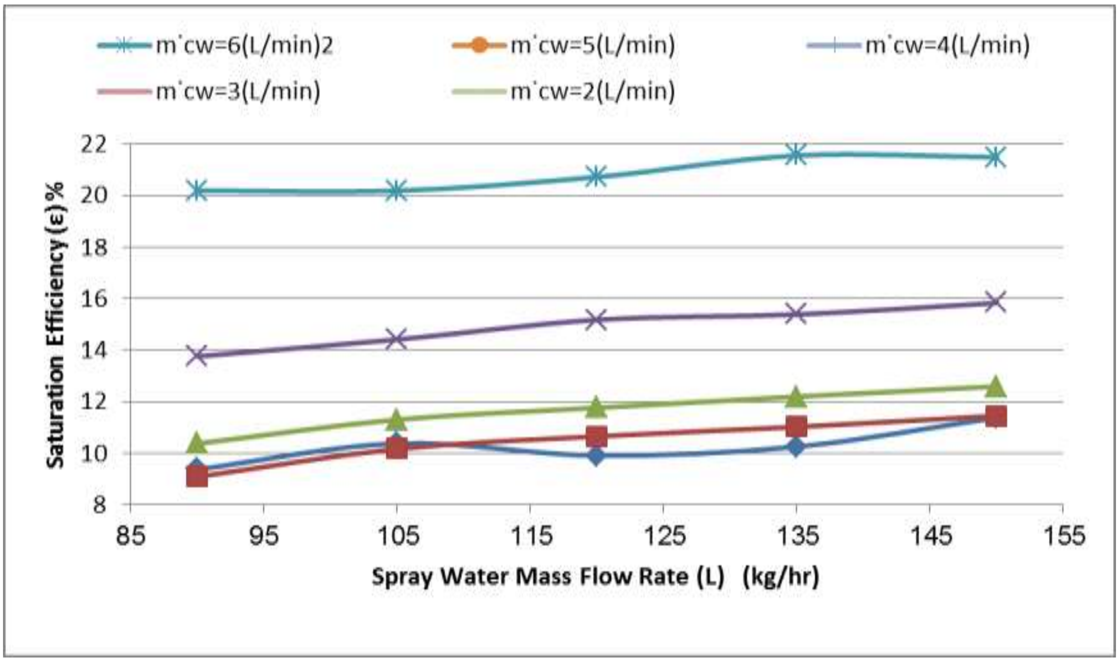


Figure. 11 Influence of the spray water mass flow rate on saturation efficiency

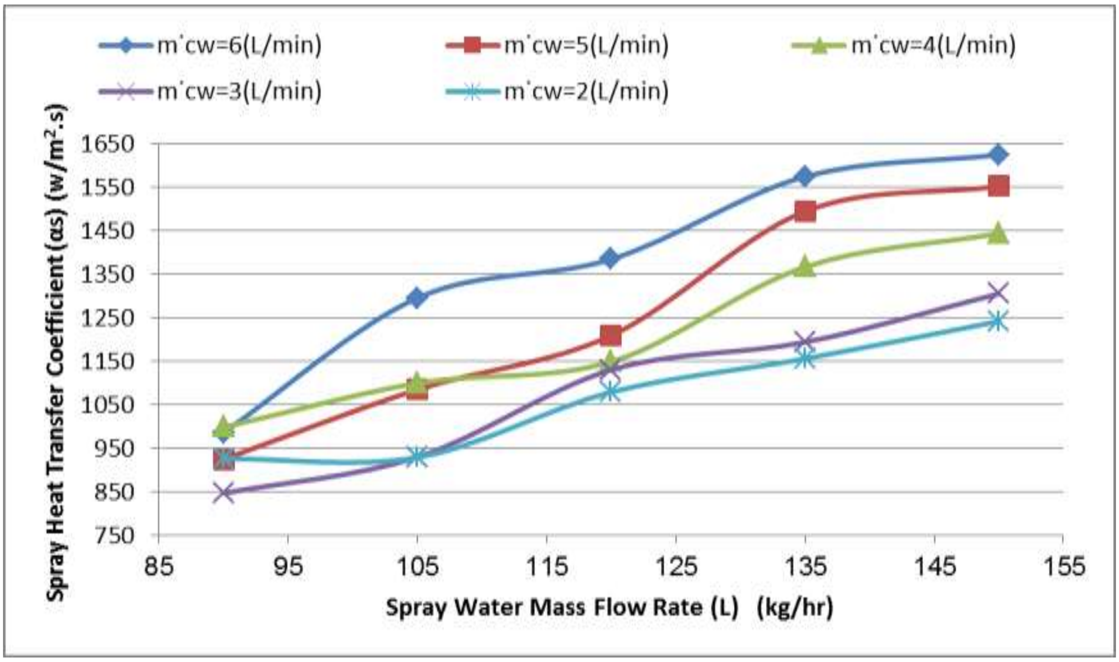


Figure. 12 Influence of the spray water mass flow rate on spray heat transfer coefficient.

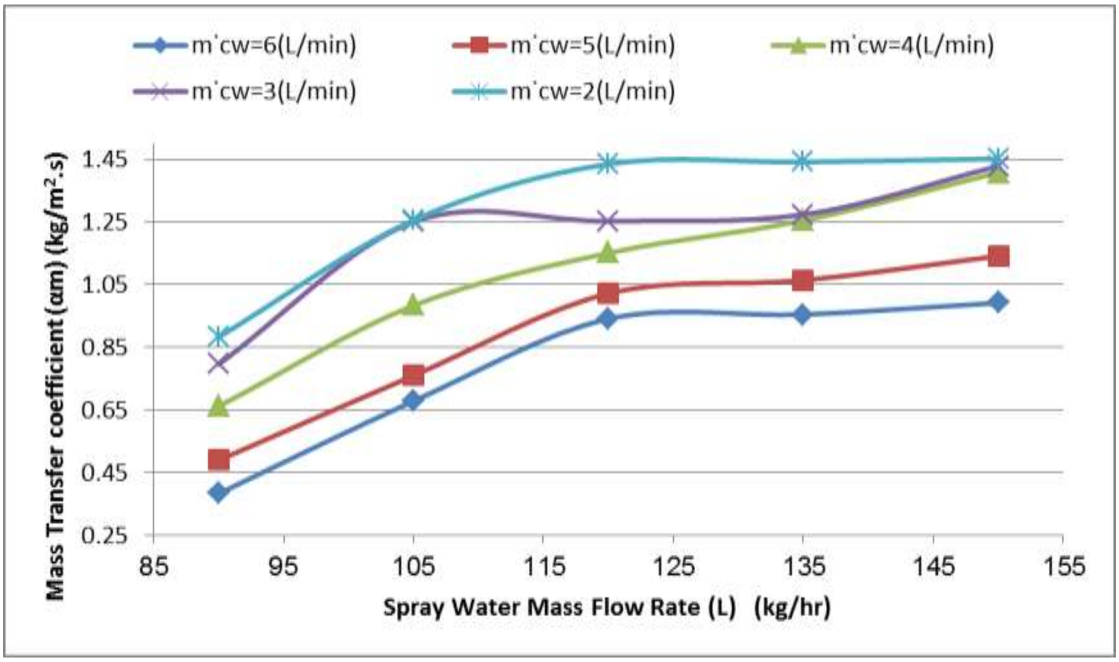


Figure. 13 Influence of the spray water mass flow rate on mass transfer coefficient.

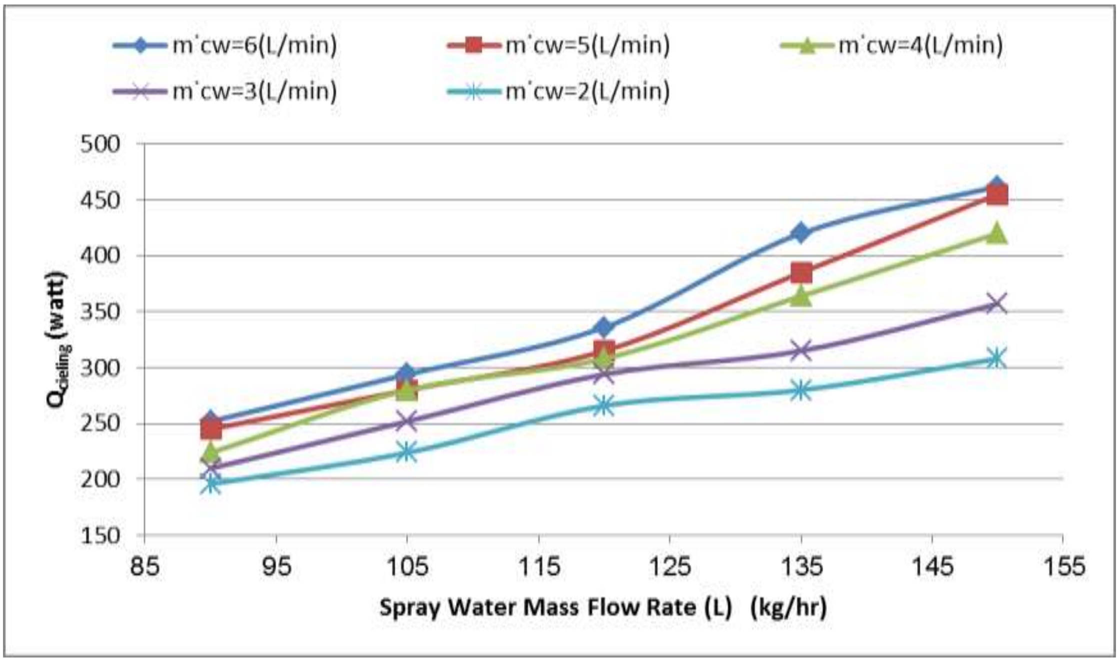


Figure. 14 Influence of the spray water mass flow rate on chilled ceiling cooling capacity

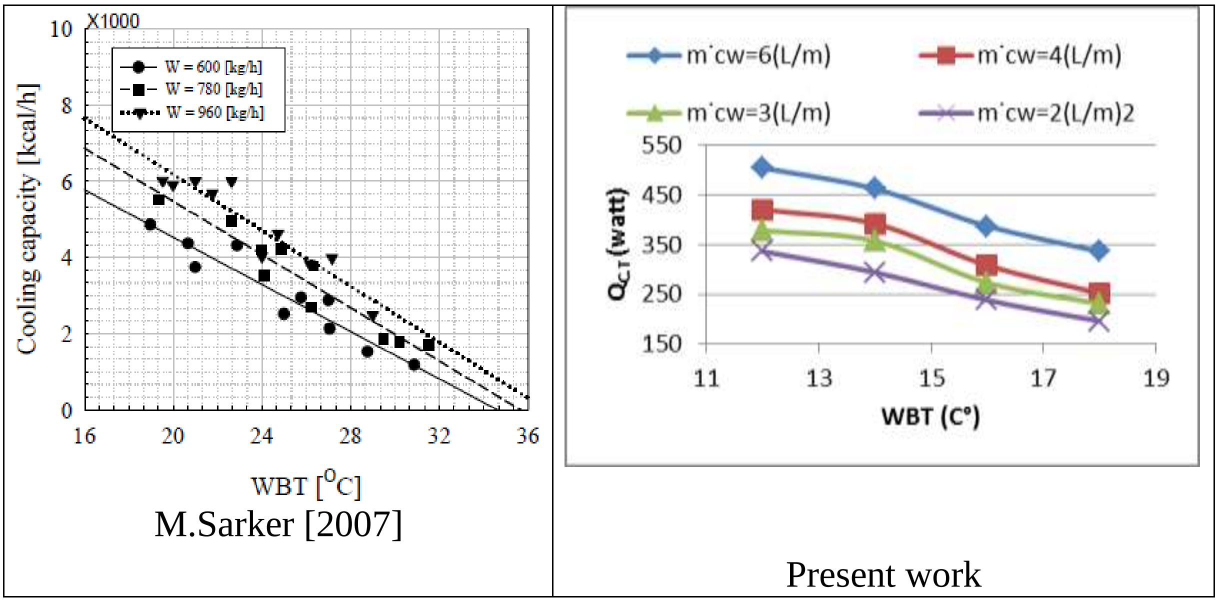


Figure. 15 Comparison of the concluded effect of inlet air wet bulb temperature on cooling capacity with other works

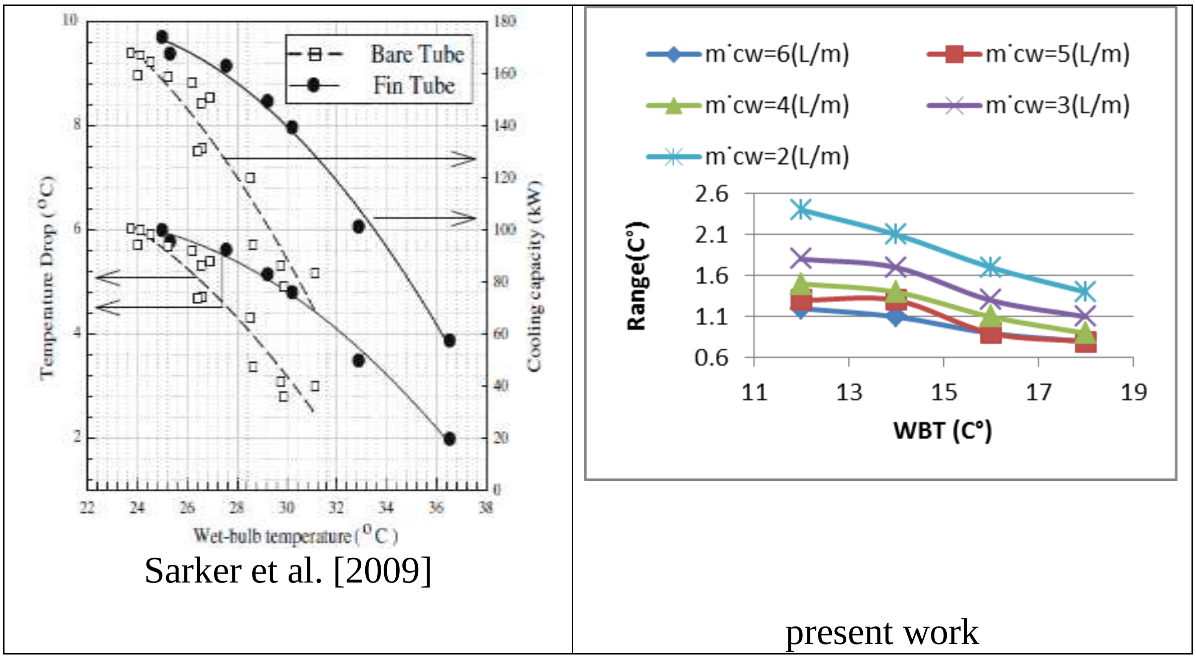


Figure. 16 Comparison of the concluded effect of inlet air wet bulb temperature on cooling water range with other works

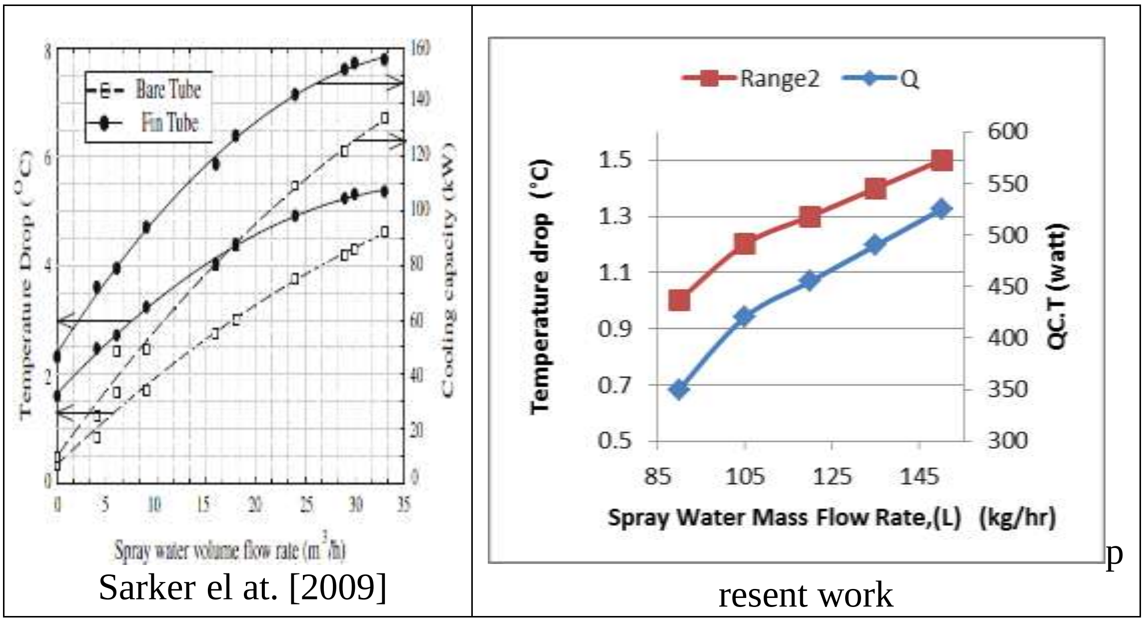


Figure. 17 Comparison of the concluded effect of spray water flow rate on both cooling capacity & temperature drop with other work

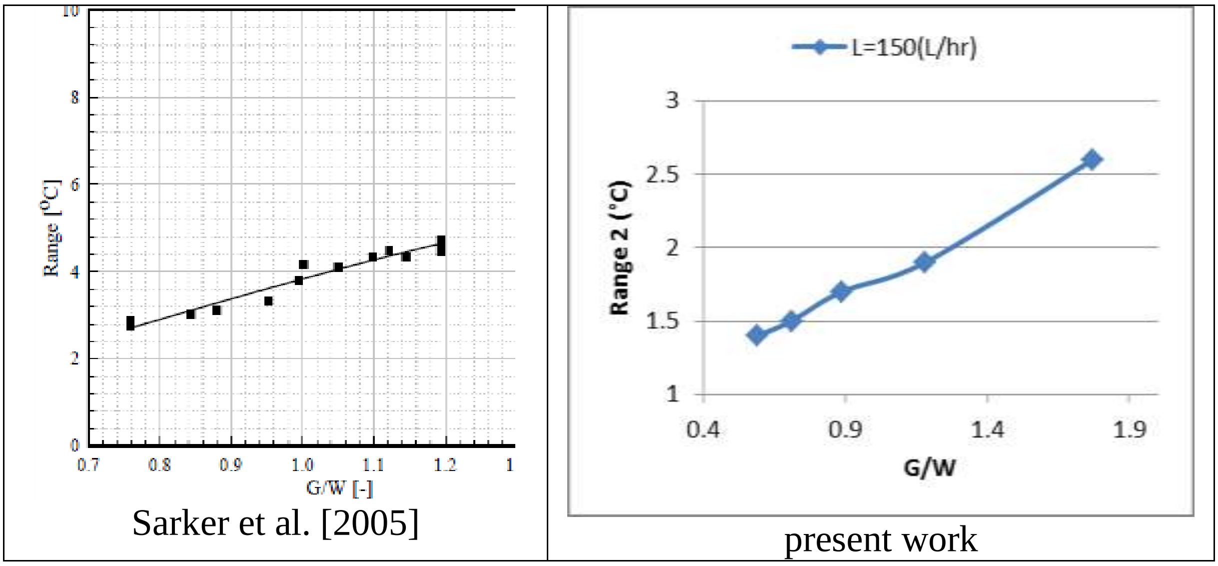


Figure. 18 Comparison of the concluded effect of (air mass flow rate to cooling water ratio) on cooling water range with other work

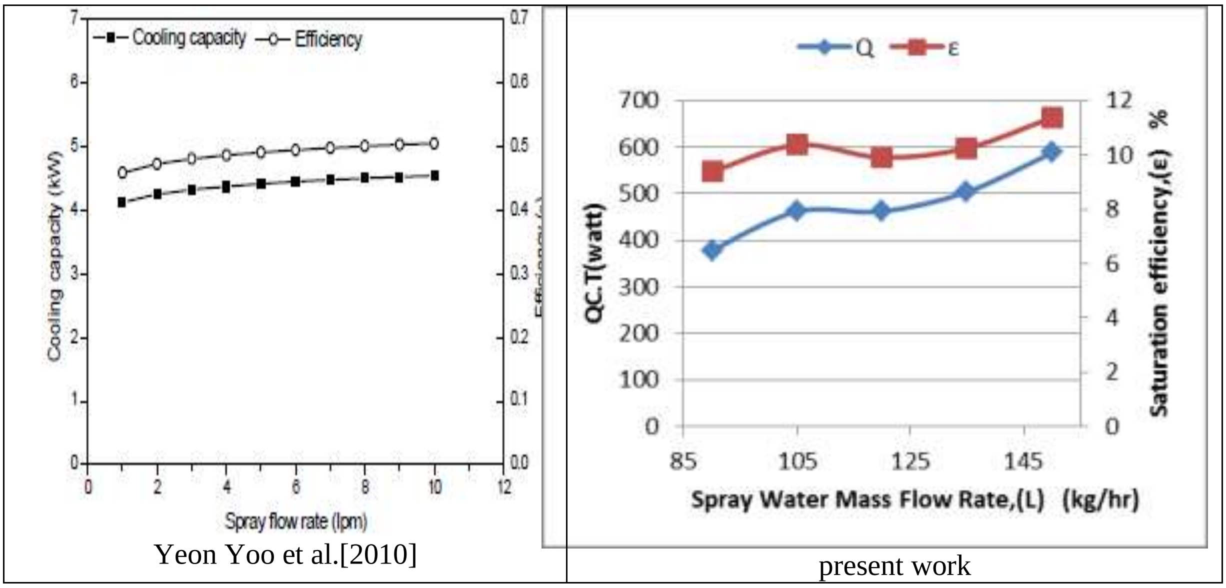


Figure. 19 Comparison of the concluded effect of spray water flow rate on both cooling capacity & saturation efficiency with other work

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NOMENCLATURE

A Area (m^2)
Cp Specific heat at constant pressure (kJ/kg ^oC)
D Outer tube diameter (m)
d Inner tube diameter (m)
G Air mass flow rate (kg/hr)
$\dot{G}$ Air mass velocity based on minimum Section = ρ v (kg/m^2.s)
h Specific enthalpy (kJ/kg)
k Thermal conductivity (W/m ^oC)
L Spray water mass flow rate (kg/hr)
$\dot{m}$ Mass flow rate (kg/hr)
Q Cooling capacity (Watt)
Pr Prandtl number
Re Reynolds number
T Temperature (^oC)
U_o Overall heat transfer coefficient (W/m^2 ^oC)
V Velocity (m/s)
WBT_{in} inlet wet bulb temperature (^oC)

Greek letter

α_m Mass transfer coefficient for water vapor, between spray water film and air (kg/m^2 s)
α_s Heat transfer coefficient between tube surface and spray water film (W/m^2 ^oC)
α_w Heat transfer coefficient for water inside the tubes (W/m^2 ^oC)
Γ Spray water mass rate per length of tube (kg/m s)
ρ Density (kg/m^3)

Sub-Script

ave Average
air Air flow (a)
cw Cooling water
in Inlet
out Outlet
i Interface between spray water film & air
f Saturated air-spray water film
sat Saturation properties
sp Spray water (w)

Abbreviations

CFC Chlorofluorocarbon
LMhD Logarithmic mean enthalpy difference (kJ/kg)
LMTD Logarithmic mean temperature difference (^oC)
CWCT Closed wet cooling tower
ICCCCT Indirect contact closed circuit cooling tower
HVAC Heating ventilation air conditioning