Experimental Investigation into the Heat and Mass Transfer in an Indirect Contact Closed Circuit Cooling Tower

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Abstract

The heat and mass transfer coefficients of the indirect contact closed circuit cooling tower, ICCCCT, were investigated experimentally. Different experiments were conducted involving the controlling parameters such as air velocity, spray water to air mass flow rate ratio, spray water flow rate, ambient air wet bulb temperature and the provided heat load to investigate their effects on the performance of the ICCCCT. Also the effect of using packing on the performance of the ICCCCT was investigated. It was noticed that these parameters affect the tower performance and the use of packing materials is a good approach to enhance the performance for different operational conditions. Correlations for mass and heat transfer coefficients are presented. The results showed a good agreement with other published works. Correlations are showed that the spray heat transfer coefficient is a function flow rates of spray water and air as well as spray water temperature while mass transfer coefficient is a function of spray water and air flow rates only.

Keywords

Closed Circuit Cooling Tower, Experiments, Heat & Mass Transfer Coefficients, Correlations

التحقق العملي من انتقال الحرارة والكتلة في برج تبريد نوع مغلق و ذي اتصال غير مباشر

استاذ مساعد د. نجم عبد جاسم، محمد عبد الخالق الطيار

الخُلاصة

تضمن هذا البحث دراسة معاملي إنتقال الحرارة والكتلة لبرج تبريد من النوع المغلق ذي الإتصال غير المباشر عملياً. حيث تم إجراء عدد من التجارب بتغيير العوامل الحاكمة في أداء برج التبريد كسرعة الهواء، نسبة معدل التدفق الكتلي بين ماء الرش والهواء، معدل تدفق ماء الرش، درجة حرارة البصلة الرطبة للهواء الخارجي و الحمل الحراري المجهز من أجل تحري تأثيرات هذه العوامل على أداء البرج. كذلك فقد تم أيضاً تحري تأثير إستخدام الحشوات على الأداء الحراري للبرج. لقد بينت النتائج أن لهذه العوامل تأثيرات واضحة على أداء البرج وإن استخدام الحشوات طريقة جيدة لتحسين الأداء ولظروف تشغيلية مختلفة. تم تقديم معادلات تجريبية لمعاملي إنتقال الكتلة والحرارة وأظهرت هذه المعادلات توافقاً جيداً مع الأعمال المنشورة في الأدبيات. بينت المعادلات التجريبية أن معامل إنتقال الحرارة يعتمد على معدل تدفق كل من ماء الرش والهواء إضافة إلى درجة حرارة ماء الرش في حين إن معامل إنتقال الكتلة يعتمد على معدل تدفق ماء الرش والهواء فقط.

Introduction

Cooling towers are heat exchangers which are used to dissipate large heat loads to the atmosphere. They are equipments used to reduce the temperature of a water stream by extracting heat from water and emitting it to the atmosphere. Heat is transferred in a cooling tower by two major mechanisms as sensible heat from water to air (convection) and latent heat transfer by the evaporation of water (diffusion). These mechanisms operate at the air-water interface. The total heat transfer is the sum of the effect of these mechanisms.

One considers an elementary control-volume as that shown in **Fig. 1**. This is crossed by a water flow, an air flow and a water spray flow. At the interface between the air and the water spray, there is a film of saturated air, in close contact with the water. This film of saturated air is at the average temperature of the water spray film in this small element of volume. Since the water-vapor partial-pressure at this interface is higher than the water-vapor partial pressure in the air, there is a transfer of water-vapor towards the air. This mass transfer brings a heat transfer related to the water vaporization, called transfer of latent heat. At the same time, because of the difference in temperature between surface of the water and the air, there is transfer of heat by convection (Stabat & Marchio 2004). Radiation effect is likely to be very small at normal conditions and it is generally neglected.

There are numerous types of cooling towers according to the conditions such as climate, place, capacity…etc. The indirect contact closed type cooling tower has been traditionally used in various industrial and HVAC systems. It contains two separate fluid circuits: (1) an external circuit, in which water is exposed to the atmosphere as it cascades over the tubes of a coil bundle, and (2) an internal circuit, in which the fluid to be cooled circulates inside the tubes of the coil bundle. In operation, heat flows from the internal fluid circuit, through the tube walls of the coil, to the external water circuit and then, by dual heat & mass transfer, to the atmospheric air.

The main advantageous of this type compared with an open cooling tower are the contamination risks with airborne dusts & corrosion are limited since the process water never contacts the outside air, the possibility of using it to cool

Fluids other than water as the internal fluid never contacts the atmosphere and it minimizes contamination and maintenance of heat exchangers, chiller condensers and other equipments. The main drawback compared to an open cooling tower is that the cost & the size are increased since a large heat exchange is required to reach the same heat transfer.

The objective of this work is to investigate the thermal performance of the ICCCCT experimentally. This was represented by the mass transfer coefficient between spray water interface and air, α_m, and the heat transfer coefficient between tubes and spray water, α_s.

1. Literature Review

The first basic theory of cooling tower was proposed by Walker in 1923. Several authors presented some correlations of mass transfer coefficient between air and spray water film and heat transfer coefficient between tube external surface and spray water film that take place in closed circuit cooling tower and evaporative cooler.

Parker & Treybal in 1961 were the first researchers presented a detailed analysis of counter flow evaporative liquid coolers. The analysis assumed that the amount of water evaporated is negligibly small. Empirical correlations for heat & mass transfer were presented for 19mm outside diameter staggered tubes as in equs. (1) & (2), respectively. They assumed that the Lewis factor is equal to unity.

$$\alpha_s = 704\left(1.39 + 0.022T_{sp}\right)\left(\frac{\Gamma}{D}\right)^{\frac{1}{3}} \tag{1}$$

$$\alpha_m = 0.049G_{air}^{0.049} \tag{2}$$

Mizushinha et al., in 1967, conducted tests on an evaporative cooler for three different tube diameters 12.7, 19.05 and 40 mm to predict the effect of tube diameter variation on heat & mass transfer coefficients. An assumption of constant spray water temperature inside the tower was applied. The results of mass transfer coefficient were presented as a function of air Reynolds number (Re_{air}) and spray water Reynolds number (Re_{sp}) while heat transfer coefficient was presented as a function of spray water flow rate per unit length (Γ):

$$\alpha_s = 2100\left(\frac{\Gamma}{D}\right)^{\frac{1}{3}} \quad (3)$$

$$\alpha_m A_v = 5.028 * 10^{-8}\, Re_{air}^{0.9}\, Re_{sp}^{0.15}\, D^{-0.26} \quad (4)$$

where A_V is the contact area per unit volume and these correlations are valid for the ranges of $1.5 * 10^3 < Re_{air} < 8 * 10^3$ and $50 < Re_{sp} < 240$.

Nistu et al. in 1969 suggested the heat and mass transfer correlations of an evaporative cooler having both smooth and finned tubes with 16 mm outside diameter in a staggered arrangement. The correlations for plain tubes were as shown below:

$$\alpha_s = 990\left(\frac{\Gamma}{D}\right)^{0.46} \quad (5)$$

$$\alpha_m = 0.076 G_{air}^{0.8} \quad (6)$$

Hasan & Sirén in 2002 developed a computational model for a 10 kW nominal power CWCT with chilled ceilings consisted of 19 tubes of 10 mm outside diameter arranged in 12 rows in a staggered arrangement. They also presented a correlation for the mass transfer coefficient concluded for a total of 60 sets of measurements as shown in equ. (7).

$$\alpha_m = 0.065\ \dot{G}_{air}^{0.773} \quad (7)$$

Oliveira & Facao in 2004 tested a small-size indirect contact cooling tower and correlations for heat & mass transfer were experimentally determined. Experimental results obtained heat transfer coefficient as a function of spray flow rate as shown in equ. (8), while mass transfer coefficients was presented as a function of air flow as shown in equ. (9). The model showed that the correlations have a good degree of accuracy when applied to all possible operating conditions.

$$\alpha_s = 602\left(\frac{\Gamma}{D}\right)^{0.358} \quad (8)$$

$$\alpha_m = 0.064\ \dot{G}_{air}^{0.81} \quad (9)$$

Gyu-Jin Shim et al. in 2008 investigated experimentally the effect of changing the heat exchanger in a CWCT on the heat & mass transfer coefficients and also on cooling capacity. Two heat exchangers consisting of bare-type copper tube of 15.88mm & 19.05mm were used with multi path. It was found that the range of CWCT using two paths is higher approximately 20% than those using one path.

Heyns & Kroger in 2010 investigated the thermal-flow performance characteristics of an evaporative cooler consisting of 15 tube rows with 38.1 mm outer diameter galvanized steel tubes. From the experimental results, correlations for the water film heat transfer coefficient, air–water mass transfer coefficient were developed. Their results showed that the spray water mass flow rate has the greatest influence on the spray heat transfer coefficient but this coefficient is also a function of the air mass flow rate and the spray water temperature as given by equ. (10). It was also found that the air–water mass transfer coefficient is a function of the air mass velocity and the spray water mass velocity as given by equ. (11).

$$\alpha_s = 470\ \dot{G}_{air}^{0.1}\left(\frac{\Gamma}{D}\right)^{0.1} T_{sp}^{0.3} \quad (10)$$

$$\alpha_m = 0.038 \dot{G}_{air}^{0.773}\left(\frac{\Gamma}{D}\right)^{0.2} \quad (11)$$

Yoo et al. in 2010 analyzed the performance of the heat exchanger for the CWCT. Two heat exchangers in inline arrangement were investigated: a 22 row by 11 column with diameter of 9.52 mm (heat exchanger 1) and an 8 row by 5 column with diameter of 25.4 mm (heat exchanger 2). They indicated that the heat transfer coefficient can be obtained from the equation for external heat transfer of tube banks and the mass transfer coefficient was affected by the air velocity and spray water flow rate. This study provides the correlation equation for mass transfer coefficient based on the analogy of the heat and mass transfer and the experimental data. The result from the correlation equation showed accuracy within 5% with the experimental data.

The objects of this study is to analyze the influence of inlet cooling water temperature, inlet air wet and dry bulb temperatures, spray water and air flow rates and heat load on the thermal performance of the indirect contact closed circuit cooling tower. The mass transfer coefficient calculated from heat and mass transfer analogy was compared with experimental data. The regulated correlation was obtained from the result of the comparison. The cooling capacity and thermal efficiency of the closed wet cooling tower were calculated from provided equation and the performance of the tower were investigated.

2. Experimental Apparatus And Method

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. A schematic diagram & a photograph of the experimental apparatus are shown in **Figs**. **2** & **3**, respectively.

The cooling column is the most important portion in the experimental apparatus where the cooling process takes place inside it. It is a duct made from transparent glass with (150*150*800 mm) to allow the spray water paths inside it to be observed.

Cooling water circuit transfers the water that to be cooled through the tower inside the heat exchanger. After it reheated, the cooling water is recirculated through this circuit to cool again. This circuit consists of tank with a heater to reheat the cooling water, pump and the heat exchanger.

Spray water circuit transfers the spray water through the tower to enable the tower from operating in wet mode operation. This circuit consists of tank, pump and spray nozzle.

By the air circuit the air is brought into the cooling tower and it is then blow through the cooling column to absorb heat from the other process fluids. This circuit consists of a blower to blow the air and humidifier and dehumidifier to adjust the conditions of the supplied air such as relative humidity, dry and wet bulb air temperatures.

Measuring devices were used to sense the variations of cooling water temperatures & flow rate, spray water temperatures, flow rate, air temperatures, relative humidity, air velocity, flow rate and electrical voltage and current.

Several experiments were conducted by changing the controlling parameters involving air velocity, spray water to air mass flow rate ratio, spray water flow rate, ambient air wet bulb temperature and the provided heat load to investigate their effects on the performance of the ICCCCT. In addition to this, each experiment was repeated twice changing the above parameters, first without using packing and then packing were fixed inside the cooling column above the heat exchanger to study the effect of packing on the tower transfer coefficients.

After all the experimental data for each experiment were collected and recorded, a set of calculations was started to obtain the experimental mass and heat transfer coefficients for the ICCCCT.

The mass transfer coefficient of water vapor between spray water film and air was calculated after experimental measurements using eq. (12) which was presented by Olivera & Facao 2004:

$$G\left(h_{air,out} - h_{air,in}\right) = \alpha_m \, A \, LMhD \qquad (12)$$

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

$$LMhD \frac{h_{air.outi} - h_{air,in}}{Lin \dfrac{h_{sat.Ti} - h_{air,in}}{h_{sat.Ti} - h_{air.outi}}} \qquad \textbf{(13)}$$

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 2004 as well as Stabat & Marchio 2004 while the inlet and outlet air enthalpies were taken from the psychrometric chart according the measured data.

Spray heat transfer coefficient which takes place between tubes external surface and spray water was calculated by equ. (16) which presented by Olivera & Facao, 2004. Experimental data were used to calculate this coefficient after calculating the overall heat transfer coefficient, U_o, between water inside tubes and the interface based on the outer area of the tube according to equ. (14):

$$Q = \dot{m}_{cw} C_{p,cw} dT_{cw} = U_o \, A \, LMTD \tag{14}$$

where LMTD: is the logarithmic mean temperature difference (^oC) defined as:

$$LMTD \frac{T_{CW.out} - T_{CW.in}}{Lin \dfrac{T_{CW.out} - T_{sp,av}}{T_{CW.in} - T_{sp,av}}} \tag{15}$$

where $T_{sp,ave}$ is the average spray water temperature, ^oC.

After the overall heat transfer coefficient was calculated, it was used to calculate the spray heat transfer coefficient between the tubes external surface and spray water film:

$$\alpha_s = \left[\frac{\dfrac{1}{U_o} - \dfrac{D}{\alpha_w * d} - \dfrac{D}{2k_{tube}} \ln D}{d} \right]^{-1} \tag{16}$$

where α_w is the heat transfer coefficient for water inside the tubes ($W/m^2.^oC$) and it was calculated according to Stabat & Marchio 2004 by the following equation:

$$\alpha_w = 0.023 \, \mathrm{Re}^{0.8} \, \mathrm{Pr}^{0.3} \, k_{cw} / d \tag{17}$$

Reynolds number and Prandtl number were taken for the water inside tubes with values of 7.13 and 11767 respectively.

4. Results And Discussion

4.1 Spray Heat Transfer Coefficient, α_s

Fig. 4 shows the effect of air velocity on the spray heat transfer coefficient, **αs.** It is clear that spray heat transfer coefficient increased slowly with increasing of air velocity. This is because that when air velocity increases, the cooling water range increases and this leads to increase the overall heat transfer coefficient and then to increase the spray heat transfer coefficient. This figure shows that the use of packing has a good effect in enhancing the spray heat transfer coefficient because the packings make the cooling water range to be larger. A correlation for spray heat transfer coefficient was concluded from the experimental results for the case of cooling tower without packings and it conforms well to the experimental values of this experiment as seen in this figure. This correlation is given by:

$$\alpha_s = 555 \left(\dot{G}_{air}\right)^{0.075} \left(\frac{\Gamma}{D}\right)^{0.48} \left(T_{sp,ave}\right)^{0.1} \tag{18}$$

Fig. 5 shows the effect of the ratio of L/G on the spray heat transfer coefficient. This figure illustrates that the spray heat transfer coefficient decreases with increasing of L/G. This can be attributable to the fact that when L/G increases; the air flow becomes insufficient to transfer the same amount of heat. In case of using packing in the tower operates the spray heat transfer coefficient is increased due to increasing of the surface area.

Fig. 6 indicates that the heat transfer coefficient increases with increasing spray water flow rate. This can be explained by equ. (14). When spray flow rate increases, the cooling capacity increases too leads to increase the overall heat transfer coefficient, $\mathbf{U_o}$, and then $\boldsymbol{\alpha_s}$ increases according to equ. (16). This figure also shows that the presented correlation of spray heat transfer coefficient conforms well to the values of spray heat transfer coefficient of this experiment when packing were not used. If this figure compared with **Fig. 4**, it can be noticed that the spray heat transfer coefficient depends greatly on spray flow rate and little on air velocity, this is clear by the difference between the exponents of spray and air flow rates in equ. (18). The average spray water temperature changes with respect to spray flow rate; this makes spray water temperature affects the spray heat transfer coefficient as it is clear in the presented correlation and this is conform with the correlation of the spray heat transfer coefficient that presented by Heyns & Kroger 2010.

When the wet bulb temperature increases, the spray heat transfer coefficient decreases as shown in **Fig. 7**. This is simply because of the fact that the cooling capacity decreases with respect to the wet bulb temperature which leading to decrease the overall heat transfer coefficient and thus the spray heat transfer coefficient decreases. Spray heat transfer coefficient becomes larger with using packing as it compared with that when packing were not used as shown in this figure, but the difference in the values between the two cases decreases for high wet bulb temperatures.

Spray heat transfer coefficient influenced by the heat load as shown in **Fig. 8**. When the load increases the cooling water range was increased leading to increase the overall heat transfer coefficient according to equ. (14) consequently the spray heat transfer coefficient was increased. This figure also shows that the spray heat transfer coefficient as the packing used is 10.25 % higher than its values when the packing were not used.

In **Fig. 9** the spray heat transfer coefficient correlation which concluded in the present work, equ. (18), is compared with other previous works conducted by Nistu et al. 1969 for plain tubes of 16 mm outside diameter in a staggered arrangement and Olivera & Facao, 2004 for 222 staggered tubes of 10 mm outside diameter. This figure shows that the presented correlation falls with about to 11.6 % from Nistu et al. correlation and rises with about to 26.6 % above Olivera & Facao correlation .

4.2 Mass Transfer Coefficient, α_m

In **Fig. 10** it is shown that the mass transfer coefficient for water vapor between spray water film and air increases greatly as air velocity increases. This increasing in mass transfer coefficient can be attributable to the increase of water evaporation rate as air flow increases. When the packing added to the tower, mass transfer coefficient shows increasing in its values because of increasing in the surface area of mass transfer. A correlation for mass transfer coefficient was concluded from the experimental results for the case of cooling tower without packing as in equ. (19). This correlation conforms well to the experimental values seen in this figure.

$$\alpha_m = 0.063\,(\dot{G}_{air})^{0.81}\left(\frac{\Gamma}{D}\right)^{0.1} \qquad (19)$$

The mass transfer coefficient of water vapor between spray water and air with respect to the ratio of mass flow rate of spray water and air, **L/G**, is shown in **Fig. 11**. The experiment was conducted keeping L constant and letting **G** to vary. From this figure, it is evident that mass transfer coefficient decreases with increasing of **L/G**. This is mainly return to that when **L/G** increases this means that air flow inside the tower will be unproportionate with spray flow which leads to decrease the capability of air to gain more water vapor and this decreases the mass transfer coefficient.

Fig. 12 indicates that the mass transfer coefficient is influenced by spray water flow rate. As spray flow increases the 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. This figure also shows that the presented correlation of mass transfer coefficient, equ. (19), conforms well to the values of this experiment when the tower operates without packing. The increasing rate in mass transfer coefficient with respect to spray flow is much less than that with respect to air flow, as seen in figure (10) and this is clear by the difference between the exponents of spray and air flow rates in the presented correlation of mass transfer coefficient equ. (19). This figure also shows that mass transfer coefficient increases when the tower operates with using packing material, due to increase the surface area of mass transfer.

When the wet bulb temperature of the inlet air increases, mass transfer coefficient increases slightly as shown in **Fig. 13**. But when the wet bulb temperature increased 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 in increasing mass transfer coefficient. For example, for the case without packing, when wet bulb temperature increased from 20 to 32°C, $\mathbf{\Delta h_{air}}$ decreased from 39.75 to 5.6 kJ/kg with a ratio of 7.1 while **LMhD** decreased from 21.88 to 2.536 kJ/kg with a ratio of 8.63 and this was the reason to increase mass transfer coefficient from 0.241 to 0.293 kg/m^2.s. This figure shows also that the tower operates with packing has higher values for mass transfer coefficient as it operates without packing.

In **Fig. 14** the influence of the heat load on the mass transfer coefficient is shown. It is clear that mass transfer coefficient increases slightly with increasing the heat load. This can be attributable to the increase in the outlet air enthalpy as heat load increases which increasing the difference between the outlet & inlet air enthalpies which then affects the mass transfer coefficient according to equ. (12). This figure indicates that the mass transfer coefficient increases with about 7.78 % in case of the tower operates using packing.

In **Fig. 15** the mass transfer coefficient correlation which concluded in the present work, eq. (19), is compared with other previous works conducted by Parker and Treybal, 1961 for 19 mm outside diameter staggered tubes, and Hasan and Serin, 2002 for 34 mm outside diameter staggered tubes arranged in 13 rows*20 columns, and Olivera and Facao, 2004 for 222 staggered tubes of 10 mm outside diameter. This figure shows that the presented correlation falls within the range of other correlations and it conforms well to them especially Hasan and Serin 2002.

5 CONCLUSIONS

The thermal performance of a small size indirect contact closed type cooling tower was investigated experimentally. It was found that the spray water flow rate has the greatest influence on the spray heat transfer coefficient, α_s, but it is also a function of the air flow rate and the spray water temperature. In addition to this, both air and spray water flow rate affect the mass transfer coefficient, α_m, but the great effect belongs to the air flow rate. The results of increasing the ratio of spray water to air mass flow rate indicate that it decreases the tower transfer coefficient. Correlations for spray heat transfer coefficient and mass transfer coefficient for ICCCCT were concluded. These correlations was found to represent the experimental results very and also in a good agreement with the previous works. Wet bulb temperature w2as found to has a great influence on the characteristics of the tower. It decreases spray heat transfer coefficient while it increases mass transfer coefficient. The cooling water heat load increases both heat and mass transfer coefficients. The effect of packing material and then repeated using packing was found to has a relatively good enhancement on the cooling tower performance.

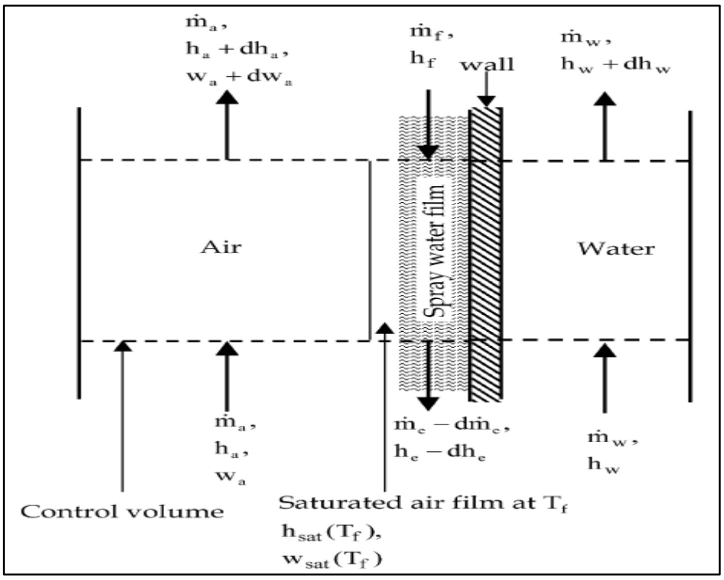

Fig. 1 Heat & mass transfer mechanisms in the Indirect contact cooling tower

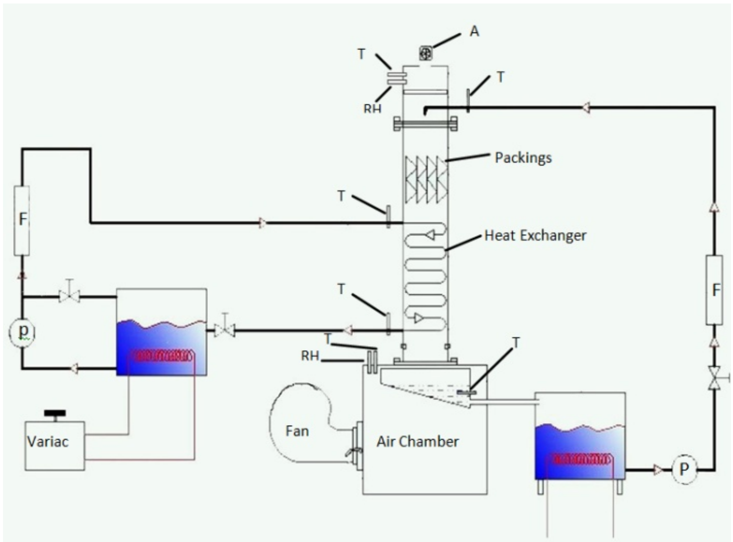

Fig. 2 Schematic representation of the experimental apparatus

A: Anemometer T: Thermocouple RH: Relative Humidity Sensor F: Flow Meter P: Pump

Fig. 3 Photograph of the experimental apparatus

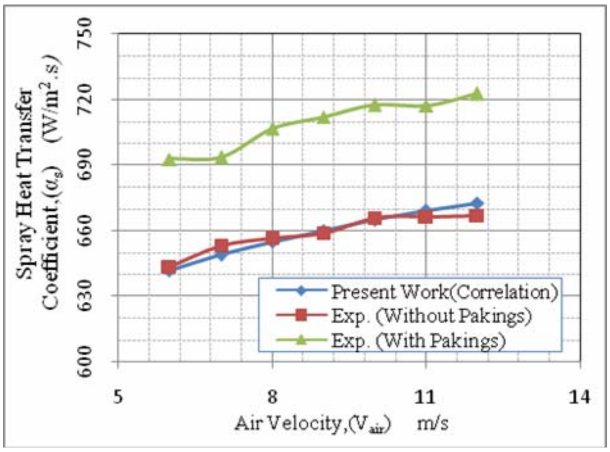

Fig. 4 Influence of the air velocity on spray heat transfer coefficient

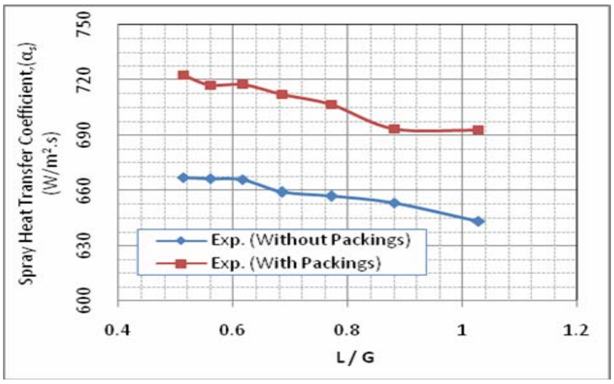

Fig. 5 Influence of spray water to air mass flow rate ratio on spray heat transfer coefficient

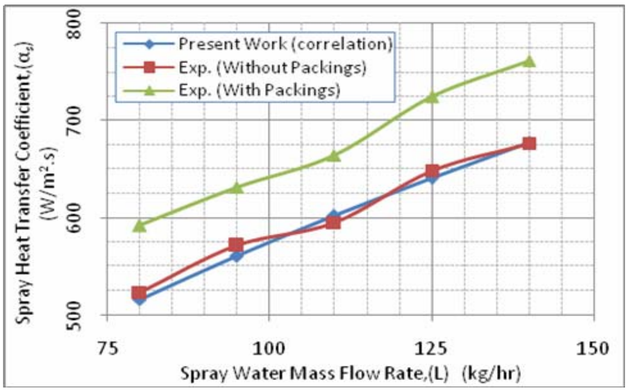

Fig. 6 Influence of the spray water mass flow rate on spray heat transfer coefficient

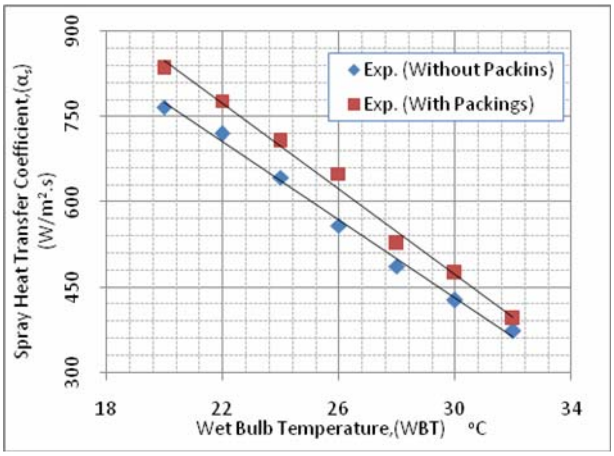

Fig. 7 Influence of the inlet air wet bulb temperature on spray heat transfer coefficient

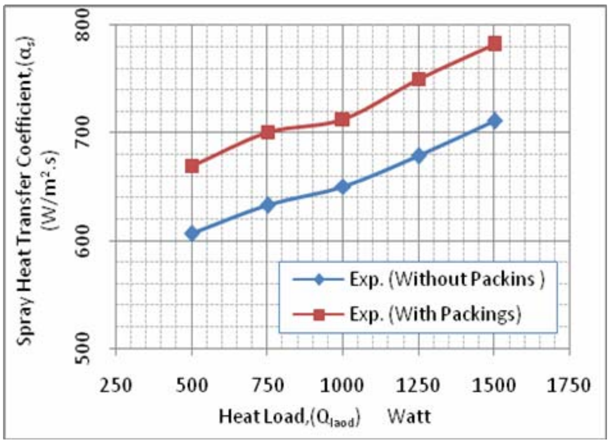

Fig. 8 Variation of spray heat transfer coefficient with respect to the heat load

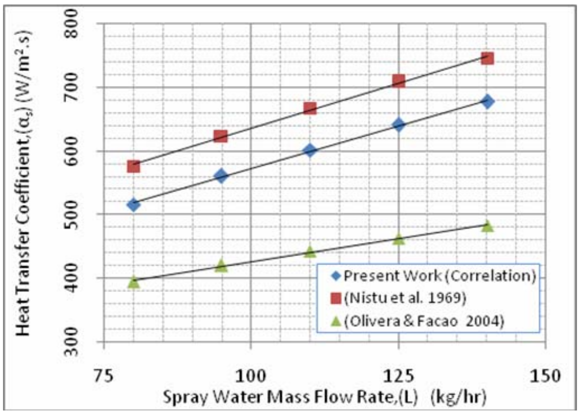

Fig. 9 Comparison of the presented correlation for the spray heat transfer coefficient with other works

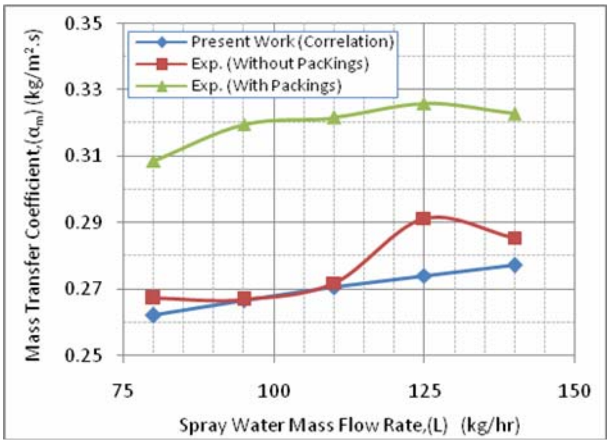

Fig. 12 Influence of the spray water mass flow rate on mass transfer coefficient

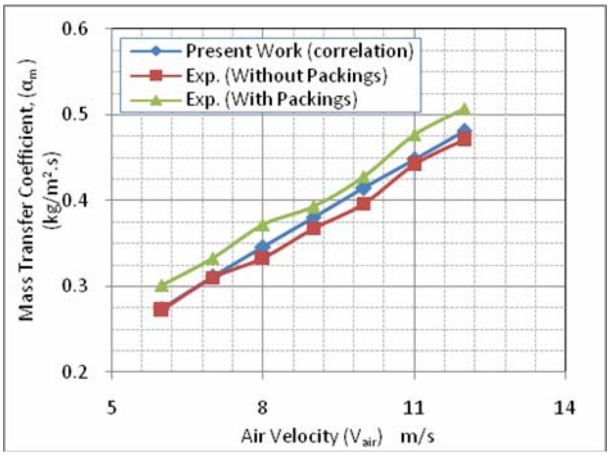

Fig. 10 Influence of the air velocity on mass transfer coefficient

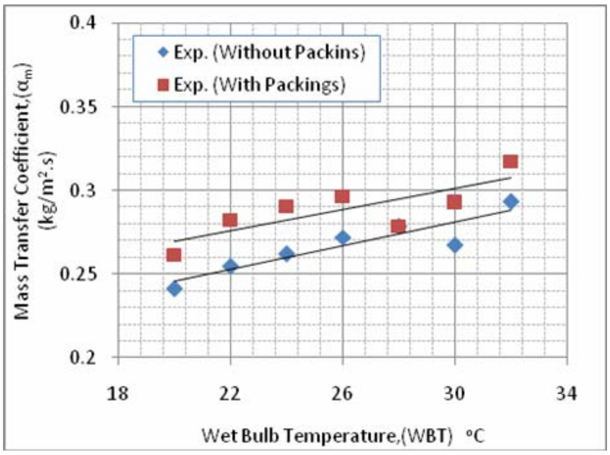

Fig. 13 Influence of the inlet air wet bulb temperature on mass transfer coefficient

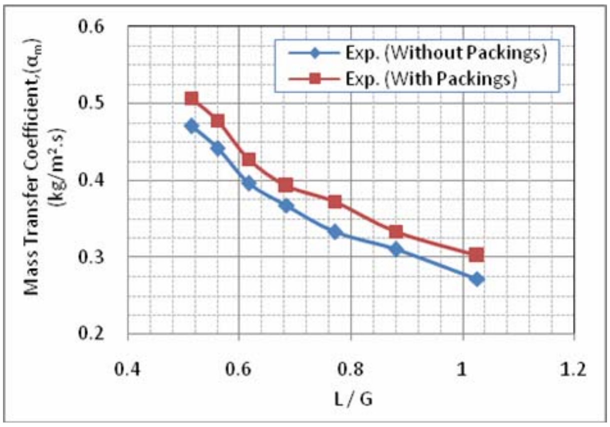

Fig. 11 Influence of spray water to air mass flow rate ratio on mass transfer coefficient

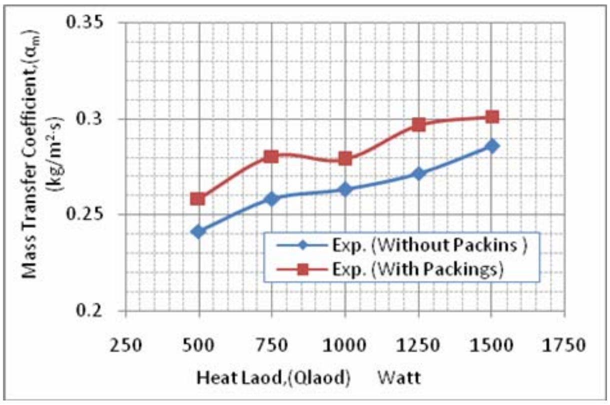

Fig. 14 Variation of mass transfer coefficient with respect to the heat load

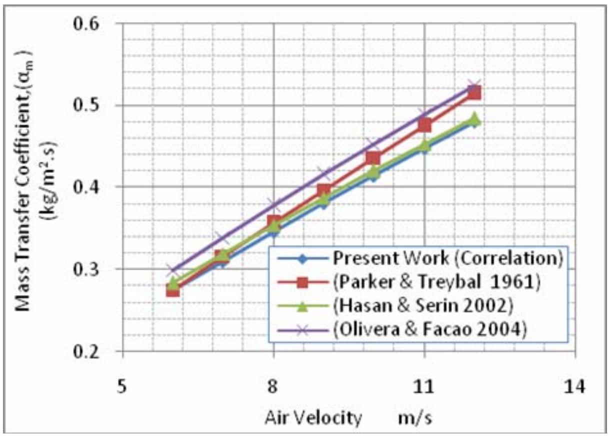

Fig. 15 Comparison of the presented correlation for the mass transfer coefficient with other works

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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 = ρv ($kg/m^2.s$)
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)

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

LMhD Logarithmic mean enthalpy Difference (kJ/kg)
LMTD Logarithmic mean temperature Difference (^{o}C)
CWCT Closed wet cooling tower
ICCCCT Indirect contact closed circuit cooling tower
HVAC Heating ventilation air conditioning