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THEORETICAL AND EXPERIMENTAL STUDY OF THERMAL PERFORMANCE WITHIN A COUNTER-FLOW WET COOLING TOWER

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Abstract

This study is interested on experimental and theoretical of heat and mass transfer phenomena in a counter-flow wet cooling tower. A simplified thermal model has been used to simulate the thermal behavior and the performance of the tower. The calculation of the heat and mass transfer parameters to determine the characteristics of the good operation of the tower. Eight experimental manipulations were conducted to study the influence of various parameters (such as: thermal cooling load, the flow of water and air) on the thermal performance of the tower. The results obtained show that the increase in cooling load contributes to a considerable reduction in the effectiveness of the tower, as well as the increase in the water flow is used to decrease the performance of the tower, while the increase in airflow is used to improve considerably the different parameters of performance. It can be seen that the results obtained by simulation coincide quite well with the experimental results. The minor discrepancies between the present simulation and experimental measurements can be attributed to the measurements uncertainties and simulation accuracy.

Keywords: Cooling tower, Experiments, Heat transfer, Mass transfer, Modeling.

1. Introduction

A considerable amount of energy is converted into heat during all manufacturing and industrialized processes, irrespective of its form (mechanical, electrical, chemical). This energy being always neither recovered nor recycled, but it is extracted from the process and released to the surrounding environment through some cooling process. This non-recoverable heat called residual heat may be alters the proper functioning of the equipment. Cooling is naturally done by exchange with the surrounding environment, but this is not always sufficient, however, it is necessary to use force cooling. Among the employed solutions, a wet cooling tower can be can be used to evacuate the heat of cooling system the outside environment (as shown in Fig. 1) by pulverizing hot water in air-flow. This water sprays by gravity inside a cool air stream back into the tower. The air stream call water by evaporation part of sprayed water. Evaporated part of water is sometimes visible as panache above the tower.

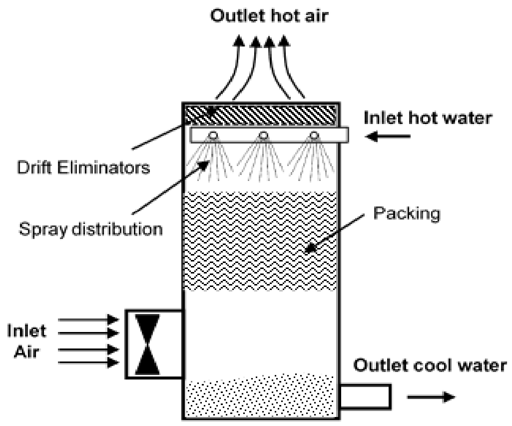

Fig. 1. Schematic representation of a counter-flow wet cooling tower.

Robinson [1] was the first who considered the problem of cooling tower in 1922. Walker et al. [2] developed the basic equations for heat and mass transfer by considering them separately. They used the ambient air humidity as the sole driving force for the cooling process in cooling towers. Merkel [3] however, developed the most widely used theory for cooling tower calculation. Various mathematical models have been developed to predict the thermal behavior of wet cooling towers. Merkel [3] proposed the first practical model to describe the heat and mass transfer mechanisms in wet cooling towers, in which he combined the equations for heat and water vapor transfer. He demonstrated the utility of total heat or enthalpy difference as a driving force to allow for both sensible and latent heats. Majumdar et al. [4] studied numerically the performance of natural and forced cooling tower in two dimensions. Bedekar a et al. [5] have studied experimental investigation of the performance of counter flows in packed bed mechanical cooling and showed that the tower performance decreases with an increase in the (L/G) ratio. Goshayshi and Missenden [6] studied experimentally the mass transfer and the pressure drop characteristics of many types of packing, including smooth and rough surface corrugated packing in the atmospheric cooling tower. Stabat and Marchio [7] presented a simplified model for indirect cooling towers behavior. This model is devoted to building simulation tools and satisfies several criteria such as simplicity of parameterization, accuracy, and possibility to model the equipment under different operation condition. Fisenko et al. [8] developed a mathematical model of mechanical draft cooling tower, and took into account the radii distribution of the water droplets. Kloppers and Kröger [9] analysed the derivation of heat and mass transfer equations in counter flow wet cooling towers in detail. They described Merkel, NTU and Poppe methods and concluded that Poppe method yields higher Merkel numbers.

Several researchers have investigated this subject through theoretical and experimental analysis of the heat and mass transfer processes. Lemouari et al.[10] presented an experimental investigation of the thermal performances of a forced draft counter flow wet cooling tower filled with a VGA (Vertical Grid Apparatus) type packing. They studied the effects of the air and water flow rates on the cooling water range as well as the tower characteristic, for different inlet water temperatures. Lemouari et al. [11] investigated experimentally the thermal performance of a forced cooling tower used in a solar desalination system based on humidification–dehumidification of air. They used a counter flow wet cooling tower filled with film packing materials. They obtained the measured variables for wide ranges of mass flow rates of air and water as well as for several inlet water temperatures. Those researchers evaluated the tower characteristic and efficiency in terms of water to air mass flow rate ratio. Kara [12] investigated experimentally the thermal performances of a forced draft counter flow wet cooling tower. The factors affecting on the cooling tower performance, such as water inlet and outlet temperatures, air and water mass flow rates, heat load, and effectiveness of the cooling tower are investigated. The effect of air mass-flow rate on approach and range of the cooling tower, for different water mass-flow rates has been investigated. These results show that cooling tower performance increases with the increase in air mass-flow rate. Braun et al. [13] developed effectiveness models for cooling towers, which utilized the assumption of a linearized air saturation enthalpy and the modified definition of the number of transfer units. The models were useful for both design and system simulation. However, Braun’s model needs iterative computation to obtain the output.

Ramkumar and Ragupathy [14] investigated the thermal performance of forced draft counter flow wet cooling tower experimentally with expanded wire mesh vertical and horizontal orientation types packing, the vertical orientation wire mesh packing is having better performance than horizontal orientation wire mesh packing. Lucas et al. [15] presented an experimental study of thermal performance of a wet cooling tower with forced draft in counter current equipped with a gravity water distribution system (GWDS) for six-drop separators.

Hernandez-Calderon et al. [16] most recently adopted an orthogonal technique to solve the Poppe method equations for heat and mass transfer in counter flowing wet cooling towers. They introduced the air humidity ratio as a finite power series at water temperature. The air enthalpy is expressed as a function of the water temperature and unknown coefficients of the expansion from the humidity ratio. They applied this methodology to eight examples, and their results were compared to the results obtained when the governing equations are integrated with the Dormande Prince method. Mansour and Hassab [17] presented the new correlations for calculating the thermal performance of counter flow wet-cooling tower. Those correlations are based on solving heat and mass-transfer equation coupling with energy equations simultaneously. They obtained results showed a very good agreement with a deviation less than 10% with those obtained from the literature for a temperature difference between the inlet water temperature and inlet air wet-bulb temperature equal to or less than 10 K.

The objective of this article is to study experimentally and theoretically, the heat and mass exchange within a counter-flow wet cooling tower.

Investigational testing was carried out to show the influence of some operating parameters on the thermal performance of the tower. The influence of water and air mass-flow rates as well as the amount of heat to evacuate; on cooling tower efficiency will be evaluated.

2. Mathematical Modeling

In this study, a mathematical model is developed for a counter-flow wet cooling towers. In this case, the saturation enthalpy of air is assumed to be linear. (by employing the assumption of a linearized air saturation enthalpy). The model is established based on the following assumptions (Braun model) [18].

- No heat and mass transfer in the water and air flow directions.
- The heat and mass transfers through the walls are negligible.
- The heat inputs due to the fan are negligible.
- The specific heat of water and the dry air are assumed to be constant.
- Water loss is negligible compared to its input flow.

A schematic of the counter-flow wet cooling tower with the different flows of water and moist air in an elementary volume is shown in Fig. 2.

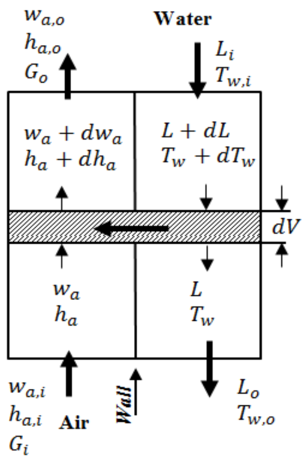

Fig. 2. Flow steams inside an elementary volume.

The steady-state energy balance between the water and the air is given by the following equation:

$$Gdh_a = Ldh_{f,w} + h_{f,w}dL \tag{1}$$

The steady-state mass balance allows as describing the water loss and the water mass-flow within the tower. There, are given by the following equations:

$$dL = GdW_a \quad (2)$$

$$L = L_i - G\left(W_{a,o} - W_a\right) \quad (3)$$

From Eqs. (1), (2), and (3), the water temperature in the control volume is determined by the following relation:

$$dT_w = \frac{dh_a - C_{P_w}\left(T_w - T_r\right)dW_a}{\left[\frac{L_i}{G} - \left(W_{a,o} - W_a\right)\right]C_{pw}} \quad (4)$$

where T_r is the reference temperature of zero enthalpy of liquid water.

The enthalpy variation of the air stream is equal to the rate of energy transfer from the water droplets due to both heat and mass transfer by convection and evaporation. It is re-written as following equation:

$$Gdh_a = h_c A\left(T_w - T_a\right)dV + h_{g,w}GdW_a \quad (5)$$

Assuming that the mass element of water vapor in the mixture of air and vapor is approximately equal to the humidity ratio, the rate of mass transfer of water vapor to the air stream can be written as:

$$GdW_a = h_d A\left(W_{s,w} - W_a\right) \quad (6)$$

By introducing the Lewis number L_e, the enthalpy of water vapor can be written as:

$$Gdh_a = h_d A\left[L_e C_{pm}\left(T_w - T_a\right) + \left(W_{s,w} - W_a\right)h_{g,w}\right]dV \quad (7)$$

$$L_e = \frac{h_c}{(h_d C_{pm})} \quad (8)$$

C_{Pm} Pressure specific heat of moist air.

The number of transfer unit is correlated by the following relation [18]:

$$NTU = \frac{h_d .A.V}{G} = c.\left(\frac{L}{G}\right)^n \quad (9)$$

To determine the coefficients *c* and *n*, we have to know either two operating points or the characteristics of the packing.

$$n = \frac{In\left(NTU_1\right) - In\left(NTU_2\right)}{In\left(\frac{L_1}{G_1}\right) - In\left(\frac{L_2}{G_2}\right)},\ c = exp\left(In\left(NTU_1\right) - n.In\left(\frac{L_1}{G_1}\right)\right)$$

By employing the NTU definition, the Eqs (6) and (7) can be reduced to:

$$dW_a = -\frac{NTU}{V}\left(W_a - W_{s,w}\right)dV \quad (10)$$

$$\frac{dh_a}{dV} = -\frac{L_e NTU}{V}\left[\left(h_a - h_{s,w}\right) + \left(W_a - W_{s,w}\right)\left(1/L_e - 1\right)h_{g,w}\right] \quad (11)$$

If the Lewis number is equal to 1 in the considered field, the Eq. (7) of the enthalpy variation of moist air, becomes:

$$\frac{dh_a}{dV} = -\frac{NTU}{V}\left(h_a - h_{s,w}\right) \tag{12}$$

Nevertheless, neglecting the water losses compared to the inlet water flow, Eq. (4) reduces to:

$$dT_w = \frac{G.dh_a}{L.C_{P_w}} \tag{13}$$

Equation (13) can be rewritten in terms of enthalpy by introducing the concept of enthalpy air saturation at the water temperature, as follows:

$$dh_{s,w} = \frac{G.C_{P_s}.dh_a}{L.C_{P_w}} \tag{14}$$

$$C_{Ps} = \frac{dh_s}{dT} \tag{15}$$

Equations (12) and (14) are equivalent to those of a thin-wall heat exchanger (sensitive). The specific heat at saturation was assumed to be constant in the operation conditions of the tower. Using the NUT-ε, method, the heat transfer flux can be written as follows:

$$\dot{Q} = \xi_a.G.\left(h_{s,wi} - h_{a,i}\right) \tag{16}$$

$$\xi_a = \frac{1-exp\left[-\left(1-C\right).NTU\right]}{1-C.exp\left[-\left(1-C\right).NTU\right]} \tag{17}$$

$$C = \frac{\dot{C}_a}{\dot{C}_w}, \dot{C}_a = G \text{ and } \dot{C}_w = L.\frac{C_{pw}}{C_{ps}}$$

The exit air enthalpy and the outlet water temperature of the tower are determined from a global energy balance for both fluids:

$$h_{a,o} = h_{a,i} + \varepsilon_a\left(h_{s,w,i} - h_{a,i}\right) \tag{18}$$

$$T_{w,o} = \frac{L_i\left(T_{w,i} - T_r\right)C_{Pw} - G\left(h_{a,o} - h_{a,i}\right)}{L_o C_{Pw}} \tag{19}$$

The specific heat mass at saturation is a thermo-physical parameter estimated as the average gradient between the inlet and outlet water conditions. It is given by the following equation:

$$C_{ps} = \frac{h_{s,w,i} - h_{s,w,o}}{T_{w,i} - T_{w,o}} \tag{20}$$

The exit air specific humidity ratio is calculated from Eq. (10):

$$W_{a,o} = W_{s,w,o} + \left(W_{a,i} - W_{s,w,o}\right)exp\left(NTU\right) \tag{21}$$

where $W_{s,w,o}$: is the average specific humidity at saturation.

The average enthalpy at saturation is given by the following relation:

$$h_{s,w,o} = h_{a,i} + \frac{\left(h_{a,o} - h_{a,i}\right)}{1 - exp\left(-NTU\right)} \tag{22}$$

The thermal efficiency of the tower and the outlet wet bulb air temperature are given by the following relations:

$$\varepsilon = \frac{T_{w,i} - T_{w,o}}{T_{w,i} - T_{ah,i}} \tag{23}$$

$$T_{a,o} = \frac{\left(h_{a,o} - W_{a,o} h_{fg}\right)}{\left(C_{pa} + W_{a,o} C_{pv}\right)} \tag{24}$$

The model data are: air mass-flow rate, water mass-flow rate, dry air inlet and wet bulb temperatures, inlet water temperature, inlet air specific humidity. Thus, the thermo-physical constants used are those specific to the two fluids (water and air).

The problem resolution requires an iterative method. A computer calculation code based on data experiences carried out in the laboratory is used for resolving numerically the Eqs. (1-24). The flow chart of the mentioned code is illustrated in Fig. 3 (Appendix A). According to ASHRAE [19], the thermo-physical properties of the mixed water steam and moist air, required in every calculation step.

3. Experimental Study

3.1. Cooling tower presentation

In this study, we have used the experimental device shown schematically in Fig. 4, and the photographically Fig. 5. Essentially, it consists of two major parts; a packed column (1), which represents the most important region where heat and mass exchanges occur between both fluids (air and water), and a base unit. All components are mounted on a robust base with linked instrument panel. These base unit components are; cold water basin (2), a storage tank (3) which contains two electric heaters (12), a water pump (4), a flow meter device (5) a by-pass pipe (6), a water distributor (7), a centrifugal fan (8), an air distribution chamber (9), a drift eliminator (10), A thermostat for measured the temperature of the water in the tank (11). Auxiliaries items are also used such as temperatures and pressures measuring devices (13) (14), as well as system for the regulation of water levels (15) in the feed basin (16). As reported by Lemouari et al [20], it is a cooling tower. The basic characteristics of the cooling tower are summarized at Table 1.

The cooling operation description of the present device is achieved by circuit loop of both used fluids (air and water).

3.1.1. Water circuit loop

The water capacity of the cooling tower system used throughout this study is 3 liters. This quantity of water is heated by means of 0.5 to 1.5 kW electric heats. The “hot” water enters from the top of the tower, $T_{w,i}$ where it is fed into troughs, from which it flows via notches above the packing material in the tower. Those troughs at the top of the tower were designed to uniformly distribute water over the packing with a minimum splashing. The packings encompass an easily wetted surface to spread the water over it and to expose a maximum surface to the air stream.

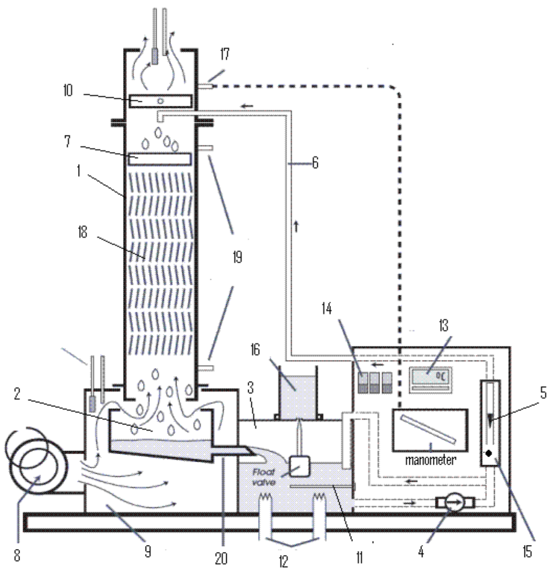

Fig. 4. Cooling tower (Hilton H891) of laboratory [21].

(1) the cooling tower , (2) load tank, (3) water basin, (4) water circulation pump, (5) flow meter, (6) by-pass pipe, (7) water distributor, (8) fan, (9) air distribution chamber, (10) drift eliminator, (11) thermostat, (12) electric heaters, (13) digital temperature indicator, (14) switch of heaters, (15) float valve, (16) make-up tank, (17) connection for orifice differential pressure, (18) packing, (19)connections for pressure Drop Across Packing, (20) control valve.

The water exiting the cooling tower at temperature, $T_{w,o}$ falls firstly from the lowest packing into the collection basin, before being pumped to heaters. The water quantity contained in the cooling system must be maintained in an accumulator called "make-up tank", which is owed to the water evaporation and the supplementary water volume for the system can be determined by the lost water inside the make-up tank. Water mass-flow rate, L is controlled by the control valve on the float-type flow meter with a range of 0 to 50 gram meter per second, as shown in Fig. 4.

Fig. 5. Picture of the experimental device, type Bench P.A. Hilton H891.

3.1.2. Air Circuit

Using a small centrifugal fan with a damper to aspires the ambient air from the atmosphere, and passes it through the cooling tour, air is driven up through the wet packings. Air enters the bottom of the tower and flows past a dry bulb temperature sensor $(T_{db,i})$ and a wet-bulb temperature sensor $(T_{wb,i})$. At the exit of the cooling (at the top) the exit air dry-bulb temperature $(T_{wb,o})$and wet-bulb temperature $(T_{db,o})$are measured. Also, controlling the intake damper leads to regulate the air mass-flow rate, this could be estimated at outlet by a pre calibrated 80 mm-diameter thin-wall orifice.

The measure of the orifice differential (x) in mm H_2O using an inclined tube manometer could determine air mass-flow rate, G, passing through the packing inside the cooling tower [20]. This is given by the following equation [12]:

$$G = 0{,}0137\sqrt{\frac{X}{v_b}} \tag{25}$$

where v_b is the steam and air mixture specific volume of leaving the cooling tower from the top (m^3/kg dry air), evaluated via the formula given in [22]:

$$v_b = (1 + w_b)v_{ab} \tag{26}$$

where w_b is the specific humidity, and v_{ab} is the air specific volume leaving the cooling tower from the top.

Inlet and outlet temperatures of air and water within of the cooling tower were measured carefully with K-type thermocouple sensors having six-point digital temperature indicators. Those thermocouples with uncertainty of ±0.5 ºC were employed to measure the dry and wet bulb air temperatures at the packed column top and bottom, as well as inlet and outlet water temperatures, as point out in Fig. 4.

Table 1. Characteristics of the cooling tower [22].

Characteristics	Value
Dimensions of cooling tower	**150 mm × 150 mm × 600 mm high.**
Energy transferred to water by pump	0.1 kW
Water capacity of system	3 liter
Number of decks of packing	8
Number of plates per deck of packing	10
Total surface area of packing	1.19 m^2
Height of packing	0.48 m
Packing "Density" area/volume	110 m^{-1}

3.2. Experimental method

In the beginning, we have started the cooling tower until reaching the stable conditions as follows:

- Water mass-flow rate: L = 3 kg/min,
- Differential orifice: X = 25 mmH_2O,
- Cooling load = 1.5 kW.

For every time interval of 10 min, we read all temperatures values. The readings are repeated for different air mass-flow rates ranging from 25 to 5 mmH_2O. This test procedure was repeated for 5 min time intervals, with different water mass-flow rates starting 50 down to 10 g/s.

Filling the make-up tank to the gauge mark is carried out at the end of every 5 minutes period with distilled water. The water amount lost by evaporation represents the required compensation water for the selected time interval. The obtained results are shown on the following data Tables (2) and (3).

Table 2. Experimental measurement of the final states of the water and the air under a variable water mass-flow rate; when: cooling load=1.5 kW.

			Inlet temperatures (°C)			Outlet temperatures (°C)		
	Time (min)	L (g/s)	$T_{db,i}$	$T_{wb,i}$	$T_{w,i}$	$T_{wb,o}$	$T_{db,o}$	$T_{w,o}$
X = 25 (mmH$_2$O)	5	45	11.6	8.9	25.8	17.1	18.8	19.5
	10	40	11.9	9.2	27.1	17.2	19	19.7
	15	35	12.2	9.6	28.9	17.2	19.1	20.5
	20	30	12.5	9.9	31.2	17.2	18.8	21.2
	25	25	12.7	10.1	33.5	17	18.9	21.7
	30	20	12.8	10.3	36.1	17.2	19.3	21.5
	35	15	13	10.5	39.7	17.2	19.6	21.4
	40	10	13.1	10.7	46.3	16.6	20	20.8
X = 20 mmH$_2$O)	5	45	13.7	11.1	28.8	20.1	22.4	21.9
	10	40	13.9	11.4	29.3	19.8	22.1	22.1
	15	35	14	11.5	30.8	19.4	22	22.5
	20	30	14.1	11.6	32.7	19.2	21.8	22.8
	25	25	14.1	11.7	35.1	19.1	22.7	23.3
	30	20	14	11.7	37.7	19.3	22.9	23.1
	35	15	14.1	11.8	41.2	19.1	22.2	22.6
	40	10	14.1	11.8	46.7	18.5	22.6	21.7
X = 15 (mmH$_2$O)	5	45	14.7	12.4	30.9	22	24.4	23.7
	10	40	14.7	12.6	31.7	22	24.1	23.7
	15	35	14.8	12.7	32.8	21.8	24	24.2
	20	30	14.9	12.7	34.9	21.6	23.5	24.5
	25	25	15.1	12.8	37.1	21.5	23.5	24.8
	30	20	15	12.7	39.7	21.4	23.8	24.6
	35	15	15	12.7	43.2	21.3	23.9	23.8
	40	10	15	12.8	47	19.9	23.9	22.7
X = 10 (mmH$_2$O)	5	45	15.5	13.2	33.5	25.2	27.2	25.6
	10	40	15.6	13.2	34	25.3	27.7	25.9
	15	35	15.7	13.3	35.3	25	27.7	26.1
	20	30	15.7	13.4	37.4	24.8	27.7	26.3
	25	25	15.7	13.3	39.6	25.5	28.4	26.6
	30	20	15.8	13.3	42.1	25.3	29.1	26.4
	35	15	15.7	13.3	45.4	25.2	29.7	25.6
	40	10	15.7	13.3	47.4	23	28.9	23.9
X = 5 (mmH$_2$O)	5	45	16.5	13.9	37.4	30.6	31.8	28.8
	10	40	16.6	14	38	30.5	32.5	29
	15	35	16.8	14	39.2	30.1	32.6	29.2
	20	30	16.8	14.1	40.8	30.1	32.7	29.4
	25	25	16.8	14	42.6	30.3	33.3	29.1
	30	20	16.7	14	44.7	30.1	34	28.8
	35	15	16.8	14.1	47.4	28.1	34	27.9
	40	10	16.7	14	48.7	26.5	32.4	25.8

4. Results and Discussion

4.1. Effect of water to air mass-flow rate ratio (*L*/*G*) on the performance of cooling tower

The performance of a cooling tower depends on several parameters such as the range of cooling temperature, the inlet water temperatures, (*L*/*G*) variation ratio and wet bulb temperature approach. At given operating conditions, the outlet-water temperature provides a measure of tower capabilities. Right axis of Fig. 6 shows the variation of the outlet water temperature with (*L*/*G*) ratio for different water mass-flow rates. In average 0.4 – 1.5 of (*L*/*G*) ratio the water temperature exhibits slight increase with water mass-flow rate at low (*L*/*G*) ratios. Furthermore, the outlet water temperature increases monotonically with increasing (*L*/*G*) ratio. This is due to the variation of the amount of exchanged energy between the water and the air. At a greater water mass-flow rate (i.e., the ratio (*L*/*G*) increases) the range of cooling tower increases.

The heat absorption effectiveness in terms of flow rate ratios (*L*/*G*) with different inlet water temperatures has been illustrated in the left axis of Fig. 6. This figure shows that the effectiveness of the tower progressively decreases with the increase in the ratio (*L*/*G*) for different inlet water temperatures, whereas the better efficiency was observed with high temperature values. A minimum efficiency was recorded for the rate ratio (*L*/*G*) greater than 1 and lower water temperatures.

Figures 7 and 8 present the variation of the cooling tower effectiveness with the mass-flow rate ratios (*L*/*G*), for different values of the water mass-flow rate, performed at two inlet water temperatures: 30 and 35°C, respectively. The obtained results show that the effectiveness decreases with increasing the mass-flow ratio (*L*/*G*) for all inlet water temperatures. It was seen that higher effectiveness was obtained at lower mass-flow rate ratio (*L*/*G*) less than unity and when (*L*/*G*) is greater than 1, the thermal effectiveness is less than 40%. Values of (*L*/*G*) become noticeable only for higher water mass-flow rate. This result is agreement with the experimental result of Bedekar a et al. [5] and Lemouari and Boumaza. [23].

Table 3. Experimental measurement of the final states of water and air under a variable air mass-flow rate, where: cooling load=1.5 kW.

			Inlet temperatures (°C)			Outlet temperatures (°C)		
L **(g/s)**	**Time** (min)	***X*** (mmH_2O)	$T_{db,i}$	$T_{wb,i}$	$T_{w,i}$	$T_{wb,o}$	$T_{db,o}$	$T_{w,o}$
50	**10**	**25**	15.4	13	29.2	21.1	22.5	22.7
	20	**20**	15.6	13.2	30	22.2	23.5	23.6
	30	**15**	15.8	13.3	31.3	23.6	24.9	24.5
	40	**10**	16	13.6	33.6	26.8	27.5	26.4
	50	**5**	16.6	14	37.4	31.1	31.5	29.6
40	**10**	**25**	15.7	13.1	30.5	20.8	22.6	22.6
	20	**20**	16.1	13.6	31.8	22.2	24.2	23.8
	30	**15**	16.4	13.8	33	23.7	25.7	25
	40	**10**	16.2	13.8	34.9	26.5	28.2	26.4
	50	**5**	16.9	13.9	38	30.7	31.8	28.9
30	**10**	**25**	16.2	13.7	35	21.2	22.8	24.1
	20	**20**	16.4	13.8	35.5	22.1	23.6	24.9
	30	**15**	16.7	13.9	36.5	23.6	25.1	25.6
	40	**10**	16.9	14.1	38.2	26.7	27.7	27
	50	**5**	17.6	14.3	41.1	31	31.8	29.4

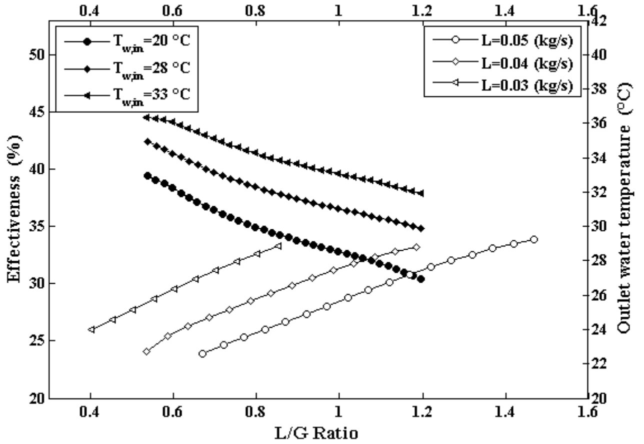

Fig. 6. Variation of outlet water temperature for different water mass-flow rates and cooling tower effectiveness for different inlet water temperature, with different (*L/G*) ratio.

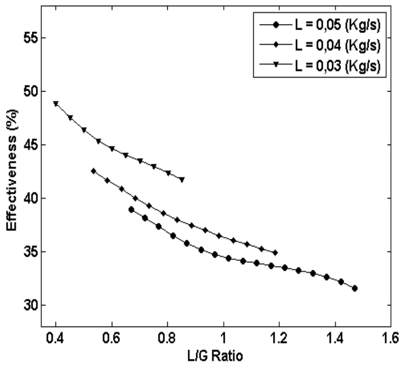

Fig. 7. Variation of cooling tower effectiveness with *L*/*G* ratio for different water mass-flow rates. With inlet water temperature, $T_{w,i}$ = 30 °C.

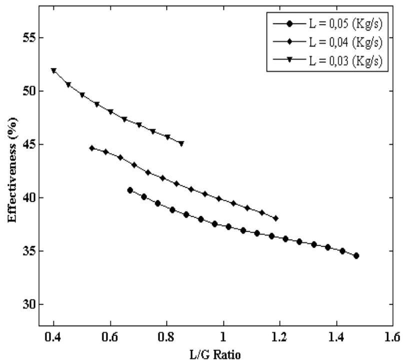

Fig. 8. Variation of cooling tower effectiveness with *L*/*G* ratio for different water mass-flow rates. With inlet water temperature $T_{w,i}$ = 35 °C.

4.2. Effect of air mass-flow rate on the performance of cooling tower

With different water mass-flow rates, the relationship between wet bulb approach and air mass-flow rates are illustrated in Fig. 9. This figure showed that increase in the air mass-flow rate causes a progressive decrease of the wet bulb temperature approach; this is explained by increasing of the air wet bulb temperature. More the air is hot, it can contain more moisture. Consequently, when the temperature is very low, less water quantity evaporates into the air, which reduces the amount of exchanged heat.

Figure 10 shows the influence of air mass-flow rate on the outlet water temperature with different water mass-flow rates. We note that the outlet water temperature decreased with increasing air mass-flow rate. Also the cooling field is reduced significantly when the water mass-flow rate decreases, For instance, as the range of air mass-flow varied from 0.035 to 0.075 kg/s, we found the cooling field varies from (24 °C – 29.1 °C), (22.6 °C – 29.4 °C), with water mass-flow rate quantities of 0.03 kg/s and 0.05 kg/s respectively. It was found that mass-flow rates of air and water have a high influence on the tower water temperature. These factors influence the amount of energy exchanged and the tower cooling capacity.

Variations of cooling tower effectiveness with air mass-flow rate for different values of water mass-flow rate are illustrated in Fig. 11. As can be shown in this figure the cooling tower effectiveness increases with increasing air mass-flow rate and decreases with increasing water mass-flow rate. Therefore, the best cooling tower effectiveness is achieved at the highest air mass-flow rate and of course at the lowest water mass-flow rate.

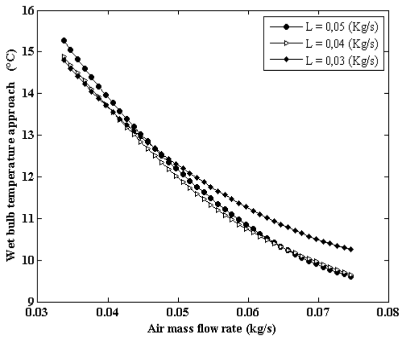

Fig. 9. Variation of wet bulb temperature approach with air mass-flow rate, for different water mass-flow rates.

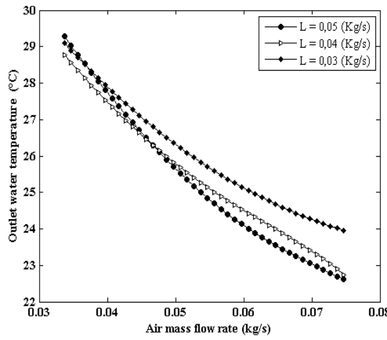

Fig. 10. Variation of outlet water temperature with air mass flow rate for different water mass flow rates.

4.3. Effect of water mass flow rate on the performance of cooling tower

Figures 12 and 13 represent the effect of water mass-flow rate on the performance of cooling tower. As shown in Fig. 12 the cooling tower performance in terms of effectiveness decreases progressively with increasing water mass-flow rates and increases with air mass-flow rates which means that high degree of tower effectiveness corresponds to better cooling performance and higher heat removal. In addition, it can be seen in this figure that when the inlet cooling air has a high mass-flow rate with low water mass-flow rate, the effectiveness of the cooling tower increases. This result is agreement with the experimental results of Bedekar a et al. [5] and Marmouch et al. [11].

Figure 13 shows that the outlet water temperature increases slightly with increasing water mass-flow rates. However, for high values of air mass-flow rates, it can be observed that the outlet water temperature was increased when water mass-flow rates raises from 0,010 kg/s to 0.025 kg/s, but it was decreased slightly when water mass-flow rates were above 0.025 kg/s. This may be due to the decrease in heat and mass transfer coefficients.

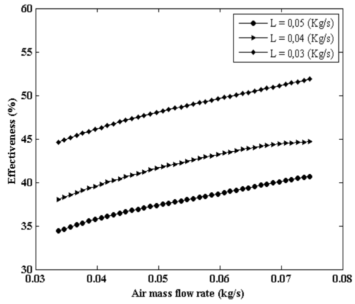

Fig. 11. Variation of cooling tower effectiveness with air mass-flow rate, for different water mass-flow rates.

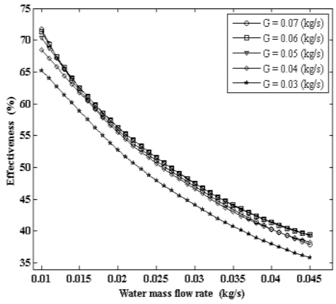

Fig. 12. Variation of cooling tower effectiveness with water mass-flow rate for different air mass-flow rates.

5. Model Validation

The validations of theoretical results against the results of conducted tests in the laboratory are illustrated in Figs 14, 15, 16 and 17. Fig.14 illustrates that the difference between experimental and theoretical results of the outlet water temperatures with different values of inlet air mass-flow rate are almost below 1.2 °C, the maximum and minimum values of the average absolute difference are 0.6 and 0.4 °C, these values are correspondent to 0.04 and 0.07 kg/s, respectively. This figure shows a satisfactory agreement between the theoretical and experimental values with a maximum relative error of 2.46 %, which indicates the accuracy of obtained results.

It is observed from Fig. 17, that the difference between experimental and theoretical results of effectiveness shows an error less than 10 %, with air mass-flow value of 0.05 kg/s. This can be due to the error committed while reading of experiment measurements.

It is noted that all marked errors are included in the range of experimental uncertainty. The uncertainty of experimental data and the approximation of the present theoretical results are considered to have a good agreement between them. This shows that the used mathematical model presents the real situation inside the counter-flow wet cooling tower; this means that the developed program can be used as a tool to optimize thermal performances and most favorable conditions for tower operation.

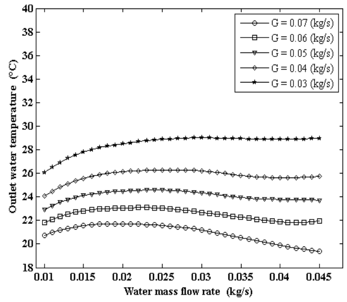

Fig. 13. Variation of outlet water temperature with water mass-flow rate for different air mass-flow rates.

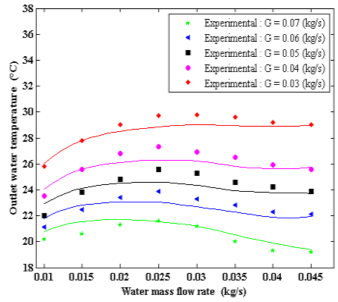

Fig. 14. Comparisons between the experimental and theoretical results of outlet water temperatures with water mass-flow rate, for different inlet air mass-flows rates.

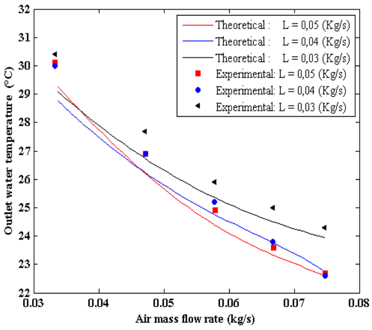

Fig. 15. Comparisons between experimental and theoretical results of outlet water temperatures versus air mass-flow rate, for different inlet water mass-flow rates.

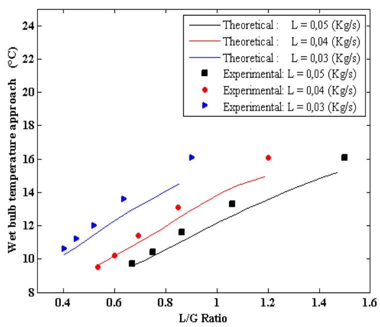

Fig. 16. Comparisons between the experimental and theoretical results of wet bulb temperature approach with air to water rate ratio (*L*/*G*), for different inlet water mass-flows rates.

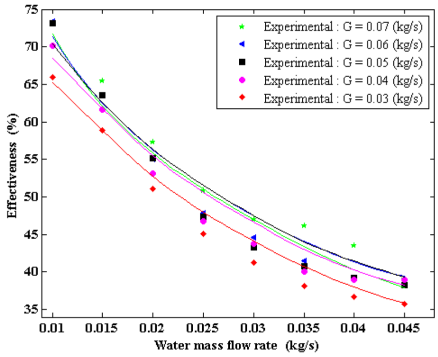

Fig. 17. Comparisons between the experimental and simulated results of effectiveness versus water mass-flow rate, for different inlet air mass-flows rates.

6. Conclusions

In this article, an experimental and theoretical investigation of the thermal performance of heat and mass exchanges within counter-flow wet cooling tower is presented. An exchange phenomenon of heat and mass between two fluids in contact, which are distilled water, and air, has been considered.

A mathematical model has been used to characterize the heat and mass exchange within the tower. This model emphasizes the influence of the input parameters on the output parameters and the tower thermal performance. It can be concluded that the increase in inlet water temperature leads to a considerable increase of the tower efficiency. Also, the tower efficiency decreases as well as the water mass-flow rate increases, while an increase in the air mass-flow rate yields to a considerable improvement of the different performance parameters. The calculation of these parameters allows determining the characteristics of the good tower operation.

The results obtained show that the efficiency and the flow of heat exchanged within the tower increases with the air mass-flow rate increase, and decreases rapidly with considerable cooling loads. The validation of the theoretical results by those obtained from the carried out experiments shows a good agreement.

Nomenclatures

A	Area, m^2
$\dot{C}$	capacity rate, W /K
C_{p_a}	Air specific heat at constant pressure, kJ/kg.°C
C_{p_v}	Vapor specific heat, kJ/kg. °C
C_{p_w}	water specific heat at constant pressure, kJ/kg.°C
C_{p_s}	Specific heat mass at saturation,, kJ/kg.°C
G	Air mass flow rate, kg/s
h_s	Enthalpy of saturated air, J/kg
$h_{s,w}$	Enthalpy air saturation at the water temperature,, J/kg
$h_{f,w}$	Enthalpy of liquid water,, J/kg
h_a	Enthalpy of moist air, J/kg
$h_{g,w}$	Enthalpy of water vapor,, J/kg
h_c	Convective heat transfer coefficient of water,, kW/m ^{2} K
L	Water mass flow rate, kg/s
L_e	Lewis number
NTU	Number of transfer units
$\dot{Q}$	Heat change between air and water, W
$T_{wb,i}$	Wet bulb temperature of the inlet air, °C
$T_{db,i}$	Dry bulb temperature of the inlet air, °C
$T_{w,i}$	Inlet water temperature, °C
$T_{w,o}$	Outlet water temperature, °C
$T_{db,o}$	Dry bulb temperature of the outlet air, °C
$T_{wb,o}$	Wet bulb temperature of the outlet air, °C
w_a	Specific humidity of air, kg$_w$/kg$_a$
w_s	Specific humidity of saturated air, kg$_w$/kg$_a$
$w_{s,w}$	Saturated air humidity ratio at local water temperature, kg$_w$/kg$_a$
ε	Effectiveness
V	volume of tower,, m^3)

Subscripts

a	Air
o	Outlet
i	Inlet
w	water
v	Vapor
s	saturation

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Appendix A

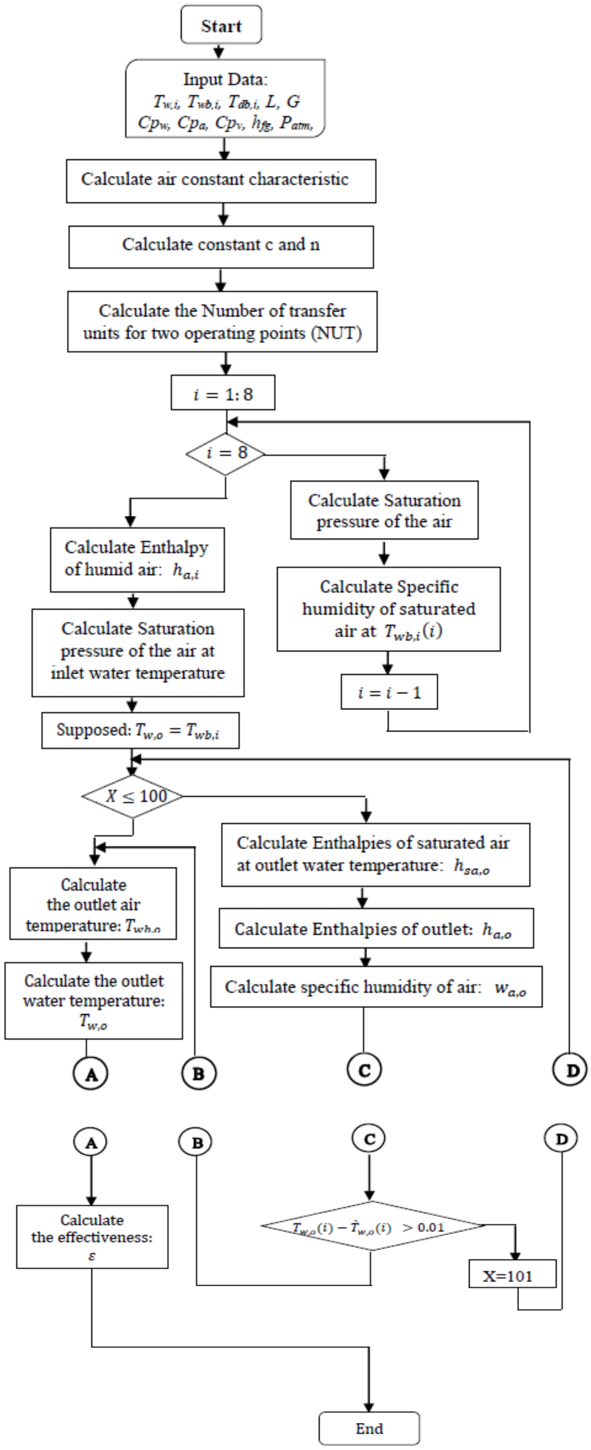

Fig. 3. Flow chart of the program model.