Numerical and Experimental Studies of Heat and Mass Transfer Process Through Packing Zone in a Counter-Flow Wet Cooling Tower

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Abstract—since few decades, cooling towers are used to cool hot water by evaporating a part of water into an air stream. This system consists of three parts; namely: spray, rain zones, and the very important zone called packing zone. This work aims to numerical modelling of heat and mass transfer through packing zone via a computer calculation code based on five ordinary differential equations. The present model is validated using experimental data obtained from a realized prototype of a cooling tower in LEVRES laboratory in which, the maximum error is less than 5%. The obtained results showed that the optimum height and tower effectiveness are affected by the inlet air humidity; in addition, the heat transfer mode in the packing zone is dominated by evaporation.

Index Term-- Heat and mass transfer, cooling tower, modeling, evaporation and packing height.

I. INTRODUCTION

All industrial and manufacturing processes use energy in different forms (mechanical, chemical, electrical, etc.), which is converted into heat. This energy cannot always be recovered or reused, but it is removed from the process and discharged to the environment by cooling. This non-recoverable heat is called residual heat and it may be altered by the proper functioning of 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 used to evacuate the heat of cooling systems to the outside environment (air condition in goret industrial process) by pulverizing hot water in airflow. These water sprays by gravity inside a cool air stream back into the tower. The air stream cools water by evaporation part of sprayed water. Evaporated part of water is sometimes visible as panache above the tower.

There have been several previous studies related to the heat and mass transfer within a wet cooling tower [1]. Among them, we can cite the forced convection between water and air [2], the variation of medium humidity with air temperature [3], and the energy consumed by the evaporation process [4, 5].

The first problem of cooling tower was examined in 1922 by Robinson [6]. Others, Walker et al. [7] was suggested a basic theory of cooling tower operation. They used the ambient air humidity as the sole driving force for the cooling process in cooling towers. However, the most widely used theory for cooling tower calculation is developed by Merkel [8].

A simplified Merkel theory has been used for the analysis of cooling tower performance. Simpson and Sherwood in [9] studied the performances of forced draft cooling towers with a 1.05m packing height consisted of wood slats. Baker and Shyrock [10] tried to minimize the error due to the assumption of Merkel’s theory and they gave a detailed explanation for the Merkel’s method. Kloppers and Kröger [1] analyzed 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.

Otherwise, an experimental study on packings made of PVC is accomplished by Goshayshi et al. [11]. The influence of distance and roughness of packings on heat and mass transfer coefficient and pressure drop was reported in their study. Again, Goshayshi and Missenden [12] 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. Gharagheizi et al. [13] presented an experimental and a comparative study on terms of tower characteristics (KaV/L), water to air rate ratio (L/G) and efficiency for two-film type packings.

They have used a vertical and horizontal corrugated packing’s. Consequently, the authors in [13] showed that the tower with vertical corrugated packaging has higher efficiency than the one with horizontal corrugated packaging. Qureshi and Zubair [14] developed models of three zones; namely, spray zone, packing and rain zones of a basically cooling tower. They used engineering equation solver for modeling and solving simultaneously these zones. It also found that, the error in calculation of the tower volume is 6.5% when the spray and rain zones are neglected. This error is reduced to 3.15% and 2.65%
as the spray and rain zones are incorporated in the model, respectively.

Recently, Khorshidi et al. [15] studied deals with calculation and comparison of performance parameters of three types of horizontal, and mixed corrugated film fill packing, to their actual values in cooling tower of the Bisotun power plant of Kermanshah Province in Iran. A numerical modelling of heat and mass transfer in the counter-flow wet cooling tower fill is presented by Hyhlík [16]. In this work, simplified model based on the set of four ODEs was chosen. Numerical solution of this model was performed using Dormand-Prince method combined with shooting method. Klimanek [17] presented a numerical modelling of natural draft wet-cooling towers. It has developed a complete CFD model capable to predicted cooling tower performance under various operating conditions.

More recently, many researchers have focused on the thermal performance of packings for wet cooling towers, Gao et al. [18] performed a thermal-state model experiment to investigate the influence of filling layout pattern on thermal performance of a wet cooling tower. Various parameters included in this model such as; the cooling temperature difference, cooling efficiency, Merkel number, Lewis number, and ratio of evaporative heat rejection, and obtained one kind of optimal non-uniform pattern. However, these authors did not research the change rules of heat transfer coefficient and total heat rejection of circulating water, which are extremely significant for analysis of the thermal performance of wet cooling towers [19].

Taken into consideration the previous works, the main objective of this paper is to study, numerically and experimentally, the heat and mass exchange within a counter-flow wet cooling tower. A mathematical model is used for determines the optimum height of packing zone, evaporation rate, relative humidity, water and air temperatures distribution along the packing height. Accordingly, investigational testing was carried out to show the influence of mass flow and packing zone height on thermal performance of the tower.

II. MATHEMATICAL MODELING

The packing zone is an essential part of the cooling tower, because it provides a large contact area between the water and air by distributing sprays of water uniformly to enhance evaporation and heat transfer. However, in this study the processes of heat and mass transfer through the packing zone are mathematically modeled. Due to the complexity of two-phase flow within packing zone, one-dimensional model of heat and mass transfer was used. This model consists of combined assumptions of different models, notoriously Merkel[8], Kloppers and Kröger[1], and Klimanek[17], which are:

- Negligible heat and mass transfer through the tower walls to the surroundings.
- The temperature throughout the water steam at each cross-section is uniform.
- The cross-sectional area of the tower is considered uniform.
- The thermal properties of the cooling tower packing material are constants.
- The temperature at the interface between water film and air is supposed to be equal to the mean water temperature.

A. Model description

The coupling between heat and mass transfer processes within a direct counter-flow wet cooling tower was modeled using macroscopic conservation laws. The governing equations of the present model relate five dependent variables, which are inlet water temperature, inlet air temperature, humidity ratio of air, air enthalpy, and air and water mass-flow rates. Fig. 1 represents the schematic of the studied problem.

The surface of the interface is

\[ dA = \psi A \, dz \]  

where \( \psi \) is the area density, and \( A \) is the cross sectional area of the packing, perpendicular to the air and water flow.

The mass and energy balances on a differential portion of the packing perpendicular to the air and water mass-flow within the tower is given by the following equations:

\[ \frac{dW}{dz} = \frac{1}{m_w} \frac{dm_w}{dz} \]  
\[ m_w \frac{dh_w}{dz} = C_{p_r} m_w \frac{dT_w}{dz} + C_{p_r} T_w \frac{dm_w}{dz} \]

where \( h_w \) is the specific enthalpy of the moist air, approximated by a linear distribution of air temperature. It can be evaluated as [20]:

\[ h_w = C_{p_r} T_w + W \left( h_{f_w} + C_{p_r} T_w \right) \]

The mass transfer flow rate from the water as a result of evaporation into the air is estimated by the moisture concentration difference of the air, which is expressed as:

\[ dm_w = \alpha \left( W_{s_w} - W \right) dA \]

where \( \alpha \) is the average mass transfer coefficient and \( W_{s_w} \) is the saturated specific humidity at \( T_w \).

The total heat transferred from the water to the air can be expressed by [17]:

\[ dQ = dQ_{sw} + dQ \]

Where \( dQ_{sw} \) is the heat transmitted from water to air related with mass convection, and given by:

\[ dQ_{sw} = \alpha h_i (W_{s_w} - W) dA \]
where \( h \) is the specific enthalpy of water vapor evaluated at the local water temperature given by [1]:

\[
h = h_f + C_P T_w
\tag{8}
\]

The convection heat transfer is given by the Newton’s law of cooling defined as:

\[
dQ = \beta(T_w - T_a)\,dA
\tag{9}
\]

where \( \beta \) is the average heat transfer coefficient.

The range of temperature in (9) can be replaced by an enthalpy differential. This enthalpy evaluated at the local bulk water temperature is expressed by:

\[
h = C_P T_w + W(h_f + C_P T_w)
\tag{10}
\]

Substitute (10) into (8) can get:

\[
h = C_P T_w + W(h_f + C_P T_w)h
\tag{11}
\]

By subtracting (4) from (11), the resulting equation can be simplified if the weak differences in specific heats are ignored, because are evaluated for each inlet temperatures (water and air) in the packing.

\[
(T_w - T_a) = \frac{(h_f - h_a) - (W-sw - W)}{C_P}\,h
\tag{12}
\]

where \( C_Pw_a \) is the specific heat of air and water vapor mixtures.

By substituting (7) and (9) into the total rejected heat (6), we can obtain:

\[
dQ = \beta(T_w - T_a) + \alpha h_a (W-sw - W)\,dA
\tag{13}
\]

Also, by substituting equation (12) into (9). After, we substitute both equations (the resulting equation and equation (7)) into (5), after simplifying we find:

\[
dQ = \alpha \left[ \frac{\beta}{C_Pw_a} (h_f - h_a) - \left( \frac{1 - \beta}{C_Pw_a} \right) h_a \right] (W-sw - W)\,dA
\tag{14}
\]

By definition, the Lewis factor \( L_e = \beta / \alpha \left( C_Pw_a + WC_P \right) \) is a measure parameter that describes the ratio of heat and mass transfer coefficients in an evaporative process. For unsaturated air-water vapor systems, Bosnjakovic in [21] proposed the following empirical relation:

\[
L_e = 0.865^{3/5} \left[ \frac{(W_{sw} + 0.622)/(W + 0.622) - 1}{\ln[(W_{sw} + 0.622)/(W + 0.622)]} \right]
\tag{15}
\]

The enthalpy transfer to the air stream from (14) is:

\[
\frac{dh}{dz} = \frac{1}{m_a} \frac{dQ}{dz} = \frac{h_f W_a}{m_a} \left[ L_e (h_f - h_a) + (1 - L_e) \right] \frac{dh_a}{(W_sw - W)}
\tag{16}
\]

By introducing (1) into (4), we can conclude that the driving potential for mass transfer takes the form:

\[
\frac{dm}{dz} = \alpha (W_sw - W) \psi A_c
\tag{17}
\]

Substituting (13) into mass balance equation (2) yields:

\[
\frac{dW}{dz} = \alpha \frac{W}{m_a} (W_sw - W) \psi A_c
\tag{18}
\]

The total rejected heat from the air/water interface must be equal to the enthalpy variation of water and air, it is represented by the equation (3). By equalizing the enthalpy variation with the total heat transfer rate of the (13), we can get:

\[
\frac{dh}{dz} = \psi A \left[ \beta(T_w - T_a) + \alpha h_a (W_sw - W) \right]
\tag{19}
\]

Substituting the specific enthalpy of water vapor (8) into (19) yields:

\[
\frac{dh}{dz} = \psi A \left[ \beta(T_w - T_a) + \alpha h_f + C_P T_w (W_sw - W) \right]
\tag{20}
\]

\[
\frac{dh}{dz} = \left( C_P + W C_P \right) \frac{dT}{dz} + \left( h_f + C_P T_w \right) \frac{dW}{dz}
\tag{21}
\]

Combining (21), (20) and after introduction of the Lewis number definition we can get:

\[
\frac{dT}{dz} = \frac{\psi A \left[ L_e(T_w - T_a) (C_P + C_P T_w) + \left( C_P + C_P T_w - C_P^o T_w \right) (W_sw - W) \right]}{m_a (C_P + C_P T_w)}
\tag{22}
\]

Using equation (3) after substitution of (17) and (20), we derive the equation of water temperature variation in the packing as follows:

\[
\frac{dT}{dz} = \frac{\psi A \left[ L_e(T_w - T_a) (C_P + C_P T_w) + \left( h_f + C_P T_w - C_P^o T_w \right) (W_sw - W) \right]}{m_a (C_P + C_P T_w)}
\tag{23}
\]

Equations (16), (17), (18), (22) and (23) consist of a system of five ordinary differential equations with explicit form in the derivatives of the unknown dependent variables. These ODEs are used to determine the processes of heat and mass transfer between the falling water droplets and the ascending air in the packing area. It can be summarized in the following differential equations system:

\[
\frac{dm}{dz} = \alpha \psi A_c (W_sw - W)
\]

\[
\frac{dW}{dz} = \alpha \frac{W}{m_a} (W_sw - W)
\]

\[
\frac{dT}{dz} = \frac{\psi A \left[ L_e(T_w - T_a) (C_P + C_P T_w) + \left( C_P + C_P T_w - C_P^o T_w \right) (W_sw - W) \right]}{m_a (C_P + C_P T_w)}
\]

\[
\frac{dh}{dz} = \alpha \frac{h_f}{m_a} (W_sw - W)
\]

Numerical resolution of these ODEs represents the boundary value problem, processing two boundary conditions specified at two points (TPBVP), i.e. consists of the differential equations with conditions specified on both sides (bottom and top of the packing).

The five boundary conditions required for those ODEs comprise the initial value of dependent variables such as the specific humidity, the dry air temperature, the air enthalpy, the water temperature and the mass flow rate. Some values of
dependent variables \( m_{x}(z), h_{x}(z), W(z), T_{x}(z), T_{e}(z) \) and its derivatives are specified at the original spatial value where \( z = 0 \), the others are specified at the final spatial value, i.e. height \( z = H \), which can be summarized by the following expression as:

\[
\begin{align*}
  h_{x}(z = 0) &= h_{x,i} \\
  W(z = 0) &= W_{i} \\
  T_{a}(z = 0) &= T_{a,i} \\
  T_{e}(z = H) &= T_{e,i} \\
  m_{x}(z = H) &= m_{x,i}
\end{align*}
\]

(25)

There are unknown parameters in these ODEs (24), such as \( \psi, \alpha \) and \( A \). These parameters have to be determined by using experimentally obtained characteristics of the packing. The Merkel’s model can be used to determine iteratively those parameters. It is given by the following relation [16]

\[
M_{ek} = \int_{z=0}^{H} \frac{\alpha \psi A_{e}}{m_{w}} dz
\]

(26)

The initial estimate for the iterative technique can be

\[
(\alpha \psi A_{e})_{0} = M_{ek} \frac{m_{x,i}}{H}
\]

(27)

where \( M_{ek} \) is the Merkel number identified from experiments. The ODEs are solved with the initial guess value of \( (\alpha \psi A_{e})_{0} \), where the left-hand side of equation (26) can be calculated by differentiating the integrated set in the sense of the following [22]:

\[
\frac{dM_{ek}}{dz} = \frac{\alpha \psi A_{e}}{m_{w}}
\]

(28)

The outlet water mass-flow rate can be evaluated iteratively by the following experimental relation

\[
m_{w,o} = m_{x,i} - m_{e} (W_{w} - W_{i})
\]

(29)

The differential equations system (24) above describes the heat and mass exchange processes, is valid only in the case of unsaturated air. Assuming that the air exiting from the tower cannot be saturated before leaving the packing section.

A computer program code written and compiled in the MATLAB environment, based on tower packing characteristics (shown in Table 1), and the data experiences carried out in the laboratory, is used for resolving numerically the ODEs (24). The thermo-physical properties of the mixed water, steam and moist air, required in every calculation step is taken from ASHRAE [23]. The initial values of the specific humidity and air enthalpy are obtained by the Psychrometrics chart of moist air, according to the values of dry and wet bulb air temperatures [24]. Three numerical studied cases are presented in Table 2. They show particular situations that occur when air and water temperatures and mass-flows rates change. The experimental measurement was carried out in a prototype of a cooling tower with a packing height of \( H = 0.60 \, m \) and an area of \( A_{c} = 1.19 \, m^2 \).

### III. EXPERIMENTAL APPROACH

The experimental design used in this study is shown in fig.2, and the photograph of the apparatus is shown in Fig 3. It consists mainly of both parties, a packed column \((1)\), which represents the main region of heat and mass exchange between two fluids (air and water), and a base unit.

![Fig. 2. Cooling tower (Hilton H891) of laboratory [25].](image-url)


All components mounted on a robust base with connected instrument panel. The base unit components consist of a 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 \((16)\). This cooling tower, as reported by Lemouari et al [26]. The basic characteristics of the cooling tower are summarized at the table I.
The cooling mode of this tower is reflected by circuit loop description of both used fluids water and air.

A. Water circuit

The water capacity of the cooling tower system used in this test is three liter. Water is heated using 0.5 to 1.5 kW electric heats. The heated water enters the top of the tower where its temperature ($T_{w,o}$) is measured and it is dispersed uniformly over the volume of the tower through the top packing. As the water spreads over the plates under gravity with minimum splashing, a large thin film of water is exposed to air stream. During its descending passage through the packing, water is cooled, mainly by evaporation of a small part of the total flow.

The collected cooled water in the basin is returned to the load tank, where its temperature ($T_{w,u}$) is measured again and where it is re-heated before re-circulation. Due to an amount of evaporated water, an accumulator or “make-up tank” is provided that is connected to the load tank via a float operated indicator valve, which must maintain the quantity of water in the cooling system. The volume of water added to the system can be measured by the loss of water in the make-up tank.

The water mass-flow rate, $m_w$, circulating in the cooling tower was measured by a variable area flow meter with a range of 0 to 50 gram meter per second, as shown in Fig 2.

B. Air circuit

The fan aspires the ambient air from the atmosphere, and pass it through the cooling tower. On top of the packed column, a droplet arrester catches the most droplets contained in wet air. The air mass-flow rate is adjusted by controlling the intake damper setting. This parameter can be estimated in outlet by a previously calibrated thin-wall orifice of 80 mm of diameter.

Air mass-flow rate, $m_a$, passing through the packing within the cooling tower was determined by measuring orifice differential ($X$) in $mmH_2O$ with an inclined tube manometer [26]. The air mass-flow rate through the orifice is given by the following equation[27]:

$$m_a = 0.0137 \sqrt{\frac{X}{V_b}} \quad (30)$$

where $V_b$ is the specific volume of steam and air mixture leaving top of the cooling tower (m$^3$/kg dry air), is evaluated by using the formula given in[28]

$$V_b = (1 + w_a) \nu_{ab} \quad (31)$$

where $w_a$ is the specific humidity, and $\nu_{ab}$ specific volume of air leaving top of cooling tower.

All temperatures of water and air (entering and leaving) inside of the cooling tower were measured with six point’s digital temperature indicator of type-K thermocouple sensors. Thermocouples with the uncertainty of ±0.5°C employed to measure the dry and wet bulb temperatures at the base ($T_{db,i}$ and $T_{db,o}$) and at the top of the packed column ($T_{db,o}$ and $T_{db,a}$), and the inlet and outlet water temperatures ($T_{w,i}$ and $T_{w,o}$) respectively as indicated in Fig 2.

C. Water circuit

Initially, we started the experimental devise until reaching the following stable conditions: Differential orifice: $X = 25 \ mmH_2O$, cooling load =1.5 KW and water mass-flow rate $m_w = 50 \ g/s$.

For every time interval of 600 s, we take the readings of all temperatures. We repeat the readings with different air mass-flow rates from 25 to 5 $mmH_2O$. This test procedure was repeated for every time interval of 300s, with different water mass-flow rates from 50 to 10 g/s.

At the end of every 300s-period, we fill the make-up tank to the gauge mark with distilled water. The amount of water lost (by evaporation) represents the necessary compensating water for the chosen time interval. The results obtained are indicated on the following measure table II.

### TABLE I

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dimensions of cooling tower</td>
<td>50mm x 150mm x 600 mm high.</td>
</tr>
<tr>
<td>Energy Transferred to Water by Pump</td>
<td>0.1 KW</td>
</tr>
<tr>
<td>Water Capacity of System</td>
<td>3 liter</td>
</tr>
<tr>
<td>Number of Decks of packing</td>
<td>8</td>
</tr>
<tr>
<td>Number of Plates per Deck of packing</td>
<td>10</td>
</tr>
<tr>
<td>Total surface Area of Packing</td>
<td>1.19 m$^2$</td>
</tr>
</tbody>
</table>

IV. RESULTS AND DISCUSSION

In this section, simulation results of heat and mass transfer inside packing zone are carried out to validate the proposed mathematical model. Figures 4 and 5 show the water and air temperatures distribution along the packing height for different inlet air mass-flow rates with a constant value of water mass-flow rate of 0.050 kg/s. As shown in Fig. 4, the outlet water temperature decreases while increasing in air mass-flow rate.
Moreover, it can be seen that the distribution profile of water temperature progressively decreased from the inlet to exit packing zone, which occurs as a result of continuous evaporation from water to air since the evaporation process is dominant in convective heat transfer. Also, we can see in Fig. 5 that the air temperature slightly increases from the bottom to the top of the packing zone. This increase is accompanied by a significant rise in air moisture content in the same part of the packing zone.

![Fig. 5. Distribution of air temperature along the packing height for different inlet air mass-flow rates.](Image)

The distribution of specific humidity along the packing height for different water to air mass-flow ratio is illustrated in Fig. 6. Obviously, the increase of air mass-flow rate leads to the reduction of outlet dry bulb temperature. Moreover, it can be seen that the relationship between outlet dry bulb air temperature and water to air mass-flow ratio is nearly linear, and that the water to air mass-flow ratio has the same effect on the variation of air specific humidity; but this effect is more obvious when the value of water to air mass-flow ratio is greater than 0.86. As a result, the evaporation process inside the packing zone is improved by the mean of increasing air mass-flow rate.

Fig. 7 shows a proportional relationship between the air enthalpy and water to air rate ratios through the packing height. The increase of air enthalpy is a measure of the total heat content of air based on air’s temperature and its humidity.

![Fig. 4. Distribution of water temperature along the packing height for different inlet water temperatures.](Image)

The driving potential of evaporative heat transfer \( (W_{wu} - W) \) along the packing height for different water to air rate ratios is illustrated in Fig. 8. We note that the potential evaporation progressively decreases along the packing height with increasing of water to air ratios, from the bottom to the top of packing, this decreases is clearly noticeable when the value of water to air ratio is greater than one. This explains that, the evaporation process decreases as the air moves from the bottom to the top of the tower. In addition, we can see in Fig. 9 a linear inversion relationship between the evaporation potential and water temperature, which means that the evaporation mechanism is decelerated by the increase of the inlet water temperature, and accelerated by increases on water to air ratio.

### TABLE II
COMPARISON OF NUMERICAL RESULTS AND EXPERIMENTAL MEASUREMENTS OF THE FINAL STATES OF WATER AND AIR UNDER A VARIABLE AIR AND WATER MASS-FLOW RATES.

<table>
<thead>
<tr>
<th>Case</th>
<th>( m_w ) Kg/s</th>
<th>( m_{w,1} ) Kg/s</th>
<th>( h_{iw} ) (KJ/kg)</th>
<th>( W_i ) (Kg/kg)</th>
<th>Inlet temperature(°C)</th>
<th>Outlet temperature(°C)</th>
<th>Error %</th>
</tr>
</thead>
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<tr>
<td></td>
<td></td>
<td></td>
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<td></td>
<td>( T_{db,i} )</td>
<td>( T_{wb,i} )</td>
<td>( T_{w,i} )</td>
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<td>01</td>
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<td>0.045</td>
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<td>0.00600</td>
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<td>26.1</td>
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<td>0.030</td>
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<td>0.00879</td>
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</tr>
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<td>0.050</td>
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Based on the experimental measurement, we found that water loss by evaporation along the packing height is very small in comparison with water flow rate ($m_w$). Fig. 10 shows that the amount evaporated water through the packing zone is slightly decreased from the top to the bottom in the packing zone, and that this amount is increased while increasing water to air mass-flow rate ratio.

Figs. 11 and 12 show the distribution of humidity and potential evaporation inside the packing zone with height, and water flow rate, respectively. Clearly, the specific humidity is less than the evaporation potential until the height of 0.40 m, and becomes greater than the evaporation potential after this value. The intersection between the two curves represents the saturation point of air and optimum performances of cooling tower. In Fig. 12, a linear variation is observed between humidity and water mass flow rate from entry to exit of tower. In addition, it is clear seen that there is an inversed relationship between the potential evaporation and the flow rate of water. Water mass flow losses by evaporation inside the packing zone can be evaluated as the difference between the inlet and outlet flow rate values (0.050 to 0.0492 kg/s for the present study).
Results given in Table II show that the relative error between numerical and experimental measurements is less than 5% in term of outlet temperatures for the three cases. Hence, there is a good agreement between numerical and experimental results. Moreover, it is noted that the produced errors may be due to experimental measurements reading. Consequently, the proposed mathematical model can effectively represent the real situation inside the packing zone in a counter-flow wet cooling tower.

This means that the developed calculation program can be used as an effective tool to optimize thermal performances and most favorable conditions for tower operation.

V. CONCLUSION

In this paper, numerical and experimental studies of heat and mass transfer process through the packing zone in a counter-flow wet cooling tower have been investigated. Under this context, we have proposed a mathematical model that characterizes the heat and mass transfer along the packing height using five ordinary differential equations. This model emphasizes the influence of the input parameters on the output parameters and the tower thermal performance. Accordingly, it can be used to determine the cooling tower optimum height and to estimate the amount of water evaporated rate, humidity, distribution water and air temperatures, as well as water mass flow rate along the packing height.

The obtained results showed that the optimum height and tower effectiveness are affected by the inlet air humidity; in addition, the heat transfer mode in the packing zone is dominated by evaporation. Numerical results are validated using experimental data obtained from a realized prototype of a cooling tower in LEVRES laboratory, for which the maximum error is less than 5%.

**NOMENCLATURE**

\( A \) \quad \text{The cross sectional area of the packing } \ m^2

\( \varrho \) \quad \text{The area density}

\( h_a \) \quad \text{The specific enthalpy of the moist air}

\( h_c \) \quad \text{Enthalpy at the local bulk water temperature } J

\( C_{p_{wa}} \) \quad \text{Specific heat capacity of dry air } \ J/\text{kg.}{ }^\circ\text{C}

\( C_{p_v} \) \quad \text{Specific heat capacity of water vapor } \ J/\text{kg.}{ }^\circ\text{C}

\( h_l \) \quad \text{latent heat of vaporization } \ J/\text{kg}

\( C_{p_{wua}} \) \quad \text{Specific heat of water vapor mixture } \ J/\text{kg.}{ }^\circ\text{C}

\( \alpha \) \quad \text{The average mass transfer coefficient}

\( W \) \quad \text{Specific humidity } \text{kg/ kg DB}

\( \Delta W_e \) \quad \text{Evaporation potential } \text{kg/ kg Dry air}

\( W_{sv} \) \quad \text{Saturated specific humidity at } T_w, \ \text{kg/ kg Dry air}

\( m_{w,i} \) \quad \text{Inlet water mass-flow rate } \text{Kg/ s}

\( m_{a} \) \quad \text{Inlet air mass-flow rate } \text{Kg/ s}

\( h_i \) \quad \text{specific enthalpy of water vapor } J

\( M_{ek} \) \quad \text{Merkel number}

\( \nu_{ab} \) \quad \text{specific volume of air leaving top of cooling tower}

\( \beta \) \quad \text{the average heat transfer coefficient}

\( Q \) \quad \text{Total heat rejected } W

\( L_e \) \quad \text{Lewis number}

\( T_{db,i} \) \quad \text{Dry bulb temperature of the inlet air } (\text{°C})

\( T_{wb,i} \) \quad \text{Wet bulb temperature of the inlet air } (\text{°C})

\( T_{w,i} \) \quad \text{Inlet water temperature } (\text{°C})

\( T_{o,w} \) \quad \text{Outlet water temperature } (\text{°C})

\( T_{db,o} \) \quad \text{Dry bulb temperature of the outlet air } (\text{°C})

\( T_{wb,o} \) \quad \text{Wet bulb temperature of the outlet air } (\text{°C})

**Subscripts**

\( i \) \quad \text{inlet}

\( o \) \quad \text{outlet}

\( w \) \quad \text{water}

\( a \) \quad \text{air}

\( v \) \quad \text{Vapor}

\( s \) \quad \text{saturation}
REFERENCES


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