Heat and Mass Transfer Evaluation and Exergy Analysis in the Counter Flow Wet Cooling Tower of Khuzestan Steel Company (KSC)

Navid Bozorgan*
Department of Mechanical Engineering,
Abadan Branch, Islamic Azad University, Abadan, Iran
E-mail: N.Bozorgan@gmail.com
*Corresponding author

Nariman Bozorgan
Department of Mechanical Engineering,
Abadan Branch, Islamic Azad University, Abadan, Iran
E-mail: N.Bozorgan@yahoo.com

Received: 8 September 2011, Revised: 18 December 2011, Accepted: 26 February 2012

Abstract: A thermodynamic analysis of the counter flow wet cooling tower (CWCT) in the Steelmaking unit of the Khuzestan Steel Company (KSC) is performed in this paper. We evaluated both energy and exergy formulations for analyzing heat and mass transfer, exergy and second-law efficiency in this cooling tower. The Lewis factor Le_f is an indication of the relative rates of heat and mass transfer in an evaporative process. In many performance analyses of the cooling towers, Le_f is assumed to be 1 in order to simplify the heat and mass transfer equations between air and water. If so, evaporative loss is negligible. Taking Le_f into account complicates heat and mass transfer equations and their numerical solutions, despite this fact, in this paper, Le_f is calculated for all parts of the tower in order to predict air and water conditions in the cooling tower more accurately.

Keywords: Cooling Tower, Exergy, Heat and Mass Transfer, Lewis Factor, Second-law Efficiency

Reference: Bozorgan, Navid and Bozorgan, Nariman, 'Heat and Mass Transfer evaluation and Exergy analysis in the Counter Flow Wet Cooling Tower of Khuzestan Steel Company (KSC)', Int J of Advanced Design and Manufacturing Technology, Vol. 5/No. 3, 2012, pp. 83-92 .

Biographical notes: Navid Bozorgan received his M.Sc. in mechanical engineering from Islamic Azad University, Dezfoul branch, Iran 2009. He is currently instructor at the Islamic Azad University, Abadan branch, Abadan, Iran. **Nariman Bozorgan** is mechanical engineer at Iranian Drilling Services Company (IDSC). He is a member of young researchers club, Islamic Azad University of Ahvaz.

1 INTRODUCTION

Cooling towers are enclosed vessels, absorbing large heat loads from industrial processes such as power generation units, chemical and petrochemical plants and refrigeration and air-conditioning system and dissipating this heat to the atmosphere. Heat rejection in cooling towers is accomplished by heat and mass transfer between hot water droplets and ambient air. In the present paper, the performance of one of the counter flow wet cooling towers of Khuzestan Steel Co. is analyzed in order to improve the function of this unit and its effect on the related units. Since energy analysis emphasizes only on the quality of energy, it cannot fully define the performance of the cooling tower. Therefore, in order to optimize the performance of the tower and also the overall process in which cooling is a part of it, it is necessary to analyze energy and exergy of this cooling tower simultaneously.

As a result, heat and mass transfer between ambient air and hot water, exergy and second-law efficiency in this cooling tower was evaluated in order to analyze its heat transfer performance. Examination of air and water conditions in exergy analysis of cooling tower is of high importance. Khan et al. [1] examined the operation of cooling tower with counter flow and they used the thermodynamic property equations that are formulated by Hyland and Wexlar [2], [3] in their analysis of heat and mass transfer mechanism in cooling tower.

Bahaidrah [4] examined the operation of cooling towers consideration water and air conditions throughout the tower. EI-Dessouky et al. [5] proposed a model for heat analysis of counter flow wet cooling tower in a stable state. Nimr [6] presented a mathematical model to describe the thermal behavior of packed cooling towers. A closed form solution was obtained for both the transient and steady temperature distribution in a cooling tower. Khan et al. [7] investigated the heat and mass transfer mechanism and performance characteristics using a detailed theoretical model in counter flow cooling towers.

In this model, an approximate equation was used to calculate wet air enthalpy. This equation was obtained from the thermodynamic properties of saturated air-water vapor mixture. To obtain proper results calculating accurately the properties of wet air appears to be essential. This paper, presents a mathematical model by which heat and mass transfer equations throughout the tower and numerical solutions of these equations, and also properties of wet air such as humidity ratio, enthalpy and temperature are precisely computed.

2 MATHEMATICAL MODELING

Currently in counter cooling towers, air motion, as shown schematically in Fig. 1, is due to the natural chimney effect of the warm moist air in the tower or it may be caused by fans at the bottom (forced draft) or at the top (induced draft) of the tower.

Using Poppe heat and mass transfer mathematical model [8] and its numerical solution the internal air and water conditions in the cooling tower can be analyzed. Calculating wet air property is important to obtain proper results in analyzing exergy. Therefore, humidity ratio, enthalpy and air temperature in different points of the cooling tower are calculated through analyzing heat and mass transfer equations at the screening surface between air and water. The assumptions used in analysis of air and water conditions in the counter flow wet cooling tower are as follows:

- Heat and mass transfer from tower walls to external environment is negligible.
- Heat transfer from fan of the tower to air or water vapor is negligible.
- Water and dry air specific heat are constant.
- The water temperature at each cross section is uniform.
- Cross sectional area of the tower is uniform.
- Discharged water drops from the tower caused by outlet air stream are negligible.

The enthalpy of the air-water vapor mixture per unit mass of dry air, is expressed by

$$i_{ma} = c_{pa}T_a + \omega(i_{fgwo} + c_{pv}T_a) \quad (1)$$

Knowing the humidity ratio and temperature of the cooling tower, the enthalpy of the inlet air can be calculated according to Eq. (1). In this paper in order to determine the enthalpy, humidity ratio and air temperature in different places of the cooling tower, variations in air humidity relative to water temperature and enthalpy differential of the air-water vapor mixture to water temperature are calculated assessing heat transfer rate at the interface between air and water. At the outset, heat exchange between air and water in the cooling tower as depicted in Fig. 2 is the total sum of heat transfer due to mass transfer and convective heat transfer rate due to differential temperature between air and water. At first, we will calculate heat transfer between air and water resulting from mass transfer.

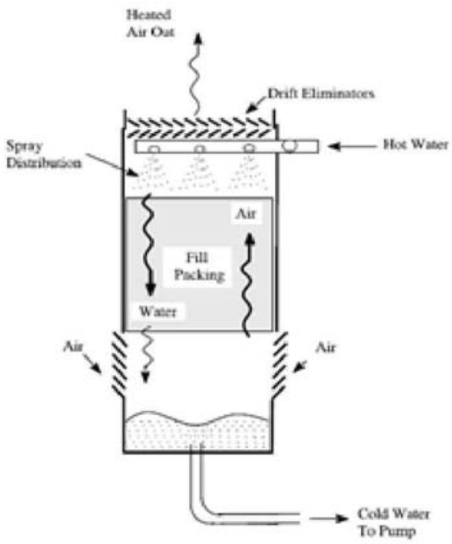

Fig. 1 Schematic of a wet counter flow cooling tower.

Mass transfer at the interface between air and water is formulated as follows [9]:

$$dm_w = h_d(\omega_{sw} - \omega)dA \tag{2}$$

Knowing water vapor pressure at the dew point, we can calculate humidity ratio of saturated air at water temperature.

$$\omega_{sw} = 0.622P_g / (P - P_g) \tag{3}$$

$$p_g = e^{18.6 - \frac{5206.9}{T_w}} \tag{4}$$

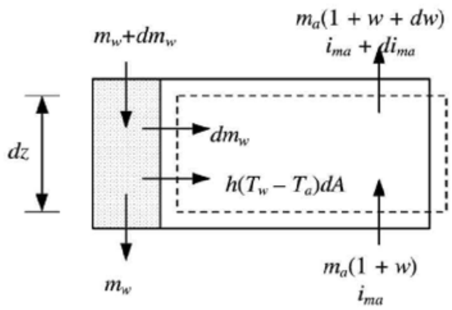

Fig. 2 Air side control volume of the fill [9].

Considering Fig. 2, rate of heat transfer resulted from mass transfer is given by

$$dQ_m = i_v dm_w = i_v h_d(\omega_{sw} - \omega)dA \tag{5}$$

If the latent heat of water is i_{fgwo}=2500 kJ/Kg_w, then the enthalpy of water vapor, is given by

$$i_v = i_{fgwo} + c_{pv}T_w \tag{6}$$

Convective heat transfer originated from temperature difference between air and water can be expressed as follows:

$$dQ_c = h_c(T_w - T_a)dA \tag{7}$$

In Eq. (7), temperature differential between air and water is the most influential parameter in the convective heat transfer. By considering the equations of enthalpy of the saturated air and enthalpy of the air-water vapor mixture, the temperature differential is calculated.

Knowing the temperature of water, we can obtain enthalpy of saturated air.

$$i_{masw} = c_{pa}T_w + \omega_{sw}(i_{fgwo} + c_{pv}T_w) \tag{8}$$

By considering Eqs. (1) and (8), temperature differential between air and water in the cooling tower is given by:

$$T_w - T_a = \frac{(i_{masw} - i_{ma}) - (\omega_{sw} - \omega)i_v}{c_{pma}} \tag{9}$$

In Eq. (9) specific heat of air-water vapor mixture is expressed by:

$$c_{pma} = (c_{pa} + \omega c_{pv})kJ\ /\ Kkg\ dry\ air \tag{10}$$

By substitution of Eq. (9) in Eq. (7), the following equation will be obtained:

$$dQ_c = h_c \frac{(i_{masw} - i_{ma}) - (\omega_{sw} - \omega)i_v}{c_{pma}} dA \tag{11}$$

Therefore, total heat transfer in the cooling tower is as follows:

$$\begin{aligned} dQ = dQ_m + dQ_c = h_d[&\frac{h_c}{c_{pma}h_d}(i_{masw} - i_{ma}) \\ &+(1 - \frac{h_c}{c_{pma}h_d})i_v(\omega_{sw} - \omega)]dA \end{aligned} \tag{12}$$

$h_c/c_{pma}\ h_d$ in Eq. (12) is known as the Lewis factor Le_f and is an indication of the relative rates of heat and mass transfer in an evaporative process. Bosnjakovic [10] proposed the Lewis factor Le_f for air-water vapor systems, therefore:

$$Le_f = 0.865^{0.667}(\frac{\omega_{sw} + 0.622}{\omega + 0.622} - 1) / [\ln(\frac{\omega_{sw} + 0.622}{\omega + 0.622})] \tag{13}$$

By considering Le_f and heat transfer rate between air and water in the cooling tower, the enthalpy transfer to the air system is given by:

$$di_{ma} = \frac{1}{m_a} dQ = \frac{h_d dA}{m_a}[Le_f(i_{masw} - i_{ma}) + (1 - Le_f) i_v(\omega_{sw} - \omega)] \quad (14)$$

When the enthalpy transfer in air stream is obtained through the examination of heat exchange between air and water in the cooling tower, we attempt to obtain variations of humidity ratio of air and variations enthalpy of air-water vapor mixture to water temperature by which we can obtain humidity ratio and enthalpy of air in the cooling tower. A mass balance for the control volume in Fig. 3 yields

$$dm_w = m_a d\omega \quad (15)$$

The energy balance for the control volume in Fig. 3 is as follows:

$$m_a di_{ma} - m_w di_w - i_w dm_w = 0 \quad (16)$$

where, di_w in Eq. (16) is:

$$di_w = c_{pw} dT_w \quad (17)$$

Therefore, the temperature differential of water can be calculated considering Eqs. (15-17) as follows:

$$dT_w = \frac{m_a}{m_w}\left(\frac{1}{c_{pw}} di_{ma} - T_w d\omega\right) \quad (18)$$

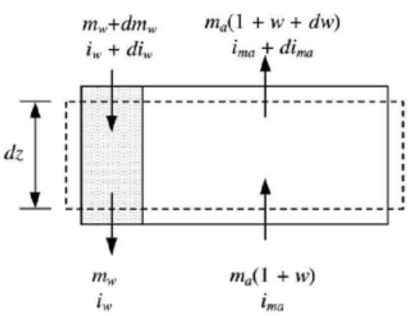

Fig.3 Control volume of counter flow fill [9].

Also, by dividing both sides of Eq. (18) by $T_w dT_w$, after rearrangement it is found:

$$\frac{d\omega}{dT_w} = \frac{1}{c_{pw}T_w}\frac{di_{ma}}{dT_w} - \frac{1}{T_w}\frac{m_w}{m_a} \text{ or}$$
$$\frac{d\omega}{dT_w} = \frac{di_{ma}}{T_w di_w} - \frac{1}{T_w}\frac{m_w}{m_a} \quad (19)$$

Substitute Eqs. (2) and (14) in Eq. (16), after rearrangement it is found:

$$m_w di_w = h_d dA[i_{masw} - i_{ma} + (Le_f - 1)[i_{masw} - i_{ma} - (\omega_{sw} - \omega)i_v] - (\omega_{sw} - \omega)c_{pw}T_w] \quad (20)$$

Variations of humidity ratio of the air in relation to temperature of the water results from substitution of Eqs. (14) and (20) in Eq. (19) as follows:

$$\frac{d\omega}{dT_w} = \frac{c_{pw}\frac{m_w}{m_a}(\omega_{sw} - \omega)}{i_{masw} - i_{ma} + (Le_f - 1)[i_{masw} - i_{ma} - (\omega_{sw} - \omega)i_v] - (\omega_{sw} - \omega)c_{pw}T_w} \quad (21)$$

Substitution of Eq. (21) in Eq. (19) results in:

$$\frac{di_{ma}}{dT_w} = \frac{m_w c_{pw}}{m_a}\left(1 + \frac{(\omega_{sw} - \omega)c_{pw}T_w}{i_{masw} - i_{ma} + (Le_f - 1)[i_{masw} - i_{ma} - (\omega_{sw} - \omega)i_v] - (\omega_{sw} - \omega)c_{pw}T_w}\right) \quad (22)$$

In order to obtain the ratio of humidity and enthalpy of air in different sections of the cooling tower, these variables in the above sections must be clear. Therefore, computations will start considering variations of humidity ratio of wet air to water temperature and variations of enthalpy of wet air to water temperature according to Eqs. (21) and (22) and by utilizing of Runge-Kutta method, it is possible to compute humidity ratio, enthalpy and temperature of air in different sections of the cooling tower.

3 EXERGY AND SECOND-LAW EFFICIENCY CALCULATIONS

The main objective of the exergy analysis is to find and quantitatively survey the reasons for the thermodynamic defections of the stated process. Analysis of air and water conditions in the cooling tower using mathematical models is highly important in studying exergy. According to Shukuya and Hammache [11], the total exergy in the cooling tower equals the sum of thermomechanical and chemical exergies, both of which play a significant function in the evaluation of the real thermodynamic value of humidity assessment process. This paper, aims to analyze the exergy relations between water and air as well as the factors affecting them. In the cooling towers, disregarding the effect of potential and kinetic energies, special exergy in the stable state is as follows [12-14]:

$$\psi = \psi_{tm} + \psi_{ch} \quad (23)$$

Substituting the current state for the inlet state and also the dead state for the outlet state, the thermo-mechanical exergy will be obtained as follows:

$$\psi_{tm} = (i - i_0) - T_0(s - s_0) \tag{24}$$

which for an ideal gas, the formula will be:

$$\psi_{tm} = c_p(T - T_0) - T_0(c_p \ln\frac{T}{T_0} - R\ln\frac{p}{p_0}) \tag{25}$$

Pressure differential in the cooling tower is negligible, so:

$$\psi_{tm} = c_p(T - T_0 - T_0 \ln\frac{T}{T_0}) \tag{26}$$

Chemical exergy, which is an exergy change based on the concentration change, is defined as follows:

$$\psi_{ch} = \sum_{k=1}^{n} x_k(\mu_{k,0} - \mu_{k,00}) \tag{27}$$

In Eq. (27), x_k is the mole portion of k material in the mixture. Assuming the concentration change with a slight change of pressure, the chemical potential for the ideal gas mixture will be as follows:

$$\mu_{k,0} - \mu_{k,00} = RT_0 \ln\frac{P_{k,0}}{P_{k,00}} \tag{28}$$

The total exergy is the result of both thermo-mechanical exergy, which has turned from real state into dead state, and chemical exergy, which has turned from dead state into environmental state. Accordingly:

$$\psi = (i - i_0) - T_0(s - s_0) + \sum_{k=1}^{n} x_k(\mu_{k,0} - \mu_{k,00}) \tag{29}$$

As mentioned earlier, water and air are the fluids that have a significant function in the cooling towers, and so in order to perform exergy analysis, it is necessary to obtain the exergy equations of both water and air.
Temperature differential of water in the cooling tower and chemical exergy of environmental humidity are influential in exergy analysis of water. Drawing on Eq. (29), since water is an incompressible fluid, its exergy can be obtained as follows:

$$X_w = m_w[(i_{f,w} - i_{f,0}) - T_0(s_{f,w} - s_{f,0}) - R_v T_0 \ln\theta_0 \tag{30}$$

When exergy analysis of air in the cooling tower is aimed at, the changes in temperature and humidity are considered as effective factors.

Assuming pressure changes in the cooling tower as negligible and drawing on the Eqs. (25-27) the exergy of air is as follows:

$$\begin{aligned} X_{air} &= m_a[(c_{pa} + \omega c_{pv})(T - T_0 - T_0 LnT / T_0) \\ &+ R_a T_0((1 + 1.608\omega)Ln(1 + 1.608\omega_{00}) / \\ &(1 + 1.608\omega) + 1.608\omega Ln\omega / \omega_{00})] \end{aligned} \tag{31}$$

Taking the exergy equations of water and air into account, the analysis of water and air conditions in the cooling tower are of significant importance in the exergy analysis. In this paper, water and air conditions in the cooling tower are analyzed using the mathematical models of heat and mass transfer. For a real process the exergy input always exceeds the exergy output, this unbalance is due to irreversibilities, which we name exergy destruction. Therefore, the second-law efficiency, which is a measure of irreversible losses in a given process, is defined as follows:

$$\eta_{II} = \frac{(Total\ exergy\ leaving)}{(Total\ exergy\ entering\ )} \tag{32}$$

Considering the air-water thermodynamic properties and exergy distribution at discrete points along the tower height, we calculate [15]

$$Total\ exergy\ entering = [X_{w,H(j+1)} + X_{air,H(j)}] \tag{33}$$

$$Total\ exergy\ leaving\ = [X_{w,H(j)} + X_{air,H(j+1)}] \tag{34}$$

Consequently, the second-law efficiency was evaluated to measure the actual performance of the counter flow wet cooling tower in the Steelmaking unit of the Khuzestan Steel Company in this study.

4 HEAT AND MASS TRANSFER CALCULATIONS

Accurate calculation of heat and mass transfer coefficients is significant to predict the cooling tower performance and outlet conditions of air and water as well as cooling tower design.
Several researchers have theoretically calculated heat and mass transfer coefficients using experimental results and have defined the effect of some new parameters on these coefficients. For example, Thomas and Houston [16] considered the mass flow rates of air and water as the only factors affecting heat and mass transfer and gave the following relations for the heat and mass transfer coefficients as:

$h_c A_V = 3(m_w / A)^{0.26}(m_a / A)^{0.72}$ and $h_D A_V = 295(m_w / A)^{0.26}(m_a / A)^{0.72}$

In the present paper, these coefficients were calculated considering important factors such as the mass flow rates of air and water, inlet water temperature and enthalpy of air entering the cooling tower and conditions of air and water in the tower. Consequently, heat and mass transfer coefficients have been calculated using the mathematical model presented in this paper based on the principles of mass and heat transfer at the interface between water and air and finally, numerical solution for this model.

Jabber and Webb [17] presented an analysis that shows how the theory of heat exchanger design may be applied to cooling towers. They demonstrated that number of transfer units (*NTU*) definition used for heat exchanger design is applicable to all cooling tower operating conditions. Consequently, NTU has been calculated numerically according to the model presented in this paper as follows [7]:

$$NTU = h_d A_v V / m_a = \int_{w_i}^{w_o} \frac{d\omega}{\omega_{sw} - \omega} \tag{35}$$

If the value of the integral in the above equation is known, convective mass transfer coefficient (h_d) can be obtained. To calculate the integral in Eq. (35), we used the Merkel number (transfer coefficient) according to the Poppe method as follows:

$$Me_p = \int \frac{m_a}{m_w} \frac{d\omega / dT_w}{\omega_{sw} - \omega} dT_w \tag{36}$$

Merkel number plays a significant role in the investigation of heat and mass transfer process in the cooling tower. Therefore, the integration of Eq. (35) is solved numerically considering the Merkel number and substituting Eq. (36) into Eq. (35), we get

$$\int_{w_i}^{w_o} \frac{d\omega}{\omega_{sw} - \omega} = Me_p \left(\frac{m_w}{m_a}\right) \tag{37}$$

To determine the numerical value of the Merkel number, the gradient of the Merkel number-temperature is first calculated as follows:

$$\frac{dMe_p}{dT_w} = \frac{c_{pw}}{i_{masw} - i_{ma} + (Le_f - 1)[i_{masw} - i_{ma} - (\omega_{sw} - \omega) i_v] - (\omega_{sw} - \omega) c_{pw} T_w} \tag{38}$$

Consequently, the Merkel number can be calculated for calculating the mass transfer coefficient using the mathematical model presented in this paper and numerical solution of Eqs. (21), (22) and (38) using the Runge-Kutta method.

If the value of (h_d) is known, the heat transfer coefficient (h_c) may be obtained using Lewis factor ($Le_f = h_c / c_{pma} h_d$). In order to simplify heat and mass transfer equations between air and water , Le_f is assumed to be 1 by researchers, in this paper, Le_f is calculated for different parts of the tower in order to achieve more accuracy. With this simplification, evaporation is neglected in the energy balance. Indeed, water evaporation is a function of the actual value of the Lewis factor, especially when the ambient air is relatively warm. Moreover, in this research, convection and evaporation heat transfer has been evaluated by precise prediction of air and water stream conditions and also heat and mass transfer coefficients inside the cooling tower. The evaporation and convection heat transfer rates Q_{evap} and Q_{conv} can be calculated as follows:

$$Q_{evap} = h_d A_v (\omega_{sw} - \omega) i_{fg,w} \tag{39}$$

$$Q_{conv} = h_c A_v (T_w - T_{db}) \tag{40}$$

In Eqs. (39) and (40), the driving potentials for evaporative heat transfer and convective heat transfer are $(\omega_{sw} - \omega)$ and $(T_w - T_{db})$, respectively.

Table 1 Reference operating parameters and ambient conditions

Conditions	Reference
Height of column packing (m)	1.8
Total surface area of packing (m^2)	207
Atmospheric pressure (kPa)	101
Air flow rate (kg s^{-1})	157.2
Air dry-bulb temperature ($^\circ$C)	36.9
Air wet-bulb temperature ($^\circ$C)	27.8
Humidity ratio of the air ($kg_w\, kg_a^{-1}$)	0.02
Water flow rate (kg s^{-1})	693.22
Water temperature ($^\circ$C)	50

5 EXERGY AND SECOND-LAW EFFICIENCY ANALYSIS

As it was mentioned earlier, exergy analysis of water and air in the cooling tower of Khuzestan Steel Company was carried out. In this analysis, based on references presented in Table 1, the following presumptions of T_0 = 298.15°K, θ = 0.5, and ω_{00} = 0.009923 $kg_w kg_a^{-1}$were made.

Also, grid type cooling tower packing is made from PVC. The results of this analysis are displayed in Figs.

4-8. As shown in Fig. 4, at first the temperature of the inlet air decreases in the tower, but it increases when it reaches the height of over 0.3 meters. This height is actually the contact point in the curves of water and air temperatures. Before the contact point, the air temperature is higher than the water temperature; therefore, the heat transfer streams from air to water. In fact, water and air have the same temperature when the curves of water and air temperature meet.

At this point the heat difference between water and air is zero. After this point the heat transfer initiates from water to air, since the temperature of water is higher than that of air. As shown in Fig. 5, at the point of air inlet in to the tower, air exergy decreases, while it constantly increases when heat is transferred due to evaporation and convection. Heat transfer through evaporation and convection is an effective factor in the analysis of air exergy.

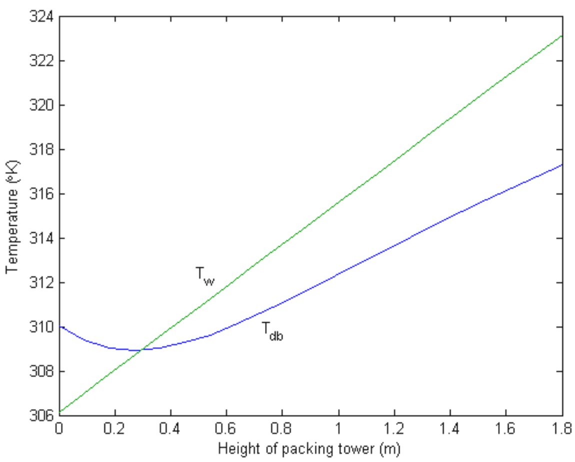

Fig. 4 Temperature profiles of water and air in the cooling tower

At first, the inlet air undergoes a preliminary temperature reduction flowing through the lower parts of the tower towards its upper sections. Though, it later increases when through convective heat transfer, the exergy makes the flow of heat energy in the air possible. As air temperature increases, so does the air exergy. As shown in Fig. 6, flowing from the lower part of the tower to its upper part, the air exergy increases as the air humidity rises. The temperature of water is a determining factor in the energy of water, according to Fig. 7.

When hot water from the upper part of the tower is splashed on the packing, then the temperature of the water decreases constantly. The energy of water, therefore, decreases from the upper part of the cooling tower to its lower part as the temperature of water decreases. As indicated by Fig. 8, since the system generates entropy, the exergy of water is more than the exergy of air. The exergy of air, however, constantly increases from the lower part of the tower to its upper part. As the temperature decreases, so does the exergy of the water. While as the temperature of air increases, so does the exergy of air. Moreover, thermodynamic analysis of this cooling tower is carried out using second-law efficiency for different inlet wet-bulb temperature. According to Fig. 9, second-law efficiency increases as the exergy destruction decreases with increasing wet-bulb temperature. As shown in Fig. 10, the effect of the ratio of mass flow rate of water to mass flow rate of air is investigated on the second-law efficiency for different inlet wet-bulb temperature. Note that the air flow rate is kept constant. When the mass flow ratio increases, the total heat transfer ($Q_{evap} + Q_{conv}$) also rises, therefore, the second-law efficiency is higher as well.

6 HEAT AND MASS TRANSFER ANALYSIS

Regarding sections 2, 4 and Table 1, the heat performance of counter flow wet cooling tower of Khuzestan Steel Co. is analyzed in this section. Convection and evaporation heat transfer rates Q_{conv}, Q_{evap} and Q_{total} (= Q_{conv}+ Q_{evap}) are plotted against the height of the tower as shown in Fig. 11. Poured water from the top of the tower, upon reaching lower parts of the tower has T_w much lower than T_{db} .This implies until the tower height of 0.3m. Results show negative convection, according to Fig. 11. At intersection point of T_{db} and T_w curves, there is no indication of temperature difference; hence, there is no convective heat exchange between water and air. A further inspection of Fig. 11 shows that the driving potential for convective heat transfer $(T_w - T_{db})$ increases in the top portions of the tower, which means that convective heat transfer increases as well. Furthermore, the driving potential for evaporative heat transfer is $(\omega_{sw} - \omega)$. Fig. 11 shows that the potential for evaporation decreases first and then increases after the tower height of 0.6m, which means that evaporative heat transfer also increases. Fig. 11 clearly shows that the evaporative heat transfer rate inside the cooling tower is much greater than the convective heat transfer rate. As a result, the evaporative heat transfer rate is the major contribution of heat transfer inside the cooling tower. Moreover, the total heat transfer rate, Q_{total}, decreases descending from the upper part of the tower towards its lower part.

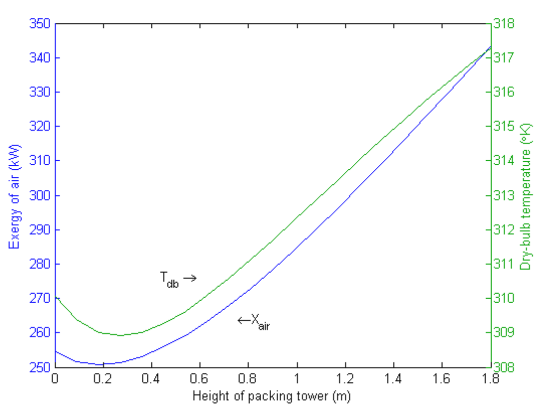

Fig. 5 Exergy of air and air temperature profiles in the cooling tower.

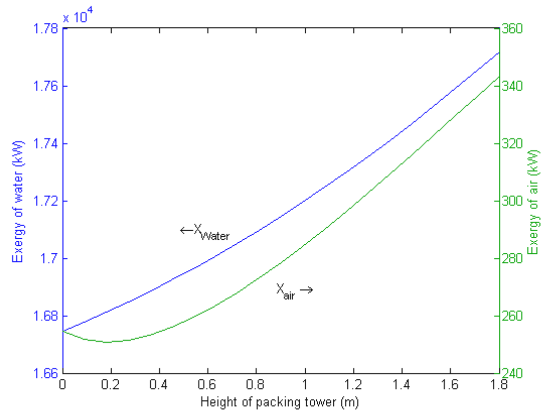

Fig. 8 Exergy profiles of water and air in the cooling tower.

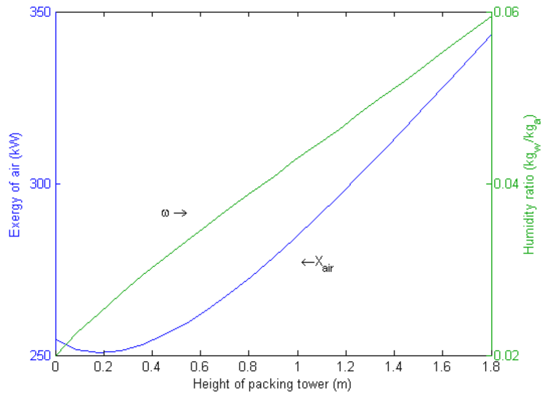

Fig. 6 Exergy of air and humidity ratio of air profiles in the cooling tower.

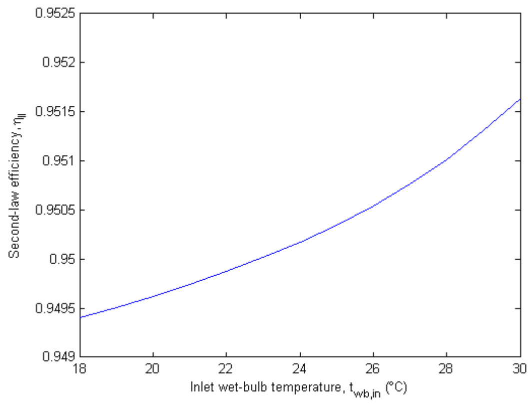

Fig. 9 Variation of second-law efficiency versus inlet wet-bulb temperature in the cooling tower study.

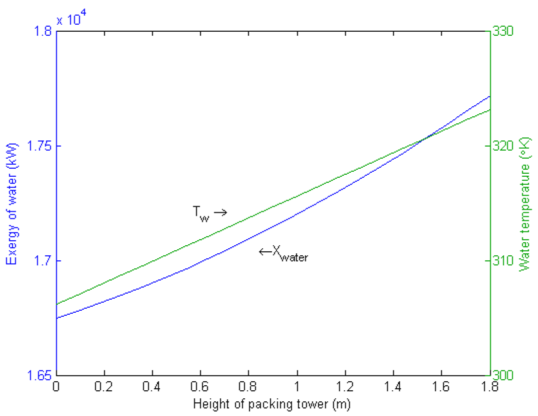

Fig. 7 Exergy of water and water temperature profiles in the cooling tower.

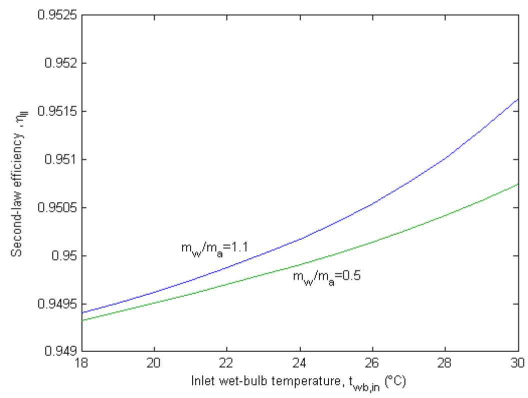

Fig. 10 Variation of second-law efficiency versus inlet wet-bulb temperature for different ratios of m_w/m_a

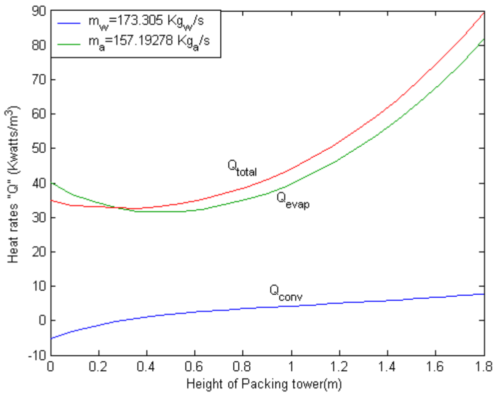

Fig. 11 Variation of heat rates against the height of packing tower.

7 CONCLUSIONS

In this research, a mathematical model based on the principles of heat and mass transfer between water and air has been presented in order to predict air and water conditions inside a counter flow wet cooling tower. The performance of the counter flow wet cooling tower in the Steelmaking unit of the Khuzestan Steel Company has undergone heat and mass transfer, exergy, and second-law efficiency analysis according to the mathematical model presented in this study. Also, *NTU* definition is used to analyze heat exchange in the cooling tower.

For the sake of simplicity, state of art implies assuming Le_f, in heat and mass transfer equations between air and water to be one; but in this paper, Le_f is calculated at specific points all through the tower in order to achieve more accurate results in predicting the heat performance of this cooling tower. As a result, heat and mass transfer equations and their numerical solutions presented are more complicated.

NOMENCLATURE

A_v Surface area of water droplets per unit of tower, m^2/m^3

c_{pa} Specific heat of dry air at constant pressure, kJ/kgK

c_{pw} Specific heat of water at constant pressure, kJ/kgK

c_{pv} Specific heat of water vapor at constant pressure, kJ/kgK

h_c Convective heat transfer coefficient, kW/m^2K

h_d Convective mass transfer coefficient, kW/m^2K

$i_{fg,w}$ Phase change enthalpy at T_w , kJ/kg

$i_{f,w}$ Enthalpy of saturated liquid water, kJ/kg

i_{ma} Enthalpy of air in the cooling tower at T_a , kJ/kg

i_{masw} Enthalpy of saturated air at T_w , kJ/kg

i_v Enthalpy of water vapor at T_w , kJ/kg

Le_f Lewis factor

m_a Mass flow rate of the air, kg_a/s

m_w Mass flow rate of the water, kg_w/s

Me_p Merkel number according to Poppe method

NTU Number of transfer units

P Pressure, kPa

P_g Dew point pressure of saturated water vapor at mean water temperature, kPa

Q_{conv} Convection heat transfer rate, kW/m^3

Q_{evap} Evaporation heat transfer rate, kW/m^3

R_a Gas constant of dry air, kJ/kgK

R_v Gas constant of water vapor, kJ/kgK

s Entropy, kJ/kgK

$s_{f,w}$ Entropy of water, kJ/kgK

T_a Air temperature, °K

T_w Water temperature, °K

T_0 Temperature in restricted dead state, °K

ω Humidity ratio, kg_w /kg_a

ω_{sw} Humidity ratio of saturated air at T_w , kg_w/kg_a

ω_{00} Humidity ratio of environment, kg_w/kg_a

X_{air} Air exergy, kW

X_w Water exergy, kW

θ Relative humidity

Ψ_w Specific water exergy, kJ/kg

Ψ_{tm} Specific thermomechanical exergy, kJ/kg

Ψ_{ch} Specific chemical exergy,kJ/kg

μ Chemical potential, kJ/kg

Subscripts

0 Restricted state

00 Environment

a Air

s Saturation

sw Saturated moist air evaluated at T_w

v Vapor

w Water

wb Wet-bulb

REFERENCES

[1] Khan, J. R., Qureshi, B. A., and Zubair, S. M., "A comprehensive design and performance evaluation study of counter flow wet cooling towers", International Journal of Refrigeration, Vol. 27, Iss. 8, 2004, pp. 914–923.

[2] Hyland, R. W. and Wexler, A., "Formulations for the thermodynamic properties of the saturated phases of H_2O from 173.15 to 473.15 K", ASHRAE transactions, Vol. 89, No. 2A, 1983, pp. 500-519.

[3] Hyland, R. W and Wexler, A., "Formulations for the thermodynamic properties of dry air from 173.15 to 473.15 K at pressure to 5 MPa", ASHRAE transactions, Vol. 89, No. 2A, 1983, pp. 520-535.

[4] Bahaidrah, H. M. S., "Design and performance evaluation of evaporative cooling towers", M.Sc. thesis, faculty of the collage of graduate studies, king fahd university of petroleum and minerals, Dhahran, 1999.

[5] EI-Dessouky, H. T. A., AI-Haddad, A. and AI-Juwayhel, F., "A modified analysis of counter flow cooling towers", ASME journal of heat transf., Vol. 119, No. 3, 1997, pp. 617-626.

[6] Nimr, M. A., "Modeling the dynamic thermal behavior of cooling towers containing packing materials", heat transfer engineering, Vol. 20, Iss. 1, 1999, pp. 91-96.

[7] Khan, J. R., Yaqub, M. and Zubair, S., "Performance characteristics of counter flow wet cooling towers", energy conversion and management, Vol. 44, Iss. 13, 2003, pp. 2073–2091.

[8] Poppe, M. and Rögener, H., Berechnung Von Rückkühlwerken, VDI-Wärmeatlas, 1991, pp. Mi 1–Mi 15.

[9] Kloppers, J. C. and Kroger, D. G., "A critical investigation into the heat and mass transfer analysis of counterflow wet-cooling towers", International Journal of Heat and Mass Transfer, Vol. 48, Iss. 3-4, 2005, pp. 765–777.

[10] Bosnjakovic, F., "Technische thermodinamik", theodor steinkopf, dresden, 1965.

[11] Shukuya, M. and Hammache, A., "Introduction to the concept of exergy", paper presented in the low exergy systems for heating and cooling of buildings, IEA ANNEX37 finland, 2002, pp. 41-44.

[12] Wang, L. and Li, N., "Exergy transfer and parametric study of counter flow wet cooling towers", applied thermal engineering, Vol. 31, Iss. 5, 2011, pp. 954-960.

[13] Muangnoi, T., Asvapoositkul, W. and Wongwises, S., "An exergy analysis on the performance of a counterflow wet cooling tower", applied thermal engineering, Vol. 27, Iss. 5-6, 2007, pp. 910-917.

[14] Bejan, A., "Advanced engineering thermodynamics", second edition, wiley, Singapore, 1997.

[15] Qureshi, B. A. and Zubair, S. M., "Second-law-based performance evaluation of cooling towers and evaporative heat exchangers", International Journal Thermal Sciences, Vol. 46, No. 2, 2007, pp. 188–198.

[16] Thomas, W. J. and Houston, P., "Simultaneous heat and mass transfer in cooling towers", British Chemical Engineering, 1959a and 1959b, pp. 160-163 and 217-222.

[17] Jaber, H. and Webb, R. L., "Design of cooling towers by the effectiveness-NTU method", ASME journal of heat transfer, Vol. 111, 1989, pp. 837-843.