Entropy generation analysis in an evaporative air-cooled heat exchanger

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Abstract
In the present study, air-side entropy generation rate for an evaporative air-cooled heat exchanger has been investigated using thermodynamic second-law analysis. For this purpose, entropy generations due to heat transfer, friction loss and evaporation were taken into consideration. From the results of this study, it was observed that the total air-side entropy generation was increased by increasing the air-side Reynolds number. Actually, increasing of air mass flow rate, increases irreversibilities due to both evaporation and friction. Stepping up of ambient temperature leads to increasing of irreversibility due to water evaporation. The results showed when the ambient temperature approaches to inlet process fluid temperature, the contribution of the entropy generation rate due to water evaporation becomes the dominant term in the total entropy generation rate. Moreover, it was seen that the effect of deluge water mass flow rate changes on entropy generation rate is slight.

Key words: Thermodynamic analysis, Entropy generation, Second law, Evaporative, Air cooled heat exchanger, Deluge water

1. Introduction

Air-cooled heat exchangers are utilized in various applications such as air-conditioning and refrigeration systems, chemical plants, and power plants. In an evaporative cooler, heat is usually transferred from the process fluid to the cooling air stream via spraying water over the tubes and extended surfaces or finned tubes. As water flows down the surface of tube, it is evaporated by air flowing over it, resulting in cooling of the process fluid.

In a thermodynamic process, the loss of exergy is primarily due to the associated irreversibilities which generate entropy. There is a direct proportion between the wasted power and the entropy generation rate. If engineering systems are to operate such that their lost work is to be minimized, then the conceptual design of such systems and components must comply with the minimization of entropy generation (Bejan, 1994, 1996). The minimization of entropy generation in evaporative systems attributed to fluid flow, heat transfer and evaporation of water is of considerable practical interest with regard to heat transfer augmentation techniques. Our main objective is to increase the convective heat transfer coefficient as compared to that of an unaugmented surface. A parallel objective, however, is to achieve this improvement without an increase in pumping power as demanded by the forced convection arrangement. These two conflicting objectives lead to concept of optimal design of an engineering system. Optimization of these systems in general and heat exchangers in particular plays a significant role in the profitibility of the overall systems. The current trend in the optimization of thermal systems considers both the first and second laws of thermodynamics as the criteria; i.e. the efficient usage of both energy and exergy shall be ensued (Bejan, 1996).

Based on the entropy generation minimization principle, considerable optimal designs of thermal systems have been proposed. For example, Morosuk (2005) worked on entropy generation in conduits filled with porous medium.
The irreversibility analysis in various duct geometries with constant wall heat flux and laminar flow was done by Sahin (1998). Sara et al. (2001) performed the optimal analysis of rectangular channels with square pin-fins. Salimpour and Zahedi (2012) carried out thermodynamics second law analysis to optimize some proposed novel cycles suitable for recovery of liquefied natural gas. Shokouhmand and Salimpour (2007a) analyzed heat transfer convection inside helical coiled tubes with uniform wall temperature. They further found the optimal Reynolds number of the flow in helical coils based on the minimal entropy generation principle (Shokouhmand and Salimpour, 2007b). Shokouhmand et al. (2009) performed an entropy generation analysis to find optimal porosity in an air heater conduit filled with a porous matrix.

In a recent work, Salimpour and Bahrami (2011) investigated the entropy generation in a dry air-cooled heat exchanger. In this study, it was observed that the total entropy generation has a minimum at special Reynolds numbers of tube-side. Also based on the computed results, they proposed a new correlation to predict the optimum Reynolds number of the tube-side fluid flow.

The present paper intends to investigate entropy generation in an evaporative air-cooled heat exchanger using thermodynamic analysis. The effect of various parameters on the air-side entropy generation is the main principal concern. Similar to other thermal systems, the forced convection in evaporative air-cooled heat exchanger also faces the challenge of inventing the optimal design to have the minimal entropy generation. However, to our knowledge, the optimization work of evaporative air-cooled heat exchangers using the minimum entropy generation concept has not been explored in existing literature. Nonetheless, the investigation of the thermodynamic optimal design of evaporative air-cooled heat exchangers is worthwhile for practical applications. The objective of this paper is to present second-law based evaluation of evaporative air-cooled heat exchangers undertaking different operation conditions.

**Nomenclature**

- \( A_c \) Free flow area through the heat exchanger
- \( a \) Half of hydraulic diameter
- \( c_p \) Specific heat capacity
- \( d_f \) Fin diameter
- \( d_t \) Tube inside diameter
- \( d_o \) Tube outside diameter
- \( d_r \) Fin root diameter
- \( F \) Pressure loss per unit length
- \( e \) Exergy
- \( e_t \) Total flow exergy
- \( f \) Friction factor
- \( G_c \) Mass velocity
- \( h \) Average heat transfer coefficient
- \( h \) Special enthalpy
- \( k \) Thermal conductivity
- \( L \) Length of finned tube
- \( \dot{m} \) Mass flow rate
- \( N_s \) Entropy generation number
- \( Nu \) Nusselt number
- \( n_r \) Number of tube rows

The text contains a list of symbols and their meanings. Here is the list in natural text format:

- $n_r$: Number of tubes per row
- $P$: Pressure
- $Pr$: Prandtl number
- $P_f$: Fin pitch
- $P_L$: Longitudinal pitch
- $P_t$: Transversal tube pitch
- $\dot{Q}$: Heat transfer rate
- $r_i$: Inner radius of tube
- $Re$: Reynolds number
- $\dot{S}_{gen}$: Entropy generation rate
- $St$: Stanton number
- $s$: Specific entropy
- $T$: Bulk temperature
- $T_i$: Inlet temperature
- $T_{wm}$: Mean deluge water film temperature
- $t_f$: Fin thickness (mean)
- $V$: Average velocity
- $v$: Specific volume
- $v_c$: Air velocity
- $x$: Mole fraction

**Greek letters**

- $\beta_1'$: Dimensionless parameter, $= 4k/(\mu c_p)$
- $\varepsilon_R$: Rational effectiveness Eq. (15)
- $\eta_{w-s}$: Witte and Shamsunder's efficiency Eq. (16)
- $\lambda$: Dimensionless passage length
- $\mu$: Chemical potential
- $\rho$: Density
- $\tau$: Dimensionless temperature difference
- $\varphi_0$: Ambient relative humidity
- $\phi_1, \phi_2, \phi_3$: Dimensionless parameters
- $\omega$: Humidity ratio
- $\bar{\omega}$: Mole fraction ratio

**Subscripts**

- $a$: Refers to air
- $C$: Cold stream
ev  Evaporative

H   Hot stream

in Inlet

o  Outer-side

out  Out

p Refers to process fluid or pressure loss

T Refers to temperature difference

v Refers to vapor

w Refers to water or wall

θ Ambient conditions

Superscripts

(−) Molar quantity

(*) Properties at ambient conditions

2. Geometry of an evaporative air-cooled heat exchanger

In the evaporative air-cooled heat exchanger shown in Fig. 1, air is flowed over the tubes by means of a fan and cooling process of fluid flowing in tubes is achieved by spraying water on the tubes, essentially deluging them and forming a film of water flowing downwards under the action of gravity. As the water flows down the surface of the tube, it is evaporated by air flowing over it, results in cooling down the fluid. The tubes are equipped with circular fins to enhance the heat transfer surface. In Fig.1, $d_i$ is the inner diameter of tube, $d_o$ is the outer diameter of tube, $d_r$ is the root diameter of fin, $d_f$ is the diameter of the fin, $P_f$ is the fin pitch, $t_f$ is the thickness of the fin, $n_t$ is number of tube rows, $n_o$ is the number of tubes per row, $P_L$ is the longitudinal pitch and $P_t$ is the transversal tube pitch.

![Geometry of an evaporative air-cooled heat exchanger](image)

Fig. 1 Geometry of an air-cooled heat exchanger
3. Entropy generation analysis

The total entropy generation can be expressed as the summation of the entropy generation contributions of internal and external flows of the heat exchanger and water evaporation into the air. However, as the entropy generation analysis of tube side has been previously presented by Salimpour and Bahrami (2011), in this study only the air-side entropy generation is investigated.

3.1 Entropy generation due to heat transfer and pressure loss

The differential element of Fig. 2 was considered as the control volume to derive the entropy generation relationships. Second law of thermodynamics \( d\dot{S}_{gen} = \dot{m}c_p \left( \frac{T_{w} - T}{T_{w}} - \frac{dP}{\rho c_p dT} \right) \) and the thermodynamics relationship \( T d\dot{S} = c_p dT/dT \) are combined to give,

\[
d\dot{S}_{gen} = \dot{m}c_p \left( \frac{T_{w} - T}{T_{w}} - \frac{dP}{\rho c_p dT} \right)
\]  

(1)

Writing bulk temperature, \( T \), in terms of inlet and wall temperatures, and integrating the resultant equation along the path, Eq. (2) is achieved. This formulation is fully described by Shokouhmand and Salimpour (2007a).

\[
\dot{S}_{gen} = \dot{m}_a c_p \left[ \ln \left( \frac{1 - e^{-\beta_1' \lambda Nu / Re}}{1 - \tau} \right) - \tau \left( 1 - e^{-\beta_1' \lambda Nu / Re} \right) + \frac{FV_a}{h_a T_{wm}} \ln \left( \frac{e^{\beta_1' \lambda Nu / Re - \tau}}{1 - \tau} \right) \right]
\]  

(2)

where \( \tau = \frac{T_{w} - T_i}{T_{w}} \) is dimensionless temperature difference, \( \lambda = L/2a \) is dimensionless passage length, \( St = h_a / \rho V c_p \) is Stanton number, and \( V \) is velocity of air through heat exchanger, \( h_a \) is average heat transfer coefficient of the air and \( T_{wm} \) is mean deluge water film temperature. In Eq. (2), \( a \) is defined as \( a = 2A_c / P_c \), where \( P_c \) is the wet perimeter that air is contacted by tubes and fins and \( A_c \) is cross section area. In other word, \( a \) is the half of hydraulic diameter. Also, \( F \) is defined as \( F = \Delta P / (n_r P_c) \), where \( \Delta P \) is pressure loss in heat exchanger and \( n_r \) is the number of tube rows in the flow direction. Entropy generation number, \( N_S \), is defined as \( N_S = \dot{S}_{gen} / (\dot{Q} / \Delta T) \). By defining the dimensionless parameter \( \beta_1' = 4k / (\mu c_p) \) and simplification, \( N_S \) can be written as

\[
N_S = \ln \left( \frac{1 - e^{-\beta_1' \lambda Nu / Re}}{1 - \tau} \right) - \tau \left( 1 - e^{-\beta_1' \lambda Nu / Re} \right) + \frac{FV_a}{h_a T_{wm}} \ln \left( \frac{e^{-\beta_1' \lambda Nu / Re - \tau}}{1 - \tau} \right)
\]  

(3)

Equation (3) can be regarded as following.

\[
N_S = (N_S)_{\tau} + (N_S)_P
\]  

(4)

\[
(N_S)_{\tau} = \ln \left( \frac{1 - e^{-\beta_1' \lambda Nu / Re}}{1 - \tau} \right) - \tau \left( 1 - e^{-\beta_1' \lambda Nu / Re} \right)
\]  

(5)

\[
(N_S)_P = \frac{FV_a}{h_a T_{wm}} \ln \left( \frac{e^{-\beta_1' \lambda Nu / Re - \tau}}{1 - \tau} \right)
\]  

(6)
For a finned-tube air-cooled heat exchanger with six tube rows, Briggs and Young (1963) recommended the following equation for calculation of Nusselt number.

\[
Nu = \frac{h_a d_r}{k} = 0.134 \Pr^{0.33} Re^{0.681} \left( \frac{(P_f - t_f)}{d_r} \right)^{0.2} \left( \frac{(P_f - t_f)}{t_f} \right)^{0.1134}
\]  

(7)

where \( Re = G_d d_r / \mu \) and \( G_d = \dot{m}_a / A_c \). This equation is valid for \( 1000 < Re < 18000 \), \( 11.13 \text{ mm} < d_r < 40.89 \text{ mm} \), \( 2.84 \text{ mm} < (d_f - d_r) < 33.14 \text{ mm} \), \( 0.33 \text{ mm} < t_f < 2.02 \text{ mm} \), \( 1.30 \text{ mm} < P_f < 4.06 \text{ mm} \) and \( 24.49 \text{ mm} < P_t < 111 \text{ mm} \).

Ward and Young (1959) found the row-effect to be a function of the air velocity, \( v_c \), at the smallest cross section area of the finned tube bundle. According to Gionolio and Cuti (1981) this trend is correlated approximately by

\[
h_{n_r} = h_0 (1 + v_c/n_f)^{-0.14}
\]

(8)

where \( h_0 \) is the mean heat transfer coefficient for a six-row bundle and \( n_r \) is the number of tube rows in the flow direction.

Robinson and Briggs (1966) proposed the following equation for the pressure loss for tubes arranged in a staggered pattern

\[
\Delta P = 18.93 n_f Re^{-0.316} \left( \frac{P_f}{d_r} \right)^{0.927} \left( \frac{P_f}{d_t} \right)^{0.515} \frac{\alpha^2}{\rho}
\]

(9)

where, \( P_d = [(P_f/2)^2 + P_f^2]^{0.5} \). This equation is valid for \( 2000 < Re < 5000 \), \( 18.64 \text{ mm} < d_r < 40.89 \text{ mm} \), \( 39.68 \text{ mm} < d_t < 69.85 \text{ mm} \), \( 21.04 \text{ mm} < (d_f - d_r) < 28.96 \text{ mm} \), \( 37.11 \text{ mm} < P_f < 98.99 \text{ mm} \), \( 1.98 \text{ mm} < P_f < 28.22 \text{ mm} \), \( 42.85 \text{ mm} < P_t < 114.3 \text{ mm} \) and \( 1.8 < P_t/d_r < 4.6 \).

### 3.2 Entropy generation due to water evaporation

Total exergy of humid air (per mole) is calculated as

\[
\bar{e}_t = x_a \left[ \bar{h}_a - \bar{h}^\prime_a - T_0 (\bar{s}_a - \bar{s}^\prime_a) + \mu_a^\prime - \mu_{0, a} \right] + \left[ \bar{h}_v - \bar{h}^\prime_v - T_0 (\bar{s}_v - \bar{s}^\prime_v) + \mu_v^\prime - \mu_{0, v} \right]
\]

(10)

Considering ideal gas model for water vapor, Eq. (10) is rendered to

\[
\bar{e}_t = (x_a \bar{e}_{p, a} + x_v \bar{e}_{p, v}) T_0 \left( \frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + \bar{R} T_0 ln \frac{\rho}{\rho_0} + \bar{R} T_0 \left( x_a ln \frac{x_a}{x_a_0} + x_v ln \frac{x_v}{x_v_0} \right)
\]

(11)

Substituting \( x_a = (1 + \bar{\omega})^{-1} \) and \( x_v = (1 + 1/\bar{\omega})^{-1} \) in Eq. (10), we have

\[
e_t = (c_{p, a} + \omega c_{p, v}) T_0 \left( \frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + (1 + \bar{\omega}) R_a T_0 ln \frac{\rho}{\rho_0} + R_a T_0 \left[ (1 + \bar{\omega}) ln \frac{1+\bar{\omega}}{1+\bar{\omega}} + \bar{\omega} ln \frac{\bar{\omega}}{\omega_0} \right]
\]

(12)

Putting \( \omega = \bar{\omega} = 0 \), the exergy of dry air, \( e_{t,a} \), is calculated as

\[
e_{t,a} = c_{p, a} T_0 \left( \frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + R_a T_0 ln \frac{\rho}{\rho_0} + R_a T_0 ln (1 + \bar{\omega})
\]

(13)

Doing some simplifications, liquid water exergy can be stated by Eq. (14).

\[
e_{t,w} \equiv h_f(T) - h_g(T_0) - T_0 s_f(T) + T_0 s_g(T_0) + [P - P_{sat}(T)] v_f(T) - R_v T_0 ln \phi_0
\]

(14)

where, \( \phi_0 \) is ambient relative humidity. The entropy generation of an evaporative system can be calculated from the exergy balance as (Bejan, 1997).

\[
\dot{S}_{gen} = \frac{m_a}{T_0} \left( e_{t,a} + \omega e_{t,w} - e_t \right)
\]

(15)
In the above equation, $T_0$ is ambient temperature, $S_{gen}$ is entropy generation rate due to water evaporation, $\dot{m}_a$ is air mass flow rate, $e_{tw}$ is exergy of the dry air, $e_t$ is total exergy of the outlet air and $\omega$ is humidity of the outlet air (humidity of inlet air is neglected).

### 4. Results and discussion

In this section, investigation of the effect of different air-cooled heat exchanger geometrical and flow parameters on the entropy generation was accomplished. It should be noted that in this study, we focused on the air-side entropy generation. In the calculations of this section, all the parameters were considered as the baseline case unless specified otherwise. Table 1 presents the conditions of the baseline case.

#### Table 1 Specifications of the baseline case

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet temperature of process fluid, $T_{in}$ (K)</td>
<td>353.75</td>
<td>Thermal conductivity of tube material, $k_f$ (W/mK)</td>
<td>50</td>
</tr>
<tr>
<td>Mass flow rate of process fluid, $\dot{m}_p$ (kg/s)</td>
<td>35.67</td>
<td>Outside diameter of tube, $d_o$ (mm)</td>
<td>24.5</td>
</tr>
<tr>
<td>Ambient temperature, $T_o$ (K)</td>
<td>293.15</td>
<td>Inside diameter of tube, $d_i$ (mm)</td>
<td>21.1</td>
</tr>
<tr>
<td>Thermal conductivity of fin, $k_f$ (W/mK)</td>
<td>204</td>
<td>Number of tube rows, $n_r$</td>
<td>6</td>
</tr>
<tr>
<td>Fin diameter, $d_f$ (mm)</td>
<td>57.2</td>
<td>Number of tubes per row, $n_p$</td>
<td>55</td>
</tr>
<tr>
<td>Fin root diameter, $d_r$ (mm)</td>
<td>27.6</td>
<td>Transversal tube pitch, $P_t$ (mm)</td>
<td>58</td>
</tr>
<tr>
<td>Fin thickness, $t_f$ (mm)</td>
<td>0.5</td>
<td>Longitudinal pitch, $P_L$ (mm)</td>
<td>50.22</td>
</tr>
<tr>
<td>Fin tip thickness, $t_p$ (mm)</td>
<td>0.25</td>
<td>Length of tube, $L$ (m)</td>
<td>10.203</td>
</tr>
<tr>
<td>Fin pitch, $P_f$ (mm)</td>
<td>2.8</td>
<td>Fan rotational speed, $N_F$ (rpm)</td>
<td>41.5</td>
</tr>
</tbody>
</table>

#### 4.1 Effect of air-side Reynolds number on air-side entropy generation

Figure 3 shows the variations of entropy generation rates due to pressure loss, $(N_{S})_{Re}$, temperature difference, $(N_{S})_{\Delta T}$, water evaporation, $(N_{S})_{ev}$, and total air-side entropy generation $(N_S=(N_{S})_{Re}+(N_{S})_{\Delta T}+(N_{S})_{ev})$ versus the outer-side Reynolds number. It should be noted that this Reynolds number ($Re_o$) is defined based on the half of hydraulic diameter, $a$. Therefore, the range of this Reynolds number is much less than the range of Reynolds number used in Eqs. (7) and (9). From this figure it is obvious that variation of $(N_{S})_{Re}$ is very slight, which proves that by increasing $Re_o$, pressure loss irreversibility does not grow, considerably. Also, $(N_{S})_{\Delta T}$ first is increased (to $Re_o = 200$) and then is decreased. It can be explained that, although the temperature difference is reduced due to increasing of air-side Nusselt number, but increasing of $Re_o$ increased the heat transfer rate which leads to the growth of $(N_{S})_{\Delta T}$. By further rising $Re_o$, reduction in $\Delta T$ exceeds the increase of heat transfer and as a result, $(N_{S})_{\Delta T}$ is reduced. $(N_{S})_{ev}$ grows by the increase of $Re_o$ and in the range $Re_o > 1000$ the curve of $(N_{S})_{ev}$ gets steeper. This issue can be explained such that by increasing $Re_o$, evaporation rate of deluge water is increased which results in the increase in entropy generation rate due to water evaporation. According to this figure, the effect of $(N_{S})_{ev}$ on $N_S$ in high Reynolds numbers is more significant. Totally saying, increasing of $Re_o$ leads to increasing of air-side entropy generation rate.

To examine the effect of each contribution of irreversibility, the ratio of its entropy generation to the total air-side entropy generation is graphed in Fig. 4. This figure shows the variations of $\phi_1 = (N_{S})_{\Delta T}/N_S$, $\phi_2 = (N_{S})_{Re}/N_S$ and $\phi_3 = (N_{S})_{ev}/N_S$ with air flow Reynolds number. According to this figure, $\phi_2$ is almost constant; thus one can conclude that the effect of $(N_{S})_{Re}$ on $N_S$ is negligible; this matter can also be seen in Fig. 3. Simultaneous review of Figs. 3 and 4 shows that by increasing $Re_o$, entropy generation due to water evaporation is increased; and its share of total entropy generation is augmented so that we can say in $Re_o > 1200$ all of the air-side entropy generation is induced by water evaporation.

To investigate the irreversibility in more details, two irreversibility parameters were proposed by Bruges and Reistad and Witte and Shamsundar which are expressed by Eqs. (15) and (16), respectively (Bejan, 1997).
\[ \varepsilon_R = \frac{m_C(e_{\text{out}} - e_{\text{in}})_C}{m_H(e_{\text{in}} - e_{\text{out}})_H} = 1 - \frac{T_0 S_{\text{gen}}}{m_H(e_{\text{in}} - e_{\text{out}})_H} \]  
(16)

\[ \eta_{w-s} = \frac{T_0 S_{\text{gen}}}{Q} \]  
(17)

where in Eq. (16), subscripts $C$ and $H$ refer to cold and warm streams, respectively. Figure 5 illustrates the variations of $\varepsilon_R$ and $\eta_{w-s}$ versus outer-side Reynolds number. This figure shows that increasing of $Re_o$ leads to the decrease in both of these parameters which means that higher irreversibilities occur at higher air-side Reynolds numbers. This conclusion is compatible with the results of Figs. 3 and 4. This figure also reveals that the two mentioned irreversibility parameters show very little difference from each other.

Fig. 3 Variation of air-side entropy generation versus the air-side Reynolds number

Fig. 4 Variations of $\phi_1$, $\phi_2$ and $\phi_3$ with air-side Reynolds number
4.2 Effect of ambient temperature on air-side entropy generation

Figure 6 plots the entropy generation rate induced by pressure drop \((N_{\phi})_p\), temperature difference \((N_{\phi})_\Delta T\), water evaporation \((N_{\phi})_{ev}\), and the total air-side entropy generation rate \(N_{\phi}s\), versus ambient temperature \(T_0\). From Fig. 6 it is apparent that increasing the ambient temperature reduces the entropy generation due to temperature difference; because \(\Delta T\) between air flow and water-film is reduced. Also, \((N_{\phi})_p\) increases very slightly. By increasing of ambient temperature, \((N_{\phi})_{ev}\) is raised, severely. Increment of ambient temperature leads to water evaporation increase which results in the increase of entropy generation rate due to the water evaporation.

Figure 7 shows the effect of ambient temperature on the different irreversibility contributions. This figure illustrates that in low ambient temperature, most share of irreversibility is due to temperature difference. Also, by increasing \(T_0\), this contribution reduces while the share of irreversibility due water evaporation enhances. From this figure it can be concluded that at high ambient temperatures, the only effective mechanism of entropy generation is water evaporation.
The effect of ambient air temperature on the irreversibility parameters $\varepsilon_R$ and $\eta_{w-s}$ are represented in Fig. 8. In this figure it is visible that both of diagrams coincide. This figure illustrates that increasing $T_0$ leads to the decrease in both of these parameters which means that higher irreversibilities occur at higher ambient temperatures in an evaporative air-cooled heat exchanger. This conclusion confirms the results of Fig. 6 well.

4.3 Effect of deluge water mass flow rate on air-side entropy generation

Graphed in Fig. 9, are the variations of $(N_S)_P$, $(N_S)_T$, $(N_S)_e$, and $N_S$ versus deluge water mass flow rate. According to this figure, increasing of $\dot{m}_w$, does not affect the value of $(N_S)_P$, $(N_S)_T$ and $(N_S)_e$; therefore, $N_S$ does not change. Moreover, it can be concluded that air-side entropy generation does not change by the variation of $\dot{m}_w$.

Figure 10 reports the variations of $\varepsilon_R$ and $\eta_{w-s}$ with deluge water mass flow rate. This figure shows that the effect of $\dot{m}_w$ on $\varepsilon_R$ and $\eta_{w-s}$ is slight, hence by increasing deluge water mass flow rate, irreversibility does not change considerably.
5. Conclusion

An analytical study has been carried out to investigate the effect of various air-cooled heat exchanger parameters on the total air-side entropy generation in an evaporative air-cooled heat exchanger. The effects of several parameters such as air-side Reynolds number, ambient temperature and deluge water mass flow rate on the air-side entropy generation rate and its contributions to heat transfer, frictional loss and water evaporation have been studied. Investigating the variations of entropy generation rate with Reynolds number reveals that entropy generation is increased with the growth of Reynolds number. It was also found out that increasing of ambient temperature augments the entropy generation rate at high values of ambient temperature. In fact in this condition, entropy generation rate due to water evaporation is too high. Because, trying to decrease entropy generation due to heat transfer leads to the increase in water evaporation entropy generation. Furthermore, the increase in deluge water mass flow rate does not affect entropy generation rate, remarkably. Finally, it can be concluded that deluge water mass flow rate does not play a noticeable role in entropy generation rate.
References