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Exergy analysis of humidification–dehumidification desalination systems using driving forces concept

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ABSTRACT

In this paper, a new method for exergy analysis of humidification–dehumidification (HD) desalination systems is presented. It is based on the principle that there are exergy losses wherever the driving forces exist. A methodology was developed for investigating various parametric effects on exergy losses. The method involved developing a sink and source model as well as basic relations in the system. Results showed that the mass transfer phenomenon does not have any effect on the total exergy losses of the HD systems. Heater was the largest irreversibility resource. Flow rate of the Un-heated fluid and the maximum temperature of the system had key roles in the total exergy losses. An optimum point for the water-heated HD desalination system is also introduced. Finally, some comparisons are proposed between the water-heated and the air-heated HD desalination systems.

1. Introduction

While the world's population has tripled in the 20th century, the use of renewable water resources has grown six-fold. Within the next 50 years, the world population will increase by another 40 to 50%. This population growth – coupled with industrialization and urbanization – will result in an increasing demand for water and will have serious consequences for the environment. There are many countries that are severely influenced with regard to human health and inadequate drinking water. Humidification–dehumidification desalination is an innovative technology that has the potential for providing safe drinking water to the people of developing countries.

It is crucial in terms of energy and environmental considerations to apply a method through which highly efficient processes can be performed. The application of second law of thermodynamics can lead to the improvement in the quality of processes. In recent years, thermal systems have been analyzed and optimized using this law. Thermodynamic laws provide the designers and engineers with a powerful and efficient tool known as the exergy analysis. However, energy balance can provide useful information about the state of the system as far as the losses from the plant body and incomplete combustion are concerned. However, it does not display any difference between the types of energy nor does it consider the losses due to decrease in energy quality. By definition, exergy is referred to as the maximum shaft or electrical work in a reversible process when the system reaches the environment conditions. Exergy analysis makes it possible to provide methods for thermodynamic development. However, economical and environmental considerations will determine the selection of the ultimate choice.

There are massive literature materials on the subject. Bejan [1] and Bejan *et al.* [2], for example, linked the principles of heat transfer to the second law of thermodynamics and entropy generation.

There are two main phenomena in the HD desalination systems: 1— mass transfer and 2— heat transfer. Some researchers have focused on the second law analysis in the combined heat and mass transfer. Bejan [3, 4] created and motivated a large new body of knowledge linking second law analysis to heat and mass transfer processes. San et al. [5] showed that the temperature profile of the pure heat transfer problem can be utilized in all calculations for entropy generation in the combined heat and mass transfer in an air–water vapor system. Carrington and Sun performed the second law analysis for entropy generation due to heat and mass transfer processes in a multi-componential fluid. Demirel and sandler [6] proposed application of the linear non-equilibrium thermodynamics theory for the coupled heat and mass transfer. Muangnoi and Asvapoositkul [7] performed the exergy analysis of a counter-current flow wet cooling tower. Narayan et al. [8] introduced 'modified heat capacity rate ratio' in the combined heat and mass transfer devices and showed that the entropy generation of a combined heat and mass exchange device is minimized (at a constant value of effectiveness) when the modified heat capacity rate ratio is equal to one, irrespective of the value of other independent parameters.

Some other researchers have focused on the optimization of the HD desalination system. A number of them performed cost or performance optimization from energy point of view [10, 11]. A few researchers have previously attempted to apply the second law analysis of HD

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Nomenclature

Symbols

c	specific heat capacity (J/kg. K)
c	concentration (mol/L)
EL	Exergy losses (J)
EX	Exergy (J)
h_{fg}	Latent heat of vaporization (J/kg)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{n}$	Mole flow rate (gmol/s)
p	Pressure (Pa)
p^*	Vapor pressure (mmHg)
$\dot{q}$	Rate of heat transfer (W)
R	Ideal gas constant (J/mole K)
$\dot{S}_{gen}$	Rate of entropy generation (J/K)
T	temperature (K)
y	Mole fraction

Greek symbols

ω	Absolute humidity of dry air (kg_w/kg_a)
Δ	change or difference

Subscript

a	Air
D	Dehumidifier
da	Dry air
H	Humidifier
HE	Heater
LM	Log. Mean
m	Mean
p	Pressure constant
pw	Pure water
W	Water
wa	Water in air

desalination systems. Alhazmy [12] presented a theoretical analysis based on the second law of thermodynamics for estimating the minimum work required for air dehumidification process in order to produce potable water in a humidification–dehumidification HD desalination cycle. Narayan et al. [13] analyzed the thermodynamic performance of various HD systems using a theoretical cycle analysis. In addition, they proposed novel high-performance variations on those cycles, which included multi-extraction, multi-pressure and thermal vapor compression cycles. Mistry et al. [14] investigated the effect of entropy generation on the performance of HD desalination cycles. Narayan et al. [8] examined the concept of 'balancing' of combined heat and mass exchangers on the entropy generation of the HD desalination system. Recently, the design and analysis of systems is conducted by applying the concept of constructal theory proposed by A. Bejan. Mehrgoo and Amidpour [15] conducted the constructal design and optimized the HD desalination system by applying the concept of constructal theory. In their study, a procedure is developed to optimize the fresh water production rate by applying the Lagrange multipliers method.

Simultaneous heat and mass transfer leads to complicated effects in the systems. Interactions between heat and mass transfer are the most important factors in this complication. Linear-nonequilibrium thermodynamics (LNET) is the main tool for analyzing these systems. But in the literature such coupling has been formulated incompletely and sometimes in a confusing manner. The reason for this is the lack of a proper combination of LNET theory with the phenomenological theory. To reduce the complications, it is essential to apply more simple methods.

For a real process the exergy input always exceeds the exergy output, this unbalance is due to irreversibilities, which we name exergy destruction. The exergy output consists of the utilized output and the non-utilized output, i.e. exergy of waste output. This latter part we entitle the exergy waste. It is very important to distinguish between exergy destruction caused by irreversibilities and exergy waste due to unused exergy, i.e. exergy flow to the environment. In the literature, sum of the exergy destruction and exergy waste is known exergy losses. By calculating the exergy losses i.e. destruction and waste, we can visualize possible process improvements. Obviously, if the exergy waste is negligible, the exergy losses will equal to exergy destruction.

The common method to exergy analysis of thermal systems is stream-wise method in which there is a focus on the exergy of input and outputs. Obviously, the dealing method of one case is different from others. The first step of making the correct decision in dealing with irreversibility factors is to indicate the main sources of exergy losses. Driving forces are needed for performing real phenomena. These forces together with transfer coefficients, on the one hand, affect the rate of transfer phenomena and, given the relation of driving forces with irreversibilities, they influence the exergy losses on the other hand. The driving forces can be, therefore, used as a relation among the rate of transfer phenomena, transfer coefficients and exergy losses.

The originality of the paper is in the new looking at HD desalination systems' exergy analysis. The philosophy underlying such an approach is based on the principle that wherever there are driving forces, there are exergy losses and the total exergy losses in the system would be equal to the sum of the exergy losses due to these driving forces. In the present research, the relations of the exergy and state functions are combined so that it can be possible to include the effect of various parameters such as temperature and flow rates on the exergy losses of HD desalination systems. Another difference between this work and previous studies is the ability to separately compute the exergy losses caused by different kinds of irreversibilities, such as mass and heat transfer, in the system. Thus, the source of irreversibility will be comparable. In addition, exergy analyses based on the driving forces can reduce complication of the analysis and give some helpful results.

As we know, enthalpy, entropy and exergy are the state functions which dependence on the first and final points. Fig. 1 shows the manner which is used to calculation of the enthalpy change due to temperature and pressure variation.

It can be written:

$$dh = \left(\frac{\partial h}{\partial T}\right)_P dT + \left(\frac{\partial h}{\partial T}\right)_T dP \Rightarrow \Delta h_{\Delta P,\Delta T} = (\Delta h_{\Delta T})_{\Delta P=0} + (\Delta h_{\Delta P})_{\Delta T=0}. \quad (1)$$

Using the same method for exergy change calculation due to temperature and concentration variation, gives:

$$\Delta EX_{\Delta C,\Delta T} = (\Delta EX_{\Delta T})_{\Delta C=0} + (\Delta EX_{\Delta C})_{\Delta T=0}. \quad (2)$$

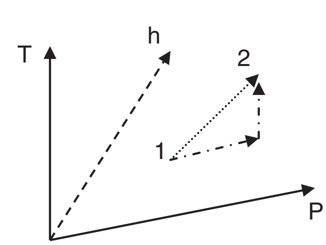

Fig. 1. Enthalpy change calculation due to temperature and pressure variation.

Eq. (2) offers the ability to separately calculation the exergy losses caused by mass and heat transfer, in the system. This method can be developed for other thermal systems.

2. System description

The HD desalination system is based closely on the natural water cycle. In this cycle, the sun's energy evaporates water from large bodies of water (oceans, seas and …) then the water vapor mixes with the air. When the air temperature drops, the amount of moisture content that the air can hold decreases and the vapor precipitates out as rain (fresh water). The same processes can be used in the HD desalination systems. Sun's energy or other heating sources can be used for evaporating of water; the moist air then condenses on a cooler surface and utilized as fresh water. By separating the evaporation and condensation processes and by incorporating regenerative heating, the efficiency of the system is improved. This is the foundation for the HD desalination systems. This system consist of three subsystems: (a) the air and/or water heater, which can use various sources of heat like solar, thermal or combinations of them; (b) the humidifier or evaporator and (c) the dehumidifier or condenser. Depending on how these three components are arranged, various types of systems can be formed. The cycles are classified based on the nature of the flow pattern and the flow which is heated.

The common systems are:

1- Close Air, Open Water, Water Heated (CAOW-WH) system
2- Close Air, Open Water, Air Heated (CAOW-AH) system

These two systems are shown in Figs. 2 and 3.

Direct contact air–water in the humidifier leads to the heat and mass transfer between them because there are driving forces for these two phenomena (temperature and concentration differences). In other words, heat and mass transfer both lead to the water changes in the phase. The temperature difference between the hot–humid air and cooled-salt feed water tubes leads to a heat transfer in the dehumidifier. This heat transfer causes the condensation of the vapor in the air and transfers the molecules of the water from gas to liquid phase. Therefore, there are mass and heat transfers in both humidifier and dehumidifier.

3. Sink–source model exergy analyses in heat transfer processes

In all heat transfer processes which take place above the environmental temperature, the higher and lower temperature heat sources can be considered as exergy source and sink, respectively. According to the second law of thermodynamics, exergy sink cannot absorb

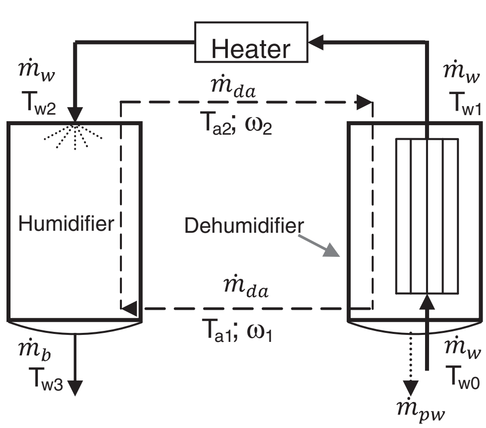

Fig. 2. Close Air-Open Water, Water Heated (CAOW-WH) system.

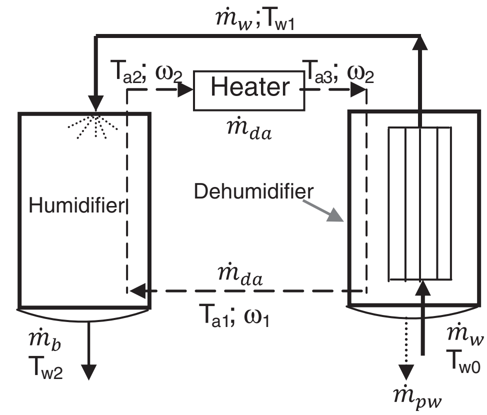

Fig. 3. Close Air–Open Water, Air Heated (CAOW-AH) system.

the whole source exergy. Thus, in the sink–source model, exergy losses are defined as follows:

$$EL_{\Delta T} = \Delta EX_{source_{\Delta T}} - \Delta EX_{sink_{\Delta T}} \tag{3}$$

where, $EL_{\Delta T}$ is the exergy loss, $\Delta EX_{source_{\Delta T}}$ and $\Delta EX_{sink_{\Delta T}}$ are the source and sink exergy change due to heat transfer respectively. Consider the two heat sources having different temperatures ($T_1 > T_2$) and the heat transfer rate between them equaling "$\dot{q}$". Based on these, the flowing equations can be written in order to calculate the exergy variation:

$$\Delta EX_{source_{\Delta T}} = \dot{q}\left(1 - \frac{T_0}{T_1}\right) \tag{4}$$

$$\Delta EX_{sink_{\Delta T}} = \dot{q}\left(1 - \frac{T_0}{T_2}\right) \tag{5}$$

where, T_0 is the environment temperature. Inserting Eqs. (4) and (5) into Eq. (3) gives the flow relation:

$$EL_{\Delta T} = \dot{q}\, T_0 \left(\frac{T_1 - T_2}{T_1 T_2}\right). \tag{6}$$

It proves that, under circumstances where the temperature of sink and source during the process varies, their logarithmic mean temperature can be used in Eqs. (4), (5) and (6).

4. Exergy losses due to concentration change for ideal gas mixture

Standard chemical exergy of ideal gas mixture can be calculated by Eq. (7) [2].

$$e^{-CH} = \sum x_k e_k^{-CH} + RT_0 \sum \ln x_k \tag{7}$$

where, e^{-CH} is the Standard chemical exergy of ideal gas mixture and x_k is the mole fraction of the components.

Consider a binary mixture of ideal gas in contact with a liquid phase. There is a change in gas concentration due to mass transfer between phases. It is assumed that the temperature and pressure are constant and equal to T_0 and P_0. There is no chemical reaction and the change in liquid concentration is negligible. Fig. 4 shows this system schematically.

Exergy losses due to mass transfer in the above system can be calculated by Eq. (8):

$$EL = EX_{in} - EX_{out}. \tag{8}$$

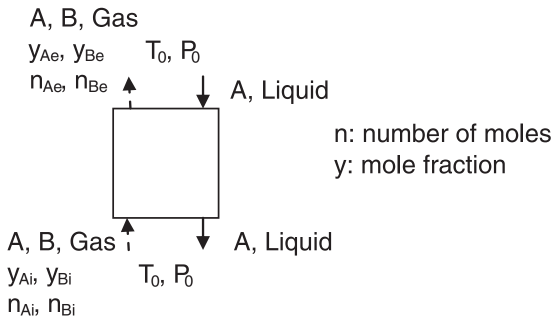

Fig. 4. Ideal gas mixture mass transfer with a liquid phase.

Eq. (9) and (10) give input and output exergy of the system, respectively.

$$\mathrm{EX_{in}} = RT_0[\mathrm{n_{Ai}Ln(y_{Ai}) + n_{Bi}Ln(y_{Bi})}] = RT_0 Ln\left(\mathrm{y_{Ai}}^{\mathrm{n_{Ai}}} \times \mathrm{y_{Bi}}^{\mathrm{n_{Bi}}}\right) \tag{9}$$

$$\mathrm{EX_{out}} = RT_0[\mathrm{n_{Ae}Ln(y_{Ae}) + n_{Be}Ln(y_{Be})}] = RT_0 Ln\left(\mathrm{y_{Ae}}^{\mathrm{n_{Ae}}} \times \mathrm{y_{Be}}^{\mathrm{n_{Be}}}\right) \tag{10}$$

Substituting Eqs. (9) and (10) in Eq. (8), gives:

$$\mathrm{EL} = RT_0 Ln\left(\frac{\mathrm{y_{Ai}}^{\mathrm{n_{Ai}}} \times \mathrm{y_{Bi}}^{\mathrm{n_{Bi}}}}{\mathrm{y_{Ae}}^{\mathrm{n_{Ae}}} \times \mathrm{y_{Be}}^{\mathrm{n_{Be}}}}\right) \tag{11}$$

Eq. (11) gives the exergy losses due to concentration change for binary ideal gas.

5. Exergy analysis of HD desalination system

In this analysis, the following approximations were made:

1- The system condition was taken to be steady state–steady flow.
2- All calculations were performed for atmospheric pressure.
3- Changes of kinetic and potential energy were neglected.
4- Exergy losses due to fluids' flow friction were neglected.
5- Heat transfer between the system and environment was negligible.
6- The moist air streams were taken to be saturated at the exit of both humidifier and dehumidifier.
7- The concentration change for the salt water was negligible.
8- The gas phase was considered as ideal gas.

Total exergy losses of the system (EL_{total}) were the sums of the exergy losses in the humidifier (EL_H), dehumidifier (EL_D) and heater (EL_{HE}), then:

$$EL_{\mathrm{total}} = EL_H + EL_D + EL_{HE}. \tag{12}$$

5.1. Humidifier

There are two sources for exergy losses in the humidifier: 1– mass transfer and 2– heat transfer. There are complex flows inside the column which transfer heat and mass simultaneously. These two phenomena have complicated interactions. Since exergy is a state function, the exergy losses due to mass and heat transfer can be calculated separately:

$$EL_H = EL_{H_{\Delta T}} + EL_{H_{\Delta C}} \tag{13}$$

where, $\mathrm{EL}_{\mathrm{H}_{\Delta T}}$ and $\mathrm{EL}_{\mathrm{H}_{\Delta C}}$ are exergy losses due to heat and mass transfer respectively in the humidifier. According to the sixth approximation, the rate of heat transfer in the humidifier equals to rate of heat vaporization $\left(\dot{q} = \dot{m}_{\mathrm{pw}} \mathrm{h}_{\mathrm{fg_{water}}}\right)$. Where, $\mathrm{h}_{\mathrm{fg_{water}}}$ is the water heat of vaporization. Then sink–source model can be used for calculating the exergy losses due to heat transfer as below:

$$EL_{H_{\Delta T}} = \dot{m}_{pw} h_{fg_{water}} T_0 \left(\frac{T_{LM_{water}} - T_{LM_{air}}}{T_{LM_{water}} \times T_{LM_{air}}}\right)_H \tag{14}$$

where, $T_{LM_{water}}$and $T_{LM_{air}}$ are water and air logarithmic mean temperature in the humidifier, respectively, which can be calculated by Eq. (15) and (16).

$$\left(T_{LM_{water}}\right)_H = \begin{cases} \dfrac{(T_{w3} - T_{w2})}{Ln\left(\dfrac{T_{w3}}{T_{w2}}\right)} & ,(\mathrm{CAOW-WH}) \\ \dfrac{(T_{w2} - T_{w1})}{Ln\left(\dfrac{T_{w2}}{T_{w1}}\right)} & ,(\mathrm{CAOW-AH}) \end{cases} \tag{15}$$

$$\left(T_{LM_{air}}\right)_H = \frac{(T_{a2} - T_{a1})}{Ln\left(\frac{T_{a2}}{T_{a1}}\right)} \tag{16}$$

where, the notations are in correspondence with Figs. 2 and 3. According to the two last approximations, and Eq. (11), the exergy losses due to mass transfer in the humidifier can be calculated by Eq. (17).

$$EL_{\Delta c} = RT_0 Ln\left(\frac{y_{da1}^{\dot{n}_{da}} \cdot y_{wa1}^{\dot{n}_{wa1}}}{y_{da2}^{\dot{n}_{da}} \cdot y_{wa2}^{\dot{n}_{wa2}}}\right) \tag{17}$$

where, $\dot{n}$ and y are mole flow rate and mole fraction respectively. Subscripts "da" and "wa" denoted "dry air" and "water in air" respectively. The parameters in Eq. (17) can be calculated using Eqs. (18), (19) and (20)

$$\dot{n}_{da} = \frac{\dot{m}_{da}}{29} \quad ,(air\ molecular\ wieght = 29) \tag{18}$$

$$\dot{n}_{wai} = \frac{\dot{m}_{da} \times \omega_{ai}}{18} \quad ,(water\ molecular\ wieght = 18) \tag{19}$$

$$y_{dai} = \dot{n}_{da}/(\dot{n}_{da} + \dot{n}_{wai}), \quad y_{wai} = 1 - y_{dai}. \tag{20}$$

5.2. Dehumidifier

Similar to the humidifier, heat and mass transfers are irreversibility sources in the dehumidifier. So, the following can be written:

$$EL_D = EL_{D_{\Delta T}} + EL_{D_{\Delta C}} \tag{21}$$

where, $EL_{D_{\Delta T}}$ and $EL_{D_{\Delta C}}$ are exergy losses due to heat and mass transfer respectively in the dehumidifier. In the (CAOW−WH) systems, the moist air entering in the dehumidifier can be assumed saturated. Then the rate of heat transfer in the dehumidifier equals to rate of heat vaporization $\left(\dot{q} = \dot{m}_{\mathrm{pw}}, \mathrm{h}_{\mathrm{fg_{water}}}\right)$. Where, $\mathrm{h}_{\mathrm{fg_{water}}}$ is the water heat of vaporization. Then the exergy losses due to heat transfer by sink–source model can be calculated as below:

$$EL_{D_{\Delta T}} = \dot{m}_{pw} h_{fg_{water}} T_0 \left(\frac{T_{LM_{air}} - T_{LM_{water}}}{T_{LM_{air}} \times T_{LM_{water}}}\right)_D ,(\mathrm{CAOW-WH}) \tag{22}$$

But, in the (CAOW−AH) systems, heating air before the dehumidifier leads to superheating the moist air. Therefore, there is a sensible heat transfer for the moist air until its temperature reaches the saturated point (humidifier outlets temperature). Water condensation starts after this point. Then, using the sink–source model for heat transfer exergy losses, the following can be presented:

$$EL_{D_{\Delta T}} = \left[\dot{m}_{da}(1+\omega_2)(1+1.8\omega_2)(T_{a3}-T_{a2}) + \dot{m}_{pw}h_{fg_{water}}\right]T_0\left(\frac{T_{LM_{air}}-T_{LM_{water}}}{T_{LM_{air}}\times T_{LM_{water}}}\right)_D, \text{(CAOW−AH)} \quad (23)$$

The first term in the bracket is the sensible heat. The quantity of "$(1+\omega_2)(1+1.8\omega_2)$" is the heat capacity of the moist air (J/kg dry air.K). "$T_{LM_{water}}$" and "$T_{LM_{air}}$" are the water and air logarithmic mean temperature in the dehumidifier, respectively, which can be calculated using Eqs. (24) and (25).

$$\left(T_{LM_{air}}\right)_D = \begin{cases} \dfrac{(T_{a1}-T_{a2})}{Ln\left(\dfrac{T_{a1}}{T_{a2}}\right)} & \text{, (CAOW−WH)} \\ \dfrac{(T_{a1}-T_{a3})}{Ln\left(\dfrac{T_{a1}}{T_{a3}}\right)} & \text{, (CAOW−AH)} \end{cases} \quad (24)$$

$$\left(T_{LM_{water}}\right)_D = \frac{(T_{w1}-T_{w0})}{Ln\left(\frac{T_{w1}}{T_{w0}}\right)} \quad (25)$$

According to the two last approximations, and Eq. (11), the exergy losses due to mass transfer in the dehumidifier can be obtained using Eq. (26).

$$EL_{\Delta c} = RT_0 Ln\left(\frac{y_{da2}^{\dot{n}_{da}}\cdot y_{wa2}^{\dot{n}_{wa2}}}{y_{da1}^{\dot{n}_{da}}\cdot y_{wa1}^{\dot{n}_{wa1}}}\right). \quad (26)$$

It is notable that the mole numbers and fractions of moist air components before and after the heater in the (CAOW–AH) system are equal. Then, Eq. (26) is usable for this system. The parameters in Eq. (26) can be obtained using Eqs. (18), (19) and (20).

A comparison of Eqs. (17) and (26) shows that the sum of the mass transfer exergy losses in the humidifier and dehumidifier is zero. Therefore, it can be concluded that the mass transfer does not have any effect on the total exergy losses in the HD desalination systems.

5.3. Heater

The exergy losses due to heat transfer in the heater can be calculated by Eq. (27).

$$EL_{HE} = T_0\dot{S}_{gen_{heater}}. \quad (27)$$

According to the fourth, fifth and eighth approximations entropy generation in the heater can be calculated by Eq. (28).

$$\dot{S}_{gen_{heater}} = \dot{m}_{fluid}c_{pm}Ln\left(\frac{T_{out}}{T_{in}}\right) \quad (28)$$

where, $\dot{m}_{fluid}$ and c_{pm} are the mass flow rate and mean specific heat capacity of the fluid which is passed along the heater, T_{out} and T_{in} are outlet and inlet temperature of the fluid respectively. Then the exergy losses in the heater can be obtained from Eqs. (29) and (30).

$$EL_{HE} = \dot{m}_w c_{p_w} T_0 Ln\left(\frac{T_{w2}}{T_{w1}}\right), \text{(CAOW−WH)} \quad (29)$$

$$EL_{HE} = \dot{m}_{da}(1+\omega_2)(1+1.8\omega_2)T_0 Ln\left(\frac{T_{a3}}{T_{a2}}\right), \text{(CAOW−AH)} \quad (30)$$

The total exergy losses can be calculated by Eqs. (13) to (16), (21), (22), (24), (25) and (29) for the (CAOW- WH) system and Eqs. (13) to (16), (21), (23), (24), (25) and (30) can be used for (CAOW- AH) system. The required parameters for these calculations are the temperatures, composition and flow rates for air and water streams.

As can be seen in Figs. 2 and 3 and the obtained relations, twelve parameters must be identified for calculating the exergy losses in the system. If the pure water flow rate and the salt feed water temperature are known, these parameters will be reduced to ten numbers. The governing equations are well described by Narayan *et al.* [13], (Eqs. 1 to 8 in reference [13]). Here, two other relations are introduced by combining the Raoults' law and the Antoines' equation for the water–air solution.

The Antoines' equation gives the water vapor pressure and temperature relation as below [16]:

$$LnP^*_{H_2O} = 18.3036 - \frac{3816.44}{-46.13+T} \quad (31)$$

where, $P^*_{H_2O}$ is the vapor pressure of the water at the given temperature (T). According to the sixth assumption, the raoults' law [17] is true for the outlet air from the humidifier and dehumidifier.

$$y_{H_2O} = \frac{P^*_{H_2O}}{P_{total}}. \quad (32)$$

The relation between the absolute humidity of the air (ω) and mole fraction of the water (y_{H_2O}) is given by Eq. 33.

$$\omega = \frac{18y_{H_2O}}{29\left(1-y_{H_2O}\right)}. \quad (33)$$

Combining Eqs. (31) through (33) gives Eq. (34).

$$\omega_{ai} = \frac{18\times\exp[18.3036-3816.44/(-46.13+T_{ai})]}{29\times\{760-\exp[18.3036-3816.44/(-46.13+T_{ai})]\}}, (i=1,2). \quad (34)$$

Eq. (34) can be applied for air output from the humidifier and dehumidifier. Then, two new equations are resulted.

A model has been developed for investigation of various parameter effects on exergy losses in the MATLAB environment. A comparison between the model results and a case in the reference [9] (Fig. 13 in this reference) is shown by Table 1:

The parameters in the above table are in correspondence with Fig. 3.

6. Results and discussion

The first step of the exergy analysis is to identify the magnitudes and locations of real exergy losses in order to improve the existing

Table 1
Comparison between the model outputs and a case in the reference [9]*.

Parameters	The model results	Given by ref. [9]
Sea water flow rate (**kg/s**)	**0.1607**	**0.15**
Dehumidifier water outlet temperature (**°C**)	**60**	**62.79**
Brain water flow rate (**kg/s**)	**0.1548**	**0.144**
Humidifier water outlet temperature (**°C**)	**37**	**37.05**
Dry air flow rate (**kg/s**)	**0.0959**	**0.1**
Dehumidifier air outlet temperature (**°C**)	**34**	**34.2**
Humidifier air outlet temperature (**°C**)	**51**	**51.4**
Heater air outlet temperature (**°C**)	**90**	**90**
Bottom air absolute humidity (**kg_w/kg_a**)	**0.0339**	**Is not given.**
Top air absolute humidity (**kg_w/kg_a**)	**0.0955**	**Is not given.**

*The model input data (equal to the quantities of Fig. 13 in the reference [9]):
$\dot{m}_{pw} = 0.0059$Kg/sec.
$T_{w0} = 30^oC$.

systems, processes or components. Figs. 5 to 8 show the exergy losses' distribution for CAOW-HD desalination systems (the parameters in all figures are in correspondence with Figs. 2 and 3).

Figs. 4 to 7 show that the heater is the largest irreversibility resource in the HD desalination systems. Dehumidifier and humidifier have the second and third quantity of the exergy losses, respectively. Therefore, the heater's performance improving has the highest effect on the system efficiency. These figures also show that the percent of the heater in the exergy losses increases by the reduction of the maximum temperature of the system. This temperature is the outlet fluid temperature from the heater which is equal to 'T_{w2}' and 'T_{a3}' in the water-heated and air-heated systems, respectively. From these figures, it can be seen that the dehumidifier exergy losses' percentage in the air-heated systems is more than the one in the water-heated systems. This can be predicted because the dehumidifier air inlet is superheated and needs some sensible heat transfer for reaching the saturated point and starting condensation. However, there is not this sensible heat transfer in the water-heated systems because the dehumidifier air inlet is saturated. Therefore, for equal pure water production, the AH-system needs more heat transfer in the dehumidifier.

The top temperature of the fluid which does not pass in the heater (un-heated fluid) is an effective factor on the heater performance. For the water-heated system, this temperature is the humidifier outlet air temperature (t_{a2}). Figs. 9 and 10 show this effect on exergy losses and gained-output-ratio (GOR), for a case of CAOW-WH- HD system.

It is notable that the rate of the humidifier exergy losses against the maximum temperature is inverse with respect to the same one for the dehumidifier. On the other hand, the heater exergy losses' reduction is the only reason for improving the total system.

As can be seen, increasing the top temperature of the un-heated fluid has positive effects on the system. The most important reason for this effect is the heater load reduction.

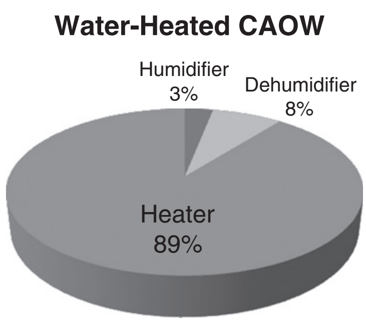

Fig. 5. Exergy losses distribution in the CAOW-WH systems, $T_{w0} = 30$ °C; $T_{w2} = 90$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

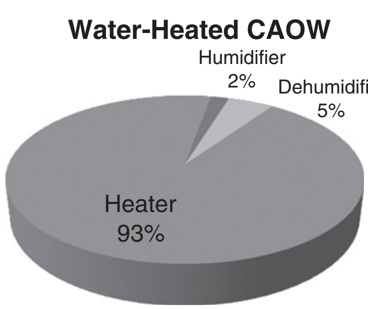

Fig. 6. Exergy losses distribution in the CAOW-WH systems, $T_{w0} = 30$ °C; $T_{w2} = 80$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

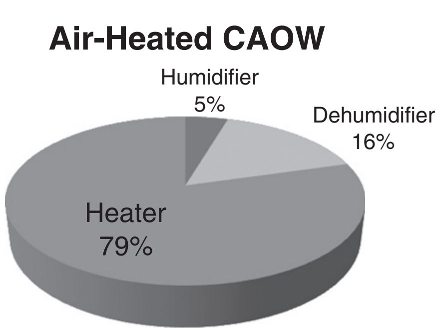

Fig. 7. Exergy losses distribution in the CAOW-AH systems, $T_{w0} = 30$ °C; $Ta_3 = 90$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

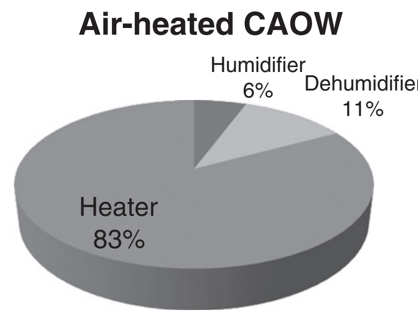

Fig.8. Exergy losses distribution in the CAOW-AH systems, $T_{w0} = 30$ °C; $T_{a3} = 80$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

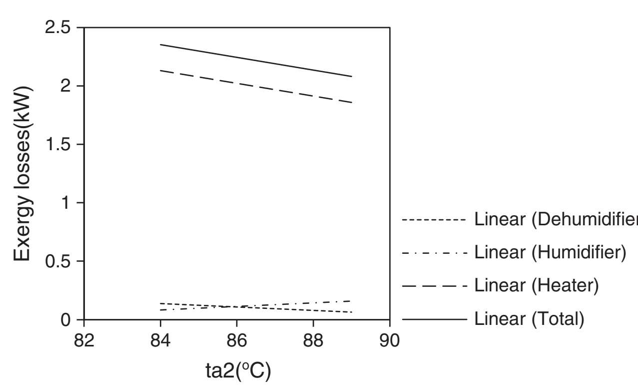

Fig. 9. Effect of humidifier air outlet temperature on water-heated CAOW system exergy losses, $T_{w0} = 30$ °C; $T_{w2} = 90$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

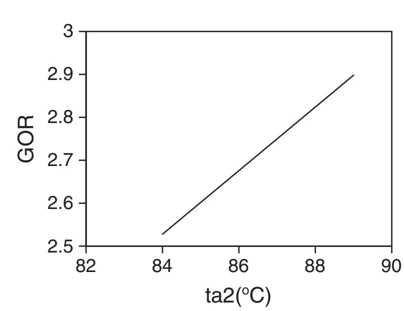

Fig. 10. Effect of humidifier air outlet temperature on water-heated CAOW system gained output ratio, $T_{w0} = 30$ °C; $T_{w2} = 90$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

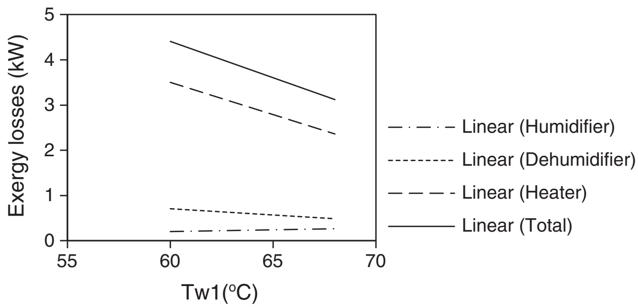

Fig. 11. Effect of dehumidifier water outlet temperature on air-heated CAOW system exergy losses, $T_{w0} = 30$ °C; $T_{a3} = 90$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

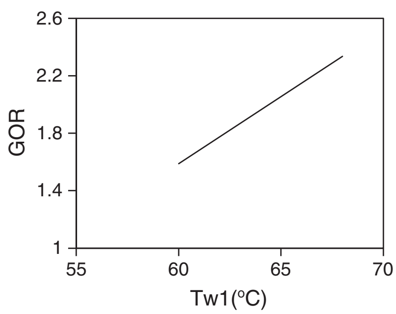

Fig. 12. Effect of dehumidifier water outlet temperature on air-heated CAOW system gained output ratio, $T_{w0} = 30$ °C; $T_{w2} = 90$ °C ; $\dot{m}_{pw} = 0.005\ kg/s$.

In air-heated systems, dehumidifier water outlet temperature is an effective factor on the heater exergy losses. Figs. 11 and 12 show the same results for this system.

According to Figs. 9 to 12, under conditions which the pure water flow rate, maximum temperature and water inlet temperature are identified, the flow rate of the un-heated fluid should be minimized. On the other hand, the air and water flow rates should be minimized for water-heated and air-heated systems, respectively.

Another effective factor on the exergy losses of the HD desalination systems is the maximum temperature in the system. The heater outlet fluid temperature is this factor (T_{w2} and T_{a3}). Figs. 13 and 14 show the effect of this factor on exergy losses and the gained-output-ratio for water- and air- heated CAOW systems.

As can be seen, the exergy losses are reduced and 'GOR' is increased by increasing the maximum temperature of the system. This shows that higher temperature has positive effects on the system performance. Another result from Figs. 13 and 14, mentioned in Narayan et al. [15], is that the air-heated system has higher performance than the water-heated system in the same conditions.

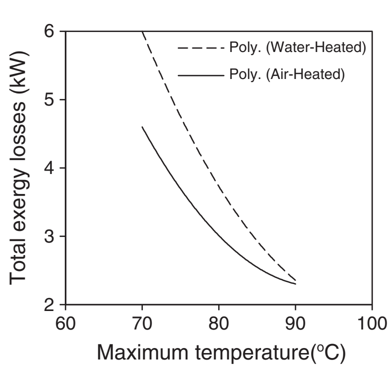

Fig. 13. Effect of maximum temperature on CAOW systems exergy losses. $T_{w0} = 30$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

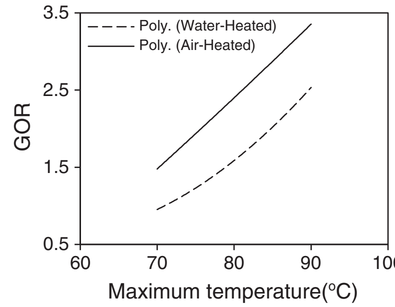

Fig. 14. Effect of maximum temperature on CAOW systems gained output ratio, $T_{w0} = 30$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

Yet another effective factor is the input water temperature. Fig. 15 shows the effect of this factor on the exergy losses, gained-output-ratio and feed water flow rate in the water-heated CAOW system for the maximum temperature equaling 90 °C.

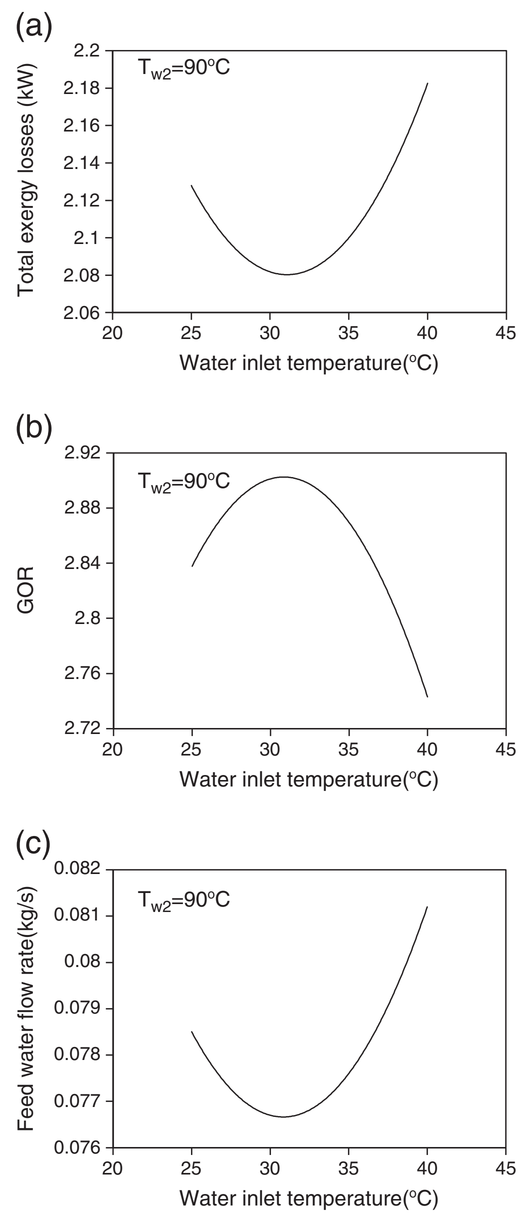

Fig. 15. Water inlet temperature effect on water-heated CAOW system (a) exergy losses (b) performance (c) feed flow rate, $T_{max} = 90$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

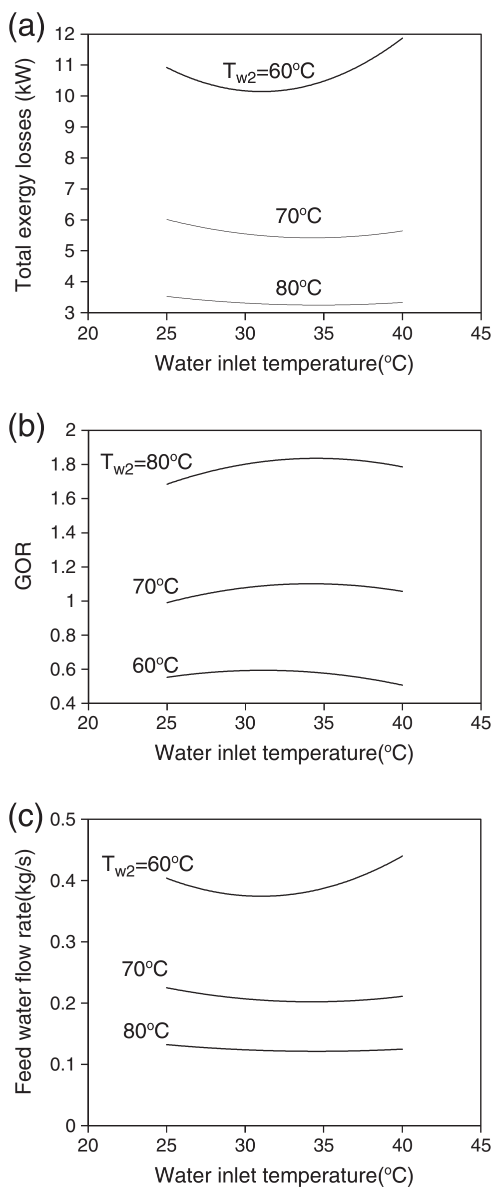

Fig. 16. Water inlet temperature effect on water-heated CAOW system (a) exergy losses (b) performance (c) feed flow rate, $\dot{m}_{pw} = 0.005\ kg/s$.

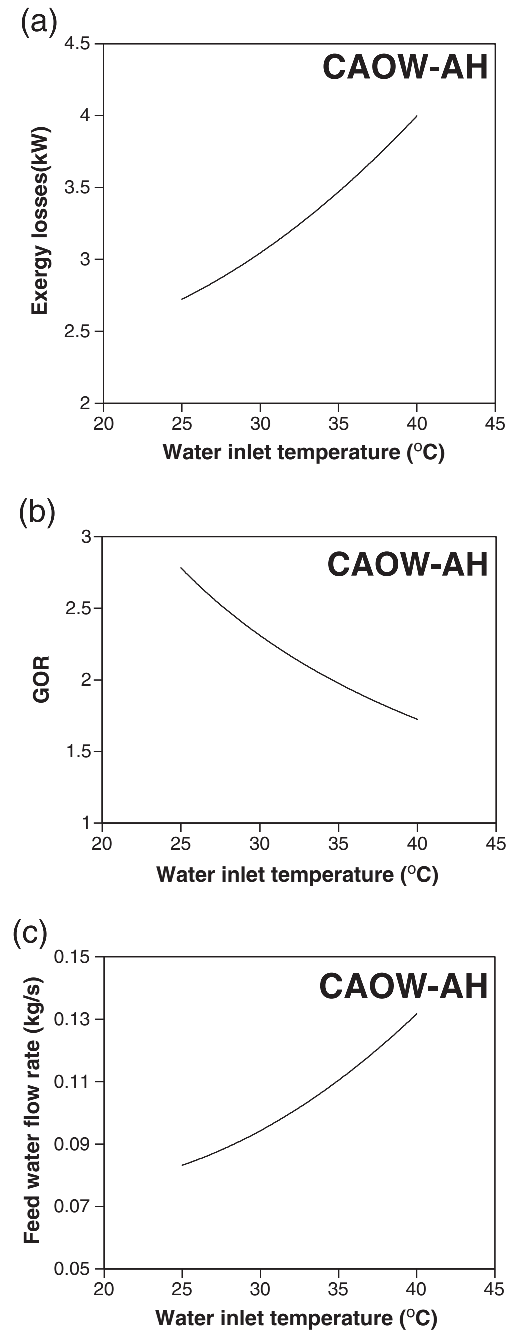

Fig. 17. Water inlet temperature effect on air-heated CAOW system (a) exergy losses (b) performance (c) feed flow rate, $T_{max} = 80$ °C; $\dot{m}_{pw} = 0.005\ kg/s$.

As can be seen, there is an optimum point for the input water temperature. This point is between 30 and 35 °C. Fig. 16 shows the effect of the input water temperature for other maximum temperatures.

The air-heated systems have different behaviors against the water inlet temperature. Fig. 17 shows the exergy losses, gained-output-ratio and feed water flow rate versus T_{w0} in these systems.

As can be seen, the inlet water temperature increase has a negative effect on the air-heated systems. One may be surprised by this result. But, it must be considered that the rate of condensation in the dehumidifier will be reduced by this rising. Then, more flow rates of the water and air are needed for a constant pure water flow rate. More air flow rate leads to more load of the heater and more exergy losses. However, the inlet water temperature increase leads to the heater's load reduction.

7. Conclusion

Exergy losses' relations were developed for CAOW-WH and CAOW-AH-HD desalination systems based on individually irreversibility factors. The new approach leads to following results:

1. The new approach adapted to the exergy analysis of HD desalination system in terms of driving forces found new simple equations for analyzing exergy in the system, which simplifies and reduces the complication of the analysis.
2. Through the methodology, the exergy losses caused by different kinds of irreversibilities, such as mass and heat transfer can be calculated in the HD systems separately. Thus, the sources of irreversibility are identified.
3. The mass transfer phenomenon does not have any effect on the total exergy losses of the HD systems because it has equal inverse directions in the humidifier and dehumidifier.
4. The heater is the largest irreversibility resource while the dehumidifier and humidifier have the second and third percent of the total exergy losses in the HD systems.

5. The percentage of exergy losses of the heater will increase if the maximum temperature of the system decreases.
6. The dehumidifier in the air-heated systems has a larger role than the water-heated systems in the irreversibilities.
7. For the heater exergy losses' reduction, the flow rate of the unheated fluid should be minimized.
8. The exergy losses decrease and the gained-output-ratio increase with the increasing the maximum temperature.
9. In the same conditions, the air-heated system has higher performance and lower exergy losses than the water-heated system.
10. For a certain pure water flow rate and the maximum temperature, there is an optimum point for CAOW-WH DH systems with respect to the water inlet temperature. In this point, the exergy losses and water flow rate are at the minimum points and the gained-output-ratio is at the maximum point.
11. For a certain pure water flow rate and the maximum temperature, inlet water temperature increase has a negative effect on the air-heated systems. The reason of this is that the rate of condensation in the dehumidifier will be reduced and more flow rates of the water and air are needed for a constant pure water flow rate.

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