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Second-law-based performance evaluation of cooling towers and evaporative heat exchangers

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Abstract

In this paper, we present thermodynamic analysis of counter flow wet cooling towers and evaporative heat exchangers using both the first and second laws of thermodynamics. A parametric study is carried out to determine the variation of second-law efficiency as well as exergy destruction as a function of various input parameters such as inlet wet bulb temperature. Irreversible losses are determined by applying an exergy balance on each of the systems investigated. In this regard, an engineering equation solver (EES) program, with built-in functions for most thermodynamic and transport properties, is used. The concept of total exergy as the sum of thermomechanical and chemical parts is employed in calculating the flow exergies for air and water vapor mixtures. For the different input variables investigated, efficiencies were, almost always, seen to increase or decrease monotonically. We notice that an increase in the inlet wet bulb temperature invariably increases the second-law efficiency of all the heat exchangers. Also, it is shown that Bejan's definition of second-law efficiency is not limited in evaluating performance. Furthermore, it is understood that the variation in the dead state does not significantly affect the overall efficiency of the system.

Keywords: Exergy; Evaporative cooler; Evaporative condenser; Dead state

1. Introduction

Energy exists in different forms. It is a measure of quantity but an energy source cannot be evaluated on its quantity alone. A measure of the quality of energy is defined as *exergy*, which is the work potential of energy in a given environment [1]. Exergy (or availability) analysis is defined as a method of performing system analysis according to the conservation of mass, conservation of momentum and second law of thermodynamics. It consists of using the first and second law together, for the purpose of analyzing performance in the reversible limit, and estimating the departure from this limit [2]. We note that it is exergy, not energy that represents the true potential of a system to perform an optimal work with respect to a dead state or surrounding. The greater the difference between the energy source and its surroundings, the capacity to extract work from the system increases. It is important to understand that before analyses can be applied with confidence to engineering systems, the significance of the sensitivities of exergy analysis results to reasonable variations in or selection of dead state properties should be evaluated. Szargut et al. [3] introduced the concept of reference substances different for every chemical element that are most common in the real environment. Different dead-states have been used in humid-air exergy calculations [4–6]. Although the exergy of a particular stream is very sensitive to the values assigned to the dead-state conditions, this might not be the case with the overall system performance. We emphasize that exergy destruction represents the waste of energy resources. Therefore, the method of exergy analysis aims at the quantitative evaluation of the exergy destruction associated with a system since the values of the rates of exergy destruction provide direct measure of thermodynamic system inefficiencies. Exergy analysis additionally often involves the calculation of the system performance in the form of second-law efficiency. When defining second-law efficiency, it should be kept in mind that the definition must be meaningful from both the thermodynamic and economic viewpoints.

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Nomenclature

A	outside surface area of cooling tubes	m^2
A_V	surface area of water droplets per unit volume of the tower	$m^2\,m^{-3}$
c_p	specific heat capacity at constant pressure	$kJ\,kg^{-1}\,K^{-1}$
d	diameter	m
EES	engineering equation solver	
h	specific enthalpy	$kJ\,kg^{-1}$
h_c	convective heat-transfer coefficient of air	$kW\,m^{-2}\,K^{-1}$
h_D	convective mass-transfer coefficient	$kg_w\,m^{-2}\,s^{-1}$
$h_{f,w}$	specific enthalpy of water evaluated at t_w	$kJ\,kg_w^{-1}$
h_g	specific enthalpy of saturated water vapor	$kJ\,kg_w^{-1}$
h_g^0	specific enthalpy of saturated water vapor evaluated at 0 °C	$kJ\,kg_w^{-1}$
k	thermal conductivity	$kW\,m^{-1}\,K^{-1}$
HVAC	heating ventilation and air conditioning	
L	length of tube	m
Le	Lewis number	
m	mass	kg
$\dot{m}$	mass flow rate	$kg\,s^{-1}$
n_{tr}	number of tube rows	
N	number of moles	mol
$\dot{N}$	molal flow rate	$mol\,s^{-1}$
P	pressure	kPa
$\dot{Q}$	rate of heat transfer	kW
R	ideal gas constant	$kJ\,kg^{-1}\,K^{-1}$
RDS	restricted dead state	
s	specific entropy	$kJ\,kg^{-1}\,K^{-1}$
$\dot{S}_{gen}$	rate of entropy generation	kW
t	temperature,	°C
T	temperature	K
U	overall heat transfer coefficient	$kW\,m^{-2}\,K^{-1}$
V	volume of tower	m^3
x	specific flow exergy	$kJ\,kg^{-1}$
$\bar{x}$	specific molal flow exergy	$kJ\,kmol^{-1}$
$\dot{X}$	rate of exergy transport	kW
y	mole fraction	
$()^*$	properties evaluated at the restricted dead state	

Greek symbols

η_{II}	second-law efficiency	
ω	specific humidity ratio	$kg_{water}\,kg_{air}^{-1}$
$\tilde{\omega}$	mole fraction ratio	$kmol_{water}\,kmol_{air}^{-1}$
ϕ	relative humidity	
μ	chemical potential	$kJ\,kmol^{-1}$

Subscripts

a	moist air
da	dry air
ch	chemical
D	destruction
in	inlet
is	inside
o	dead or reference state
os	outside
out	outlet
p	process fluid
Q	heat transfer
r	refrigerant
st	steam
t	tube
tot	total
v	water vapor
W	mechanical power
Wep	definition used by Wepfer et al.
w	water
wb	wet bulb
x	thermomechanical

Various references [5–9] contain examples that illustrate application of the second law of thermodynamics to a variety of Heating Ventilating and Air-Conditioning (HVAC) processes. In these references, a ratio of exergy of the products to the exergy supplied was used to measure the second-law efficiency of the processes. This was found to be confusing; for example, certain quantities were not used in the calculations even though they were contributing to the overall efficiency of the process. Specifically, in the steam-spray humidification process discussed by Wepfer et al. [6], efficiency was seen to become negative under certain operating conditions, which will be demonstrated later. Krakow [10] and Kestin [11] used the actual and ideal cycle values to estimate how well the actual cycle approaches a thermodynamic perfection. Akau and Schoenhals [12] described various methods for calculating the second-law efficiency for a heat pump system using water as a heat source and a heat sink. The second-law efficiency was defined as the ratio of the required minimum energy input for an ideal system to the actual energy input of a real system when achieving the desired task, which, in effect, was essentially the same definition used by Krakow [10] and Kestin [11]. Bejan [2] defined the second-law efficiency as a ratio of the total exergy leaving the system to the total exergy entering the system, which confines the efficiency between 0 and 1. Furthermore, he demonstrated, through many examples, the method of exergy analysis when considering various open and closed systems as well as some psychrometric processes. Lu et al. [13] performed an exergetic analysis of cooling systems with ozonation water treatment with the purpose of calculating exergy changes under varying conditions. It was shown that the minimum change in exergy with respect to the entering temperatures of air and water does exist. Chengqin et al. [14] evaluated four evaporative cooling schemes and, in the case of cooling and dehumidification, second-law efficiency was found to be about 45%, which was based on the definition that the exergy efficiency is the ratio of the exergy produced to the exergy depleted. This efficiency is 20% higher than that found by Wepfer. Kanoglu et al. [15] performed an exergetic analysis of a desiccant cooling system in ventilation mode. The exergy efficiency of the evaporative coolers in the system was defined as the ratio of the exergy recovered to the exergy input, which led to efficiencies ranging from 15 to 58%. Dincer and Sahin [16] found exergy efficiency to be about 10% in the case of a drying process. The exergy efficiency for this drying process was defined as the ratio of exergy use (investment) in the drying of the product to exergy of the drying air supplied to the system. It is important to note that the definition of exergy efficiency in the above studies is not consistent. Therefore, it is appropriate to use the concept presented by Bejan [2] as a common basis for evaluating thermodynamic performance of thermal systems.

The objective of this paper is to present second-law-based evaluation of cooling towers and evaporative heat exchangers under varying operating conditions. In this regard, we first review the necessary thermodynamic equations required for the exergy analysis of open systems, which is then followed by an analysis of the selected heat exchangers using an engineering-equation solver (EES) program [17,18]. Then the effect of input variables and selection of the dead state is studied.

2. Analytical framework

For a steady-state steady-flow system, the work transfer rate equation in a generalized form for an open system that also experiences mass-transfer interactions with the environment can be written as [2]

$$\dot{X}_W = \sum_{i=1}^{n}(\dot{X}_Q)_i + \sum_{j=1}^{q}(\dot{N}\bar{x}_t)_j - \sum_{k=1}^{r}(\dot{N}\bar{x}_t)_k - T_o\dot{S}_{gen} \tag{1}$$

where

$$\dot{X}_Q = \dot{Q}(1 - T_o/T) \tag{2}$$

and the j's and k's refer to inlet and outlet ports, respectively. $\dot{X}_W$ is the exergy delivery rate or useful mechanical power output by the control volume as an open system and $\dot{X}_Q$ is the exergy content due to heat transfer interactions.

The steady flow exergy balance for an open system is simply written as

$$\sum_{in}\dot{X} = \dot{X}_D + \sum_{out}\dot{X} \tag{3}$$

The exergy flow of an open system is represented by the second and third terms on the right-hand side of Eq. (1), where $\bar{x}_t$ is the total molal flow exergy of the mixture stream, given by [2]

$$\bar{x}_{tot} = (\bar{h} - \bar{h}^*) - T_o(\bar{s} - \bar{s}^*) + \sum_{i=1}^{n}(\mu_i^* - \mu_{o,i})y_i \tag{4}$$

where $(\ )^*$ indicates properties evaluated at the restricted dead state (RDS). This dead state means that the stream is brought to thermal and mechanical equilibrium (only) with the environment.

As stated before, the total flow exergy is the sum of the thermomechanical and chemical flow exergies, i.e.

$$\bar{x}_{tot} = \bar{x}_x + \bar{x}_{ch} \tag{5}$$

However, with reference to the RDS (T_o, P_o), thermomechanical specific molal flow exergy is given by

$$\bar{x}_x = (\bar{h} - \bar{h}^*) - T_o(\bar{s} - \bar{s}^*) \tag{6}$$

while the specific molal chemical flow exergy,

$$\bar{x}_{ch} = \sum_{i=1}^{n}(\mu_i^* - \mu_{o,i})y_i \tag{7}$$

Considering dry air and water vapor as an ideal gas, an alternative formula presented by Wepfer et al. [6], gives the total flow exergy of humid air per kilogram of dry air as

$$\begin{aligned} x_{tot} &= (c_{p,da} + \omega c_{p,v})T_o\big(T/T_o - 1 - \ln(T/T_o)\big) \\ &\quad + (1+\tilde{\omega})R_{da}T_o\ln(P/P_o) + R_{da} \\ &\quad \times T_o\big[(1+\tilde{\omega})\ln\big((1+\tilde{\omega}_o)/(1+\tilde{\omega})\big) + \tilde{\omega}\ln(\tilde{\omega}/\tilde{\omega}_o)\big] \end{aligned} \tag{8}$$

where the last term is the chemical exergy. The proportionality between specific humidity ratio ω and specific humidity ratio on a molal basis $\tilde{\omega}$ is given by

$$\tilde{\omega} = 1.608\omega \tag{9}$$

where the specific humidity ratio is

$$\omega = \dot{m}_v/\dot{m}_{da} \tag{10}$$

It represents number of kilograms of water that are present in one kilogram of dry air in the air–water vapor mixture.

Wepfer et al. [6] used approximate formulations to calculate stream exergies. A parametric study was carried out, using the cooling tower case (see next section), to ascertain the amount of error in the exergy of the inlet and outlet humid air streams calculated by Eq. (8) as compared to Eq. (4) by varying the inlet wet-bulb temperature. For the three different water-to-air flow ratios investigated, i.e. $\dot{m}_{w,in}/\dot{m}_{da} = 1, 0.75$ and 0.5, the maximum error was found to be less than 0.65%. It was noted that the errors were negligible. Therefore, due to the presence of thermodynamic and transport properties in the equation-solving software [17] that was used in the present work, these approximate equations were not employed. Eqs. (6) and (7) were simply applied with the understanding that pure streams, such as water, do not have a chemical exergy contribution.

The second-law efficiency, which is a measure of irreversible losses in a given process, is defined as [2]

$$\eta_{II} = \frac{\textit{total exergy leaving}}{\textit{total exergy entering}} \tag{11}$$

On using Eq. (3), we can define the efficiency as

$$\eta_{II} = 1 - \frac{\textit{exergy destruction}}{\textit{total flow exergy entering}} \tag{12}$$

Wepfer et al. [6] defined the second-law efficiency in terms of product and supply exergy. This was also calculated to make a comparison between the two definitions. Here, in dealing with the inefficiencies associated with a component, we should

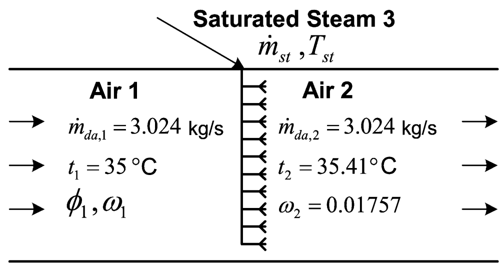


Fig. 1. Steam-spray humidification process.

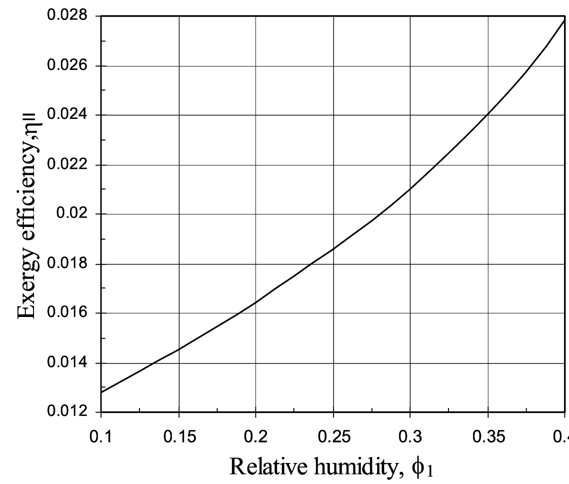
recognize that the exergies of all material streams exiting a component are considered either at the product side or (with a negative sign) at the fuel side [19].

$$\eta_{II,Wep} = \frac{Change\ in\ product\ exergy}{Change\ in\ supply\ exergy} \tag{13}$$

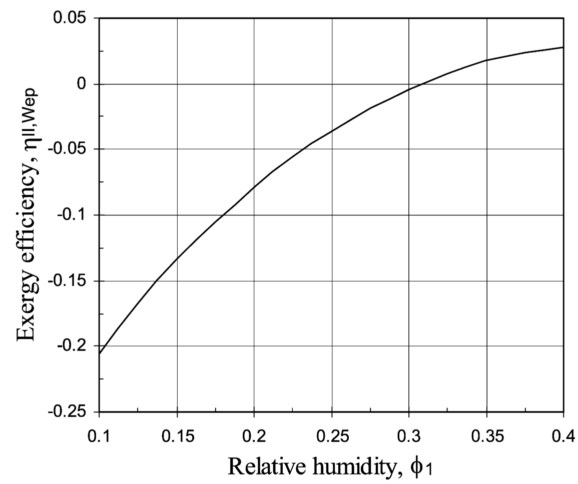
As mentioned earlier, we will now consider the case of steam-spray humidification process discussed by Wepfer et al. [6] that is shown in Fig. 1. The same air properties and dead state are used wherein the saturated steam spray is at moderate pressure (i.e. 143 kPa). Only the humidity ratio of the inlet air is varied which causes the mass flow rate and temperature of the steam to change accordingly so that the required outlet air condition is achieved. Considering this process, a comparison was made between the two efficiency definitions mentioned earlier (see Eqs. (11) and (13)). In this regard, for Fig. 2(b), the second-law efficiency was defined by Eq. (13). It can be seen that the efficiency is negative and becomes greater than zero after a relative humidity of approximately 32%. Moreover, it is evident from Fig. 2(a) that efficiency does not assume a negative value when Bejan's [2] definition is used.

It is important to note that when defining second-law efficiency in Wepfer et al.'s [6] way, special considerations are applicable. For example, a throttling valve is a component for which a product is not readily defined when the valve is considered alone. Throttling valves typically serve other components. Accordingly, when formulating efficiency based on the second-law, the throttling valve and the components it serves should be considered together. Similar considerations apply to heat exchangers (coolers) that achieve the cooling of a stream by heating of another stream. When the purpose of owning and operating a plant component also involves other components, second-law efficiency generally should be defined for an enlarged system consisting of the component and the other components it directly affects. Also, regarding the dead state, in simple heat exchangers or refrigeration applications, it is assumed, in Wepfer et al.'s [6] definition, that all heat transfers occur above or below, respectively, the dead state temperature (T_o). No meaningful product, and thus, no meaningful second-law efficiency, can be defined for a heat exchanger that allows heat transfer across the temperature of the environment i.e. heat transfer between two streams at $T > T_o$ and $T < T_o$, respectively [9]. The above discussion indicates that, compared to Eq. (11), Eq. (13) has some limitations and cannot be used for cooling towers and evaporative heat exchangers.

All analyses were carried out for three different water-to-air flow ratios, $\dot m_{w,in}/\dot m_{da} = 1, 0.75$ and 0.5, whereas the air-



(a)



(b)

Fig. 2. (a) Variation of second-law efficiency versus relative humidity (Eq. 12)); (b) Variation of second-law efficiency versus relative humidity (Eq. (13)).

flow rate is kept constant. It should be noted that the following constant values of air and water vapor were used: $c_{p,da} = 1.004\ \mathrm{kJ\,kg^{-1}K^{-1}}$, $c_{p,v} = 1.872\ \mathrm{kJ\,kg^{-1}K^{-1}}$, $R_{da} = 0.287\ \mathrm{kJ\,kg^{-1}\,K^{-1}}$, and $R_v = 0.461\ \mathrm{kJ\,kg^{-1}\,K^{-1}}$. The dead state was chosen as $T_o = 298.16$ K, $P_o = 101.325$ kPa, and $\phi_o = 0.5$. Keeping in mind that the dead state conditions that have been used are those traditionally associated with standard atmospheric conditions [2].

To completely define the problem, the following two sections briefly outline the model used for cooling towers and evaporative heat exchangers.

3. Mathematical model for wet cooling towers

The major assumptions applied to derive the basic modeling equations are summarized in [20,21], although it should be noted that evaporation was not neglected. From steady-state energy balance between air and water (refer to Fig. 3), we get

$$\dot m_{da}\,dh_a = -\left[\dot m_{w,in} - \dot m_{da}(\omega_{out} - \omega)\right] dh_{f,w} + \dot m_{da}\,d\omega\, h_{f,w} \tag{14}$$

The air-side water–vapor mass balance is governed by

$$\dot{m}_{da}\, d\omega = h_D A_V\, dV(\omega_{s,w} - \omega) \tag{15}$$

The condition line on the psychrometric chart for changes in state for moist air passing through the tower is given by

$$\frac{dh_a}{d\omega} = Le\frac{(h_{s,w} - h_a)}{(\omega_{s,w} - \omega)} + (h_{g,w} - h_g^0 Le) \tag{16}$$

4. Mathematical model for evaporative fluid cooler and condenser

The major assumptions that are used to derive the basic modeling equations are summarized in [22–24] where evaporation was not neglected. The water mass balance, for both the evaporative cooler and condenser (refer to Figs. 6 and 9), yields

$$\dot{m}_{da}\, d\omega = d\dot{m}_w \tag{17}$$

The mass flow of recirculating water evaporating into air, in terms of the mass-transfer coefficient, h_D, for both the evaporative cooler and condenser, can be written as

$$d\dot{m}_w = h_D(\omega_{s,int} - \omega)\, dA \tag{18}$$

Eqs. (17) and (18) indicate that the mass flow rate does not remain constant as some of the water evaporates.

In the evaporative cooler and condenser, at the air–water interface, simultaneous heat and mass transfer takes place, which can be expressed as shown below if Lewis number is taken as unity [24]

$$dh_a = \frac{h_D}{\dot{m}_{da}}(h_{s,int} - h_a)\, dA \tag{19}$$

For the evaporative cooler, the energy balance on the process fluid can be expressed as:

$$dt_p = -\frac{U_{os}}{\dot{m}_p c_{p,p}}(t_p - t_{int})\, dA \tag{20}$$

where U_{os} is the overall heat transfer coefficient. The overall energy balance on the evaporative cooler (refer to Fig. 6), gives

$$dt_w = \frac{1}{\dot{m}_{w,in} c_{p,w}}\left[\dot{m}_{da}\, dh_a - c_{p,w} t_w\, d\dot{m}_w + \dot{m}_p c_{p,p}\, dt_p\right] \tag{21}$$

It is important to note from Eq. (21) that some of the heat removed from the process fluid goes to heating (or cooling) the water film.

Similarly, the overall energy balance for the evaporative condenser (refer to Fig. 9), results in

$$dt_w = \frac{1}{\dot{m}_{w,in} c_{p,w}}\left[\dot{m}_{da}\, dh_a - c_{p,w} t_w\, d\dot{m}_w - \dot{m}_r\, dh_r\right] \tag{22}$$

Applying a similar procedure to the evaporative condenser as was used earlier to formulate Eq. (20), keeping in view the fact that enthalpy changes but the fluid temperature remains constant and also the direction of flow of the condensing fluid, we get

$$dh_r = \frac{U_{os}}{\dot{m}_r}(t_r - t_{int})\, dA \tag{23}$$

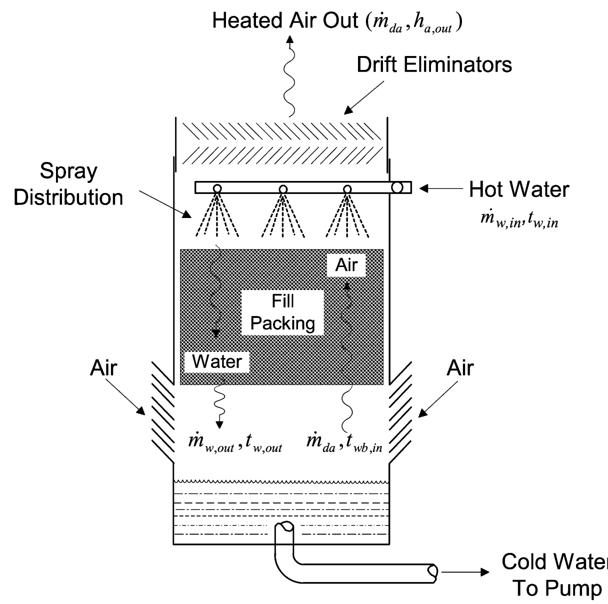


Fig. 3. A counter flow wet-cooling tower.

Based on outside surface area of the tubes, the overall heat transfer coefficient U_{os} can be written as

$$\frac{1}{U_{os}} = \left[\frac{1}{h_{c,is}}\right]\frac{d_{t,os}}{d_{t,is}} + \left(\frac{d_{t,os}}{2k_t}\right)\ln\left[\frac{d_{t,os}}{d_{t,is}}\right] + \frac{1}{h_{c,w}} \tag{24}$$

Now, if the temperature of the interface film is considered the same as the bulk water temperature, then all the terms with the subscripts (s, int) will be replaced by (s, w). This approach was used in the current work.

5. Exergetic analysis for the wet cooling towers

In this section, we analyze the wet cooling tower, which is a very important heat rejection system. The exergy destruction (refer to Eq. (3)), is given by

$$\dot{X}_D = (\dot{X}_{a,in} + \dot{X}_{w,in} + \dot{X}_{makeup}) - (\dot{X}_{a,out} + \dot{X}_{w,out}) \tag{25}$$

The second-law efficiency (refer to Eq. (12)), of the entire system shown schematically in Fig. 3, can be written as

$$\eta_{II} = 1 - \frac{\dot{X}_D}{\dot{X}_{a,in} + \dot{X}_{w,in} + \dot{X}_{makeup}} \tag{26}$$

A sensitivity analysis with respect to the second-law efficiency was performed in which normalized sensitivity coefficients were calculated by equally perturbing each input variable one by one [25]. For the cooling tower, the inlet wet-bulb and water temperatures were found to be the two most important input parameters. Therefore, variation in second-law efficiency is studied with respect to these two parameters. The graphs are drawn for the following set of input data: $t_{db,in} = 29.0\,^\circ\mathrm{C}$, $V = 0.699\ \mathrm{m}^3$, $Le = 0.9$ and $\dot{m}_{da} = 1.19\ \mathrm{kg\,s^{-1}}$.

5.1. Variation in inlet wet bulb temperature

Fig. 4(a) illustrates the exergy destruction variation, and Fig. 4(b) shows the second-law efficiency, using Eq. (26), as

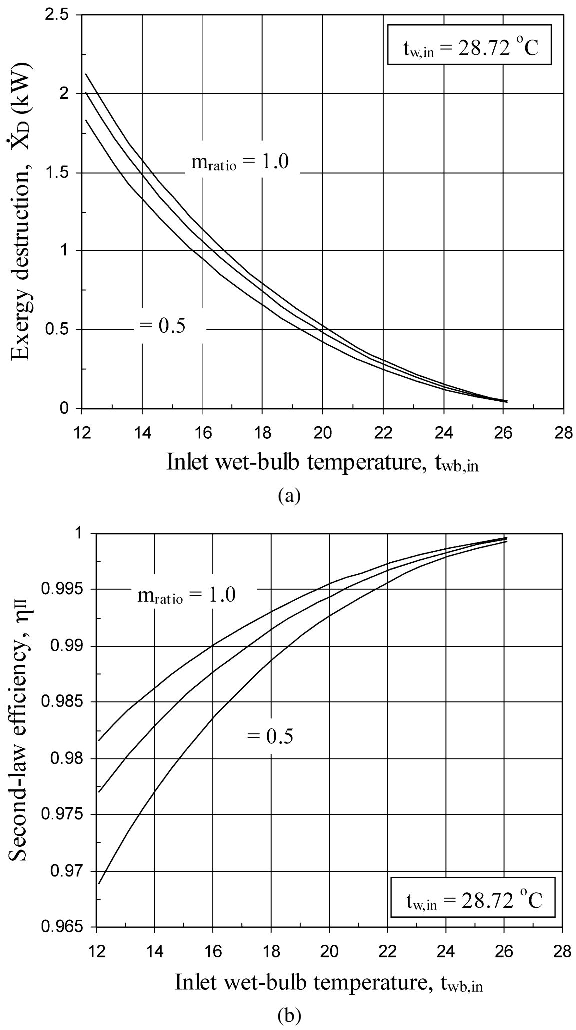


Fig. 4. (a) Variation of exergy destruction versus inlet wet-bulb temperature; (b) Variation of second-law efficiency versus inlet wet-bulb temperature (Eq. (26)).

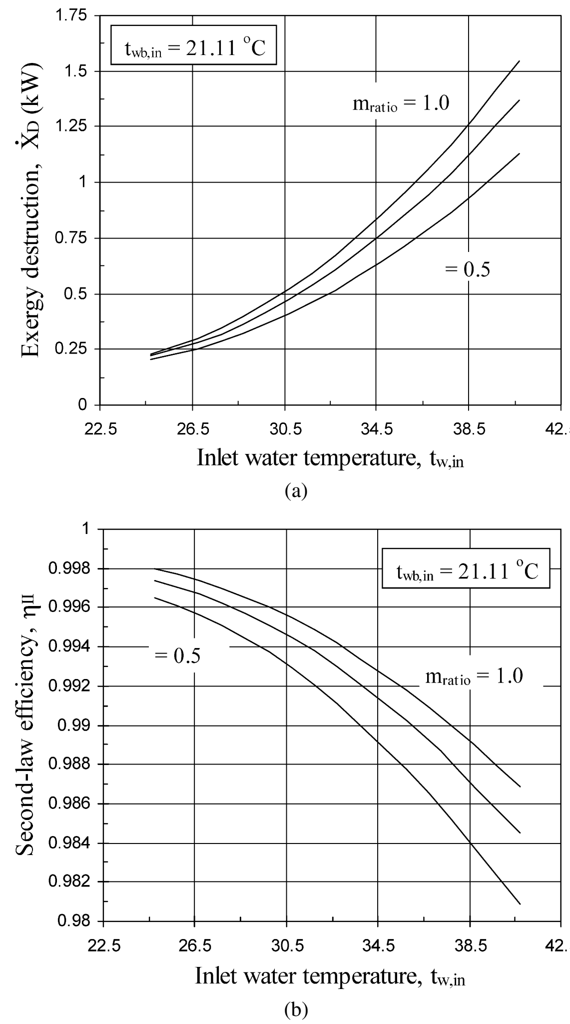


Fig. 5. (a) Variation of exergy destruction versus inlet water temperature; (b) Variation of second-law efficiency versus inlet water temperature (Eq. (26)).

a function of the inlet wet-bulb temperature (from 12.11 to 26.11 °C), for different mass flow ratios. From Fig. 4(a) and (b), it is noted that second-law efficiency increases as the exergy destruction decreases for the increasing inlet wet-bulb temperature. The exergy of the outlet air stream constantly increases due to higher dry-bulb temperature as well as humidity ratios that are achieved. Also, since the water loss decreases with the increasing inlet wet bulb temperature, exergy of the makeup water also decreases. As $t_{wb,in}$ increases, the outlet water temperature also rises and, thus, the exergy of the outlet water stream increases. On the other hand, the exergy of the incoming water is constant. The exergy destroyed decreases due to the continuously decreasing value of $(t_{db,in} - t_{wb,in})$. These factors combine so that the second-law efficiency η_{II} increases and can be attributed to the decreasing value of $(t_{w,out} - t_{wb,in})$, as the volume of the tower is kept constant.

5.2. Variation in inlet water temperature

Fig. 5(a) and 5(b) shows the variation in the exergy destruction and second-law efficiency, respectively, as the inlet water temperature changes from 24.72 to 40.72 °C, for different mass flow ratios. In Fig. 5(b), it is noted that second-law efficiency decreases as the exergy destruction (Fig. 5(a)) increases for the increasing inlet water temperature. The exergy of the exiting air stream continuously increases as it gets farther from the dead state humidity ratio. On the other hand, exergy of the entering air stream is constant. Also, since the water loss increases due to the increasing difference of the inlet water and wet-bulb temperatures, exergy of the makeup water also increases. The exergy of the outlet water stream decreases as its temperature approaches T_o. However, the exergy of the incoming water stream constantly increases due to higher water temperatures used. The increase in the exergy destruction is due to the continually increasing difference between the inlet and outlet water temperatures. These factors cause the second-law efficiency η_{II} to decrease and can also be understood from the fact that the effectiveness is also decreasing.

6. Exergetic analysis for the evaporative fluid cooler

In this section, we analyze the evaporative fluid cooler. The exergy destruction (refer to Eq. (3)) can be expressed as

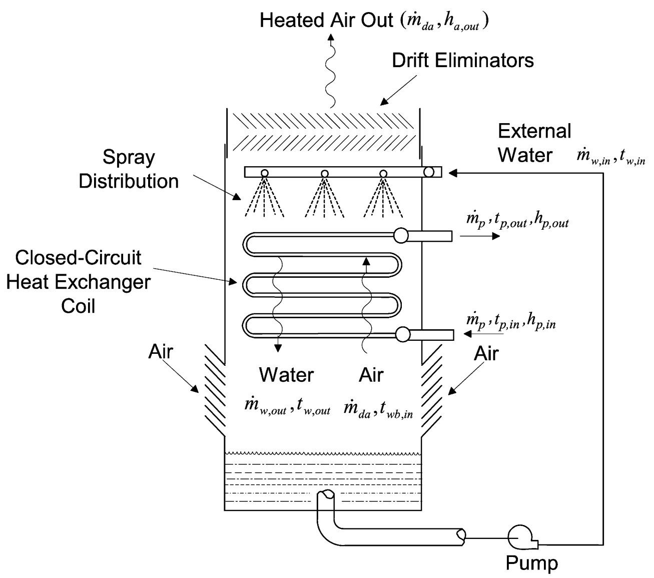


Fig. 6. An evaporative cooler.

$$\dot X_D = (\dot X_{a,in} + \dot X_{w,in} + \dot X_{p,in} + \dot X_{makeup}) \\ - (\dot X_{a,out} + \dot X_{w,out} + \dot X_{p,out}) \quad (27)$$

The second-law efficiency (refer to Eq. (12)), of the entire system, shown schematically in Fig. 6, can be written as

$$\eta_{II} = 1 - \frac{\dot X_D}{\dot X_{a,in} + \dot X_{w,in} + \dot X_{p,in} + \dot X_{makeup}} \quad (28)$$

The sensitivity analysis revealed that the normalized sensitivity coefficients were highest for the inlet wet-bulb and process fluid temperatures [25]. Therefore, the changes in second-law efficiency were studied with respect to these variables. The graphs are drawn for the following set of input data: $t_{db,in} = 29.0\,°\mathrm{C}$, $A = 1.915\ \mathrm{m}^2$, $Le = 1$, $d_{t,is} = 16.05$ mm, $d_{t,os} = 19.05$ mm, $L = 0.5$ m, $n_{tr} = 6$, $\dot m_p = 0.325\ \mathrm{kg\,s^{-1}}$ and $\dot m_{da} = 0.167\ \mathrm{kg\,s^{-1}}$ where the water temperature at the inlet and outlet is considered the same.

6.1. Variation in inlet wet bulb temperature

Fig. 7(a) illustrates the exergy destruction and Fig. 7(b) variation in the second-law efficiency, using Eq. (28), as the inlet wet-bulb temperature varies from 12.11 to 23.11 °C, for different mass flow ratios. Though it is not apparent from the figures, we can see clearly from Table 1 that second-law efficiency decreases and the exergy destruction increases as the inlet wet-bulb temperature decreases. It should be noted that the inlet wet-bulb temperature is, theoretically, the lowest possible temperature that can be achieved in this system. Therefore, with the inlet process fluid temperature constant, the behavior seen in these figures can be understood from the fact that an increase in the wet-bulb temperature has a direct effect of raising the outlet process fluid temperature as well as the water temperature.

Similar to the case of cooling tower discussed above, the exergy of the inlet moist air minimizes at a wet-bulb temperature of approximately 19.2 °C when it reaches the dead state

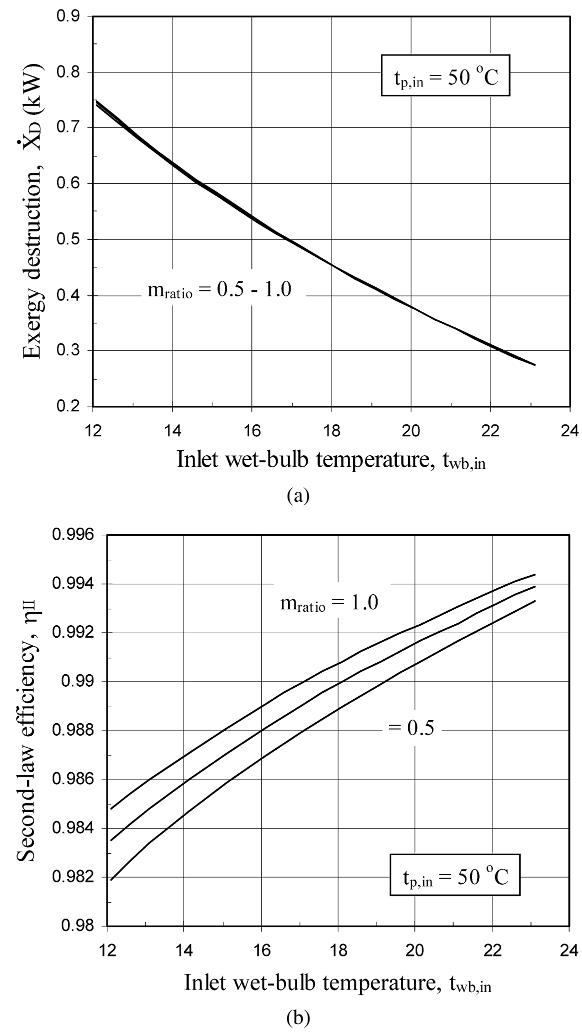


Fig. 7. (a) Variation of exergy destruction versus inlet wet-bulb temperature; (b) Variation of second-law efficiency versus inlet wet-bulb temperature (Eq. (28)).

Table 1
Calculated values for Fig. 7(a) and (b)

$t_{wb,in}$ [°C]	$\dot X_D$ [kW]			η_{II}		
	$m_{ratio} = 1$	$= 0.75$	$= 0.5$	$m_{ratio} = 1$	$= 0.75$	$= 0.5$
23.11	0.2744	0.2753	0.2756	0.9944	0.9939	0.9933
21.11	0.3401	0.3403	0.34	0.9931	0.9924	0.9917
19.11	0.4126	0.4123	0.4113	0.9916	0.9908	0.9899
17.11	0.4929	0.4923	0.4904	0.99	0.9891	0.988
15.11	0.583	0.5817	0.5792	0.9881	0.9871	0.9859
12.11	0.7469	0.7448	0.741	0.9848	0.9835	0.9819

humidity ratio. It then continuously increases with the increasing wet-bulb temperature. The exergy of the outlet air stream constantly increases due to higher dry-bulb temperature as well as humidity ratios that are achieved. Since the water loss decreases with the increasing inlet wet bulb temperature, exergy of the makeup water also decreases. Keeping in mind that the water temperature at the inlet and outlet are same, we find that the rising wet-bulb temperature increases the water temperature due to the decreasing rate of evaporation and, consequently, the exergy of the water streams. The process fluid exergy at the inlet is constant but it increases at the outlet due to higher temperatures achieved there, the exergy destroyed decreases due to the continuously decreasing value of $(t_{db,in} - t_{wb,in})$. These factors cause the second-law efficiency η_{II} to increase. With the surface area of the tubes constant, this can be attributed to the decreasing value of $(t_{p,in} - t_{wb,in})$.

6.2. Variation in inlet process fluid temperature

Fig. 8(a) shows the exergy destruction and Fig. 8(b) variation in the second-law efficiency as the process fluid inlet temperature increases from 40 to 60 °C, for different mass flow ratios. It is noted that second-law efficiency decreases and the exergy destruction increases monotonically as the inlet process fluid temperature increases. We see that the mass flow rate ratio has a small effect on the exergy destruction. The exergy of the outlet air stream constantly increases, as it gets farther away from the dead state humidity ratio. On the other hand, the exergy of the entering air stream is constant. As the inlet process fluid temperature increases, its exergy value rises as well. Furthermore, this causes higher water temperatures and an increase in the rate of evaporation due to the increased heat transfer, which increases the exergy of the makeup and re-circulating water. As discussed above, the temperature of the water is considered same at the inlet and outlet of this heat exchanger. However, the exergy difference of the inlet and outlet process fluid streams constantly increases due to higher process fluid temperatures at the inlet. This causes the exergy destruction to increase and can be attributed to the continually increasing difference between the inlet and outlet process fluid temperatures. With the exergy destruction increasing, the second-law efficiency η_{II} decreases.

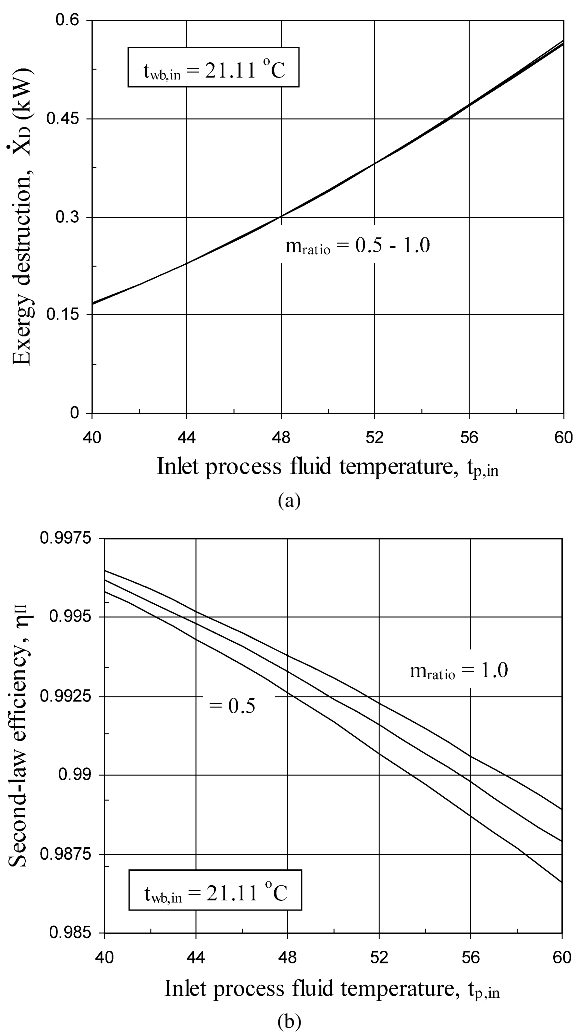


Fig. 8. (a) Variation of exergy destruction versus inlet process fluid temperature; (b) Variation of second-law efficiency versus inlet process fluid temperature (Eq. (28)).

7. Exergetic analysis for the evaporative condenser

In this section, we analyze the evaporative condenser. The exergy destruction (refer to Eq. (3)), can be written as

$$\dot{X}_D = (\dot{X}_{a,in} + \dot{X}_{w,in} + \dot{X}_{r,in} + \dot{X}_{makeup}) - (\dot{X}_{a,out} + \dot{X}_{w,out} + \dot{X}_{r,out}) \tag{29}$$

The second-law efficiency (refer to Eq. (12)), of the entire system shown schematically in Fig. 9, can be written as

$$\eta_{II} = 1 - \frac{\dot{X}_D}{\dot{X}_{a,in} + \dot{X}_{w,in} + \dot{X}_{r,in} + \dot{X}_{makeup}} \tag{30}$$

Based on the sensitivity analysis [25], the variation in second-law efficiency was studied with respect to the varying inlet wet bulb temperature as well as refrigerant temperature. The graphs are drawn for the following set of input data: $t_{db,in} = 29.0\,°C$, $A = 0.3788\ m^2$, $Le = 1$, $d_{t,is} = 14.1$ mm, $d_{t,os} = 15.9$ mm, $L = 0.343$ m, $n_{tr} = 4$, $\dot{m}_r = 0.0132\ kg\,s^{-1}$ and $\dot{m}_{da} = 0.0619\ kg\,s^{-1}$ where water temperature at the inlet and outlet is considered the same.

7.1. Variation in inlet wet bulb temperature

Fig. 10(a) illustrates the exergy destruction and Fig. 10(b) the variation in the second-law efficiency, using Eq. (30),

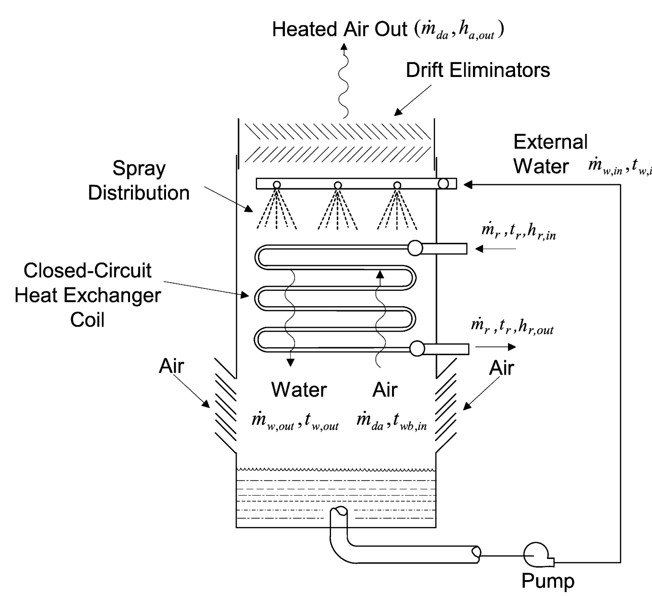


Fig. 9. An evaporative condenser.

as a function of the inlet wet bulb temperature (from 12.11 to 23.11 °C) and different mass flow ratios. From Fig. 10(a) and (b), we see that second-law efficiency increases and the exergy destruction decreases as the inlet wet bulb temperature increases. Similar to the cooling tower and evaporative cooler, the exergy of the inlet moist air minimizes at a wet-bulb temperature of approximately 19.2 °C when it reaches the dead state humidity ratio. Thereafter it constantly increases. Again, the exergy of the outlet air stream constantly increases due to the higher dry-bulb temperature and humidity ratios attained. With the water loss decreasing, the exergy of the makeup water also decreases. The rising wet-bulb temperature increases the steady-state water temperature and, consequently, the exergy values of water. Also, lesser heat is transferred from the condensing fluid and, thus, the exergy of the refrigerant at the outlet also increases. These factors decrease the exergy destruction and can be attributed, in general, to the continuously decreasing value of $(t_{db,in} - t_{wb,in})$. At smaller mass flow ratios, exergy destruction is lower mainly due to smaller exergy values of the outlet air and water streams. Subsequently, these factors cause the second-law efficiency η_{II} to increase and can be attributed to the decreasing value of $(t_r - t_{wb,in})$ because the surface area of the tubes is constant. As the mass flow rate ratio increases, the overall heat transfer coefficient also rises and, therefore, the second-law efficiency is higher as well.

7.2. Variation in refrigerant temperature

Fig. 11(a) shows the exergy destruction and Fig. 11(b) the variation in the second-law efficiency as the refrigerant temperature changes from 35 to 50 °C, for different mass flow ratios. It is noted that second-law efficiency decreases and the exergy destruction increases as the refrigerant temperature increases. The exergy of the outlet air stream constantly increases as it moves away from the dead state humidity ratio due to the increasing value of $(t_r - t_{wb,in})$. It allows the air to become more and more humid. On the other hand, the exergy of the entering air stream is constant, as the conditions there are not changing. As the water loss increases, due to the increasing value of $(t_r - t_{wb,in})$, exergy of the makeup water also increases. We note that the exergy value of the refrigerant increases at the

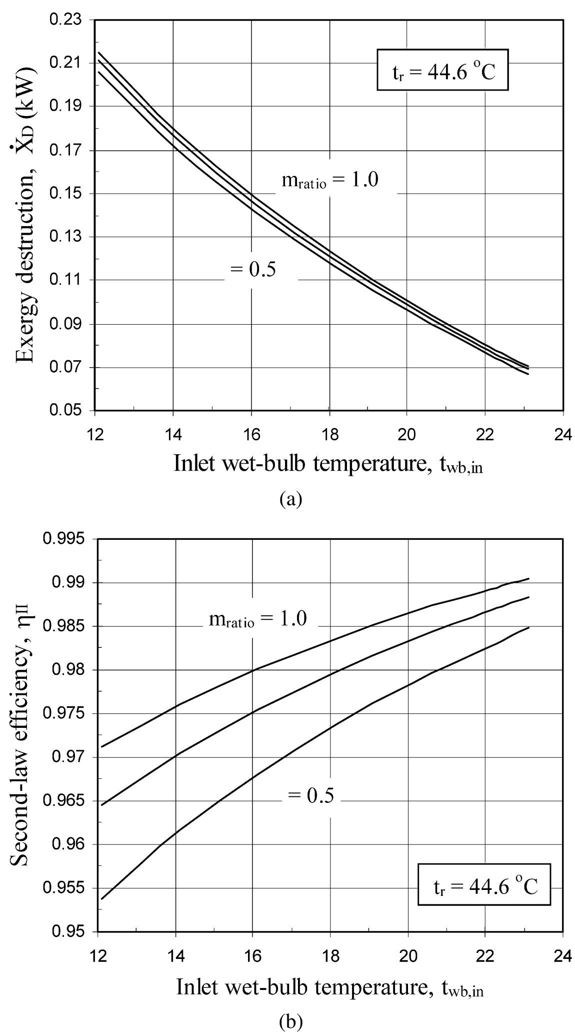


Fig. 10. (a) Variation of exergy destruction versus inlet wet-bulb temperature; (b) Variation of second-law efficiency versus inlet wet-bulb temperature (Eq. (30)).

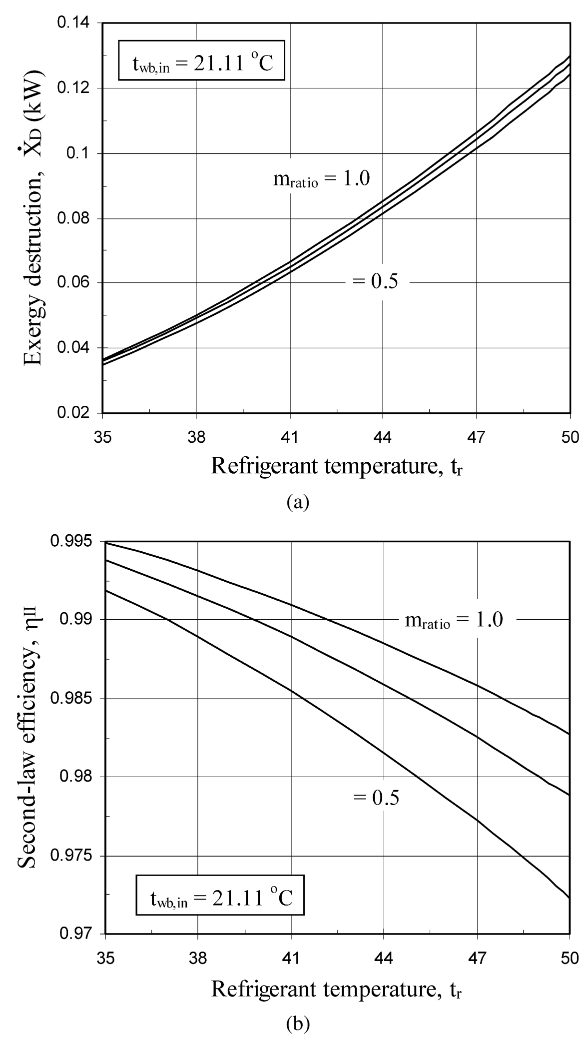


Fig. 11. (a) Variation of exergy destruction versus refrigerant temperature; (b) Variation of second-law efficiency versus refrigerant temperature (Eq. (30)).

inlet with the increase in its temperature. A combination of these factors causes the exergy destruction to increase, which can be attributed to the increasing value of $(t_r - t_{wb,in})$. The mass flow ratio does not have a significant effect on the exergy destruction mainly due to the fact that it largely affects the exergy of the water stream, which cancels out since they are equal (see Eq. (29)). Subsequently, the second-law efficiency η_{II} correspondingly decreases since the surface area of the tubes is constant. As the mass flow rate ratio increases, the overall heat transfer coefficient also rises and, therefore, the second-law efficiency is higher as well.

8. On second-law efficiency values

In general, regarding the high second-law efficiencies shown in the above figures, the most important thing, is to note that the formula used to calculate the values (refer to Eq. (11)) is different from those found in the literature regarding similar processes. Considering specifically the case of cooling towers, Moran [5] discussed the availability (exergy) analysis of a cooling tower through an example problem. If we use our definition, the efficiency comes out to be 95.76%. Also, it is well known that an increase in the temperature drop for the hot fluid is accompanied by higher destruction of exergy. Therefore, using the same conditions as in Fig. 5, results for an inlet water temperature of 42.72 °C and 80 °C were compared and it was found that the exergy destruction increased 10 times for a 3.8 times rise in temperature drop for the water. As expected, the efficiency for the latter case was less compared to the former where it decreased by almost 7%. The reason for this small reduction is the fact that, even though exergy destruction has increased, the total inlet exergy is also greater than before. Similar observations were also noticed for evaporative coolers and condensers.

9. Effect of variation in the dead state

The selection of dead state is an important aspect of exergy calculations. The exergy values of individual streams are significantly affected by variations in the dead state but it is not necessary that this must be the case for second-law efficiency. Therefore, it is important to understand the significance of reasonable variations in or selection of the dead state properties on it. Wepfer et al. [6] provide some guidelines in the selection of dead state values. They emphasized that since a system operates while the ambient conditions are different; either at different design conditions or off-design conditions, the instantaneous value of T_o, P_o and ϕ_o should be used. Therefore, it is important to ascertain the effect of selection and variations of the dead state on the second-law efficiency. From Tables 2 and 3, we can see that the variations in the dead state temperature and relative humidity do not significantly affect the overall efficiency of the cooling tower and evaporative heat exchangers, which is in line with the findings of Rosen and Dincer [26]. It is seen that there exists a maximum difference of 2.11% in the efficiencies calculated for the varying dead state temperature and 0.88% only for the varying dead state relative humidity.

Table 2
Effect of variation of dead state temperature on second-law efficiency

$T_o = 273.16$ °C	η_{II}		
	Cooling tower	Evaporative fluid cooler	Evaporative condenser
45	0.9966	0.9937	0.9771
35	0.9965	0.9948	0.9816
25	0.9965	0.9961	0.9869
15	0.9966	0.9976	0.9925
5	0.9968	0.9991	0.9982

Table 3
Effect of variation of dead state relative humidity on second-law efficiency

ϕ_o	η_{II}		
	Cooling tower	Evaporative fluid cooler	Evaporative condenser
0.1	0.9906	0.9988	0.9957
0.2	0.9901	0.9983	0.9940
0.3	0.9896	0.9978	0.9921
0.4	0.9890	0.9971	0.9898
0.5	0.9881	0.9961	0.9869

10. Concluding remarks

Exergetic analysis is conducted on cooling towers and evaporative heat exchangers. In this regard, two definitions for the second-law efficiency are used. All computations are conducted with an engineering equation solver (EES) program that has built-in functions for thermodynamic and transport properties. These built-in properties make it possible for a complete equation to be used for all streams and not only reduce the computational effort, but avoid the need for approximate solutions. For different input variables investigated, it is shown that the efficiencies are often increasing or decreasing monotonically. We also notice that an increase in the inlet wet bulb temperature consistently increases the second-law efficiency of all the systems investigated. It is shown that Bejan's definition of second-law efficiency is not limited in evaluating performance. The high second law efficiency of the evaporative heat exchangers indicates that, from thermodynamic standpoint, the processes occurring in these devices are approaching reversible. Furthermore, we see that the selection of the dead state properties and changes in it due to operation at off-design conditions does not significantly affect the overall efficiency of the system but it is noticed that the individual stream exergies are appreciably affected nonetheless.

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