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# Exergy Analysis of a High-Temperature-Steam-Driven, Varied-Pressure, Humidification-Dehumidification System Coupled With Reverse Osmosis

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## Abstract

*In this study, exergy analysis of a novel desalination system is presented and discussed. The water desalination is carried out using combined humidification-dehumidification and reverse osmosis technologies. Six system performance parameters are examined: overall exergetic efficiency, equivalent electricity consumption, specific exergy destruction, specific exergy lost, and total true specific exergy lost, as well as the exergy destruction ratios of the main components. The total true specific exergy lost is a new parameter presented in this study. It is a function of summation of total the exergy destruction rate and loss per total mass flow rate of the total pure water produced. This parameter is found to be a useful parameter to assess the exergetic performance of the system considered. By contrast, use of overall exergetic efficiency as an assessment tool can result in misleading conclusions for such a desalination system and, hence, is not recommended. Furthermore, this study reveals that the highest exergy destruction occurs in the thermal vapor compressor, which accounts for 50% of the total exergy destruction of the system considered. This study, in addition, demonstrates that the specific exergy destruction of the dehumidifier and TVC are the parameters that most strongly affect the performance of the system.*

*Key words:* seawater desalination, thermal vapor compressor (TVC), reverse osmosis (RO), humidification dehumidification (HDH), total true specific exergy lost, exergy destruction, equivalent electricity consumption, exergetic efficiency.

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## List of Symbols

$\dot{E}c$	: Equivalent electricity consumption, kWh <sub>e</sub> /m <sup>3</sup>
$ER$	: Entrainment ratio
$ex$	: Specific exergy, kJ/kg
$\bar{e}x$	: Specific exergy, kJ/kmol
$ex_{d,i,mp}$	: Specific exergy destruction, kJ/kg-product water
$ex_{DL,mp}$	: Total true specific exergy lost, kJ/kg-product water
$\dot{E}x_d$	: Exergy destruction rate, kW
GOR	: Gained-output ratio
$h$	: Enthalpy per unit mass, kJ/kg
$h^*$	: Enthalpy per unit mass, kJ/kg-dry-air
$\dot{H}$	: Enthalpy rate, kW
HCR	: Heat capacity ratio
HDH	: Humidification-dehumidification
$M$	: Molecular weight, kg/kmol
$\dot{m}_{da}$	: Dry air mass flow rate, kg/s
$\dot{m}_j$	: Mass flow rate at state $j$ , kg/s
$\dot{m}_p$	: Total mass flow rate of water produced, kg/s
$p$	: Pressure, kPa
$PR$	: TVC pressure ratio
$R$	: Universal gas constant, 8.314 kJ/kg·K
RO	: Reverse osmosis
$s$	: Entropy per unit mass, kJ/kg·K
$s^*$	: Entropy per unit mass, kJ/(kg-dry-air ·K)
$\dot{S}_{gen}$	: Entropy generation rate, kW
$T$	: Temperature, K
$\dot{W}$	: Power (work rate), kW
$x_j$	: Molar fraction of species $j$
$Y_D$	: Exergy destruction ratio

### ***Greek letters***

$\eta$	: Efficiency
$\omega$	: Absolute humidity, kg-H <sub>2</sub> O/kg-dry-air
$\varepsilon$	: Effectiveness

### ***Subscripts***

0	: Atmospheric conditions
$a$	: Air
$c$	: Cold steam
$cv$	: Control volume
$d$	: Destruction
$D$	: Dehumidifier
$e$	: Exit

*exp* : Expander  
*h* : Hot stream  
*H* : Humidifier  
*i* : Property of component *i*  
*in* : Inlet  
*j* : Indicate a property of a value at state *j*  
*L* : Loss (exergy lost)  
*mp* : Mass of water produced  
*p* : Pure water product  
 $\dot{Q}$  : Heat rate, kW  
*rev* : Reversible process  
*RO* : Reverse osmosis  
*st* : Steam  
*TVC* : Thermal vapor compressor

*w* : Water

### ***Superscripts***

*CH* : Chemical exergy  
*PH* : Physical exergy  
*s* : Isentropic

## 1. Introduction

The need for potable water is increasing as world population and standards of living rise. It is expected that the world population will grow to 9.1 billion in 2050 [1]. Hence, there is a crucial need to find energy efficient desalination systems.

Operating a desalination system with a relatively high brine temperature improves the thermodynamic performance; however, there is a practical limit for the top brine temperature. A high top brine temperature can lead to the formation of non-alkaline scales (calcium sulphate). Top brine temperatures do not normally exceed 120 °C and are often held much lower, depending on the desalination technology used. The system examined in this study is a novel gas carrier thermal desalination system that can operate with a top system temperature above 120 °C [2, 3]. The concern for scale formation is avoided because the highest temperature carrier gas is not brought into direct contact with the brine, as described in the next section.

A few studies have been conducted using exergy analysis to examine either a humidification - dehumidification (HDH) desalination [4, 5, 6, 7] or reverse osmosis (RO) desalination [8, 9, 10, 11, 12, 13] technology for water production. Further, a recent study considering energy analysis and design of a novel desalination system using combined HDH and RO was conducted by Narayan et al. [2]. In the present study, a detailed exergy analysis for this novel system is conducted.

The objective of this study is to provide better understanding of the exergetic performance of the novel desalination system proposed in [2, 3] and, thereby, to help identify possible improvements. The assessment of the novel desalination system proposed is conducted by considering exergetic efficiency, equivalent electricity consumption, specific exergy destruction and loss, total true exergy destruction lost, and the associated mass flow rate of purified water; all these parameters are examined under variations of key system parameters. Moreover, the fractions of the system exergy destruction rate associated with the main components in the system are identified.

## 2. System Description

HDH is an emerging technology that has recently received more attention from researchers; meanwhile RO is considered one of the most efficient water desalination technologies. A simple configuration of HDH consists of an air dehumidifier, an air humidifier, and usually a water heater. On the other hand, RO requires either mechanical or electrical power to run the water pump, which is used to pump, for example, seawater into the RO.

In the current study, a novel hybrid water desalination system consisting of a combination of HDH and RO subsystems is examined through exergy analysis. The current hybrid water

desalination system has several advantages as compared to other conventional desalination systems. It has higher GOR and lower equivalent electricity consumption as compared to constant pressure HDH or varied pressure HDH with a mechanical compressor. As compared to the mechanical compressor used in varied pressure HDH, TVC is cheaper and requires less maintenance. In addition, the current hybrid system has the same GOR or higher and the same equivalent electricity consumption or lower as compared to multi-stage flash water desalination system. Furthermore, the current hybrid system under optimum operating conditions has the same GOR or higher and the same equivalent electricity consumption or lower as compared to multi-effect distillation technology. The current hybrid desalination system appears to be a very promising water desalination, especially when a high pressure steam is available with reasonable cost. In addition, the current hybrid water desalination system, as discussed at the end of this section, can work with a relatively high temperature steam without producing a correspondingly a high top brine temperature.

The novel system considered has two desalination technologies: HDH and RO. In this system, there are two loops, a moist air loop and a liquid water loop, (Fig. 1). The moist air loop is described first. Air travels through a humidifier and dehumidifier operated at different pressures. The pressure difference is maintained through a thermal vapor compressor (TVC) and an expander. The humidified moist air leaving the humidifier (state 2) is propelled by a high pressure steam in the TVC. High pressure moist air (state 3) exits the TVC and enters the dehumidifier (condenser). Most of the water in the moist air condenses (state 8), and air at lower humidity exits the dehumidifier and enters the expander (state 4). The expander produces mechanical power, which in turn, is used to operate the RO pump that drives the RO subsystem. The moist air (state 1) exits the expander at lower pressure and temperature and enters the humidifier. The moist air, thus, operates in a closed cycle.

On the other hand, the liquid water operates in an open cycle. The seawater enters the dehumidifier (state 5) in which it cools the the moist air and absorbs the heat released during condensation. After that, the seawater enters the humidifier (state 6) and mixes with the moist air. Then the seawater exits the humidifier (state 7) and enters the RO pump. This pump is run using the energy extracted from the expander. The high pressure seawater enters the RO membrane modules where fresh water is extracted. The remaining brine is rejected to the sea.

The cycle proposed is shown on a moist air psychrometric chart in Fig. 2. In this figure, path 1-2 is the air humidification process that is approximated to follow the saturation line. Path 2-3 is the thermo-compression process where the moist air is compressed to higher pressure and temperature. Path 3-4 is the dehumidification process which is approximated to follow the saturation line at higher pressure. Path 4-1 is the air expansion process where

some of the energy input in the compressor is recovered.

It can be observed in the system considered that relatively a high pressure and temperature steam ( $T_{sat} > 120^\circ\text{C}$ ) can be used without producing a correspondingly high top brine temperature. This high temperature can be used because the heating steam is not brought in a direct contact with the seawater. Instead, it is brought in contact with the moist air. Furthermore, the pressure ratio attained in the TVC is such that the moist air which exchanges heat with the brine in the dehumidifier is at a relatively low temperature. Therefore, this new system can be designed such that the brine temperature does not exceed  $60^\circ\text{C}$ . For standard seawater salinity concentrations, this temperature is sufficient to avoid both hard and soft scale formation. On the other hand, the high temperature energy used, in the form of steam, increases the thermodynamic efficiency of the separation process [14].

### 3. Modeling

In this section, performance parameters, detailed models for the main components, and the solution technique are presented.

#### 3.1. Performance Parameters

The performance parameters used to characterize the desalination system are presented in this section. These parameters are an energy-based effectiveness, an equivalent electricity consumption, a modified heat capacity rate ratio for the heat and mass exchange devices, an isentropic efficiency for the expander, a reversible entrainment efficiency for the TVC, the exergy destruction rate, overall exergetic efficiency, the specific exergy destroyed or lost, the total true specific exergy lost, and the exergy destruction ratio.

##### 3.1.1. Energy effectiveness

The energy effectiveness of the humidifier or dehumidifier is defined as the ratio of change of the enthalpy rate to the maximum possible enthalpy rate. In equation form it can be written as

$$\varepsilon = \frac{\Delta\dot{H}}{\Delta\dot{H}_{max}} \quad (1)$$

where  $\dot{H}$  is the enthalpy rate. The maximum possible change in total enthalpy rate can be of either the cold or the hot stream, depending on the heat capacity rate of the two streams. Further discussion of this effectiveness can be found in a previous publication [15].

### 3.1.2. Equivalent electricity consumption

The equivalent electricity consumption is defined in terms of the amount of electricity that could have been produced using thermal energy that was provided to a desalination system at a given temperature. The equivalent electricity consumption parameter facilitates comparison between thermal desalination systems and work driven desalination system. It is defined as

$$\dot{E}_c = \frac{\dot{W}_{st}}{3.6 \cdot \dot{m}_{pw}} [kW_e h/m^3] \quad (2)$$

where the electrical power,  $\dot{W}_{st}$ , is calculated assuming the steam used in the desalination plant was instead expanded in a steam turbine as follow [2]:

$$\dot{W}_{st} = \dot{m}_{pw} \cdot (h_{st,tur,in} - h_{st,tur,out}) \cdot \eta_{gen} \quad (3)$$

where  $\eta_{gen}$  is the electrical generator efficiency and assumed to be 95%. To calculate the enthalpy at the exit of the turbine, it was assumed its exit temperature to be 35°C which is commonly used in Rankine power plants, and the isentropic efficiency of the turbine was assumed to be 85%.

### 3.1.3. Heat capacity rate ratio

The heat capacity rate ratio for the heat and mass exchangers in this study is defined as the ratio of the maximum possible change in the enthalpy rate of the cold stream to the maximum change of enthalpy rate of the hot stream [16]:

$$\text{HCR} = \frac{\Delta \dot{H}_{max,c}}{\Delta \dot{H}_{max,h}} \quad (4)$$

where the subscript  $c$  and  $h$  indicate cold and hot streams, respectively. The thermodynamics behind Equation 4 were discussed in detail in [16].

### 3.1.4. Isentropic efficiency of the expander

The work rate (power) of the expander is found using the isentropic efficiency definition defined as

$$\eta_{exp} = \frac{\dot{W}_{exp}}{\dot{W}_{exp,rev}} \quad (5)$$

### 3.1.5. Efficiency of the TVC

The definition of the TVC efficiency used is based on the entrainment ratio. Efficiency of the TVC is defined as the ratio of the entrainment ratio of the actual TVC to the entrainment



ratio of an ideal TVC (one with no entropy generation):

$$\eta_{TVC} = \frac{ER}{ER_{rev}} \quad (6)$$

where  $ER$  is the entrainment ratio defined as

$$ER = \frac{\dot{m}_{a,in}}{\dot{m}_{st}} \quad (7)$$

Detailed discussion of this efficiency was presented in a previous publication [17].

### 3.1.6. Exergy destruction

Exergy destruction is an essential parameter in exergy modeling. It is a measure of irreversibility in the system. It is defined as the potential work lost due to irreversibility. For a control volume at steady state, the exergy destruction rate is defined as

$$\dot{E}x_d = \sum_j \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_j - \dot{W}_{cv} + \sum_{in} \dot{m}_i ex_{in} - \sum_e \dot{m}_e ex_e \quad (8)$$

where  $T$ ,  $\dot{E}x_d$ , and  $ex$  are temperature, rate of exergy destruction, and exergy per unit mass, respectively. The subscript 0 denotes the value of a property at the conditions of the surroundings while the subscript  $j$  denotes a property value at state  $j$ . The subscripts  $in$  and  $e$  refer to inlet and exit, respectively. The total exergy rate is defined as

$$ex_{total}^{PH} = ex^{PH} + \sum ex_j^{CH} \quad (9)$$

where  $ex^{PH}$  is the physical exergy per unit mass flow rate at a given state, defined as

$$ex^{PH} = (h - h_0) - T_0(s - s_0) + \frac{V^2 - V_0^2}{2} + g(z - z_0) \quad (10)$$

where  $h$ ,  $s$ ,  $V$ ,  $z$ , and  $g$  are enthalpy per unit mass, entropy per unit mass, velocity, elevation, and gravity, respectively. In this study, the velocity and elevation effects are very small and, hence, they are neglected. The chemical exergy per unit mass of an ideal gas for a species  $j$  is defined as [18]

$$ex_j^{CH} = [x_j \cdot \bar{e}x_j^{CH} + R \cdot T_0 \cdot x_j \cdot \ln(x_j)]/M_j \quad (11)$$

where  $M_j$  is the molecular weight of species  $j$  and  $\bar{e}x_j^{CH}$  is the standard chemical exergy value of species  $j$ . The variables  $x$  and  $R$  are the mole fraction and universal gas constant, respectively. In this study, the exergy of the moist air was calculated based on the approximation developed by Wepfer et al. [19] while the exergy of the seawater water was calculated using

the equations developed by Sharqawy et al. [20].

### 3.1.7. Overall exergetic efficiency

Exergetic efficiency is a measure of the useful exergy in the system and gives an indication of exergy destruction and loss in the system, and it indicates how far is the system considered departs from a reversible one. The overall exergetic efficiency of the system is a function of total exergy destruction, exergy loss, and exergy input to the system. It is defined as [21, 22]

$$\eta_{ex} = \frac{\dot{E}x_{useful}}{\dot{E}x_{in}} = \frac{\dot{E}x_{in} - \dot{E}x_{d,total} - \dot{E}x_{loss}}{\dot{E}x_{in}} \quad (12)$$

where  $\dot{E}x$  indicates an exergy rate. The subscripts *in* and *d* refer to the inlet and destruction, respectively.

### 3.1.8. Specific exergy destroyed and lost

The specific exergy destruction is defined as the exergy destruction for a single component, *i*, per unit mass of purified water produced. In equation form, it may be represented as

$$ex_{d,i,mp} = \frac{\dot{E}x_{d,i}}{\dot{m}_{p,total}} \quad (13)$$

Similarly, the specific exergy lost per unit mass of water produced is defined as

$$ex_{Loss} = \frac{\dot{E}x_{Loss}}{\dot{m}_{p,total}} \quad (14)$$

### 3.1.9. Total true specific exergy lost

A new parameter named the total true specific exergy lost is defined in this study. It is the summation of the specific exergy destroyed by all the system components and the specific exergy lost:

$$ex_{DL} = \frac{\dot{E}x_{D,total} + \dot{E}x_{Loss}}{\dot{m}_{p,total}} \quad (15)$$

This parameter provides a better indication of the system exergetic performance as compared to the exergy destruction rate or overall exergetic efficiency of the system, as discussed in the next section.

### 3.1.10. Exergy destruction ratio

Exergy destruction ratio of a component in a thermal or desalination system is defined as the ratio of the of exergy destruction of a component to the total exergy destruction of the system:

$$Y_{D,i} = \frac{\dot{E}x_{d,i}}{\dot{E}x_{d,total}} \quad (16)$$

The exergy destruction ratio indicates the exergy destruction contributed by each component and is therefore important to understanding how best to improve system performance.

### 3.2. Component Modeling

In this subsection, models for the main components in the system considered are presented. These components are the humidifier, dehumidifier, thermal vapor compressor (TVC), and expander.

#### 3.2.1. Humidifier

The humidifier is a counterflow heat and mass exchanger where one fluid stream is water while the other fluid stream is a mixture of air and water vapor (moist air). The moist air stream enters at state 1, absorbs evaporating water and exits at state 2 with a higher water content (more humid air). The seawater enters at state 6 transfers water and energy to the moist air in the humidifier and exits at state 7 with a somewhat lower mass flow rate of water. The mass balance on the humidifier is

$$\dot{m}_7 = \dot{m}_6 - \dot{m}_{da} \cdot (\omega_2 - \omega_1) \quad (17)$$

where  $\omega$  is the absolute humidity.

The First Law for the humidifier is

$$\dot{m}_{da}(h_1^* - h_2^*) = \dot{m}_6 h_6 - \dot{m}_7 h_7 \quad (18)$$

where  $h_1^*$  and  $h_2^*$  are enthalpy per unit mass of dry-air while  $h_6$  and  $h_7$  are enthalpy per unit mass of water. One further equation is needed to solve the energy equation for the humidifier, which is the energy effectiveness. The energy effectiveness of the humidifier is defined using Equation 1. The definition of this effectiveness is defined based on the operating conditions. There are two cases [16]:

Case I,  $\Delta\dot{H}_{max,w,H} < \Delta\dot{H}_{max,a,H}$ :

$$\varepsilon_H = \frac{\dot{m}_6 h_6 - \dot{m}_7 h_7}{\dot{m}_6 h_6 - \dot{m}_7 h_7^{ideal}} \quad (19)$$

Case II,  $\Delta\dot{H}_{max,w,H} > \Delta\dot{H}_{max,a,H}$ :

$$\varepsilon_H = \frac{h_2^* - h_1^*}{h_2^{*,ideal} - h_1^*} \quad (20)$$

The Second Law for the humidifier is defined as

$$\dot{m}_6 \cdot s_6 - \dot{m}_7 \cdot s_7 - \dot{m}_{da} \cdot (s_2^* - s_1^*) = -\dot{S}_{gen_H} \quad (21)$$

where  $s$  is the specific entropy and  $\dot{S}_{gen}$  is the entropy generation rate.

### 3.2.2. Dehumidifier

The dehumidifier is counterflow heat and mass exchanger where one fluid stream is water while the other fluid stream is moist air. In the system considered, the moist air stream enters at state 3 and exits at state 4 with a lower water content (less humid air). The seawater enters at state 5 mixes with air in the dehumidifier and exits at state 6 with a relatively higher temperature. The water mass balance for the dehumidifier is

$$\dot{m}_7 = \dot{m}_6 - \dot{m}_{da} \cdot (\omega_2 - \omega_1) \quad (22)$$

Applying the First Law of Thermodynamics to the dehumidifier gives

$$\dot{m}_{da} \cdot (h_3^* - h_4^*) = \dot{m}_5 \cdot (h_6 - h_5) + \dot{m}_8 \cdot h_8 \quad (23)$$

A further equation is needed to solve the above equation and as before the energy effectiveness for the dehumidifier, Equation 1, may be used as follows:

Case I,  $\Delta\dot{H}_{max,w,D} < \Delta\dot{H}_{max,a,D}$ :

$$\varepsilon_D = \frac{h_3^* - h_4^*}{h_3^* - h_4^{*,ideal}} \quad (24)$$

Case II,  $\Delta\dot{H}_{max,w,D} > \Delta\dot{H}_{max,a,D}$ :

$$\varepsilon_D = \frac{\dot{m}_{da}h_4^* - \dot{m}_{da}h_3^* + \dot{m}_8h_8}{\dot{m}_{da}h_4^{*,ideal} - \dot{m}_{da}h_3^* + \dot{m}_8^{ideal}h_8^{ideal}} \quad (25)$$

The Second Law for the dehumidifier can be defined as

$$\dot{m}_{da} \cdot (s_3^* - s_4^*) - \dot{m}_5 \cdot (s_6 - s_5) - \dot{m}_8 \cdot s_8 = -\dot{S}_{gen_D} \quad (26)$$

### 3.2.3. Thermal vapor compressor

The TVC is used to compress the moist air by mixing the moist air with high pressure steam. The moist air enters at state 4 mixes with the steam, and this mixture exits the TVC as moist air with higher pressure and humidity, state 3. Some water may condense out at the exit of the TVC, state 10. The dry air has a constant mass flow rate. Therefore, the mass flow rate of condensed water at the exit of the TVC can be found using this equation

$$\dot{m}_{da} \cdot (\omega_3 - \omega_2) = \dot{m}_{st} - \dot{m}_{10} \quad (27)$$

The TVC efficiency equation, Equation 7, is used to complete the calculation

$$\eta_{TVC} = \frac{\dot{m}_{st}^{rev}}{\dot{m}_{st}} \quad (28)$$

where  $\dot{m}_{st}^{rev}$  is the mass flow rate for the reversible case which can be calculated using the First and Second Laws of Thermodynamics:

$$\dot{m}_{st}^{rev} h_{st,in} = \dot{m}_{da} h_3^{*,rev} - \dot{m}_{da} h_2^* + \dot{m}_{10} h_{10} \quad (29)$$

$$\dot{m}_{st}^{rev} s_{st,in} = \dot{m}_{da} s_3^{*,rev} - \dot{m}_{da} s_2^* + \dot{m}_{10} s_{10} \quad (30)$$

For the reversible case, the discharged pressure (state 3) is the same as the real case. We may again apply the First and Second Laws of Thermodynamics to obtain

$$\dot{m}_{st} h_{st,in} = \dot{m}_{da} h_3^* - \dot{m}_{da} h_2^* + \dot{m}_{10} h_{10} \quad (31)$$

$$\dot{S}_{gen,TVC} = \dot{m}_{da} s_3^* - \dot{m}_{da} s_2^* + \dot{m}_{10} s_{10} - \dot{m}_{st} s_{st,in} \quad (32)$$

The pressure ratio of the TVC is defined as

$$PR = P_3/P_2 \quad (33)$$

### 3.2.4. Expander

Moist air enters the expander and is cooled at the exit. Due to the cooling, some water may condense out. The mass flow rate of this condensed water can be calculated using the following mass balance:

$$\dot{m}_9 = \dot{m}_{da} \cdot (\omega_4 - \omega_1) \quad (34)$$

Apply the First Law for the expander to obtain

$$\dot{W}_{exp} = \dot{m}_{da} \cdot (h_4^* - h_1^*) - \dot{m}_9 \cdot h_9 \quad (35)$$

where  $\dot{W}_{exp}$  is the power produced by the expander. A further auxiliary equation is needed. Use the isentropic efficiency definition of the expander, Equation 5, to obtain

$$\eta_e = \frac{\dot{m}_{da} \cdot (h_4^* - h_1^*) - \dot{m}_9 \cdot h_9}{\dot{m}_{da} \cdot (h_4^* - h_1^{*,s}) - \dot{m}_9 \cdot h_9} \quad (36)$$

Now, apply the Second Law to the expander to obtain

$$\dot{m}_{da} \cdot (s_4^* - s_1^*) - \dot{m}_9 \cdot s_9 = -\dot{S}_{gen,exp} \quad (37)$$

The brine that exits from the humidifier, state 7, is further used to produce pure water through the RO. The work produced by the expander is used to run a pump; and the pump in turn is used to run the RO. The energy consumed by the RO system is assumed to be 3.5 kWh/m<sup>3</sup> at a water recovery ratio of 50% [2]. These two assigned performance values are representative of RO desalination systems from medium to large scale [23]. The mass flow rate of the pure water produced through such RO systems may be calculated as

$$\dot{m}_{pw,RO} = \frac{\dot{W}_{out}}{3.6 \cdot \dot{E}_{c,RO}} \quad (38)$$

### 3.3. Solution technique

The solution of the governing equations was carried out using Engineering Equation Solver (EES) [24]. EES is a numerical solver that identifies the equations and then groups them. It solves the grouped equations by an iterative procedure. The solution is considered converged if two conditions are satisfied for each equation solved in the model. The first condition is that the magnitude of the relative residual is less than 10<sup>-6</sup>. The relative residual is defined as the difference between the left-hand and right-hand sides of the equation solved in EES divided by the left-hand side of this equation. The second condition is the change in variables is less than 10<sup>-9</sup>. It is defined as the change in the value of a variable between the current and previous iteration. The value of these conditions are the standard conditions in EES and have been used in several previous thermodynamic studies, e.g. [2, 25, 15, 17, 26, 27].

Dry air properties used in EES are evaluated using the ideal gas formulations presented by Lemmon [28]. Moist air properties in EES are evaluated assuming an ideal mixture of air and steam as presented by Hyland and Wexler [29]. Seawater properties are evaluated using the correlations developed by Sharqawy et al. [20].

## 4. Results and Discussion

In this section, effects of varying steam pressure, TVC pressure ratio, TVC efficiency, and expander efficiency on key performance parameters are presented and discussed. These parameters are overall exergetic efficiency, electricity equivalent consumption, specific exergy destruction, specific exergy lost, and total true specific exergy lost. Furthermore, the ratios of exergy destruction by individual components are presented and discussed at the end of this section. The operating conditions of the system stations are given in Table 1.

The performance parameters are evaluated under the baseline conditions. The baseline conditions are given in Table 2:  $P_{st} = 5000$  kPa,  $T_0 = 25^\circ\text{C}$ ,  $PR = 1.2$ ,  $\eta_{TVC} = 30\%$ , and  $\eta_{exp} = 50\%$ . The selected baseline pressure, 5000 kPa, is a representative value for a relatively low- or medium-grade heat value, such as waste heat, geothermal energy, or solar energy. The selected TVC pressure ratio, 1.2, is a typical value for multi-pressure HDH desalination systems [2]. The efficiency of the expander is selected to be 50% which is representative of typical expanders at the selected steam pressure and pressure ratio through the expander ( $1/PR$ ).

Another parameter that can be used to assess the exergetic performance is the total exergy destruction rate. However, this parameter was not used in this study since the present simulation is a design simulation and not a rating simulation. Therefore, examining total exergy destruction rate is not appropriate. It can be used for a fixed size system and the resulting value would be meaningful only for that system. Therefore, there is a need to find an exergetic parameter having a value that can be compared to other systems. Hence, total true specific exergy lost is introduced.

#### 4.1. Steam pressure effect

The effect of steam pressure is studied in this section. The steam pressure is the driving force of the system. Therefore, it is important to examine the steam pressure variation and to identify under which steam pressure the system proposed works most efficiently. The effect of steam pressure on the selected parameters is shown in Figs. 3 and 4.

The overall exergy efficiency of the system decreases as the steam pressure increases, as shown in Fig. 3. It decreases from almost 12% at 1000 kPa to 10% at 10,000 kPa. This decrease in the overall exergy efficiency is attributed to an increase in the amount of energy entering the system as the steam pressure increases and, consequently, the amount of exergy entering the system increases. Note that the exergy entering the system is the denominator of the overall exergy efficiency, Equation 12. The other  $y$ -axis shows the electricity equivalent consumption and as defined before it is the electricity consumption per water produced. Therefore, a decrease in this parameter should result in saving, and, consequently, increases efficiency for both energy and exergy. Nevertheless, this figure illustrates that the increase in steam pressure results in an exergetic efficiency decrement while the equivalent electricity consumption decreases. This figure gives an indication that the exergy efficiency is not a good parameter to measure the performance of a water desalination system.

To demonstrate that further, the effect of steam pressure on the total mass flow rate of the water produced, and specific exergy destroyed and lost is examined as shown in Figure 4. It can be observed that the total mass increases from 6 to 8.8 kg/s as the steam pressure increases from 1,000 to 10,000 kPa. Furthermore, it was shown in a previous publication

[2], that increasing the steam pressure raises the gained-output ratio (GOR) and decreases the equivalent electrical consumption. These results recommend operation at high steam pressure. However, considering the previous figure, Fig. 3, the overall exergetic efficiency decreases as the steam pressure increases, which could result in a misleading conclusion. Therefore, overall exergetic efficiency for such a desalination system is not recommended as an exergetic assessment parameter.

A different exergetic parameter is recommended instead, which is the total true specific exergy lost,  $ex_{DL,mp}$ . Figure 4 demonstrates that as the steam pressure increases, this parameter decreases. This parameter indicates that the system's exergetic performance improves as the steam pressure increases. The conclusion that is drawn from this parameter is similar to the conclusion drawn from the effect of steam pressure on the total mass flow rate of the water produced, from the GOR, and from the equivalent electrical consumption. On the other hand, the figure shows that the specific exergy destruction of all components are almost constant except for that of the dehumidifier. The specific exergy destruction of the dehumidifier decreases from 24 kJ/kg at 1000 kPa to 14 kJ/kg at 10,000 kPa. It decreases noticeably up to 2,000 kPa; and then it keeps decreasing marginally as the steam pressure increases. Therefore, it is recommended to focus on improving the design of the dehumidifier when a similar system is intended to be operated at low steam pressure, below 2,000 kPa. The total true specific exergy decreases noticeably from 105 kJ/kg at 1,000 kPa to 100 kJ/kg at 2,000 kPa; and then it decreases marginally as the pressure increases to reach 94 kJ/kg at 10,000 kPa. This figure reveals that the trend of the total true specific exergy destruction is clearly governed by the trend of the specific exergy destruction of the dehumidifier. On the other hand, the TVC has the highest specific exergy destruction and, therefore, careful design and selection of a TVC for such a desalination system is most essential.

#### 4.2. TVC pressure ratio effect

The variation of the TVC pressure ratio is examined between 1.1 and 1.8, as shown in Figs. 5 and 6. Figure 5 illustrates the effect of pressure ratio on the overall exergy efficiency and equivalent electricity consumption. It can be noticed that the overall exergy efficiency increases from around 9.4 to 11.8% as the pressure ratio increases from 1.1 to 1.8. In addition, this figure demonstrates that the equivalent electricity consumption increases as the pressure ratio increases. This result suggests operating the system at high TVC pressure ratio from exergetic point of view; however, this apparent conclusion is not correct, as discussed in the next paragraph.

The variation of the specific exergy destruction and lost, and the total mass flow rate of the water produced as the pressure ratio of the TVC varies are shown in Figure 6. This figure reveals that the total mass flow rate of the water produced decreases from 8.4 to 7.4



kg/s as the pressure ratio increases from 1.1 to 1.8. On the other hand, as expected, the total true specific exergy lost increases as the pressure ratio increases from 1.1 to 1.8; it increases from 90 to 112 kJ/kg. That is, this parameter recommends operation of the system under low TVC pressure ratio to get a high mass flow rate of pure water per unit energy input. This conclusion agrees with the results from the previous study on energy analysis of the desalination system considered where the highest GOR and lowest equivalent electrical consumption occur at the a pressure ratio of 1.1 [2]. These conclusions indicate again that overall exergetic efficiency is not an appropriate exergetic performance parameter for such a desalination system. On other hand, it can be noticed that the specific exergy destruction of all the main components are almost constant except for the dehumidifier. The dehumidifier specific exergy destruction increases from around 10 kJ/kg at a pressure ratio of 1.1 to 30 kJ/kg at 1.8. This figure demonstrates that the exergy destruction of the dehumidifier is the main factor of increasing the total true specific exergy destruction. In addition, this figure shows that the dehumidifier is sensitive to the pressure ratio change and that it needs further improvement to reduce this sensitivity and, thus, to increase the mass of water produced and reducing the total true specific exergy destruction.

#### *4.3. TVC efficiency effect*

The effect of varying the TVC efficiency is shown in Figs. 7 and 8. Figure 7 illustrates that the overall exergy efficiency increases from 8.6% at TVC efficiency of 10% to 10.8% at TVC efficiency of 40%. The other  $y$ -axis shows that the equivalent electricity consumption decreases as the TVC efficiency increases. It decreases from 72 kW<sub>e</sub>h/m<sup>3</sup> at TVC efficiency of 10% to 16 kW<sub>e</sub>h/m<sup>3</sup> at TVC efficiency of 40%.

On the other hand, the total mass flow rate of water produced increases significantly from 2.7 to 10.3 kg/s as the TVC efficiency increases from 10 to 40%, as seen in Fig. 8. That is, the highest exergy efficiency occurs at the highest water mass flow rate produced which is unlike the trend of the overall exergy efficiency from the previous two figures, Figs. 3 and 5, where the highest exergetic efficiency occurs at the lowest water mass flow rate produced. This conclusion demonstrates that the overall exergetic efficiency is not consistent and, therefore, not recommended as an assessment parameter for a desalination system. Alternatively, the specific exergy destruction and loss are exergetic performance parameters which are always consistent. The total true specific exergy lost decreases significantly from 315 kJ/kg at a TVC efficiency of 10% to 70 kJ/kg at 40%. This significant drop in the total true specific exergy lost is attributed to the significant drop in the specific exergy destruction of the TVC as shown in this figure. This figure reveals that both the TVC and dehumidifier specific exergy destruction values are sensitive to the variation of the TVC efficiency. Note that it is

expected that the specific exergy destruction of the TVC will be sensitive to the variation of the TVC efficiency since it is direct function of it.

#### 4.4. Expander efficiency effect

The effect of varying expander efficiency is shown in Figs. 9 and 10. Figure 9 illustrates the effect of the expander efficiency on the overall exergetic efficiency and equivalent electricity consumption. It is shown that the overall exergetic efficiency increases from approximately 9.5 to 13.5% as the expander efficiency increases from 40 to 90%. In addition, it is shown that the equivalent electricity consumption decreases from 27 to 13 kW<sub>e</sub>h/m<sup>3</sup> as the expander efficiency increases from 40 to 90%. The effect of expander efficiency variation on the total mass flow rate of the water produced and specific exergy destruction and lost is illustrated in Fig. 10. The total mass flow rate of water produced increases considerably from 6.2 to 14.1 kg/s as the expander efficiency increases from 40 to 90%. That is, to achieve a high water mass production rate, the overall exergetic efficiency must be high.

This result demonstrates again that the overall exergetic efficiency is not consistent with the water mass flow rate produced because this efficiency may increase or decrease as the mass flow rate of water produced increases. Alternatively, the total true specific exergy lost is always consistent. The total true specific exergy lost decreases noticeably from 120 to 52 kJ/kg as the expander efficiency increases from 40 to 90%. This figure reveals that the main reason for the reduction in the total true specific exergy lost as the expander efficiency increases is the decrement in the specific exergy destruction of the TVC; and to a lower extent it is also affected by the decrement of the expander and dehumidifier specific exergy destructions.

#### 4.5. Overall exergy destruction ratio

The exergy destruction ratios of the main components, as well as the exergy lost ratios are shown in Fig. 11 at baseline values. This figure reveals that the main source of exergy destruction is the TVC, accounting for about 50% of the total. Moreover, this figure demonstrates that the dehumidifier, expander, and RO have relatively significant exergy destruction ratios, which are 0.15, 0.1, and 0.08, respectively. The fraction of the exergy lost is small at around 0.05, which is attributed to the small magnitude of the thermal disequilibrium for the desalination system considered [22].

## 5. Conclusions

In this study, an exergy analysis of a novel desalination system was conducted. The main findings from this study are as follow.

- A new parameter that provides useful assessments of the exergetic performance of the desalination system is introduced. This parameter is the summation of specific exergy destruction and loss and is called total true specific exergy lost.
- For the desalination system considered, examining the overall exergetic efficiency as a performance parameter can yield misleading conclusions. It was shown that conditions with the total mass flow rate of pure water produced do not always result in increases in exergetic efficiency. Therefore, it is not recommended to use the exergetic efficiency as an assessment parameter for such a desalination system.
- The specific exergy destruction of the dehumidifier and of the TVC are the most sensitive parameters driving the operating or design variables. Therefore, further design improvements are needed for the dehumidifier and TVC to improve the performance of the system.
- The highest specific exergy destruction is due to the TVC. This exergy could be reduced significantly if a high efficiency TVC were available.
- Increasing the TVC pressure ratio from 1.1 to 1.8 results in a reduction of the total mass of the pure water produced and an increase in the total true specific exergy lost. Conversely, increasing the other parameters, including steam pressure, efficiency of the TVC, or efficiency of the expander, results in a higher total water production and lower total true specific exergy loss.

## 6. Acknowledgement

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Table 1: Thermodynamic properties at each station.

Station	Fluid type	$T$ ( $^{\circ}\text{C}$ )	$\dot{m}$ (kg/s)	$h$ (kJ/kg)	$s$ (kJ/kg $\cdot$ K)	$\omega$ (kg-H <sub>2</sub> O/ kg-dry-air)
Inlet steam	Steam	264	0.8	2794	5.97	
1	Moist air	37.22	11.8	169.3	6.241	0.051
2	Moist air	43.28	12.04	232.4	6.443	0.073
3	Moist air	58.53	12.82	430.1	6.994	0.142
4	Moist air	41.21	11.82	178.3	6.213	0.053
5	Seawater	25.0	30.73	100.5	0.3535	
6	Seawater	46.44	30.73	186.8	0.6332	
7	Seawater	41.06	30.49	165.1	0.5647	
8	Water Product	41.21	1.0	172.7	0.5885	
9	Water Product	37.22	0.02	156	0.5351	
10	Water Product	58.53	0.021	245.1	0.8127	
11	Seawater	41.06	30.49	165.1	0.5647	
12	Brine	41.06	22.69	165.1	0.5647	
13	Water Product	41.06	7.794	165.1	0.5647	

Table 2: Baseline data [2]

Expander efficiency	50%
TVC efficiency	30%
TVC pressure ratio	1.2
Steam pressure	5000 kPa
Pump efficiency	70%
$\text{HCR}_H$	1
Seawater Salinity	35 g/kg
Ambient pressure	101.3 kPa
Ambient relative humidity	0.62
Ambient temperature	25 $^{\circ}\text{C}$



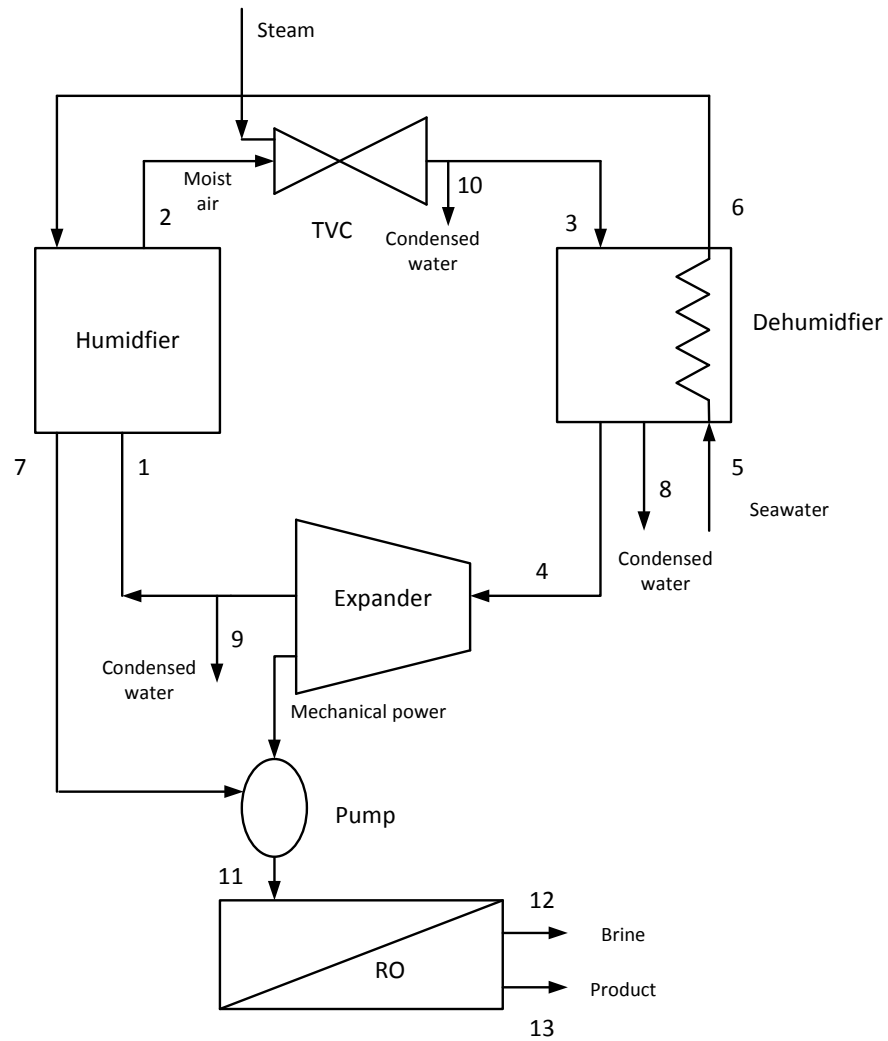


Figure 1: Schematic diagram of the HDH-RO-TVC desalination system considered.

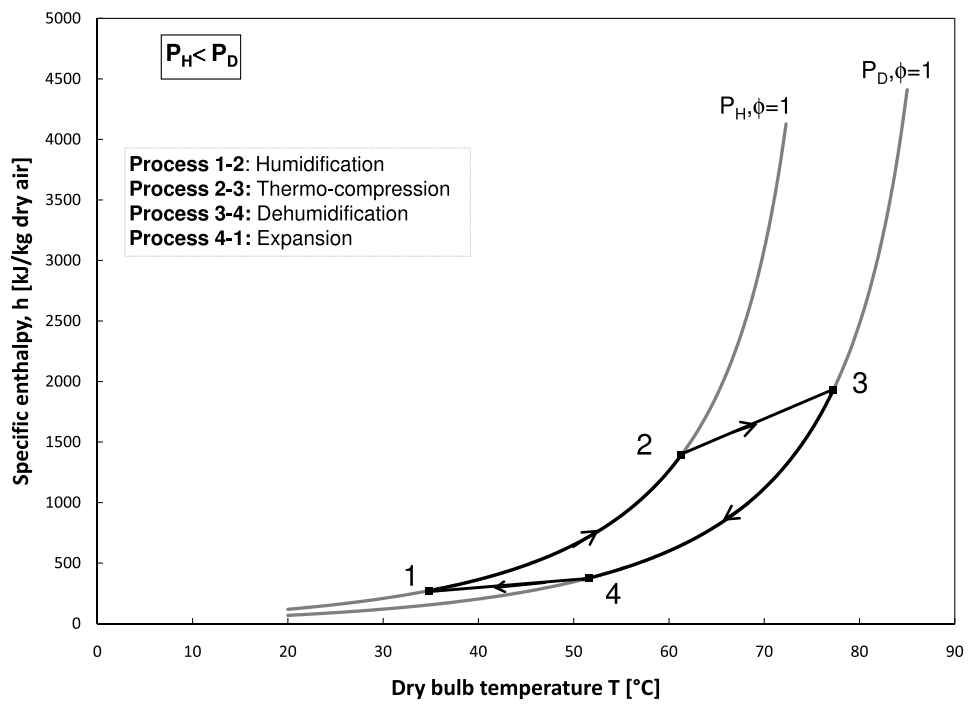


Figure 2: Psychrometric chart of the moist air cycle in the desalination system considered.

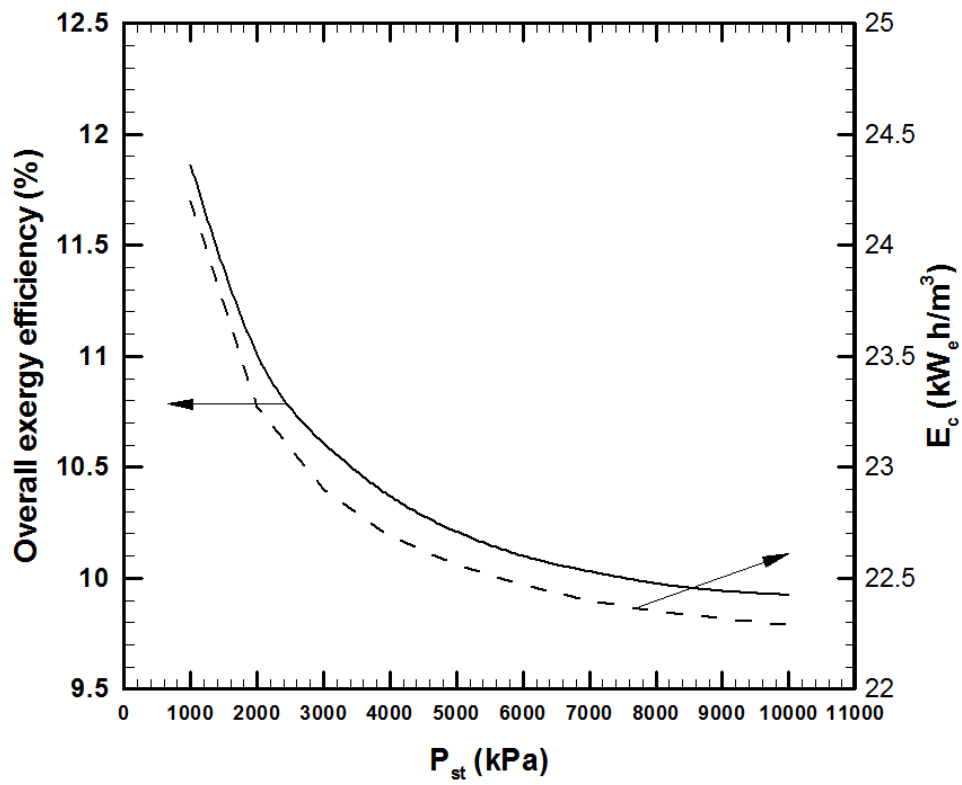


Figure 3: Effect of the steam pressure on the overall system exergetic efficiency.

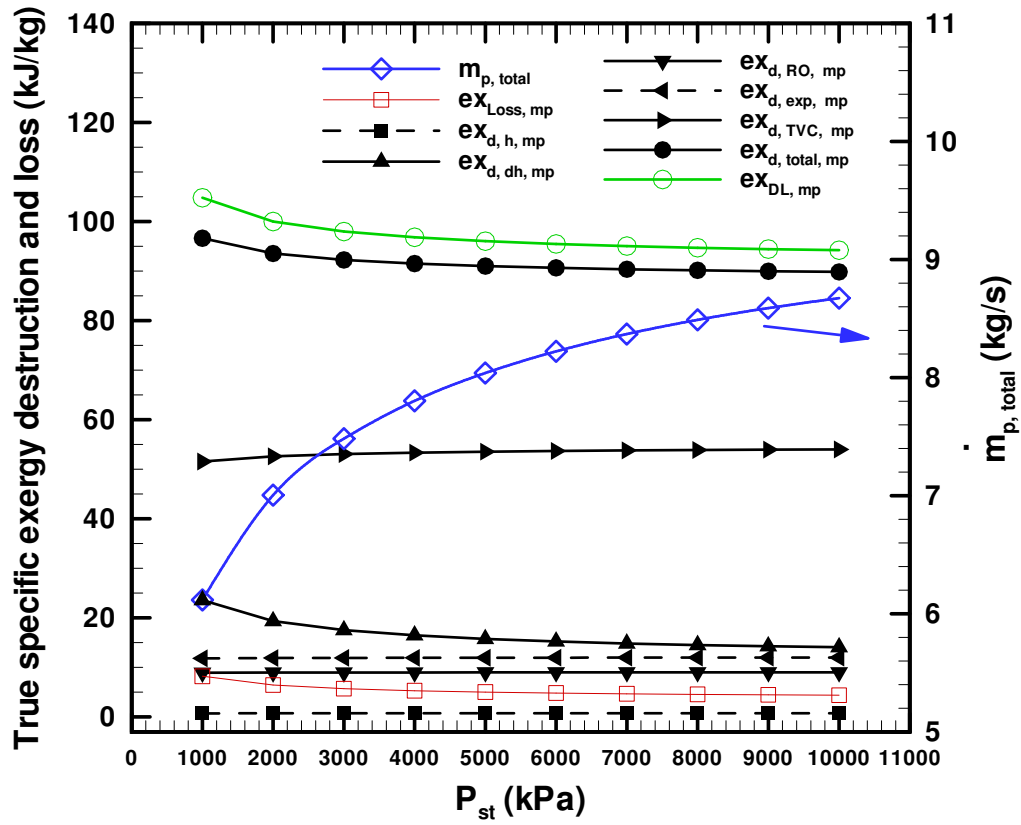


Figure 4: Effect of the steam pressure on the total mass flow rate of the water produced and specific exergy destruction and lost.

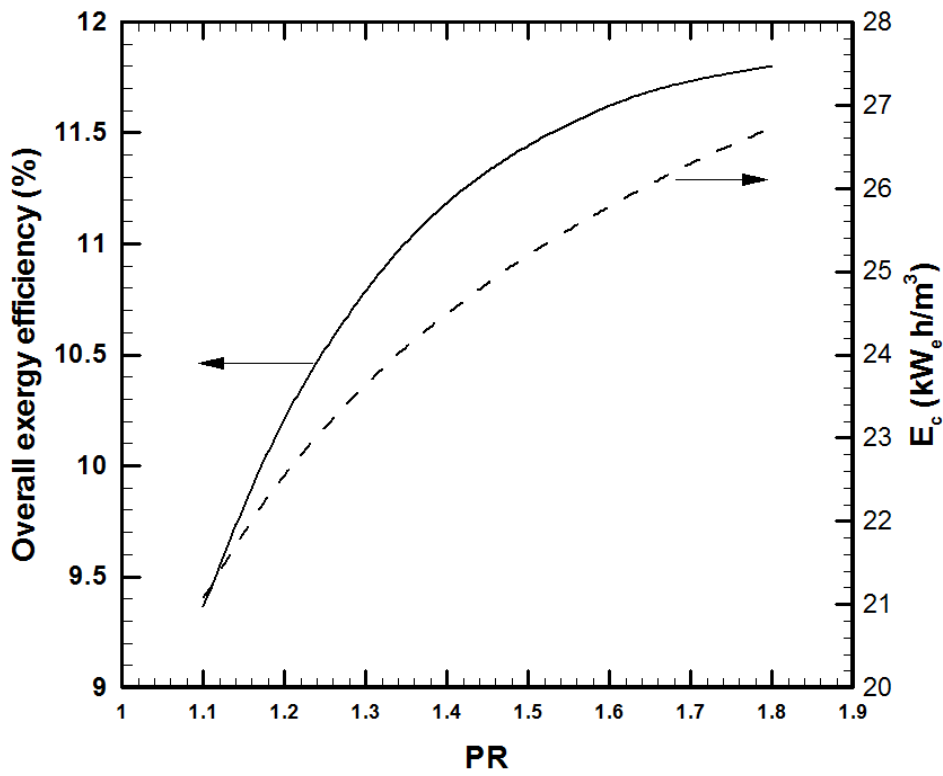


Figure 5: Effect of the TVC pressure ratio on the overall system exergetic efficiency.

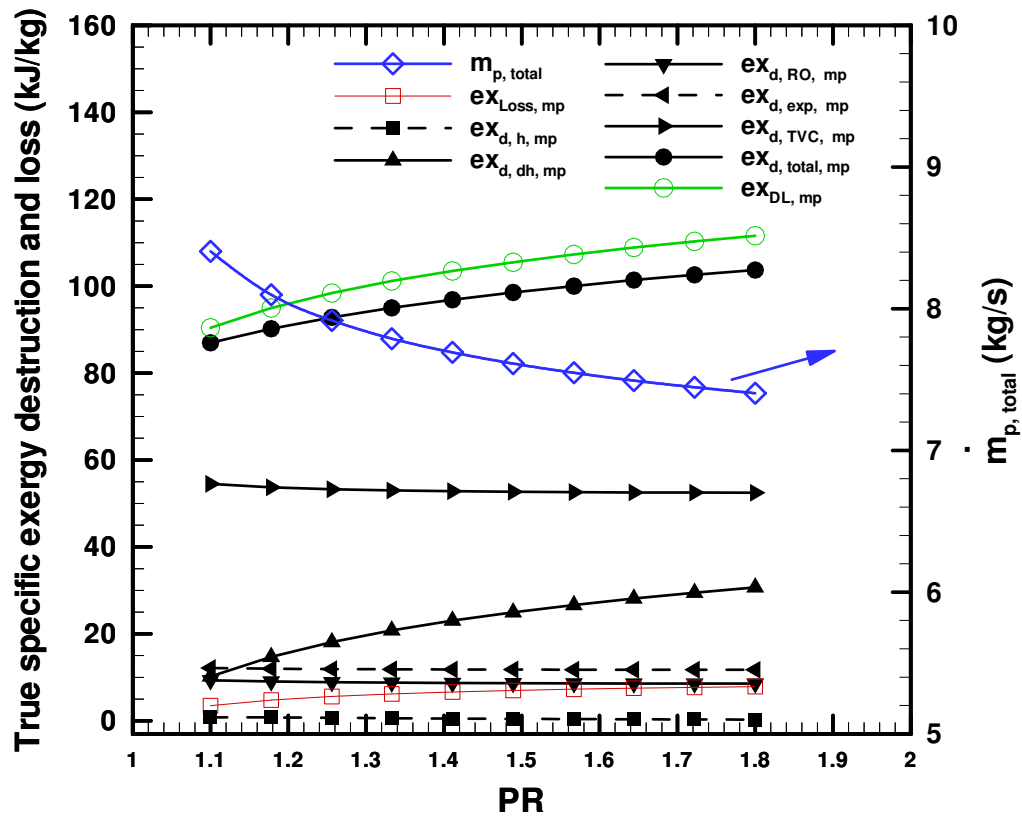


Figure 6: Effect of the TVC pressure ratio on the total mass flow rate of the water produced and specific exergy destruction and lost.

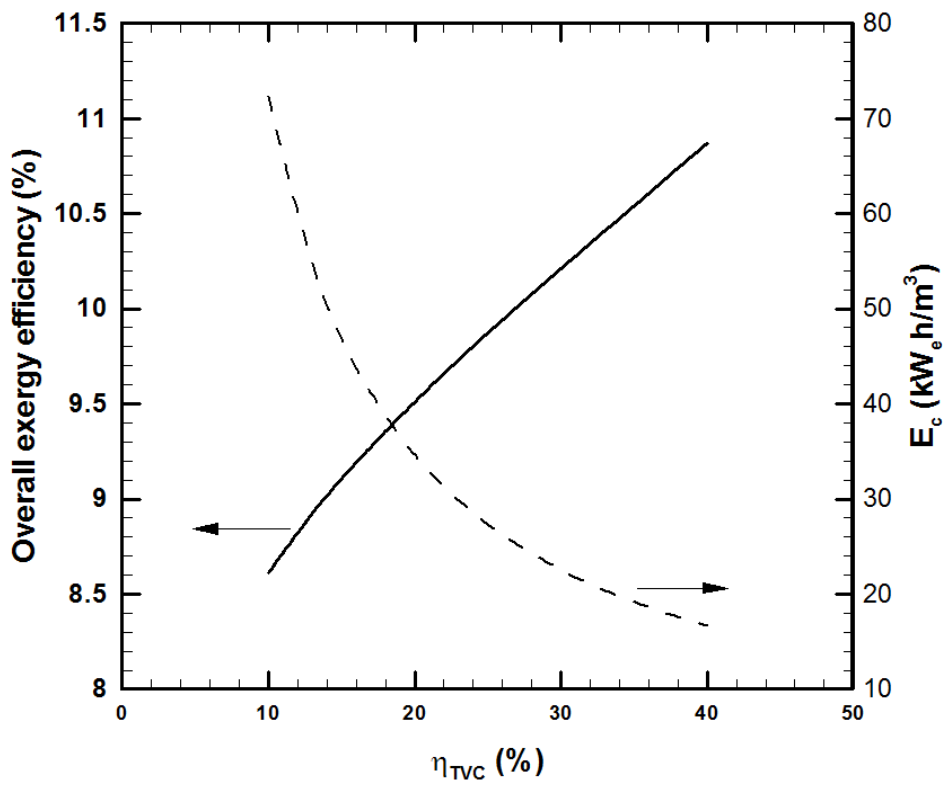


Figure 7: Effect of the TVC efficiency on the overall system exergetic efficiency.

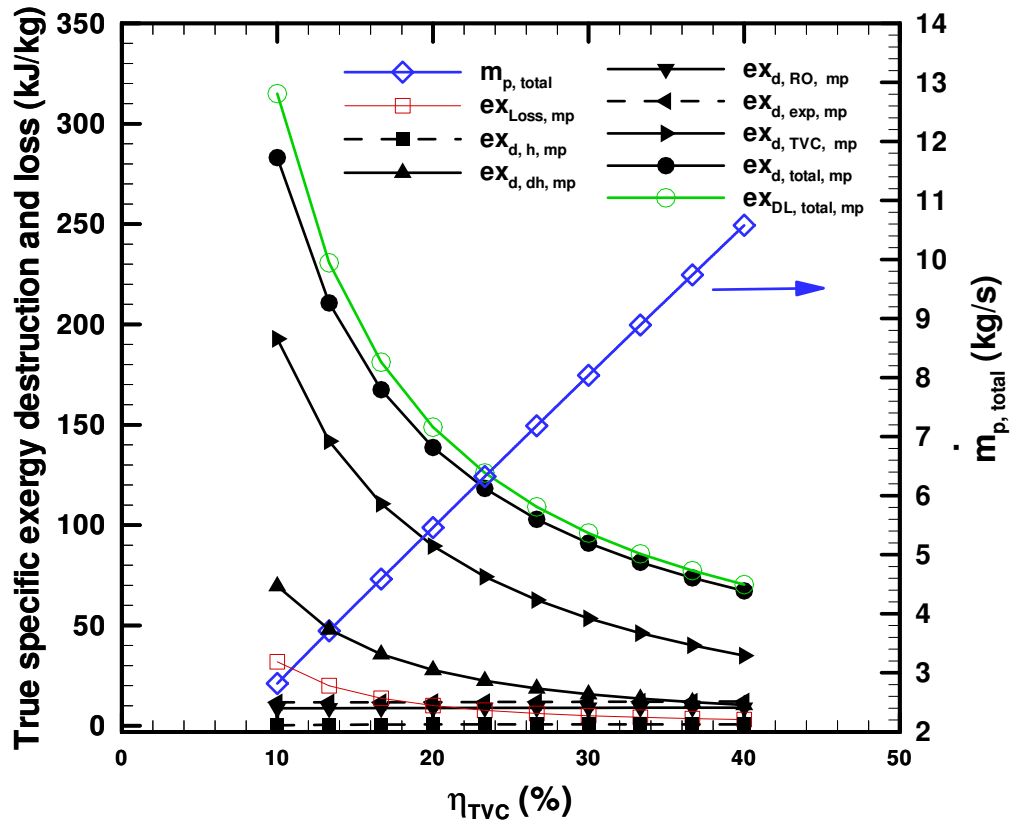


Figure 8: Effect of the TVC efficiency on the total mass flow rate of the water produced and specific exergy destruction and lost.



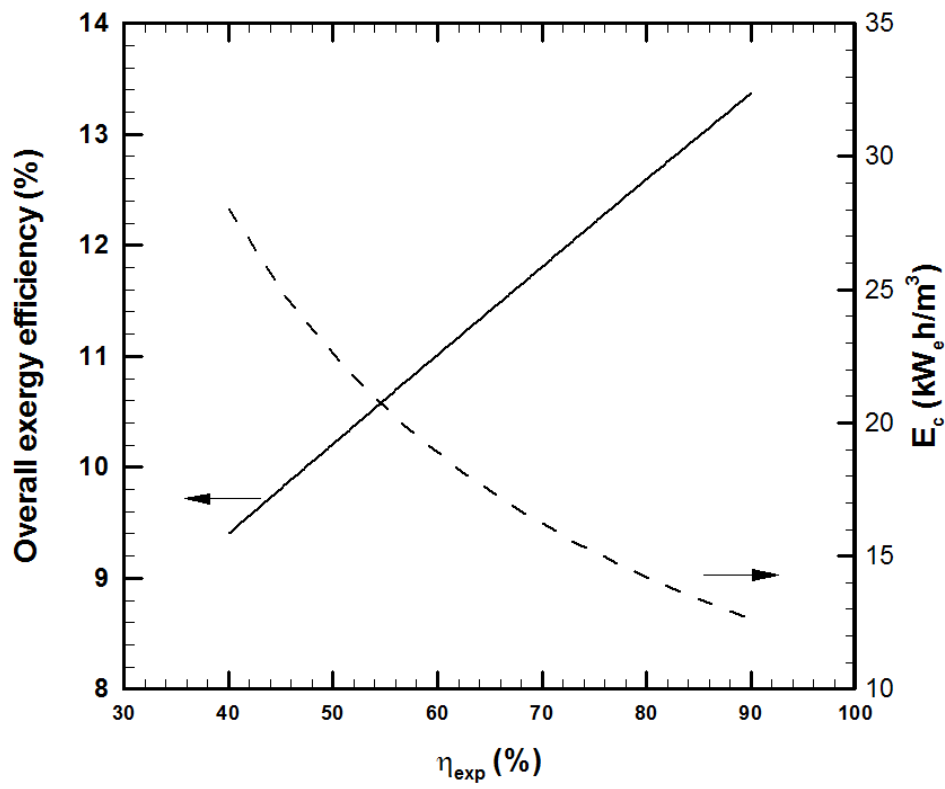


Figure 9: Effect of the expander efficiency on the overall system exergetic efficiency.

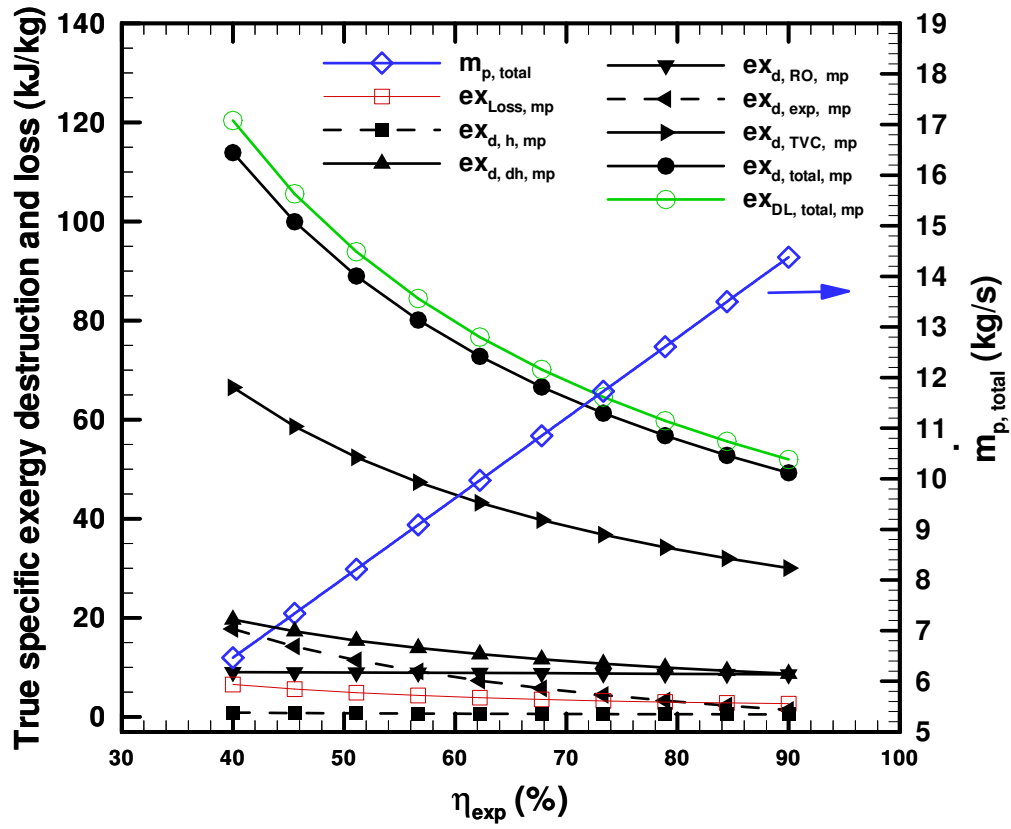


Figure 10: Effect of the expander efficiency on the total mass flow rate of the water produced and specific exergy destruction and lost.

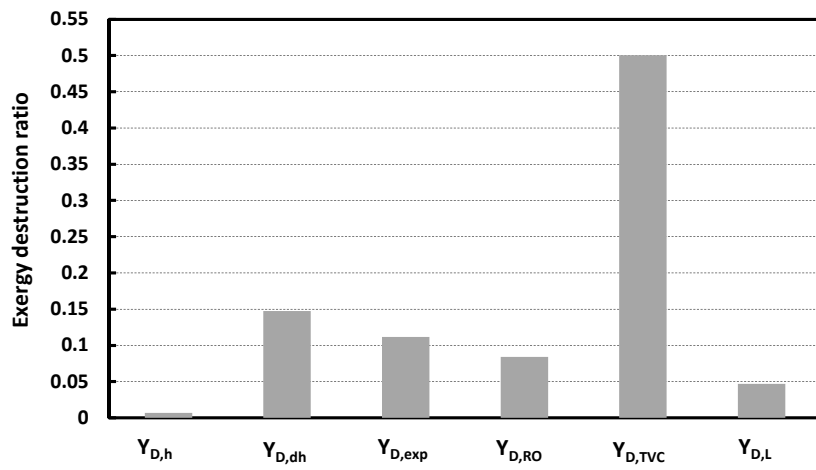


Figure 11: Exergy destruction ratios and exergy loss ratio of the desalination system considered at baseline values.