The Journal of Energy: Engineering & Management
Vol. 2, No. 4, Winter 1391, P. 32-39

Thermodynamic and Exergy Analysis of a Combined Power and Desalination Plant

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Received: 13 September 2011 Revised: 17 November 2011
Accepted: 20 June 2012

Abstract

Making potable water through desalination plants is a very important process in Iran where clean water is highly required. On the other hand, large amount of fossil fuel sources leads to the development of gas turbine power plants all over the country. Furthermore, Persian Gulf in the south and Caspian Sea in the north could be the main sources for supplying potable water in water scarcity areas, especially in the south. Dual-purpose plants are the ones that supply heat for a thermal desalination unit and produce electricity for distribution to the electrical grid. In this paper a gas turbine power plant was combined with a multi stage flash desalination system. Then, energy and exergy analysis for desalination plant, power generation cycle, heat recovery steam generator (HRSG), and combined power and water cycle were developed. Results showed that by increasing the number of desalination effect, performance ratio, exergetic efficiency, and specific heat transfer area steadily increase. Additionally, the sensitivity analysis showed the relationship between the parameters that are not known precisely or difficult to foresee such as steam pressure of HRSG and compression ratio on the performance parameters of dual purpose plant.

Keywords: Power plant, Multi stage flash desalination, Heat recovery steam generation, Co-generation, Exergy analysis.

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1. Introduction

Water is available in large quantities on earth but only a small fraction has a low enough salinity to be suitable for drinking and irrigation [1]. From the past, saving and making desalination of water has been noticed in the world. Desalination is one of the most important processes to provide water for population in water scarcity areas, especially in the Persian Gulf Area. But desalination processes consume a lot of energy; unfortunately the majority of the energy currently used for desalination is obtained from oil or natural gas [2]. Since seawater desalination plants require considerable energy resources, they are normally associated with the power plants as dual-purpose plants.

Large dual-purpose plants are built to reduce the cost of electricity production and freshwater. The dual purpose power desalination plants use thermal energy extracted or exhausted from power plants in the form of low-pressure steam to provide heat input for thermal desalinations. Such systems include multi-stage flash (MSF) or multi-effect distillation (MED) systems. The profitability of dual-purpose plants from an economic point of view is usually dependent on its impact on power system expenditures throughout the life time of the plant. They usually use waste energy of flue gas exited from gas turbine cycles or extracted steam of steam turbines as dual purpose plants. For thermodynamic evaluation of dual purpose plants, energy and exergy analysis can be useful. These thermodynamic methods can show efficiency and exergy destruction of all part of the systems and help the researchers improve system performance. Many researchers studied and evaluated power and water production systems. Some of them were basically concerned with economical aspect and have not done a complete thermodynamic evaluation.

Many researchers have studied thermal desalination from thermodynamic and economic point of view. Tadros [3], Rautenbach and Arzt [4], El-Nashar [5], Kahraman and Cengel [6], Karl et al. [7], Shih [8], Ji et al. [9], studied different aspects of thermal desalination and developed thermodynamic model to investigate effects of various parameters on the performance of systems. In all of these studies, energy or exergy analysis or heat and mass transfer simulation of thermal system without economic consideration are presented.

On the other hand, in recent years some studies focused on dual-purpose plant included of power generation cycle and a mechanical or thermal desalination from thermodynamic perspective [10-19].

In this paper, a combined gas turbine power plant and a multi stage flash desalination were simulated and energy and exergy analysis were done for all parts of the system. In addition, sensitivity analysis is studied for the identification of important parameters such as number of MSF effects, HRSG steam pressure, and pressure ratio of the compressor in the performance parameters of hybrid plant.

Fig. 1 illustrates the schematic of the combined GT-MSF system for simultaneous generation of the electric power and fresh water. Power generation cycle includes compressor, combustion chamber, and gas turbine that have a nominal output power of 65 MW. Also, a heat recovery steam boiler was used to produce saturated steam of distillation unit.

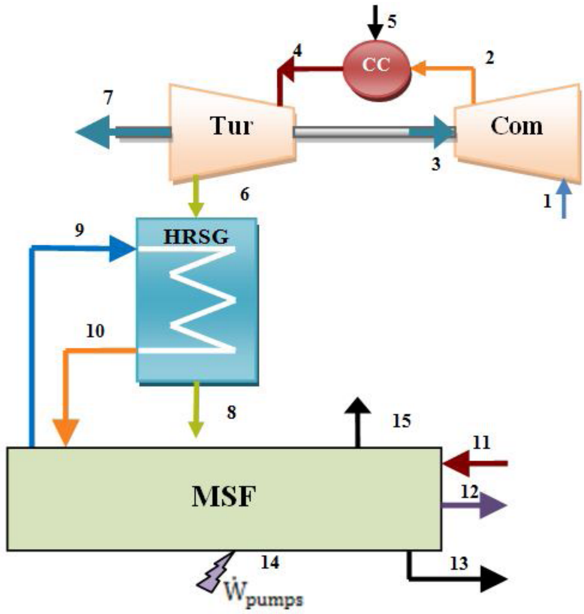

Fig. 1. Combined gas turbine and desalination
(1,2: Air; 3,7,14: Power; 4,6,8: Combustion products; 5: Methane; 9: Water; 10: Steam; 11,15: Sea water; 12: Distillate; 13: Brine)

2. Thermodynamic model

2.1. Power generation cycle

The assumptions for modeling gas turbine cycle are:

- The power cycle works in steady-state condition.
- The combination of inlet air is assumed as follows:

$0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O$

- Fuel of power cycle is methane gas.
- Physical properties of all streams are calculated in mean inlet and outlet temperature.

Making use of the isentropic P-T relation, it is possible to calculate rising of air temperature in compressor outlet:

$$T_{2s} = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k_{com}-1}{k_{com}}} \quad (1)$$

Specific heat capacity of air components is calculated by the following equation in which *a, b, c*, and *d* are constant coefficients for each component [20]:

$$c_P = a + by + cy^{-2} + dy^2 \quad , \; y = 10^{-3}T \quad (2)$$

Real temperature of air in compressor outlet by definition of isentropic efficiency is:

$$T_2 = T_1 + \frac{(T_{2s} - T_1)}{\eta_{com}} \quad (3)$$

Reaction in combustion chamber is assumed as [20]:

$$\bar{\lambda}CH_4 + [0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O] \rightarrow (1+\bar{\lambda})[x_{N_2}N_2 + x_{O_2}O_2 + x_{CO_2}CO_2 + x_{H_2O}H_2O] \quad (4)$$

$\bar{\lambda}$is fuel to air mole fraction and is given by:

$$\bar{\lambda} = \frac{n_F}{n_{air}}$$
$$1+\bar{\lambda} = \frac{n_P}{n_{air}} \quad (5)$$

By balancing of equation coefficients, the mole fraction of each component could be calculated as:

$$x_{N_2} = \frac{0.7748}{1+\bar{\lambda}}$$
$$x_{O_2} = \frac{0.2059 - 2\bar{\lambda}}{1+\bar{\lambda}} \quad (6)$$
$$x_{CO_2} = \frac{0.0003+\bar{\lambda}}{1+\bar{\lambda}}$$
$$x_{H_2O} = \frac{0.019+2\bar{\lambda}}{1+\bar{\lambda}}$$

In a similar way, exit temperature of gas turbine is:

$$\mathrm{T}_{6s} = \frac{\mathrm{T}_4}{\left(\frac{\mathrm{P}_4}{\mathrm{P}_6}\right)^{\left(\frac{k_{Tur}-1}{k_{Tur}}\right)}} \quad (7)$$

Real temperature of air in turbine outlet by definition of isentropic efficiency is:

$$\mathrm{T}_6 = \mathrm{T}_4 - (\mathrm{T}_4 - \mathrm{T}_{6s})\eta_{i,Tur} \quad (8)$$

2.2. Multi stage flash desalination

The brine circulating multi stage flash desalination (MSF-BR) has been schematically illustrated in Fig. 2.

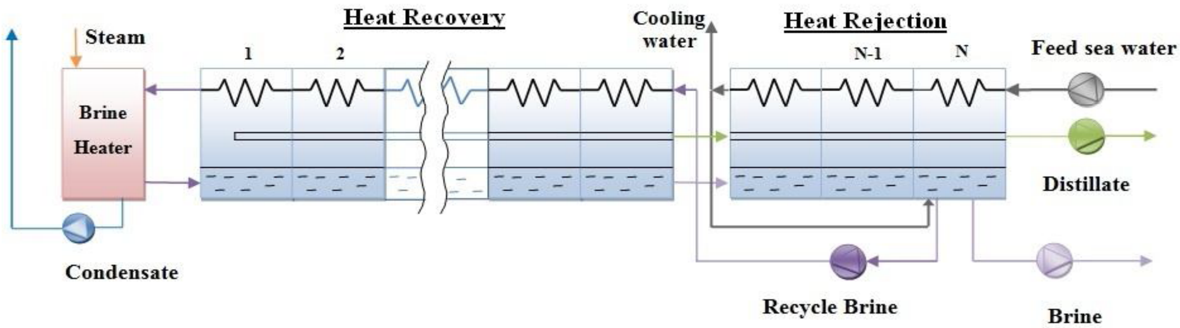

Fig. 2. Brine circulating multi stage flash desalination

As shown in this figure, the process involves heating seawater to a high temperature then passing it through a series of stages where the pressure and temperature are reduced progressively. This process causes the sea water to flash at each successive stage. The released vapor is condensed and collected as distilled water. In the MSF process, the brine recycle is invariably heated by low-pressure steam from an associated power plant.

Analysis of the MSF process can be accomplished by detailed mathematical models. These models solve a large number of equations and generate much larger information on system variables and properties. The following analysis is adopted to calculate the main design features of large scale MSF systems, which include the brine recycle flow rate, the flow rate of cooling seawater, the flow rate of distillate water, steam consumption, etc.

The following assumptions are considered in modeling of the MSF system:

- The desalination system works in steady-state condition.
- Vapor formed in each effect is salt free.
- Thermal loss from desalination to environment is negligible.
- Equal temperature drop per stage for the flashing brine.
- The latent heat of vaporization in each stage is assumed equal to the average value for the process.
- Effects of the non-condensable gases have negligible effect on the heat transfer process. (Air and other gases are non-condensable and its presence in the system may result in severe reduction in the heat transfer rates within the chamber, increase of the tendency for corrosion, and reduction of the flashing rates.)

The governing equations for thermodynamic modeling of the desalination plant are given by [21]. Following is a summary of the model equations:

First, the temperature drop per stage, ΔT, and temperature of stage i, are obtained from the relation:

$$\Delta T = (T_0 - T_n)/n$$
$$T_j = T_0 - j\Delta T \quad , \quad j = 1,2,\dots,n \quad (9)$$

Where T_0, T_n, n are defined as the top brine temperature, last effect temperature, and number of stages, respectively.

Overall water and salt mass balance equations are given by (10) where, *M* is the mass flow rate, *X* is the salt concentration, and the subscripts *b, d, f, r*, and *cw* define the brine, distillate, feed, recovery, and cooling water, respectively:

$$M_f = M_b + M_d$$
$$X_f M_f = X_b M_b \quad (10)$$

The amount of flashing vapor formed in each stage is obtained by conservation of energy within the stage, where the latent consumed by the flashing vapor is set equal to the decrease in the brine sensible heat. This is:

$$D_j = M_r y (1-y)^{(j-1)} \quad (11)$$

y is the specific ratio of sensible heat and latent heat and is equal to:

$$y = c_p \Delta T / \lambda_{av} \quad (12)$$

$\lambda_{av}$is the average latent heat calculated at the average temperature:

$$T_{av} = (T_0 + T_n)/2 \quad (13)$$

The salt concentration in the brine stream leaving stage j is given by:

$$X_j = \left(M_r - \sum_{K=1}^{j} D_k \right) \Big/ B_j \quad (14)$$

A total distillate product is given by:

$$M_d = M_r (1 - (1-y)^n) \quad (15)$$

Equation (15) is used to calculate the brine recycle flow rate. The salt concentration in the recycle stream, X_r, and the cooling water flow rate,M_{cw}, are given by:

$$X_r = \left[(X_f - X_b) M_f + M_r X_b \right] / M_r$$
$$M_{cw} = \left(M_s \lambda_s - M_f C_p (T_n - T_{cw}) \right) / \left(C_p (T_n - T_{cw}) \right) \quad (16)$$

Temperature profile of condenser tubes is obtained from energy balance equation for HR (Heat Recovery) and HJ (Heat Rejection) sections, respectively [21]. The heat transfer area for the condenser in each stage in the HR section is assumed to be the same. The same assumption is made for the condenser heat transfer area in the HJ section. Therefore, the calculated heat transfer area for the first stage is used to obtain the total heat transfer area in the HR section. The condenser heat transfer area in the first stage is obtained from:

$$A_r = M_r C_P (T_{r_1} - T_{r_2}) / (U_r LMTD_r) \quad (17)$$

3. Exergy analysis

3.1. Exergy balance equations in the power plant

Exergy is the maximum theoretical useful work attainable from an energy carrier under the conditions imposed by an environment at given pressure p_0 and temperature T_0, and with given amounts of chemical elements [22]. The purpose of an exergy analysis is generally to identify the location, the source, and the magnitude of true thermodynamic inefficiencies in power plants. Exergy flow equation for each part of the power plant is defined as [22]:

$$\sum \dot{Q} \left(1 - \frac{T_0}{T}\right) - \dot{W} + \sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad (18)$$

where $\dot{E}_D$ is the exergy destruction due to the system irreversibilities. In the absence of nuclear, electrical, and surface tension effects, the magnitude of the specific exergy in every state is determined from the following equation [22]:

$$e = e^{ph} + e^{ch} + e^{kn} + e^{pt} \quad (19)$$

In this study, the two components, which are kinetic exergy and potential exergy, are assumed to be negligible as the elevation and speed have negligible changes. The two important ones are the physical exergy and chemical exergy. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in a combustion process. For a multi component mixture, chemical exergy can be written as [22]:

$$e^{ch} = \sum x_k e_k^{ch} - \bar{R} T_0 \sum x_k ln x_k \quad (20)$$

The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state. The physical exergy of a stream is given by [22]:

$$e^{ph} = (h - h_0) - T_0 (s - s_0) \quad (21)$$

3.2. Exergy balance equations in the MSF

The specific entropy and enthalpy of a component in an ideal solution at a specified temperature T and pressure P is [6]:

$$h = mf_s h_s + mf_w h_w \quad (22)$$

$$s = mf_s s_s + mf_w s_w \quad (23)$$

The seawater inlet for desalination is given to be at 298 K, 1 atm, and a salinity of 0.042%. This condition is assumed to be conditions of the environment. The specific heat, the enthalpy, and entropy of the salt at T_0=298 K taken to be cp_s=0.8368 kJ.kg^{-1}.K^{-1}, h_{s0}=20.92 kJ.kg^{-1}, and S_{s0}=0.0732978 kJ.kg^{-1}.K^{-1}, respectively. Hence, the enthalpy and entropy of the salt at the temperature T can be determined from [6]:

$$h\_s = h\_s0 + 〖cp〗\_s (T - T\_0) = 20.92 + 0.8368 \times (T - 298) \quad (24)$$

$$S\_s = S\_s0 + \llbracket cp \rrbracket\_s\, ln(T/T\_0) = 0.0732978 + 0.8368 \times ln(T/298) \quad (25)$$

For exergy analysis, saline water can be considered to be an ‘‘ideal solution’’ with negligible error, since the effect of dissimilar molecules (molecules of salt and water) on each other is negligible [6].So:

$$\bar{s}_i = \bar{s}(P,T)_{i,pure} - R_u lnx_i \quad (26)$$

The entropy of a saline solution is the sum of the entropies of the salt and water in the saline solutions [6]:

$$\bar{s} = x_s\bar{s}_s + x_w\bar{s}_w = x_s\left[\bar{s}_{s,pure}(T,P) - R_u lnx_s\right] + x_s\left[\bar{s}_{w,pure}(T,P) - R_u lnx_w\right] = x_s\bar{s}_{s,pure}(T,P) + x_w\bar{s}_{w,pure}(T,P) - R_u(x_s lnx_s + x_w lnx_s) \quad (27)$$

With above equation, the exergy of a flow stream is given as:

$$\dot{E} = \dot{m}e = \dot{m}[h - h_0 - T_0(s - s_0)] \quad (28)$$

4. Results

For simulation and exergy analysis of the system, a computational code was developed. It was predicted the exergy destruction of different parts, the production rate of portable water, and required heat transfer area.

After modeling and simulating the combined system, lots of results were obtained. As shown in Fig. 3, the value of exergy destruction of each part is very important to analyze the thermodynamic characteristics of system. As can be seen, the maximum and minimum values of exergy destruction rate are related to combustion chamber and turbine, respectively.

Table 1 shows the reference values for all parts of system. These values are the main functional parameters of co-generation system and have been used for presenting the results.

The exergetic efficiency for a component is given by [22]:

$$\varepsilon = \frac{\dot{E}_P}{\dot{E}_F} = 1 - \frac{\dot{E}_D}{\dot{E}_F} \quad (29)$$

Where $\dot{E}_P$, $\dot{E}_F$ are defines the exergy of product and fuel, respectively.

Table 1. Reference values for power and water production plant.

Desalination plant		Power plant	
Inlet seawater temp.	25 °C	Ambient temp.	25 °C
Rejected brine temp.	40 °C	Net power	65 MW
Top brine temp.	106 °C	Inlet turbine temp.	1100 °C
Salt composition of the outlet brine	70000 ppm	Relative air humidity	60 %
Number of effects	28	Compression ratio	10
Salt composition of the inlet seawater	42000 ppm	Pressure of the steam	600 kpa
Capacity	36977 $m^3.day^{-1}$	Thermal efficiency of power cycle	29.1 %
Performance ratio*	8.3	Inlet HRSG water temp.	25 °C
Total steam consumption	51.6 $kg.s^{-1}$	Outlet HRSG flue gas temp.	160 °C
Specific area	237 $m^2.kg.s^{-1}$	Isentropic efficiency of compressor	86%
Total feed seawater	1809 $kg.s^{-1}$	Isentropic efficiency of turbine	87%
Total cooling seawater	739 $kg.s^{-1}$	Heat loss in combustion chamber	2 %
Total brine outlet	642 $kg.s^{-1}$	Pressure loss in combustion chamber	5 %

* The ratio between the mass of the produced fresh water to that of the consumed steam

So exergy efficiency for power and steam cycle can be expressed as (See Fig. 1):

$$\varepsilon_{power\,\&\,steam\,cycle} = \frac{\dot{W}_{net} + (\dot{E}_{10} - \dot{E}_9)}{\dot{E}_{fuel} + \dot{E}_1} \quad (30)$$

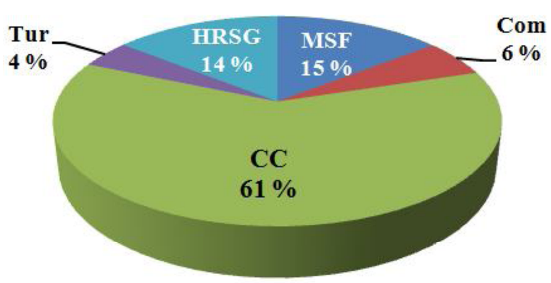

Fig. 3. Exergy destruction of the hybrid plant.

Exergy efficiency of desalination can be written as:

$$\varepsilon_{Desalination} = \frac{\dot{E}_{12} + \dot{E}_{13} + \dot{E}_{15} - \dot{E}_{11}}{\left(\dot{E}_{10} - \dot{E}_{9}\right) + \dot{W}_{pumps}} \qquad (31)$$

The overall exergetic efficiency of the dual-purpose plant is given by:

$$\varepsilon_{tot} = \frac{\dot{E}_{12} + \dot{E}_{13} + \dot{E}_{15} - \dot{E}_{11} + \dot{W}_{net} - \dot{W}_{pumps}}{\dot{E}_{fuel} + \dot{E}_{1}} \qquad (32)$$

Fig. 4 shows the exergy efficiencies of power and steam cycle, desalination and the overall of the dual-purpose plant.

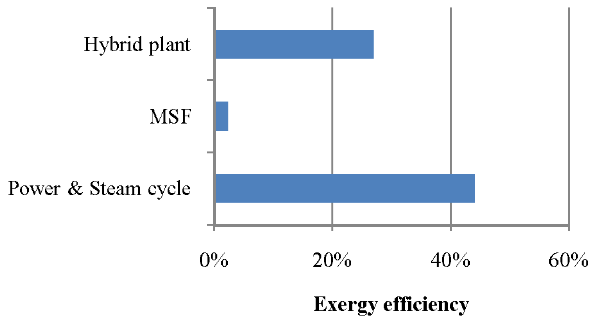

Fig. 4. Exergetic efficiency of different configurations.

As shown in Fig. 4, the magnitude of the exergetic efficiency for a thermal desalination system is quite low so coupling it with the power plant seems useful. This means that setting up a thermal desalination plant alone will be huge thermal energy consumption, but its coupling with a source of heat loss can prevent energy and exergy losses effectively.

5. Sensitivity analysis

The purpose of a sensitivity analysis is to study the impacts of important parameters on the system performance. This analysis which is performed based on changes in a related parameter as well as some other modeling parameters helps prediction of the results.

Fig. 5 shows the influence of effect number on performance ratio (PR), specific heat transfer area (SA), and exergy destruction of desalination (Ex_D). Specific heat transfer area (SA) is defined as the total heat transfer area of condenser tubes per unit mass of distillate product.

For better comparison, parameters were normalized by the reference values mentioned in the Table 1. As shown in figure 5, an increase in the number of stages, will lead to an increase in the water production, so with constant steam consumption the performance ratio increase either. It is clear that specific heat transfer area is increased by raising effect numbers.

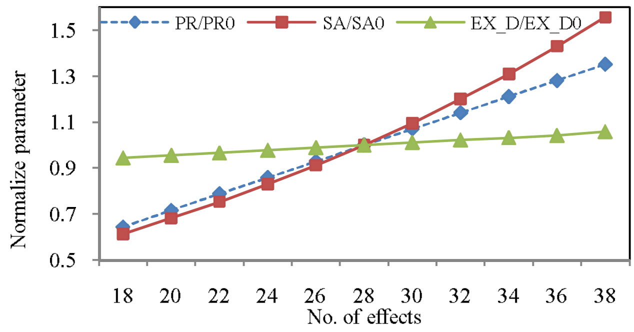

Fig. 5. Impact of number of effects on normalized of the system.

Fig. 6 shows the effect of compression ratio on power (W) and water production (D). It is clear that there is an optimum amount for compression ratio that maximizes power generation. Also, with raising turbine inlet temperature (TIT) the optimum value moves to right side of charts and it occurs in higher values of compression ratio. On the other hand, by increasing compression ratio, fresh water production decreases and probably remains constant. In addition, the figure shows the influence of TIT on power and water production too. As expected, by rising TIT, water production goes up.

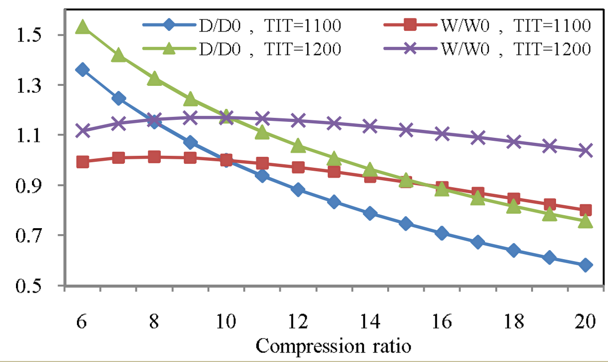

Fig. 6. Influence of compression ratio on power and water production

Fig. 7 shows the effect of HRSG steam pressure on performance ratio (PR), water production rate, and steam production rate (M_s) of HRSG. As the figure shows, by increasing the steam pressure the latent heat of the steam and the heat transfer potential decrease, due to this events amount of water production would reduce too. It is clear that exergy destruction of HRSG is decreased by increasing steam pressure, because of its temperature difference reducing, between the outlet gas turbine flue gas and saturated water.

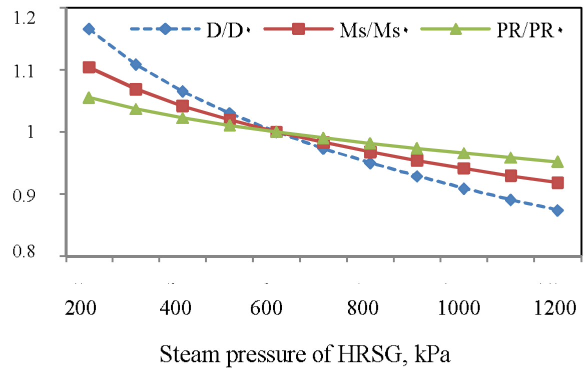

Fig. 7. Impact of HRSG steam pressure on normalized D, M_s and PR.

Fig. 8 shows the effect of top brine temperature on specific heat transfer area (SA) and exergy destruction of desalination. It is clear that by increasing the top brine temperature, the heat transfer area is reduced due to increasing the temperature difference in each effects of MSF. Also, the exergy destruction of desalination reduces, because of increasing the water production (See Eq. (31)).

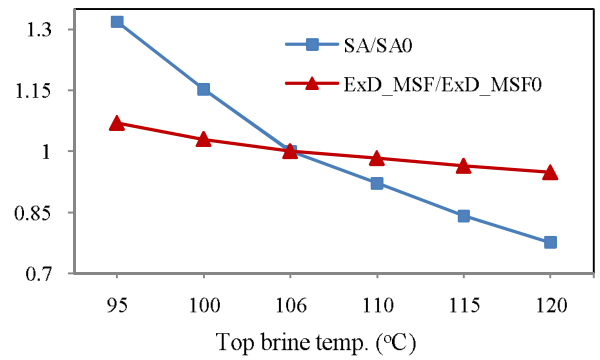

Fig. 8. Influence of top brine temperature on normalized SA, $Ex_{D,MSF}$

The sensitivity of the turbine inlet temperature to changes in power and water production is shown in Fig. 9. As can be expected, by increasing the TIT, the power production goes up due to the basic features of Briton cycle (Increasing gas turbine inlet temperature produce a higher enthalpy drop and, therefore, increase the efficiency and output of the gas turbine plant). It is clear that the water production is increased due to increasing the outlet turbine temperature and consequently, increasing the steam production of HRSG.

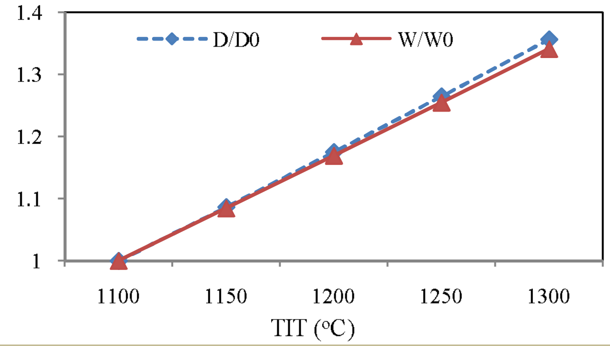

Fig. 9. Influence of TIT on power and water production,

6. Conclusions

In this paper, a combined gas turbine cycle and MSF desalination system were modeled and simulated. Energy and exergy equations for all parts of system were developed. According to the results obtained from modeling as shown in the figures, some specific conclusions are as follow:

- Combustion chamber and MSF desalination (totally 76% exergy destruction) are the two major components of the hybrid plant which can considered for reducing the exergy destructions.
- By increasing stage number of the desalination, the water production increases, so with constant steam consumption the performance ratio increase either.
- By increasing compression ratio, fresh water production decreases and probably remains constant.
- The power and water production goes up by increasing the turbine inlet temperature.
- By increasing the top brine temperature, the heat transfer area reduces because of increasing the temperature difference in each effects of MSF.

Nomenclature

A	Heat transfer area
BR	Brine circulating
CC	Combustion Chamber
Com	Compressor
D	Distillate
$\dot{E}$	Exergy rate
$\dot{E}_D$	Exergy destruction
e	Specific exergy
GT	Gas turbine
HJ	Heat Rejection
HR	Heat Recovery
$HRSG$	Heat recovery steam generator
h	specific enthalpy
j	Stage
MED	Multi Effect Distillation
MSF	Multi Stage Flash
mf	Mass fraction
n	Number of effects
PR	Performance ratio
R_u	Universal gas constant
S	Entropy

s	specific entropy
Tur	Turbine
W	Net power
X	Salinity

Greek Letters

ε	Exergetic efficiency

Superscripts

ch	Chemical
kn	Kinetic
ph	Physical
pt	Potential

Subscripts

0	Environmental state
b	brine
CC	Combustion Chamber
cw	Cooling water
D	Destruction
d	Distillate
F	Fuel
f	Feed water
i	Isentropic
j	Stage
P	Product
r	Recycle stream
s	Salt
tot	Total

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