Desalination and Water Treatment
www.deswater.com

doi: 10.1080/19443994.2012.715132

51 (2013) 1908–1914
February

Taylor & Francis
Taylor & Francis Group

Thermodynamic analysis of a cogeneration gas turbine and desalination plant

A. Al Zahrani, J. Orfi*, H. Al Ansary, B. Salim, Z. Al Suhaibani

Department of Mechanical Engineering, College of Engineering, King Saud University, P.O. Box 800, 11421 Riyadh, Saudi Arabia
Tel. +966 1 467 9798; Fax: +966 1 467 6652; email: orfij@ksu.edu.sa

Received 20 March 2012; Accepted 18 July 2012

ABSTRACT

This work deals with a thermodynamic analysis of the performance of a cogeneration gas turbine and thermal desalination plant. A heat recovery steam generator links the two units together. The analysis is based on the application of mass, energy and exergy balances on the whole cogeneration plant as well as on each of its components. The analysis is also based on an accurate evaluation of the seawater properties, which leads to correct calculations of the flow exergy and exergy efficiency of any component of the desalination plant. The main results of the study concern the influence of the pressure ratio of the compressor and the turbine inlet temperature on the exergy efficiency of the gas turbine and the whole combined system. It was found in particular that as the performance ratio of the desalination plant increases the specific exergy destruction rate decreases.

Keywords: Flow exergy; Seawater properties; Cogeneration; Waste heat; MED desalination

1. Introduction

A desalination process separates the feed saline water that can be brackish water or seawater into product water with low salinity and concentrated brine. Such a separation process requires an energy input that is the function of several parameters such as the separation process itself, the salinity and the temperature of the incoming saline water. The minimization of this required energy is very important since it reduces the cost of producing the fresh water and decreases the generation of greenhouse gases and the disposal of various pollution products into sea or atmospheric air. Therefore, the problem of reducing the energy consumption for desalination technologies is major. Extensive research under this topic has been focused on ways to solve it—such as the use of cheap alternative energy sources or waste heat. Several studies were concerned with the recuperation and the use of waste energy from power plants such as steam or gas turbine plants [1,2]. In fact, thermal desalination plants need, for producing fresh water from saline water, large quantities of low-grade thermal energy of which the temperature is usually lower than 130˚C. Wang and Lior [3] reported as an example that coupling of multi-stage flash desalination plant with steam turbine plant showed a 44.4% energy saving of water production and cost water reduction of about 45%, from 2.66 to 1.47$/m^3.

Vlachos and Kaldellis [4] presented a techno-economic study on the application of gas turbine

*Corresponding author.

Presented at the International Conference on Desalination for the Environment, Clean Water and Energy, European Desalination Society, 23–26 April 2012, Barcelona, Spain

exhaust gases for brackish water desalination in the island of Crete. The existing LM6000 gas turbine has an inbuilt capacity of 43.3 MWe and a mass flow rate of the hot out gases (at 456°C) of 123.7 kg/s. It was proposed to couple such a power plant with a multiple effect distillation accompanied with thermal vapour compression (MED-TVC) giving a rated capacity of 460 m^3/h of potable water and gained output ratio (kg of distilled per kg of steam used) of 8:1.

El Nashar [2] reviewed the technical and economic characteristics of the cogeneration plants currently commercially available as well as the cost parameters for both cogeneration and desalination plants. The same author described a methodology for selecting the optimal arrangement of cogeneration plant for a particular power and desalination capacity. He observed in particular that the power-to-water ratio has a strong influence on the optimum selection of a cogeneration plant. This optimum generation option depends strongly on the load variation throughout the year.

Wang and Lior [3] conducted a theoretical study using energy and exergy analysis of a coupled plant of a steam-injected gas turbine (STIG) and MED-TVC system. The MED-TVC unit which includes a steam jet ejector was compared to the Multi-effect evaporation (MEE) process and was selected for producing fresh water and for humidification when moderate pressure (around 3 bar or higher) steam is available. The authors concluded in particular that the steam injection rate in the STIG cycle has a strong effect on water and power production, which offers good flexibility for design and operation. Drovnikov [5] presented the results of a feasibility study on a gas turbine—multi-effect distiller power and desalination plant. The MED evaporators are of a horizontal falling film type and combustion turbines are with steam injection (STIG).

Several recent studies have been conducted on the analysis and optimization of integrated desalination and low-temperature thermal energy [6–8]. Shakib et al. [8] took into account in their optimization study of a cogeneration plant for water and power production the thermo-economic considerations. The optimal design and configuration have been found using a multi-objective genetic algorithm.

Deng et al. [9] proposed a road map for integration of low-temperature MED and power plants, and analysed the performance of several cogeneration plants under operation in China.

On other sides, analysis of power generation as well as desalination systems show an increasing interest in the combined utilization of the first and second laws of thermodynamics, using concepts such as exergy and exergy destruction in order to evaluate the efficiency with which the available energy is consumed. Exergetic analysis provides the tool for a clear distinction between energy losses to the environment and internal irreversibility in the process [10]. Kanuglu and Dincer [11] added that exergy methods allow the "value" of cogeneration products to be compared on a similar basis, whereas the more conventional energy methods treat heat and electricity the same.

It is of interest to note that the determination of exergy in a desalination plant requires the use of the reliable thermodynamic properties of the working fluid in particular saltwater properties. Sharqawy et al. [12–14] conducted detailed analysis of the properties of seawater using different models and concluded that some of these models lead to incorrect determination of the exergy flow.

This work concerns the thermodynamic analysis of a gas turbine and a desalination plant. The first part of the paper deals with the evaluation of the flow exergy of saltwater while the second part is dedicated to the analysis of performance of the cogeneration plant.

2. Modelling

2.1. Properties of saltwater

The properties of saltwater depend on its pressure, temperature and salinity. The latter can be expressed in ppm (parts per million on a mass basis), as a percentage (sal), as a salt mass fraction ($\mathrm{mf_s}$) or a salt mole fraction (x_s). The $\mathrm{mf_s}$ and x_s are defined as [15]:

$$\mathrm{mf_s} = \frac{m_s}{M_m} = x_s \frac{M_s}{M_m} \quad \text{and}$$
$$\mathrm{mf_w} = \frac{m_w}{M_m} = x_w \tag{1}$$

m_s and m_w are the molar mass of the salt and the pure water, respectively. Their values are 58.5 and 18.0 kg/kmol. M_m is the apparent molar mass of the saline water given by the following equation:

$$M_m = \frac{m_m}{N_m} = \frac{N_s M_s + N_w M_w}{N_m} = x_s M_s + x_w M_w \tag{2}$$

Therefore, the relationship between salt mass fraction and salt mole fraction can be given as Cerci [15]:

$$x_s = \frac{\mathrm{mf_s} M_w}{M_s(1 - \mathrm{mf_s}) + \mathrm{mf_s} M_w} \tag{3}$$

where: $x_s + x_w = 1$ and $\mathrm{mf_s} + \mathrm{mf_w} = 1$

2.2. Saltwater flow exergy

Exergy is defined as the maximum amount of work which can be produced by a system as it comes to equilibrium with a reference environment [16]. The dead state which refers to the environmental state must be specified in any exergy calculation.

The total exergy of a system can be divided into physical exergy, kinetic exergy, potential exergy and chemical exergy. The first three forms of exergy are called thermo-mechanical exergy. The chemical exergy refers to the maximum amount of work obtained at the dead state temperature and pressure (T_0, P_0) as the system comes to chemical equilibrium with the environment. This type of exergy is zero for a single-component system and must be included for a multi-component system.

The flow exergy of a system, $\mathrm{ex_f}$, can be expressed as follows [12]:

$$\mathrm{ex_f} = (h - h^*) - T_0(s - s^*) + \sum w_i(\mu_i^* - \mu_i^0) \tag{4}$$

where h and s are the specific enthalpy and specific entropy. The μ and w refer to the chemical potential and mass fraction, respectively.

Eq. (4) becomes for a saltwater mixture:

$$\mathrm{ex_f} = (h - h^*) - T_0(s - s^*) + w_\mathrm{s}(\mu_\mathrm{s}^* - \mu_\mathrm{s}^0) + w_\mathrm{w}(\mu_\mathrm{w}^* - \mu_\mathrm{w}^0) \tag{5}$$

The subscripts s and w stand for salt and water.

The dead state can be restricted state or global dead state. The latter refers to the case where the temperature, the pressure and the concentration correspond to the dead state (T_0, P_0 and w_0). The restricted dead state refers to the case where only the temperature and the pressure are changed to the environment values (T_0 and P_0) but at the same concentration of the initial flow stream state [14]. As reported by Sharqawy et al. [12], the chemical exergy must be taken into account when evaluating the total exergy of a multi-component system since neglecting this kind of exergy may lead to incorrect results.

Three models for seawater thermodynamic properties are used in the literature. These models refer to the seawater, sodium chloride aqueous solution and the ideal solution of sodium chloride and pure water. The latter approach is based on the fact that saltwater of less than 5% salinity is considered to be a dilute solution and can be treated as an ideal solution [15]. Such an ideal solution is a solution in which the effect of dissimilar molecules on each other is negligible. Extensive properties of a mixture are the sum of extensive properties of its individual components.

A detailed and critical analysis of the above three models has been presented by Sharqawy et al. [14]. The main conclusions of this study are:

- The thermodynamic properties of sodium chloride and seawater are different while flow exergy is very similar.
- The ideal mixture model gives incorrect values of flow exergy of seawater since the chemical exergy component of the flow exergy is omitted.
- An update and accurate correlations for the thermodynamic seawater properties have been proposed [13].

In present work, we evaluated the properties of seawater using the ideal solution approximation for the enthalpy and entropy and thus for the thermo-mechanical exergy, while for the chemical exergy terms, we used the expressions proposed by Sharqawy et al. [12,13]. This approach is found to be accurate after several comparisons with the seawater properties as given in Sharqawy et al. [12] and in El-Dessouky and Ettouney [17].

Table 1 compares the flow exergy as calculated in this work by Sharqawy et al. [12] using seawater properties model as well as by Kahrama and Cengel [18] using the ideal solution model. The environmental reference state is selected at $T_0 = 35\,^\circ\mathrm{C}$, $P_0 = 101.325\,\mathrm{kPa}$ and the salinity $w_{\mathrm{s},0} = 46.5\,\mathrm{g/kg}$.

One can see that for the conditions of pressure, temperature and salinity, the present flow exergy values are very close to those of Sharqawy et al. [12]. However, the ideal mixture model of pure water and sodium chloride as used by Kahraman and Cengel [18] leads to different trends in which the flow exergy can have negative values. Sharqawy et al. [14] proved that such a finding is correct only when the pressure goes below the reference pressure P_0, which is not the case here.

The above properties and relations are used to analyse the performance of a gas turbine combined with a thermal desalination unit using a heat recovery steam generator.

2.3. Thermodynamic analysis of a cogeneration gas turbine–MEE plant

The cogeneration plant shown schematically in Fig. 1 is composed of three subsystems. The first subsystem refers to the gas turbine (GT) power plant. The second subsystem is the thermal desalination

Table 1
Thermodynamic properties of saline water at various conditions of pressure, temperature and salinity

#	P (kPa)	T (°C)	sal (%)	ex_f (kJ/kg)[a]	ex_f (kJ/kg)[b]	ex_f (kJ/kg)[c]
0	101.3	35	4.65	0	0	0
1	168	35	4.65	0.065	0.064	0.064
2	115	43.3	4.65	0.442	0.493	0.4556
3	115	43.3	4.65	0.442	0.493	0.4556
4	9	43.3	4.65	0.339	0.392	0.354
5	9	41.4	0	4.152	10.489	4.153
6	9	43.3	7.01	0.752	−3.502	0.768
7	9	43.3	6.48	0.601	−2.672	0.6166
8	635	43.3	6.48	1.204	−2.082	1.205
9	635	85	6.48	15.07	−12.123	15.36
10	635	90.8	6.48	18.35	15.495	18.76
11	97.4	98.9	0	411.2	388.177	416.5
12	97.4	98.9	0	411.2	388.177	416.5
13	578	41.5	0	4.735	11.062	4.734
14	292	43.3	7.01	1.023	−3.238	1.033
15	101.3	35	7.01	0.42	−3.887	0.4203
16	101.3	35	0	3.972	10.268	3.972
17	101.3	35	4.65	0	0	0

[a]Sharqawy et al. [12].
[b]Kahraman and Cengel [18].
[c]This work.

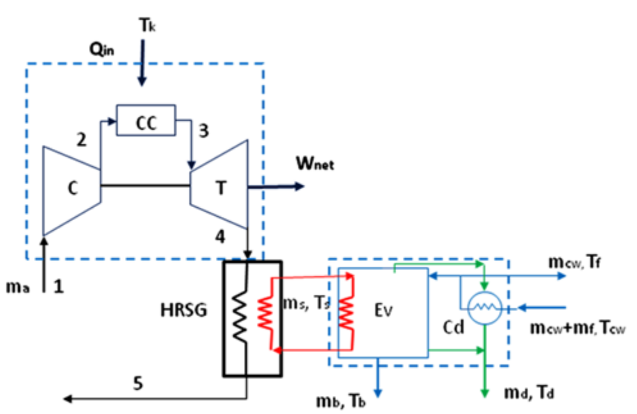

Fig. 1. Schematic diagram of the combined power and desalination plant (C: compressor, CC: combustion chamber, T: turbine, HRSG: heat recovery steam generator, Ev: evaporators and Cd: condenser).

plant (MED). These two units are connected together using a heat recovery steam generator (HRSG).

The MED unit consists of a condenser and an evaporator with several effects. The cooling and feed seawater enters the desalination unit at the temperature T_{cw} and the salinity X_{cw}. The cooling seawater is rejected at a higher temperature, T_f. The distillate with a mass flow rate m_d is collected after the evaporation and condensation process inside the plant.

The modelling is based on the application of the mass, energy and exergy balances for each component of the system as well as for the whole system.

The exergy balances for any control volume at steady state with negligible potential and kinetic variations can be expressed as follows [10]:

$$\sum\left(1-\frac{T_0}{T_k}\right)Q_k - W + \sum m_{in}ex_{in} - \sum m_{out}ex_{out} - Ex_{dest} = 0 \quad (6)$$

where T_0 refers to the reference dead state 0. The subscripts in, out, 0 and k refer to the inlet, outlet, dead and heat source states, respectively.

Different ways of formulating the exergy efficiency are proposed in the literature [16]. The exergy efficiency relates all exergy inputs as used exergy and all exergy outputs as utilized exergy.

2.3.1. Gas turbine plant

The gas turbine operates on an actual Brayton cycle with a pressure ratio r_p, inlet gas temperature, T_1 and a turbine inlet temperature T_3. The turbine and the compressor have respectively η_t and η_c as isentropic efficiencies. The working fluid is air with constant properties.

The expressions for the specific power produced W_{net}, the specific heat added, q_{in} and the specific heat of the heat recovery steam generator are obtained using the energy balance equation.

$$W_{net} = c_p \left\{ T_3 \eta_t (1 - r_p^{-A}) + \frac{T_1(1 - r_p^A)}{\eta_c} \right\} \tag{7}$$

$$q_{in} = c_p \left\{ T_3 - T_1 \left(1 + \frac{1}{\eta_c} (r_p^A - 1) \right) \right\} \tag{8}$$

$$q_{HRSG} = c_p (T_4 - T_5) \tag{9}$$

where

$$T_4 = T_3 (1 + \eta_t (r_p^{-A} - 1)) \quad \text{and}$$
$$A = (k - 1)/k \tag{10}$$

k is the ratio of the specific heats. T_5 is the gas temperature leaving the steam generator to the stack. It is assumed to be constant at 150°C.

The thermal efficiency, the utilization factor of the co-generation plant as well as the exergetic efficiency can be expressed as follows [19]:

$$\eta_{th} = \frac{W_{net}}{q_{in}}$$

$$\varepsilon_{co} = \frac{W_{net} + q_{HRSG}}{q_{in}}$$

$$\eta_{exGT} = \frac{W_{net}}{(ex_1 - ex_4) + \left(1 - \frac{T_0}{T_k}\right) q_{in}} \tag{11}$$

where

$$ex_1 - ex_4 = c_p \left(T_1 - T_4 - T_0 \ln\left(\frac{T_1}{T_4} \right) \right) \tag{12}$$

2.3.2. Thermal desalination unit

The application of the mass balances for water and salt and the first law and second law equations gives the main equations for the desalination unit:

$$m_f = m_d + m_b \tag{13a}$$

$$m_f X_f = m_b X_b \tag{13b}$$

$$m_a q_{HRSG} = m_s h_{fg_s} \tag{14}$$

$$m_s h_{fg_s} = m_d h_d + m_b h_b + m_{cw} h_f - (m_{cw} + m_f) h_{cw} \tag{15}$$

h_{fg} is the latent heat of steam.

Where the subscripts a, s, d, b, f and cw refer to air, steam, distillate, brine, feed and cooling water, respectively.

The performance ratio, the specific cooling water mass flow rate and the exergy efficiency are important performance parameters here:

$$PR = \frac{m_d}{m_s} \tag{16}$$

$$sm_{cw} = \frac{m_{cw}}{m_d} \tag{17}$$

$$\eta_{ex_i} = \frac{Ex_{out_i}}{Ex_{in_i}} = 1 - \frac{Ex_{dest_i}}{Ex_{in i}} \tag{18}$$

where i refers to each component of the whole system.

3. Results and discussion

In all the performed simulations, the following variables were taken constant:

- the air mass flow rate, $m_a = 300$ kg/s;
- the adiabatic flame temperature which is the heat source temperature for the gas turbine, $T_H = 2{,}507$°C;
- the air reference state is at 1 atm and 25°C;
- the gas leaving to the stack is fixed to 150°C; and
- the isentropic efficiencies for the turbine and the compressor are set equal to 0.85 and 0.8, respectively.

Regarding the desalination part, the temperature of the intake seawater is at 25°C, 101.325 kPa and 1.55 g/kg.

The steam temperature ranges from 82 to 100 °C. The feed and the brine salinities are 42 and 70 g/kg.

For the power system, Fig. 2 presents the thermal efficiency and the utilization factor when the cogeneration option is considered. It shows that the useful energy converted into work is lesser than 30% of the input energy, while more than 80% of that energy can be used to produce power and water. Therefore,

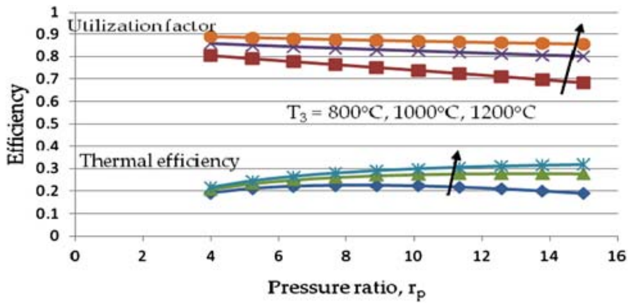

Fig. 2. Dependence of the thermal efficiency and the utilization factor on the pressure ratio for different turbine inlet temperatures.

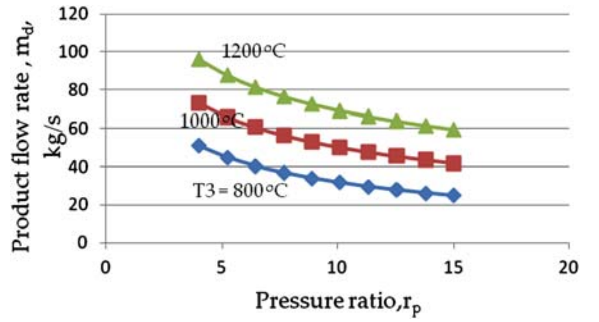

Fig. 4. Dependence of the production rate of fresh water to the pressure ratio and the turbine inlet temperature.

recovering the waste heat leaving the gas turbine leads to a better utilization of the available added energy. Fig. 3 shows the variation of the exergy efficiency of the gas turbine vs. the ambient temperature at two values of the pressure ratio, r_p and three values of the turbine inlet temperature, T_3. As T_3 increases, the exergy efficiency increases in particular at low ambient temperatures. Therefore, air carries more work potential at higher T_3 and higher r_p. The results of this figure were also presented by [19] and good agreement between the present results and those of [19] is found.

Figs. 4 shows the effect of the pressure ratio and the turbine inlet temperature on the production rate of fresh water for the simple case of single-effect evaporation unit. This behaviour is dictated by the amount of energy flowing through the heat recovery steam generator.

The exergy efficiency of the overall system is shown in Fig. 5. The desalination unit corresponds here to a single-effect evaporator. One can see that the system has better performance (lesser irreversibility losses) when the turbine inlet turbine, T_3 is higher.

The behaviour of the exergy efficiency for lower T_3 ($T_3 = 800$°C) shows an optimum when the pressure

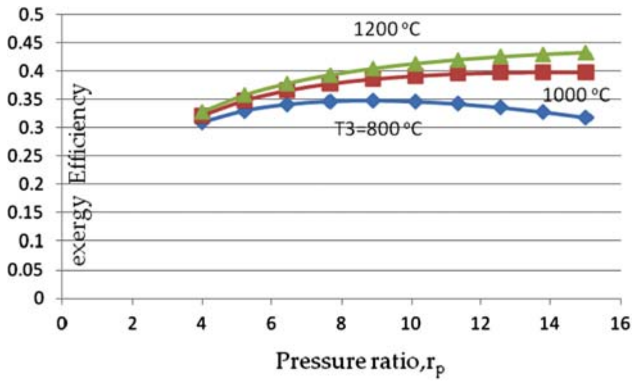

Fig. 5. Variation of the exergy efficiency of the whole cogeneration system with the pressure ratio and the turbine inlet temperature.

ratio is around 10. This behaviour is not present for high values of T_3 where the exergy efficiency increases as r_p increases from 5 to 15.

Fig. 6 depicts the variation of the exergy destruction rate per unit mass of air with the performance ratio, PR. The latter parameter defined as the amount of product fresh water per unit mass of heating steam, is a measure of the performance of the desalination system. For the cogeneration system considered here

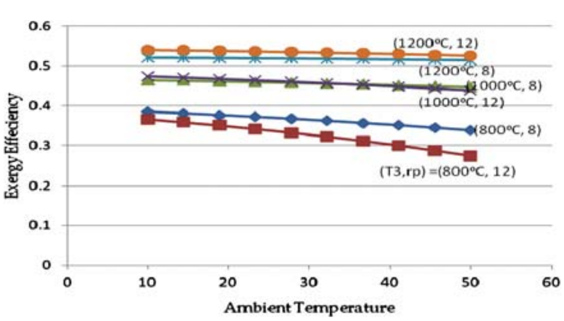

Fig. 3. Variation of the exergy efficiency of the gas turbine unit with the pressure ratio and the turbine inlet temperature.

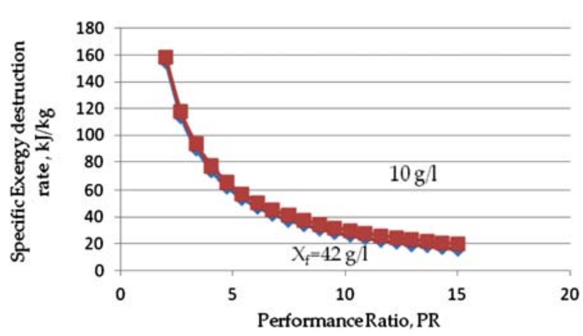

Fig. 6. Dependence of the specific exergy destruction rate on the performance ratio, PR.

and for a fixed steam temperature, PR is a function of the air mass flow rate and the temperature at the turbine exit, T_4. PR increases as m_a and/or T_4 decreases.

For a single-effect evaporator unit, the performance ratio is very close to 1 [17]. The desalination process is therefore highly irreversible in this case.

Fig. 6 shows that for higher values of PR, the exergy destruction rates are lower implying lesser irreversibility losses in the whole components of the system. The effect of the feed salinity is found to be negligible.

4. Conclusion

Several gas turbine plants located in inland regions in several countries including KSA operate with an overall efficiency lesser than 30%. The rejected heat is huge since the exhausted gas mass flow rate can be more than 500 kg/s at a temperature reaching 500°C or more. Combining this power plant with a desalination unit to produce fresh water from brackish water can be an attractive solution for a better utilization of the available added energy.

This work presents some theoretical results on such opportunities and focuses on the evaluation of the saltwater properties, in particular the flow exergy. The main conclusions of this study are:

- An accurate calculation of the saltwater properties mainly the flow exergy is necessary in order to have reliable results on the performance of a desalination plant.
- The pressure ratio of the compressor as well as the turbine inlet temperature is important parameters that have significant influence on the exergy efficieny of the gas turbine and the desalination unit.
- The specific exergy destruction rate decreases as the performance ratio of the desalination plant increases.

Further, detailed results will be published soon on the modelling and simulation of a gas turbine coupled to a MED-TVC desalination plant.

Acknowledgement

The authors would like to thank the National Plan for Science & Technology (NPST) programme and King Saud University for funding the research reported in this paper through the project number (08-ENE340-2).

References

[1] H. Aybar, Desalination system using waste heat of power plant, Desalination 166 (2004) 167–170.
[2] A.M. El Nashar, Cogeneration for power and desalination – state of the art review, Desalination 134 (2001) 7–28.
[3] Y. Wang, N. Lior, Performance analysis of combined humidified gas turbine power generation and multi-effect thermal vapor compression desalination systems—Part 1: The desalination unit and its combination with a steam-injected gas turbine power system, Desalination 196 (2006) 84–104.
[4] G.T. Vlachos, J.K. Kaldellis, Application of gas turbine exhaust gas for brackish water desalination: Techno-Economic evaluation, Appl. Therm. Eng. 24 (2004) 2487–2500.
[5] V. Drovnikov, Seawater multi-effect distillation energized by a combustion turbine, Desalination 127 (2000) 261–269.
[6] R. Chacartegui, D. Sánchez, N. di Gregorio, F.J. Jiménez-Espadafor, A. Muñoz, T. Sánchez, Feasibility analysis of a MED desalination plant in a combined cycle based cogeneration facility, Appl. Therm. Eng. 29 (2009) 412–417.
[7] S.E. Shakib, M. Amidpour, C. Aghanajafi, Simulation and optimization of multi effect desalination coupled to a gas turbine plant with HRSG consideration, Desalination 285 (2012) 366–376.
[8] S.E. Shakib, S.R. Hosseini, M. Amidpour, C. Aghanajafi, Multi-objective optimization of a cogeneration plant for supplying given amount of power and fresh water, Desalination 286 (2012) 225–234.
[9] R. Deng, L. Xie, H. Lin, J. Liu, W. Han, Integration of thermal energy and seawater desalination, Energy 35 (2010) 4368–4374.
[10] Y.A. Cengel, M.A. Boles, Thermodynamics, an Engineering Approach, Sixth ed., McGraw Hill, New York, NY, 2007.
[11] M. Kanuglu, I. Dincer, Performance assessment of cogeneration plants, Energy Convers. Manage. 50 (2009) 76–81.
[12] M.H. Sharqawy, J.H. Lienhard V, S.M. Zubair, On exergy calculations of seawater with applications in desalination systems, Int. J. Therm. Sci. 50 (2011) 187–196.
[13] M.H. Sharqawy, J.H. Lienhard V, S.M. Zubair, Thermophysical properties of seawater: A review of existing correlations and data, Desalin. Water Treat. 16 (2010) 354–380.
[14] M.H. Sharqawy, J.H. Lienhard V, S.M. Zubair, Formulation of seawater flow exergy using accurate thermodynamic data, In: IMECE-2010, British Columbia, Vancouver, 2010.
[15] Y. Cerci, Exergy analysis of a reverse osmosis desalination plant in California, Desalination 142 (2002) 257–266.
[16] Z. Utlu, A. Hepbasli, A review and assessment of the energy utilization efficiency in the Turkish industrial sector using energy and exergy analysis method, Renewable Sustainable Energy Rev. 11 (2007) 1438–1459.
[17] H.T. El-Dessouky, H.M. Ettouney, Fundamentals of Salt Water Desalination, Elsevier, Amsterdam, 2002, p. 691.
[18] N. Kahraman, Y.A. Cengel, Exergy analysis of a MSF distillation plant, Energy Convers. Manage. 46 (2005) 2625–2636.
[19] Y. Haseli, I. Dincer, G.F. Naterer, Unified approach to exergy calculation, environmental impact and sustainable development for standard thermodynamic cycles, Int. J. Green Energy 5 (2008) 105–119.