Open Access Journal

Journal of Power Technologies 92 (3) (2012) 183–191

journal homepage:papers.itc.pw.edu.pl

Thermodynamic modeling and second law based performance analysis of a gas turbine power plant (exergy and exergoeconomic analysis)

Mohammad Ameri*, Nooshin Enadi

Mechanical & Energy Engineering Department, Power and Water University of Technology (PWUT), 16765-1719, Tehran, Iran

Abstract

In this research paper, a complete thermodynamic modeling of one of the gas turbine power plants in Iran is performed based on thermodynamic relations. Moreover, a complete computer code is developed for simulation purposes using Matlab software. To assess system performance, exergy and exergo-economic analyses are conducted to determine the exergy destruction of each component and cost of each flow line of the system. A complete parametric study is also carried out to study the effect of certain design parameters such as exergy efficiency and total cost of exergy destruction on system performance variation. The exergy analysis results revealed that the combustion chamber (CC) is the most exergy destructive component compared to other cycle components. Also, its exergy efficiency is less than other components, which is due to the high temperature difference between working fluid and burner temperature. In addition, it was found that by increasing the TIT (gas turbine inlet temperature), the exergy destruction of this component can be reduced. On the other hand, the cost of exergy destruction, which is a direct function of exergy destruction, is high for the combustion chamber. The effects of design parameters on exergy efficiency showed that an increase in the air compressor pressure ratio and TIT increases the total exergy efficiency of the cycle. Furthermore, the results revealed that an increase in the TIT of about 350 K can lead to a reduction of about 22% in the cost of exergy destruction. Therefore, TIT is the best option to improve cycle losses.

Keywords: Gas turbine power plant, Exergy analysis, Efficiency, Exergy destruction, Economic analysis

1. Introduction

Energy systems involve a large number and various types of interactions with the world outside their physical boundaries. Therefore, the designer must face many issues, primarily related to energy, economy and the environment, in short “3E”. Gas turbines are a good candidate for power generation units because they are widely used in both gas cycles and combined cycles. Hence, thermodynamic modeling and performance assessment of gas turbines form a significant subject of interest for thermal system designers. Combined cycle power plants (CCPP) utilize the exhaust heat from the gas turbine engine to increase power plant output and boost overall efficiency up to 50%. Recently, exergy analysis, which is based on the second law of thermodynamics, has been found to be a potential tool for enhancing the understanding of system performance by determining the amount of irreversibilities for each component and providing better insight into system design.

*Corresponding author

Email addresses: ameri_m@yahoo.com (Mohammad Ameri*), nooshinenadi@gmail.com (Nooshin Enadi)

The exergy analysis approach is based on the simultaneous application of the first and the second laws of thermodynamics [1].The energy crisis of the 1970s and the continuing emphasis on efficiency (conservation of fuel resources) have led to a complete overhaul of the way in which power systems are analyzed and improved thermodynamically [2].

Today, many electrical generation utilities are striving to improve efficiency and the heat rate at their existing thermal electric generating stations, many of which are over 25 years old. Often, a heat rate improvement of only a few percent appears desirable, as it is thought that the costs and complexity of such measures may be more manageable than more expensive options. Thus, a better understanding is attained when a more complete thermodynamic view is taken, which uses the second law of thermodynamics in conjunction with energy analysis, via exergy methods. One of the most commonly-used methods for evaluating the efficiency of an energy-conversion process is first-law analysis although it cannot determine the location of devices in which exergy destruction would occur.

It is well-known that exergy can be used to determine the location, type and true magnitude of exergy loss (or destruction). Thus, it can play an important role in developing strategies and in providing guidelines for more effective use of energy in the existing power plants [3]. Moreover, another important issue to improve the existing system is the origin of the exergy loss and components in which the most exergy destruction take place. Hence, a clear picture, instead of only the magnitude of exergy loss in each section, is required. There are numerous research papers in the literature, which have presented exergy and exergo-economic analysis. However, they do not usually pay much attention to the effect of key parameters on the cycle components, especially the cost of exergy destruction.

According to literature, exergy analysis is a methodology for the evaluation of the performance of devices and processes, and involves examining the exergy at different points in a series of energy-conversion steps [2–5]. Exergy analysis results can aid efforts to improve the efficiency, and possibly the economic and environmental performance of gas turbine power plants. In parallel to exergy analysis, thermo-economics can also help the designers to enhance the understanding of the system performance by consideration of the system costs. Thermo-economics combines exergy analysis with economic principles and incorporates the associated costs of the thermodynamic inefficiencies in the total product cost of an energy system. These costs may lead designers to understand the cost formation process in an energy system and that can be utilized to optimize thermodynamic systems, in which the task is usually focused on minimizing the unit cost of the system product [5]. Several researchers carried out exergy analysis of and applied exergo-economics to systems in which a gas turbine played a significant role. Sahin and Ali [6] carried out an optimal performance analysis of a combined Carnot cycle (two single Carnot cycles in cascade), including internal irreversibilities for steady-state operation. Ameri et al. [3] performed an exergy analysis of supplementary firing in a heat recovery steam generator in a combined cycle power plant. Their results revealed that if a duct burner is added to the heat recovery steam generator (HRSG), the first and second law efficiencies are reduced. Also, Ameri et al. [2] performed an energy, exergy and exergo-economic analysis for one of the largest steam power plants in Iran. It was determined that the boiler has the highest exergy destruction rate. Therefore, this device should be considered for further improvements. The reason for the greatest exergy destruction in this device is due to the combustion and heat transfer processes, which take place across large temperature differences between burner temperature and working fluid. The same results were obtained in other research performed by Ameri et al. [4]. It was found that in combined cycle plants, the combustion chamber destroys the inflow exergy due to the high temperature difference. However, that paper did not pay much attention to the key parameters. Ahmadi et al. [7] performed thermodynamic and exergo-environmental analyses, and multi-objective optimization of a gas turbine power plant.They applied the multi-objective based optimization to an actual power plant in Iran and determined the optimal design parameters. The results showed that by selecting the optimized parameters, a 50% reduction in environmental impacts is obtained. Ehyaei et al. [8] carried out exergy, economic and environmental analyses of absorption chiller inlet air cooler used in gas turbine power plants. They conducted the analyses for two different regions in Iran, (i.e. hot-dry and hot-humid climate conditions). The results showed that using this system in the hot months of a year is economical. They also concluded that application of an absorption chiller increases the output power by 11.5% for the hot-dry climate and 10.3% for the hot-humid climate. The present study, which is an extended version of earlier research carried out by the authors [1, 3] mainly focuses on the followings items which are the specific contribution of the current paper in this subject:

- Complete thermodynamic modeling of a major gas turbine power plant in Iran is performed.
- A simulation computer code is developed using Matlab software to mode all parts of the power plant and this code is validated with actual data from the power plant.
- Exergy analysis of a gas turbine power plant is performed.
- Exergo-economic analysis of a gas turbine power plant is conducted.
- The effects of some major key parameters on both exergy and the exergo-economic performance of the cycle are investigated.

2. Exergy analysis

Exergy is composed of two important parts. The first one is the physical exergy and the second one is the chemical exergy. In this study, the kinetic and potential parts of exergy are negligible [4]. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacting with an equilibrium state. 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 the combustion process. It is important to observe that, unlike energy, exergy is exempt from the law of conservation [5]. Irreversibility associated with actual processes causes exergy destruction.

In order to perform the exergy analysis, mass and energy balances of the system are required to be determined. If one combines the first and second laws of thermodynamics, the exergy balance equation is formed as [4]:

Continuity equation:

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{1}$$

Energy equation:

$$\dot{Q} - \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \tag{2}$$

Exergy balance equation:

$$\dot{E}x_Q + \sum \dot{m}_i e_i = \sum \dot{m}_e e_e + \dot{E}x_D + \dot{E}x_W \tag{3}$$

where subscripts i and e refer to streams entering and leaving the control region, respectively. The exergy rate of a stream of substance (neglecting the potential and kinetic components) can be written in the form:

$$\dot{E}x = \dot{E}x_{ph} + \dot{E}x_{ch} \tag{4}$$

where:

$$\dot{E}x = \dot{m}\, e \tag{5}$$

The mixture's chemical exergy is defined as follows [6]:

$$ex^{ch}_{mix} = \left[\sum_{i=1}^{n} X_i\, ex^{ch} + RT_0 \sum_{i=1}^{n} X_i \ln(X_i) + G^E \right] \tag{6}$$

The last term, G^E, which is the excess free Gibbs energy is negligible at low pressure in a gas mixture. One can generalize the chemical exergy concept of fuel to every $C_\alpha H_\beta N_\gamma O_\varepsilon$ component [9]. The molar chemical exergy ex^{ch}_c of such a component will be:

$$ex^{ch}_c = (\mu_{c,o} - \mu^{\epsilon}_c) \tag{7}$$

Where μ^{ϵ}_c refers to the chemical potential of the component at the restricted dead state.

$$\mu^{\epsilon}_c = \alpha \bar{\mu}^{\epsilon}_{CO_2} + \frac{\beta}{2} \bar{\mu}^{\epsilon}_{H_2O} + \frac{\gamma}{2} \bar{\mu}^{\epsilon}_{N_2} + \left(-\alpha - \frac{\beta}{4} + \frac{\delta}{2} \right) \bar{\mu}^{\epsilon}_{O_2} \tag{8}$$

$\bar{\mu}^{\epsilon}_{CO_2}$ represents the chemical potential of the components at their thermo-mechanical equilibrium state with the standard ambient.

For an evaluation of fuel exergy, the above formula cannot be used. Thus, the corresponding ratio of simplified exergy is defined as the following [10]:

$$\xi = \frac{ex_f}{LHV_f} \tag{9}$$

Due to the fact that for most of the usual gaseous fuels, the ratio of chemical exergy to the Lower Heating Value is usually close to 1, one may write [4]:

$$\begin{aligned} \xi_{CH_4} &= 1.06 \\ \xi_{H_2} &= 0.985 \end{aligned} \tag{10}$$

For gaseous fuel with C_xH_y, the following experimental equation is used to calculate ξ [4]:

$$\xi = 1.033 + 0.0169\frac{y}{x} - \frac{0.0698}{x} \tag{11}$$

In this formula (Eq. 3), (e) is the total specific exergy and $\dot{E}x_D$ is the exergy destruction.

$$\dot{E}x_Q = \left(1 - \frac{T_0}{T_b}\right)\dot{Q}_i \tag{12}$$

$$\dot{E}x_W = \dot{W} \tag{13}$$

$$ex_{ph} = (h - h_0) - T_0\,(S - S_0) \tag{14}$$

Where T is the absolute temperature (K) and subscripts (i) and (o) refer to inlet and ambient conditions respectively. T_b is the boundary temperature in which heat transfer occurs.

For the exergy analysis of power plants, the exergy of each stream should be estimated for all states and the changes in exergy are determined for each major component. Unlike energy, exergy is not conserved but destroyed in the system. In the components of the power plant, exergy is dissipated during the process due to friction, mixing, combustion, heat transfer, etc. The source of exergy destruction (or irreversibility) in the combustion chamber and turbine is mainly combustion (chemical reaction) and thermal losses in the flow path respectively [11]. The objective of the present study is to perform an exergy and exergo-economic analysis and a simulation of a gas turbine power plant, which is a common cycle for producing power in Iran. Thus, for this reason after simulation and thermodynamic modeling of this cycle, the exergy balance for each component is calculated to find the exergy destruction in each component.

3. Economic analysis

Exergo-economics or thermo-economics is the branch of engineering that appropriately combines, at the level of system components, thermodynamic evaluations based on an exergy analysis with economic principles, in order to provide the designer or operator of a system with information that is useful for the design and operation of a cost-effective system, but which are not obtainable by regular energy or exergy analysis and economic analysis [12]. When exergy costing is not applied, researchers should use a different term (e.g. thermo-economics). Thermo-economics is a more general term and characterizes any combination of thermodynamic analysis with economic analysis [13, 14]. In order to define a cost function, which depends on optimization of parameters of interest, the component cost should be expressed as functions of thermodynamic design parameters [14].

For each flow line in the system, a parameter called the flow cost rate C ($/s) is defined, and the cost balance equation of each component is written as:

$$\sum_e \dot{C}_{e,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k \tag{15}$$

The cost balance equation of each component is written as:

$$\begin{aligned} \sum \left(c_e\dot{E}x_e\right)_k + c_{w,k}\dot{W} &= c_{q,k}\dot{E}x_{q,k} \\ &\quad + \sum \left(c_i\dot{E}x_i\right)_k + \dot{Z}_k \end{aligned} \tag{16}$$

$$\dot{C}_j = c_j\dot{E}x_j \tag{17}$$

In this analysis, it is worth mentioning that the fuel and product exergy should be defined. The exergy product is defined according to the components under consideration. The fuel represents the source that is consumed in generating the product. Both the product and fuel are expressed in terms of exergy. The cost rates associated with the fuel ($\dot{C}_F$) and product ($\dot{C}_p$) of a component are obtained by replacing the exergy rates ($\dot{E}x$). For example, in a turbine, fuel is the difference between input and output exergy and product is the generated output power of the turbine.

In the cost balance formulation (Eq. 15), there is no cost term directly associated with the exergy destruction of each component. Accordingly, the cost associated with the exergy destruction in a component or process is a hidden cost. Thus, if one combines the exergy balance and exergo-economic balance together, one can obtain the following equations:

$$\dot{E}x_{F,K} = \dot{E}x_{P,K} + \dot{E}x_{D,K} \tag{18}$$

Accordingly, the expression for the cost of exergy destruction is defined as follows:

$$\dot{C}_{D,k} = c_{F,k}\dot{E}x_{D,k} \tag{19}$$

Further details of the exergo-economic analysis, cost balance equations and exergo-economic factors are discussed at length in the literature [3, 14, 15].

In addition, several methods have been suggested to express the purchase cost of equipment in terms of design parameters in Eq. 15. However, we used the cost functions suggested by Ameri et al. [2].

Nevertheless, some modifications were made to tailor these results to regional conditions in Iran and to take account of the inflation rate. To convert the capital investment into cost per time unit, one may write:

$$\dot{Z}_k = Z_k \cdot CRF \frac{\varphi}{N \cdot 3600} \tag{20}$$

Where Z_k is the purchase cost of k_{th} component in US \$. N is the annual operation hours of the unit, and φ (1.06) is the maintenance factor [2, 14].The Capital Recovery Factor (CRF) depends on the interest rate as well as estimated equipment lifetime. CRF is determined using the relation [2]:

$$CRF = \frac{i\,(1+i)^n}{(1+i)^n - 1} \tag{21}$$

In which i is the interest rate and n is the total operating period of the system in years.

Finally, in order to determine the cost of exergy destruction of each component, the value of exergy destruction, $\dot{E}x_{D,k}$ is estimated using the exergy balance equation in the previous section.

3.1. Cost balance equations

In order to estimate the cost of exergy destruction for each component of the plant, first one should solve the cost balance equations for each component. Therefore, for application of the cost balance equation (Eq. 15), there are usually more than one inlet and outlet streams for some components. In this case, the number of unknown cost parameters is higher than the number of cost balance equations for that component. Auxiliary exergo-economic equations are developed to solve this problem [2, 14]. Implementing Eq. 16 for each component together with the auxiliary equations forms a system of linear equations as follows:

$$\left[\dot{E}x_k\right] \times [c_k] = \left[\dot{Z}_k\right] \tag{22}$$

Where $\left[\dot{E}x_k\right]$, $[c_k]$ and $\left[\dot{Z}_k\right]$ are the matrix of exergy rate (obtained in exergy analysis), exergetic cost vector (to be evaluated) and the vector of $\dot{Z}_k$ factors (obtained in economic analysis), respectively. The cost function for each component in the cycle is presented in table 1. After estimation of C_i, the cost of exergy destruction will be calculated based on Eq. 19.

In this equation, c_f is 0.003 \$/MJ. Therefore, by solving these sets of equations, one can find the cost rate of each line in Fig. 1. Moreover, they are used to find the cost of exergy destruction for each component of the plant.

4. Thermodynamic modeling

To find the optimum physical and thermal design parameters of the system, a simulation program was developed using Matlab software for the gas turbine power plant. Thermodynamic properties, exergy flows, exergy efficiencies and cost of exergy destruction are calculated by using this code. The energy balance equations for various parts of the gas turbine cycle (Fig. 1) are as follows:

Table 1: Purchase cost function of each piece of equipment in the gas turbine

System component	Capital or investment cost functions
Z_{AC}	$Z_{AC} = \left(\frac{c_{11}\dot{m}_a}{c_{12}-\eta_{AC}}\right)\left(\frac{P_2}{P_1}\right)\ln\left(\frac{P_2}{P_1}\right)$ $c_{11} = 71.10\,\$/(kg/s)$, $c_{12} = 0.9$
Z_{CC}	$Z_{CC} = \left(\frac{c_{21}\dot{m}_a}{c_{22}-\frac{P_4}{P_3}}\right)\left[1+\exp\left(c_{33}T_{GTIT}-c_{24}\right)\right]$ $c_{21} = 46.08$, $c_{33} = 0.995$, $c_{24} = 26.4$
Z_{GT}	$Z_{GT} = \left(\frac{c_{31}\dot{m}_g}{c_{32}-\eta_{GT}}\right)\ln\left(\frac{P_C}{P_D}\right)\left[1+\exp\left(c_{33}T_3-c_{34}\right)\right]$ $c_{31} = 479.34$, $c_{32} = 0.92$, $c_{34} = 54.4$

$$\begin{bmatrix} \dot{E}x_1 & 0 & 0 & 0 & 0 & 0 \\ \dot{E}x_1 & \dot{E}x_2 & 0 & 0 & \dot{w} & 0 \\ 0 & \dot{E}x_2 & \dot{E}x_3 & 0 & 0 & 0 \\ 0 & 0 & \dot{E}x_3 & \dot{E}x_4 & \dot{E}x_5 & \dot{E}x_6 \\ 0 & 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & -1 \end{bmatrix} \times \begin{bmatrix} c_1 \\ c_2 \\ c_3 \\ c_4 \\ c_5 \\ c_6 \end{bmatrix} = \begin{bmatrix} 0 \\ -\dot{Z}_{comp} \\ -c_f\dot{m}_f LHV - \dot{Z}_{CC} \\ -\dot{Z}_{GT} \\ 0 \\ 0 \end{bmatrix} \tag{23}$$

Air compressor.

$$T_2 = T_1\left[1+\frac{1}{\eta_{AC}}\left(r_c^{\frac{\gamma_a-1}{\gamma_a}}-1\right)\right] \tag{24}$$

$$\dot{W}_{AC} = \dot{m}_a C_{p,a}\left(T_2 - T_1\right) \tag{25}$$

Where $C_{p,a}$ is the specific heat at constant pressure and could be considered as a temperature variable function as follows [1]:

$$C_{p,a}(T) = 1.04841 - \frac{3.8371}{10^4}T + \frac{9.4537}{10^7}T^2 - \frac{5.49031}{10^{10}}T^3 + \frac{7.9298}{10^{14}}T^4 \tag{26}$$

Combustion Chamber (CC).

$$\dot{m}_a h_2 + \dot{m}_f LHV = \dot{m}_g h_3 + \left(1-\eta_{cc}\right)\dot{m}_f LHV \tag{27}$$

$$\frac{P_3}{P_2} = \left(1 - \triangle P_{cc}\right) \tag{28}$$

Gas turbine.

$$T_4 = T_3\left\{1-\eta_{GT}\left[1-\left(\frac{P_3}{P_4}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right]\right\} \tag{29}$$

$$\dot{W}_{GT} = \dot{m}_g C_{p,g}\left(T_3 - T_4\right) \tag{30}$$

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} \tag{31}$$

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{32}$$

Where $C_{p,g}$ is taken as a temperature variable function as follows [1]:

$$C_{p,g}(T) = 0.991615 - \frac{6.99703}{10^5}T + \frac{2.7129}{10^7}T^2 - \frac{1.22442}{10^{10}}T^3 \tag{33}$$

5. Case study

To validate the results of our simulation code, they are compared with actual data from an operational gas turbine power plant in Yazd Power Plant (Yazd, Iran). This power plant is located near the city of Yazd, in central Iran. The schematic of this power plant is shown in Fig. 1. Based on power plant data gathered in 2006 the incoming air is at a temperature of around 17.10°C and a pressure about 0.874 bar. The pressure increases to 10.593 bar through the compressor, which has an isentropic efficiency of 83%. The gas turbine inlet temperature is 1073°C. The turbine has isentropic efficiency of 87%. The fuel (natural gas) is injected at 17.10°C and 30 bar.

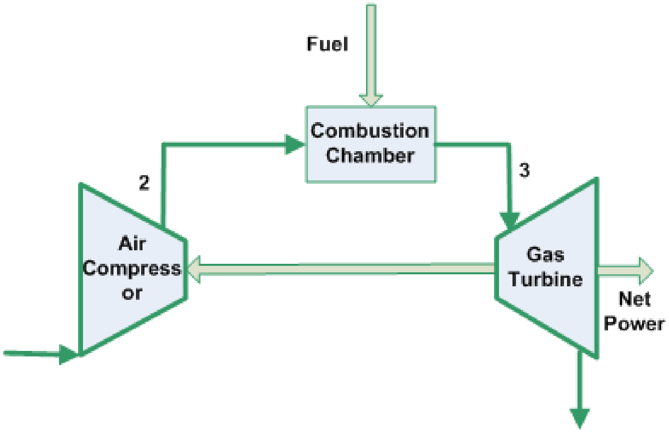

Figure 1: Schematic of a gas turbine power plant

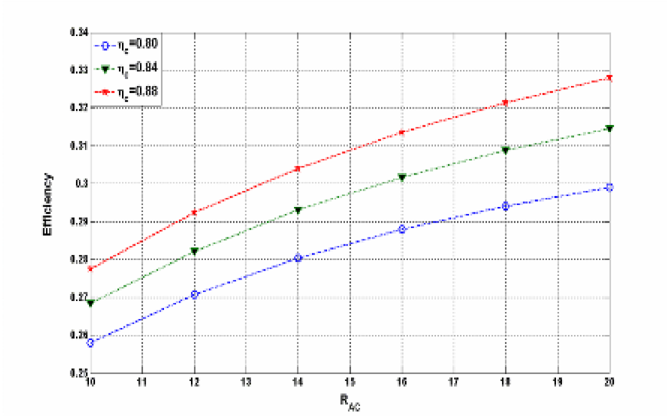

Figure 3: Effect of compressor pressure on cycle exergy efficiency

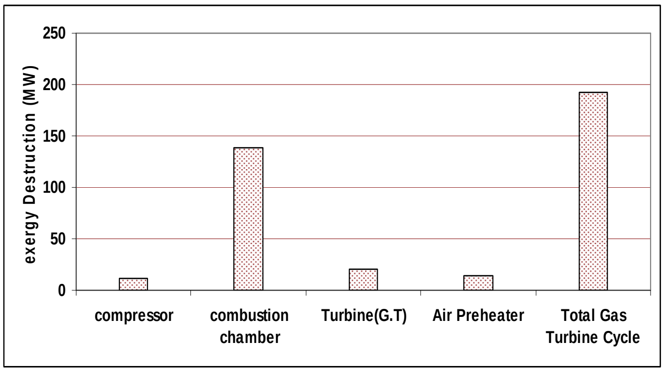

Figure 2: Exergy destruction of each component of the gas turbine cycle

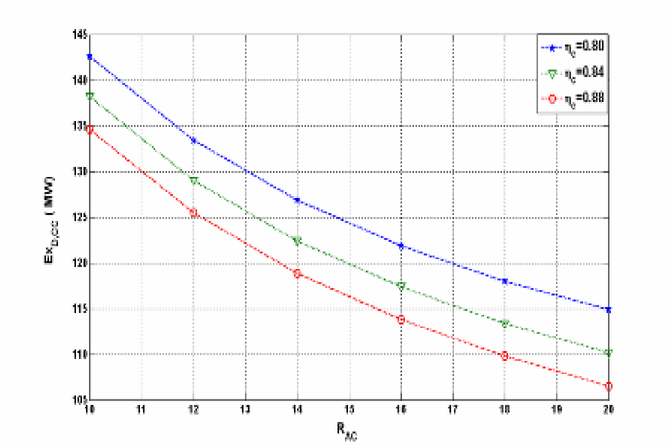

Figure 4: Effect of compressor pressure on combustion chamber exergy destruction

6. Results and discussions

6.1. Exergy analysis results

The performance analysis of the gas turbine cycle is investigated, taking into consideration realistic conditions such as temperature and pressure for each component in the unit. The air conditions at the compressor inlet are set at 0.874 bar and 298 K. In this case, the net output power of the gas turbine cycle is fixed at 106 MW. In addition, the heat losses through the combustion chamber are assumed to be 3%. The isentropic efficiency of the compressors is 83%, and the isentropic efficiency of the gas turbines is fixed at 87%. The gas turbine inlet temperature is varied between 1100 K and 1450 K, and the compressor pressure ratio of 10 to 20 is chosen in this study.

The exergy destruction of the components in the GT is shown in Fig. 2.The results from the exergy analysis show that, for the above conditions, the combustion chamber is the most significant exergy destructor in the combined cycle power plant. This is due to the fact that the chemical reaction and the large temperature difference between the burners and working fluid are the main source of irreversibility. In fact, its exergetic efficiency is lower than other components. Fig. 3 shows the effect of changes in the compressor pressure ratio versus exergy efficiency. Results show that for a gas turbine inlet temperature of around 1450 K, the gas turbine cycle exergy efficiency increases at a higher-pressure ratio.

Fig. 4 shows the effect of the compressor pressure ratio on the combustion chamber exergy destruction. It is shown that a higher-pressure ratio leads to lower exergy destruction in the whole cycle, which results

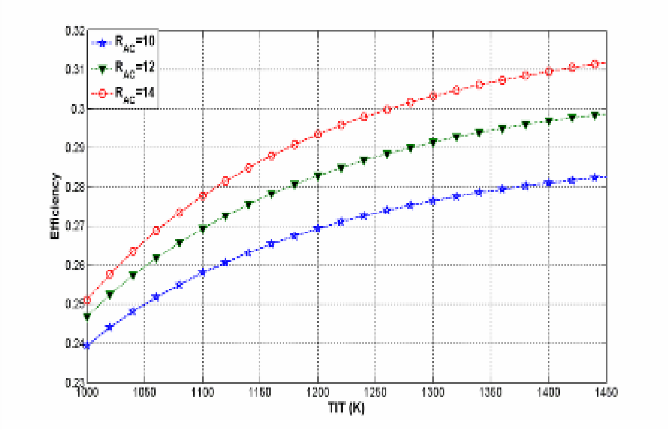

Figure 5: Effect of TIT variation on GT cycle exergy efficiency

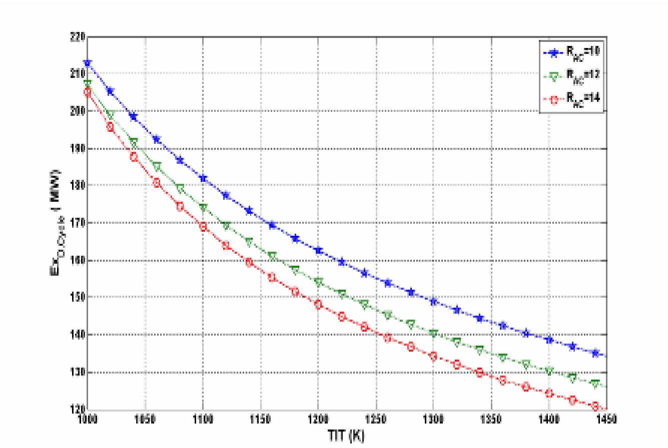

Figure 6: The effects of gas turbine inlet temperature on the total exergy destruction rate of the cycle

in less fuel supplied to the gas turbine cycle. This means that any saving in the fuel supplied has a significant impact on the total exergy destruction of the gas turbine cycle. The exergy of the fuel consists of physical and chemical exergy. However, the chemical exergy has a significant impact on the total exergy of fuel when compared to the physical exergy.

Fig. 5 shows the effect of variation of gas turbine inlet temperature on gas turbine exergy efficiency. It shows that an increase in the GTIT leads to an increase in the GT exergy efficiency due to the fact that the GT turbine work output increases. Fig. 6 confirms that an increase in the TIT leads to a reduction in exergy destruction as was concluded by Fig. 5. Therefore, it was found that TIT is the most important parameter in designing the gas turbine cycle due to the decrease in exergy destruction and increase in

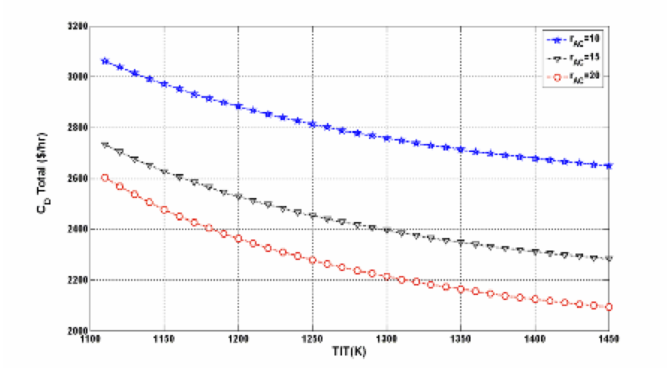

Figure 7: Total cost of exergy destruction versus TIT

cycle exergy efficiency.

6.2. *Exergo-economic Analysis Results*

To provide good insight into this study, the exergo-economic analysis is performed for the GT cycle power plant shown in Fig. 1. By solving Eq. 23, determining Ci and using Eq. 19, the cost of exergy destruction for each component is estimated. The results of the exergo-economic analysis are shown in Fig. 7. This figure shows that, like the exergy analysis results, the cost of exergy destruction for the combustion chamber decreases with an increase in the gas turbine inlet temperature (TIT). This is due to the fact that the cost of exergy destruction is proportional to the exergy destruction. Hence, an increase in the gas turbine inlet temperature can decrease the cost of exergy destruction. The results show that at constant TIT, increasing the compressor pressure ratio results in a decrease in the total cost of exergy destruction. The main reason for this is the reduction in the combustion chamber fuel mass flow rate.

7. Conclusion

Both thermodynamic modeling and exergy and exergo-economic analysis of a gas turbine cycle were performed as part of this research study. The results from the exergy analysis show that the combustion chamber is the most significant exergy destructor in the power plant, which is due to the chemical reaction and the large temperature difference between the burners and working fluid. Moreover, the results show that an increase in the TIT leads to an increase in gas turbine exergy efficiency due to a rise in the output power of the turbine and a decrease in the combustion chamber losses.

Furthermore, the results from the exergoeconomic analysis, in common with those from the exergy analysis, show that the combustion chamber has the greatest cost of exergy destruction compared to other components. In addition, the results show that by increasing the TIT the gas turbine cost of exergy destruction can be decreased.

References

[1] M. Ameri, P. Ahmadi, S. Khanmohammadi, Exergy analysis of a 420 mw combined cycle power plant, International Journal of Energy Research 32 (2008) 175–183.

[2] M. Ameri, P. Ahmadi, A. Hamidi, Energy, exergy and exergoeconomic analysis of a steam power plant (a case study), International Journal of Energy Research 33 (2009) 499–512.

[3] M. Ameri, P. Ahmadi, S. Khanmohamadi, Exergy analysis of supplementary firing effects on the heat recovery steam generator, in: Proceedings of the 15th international conference on Mechanical Eng, no. 2053, Tehran, Iran, 2007.

[4] M. Ameri, P. Ahmadi, The study of ambient temperature effects on exergy losses of a heat recovery steam generator, in: Proceedings of the International Conference on Power Eng., Hang Zhou, China, 2007, pp. 55–61.

[5] O. Balli, H. Aras, Energetic and exergetic performance evaluation of a combined heat and power system with the micro gas turbine (mgtchp), International Journal of Energy Research 31 (14) (2007) 1425–1440.

[6] B. Sahin, K. Ali, Thermo-dynamic analysis of a combined carnot cycle with internal irreversibility, Energy 20 (12) (1995) 1285–1289.

[7] P. Ahmadi, I. Dincer, Thermodynamic and exergoenvironmental analyses, and multi-objective optimization of a gas turbine power plant, Applied Thermal Engineering Journal 31 (2011) 2529–2540.

[8] M. A. Ehyaei, S. N. Hakimzadeh, N. Enadi, P. Ahmadi, Exergy and economic analysis of capacity enhancement of the gas turbine power plant using an absorption chiller, International Journal of Energy Research-doi:DOI:10.1002/er.1814.

[9] T. Kotas, The Exergy Method of Thermal Plant Analysis, Butterworths, London, 1985.

[10] A. Cihan, O. Hacıhafizoglu, K. Kahveci, Energy-exergy analysis and modernization suggestions for a combined-cycle power plant, Int. J. Energy Research 30 (2006) 115–126.

[11] P. Ahmadi, Exergy concepts and exergy analysis of combined cycle power plants (a case study in Iran), B.Sc. Thesis, Energy Engineering Department, Power & Water University of Technology (PWUT), Tehran, Iran (2006).

[12] M. Moran, Availability Analysis Guide, to Efficient Energy Use Englewood Cliffs, Prentice-Hall, N.J., 1989.

[13] G. Tsatsaronis, Definitions and nomenclature in exergy analysis and exergoeconomics, Energy 32 (2007) 249–253.

[14] M. A. Rosen, I. Dincer, Thermoeconomic analysis of power plants: an application to a coal fired electrical generating station, Energy Conversion and Management 44 (2003) 2743–2761.

[15] A. Bejan, G. Tsatsaronis, M. Moran, Thermal Design and Optimization, Wiley, New York, 1996.