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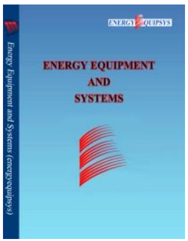

Optimization and analysis of exergy, economic and environmental of gas turbine (Case study: Kerman plant combined cycle)

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ABSTRACT

It is important to study real thermodynamic systems and evaluate the inefficiency of thermodynamic components in power plants in order to improve their performance. In this work, exergy analysis and investigation of the gas turbine segment of the Kerman combined cycle, are examined. The results reveal that the most exergy efficiency defect is related to the combustion chamber and compressor and finally to the gas turbine, respectively, and also in the combustion chamber the irreversibilities are mainly due to chemical nonequilibrium, thermal nonequilibrium, and mechanical nonequilibrium caused by combustion; 22.835% of exergy is lost during the combustion process. Therefore, this work introduces and defines ecological - cost or ecologicost as a new criterion. Ecologicost indicates that dissipated energy is more than production power; In other words, the system is defective and needs to be repaired or replaced. Ecologicost has the potential to replace traditionally criteria. Moreover, the ecological function and ecologicost value of the gas turbine are -20.418 MW and -0.05701, respectively. This Negative number indicates that system goes to destruction and needs repair or replacement or improvement. The optimization results show that the maximum power, maximum energy efficiency, maximum exergy efficiency, maximum ecological, and maximum ecologicost are 136.437 MW, 38%, 48.39%, 6.069 MW, and 0.01695, respectively.

Keywords: Ecologicost, Exergy Analysis, Optimization.

1. Introduction

Today, the development of a society depends on the development of industry and technology; industries need a lot of energy sources to develop. Without power plants, human life would be very difficult. The power plant industry is an interdisciplinary, and high technology industry [1].

Currently, the main problem with electricity is that it cannot be stored on a large scale or may become too expensive. The best way is to generate electricity when needed [2].

At present, combined cycles supply a large proportion of electricity. The combined cycle is a combination of the gas cycle (Brayton

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cycle) and the steam cycle (Rankine cycle). One of the most important parts of the combined cycle is wasted heat boiler or heat recovery steam generation (HRSG). The wasted heat boiler is located between the gas cycle and the steam cycle. The benefits of the combined cycle include flexibility, high efficiency, and fast startup and disadvantage is the complexity of the design. The lower environmental impacts are another advantage of combined cycles. Globally, stringent regulations are in place for the rate of emission of different types of pollutants. These regulations make energy producers pay particular attention to the low level of greenhouse gas emissions. The combined cycle power plants have lower investment costs and construction time compared to coal power plants and nuclear power plants.

Fakhari and et al. [3] performed Energy, exergy, economic, and environmental assessment and tri-objective optimization of a triple-pressure combined cycle power plant. The details considered in this study include combustion chamber steam injection, accurately solving combustion process utilizing the equilibrium constants. Their results indicate that HRSG has the maximum total cost rate among all the components, condenser and the combustion chamber have the lowest exergy efficiency with values of 11.23% and 68.8%, the HRSG and condenser have the maximum purchase cost rate.

Gas turbines were first used in aircraft in World War II; after a while, they were used to generate shaft power. Nowadays, gas turbines are known for their low cost, small size, fast installation and startup, high reliability. Generally, the components of gas turbines are nozzle and blades. A gas turbine or gas cycle is an internal combustion engine that uses air as the working fluid. In a gas turbine, the air first enters the compressor. As a result, the outlet air pressure increases, then the fuel is sprayed into the high pressure air in the combustion chamber, and finally, high-temperature exhaust gases enter the gas turbine, expanding the fluid and producing power. The gas turbines are smaller in size than steam power plants. Also, they are suitable for power generation during high-demand periods of grid because they can be fast startup and meet the grid requirement. One of the problems of a gas turbine is the limitation of the turbine inlet temperature (TIT) due to metallurgical constraints. Industrial gas turbines are used to generate power and can have a capacity of up to 450 MW [4]. However, the development of gas turbines continues. Common fuel in gas turbines is natural gas and diesel. Gas turbine power and efficiency are greater when compared to liquid fuels when using natural gas. In contrast, the losses in liquid fuel combustion are about 6%, but in natural gas, it is 11% [4]. Modi et al. [5] have evaluated the energy and exergy of 25 MW gas turbine that consumes ethane; they investigated the effect of ambient temperature on the exergy destruction of gas turbine components. Their results show that one-degree centigrade in environmental condition resulted in 30 kW of exergy destruction in the compressor.

For real Brayton analysis, irreversible effects such as friction and heat transfer must be taken into account; As a result, the isentropic process assumption of the turbine and compressor is not valid, and isentropic efficiency in the turbine and compressor should be used. The real Brayton cycle depends on the minimum and maximum temperatures of the cycle.

Methods to improve the performance of the Brayton cycle includes utilization of regenerator, reheater, and intercooler. Typically, the actual process efficiency is lower than the theoretical one, the reason is energy loss and destructed exergy, much of the energy loss is due to heat loss, but the exergy loss is often due to irreversibility. The efficiency can be improved by raising the maximum cycle temperature, dissipating heat at a lower temperature, or improving the process to minimize internal irreversibility [6].

One way to improve gas turbines is to reduce the temperature of the air entering the compressor. Hosseini and et al. [7] studied the evaporative cooling system at the Fars combined cycle power plant (Iran) and their results show that the output of the gas turbine of the Fars combined cycle power plant with a temperature of 38 °C and relative humidity of 8 percent is 11 MW more. Also, the temperature drop in the inlet air is about 19 °C with the evaporative cooling media. Refer to reference [8] for further study of the evaporative cooling medium. Another way to reduce the temperature of the air entering the compressor is to use an absorption chiller .In a study, Matjanov [9] used three heat sources to heat the absorption chiller. Their goal is to reduce the ambient temperature from 45 °C to 15 °C. The results show that if the existing heat dissipation of the gas turbine in the absorption chiller is used, as a result, it is not economically viable, because efficiency is greatly reduced, the use of wasted boiler waste gases for absorption chillers to reduce the inlet air temperature of the compressor is technically and economically attractive [10]. Analysis of using absorption chiller in a gas turbine for Tabas and Bushehr is performed by Ehyaei and et al. [11]. Their results showed that output power was increased 11.5 and 10.3%, second law efficiency was increased to 22.9 and 29.4% for Tabas and Bushehr cities, and the cost of electricity production for Tabas and Bushehr cities was decreased to about 5.04 and 2.97%.

Ambient temperature is an effective parameter for the combined cycle. This parameter affects on gas cycle and steam cycle (Especially HRSG). more detail about in reference [12].

The first innovation of this work is the applicability of this research; its results and performance test code can be used to improve the performance of the Kerman combined cycle power plant. This research is based on the actual conditions of the gas turbine.

The turbomachineries exit temperature such as compressor and turbine is a function of isotropic efficiency, heat capacity ratio, and gas mixture components. Then, compressor and turbine exit temperature is calculated by an iterative algorithm.

Another novelty is the introduction of a new thermodynamic analysis criterion (ecologicost) that includes more goals. Ecologicost considerations can be a good criterion in terms of production power, exergy destruction, system size (It is directly related to cost), and fuel cost.

Nomenclature

C_p	Constant-pressure heat capacity(kJ/kg.K)
Ex	Exergy (kJ)
K	Heat capacity ratio
C	Mass fraction
Y	Mole fraction
r_p	Pressure ratio
T	Temperature (K)
W	Work (kJ)

Greek symbols

η	Efficiency
Y	Exergy or rational efficiency
W	Ratio of fuel exergy to fuel low heat value
λ	Fuel air ratio

Subscripts

A	Air
$Bray$	Brayton cycle
H	Enthalpy
G	Gas
Mix	Mixture
Th	Thermal
T	Total

2. System description

The studied system is gas turbine of Kerman combined cycle power plant. Based on field research, input parameters are determined and are shown in Table 1. The performance criterion of the system is calculated based on the measured variables and the specified parameters. The measuring devices accuracy is very important; because independent variables or measurement variables greatly influence the final results. Diagram of studied gas turbine shows in Fig.1.

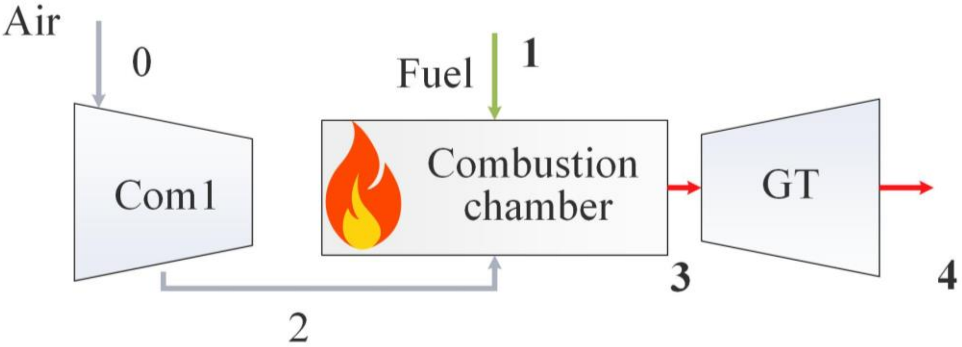
The results of the air analysis or air composition are shown in Table 2.

3. System modeling

The proposed gas turbine consists of the thermodynamic component of the combustion chamber, compressor and gas turbine. The simulation was performed using engineering equation solve (EES) and MATLAB. The hypotheses of this research include the following:

- The pressure drop at the inlet due to the filters is considered.
- The exhaust pressure drop of the stack is considered.
- Combustion chamber pressure drop is considered.
- Combustion products are considered as mixtures.

(a)



(b)

Fig.1. View and diagram of studied gas turbine

Table 1. Thermodynamic system specifications

Variable	Unit	Value
Ambient temperature	℃	10.25
Ambient pressure	*mbar*	841.3977733
Compressor pressure ratio	-	11.1779
Turbine pressure ratio	-	0.0948
Fuel	-	Natural gas
Burner	-	Premix

Table 2. Composition of air in gas turbine

Composition of air	O_2	N_2	Ar	CO_2	H_2O
Molar fraction	20.8301	77.7104	0.9113	0.0328	0.5155

The study is based on the first law of thermodynamics, the second law of thermodynamics, economics and environment. The first law of thermodynamics is as follows[13]:

$$\frac{dE}{dt}=\dot{Q}-\dot{W}+\sum_{in}\dot{m}\left(h+\frac{1}{2}V^{2}+gz\right)-\sum_{out}\dot{m}\left(h+\frac{1}{2}V^{2}+gz\right) \tag{1}$$

In Eq (1), Q, W, h, z, V work, heat transfer, enthalpy, height, and velocity respectively.

When exergy analysis is performed, thermodynamic defects can be measured as destructed exergy, which indicates a reduction in the quality of useful energy. Exergy destruction does not tell the whole truth [14, 15].

$$\dot{E}x_{in}-\dot{E}x_{out}-\dot{E}x_{des}=\frac{dEx_{CV}}{dt} \tag{2}$$

The first use of enthalpy in polynomial form was proposed by Benson [16], in this method, data are presented in the form of tables or charts. The real enthalpy of a mixture of a gas is calculated by the following equation. It used the thermodynamic relations and the solution of differential equations, and the resulting equation is obtained as a function of temperature, pressure, volume, mole fraction, and etcetera. This method uses regression and the calculation of its coefficients. Determining the effective variable is very important because it improves the accuracy of the regression [13]. For example, the effective variable is temperature enthalpy [16].

$$h_{real}=a_1+a_2T+a_3T^{2}+a_4T^{3}+a_5T^{4} \tag{3}$$

These coefficients were obtained based on experimental and laboratory results.

3.1. Air compressor

The gas turbine compressor has 16 stages. After the air flow passes through the filters, the air flow enters the compressor, the compressor increases the air pressure by consuming work. The compressor outlet temperature is calculated based on the following Equation [13]:

$$T_e=T_i\cdot\left(1+\left(\frac{1}{\eta_{com}}\right)\cdot\left(\left(\left(\frac{P_e}{P_i}\right)^{\frac{k-1}{k}}\right)-1\right)\right) \tag{4}$$

Equation (4) is very different from the real conditions. Real and theoretical gas turbines are very different. The real gas turbine has details including variable turbine isotropic efficiency, variable compressor isentropic efficiency, and gas mixture, etcetera. Isentropic efficiency compressor is calculated as below [17]:

$$\eta_{iC}=\frac{\beta^{\tilde{\mu}_C}-1}{\beta^{\tilde{\mu}_C/\eta_P}-1} \tag{5}$$

$$\tilde{\mu}_C=R/\tilde{C}_{Ps} \tag{6}$$

Polytropic efficiency compressor is determined by pressure ratio or turbine inlet temperature [17].

$$\tilde{C}_{Ps}=\frac{\left[C_{ps}(CIT)\,ln(CIT/T_0)-C_{ps}(CET)\,ln(CET/T_0)\right]}{ln(CIT/CET)} \tag{7}$$

These details have led to the use of iterative algorithms .To solve this problem, the algorithm of Fig. 2 is used.

3.2. Combustion chamber

The chemical reaction carried out in the combustion chamber is as follows.

$$\begin{aligned}&aCH_4+bC_2H_6+cC_3H_8+dC_4H_{10}(i)\\&+fnC_4H_{10}+gC_5H_{12}(i)+hC_5H_{12}(n)\\&+iC_6H_{14}(i)+jC_7H_{16}+kN_2+lCO_2\\&+nO_2+mN_2+zH_2O+yCO_2+xAr\\&\rightarrow n_{CO_2}CO_2+n_{N_2}N_2+n_{O_2}O_2+n_{H_2O}H_2O\\&+n_{NO}NO+n_{CO}CO+n_{Ar}Ar\end{aligned} \tag{8}$$

Calculations are performed assuming a complete reaction (n_{CO}=0, n_{NO}=0). Combustion products are calculated based on carbon, hydrogen, oxygen, nitrogen balance. The above equation is a general equation that is based on Hess's law. One of the important objectives of performance testing is to determine the turbine inlet temperature to be in allowable range. The analysis is performed on two methods, the first method based on enthalpy in polynomial form and the second method based on academic methods and theory. Most engineering applications and real systems have mixture working fluid. In the first method, the enthalpy of combustion gases is calculated as real mixture enthalpy based on references [18, 19], Siemens catalogues and standards. The most important point in calculating power plant performance testing is the method of calculating enthalpy as a thermodynamic potential. The second method uses mixture definitions. The enthalpy of a mixture is a function of temperature, the pressure and composition of the mixture.

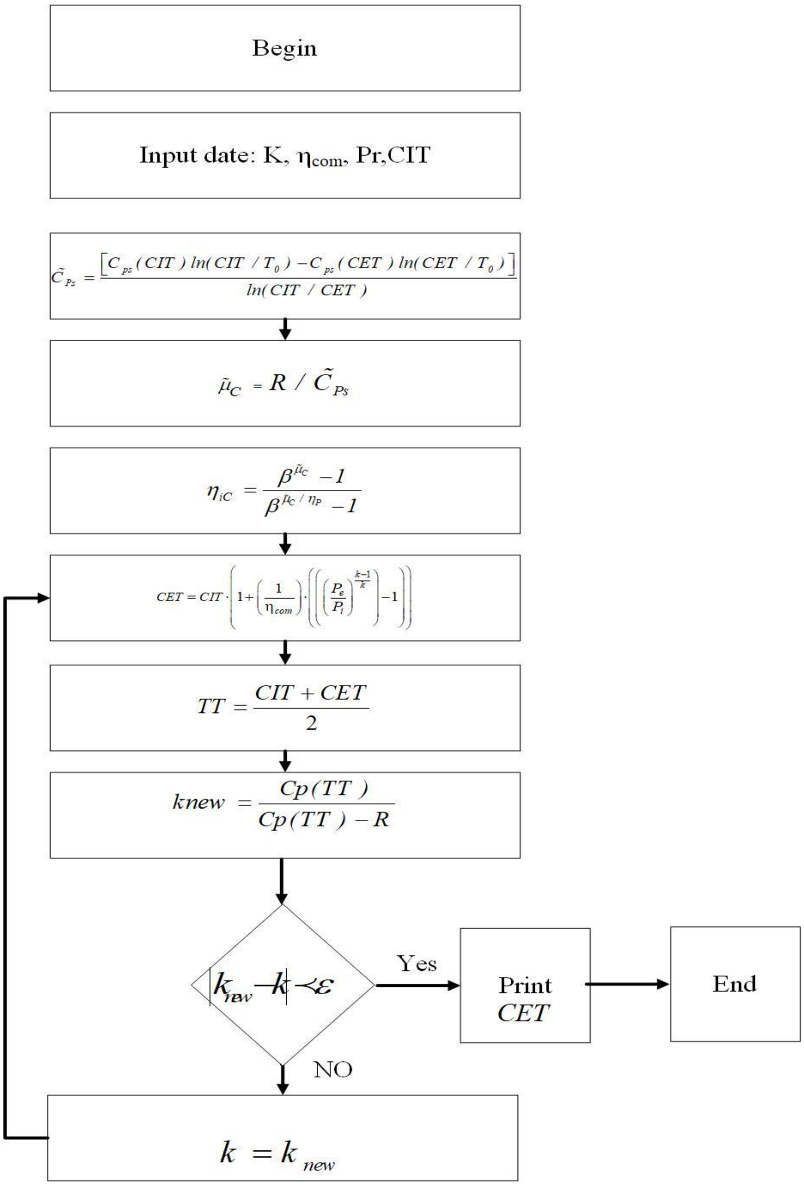


Fig. 2. iterative algorithms for calculation compressor exit temperature

It is necessary to refer to some of the thermodynamic properties of the mixtures. The mass and mole fraction of the components of a mixture are as follows [20-22]

$$c_i = \frac{m_i}{m_{tot}} \tag{9}$$

$$y_i = \frac{n_i}{n_{tot}} \tag{10}$$

The molar enthalpy of the mixture is calculated as follows.

$$\bar{h}_i = \sum_i^n y_i \bar{h}_i \tag{11}$$

Molar Heat Capacity is a function of the heat capacity of each component and its molar fraction.

$$\bar{C}_{p,mix} = \sum_i^n y_i \bar{C}_{pi} \tag{12}$$

Kurt and et al. [23] proposed the following equations to calculate the heat capacity of air and combustion gases .These equations are used in most gas cycle research, while the thermal capacity of the mixture is subject to the combinations of air and combustion gases that must vary under different conditions [24-26]

$$C_{pa}(T)=1.04841-(\frac{3.83719T}{10^4})+(\frac{9.45378T^2}{10^7})-(\frac{5.49031T^3}{10^{10}})+(\frac{9.92981T^4}{10^{14}}) \quad (13)$$

$$C_{pg}(T)=0.991615-(\frac{6.99703T}{10^5})+(\frac{2.71298T^2}{10^7})-(\frac{1.22442T^3}{10^{10}}) \quad (14)$$

3.3. Turbine

The studied turbine has four stages. Using the isentropic efficiency turbine, pressure ratio, heat capacity ratio and turbine inlet temperature, the turbine outlet temperature can be calculated[27].

$$T_e = T_i \cdot \left(1+(\eta_{tur})\cdot\left(\left(\left(\frac{P_e}{P_i}\right)^{\frac{k-1}{k}}\right)-1\right)\right) \quad (15)$$

The isentropic efficiency turbine is calculated as follows:

$$\eta_{it} = \frac{1-\beta^{\tilde{\mu}_T/\eta T}}{1-\beta^{\tilde{\mu}_T}} \quad (16)$$

$$\tilde{\mu}_T = R/\tilde{C}_{PsT} \quad (17)$$

The polytropic efficiency turbine is determined based on the pressure ratio. Therefore, $\tilde{C}_{PsT}$ is calculated as follows:

$$\tilde{C}_{PsT} = \frac{\left[C_{ps}(TIT)ln(TIT/T_0)-C_{ps}(TET)ln(TET/T_0)\right]}{ln(TIT/TET)} \quad (18)$$

Similar to the compressor in the turbine, there are details that must be considered. These details include variable isentropic efficiency turbine, cooling blade, variable heat capacity ratio, and etcetera (Fig. 3).

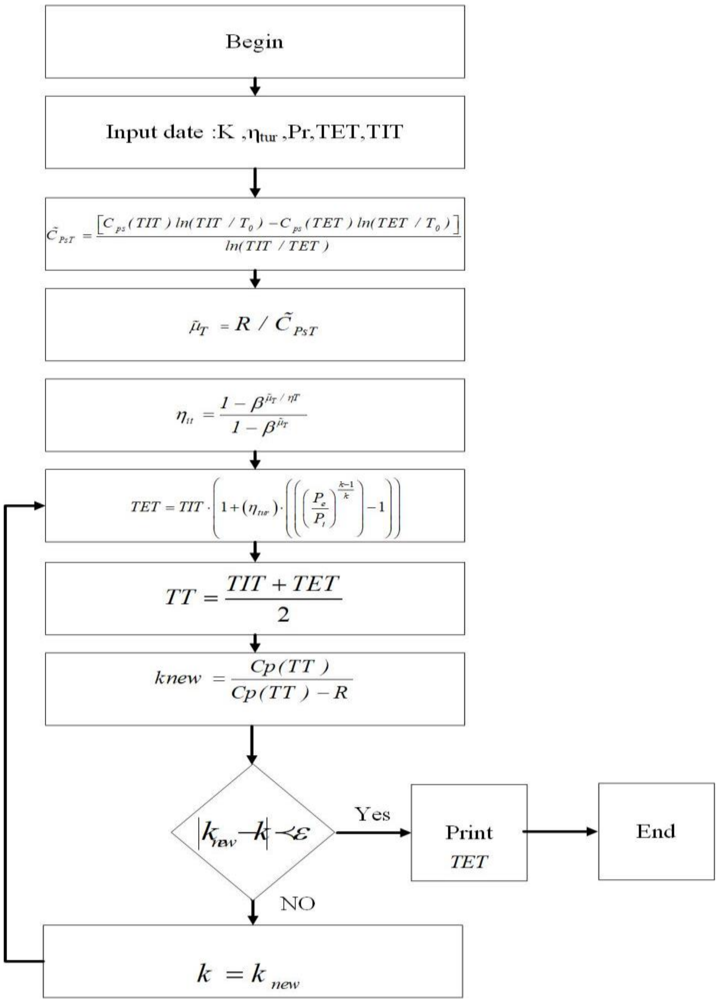


Fig. 3. Calculation of turbine exit temperature using iterative algorithm

To validation consider combustion chamber and compressor as control volume, the turbine input enthalpy is calculated based on the energy balance.

$$h_{TI} = \frac{m_{CI} h_{CI} + m_f h_f \eta_{cc}}{m_f + m_{CI}} \tag{19}$$

It is very important to determine the thermodynamic properties of combustion gases (mixture) because the output power and thermal efficiency are a function of the enthalpy of combustion gases, enthalpy reference based on International Standards Organization (ISO) is zero degrees Celsius.

4. Exergy analysis

Exergy represents useful energy, it indicates the quality of energy conversion, and exergy is based on the second law of thermodynamics. Exergy analysis for the power plant is therefore carried out in order to examine the energy more accurately or in other words to evaluate the energy qualitatively. One of the important concepts in exergy analysis is exergy efficiency or rational efficiency expressed for the purpose of comparing macroscopic forms of energy [28, 29].

Exergy has become a powerful tool for designing and analyzing thermal systems. The first step in evaluating exergy is to define the fuel and product, and fuel is the source used to generate the product, both the fuel and the product are in the form of exergy. Exergy efficiency is a thermodynamic component based on product and fuel standard for measuring system performance [30].

$$\varepsilon_k = \frac{\dot{E}x_{P,k}}{\dot{E}x_{F,k}} = 1 - \frac{\dot{E}x_{D,k}}{\dot{E}x_{F,k}} \tag{20}$$

Exergy efficiencies can also be calculated for the total system as follows.

$$\varepsilon_{tot} = \frac{\dot{E}x_{P,tot}}{\dot{E}x_{F,tot}} = 1 - \frac{\dot{E}x_{D,tot} + \dot{E}x_{L,tot}}{\dot{E}x_{F,tot}} \tag{21}$$

Exergy destruction ratio is defined as the ratio of degraded exergy to exergy fuel of the whole system.

$$y_{D,k} = \frac{\dot{E}x_{D,K}}{\dot{E}x_{F,tot}} \tag{22}$$

In addition, exergy loss ratio can be defined as follows.

$$y_{L,k} = \frac{\dot{E}x_{L,K}}{\dot{E}x_{F,tot}} \tag{23}$$

On the other hand, ratio of exergy destruction to total exergy destruction can be shown as follows.

$$y^*_{D,k} = \frac{\dot{E}x_{D,K}}{\dot{E}x_{D,tot}} \tag{24}$$

The above relationships have many applications in determining the thermodynamic component with the most destruction exergy. On the other hand, y^+ can also be defined as follow.

$$y^+_{D,k} = \frac{\dot{E}x_{D,K}}{\dot{E}x_{F,K}} \tag{25}$$

Exergy or useful energy is one of the tools of thermodynamic engineering that indicates the irreversibility of the system. Exergy consists of two parts: physical, and chemical exergy, the kinetic energy and potential exergy being neglected .As a result, the exergy of the stream is calculated as follows.

$$ex = ex_{ph} + ex_{ch} \tag{26}$$

Physical exergy can be called thermo-mechanical exergy; this is due to the nonequilibrium in mechanical energy and thermal energy [24]. Thermo-mechanical exergy is written as follows.

$$ex_{ph} = ex_T + ex_p \tag{27}$$

$$ex_T = cp\left[(T - T_0) - T_0 \ln(T/T_0)\right] \tag{28}$$

$$ex_P = RT_0 \ln(\frac{p}{P_0}) \tag{29}$$

Chemical exergy is important in chemical reactions such as combustion and mixing and phase change. Applications of chemical exergy include combustion in gas turbines, fuel cells, distillation columns and the petrochemical industry. The chemical exergy of the mixture is calculated as follows[31].

$$ex_{ch} = \sum_i^n x_i Ex_i + RT \sum_i^n x_i \ln x^i \tag{30}$$

In order to calculate fuel exergy, the ratio of fuel exergy to fuel low heat value (LHV) is used.

$$\Omega = \frac{Ex_f}{LHV_f} \tag{31}$$

Ω for liquid fuels as follows.

$$\Omega = 1.0422 + 0.011925 y/x - 0.042/x \tag{32}$$

And also the ratio of chemical exergy to fuel value for gaseous fuels (C_xH_y) is calculated based on the following experimental equation [25].

$$\Omega = 1.033 + 0.01169 y/x - 0.0698/x \tag{33}$$

The exergy balance equation and the exergy efficiency for the thermodynamic components are shown in the following Table 3

In this section, the purchase cost function, cost balance and auxiliary equations of each component of the system are introduced and calculated. The basis of the analysis method is the direct relationship between fuel and product definitions for each component, which relates them to cost equations. The fuel cost rate is \$ 1.64 per kilowatt for natural gas [32]. Cost flow rate $\dot{C}\,(\$/\mathrm{h})$ is defined as follows:

$$\dot{C} = c\dot{E}x \tag{33}$$

In the above equation, $c(\$/kw)$ the cost is per unit of exergy. The cost balance for the thermodynamic component is calculated as follows:

$$\sum_k \dot{C}_{F,k} + \dot{Z} = \sum_k \dot{C}_{P,k} \tag{34}$$

In the above equation the purchase cost rate $(\dot{Z})$ is calculated based on the following Equation:

$$\dot{Z}_k = \frac{Z_k CRF\,\varphi}{N \times 3600} \tag{35}$$

In the above equation, Z_k is the cost purchasing the kth component (\$/s), (φ) maintenance factor, (N) number of operational hours per year, capital recovery factor (CRF) is defined as follows:

$$CRF = \frac{i \times (i+1)^n}{(i+1)^{n-1}} \tag{36}$$

To calculate the cost of energy flows and mass, a cost balance must be written for each component of the system. Usually, the number of equations is not equal to the number of unknowns, which can be obtained based on economic and thermodynamic concepts of auxiliary Equations.

The cost rate per unit of fuel exergy is calculated as follows:

$$c_{f,k} = \dot{C}_{f,k} / \dot{E}x_{f,k} \tag{37}$$

The cost rate in the exergy unit of the product is written as follows

$$c_{p,k} = \dot{C}_{p,k} / \dot{E}x_{p,k} \tag{38}$$

The cost rate per destroyed exergy unit is as follows.

$$\dot{C}_{D,k} = c_{f,k}\, \dot{E}x_{D,k} \tag{39}$$

The cost rate per unit of exergy lost is calculated as follows. The cost rate of exergy loss is usually considered to be zero, and this relationship is mostly used for flows that eventually lead to the natural environment.

Table 3. Exergy balance equation and exergy efficiency for thermodynamic components of gas turbine

Thermodynamic components	Exergy balance	Exergy efficiency
Compressor	$\dot{W}_{com1} = \dot{E}x_{48} - \dot{E}x_{46} + \dot{E}x_{des,com1}$	$\eta_{ex,com1} = \frac{\dot{E}x_{48} - \dot{E}x_{46}}{\dot{W}_{com1}}$
Combustion chamber	$\dot{E}x_1 + \dot{E}x_{48} = \dot{E}x_3 + \dot{E}x_{des,cc1}$	$\eta_{ex,cc1} = \frac{\dot{E}x_3}{\dot{E}x_1 + \dot{E}x_{48}}$
Gas turbine	$\dot{E}x_3 - \dot{E}x_4 = \dot{W}_{tur1} + \dot{E}x_{des,tur}$	$\eta_{ex,tur1} = \frac{\dot{W}_{tur1}}{\dot{E}x_3 - \dot{E}x_4}$

Table 4. Exergoeconomic assessment specification

Parameter	**Value**
Annual power plant operating hours	7000 hours
Total operational period	30 years
Interest rate	12 percent
Maintence factor	1.06

Table 5. Initial capital cost of purchasing equipment

Thermodynamic component	**Equation**
Compressor	$Z_{Aircom} = \left(\frac{C_{11} \cdot \dot{m}_a}{C_{12} - \eta_{com}} \right) \cdot (PR) \cdot ln((PR))$
Combustion chamber	$Z_{cc} = \frac{C_{21} \cdot \dot{m}_a}{C_{22} - \left(\frac{P_3}{P_2} \right)} \cdot (1 + exp(C_{23} \cdot TIT - C_{24}))$
Gas turbine	$Z_{GT} = \left(\frac{C_{31} \cdot \dot{m}_g}{C_{32} - \eta_{isengt}} \right) \cdot ln\left(\frac{P_3}{P_4} \right) \cdot (1 + exp(C_{33} \cdot TIT - C_{34}))$

Table 6. Constants of equipment purchase cost functions

Thermodynamic component	**Value**
Compressor	$c_{11} = 71.10(\$ / kg\ /s)$ $c_{12} = 0.9(\$ / kg\ /s)$
Combustion chamber	$c_{21} = 46.08(\$ / (kg\ /s))$ $c_{22} = 0.995$ $c_{23} = 0.018(1/K)$ $c_{24} = 26.4$
Gas turbine	$c_{31} = 479.34(\$ / (kg\ /s))^{0.8}$ $c_{32} = 0.92(\$ / kg\ /s)$ $c_{33} = 0.036(1/K)$ $c_{34} = 54.4(\$ / (kg\ /s))^{1.2}$

$$\dot{C}_{L,k} = c_{f,k}\, \dot{Ex}_{L,k} \tag{40}$$

The economic parameters in the exergy economy analysis including the number of hours of operation of the power plant and the life of the power plant, interest rate and profit rate are shown in the Table 4. [33].

Initial capital cost of purchasing equipment related to the gas turbine is shown in the Table 5.

Constants of equipment purchase cost functions are shown in the Table 6.

One of the important concepts of exergoeconomics is the relative cost difference, the positive role of exergy degradation corresponding to its usefulness in reducing system investment costs or other costs. The relative cost difference is a useful variable for evaluation and optimization.

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} \tag{41}$$

Cost-related resources are divided into two groups: non-exergy-related costs and exergy-related costs. non-exergy-related costs include the cost of purchasing equipment or initial investment and maintenance.

Another important variable used to evaluate system performance and cost is the exergoeconomic factor. The effect and trade off of exergy-related costs and non-exergy-related costs are determined. This factor is that it is possible to save (total) costs by improving efficiency other than thermodynamically, even with an increase in investment. On the other hand, a high amount of this factor means that the investment cost should be reduced even with There is a decrease in efficiency.

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{F,k}(\dot{E}x_{des,k} + Ex_{L,k})} \tag{41}$$

5. Result

The results of the calculation of turbine inlet enthalpy based on experimental and academic methods are shown in Table 7. Table 7 shows the actual temperatures based on NASA and ISO tables close to each other.

Table 8 shows that the difference between enthalpy calculation methods is high; this causes a great deal of error in the performance test results. In the Siemens system, the output temperature of the gas turbine is used to adjust the gas turbine, and the inlet turbine temperature is used as the control parameter. The efficiency of the power plant is affected by the fuel cost and net power of the power plant. In addition, in the analysis of the power plant, they use another criterion called "heat rate", which is the amount of heat needed to produce a unit of work, and its unit is kJ/kWh. Heat rate has inverse relationship with efficiency. Heat efficiency and rate are used in both net and gross form. Table 8 shows the performance test results

The results show a reduced cycle output power compared to previous performance tests. The turbine exit temperature is set at the appropriate limit of turbine inlet temperature. Inspections have reported favorable turbine appearance. Calculating the inlet temperature of the turbine is very costly and complicated and therefore calculates it in the performance test.

The results of the calculation of turbine inlet enthalpy based on experimental and academic methods are shown in Table 9. Table 9 shows the actual temperatures based on NASA and ISO tables close to each other.

Table 9 shows that the difference between enthalpy calculation methods is high; this causes a great deal of error in the performance test results. In the Siemens system, the output temperature of the gas turbine is used to adjust the gas turbine, and the inlet turbine temperature is used as the control parameter. The efficiency of the power plant is affected by the fuel cost and net power of the power plant. In addition, in the analysis of the power plant, they use another criterion called "heat rate", which is the amount of heat needed to produce a unit of work, and its unit is kJ/kgh. Heat rate has inverse relationship with efficiency. Heat efficiency and rate are used in both net and gross form. Table 10 shows the performance test results.

Table 7. Combustion gas temperature at inlet turbine in degrees Celsius

Actual temperature based on NASA tables	Actual temperature according to International Standards Organization tables	Based on Eq. (7) assuming ideal gas	Based on Eq. (9)
1064.03376	1065.56	983.3	1042.934

Table 8. Performance analysis results of gas turbine

Variable	Unit	Value
Output power	kW	122410.36
Inlet air mass flow rate	Kg/s	408.40
Turbine inlet temperature	$^{\circ}$C	1065.56
Thermal efficiency	%	34.1045
Heat rate	kJ/kWh	10555.7831

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5.1. Exergy assessment

Exergy efficiency and efficiency defects are shown in Fig.4. Therefore, the most efficiency defects is related to combustion chamber and compressor and finally to the gas turbine, respectively. In the combustion chamber the irreversibilities are mainly due to chemical nonequilibrium, thermal nonequilibrium, and mechanical nonequilibrium caused by the combustion process. So, the more nonequilibrium there are, the greater the irreversibility. Therefore, the improvement of combustion chamber performance has a great impact on gas turbine performance.

For a closer look, temperature - entropy diagram is plotted (Fig. 5). The irreversibilities in the thermodynamic components are evident in the Fig. 6. These irreversibilities are such as heat transfer due to temperature and pressure differences and fluid friction, mechanical friction, throttling, mixing, non-equilibrium chemical.

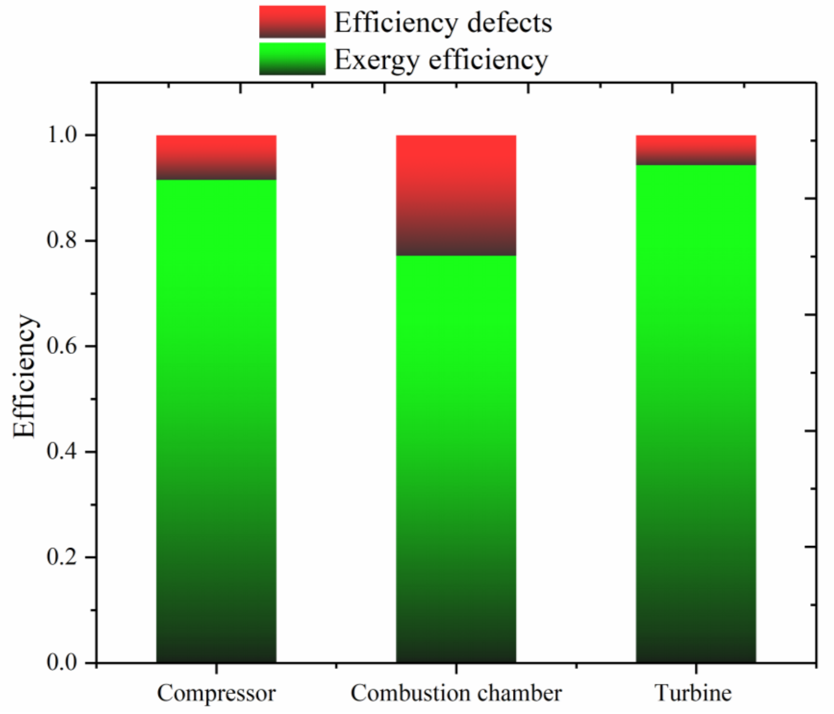

Fig. 4. Exergy efficiency and efficiency defects of studied gas turbine

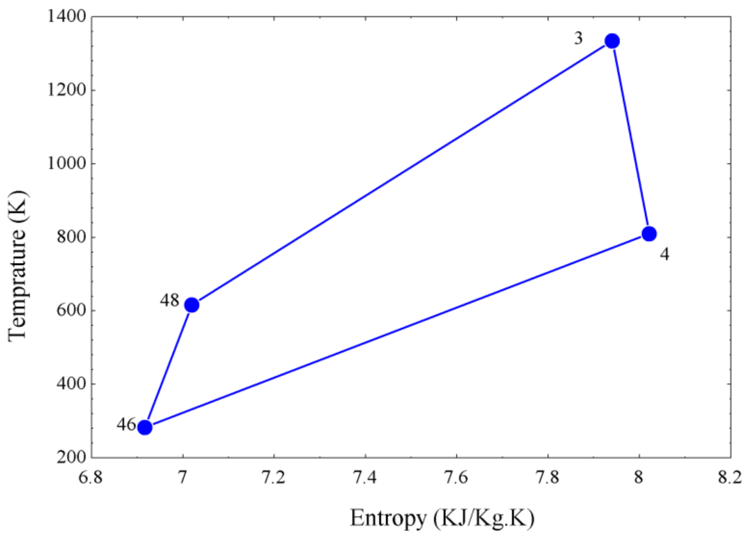

Fig. 5. Temperature - entropy plot of studied gas turbine

In order to examine more precisely the dissipated energy, destroyed exergy, each thermodynamic component is calculated. The results of the calculated exergy destruction are given in Fig. 6 Therefore, most of the exergy destroyed is related to the combustion chamber; 80.84% of exergy is lost during the combustion process; not all of this can be recovered because full energy conversion never happens. These results illustrate the importance of combustion research.

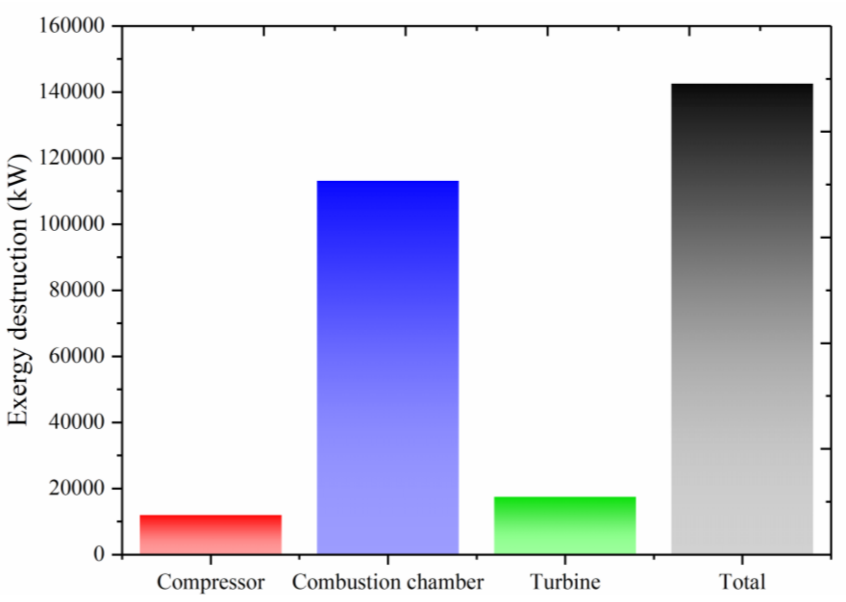

Fig. 6. Destroyed exergy of studied gas turbine

Table 11. Results of thermodynamic component exergy evaluation of gas turbine

Thermodynamic component	y^*_D	y^+_D	y_D	η_{ex}	$\dot{E}x_D$ (MW)	$\dot{E}x_P$ (MW)	$\dot{E}x_F$ (MW)
Compressor	0.06992	0.07149	0.0267	0.92	9.872	128.228	138.1
Combustion chamber	0.8084	0.2292	0.3087	0.77	114.13	383.833	497.976
Gas turbine	0.1217	0.06151	0.04647	0.9384	17.183	262.168	279.37

Table 11 shows the results of the exergy analysis that about 81% of the destructed exergy is related to the combustion chamber, which is due to the irreversibility of the combustion process and 12% of the exergy destruction is related to the gas turbine and 7% is related to the compressor. y_D and y_D^{+} are very important and practical, because it is used both for comparison and for analyzing the interactive effects between components. In the combustion chamber, about 30.87% of the total fuel exergy is destroyed, which is a very high amount. But, y_D in the gas turbine compressor 4.647 percent and 2.6 percent has been destroyed. Maximum y_D^{+} related to the combustion chamber, which is about 22.92%. After the combustion chamber, 7.149% of the compressor fuel exergy is destroyed, while only 6.151% of the gas turbine fuel exergy is destroyed.

5.2. Economic analysis of the studied gas turbine

The cost balance equations in the thermodynamic components of gas-turbine unit one are as follows. Results of exergy economics analysis gas turbine unit one reveal in Table 12.

Exergy economics integrates cost with efficiency, and this combination is highly efficient in the real system. The highest rate of investment and maintenance costs is related to the gas turbine, then the compressor, and finally the combustion chamber. On the other hand, the cost of degraded exergy is related to the combustion chamber and then to the gas turbine and compressor. The results show that the cost of destoried exergy predominates over the rate of investment and maintenance costs, which indicates the system's need to improve efficiency and performance.

r_k in the combustion chamber is about 0.3036, which indicates the high cost of exergy compared to fuel exergy, or in other words, the cost of increasing the efficiency of the combustion chamber is high, so the highest r_k is for the combustion chamber, compressor and gas turbine, respectively.

Exergoeconomic factor indicates the relationship between capital cost and energy efficiency of each component. High values of this factor indicate that in order to reduce system costs, the cost of that component should be reduced, while a low value factor indicates that equipment should be equipped with Choose better energy efficiency.

The lowest exergoeconomic factor is related to the combustion chamber, which indicates that a combustion chamber with better energy efficiency should be selected to reduce system costs.

According to the economic analysis, the Table 13 shows cost rates of gas turbine unit.

Table 12. Results of exergy economics analysis gas turbine unit one

Thermodynamic component	$\dot{C}_{des,k}$ ($/s)	$\dot{Z}_k$ ($/s)	$\dot{Z}_k+\dot{C}_{des,k}$ ($/s)	r_k	f_k
Compressor	0.0663	0.01821	0.08407	0.09812	0.2155
Combustion chamber	0.5813	0.01145	0.5928	0.3036	0.007668
Gas turbine	0.107	0.021162	0.1286	0.07878	0.1681

Table 13. Result of economic analysis

Point	**Cost rate ($/s)**
0	0
1	1.433
2	0.9457
3	2.39
4	0.6504

5.3. Environmental analysis

One of the problems with gas turbines is the production of pollutants such as carbon dioxide (CO2), carbon monoxide (CO) and nitrogen oxides (NOx) which cause environmental pollution. Consequences of environmental pollution include global warming, severe climate change, and etcetera. Therefore, the cost function of environmental pollution is calculated as follows.

$$\dot{C}_{env} = c_{CO}\dot{m}_{CO} + c_{NO_x}\dot{m}_{NO_x} + c_{NO_x}\dot{m}_{CO_2} \tag{42}$$

The cost per unit mass of emission of each pollutant is revealed in Table 14.

The amount of carbon dioxide is calculated from the results of the balance of combustion reaction. The equations for the amount of nitrogen oxide and carbon monoxide production are as follows.

$$\dot{M}_{NO} = \frac{0.15\cdot\left(10^{16}\right)\cdot\left(\tau^{0.5}\right)\cdot exp\left(-71100/T_{Pz}\right)}{\left(\left(P_2\cdot 1000\right)^{0.05}\right)\cdot\left(\frac{\Delta P}{P_2}\right)^{0.5}} \tag{43}$$

$$\dot{M}_{Co} = \frac{0.179\cdot\left(10^{9}\right)\cdot exp\left(-7800/T_{Pz}\right)}{\left(\left(P_2\cdot 1000\right)^{2}\right)\cdot\tau\cdot\left(\frac{\Delta P}{P_2}\right)^{0.5}} \tag{44}$$

These values are calculated per kg of fuel. In Eq. (46), T_{Pz} is the flame temperature, τ is the residence time of the combustion chamber, which is assumed to be 0.002 seconds. ΔP pressure drop in the combustion chamber. The flame temperature is calculated from the following equation.

$$T_{Pz} = A\cdot\left(\sigma^{\alpha}\right)\cdot exp\left(\beta\cdot\left(\sigma+\lambda\right)^{2}\right)\cdot\left(\pi^{x}\right)\cdot\left(\theta^{y}\right)\cdot\left(\psi^{z}\right) \tag{45}$$

In the above equation, ψ is the ratio of the number of carbon atoms to hydrogen. σ is calculated function of φ, where the fuel-to-air ratio is stoichiometric relative to the actual fuel-to-air ratio.

$$\begin{aligned} &\phi \le 1 \Rightarrow \sigma = \phi \\ &\phi \ge 1 \Rightarrow \sigma = \phi - 0.7 \end{aligned} \tag{46}$$

The constants of these equations are calculated from Table 15.

Therefore, the results of environmental analysis are shown in the Table 16.

Table 14. Pollutant emission cost

Pollutants	c_{CO}	c_{NO_x}	c_{CO_2}
Cost ($/kg)	6.242	8.175	0.024

Table 15. constants of Eq. (45)

Parameter	$0.3 \prec \phi \le 1$		$1 \prec \phi \le 1.6$	
	$0.92 \prec \theta \le 2$	$2 \prec \theta \le 3.2$	$0.92 \prec \theta \le 2$	$2 \prec \theta \le 3.2$
A	2361.764	2315.752	916.8261	1246.178
α	0.1157	-0.0493	0.2885	0.3819
β	-0.9489	-1.1141	0.1456	0.3479
λ	-1.0976	-1.1807	-3.2771	-2.0365
a_1	0.0143	0.0106	0.0311	0.0361
b_1	-0.0553	-0.045	-0.078	-0.085
c_1	0.0526	0.0482	0.0497	0.0517
a_2	0.3955	0.5688	0.0254	0.0097
b_2	-0.4417	-0.55	0.2602	0.502
c_2	0.141	0.1319	-0.1318	-0.2471
a_3	0.0052	0.0108	0.0042	0.017
b_3	-0.1289	-0.1291	-0.1781	-0.1894
c_3	0.0827	0.0848	0.098	0.1037

Table 16. Results of environmental analysis

$\dot{C}_{env}$ **($/s)**	$\dot{C}_{co2}$ **($/s)**	$\dot{C}_{co}$ **($/s)**	$\dot{C}_{Nox}$ **($/s)**
1.246	0.7542	0.4911	0.001099

As Table 16 shows, the highest cost rates are for carbon dioxide, carbon monoxide, and nitrous oxide, respectively. The total unit cost of pollution is about 1.245 dollars per second.

The performance of gas turbines is highly dependent on environmental conditions. Therefore, one-year data of ambient pressure, relative humidity and ambient temperature were collected over a 10-year period. Fig. 7. shows the ambient pressure, relative humidity and ambient temperature of the Kerman combined cycle. In order to examine more precisely the effects of environmental conditions on the performance of gas turbine are plotted in Fig. 7. The environmental analysis period is from 2008 to 2018. Average ambient pressure, average relative humidity average ambient temperature 83.06 kPa, 0.2735 and 17.68 °C, respectively.

Figure 8 shows energy efficiency, and exergy efficiency of gas turbine in the average environmental conditions of the months of the year. The minimum energy efficiency is in July and minimum exergy efficiency is in June.

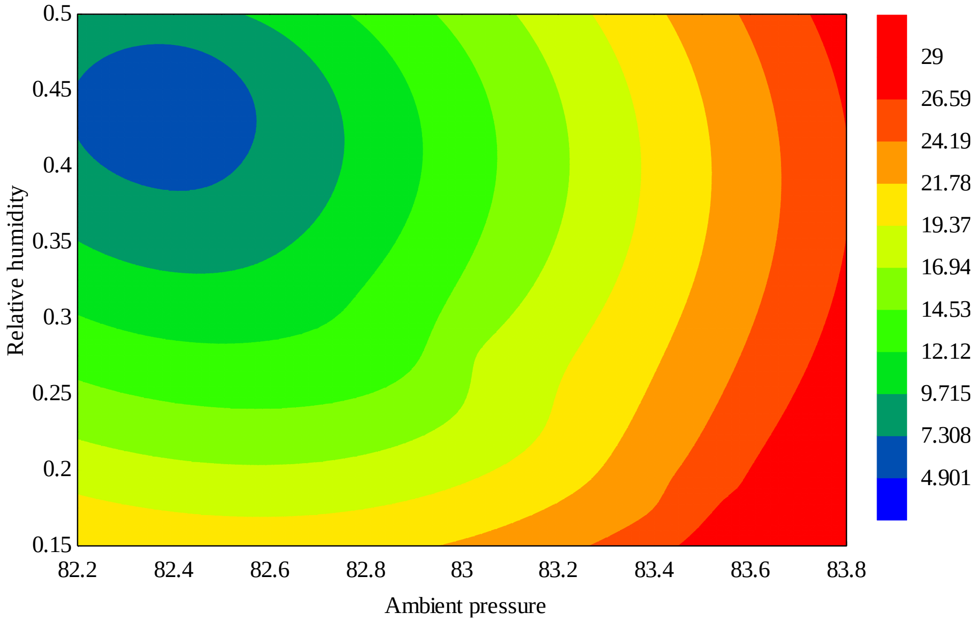

Fig. 7. Ambient pressure, relative humidity and ambient temperature in Kerman combined cycle

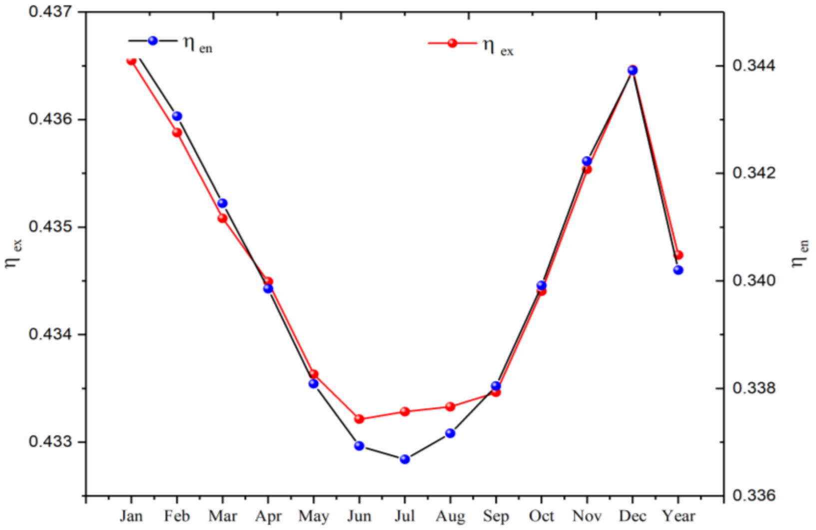

Fig. 8. Effect of environmental conditions on energy and exergy efficiencies

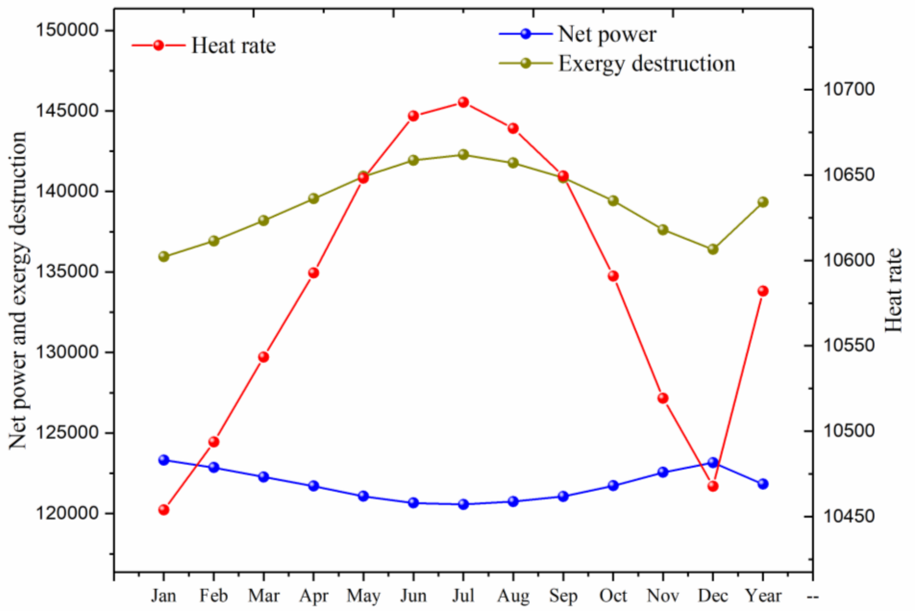

Fig. 9. Effect of environmental conditions on net power, exergy destruction and heat rate

Figure 9. shows the minimum net power in July. The main reason for the decrease in the net power is the direct relationship between the air specific volume and inlet ambient temperature.

5. Ecologicost evaluation

Finite time thermodynamics was first proposed with the idea of limiting time in thermodynamic cycles. Then, it was used finite heat transfer. Mechanical engineers, physicists, and chemical engineers take a different view [34-44]. Initially, only external irreversible were considered, but in the case studies, internal irreversible effects were considered.

One result of finite time thermodynamic analysis is the creation of a new thermodynamic criterion. Therefore, criteria such as dimensionless exergy destruction, dimensionless power, and ecological function, and so on can be mentioned. The ecological function can be calculated as a thermodynamic criterion of interest to researchers [45-51]. The ecological function is the interaction of power and production entropy. Two problems can be seen in the definition of ecological function. First, fuel cost was not considered; second, the size of heat exchangers was not considered. In addition, it represents the degree of depreciation. The ecological is achieved by subtracting the net power from the total exergy destruction (Eq. 21). The ecological function value of the gas turbine is -20.418 MW, a negative number that indicates significant environmental and power loss effects. It is therefore introduced and defined to solve two problems ecological - cost or ecologicost. The ecologicost value of the gas turbine is -0.05701.

$$Ecological=\dot{W}_{net}-T_0\dot{S}_{gen} \tag{47}$$

$$Ecologicost=(\dot{W}-T_0\dot{S}_{gen})/Q_{in} \tag{48}$$

As a new criterion, ecologicost first indicates that dissipated energy is more than production power; second, it shows 5.701% of input energy (low heat value of fuel) and primary fuel was not converted to useful energy or was wasted. Ecologicost has four terms: first term (production power); second term (dissipative power), third term (heat input to cycle related to fuel cost), in order to reduce fuel consumption), and fourth term (size of power plant).

6. Optimization of thermodynamic criteria

Optimization is one of the ways to improve thermodynamic systems. In this section, thermodynamic criteria such as power, energy efficiency, exergy efficiency, ecological function and ecologicost are optimized. Thermodynamic criteria are considered as objective functions. Decision variables include pressure ratio, turbine isentropic efficiency, compressor isentropic efficiency, and fuel-air ratio. Table 17 reveals the bounds used for decision variables. The basic constraint of the turbine inlet temperature (TIT) at the base load must not exceed 1070 ° C.

When analyzed a real system, only net power is considered, and rarely waste is considered, the ecological function and ecologicost reveal the real performance of the system with more detail. The choice of using the criterion of efficiency or net power depends on the fuel consumption cost, if the fuel consumption cost is high, it is recommended to use the energy and exergy efficiency criterion, but if the fuel consumption cost is low, net power is recommended. If fuel consumption cost is high, the use of energy and exergy efficiency criteria is recommended, but if fuel consumption cost are low, net power is recommended. However, the final decision maker is the employer to choose the criterion. In the case of ecological function, it shows the contrast between net power and loss power. In addition, it can be a measure of the performance, development and reduction of pollutants, in the new state, the system has some positive ecological function, and if depreciation, destruction and waste increase, it is negative. Now if the system conditions and fuel consumption of the system are targeted. Ecologicost function will be the criterion of success. Because it looks at three goals: net power, lost power and fuel consumption cost. The thing to note is that the optimization is based on December environmental conditions

The lower the ambient temperature and the greater the power output. It is due to the proportions of axial work with a specific volume. Optimization results shows in Table 18.

Table 17. The bounds for decision variables

Variable	**Range**
$\eta_{c,t}$	[0.8,0.91]
$\eta_{s,t}$	[0.8,0.85]
λ	[0.03264,10.02197]
PR	[9,14.59]

Table. 18. Optimization based on different criterion

Maximum net power								
$\eta_{c,t}$	$\eta_{s,t}$	λ	PR	η_{ex}	η_{en}	$Eco-\cos t$	$Power_{net}$	Eco
0.85	0.91	0.0327	9	0.483793	0.34	0.01695	136437.1485	6069.433
Maximum energy efficiency								
$\eta_{c,t}$	$\eta_{s,t}$	λ	PR	η_{ex}	η_{en}	$Eco-\cos t$	$Power_{net}$	Eco
0.85	0.909	0.0324	9.03	0.4826	0.38	0.01377	135928.86	4931.04765
Maximum ecologicost								
$\eta_{c,t}$	$\eta_{s,t}$	λ	PR	η_{ex}	η_{en}	$Eco-\cos t$	$Power_{net}$	Eco
0.85	0.91	0.0327	9	0.4838	0.343	0.01695	136437.1485	6069.433
Maximum ecological								
$\eta_{c,t}$	$\eta_{s,t}$	λ	PR	η_{ex}	η_{en}	$Eco-\cos t$	$Power_{net}$	Eco
0.85	0.91	0.0327	9	0.4838	0.343	0.01695	136437.1485	6069.433
Maximum exergy efficiency								
$\eta_{c,t}$	$\eta_{s,t}$	λ	PR	η_{ex}	η_{en}	$Eco-\cos t$	$Power_{net}$	Eco
0.85	0.91	0.0218	9	0.4952	0.368	-0.05049	131653.568	-18080.697

The results show that the following actions should be taken:

- Repairing or replacing combustion burners in order to achieve the optimum fuel-air ratio
- The increase of the turbine isentropic efficiency and isentropic compressor efficiency by replacing the damaged blades (with cracks, erosion, and corrosion)

The ecological and ecologicost function clearly shows the amount of depreciation. As the number of decision variables increases, the difference between the criteria becomes clearer. A system based on energy efficiency has the worst operating conditions.

7. Conclusion

In this work, the real gas turbine performance test, exergy assessment, and optimization were performed. This test was calculated based on two experimental, and academic methods. In the performance test, the operating variables and turbine inlet temperature are calculated. If gas turbine inlet temperature exceeds guaranteed temperature, metallurgical problems increases and the turbine blades damages, and if the temperature is below the guaranteed temperature, output power of the optimal conditions reduces. The results showed that the exergy efficiency of the turbine, compressor, and combustion chamber is 94.35%, 91.53%, and 77.165%, respectively. The results of exergy analysis show that the total destroyed exergy in gas turbine is about 142.588 MW, of which 79.38% is in the combustion chamber, 12.28 % in the turbine and 8.34% in the compressor. In this study, a new thermodynamic criterion called ecologicost was defined and applied. The ecologicost value of the gas turbine is -0.05701, which indicates the system is heading towards destruction, in other words, the amount of energy dissipated is high, and therefore, the gas turbine must be repaired and upgraded. Ecologicost is a thermodynamic criterion that has the ability to replace traditional thermodynamic criteria.

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