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Exergy Destruction Analysis of a Gas Turbine Power Plant

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ABSTRACT

According to the data released by "Republic of Turkey-the Ministry of Energy and Natural Resources", by the end of July 2017, 34% of the electricity of Turkey was produced from natural gas. As it is compared to the other resources such as coal (31%), hydraulic power (24%), wind (6%), geothermal energy (2%) and from other sources (3%), natural gas still occupies the highest place in electricity production.

The efficiency of the natural gas-fired power plants should be raised while the harmful effects of the exhaust gas emissions should be decreased. In this study, a natural gas-fired gas turbine power plant that produces electricity in a private factory in the city of Çorum-Turkey was analysed at increasing environment temperatures of -2.7 °C to 7.5 °C based on the exergy destruction.

The gas turbine data related to the operating conditions was provided by the private company. A computer code was improved with EES (Engineering Equation Solver) software to perform the exergy destruction analyses of the elements of the gas turbine cycle such as compressor, combustion chamber, turbine, boiler and economizer. At increasing environment temperatures of between -2.7 ° C and 7.5 ° C, it was found that the exergy destructions of the compressor, turbine, combustion chamber, boiler and economizer decreased. The maximum exergy destruction happened in the boiler and the minimum one happened in the combustion chamber.

Keywords:
Gas turbine; Second law; Exergy destruction; Exergetic efficiency

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INTRODUCTION

Nowadays, the importance of electricity generating systems by means of renewable energy such as wind and sun has been increasing because of decreasing the life of fossil fuels, but a large part of the need for electricity from fossil-based fuels is provided by power plants. In power generating plants, the emissions of power plants are considered as important parameters that must be taken into consideration. Interest in gas-fired gas turbine power plants has been increasing day by day because of low investment cost, efficient operation and minimal environmental impacts. It is not enough to analyse the thermal systems only in terms of the conservation of energy. Because the conservation of energy analyses systems only quantitatively in terms of energy balance. However, the second law of thermodynamics performs analysis of the thermal systems in terms of quality and enables the irreversibilities in the systems and elements of the systems to be found.

In a study , energy and exergy analyses of natural gas fired gas turbine power plants were carried and the losses in the natural gas fired gas turbine power plants were determined (Rahim and Gündüz, 2013). Sürer (2003) examined the thermodynamic and economic analysis of a cogeneration system consisting of combined gas and steam turbines and the efficiencies of this are compared. Sevilgen (2004) investigated cogeneration systems using exergy-economic analysis methods.

In the other studies carried out in the literature, gas turbine cycles have been investigated in terms

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of the first and second laws of thermodynamics. Arpacı (2002) examined exergy analysis of natural gas cogeneration systems with different data. Gürer (1997), has prepared a master of science thesis that is based on the first and second laws of analyses of the gas turbine systems used in the industrial sector in Turkey.

Sue and Chuang (2004), carried out the engineering design and exergy analysis of gas turbine systems for power generation. They found that the raise in the gas temperature in the preheated gas turbine from 22.5 °C to 118 °C caused the exergetic efficiency by 0.06%. They also resulted that the efficiency decreased by 1 % as the inlet air temperature of the compressor was reduced from 10 ° C to 5 ° C.

Ozcan et al. (2014) performed the exergy and energy analysis of the elements of the chemical cycle-based trigeneration system and derived the equations for the exergy and energy efficiencies for these elements. The highest energy consuming devices are the air separation unit and the compression air unit while the gas turbine and fuel cell are the most power producing elements.

Turan and Aydın (2014) conducted an exergy analysis of the elements of the LM6000 gas turbine engine (high and low pressure compressors and turbines, combustion unit) and analysed these elements from an exergy-economic perspective. Exergy destruction, exergetic efficiency and exergy-economic equations are written for each component. As a result of exergy analysis, it was determined that the largest exergy destruction occurred in the combustion unit (25,91 MW) and the highest exergetic efficiency (97,4%) was obtained in the high pressure turbine. As a result of the study, the total exergetic efficiency and the exergy destruction of the system were 39 and 39.3 MW respectively. In addition, they determined that the gas turbine cycle element with the highest exergy-economic factor is the high-pressure turbine. The effect of different gas turbine cycle operating parameters such as on the energy performance of two different gas turbine cycles including “basic gas turbine cycle” and “intercooled gas turbine cycle” was investigated by (Kumari and Sanjay, 2015). It was resulted that the total exergy destruction of “intercooled gas turbine cycle” was less than that of basic gas turbine cycle by 4.42%.

Ersayin and Ozgener (2015) performed a case study in order to investigate the energy and exergy analyses of a combined cycle power plant operated by a private company in Turkey and determined the energy and exergetic efficiencies of the power plant as 56 % and 50.04 %, respectively. In a similar study performed by (Ibrahim et al., 2017), it was observed that the largest exergy destruction occurred in the combustion chamber, followed by the turbine and air compressor. The exergetic efficiencies of the compressor, turbine and combustion chamber are 94.9%, 92% and 67.5%, respectively.

As the studies mentioned above in the literature are examined, it has been found that there is no a detailed case study focusing to observe the effect of the environment air temperature on the exergy destruction of the main and auxiliary elements of a gas turbine cycle. In this study, the effect of the environment air temperature on the exergy destructions and exergetic efficiencies of the elements of the gas turbine cycle such as the compressor, combustion chamber, turbine, boiler and economizer were carried out with respect to the actual data obtained from the gas turbine power plant operating in the city of Corum, Turkey. A computer program was improved with of EES (Klein, 2017) software and the actual table values are used for specific values of enthalpy, entropy and exergy.

GAS TURBINE MODEL

The gas turbine model to be used in this study is based on the natural gas-fired gas turbine power plant operating in the city of Çorum, Turkey and producing electricity and heat. The schematic of this cycle is depicted in Fig. 1.

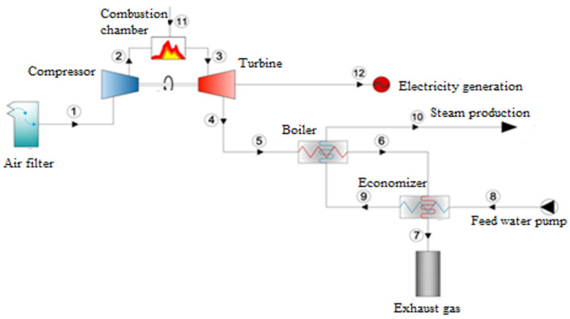

Figure 1. Gas turbine power plant

The environment air that is separated from foreign materials is sucked and compressed by the compressor from state 1 to state 2 and brought to the desired temperature and pressure conditions required for the combustion chamber. Natural gas is reacted with the environment air compressed by the compressor, and combustion products at high temperature flows to the gas turbine at state 3. The mechanical energy produced in the turbine is used to drive to the generator to produce electricity at state 12 and the exhaust gases simultaneously exhausted from the turbine at state 4. The exhaust gases enter the boiler at state 5 and leaves state 6, during this process, the steam is produced by the heat transfer between the exhaust gases and the feed water flowing throughout states 9 and 10. In the economizer, the heat is recovered from the exhaust gases flowing from state 6 to state 7 and the temperature of the feed water passing throughout state 8 and 9. At state 7, the exhaust gases exhausted to the atmosphere.

Energy equation can be written as (Van Wylen and Sonntag, 1985),

$$\begin{aligned}&\dot{Q}_{cv}+\sum\dot{m}_i\left(h_i+\frac{V_i^2}{2}+gZ_i\right)=\\&\frac{dE_{cv}}{dt}+\sum\dot{m}_e\left(h_e+\frac{V_e^2}{2}+gZ_e\right)+\dot{W}_{cv}\end{aligned} \tag{1}$$

The variation of the exergy with time for a control volume is expressed as (Moran and Shapiro, 2006),

$$\begin{aligned}&\frac{dE_{xcv}}{dt}=\sum\dot{m}_i e_{xi}-\sum\dot{m}_e e_{xe}+\sum\left(1-\frac{T_o}{T_s}\right)\dot{Q}_{cv}\\&-\left(\dot{W}_{cv}-P_0\frac{dV_{cv}}{dt}\right)-\dot{E}_D\end{aligned} \tag{2}$$

Where $\dot{E}_x$ is the exergy rate, $\frac{dE_{xcv}}{dt}$ is the variation rate for a control volume, $\frac{dE_{cv}}{dt}$ is the variation of volume of a control volume with time, $\dot{W}_{cv}$ is the power, $\dot{E}_D$ is the exergy destruction,

$\left(1-\frac{T_o}{T_s}\right)\dot{Q}_{cv}$ is exergy transfer related to the heat transfer, T_s is the boundary temperature, e_{xi}, and e_{xo} are specific exergies at the inlet and outlet, respectively.

The assumptions regarding to the gas turbine power plant are given below,

- Air standard assumptions are valid in the combustion chamber, that is, combustion process is replaced by the heat transfer from any source.
- It is assumed that the flow through the elements of gas turbine power plant such as compressor, combustion chamber, turbine, boiler and economizer is in accordance with the "Steady-State Steady Flow (SSSF)" model.
- Pressure losses in ducts and piping connecting combustion chamber, boiler, economizer and elements are neglected.
- Adiabatic compression of the air in the compressor and adiabatic expansion of the air in the turbine are assumed.
- It is also assumed that there was no heat loss from the pipes and ducts.
- The combustion chamber, boiler and economizer are considered to be insulated from the environment.
- It is assumed that the flow through the elements of the elements only includes the air.
- The variations in kinetic and potential energies are neglected in the gas turbine elements.

Compressor

In the compressor, air is adiabatically compressed and SSSF model is valid. In this case, the energy and exergy destruction equation for the compressor become

$$\dot{W}_c=\dot{m}_c\left(h_1+h_2\right) \tag{3}$$

$$\dot{E}_{D,c}=\dot{m}_c\left(e_{x1}-e_{x2}\right)-\dot{W}_c \tag{4}$$

where $\dot{m}_c$ is the mass flow rate of air flowing throughout the compressor, $\dot{W}_c$ is the power delivered to the compressor and $\dot{E}_{D,c}$ is the compressor exergy destruction.

The exergetic efficiency of the compressor is expressed as

$$\varepsilon_c=\frac{\dot{E}_{x1}-\dot{E}_{x2}}{\dot{W}_c}=\frac{\dot{m}_c\left(\left(h_1-h_2\right)-T_o\left(s_1-s_2\right)\right)}{\dot{W}_c} \tag{5}$$

where ε_c is the exergetic efficiency of the compressor.

Combustion Chamber

In the real case, the natural gas coming from the natural gas pipe line in the combustion chamber and the compressed air in the compressor chemically react and the gases which are formed as a result of the combustion leave the combustion chamber. In the gas turbine model, combustion chamber will be regarded as a heat exchanger through which air flows. The basic equations are written as,

$$\dot{Q}_{cc}=\dot{m}_{cc}\left(h_3-h_2\right) \tag{6}$$

$$\dot{E}_{D,cc}=\dot{m}_{cc}\left(e_{x2}-e_{x3}\right)+\left(1-\frac{T_o}{T_s}\right)\dot{Q}_{cc} \tag{7}$$

where $\dot{m}_{cc}$ is the mass flow rate of air flowing throughout the combustion chamber, $\dot{Q}_{cc}$ is the heat transfer rate to the combustion chamber and $\dot{E}_{D,cc}$ is the exergy destruction in the combustion chamber.

The exergetic efficiency of the combustion chamber, ε_{cc} can be expressed as

$$\varepsilon_{cc}=\frac{\left(1-\frac{T_o}{T_s}\right)\dot{Q}_{cc}}{\dot{m}_{cc}\left(e_{x3}-e_{x2}\right)}=\frac{\left(1-\frac{T_o}{T_s}\right)\dot{Q}_{cc}}{\dot{m}_{cc}\left(\left(h_3-h_2\right)-T_o\left(s_3-s_2\right)\right)} \tag{8}$$

Turbine

The mass flow of the fluid circulating in the turbine consists of the mass flow of the gases burning in the combustion chamber. It is assumed that the air expands instead of combustion gases in the expansion process in the turbine and the flow is in accordance with the SSSF model. The energy and exergy destruction equation for the turbine are expressed as follows,

$$\dot{W}_t = \dot{m}_{cc}\left(h_3 - h_4\right) \tag{9}$$

$$\dot{E}_{D,t} = \dot{m}_{cc}\left(e_{x3} - e_{x4}\right) - \dot{W}_t \tag{10}$$

where $\dot{W}_t$ is the turbine power and $\dot{E}_{D,t}$ is the turbine exergy destruction.

The exergetic efficiency of the turbine is expressed as

$$\varepsilon_t = \frac{\dot{W}_t}{\dot{E}_{x3} - \dot{E}_{x4}} = \frac{\dot{W}_t}{\dot{m}_{cc}\left(\left(h_3 - h_4\right) - T_o\left(s_3 - s_4\right)\right)} \tag{11}$$

where ε_t is the exergetic efficiency of the turbine.

Boiler

As can be seen in Fig. 2.1, the boiler is a heat exchanger in which the exhaust gases from the turbine cause the water that is preheated in the economizer to become vapor. It is assumed that the boiler is completely insulated against the environment. The energy balance and exergy destruction equations for the boiler are expressed as follows.

$$\dot{m}_{cc}\left(h_5 - h_6\right) = \dot{m}_w\left(h_{10} - h_9\right) \tag{12}$$

$$\dot{E}_{D,b} = \dot{m}_{cc}\left(e_{x5} - e_{x6}\right) = \dot{m}_w\left(e_{x9} - e_{x10}\right) \tag{13}$$

where $\dot{m}_w$ is the mass flow rate of water flowing throughout the boiler and $\dot{E}_{D,b}$ is the exergy destruction in the boiler. As the irreversibilities that are caused by the entropy flow and friction are taken into consideration, the exergetic efficiency of the boiler ε_b is expressed as

$$\varepsilon_b = \frac{\dot{m}_w\left(e_{x10} - e_{x9}\right)}{\dot{m}_{cc}\left(e_{x5} - e_{x6}\right)} = \frac{\dot{m}_w\left(\left(h_{10} - h_9\right) - T_o\left(s_{10} - s_9\right)\right)}{\dot{m}_{cc}\left(\left(h_5 - h_6\right) - T_o\left(s_5 - s_6\right)\right)} \tag{14}$$

Economizer

The economizer is a heat exvariationr in which exhaust gases from the turbine preheat the water before the water is passed to the boiler. It is assumed that the economizer is completely insulated from the environment like the boiler. The equation of energy balance and exergy for the economizer can be written as,

$$\dot{m}_t\left(h_6 - h_7\right) = \dot{m}_w\left(h_9 - h_8\right) \tag{15}$$

$$\dot{E}_{D,e} = \dot{m}_{cc}\left(e_{x6} - e_{x7}\right) = \dot{m}_w\left(e_{x8} - e_{x9}\right) \tag{16}$$

where $\dot{E}_{D,e}$ is the exergy destruction in the boiler. Same types of irreversibilities exist in the economizer. The exergetic efficiency of the economizer ε_e is expressed as

$$\varepsilon_e = \frac{\dot{m}_w\left(e_{x9} - e_{x8}\right)}{\dot{m}_{cc}\left(e_{x6} - e_{x7}\right)} = \frac{\dot{m}_w\left(\left(h_9 - h_8\right) - T_o\left(s_9 - s_8\right)\right)}{\dot{m}_{cc}\left(\left(h_6 - h_7\right) - T_o\left(s_6 - s_7\right)\right)} \tag{17}$$

Auxiliary Equations

The isentropic efficiencies of the compressor and turbine, the back work ratio and the net power can be expressed as,

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{18}$$

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{19}$$

$$BWR = \frac{\dot{W}_k}{\dot{W}_t} \tag{20}$$

$$\dot{W}_{net} = \dot{W}_t - \left|\dot{W}_c\right| \tag{21}$$

where $\eta_c, \eta_t, \dot{W}_{net}$ and BWR are the isentropic efficiencies of the compressor and turbine, net power produced by the gas turbine power plant and the back work ratio, respectively.

In terms of giving an idea about the type of data used in the analysis, the data that was obtained at an environment temperature of -1 °C in the actual operating conditions are depicted in Table 1. Table 2 shows the constant parameters used in the developed computer code.

Table 1. Operating conditions of the gas turbine power plant at -1 °C

T_1(°C)	T_3(°C)	T_4(°C)	T_5(°C)	T_6(°C)	T_7(°C)	T_8(°C)
-1	759.5	483.1	460	213	148	95
T_9(°C)	T_{10}(°C)	$\dot{W}_{net}(kW)$	P_1(Bar)	P_2(Bar)	P_1(Bar)	$\dot{m}_w(kg/s)$
184	192	7240	0.922	17.1	12	3.30

Table 2. Constant parameters in the computer code

η_c	η_t	BWR	P_{10}(kPa)	T_o(°C)
0,8	0,8	0,45	1200	25

UNCERTAINTY ANALYSIS OF THE GAS TURBINE POWER PLANT

The uncertainties of the measurements that are obtained from (Kilicarslan, 2004) and (Tore, 2016) are ±0.5 for temperature, ±3% for pressure, ±3% for mass flow rate, and ±2% for power. EES software was used to create the uncertainty progation table of the exergy destructions and exergetic efficiencies of the main and auxiliary elements as a function environment air temperatures ranging from -2.7 °C and 7.5 °C. The maximum and minimum uncertainties related to the exergy destructions and exergetic efficiencies of the main and auxiliary elements are depicted in Table 3.

Table 3. Uncertainties of exergy destructions and exergetic efficiencies

$\dot{E}_{D,c}$(%)	$\dot{E}_{D,cc}$(%)	$\dot{E}_{D,t}$(%)	$\dot{E}_{D,b}$(%)	$\dot{E}_{D,e}$(%)
±2.27	±9.92	±2.00	±4.42	±3.30
η_{ec}	η_{ecc}	η_{et}	η_{eb}	η_{ee}
±0,0009783	±0,001672	±0,00009653	±0,005504	±0,01702

RESULTS AND DISCUSSIONS

The analysis of the gas turbine power plant is based on the actual data collected from the gas turbine power plant located in the city of Çorum, Turkey. The exergy destructions and exergetic efficiencies of the main elements such as compressor, turbine, combustion chamber and those of auxiliary elements such as boiler, economizer are investigated by means of computer program developed using EES software. In order to show the validation of the model developed in this study, the present study is compared to the similar studies in the literature and Table 4 depicts the exergetic efficiencies of the compressor, combustion chamber and turbine of the present study and those obtained from (Ibrahim et al., 2017) and (Ersayin and Ozgener, 2015).

Table 4. Exergetic efficiency comparison of the main elements

	Ibrahim et al., 2017	*Ersayin and Ozgener, 2015*	*Present Study*	*Difference (%)*
η_{ec}	94.89	94.9	91	- 4.27
η_{ecc}	67.49	64	66.5	-1.48 / 3.75
η_{et}	91.96	81.7	89.5	-2.74 / 8.71

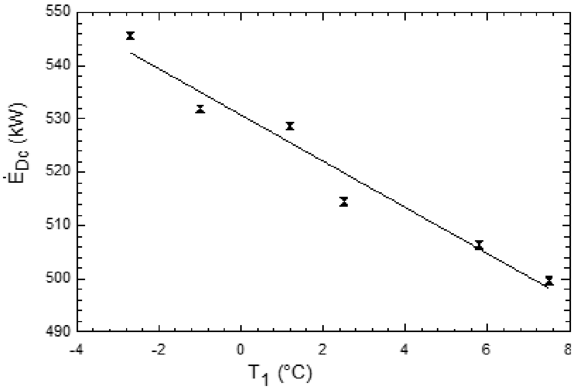

Figure 2. Compressor exergy destruction vs environment air temperature

Fig. 2 shows the compressor exergy destruction. As seen in Fig. 2, as the environment air temperature raises, the compressor exergy destruction decreases. The reason for the exergy destruction in the compressor is the entropy generation that occurs during compression of the fluid in the compressor. As mentioned earlier, at increasing environment air temperatures, entropy in the compressor decreases. As a result, the decrease in the entropy production causes the compressor exergy destruction to decrease. At increasing environment air temperatures of -2.7 ° C to 7.5 ° C, the compressor exergy destruction decreased from 545.6 kW to 499.6 kW

The variation of the exergy destruction of the combustion chamber with the environment air temperature is depicted in Fig. 3. As the environment air temperature raises, the exergy destruction of the combustion chamber decreases as seen in Fig. 3. When the environment air temperature raises, the temperature difference between the combustion chamber and the environment air decreases. This causes the amount of heat given to the combustion chamber to decrease and thereby decreasing the amount of exergy destruction in the combustion chamber as seen in Fig. 3. Fig. 3 also depicts that the exergy destruction of the combustion chamber varies between 51 kW and 38 kW. It is reduced by 11%. In the actual operating conditions of a gas turbine power plant, the combustion chamber exergy destruction occupies the largest part of exergy destruction as it is compared to the other elements such as compressor, turbine, boiler and economizer because of higher values of chemical exergy. But, the chemical exergy is not taken into consideration in this study because the air standard assumptions are assumed.

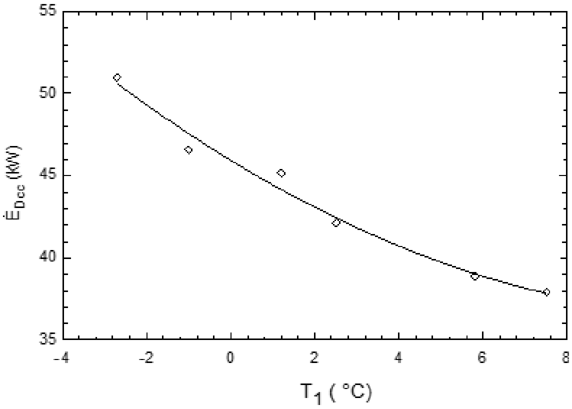

Figure 3. Combustion chamber exergy destruction vs environment air temperature

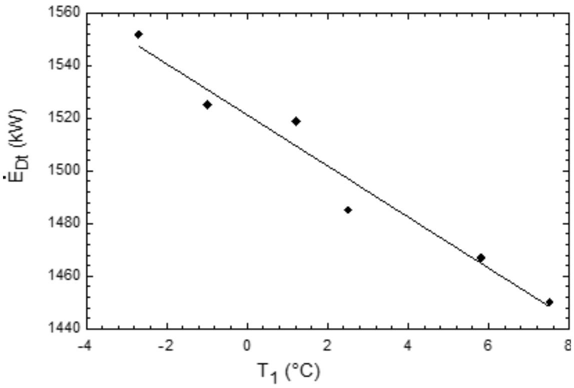

Figure 4. Turbine exergy destruction vs environment air temperature

The variation in exergy destruction of the turbine as a function of the environment air temperature is depicted in Fig. 4. As the environment air temperature raises, the exergy destruction of the turbine decreases. It was mentioned earlier that the main reasons of the entropy generation in the turbine are the sudden expansion of the fluid and friction. The production of entropy in the turbine decreases at increasing environment temperatures. This causes the exergy destruction in the turbine to decrease as depicted in Fig. 4. The maximum exergy destruction is 1552 kW at -2.7 ° C while the minimum is 1450 kW at 7.5 ° C

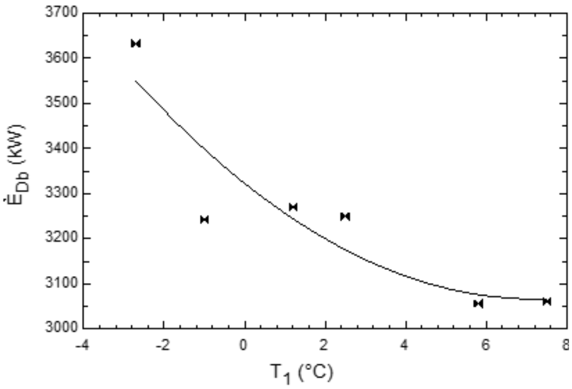

Figure 5. Boiler exergy destruction vs environment air temperature

Fig. 5 shows the variation in exergy destruction of the boiler according to the environment air temperature. As can be seen from Fig. 5, the exergy destruction in the boiler averagely decreases as the environment air temperature raises. The main reason for the exergy destruction in the boiler is the entropy flows of water and air. At -2.7 °C, the maximum exergy destruction occurs as 3632 kW, and then the exergy destruction suddenly decreased to around 3254 kW at the environment temperatures between -1° C and 2.5° C. Finally, the average exergy destruction of the boiler is 3059 kW as minimum between the environment temperature of 5.8 ° C and 7.5 ° C is. At increasing environment air temperatures between -2.7 ° C and 7.5 ° C, the exergy destruction in the boiler decreases averagely % 15.7.

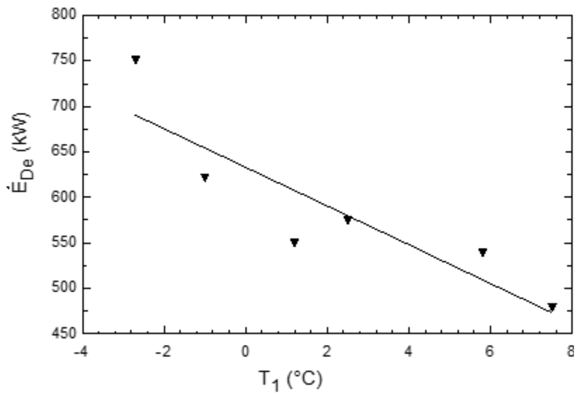

Figure 6. Economizer exergy destruction vs environment air temperature

The variation in the exergy destruction of the economizer as a function of the environment air temperature is depicted in Fig. 6. As the environment air temperature raises, the irreversibility of the economizer decreases. Entropy flows of water and air only cause the exergy destruction in the economizer because the economizer is insulated from the environment like the boiler. The exergy destruction values in the economizer are lower than those in the boiler because the average temperatures the fluids during flow in the economizer are lower than those in the boiler. so the irreversibilities that occur are also smaller. As the environment air temperature raises from -2.7 °C to 7.5 °C, the exergy destruction of the economizer decreases between 750.9 kW and 478.9 kW. At the above-mentioned environment temperatures, the irreversibility of the economizer averagely decreased % 36.

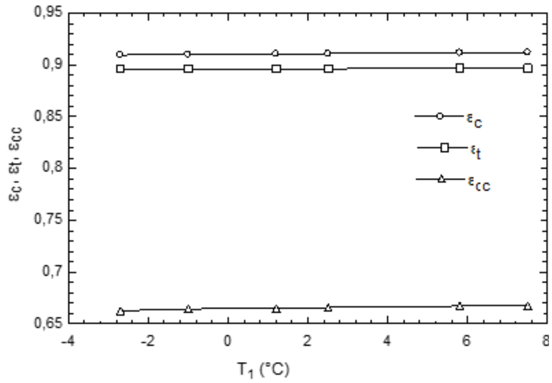

Figure 7. Exergetic efficiencies of the main elements vs environment air temperature

The variation in the exergetic efficiency of the main elements of the gas turbine power plant such as compressor, turbine and combustion chamber as a function of the environment air temperature is depicted in Fig. 7. The highest exergetic efficiency occurred in the compressor with 91%, followed by the turbine with 89% and the lowest exergetic efficiency occurred with 66% in the combustion chamber. The variation in the exergetic efficiencies of the compressor, turbine and combustion chamber are almost neglected at increasing environment temperatures of between -2.7 ° C and 7.5 ° C.

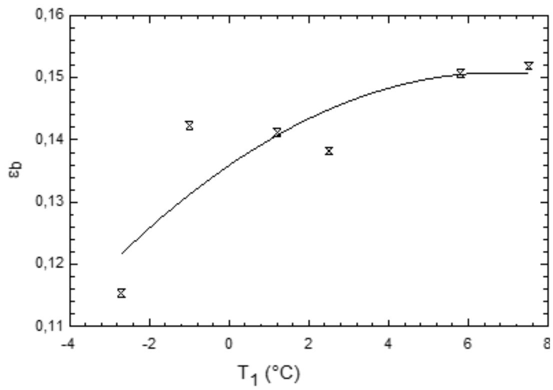

Figure 8. Boiler exergetic efficiency vs environment air temperature

Fig. 8 depicts the variation of the exergetic efficiency of the boiler as a function of the environment air temperature. As the environment air temperature raises, the exergetic efficiency of the boiler raises. At increasing environment air temperatures between -2.7 °C and 7.5 °C, the decrease in the exergy destruction of the boiler causes the exergetic efficiency of the boiler to raise. The exergetic efficiency of the boiler is 11.5 % at -2.7 °C and it is 15.2% at 7.5 ° C.

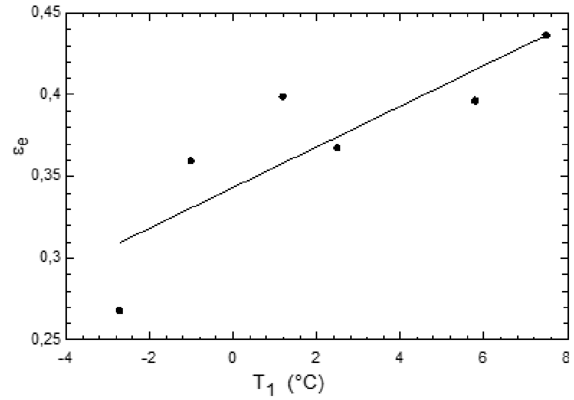

Figure 9. Economizer exergetic efficiency vs environment air temperature

Fig. 9 shows the variation in the exergetic efficiency of the economizer as a function of the environment air temperature. As the environment air temperature raises, the exergetic efficiency of the economizer averagely raises. At increasing environment air temperatures between -2.7 °C and 7.5 °C, the exergetic efficiency of the economizer variations between 26.7% and 43.61%. A decrease in the exergy destruction of the economizer causes the exergetic efficiency of the economizer to raise by 36 %. As depicted in Figure 9. The exergetic efficiency of the economizer, which tends to raise smoothly at the environment air temperatures between -2.7 °C and 1.2 °C, decreases between 1.2 ° C and 5.8 °C and then raises to its maximum at a temperature of 7.5 °C.

CONCLUSIONS

A real time data including temperature, pressure, flow rate and net power at the environment air temperatures between -2.7 °C and 7.5 °C was collected from the gas turbine power plant generating electricity and waste heat, located in Çorum-Turkey. According to the real data, the exergy destructions and exergetic efficiencies of the main and auxiliary elements were carried out with respect to the environment air temperatures.

At increasing environment air temperatures between -2.7 °C and 7.5 °C, the exergy destruction values in the compressor, turbine and combustion chamber decreased. The exergy destruction decreased from 545.6 kW to 499.6 kW for the compressor, from 51 kW and 38 kW for the combustion chamber, from 1552 kW to 1450 kW for the turbine, from 3632 kW to 3254 kW for the boiler and from 750.9 kW to 478.9 kW for the economizer at the environment air temperatures studied in this experimental work. Maximum exergy destruction occurred at -2.7 °C and minimum one occurred at 7.5 °C occurred in each of the main elements of the gas turbine power plant. At increasing environment air temperatures between -2.7 °C and 7.5 °C, the maximum exergy destruction occurred in the boiler as 3652 kW while the minimum one occurred in the combustion chamber as 37.9 kW. As it was mentioned before, the maximum exergy destruction occurs in the combustion chamber in an actual gas turbine power plant where the chemical exergies of the reactants and exhaust products are taken into consideration, but the air standard assumptions are presumed in this work.

At increasing environment temperatures between -2.7 °C and 7.5 °C, the exergetic efficiencies of the main and auxiliary elements mainly raised because of decreasing the exergy destructions. The variation in the exergetic efficiency of the main elements such as the compressor, turbine and combustion chamber can be neglected. The maximum exergetic efficiency was observed as 91% in the compressor, followed by as 89% in the turbine and 66% in the combustion chamber. Exergetic efficiency of the economizer ranged from 26% to 43% while that of the boiler ranged from 11.5% and 15%.

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NOMENCLATURE

BWR	: Back work ratio
e_x	: specific exergy (kj/kg)
$\dot{E}_x$	: Exergy rate (kW)
$\dot{E}_D$	: Exergy destruction rate (kW)
$\dot{m}$	: Mass flow rate (kg/s)
ε	: Exergetic efficiency
η_c	: Compressor isentropic efficiency
η_t	: Turbine isentropic efficiency
P_0	: Atmospheric pressure (kPa)
$\dot{Q}$	: Heat transfer (kW)
T_s	: Boundary temperature (K)
$\dot{W}$	: Power (kW)

Subscript

b	: Boiler
c	: Compressor
cc	: Combustion chamber
cv	: Control volume
e	: Economizer
i	: Inlet
o	: Outlet
t	: Turbine
w	: Water

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