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Energy and Exergy Analysis of ALkhoms Gas Turbine Power Plant Under Part Load Operating Conditions

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تحليل الطاقة والإكسرجي لمحطة توليد القدرة بالتوربينات الغازية بمدينة الخمس تحت ظروف التشغيل عند الأحمال الجزئية

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Abstract
The gradual increase in electricity demand has resulted in the expansion of power stations. Such an expansion centered on building new power plants and revamping the existing ones to improve efficiency. The objective of this study is to perform an exergy analysis on the Khoms Gas Turbine Power Plant (KGTPP), which utilizes two different types of fuel: light diesel and natural gas at different loads. Here, mass and energy conservation were applied to the power station components (gas turbine, combustion chamber, and compressor) with different loads and pressure ratios. The results showed that when natural gas was used, the largest exergy destruction of up to 290.86 MW occurred in the combustion chamber. Meanwhile, the lowest exergy destruction of 9.1 MW occurred in the compressor. According to the first law, the thermal efficiency of the gas turbine power plant was 34.3%, while exergy efficiency was 32.5%. For the light fuel power station, results showed that the largest exergy destruction of up to 219.33 MW occurred in the combustion chamber. Meanwhile, the lowest exergy occurred in the compressor at 5.68 MW. According to the first law, the thermal efficiency of the gas turbine power plant was 36.8% and exergy efficiency was 35%. The results demonstrated a significant correlation between changes in the power plant load and the overall energy and exergy efficiencies, as well as exergy destruction.

Keywords: Gas Turbine, Exergy, Exergy Destruction, part load, Exergy Analysis.

الملخص
أدى التزايد التدريجي في الطلب على الكهرباء إلى التوسع في محطات توليد الطاقة. وقد تركز هذا التوسع على إنشاء محطات جديدة وتطوير المحطات القائمة بهدف تحسين الكفاءة. تهدف هذه الدراسة إلى إجراء تحليل إكسرجي لمحطة الخمس للتوربينات الغازية ، والتي تستخدم نوعين مختلفين من الوقود هما الديزل الخفيف والغاز الطبيعي عند أحمال تشغيل مختلفة. وفي هذه الدراسة، تم تطبيق معادلات حفظ الكتلة والطاقة على مكونات محطة القدرة (التوربين الغازي، غرفة الاحتراق، والضاغط) عند أحمال تشغيل ونسب ضغط مختلفة. أظهرت النتائج أنه عند استخدام الغاز الطبيعي، حدث أكبر تدمير للإكسرجي في غرفة الاحتراق وبلغ نحو 290.86 ميجاواط، في حين سُجل أقل تدمير للإكسرجي في الضاغط وبلغ 9.1 ميجاواط. ووفقًا للقانون الأول للديناميكا الحرارية، بلغت الكفاءة الحرارية لمحطة التوربينات الغازية 34.3%، بينما بلغت الكفاءة الإكسرجية 32.5%. أما عند استخدام الوقود الخفيف، فقد أظهرت النتائج أن أكبر تدمير للإكسرجي حدث أيضًا في غرفة الاحتراق وبلغ نحو 219.33 ميجاواط، في حين سُجل أقل تدمير للإكسرجي في الضاغط وبلغ 5.68 ميجاواط. ووفقًا للقانون الأول للديناميكا الحرارية، بلغت الكفاءة الحرارية لمحطة التوربينات الغازية 36.8%، بينما بلغت الكفاءة الإكسرجية 35%.. كما أوضحت النتائج وجود علاقة واضحة بين تغيرات حمل المحطة وكلٍّ من الكفاءة الكلية للطاقة، والكفاءة الإكسرجية، ومقدار تدمير الإكسرجي.

الكلمات المفتاحية : التوربين الغازي، الإكسرجي، تدمير الإكسرجي، الأحمال الجزئية، تحليل الإكسرجي.

Introduction

The power generated by power plants should constantly align with the demand for electricity. Due to the fluctuations in electricity demand over the course of a day, power plants are occasionally required to operate at part loads. This part-load operation can lead to a reduction in the plant's efficiency. Hence, it is crucial to take proactive measures to minimize this decline in efficiency whenever possible. To do so, the utilization of energy and exergy analyses can be highly advantageous. Exergy analysis is a method of evaluating the performance of devices and processes and involves examining the exergy at different points in a series of mass and energy conversion steps [1]. It overcomes the restrictions of energy analysis and develops the work of power stations [2–7]. Furthermore, conducting exergy analysis on power generation systems is essential for the optimal utilization of energy resources. Thus, in the last ten to fifteen years, researchers have focused on component exergy analysis. At the same time, improving the efficiency of power stations depends on the operating conditions of these stations. Consequently, gaining an exact understanding of the impact of working conditions on the efficiency of power stations can help in maintaining a high-efficiency operating system, as evidenced by continuous energy and exergy analyses in recent works [8–10].

Many researchers have examined and conducted analyses involving energy and exergy in thermal power plants. In their study, Kopac and Hilalci [11] examined how variations in ambient temperature impacted the efficiency of the Catalagzi power plant in Turkey, which utilizes a regenerative and reheat system. Their findings indicated a significant correlation between boiler efficiency and the overall effectiveness of the plant. Moreover, the ambient temperature was observed to exert a notable influence on the irreversibility of the boiler. Pattanayak et al. [12] conducted an assessment of a combined cycle power plant (CCPP) incorporating reheat through both energy and exergy analyses. Their study explored various operating parameters, including excess air percentage, ambient temperature, exhaust pressure loss, different loads, and cooling water temperature. It was observed that as inlet and exhaust pressure losses increased, the CCPP's efficiency decreased. Additionally, higher inlet air temperatures resulted in a reduction in its overall power output. Aljundi [13] conducted comprehensive energy and exergy analyses on the Al-Hussein Power Plant in Jordan. The primary objective was to evaluate individual system components and pinpoint areas with the highest energy and exergy losses. The results revealed that the condenser accounted for the largest energy loss, dissipating 66% of the input energy to the environment. Additionally, the boiler exhibited the most substantial exergy destruction, with 77% of the fuel exergy input being lost. Another group of studies [14] compared the performances of different components in terms of exergy analysis to identify the sites with the largest losses and exergy destruction. They found that the combustion chamber and boiler have most of the exergy destruction. Ibrahim et al. [15] conducted a thermodynamic assessment of a simulated model for a combined cycle gas turbine. Their study incorporated data on the impact of cycle peak temperature ratio and peak compression ratio of the gas turbine. Their findings indicated that increasing the cycle peak temperature ratio and reducing the compression ratio led to enhancements in both total power output and overall thermal efficiencies. In a separate study, Martin et al. [16] characterized the performance of a 20 MW gas turbine power plant in Indonesia using exergy analysis. Their results highlighted that the combustion chamber accounted for the highest exergy destruction (71%), while the compressor exhibited the lowest exergy destruction (12%). Additionally, Maruf et al. [17] conducted an efficiency analysis encompassing both energy and exergy, utilizing data collected from various power sectors in Bangladesh. Their investigation revealed that hydropower plants exhibited higher exergy efficiency compared to thermal power plants.

The Libyan energy and electricity sector has faced numerous challenges, with the most significant ones including scarce natural resources, surging energy demand, and political instability within the nation. Despite having over 15 power generation facilities distributed across the country, only a handful have undergone thermodynamic analysis. Sassi et al. 2018 [18] conducted an exergy analysis of a South Tripoli– Libya gas turbine power plant. Their results showed that the largest exergy destruction occurred in the combustion chamber followed by the gas turbine, while the lowest amount occurred in the compressor. Elfeituri et al. [19] conducted energy and exergy analyses on a 65 MW Derna-Libya steam power plant at part load conditions. Their findings indicated that the primary source of irreversibility within the power plant was located in the steam generator, both at full and partial loads. Additionally, they observed that as the load decreased, the overall exergy efficiency decreased, accompanied by an increase in the total exergy destruction ratio. Fellah et al. [20] collected design and real-time data in order to perform exergy analysis of unit three of the Zawia-Libya CCPP. They also assessed the exergy efficiencies and the exergy destruction rates for each component. The study showed that the major exergy destructions occurred in the combustion chamber. For design and real data, the exergy destructions were 356.34 and 556.28MW, respectively, while exergy efficiencies were 54.88% and 43.75%, respectively. The same author, Fellah [21] reveals that variations in ambient temperature significantly affect the performance of combined cycle power plants in Libya. As ambient temperatures rise, the density and mass flow rate of air at the compressor entrance decrease, leading to a reduction in the net power and efficiency of the gas turbine unit. However, utilizing exhaust gases to operate a steam turbine power cycle enhances the overall performance of the combined cycle. For instance, at an ambient temperature of 15°C, first and second law efficiencies increase to 42.80% and 40.20%,

respectively, with a 53.46% gain in power. Conversely, at an ambient temperature of 40°C, efficiencies decrease to 41.11% and 38.60%, but the power gain rises to 60.98%.
In the current study, the combination of the first and second law of thermodynamics to describe exergy and to illustrate its use as a tool to improve efficiency has been implemented on KGTPP. The power plant utilizes a hybrid fuel system, primarily relying on Natural Gas (NG) as the primary source, with the option of using diesel fuel (DF) or fuel oil (FO) as secondary sources when needed. However, a literature survey has shown that the effects of part-load operating conditions and the type of fuel on the performance of the power plant have yet to be widely investigated. Therefore, the objective of this paper is to evaluate the impacts of part-load operating conditions with different fuels on the performance of each component of the KGTPP using exergy-based performance analysis.

Process Description

The KGTPP is located on the Mediterranean Sea at Khoms, Libya. It is operated by the General Electricity Company of Libya (GECOL). The gas turbine cycle associated with KGTPP has been in operation since 1992. Currently, KGTPP comprises four units, with each unit having a net output of 120 MW. The gas turbine manufacturers provide performance specifications based on ISO conditions, which include an ambient air temperature of 15°C, atmospheric pressure of 101.325 kPa, and a relative humidity of 60%. The plant was divided into different control volumes. The only main components that strongly influence the operation of the system under the study are considered. They are the compressor, combustor, and turbine, as shown in Figure 1. The analyzed operational data included the inlet temperature of the compressor, the inlet temperature of the turbine, and the pressure ratio of the compressor.

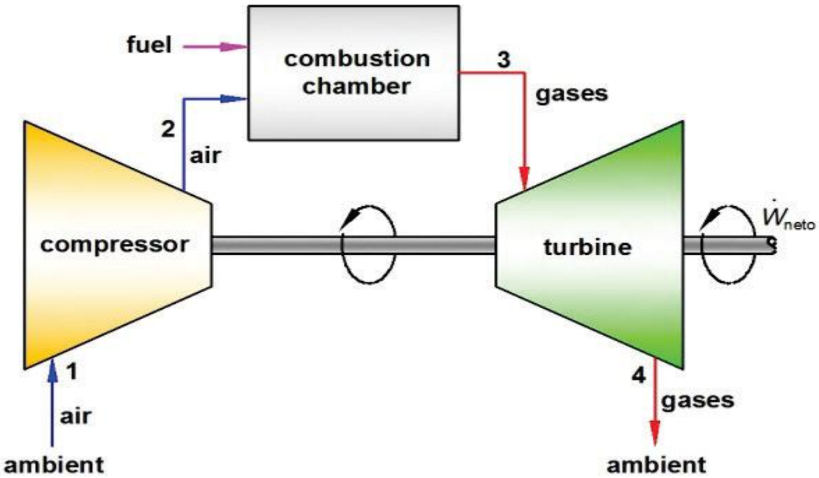

Figure 1: Gas Turbine Power Plant

Two different fuel compositions, light fuel and natural gas, were employed in the gas turbine power station. Tables 1 and 2 present the weight and mole fraction compositions of light fuel and natural gas, respectively. The working medium, typically air, undergoes pressurization in the compressor, which is powered by the turbine. As it passes through the combustor, its temperature increases due to combustion with fuel. The subsequent expansion of the hot combustion gases generates an adequate power output from the turbine, enabling it to supply useful work in addition to driving the compressor.

Table 1: Analysis weight fraction of light fuel

Component	*Weight %*
$C_9 H_{20}$	2.463
$C_{10} H_{22}$	3.626
$C_{11} H_{24}$	4.431
$C_{12} H_{26}$	9.632
$C_{13} H_{28}$	18.149
$C_{14} H_{30}$	16.623
$C_{15} H_{32}$	12.762
$C_{16} H_{34}$	8.969
$C_{17} H_{36}$	6.841
$C_{18} H_{38}$	7.282
$C_{19} H_{40}$	4.723
$C_{20} H_{42}$	2.68
$C_{21} H_{44}$	1.214
$C_{22} H_{46}$	0.6

Table 2: Analysis mole fraction of natural gas

Component	*Mole %*
N_2	0.53
CO_2	2.72
CH_4	83.14
$C_2 H_6$	9.06
$C_3 H_8$	2.67
$C_4 H_{10}$	1.19
$C_5 H_{12}$	0.44
$C_6 H_{14}$	0.25

Exergy Balance Formulations

Exergy analysis involves the evaluation of inputs, outputs, accumulation, and consumption of exergy for a system. These terms are related as follows:

Exergy input – Exergy output – Exergy consumption = Exergy accumulation

The exergy of flow in stream is given by the sum of the physical, chemical, kinetic, and potential exergies, which is calculated as follows:

$$E = E_{ph} + E_{ch} + E_{ke} + E_p \quad (1)$$

And the specific exergy rate is

$$\varepsilon = \varepsilon_{ph} + \varepsilon_{ch} + \varepsilon_{ke} + \varepsilon_p \quad (2)$$

The rate from of general exergy balance equation for a steady state single-stream system described as:

$$\sum (1 - \frac{T_o}{T_k})\dot{Q}_k - \dot{w} + \dot{m}(\varepsilon_i - \varepsilon_e) - I = 0 \quad (3)$$

The first three terms of equation represent the exergy transfer by heat, work, and by mass respectively, while the last term represents the irreversibility "exergy destroyed", and for ideal gas:

$$\varepsilon_{ph} = (h - h_o) - T_o \left[s - s_o - R\,ln({}^{P}/_{P_o}) \right] \quad (4)$$

Where h and s are enthalpy and entropy of the substance, h_o and s_o are those at standard conditions, and T_o, P_o are the temperature and pressure of the standard conditions.

The exergetic efficiency of a system defined as:

$$\eta_{ex} = \frac{Exergy\ recovered}{Exergy\ supplied} = 1 - \frac{Exergy\ destroyed}{Exergy\ suplied}$$

Exergy analysis on compressor

The exergy balance is:

$$\Delta\varepsilon = \varepsilon_e - \varepsilon_i \quad (5)$$

$$\varepsilon_i = (h_i - h_o) - T_o(s_i - s_o) \quad (6)$$

$$\varepsilon_e = (h_e - h_o - T_o) \left[s_e - s_o - R\,ln({}^{P}/_{P_o}) \right] \quad (7)$$

$$W_C = \dot{m}_{air}(h_e - h_i) \quad (8)$$

Irreversibility (Exergy destruction):

$$I = \varepsilon_i - \varepsilon_e + W_{cv} \quad (9)$$

The exergetic efficiency is:

$$\eta_{ex} = \frac{\varepsilon_e - \varepsilon_i}{|W_i|} * 100 \quad (10)$$

Exergy analysis on combustor

The exergy balance:

$$\Delta\varepsilon = \varepsilon_{i,air} + \varepsilon_f - \varepsilon_{e,fg} \quad (11)$$

$$\varepsilon_{i,air} = (h_{i,air} - h_o) - T_o \left[s_{i,air} - s_o - R\,ln({}^{P}/_{P_o}) \right] \quad (12)$$

$$\varepsilon_{e,fg} = (h_{e,fg} - h_{o,fg}) - T_o \left[(s_{e,fg} - s_{o,fg}) - R\,ln({}^{P}/_{P_o}) \right] \quad (13)$$

Irreversibility (Exergy destruction):

$$I = \dot{m}_f\varepsilon_f + \dot{m}\varepsilon_{i,air} - \dot{m}_{e,fg}\varepsilon_{e,fg} \quad (14)$$

The specific chemical exergy of hydrocarbon fuel can be determined using the following formula, which represents the ratio of fuel exergy to the lower heating value of the fuel.

$$\varphi = \frac{\varepsilon_f}{LHV}$$

The ratio φ represents the exergy factor based on the lower heating value and determined by the following relationship:

$$\varphi = 1.033 + 0.0169\frac{b}{a} - \frac{0.0698}{a}$$

The exergetic efficiency is:

$$\eta_{ex} = \frac{\dot{m}_{e,fg}\varepsilon_{e,fg}}{\dot{m}_f\varepsilon_f} \quad (15)$$

Exergy analysis on gas turbine

$$\Delta\varepsilon = \varepsilon_{i,fg} - \varepsilon_{e,fg} \tag{16}$$

$$\varepsilon_i = (h_i - h_o) - T_o\left[s_i - s_o - R\ln(P/P_o)\right] \tag{17}$$

$$\varepsilon_{e,fg} = (h_{e,fg} - h_{o,fg}) - T_o\left[s_{e,fg} - s_{o,fg} - R\ln(P/P_o)\right] \tag{18}$$

$$W_{GT} = (\dot{m}_f + \dot{m}_{air})(h_i - h_e) \tag{19}$$

Irreversibility (Exergy destruction):

(20)

The exergetic efficiency is:

$$\eta_{ex} = \frac{W_{GT}}{\varepsilon_{i,fg} - \varepsilon_{e,fg}} \tag{21}$$

The exergy and thermal efficiencies of power plant expressed as:

$$\eta_{ex} = \frac{W_{out}}{E_i} \qquad \eta_{th} = \frac{W_{out}}{\dot{m}Q}$$

Exergy analysis applied for each component of the power plants with various load conditions.

Results and Discussion

Exergy analysis is an approach that combines the principles of energy conservation (as per the first law of thermodynamics) and the non-conservation of entropy (as per the second law) to analysis, design, and enhance energy systems. This study applies exergy analysis to optimize the design of (KGTPP), which utilizes both light fuel and natural gas, under different loads (50%, 75%, 90%, and full load) while operating at constant pressure.

Table 3: Exergy analysis of KGTPP components at different loads with light fuel

Components	ε_i *(Mw)*	ε_e *(Mw)*	*I (Mw)*	η_{ex}
		Full load		
Compressor		126.64	5.688	95.7
Combustor	534.18	358.8	219.33	88
Gas turbine	358.8	83.72	73.63	78.9
		90% load		
Compressor	0	124.089	17.7	87.5
Combustor	507.46	300.4	210.03	74.7
Gas turbine	300.4	71.67	71.85	77.1
		75% load		
Compressor	0	115.25	18.5	89.1
Combustor	478.98	270.34	208.6	74.3
Gas turbine	270.34	53.92	62.6	76.1
		50% load		
Compressor	0	113.65	47	88.1
Combustor	389.78	191.8	197.96	64.3
Gas turbine	191.8	33.65	52.8	74.1

Table 3 summarizes the exergy analysis results for the main equipment of the (KGTPP) components when the light fuel is used. Observing the data, it is evident that the greatest exergy destruction in the combustor remained consistent across all four loads, with values of 198 MW, 209 MW, 210 MW, and 219 MW for 50%, 75%, 90%, and full load, respectively. Subsequently, the exergy destruction in the gas turbine exhibited values of 53 MW, 63 MW, 72 MW, and 74 MW for all loads, respectively. The compressor consistently records the lowest exergy destruction values for all loads.

The table also illustrates the plant components' second law efficiency percentages, which are referred to in this study as exergetic efficiency percentages. Among the three plant components, combustor exhibited the lowest exergetic efficiency with values of 64%, 74%, 75%, and 78% for 50%, 75%, 90%, and full load, respectively. The lowest exergetic efficiency and largest exergy destruction rate occur in the combustion chamber can be attributed to some factors such as unburnt fuels, and heat losses to the surrounding during combustion. It is clearly seen that, as the power load decreases, the exergetic efficiency of combustor and gas turbine often decreases. This is because these components are designed to operate most efficiently at or near their rated or full-load conditions.

Table 4: Exergy analysis of KGTPP components at different loads with natural gas

Components	$\mathcal{E}_i$ ***(Mw)***	$\mathcal{E}_e$ ***(Mw)***	***I (Mw)***	η_{ex}
		Full load		
Compressor	0	131.52	9.1	93.5
Combustor	653.8	363.00	290.86	69.5
Gas turbine	363	85.62	76.25	77.89
		90% load		
Compressor	0	119.38	10.16	92.1
Combustor	600.86	317	283.95	65.8
Gas turbine	317	71.06	70.6	77.0
		75% load		
Compressor	0	99.51	12.067	92.9
Combustor	506.19	236.6	269.56	58.18
Gas turbine	236.6	47.39	58.97	75.5
		50% load		
Compressor	0	78.3	15.91	92.9
Combustor	408.77	166.83	241.94	50.18
Gas turbine	166.83	29.26	47.19	73.6

Table 4 presents the exergy analysis results when natural gas is employed as a fuel. Notably, the highest exergy destruction was observed in the combustor during full load operation, exceeding 290 MW, with an exergetic efficiency of about 70 %. The huge exergy destruction in the combustor can be attributed to the huge amount of entropy generated in this equipment, which in turn, is due to the massive heat liberated in the combustor hollow. Additionally, the gas turbine exhibited an exergy destruction of 76 MW, with an exergetic efficiency of approximately 78%. Conversely, the compressor showed the lowest exergy destruction, totaling 9 MW at full load, while achieving an exergetic efficiency of about 94%. As the power load decreases, the exergy destruction values for both the combustor and gas turbine also decrease. Whereas, when the power plant operates at partial load, the compressor may not be operating in its optimal range, leading to increase in its exergy destruction. This is because compressors are typically designed for specific flow rates and pressures, and variations from these conditions can affect their performance. However, the exergy destruction values in the compressor are relatively insignificant when compared to those in the combustor. It's worth noting that this outcome aligns with the results reported by Saad Alrwashdeh [8].

In summary, based on the outcomes provided in Tables 3 and 4, both fuels consistently demonstrated a similar way regarding the influence of power load on energy efficiency and exergy destruction. Furthermore, it has been observed that the exergetic efficiency of the power plant using natural gas is lower compared to that when using light fuel oil. This could be attributed to the use of a higher air-to-fuel ratio than the optimal ratio, which is approximately 10:1. It is known that, the operating with high excess air can reduce the overall efficiency of the combustion process. This is because a portion of the energy in the fuel is used to heat and dilute the excess air, which does not contribute to useful work output. As a result, the power plant may generate less electricity or mechanical work for a given amount of fuel. Also, in some cases, excessively high air-fuel ratios can lead to incomplete combustion, resulting in unburned fuel in the exhaust. This not only wastes fuel but can also pose safety hazards. In contrast, excess air can reduce NOx emissions, it may increase emissions of other pollutants such as carbon monoxide (CO) and unburned hydrocarbons. Achieving the right balance between emissions reduction and efficiency is crucial.

Exergetic efficiency are thoroughly presented in Figures 2 and 3 of the power plant for light fuel and natural gas at various loads, respectively. As observed in both figures, when the load is decreased, both efficiencies show a slight decrease. It is clear that exergetic efficiency is consistently lower than thermal efficiency, primarily because of some energy is inevitably lost as a result of irreversibilities, such as heat transfer to the surroundings, fluid friction, and imperfect combustion. These losses are accounted for in exergy analysis but are often neglected in thermal efficiency calculations. In general, exergetic efficiency takes into account the quality of energy and the losses associated with real-world processes, which typically result in a lower efficiency value compared to thermal efficiency.

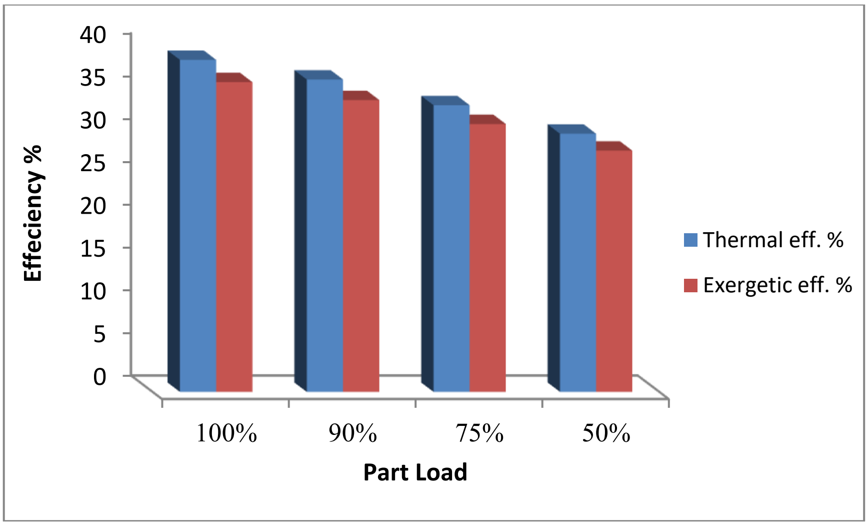

Figure 2: Thermal and exergetic efficiency at different loads for the power plant operated with light fuels.

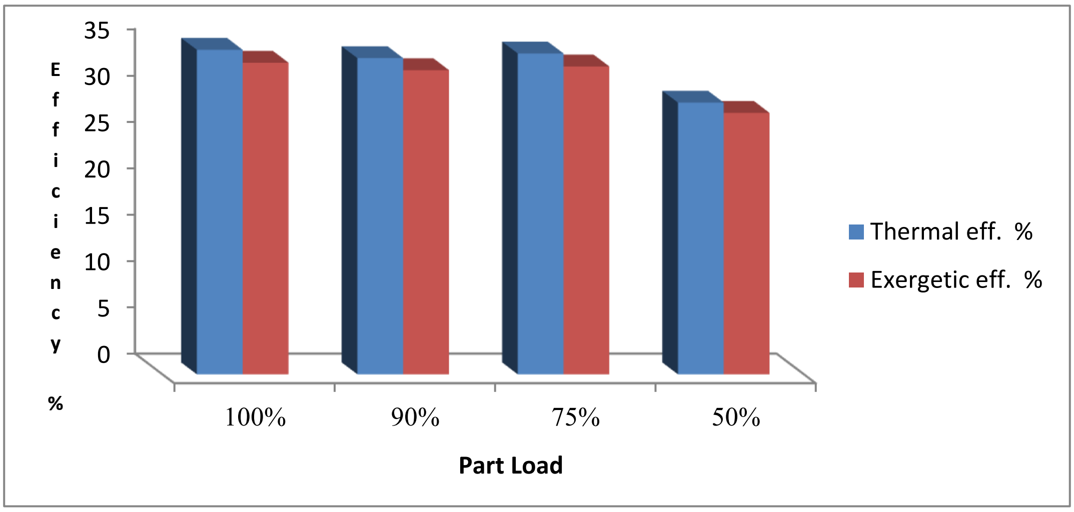

Figure 3: Thermal and exergetic efficiency at different loads for the power plant operated with natural gas.

Figures 4 and 5 exhibit distributions of exergy destruction of the main plant components as the load changes when using light oil and natural gas fuels, respectively.

As can be seen, the largest amount of exergy destruction occurred in the combustor. The turbine comes second in terms of exergy destruction compared with the combustor. However, the exergy analysis of the compressor shows that it has no impact on the overall exergy destruction. The benefit of these results lies on showing us which component we should focus on to increase the overall power plant efficiency.

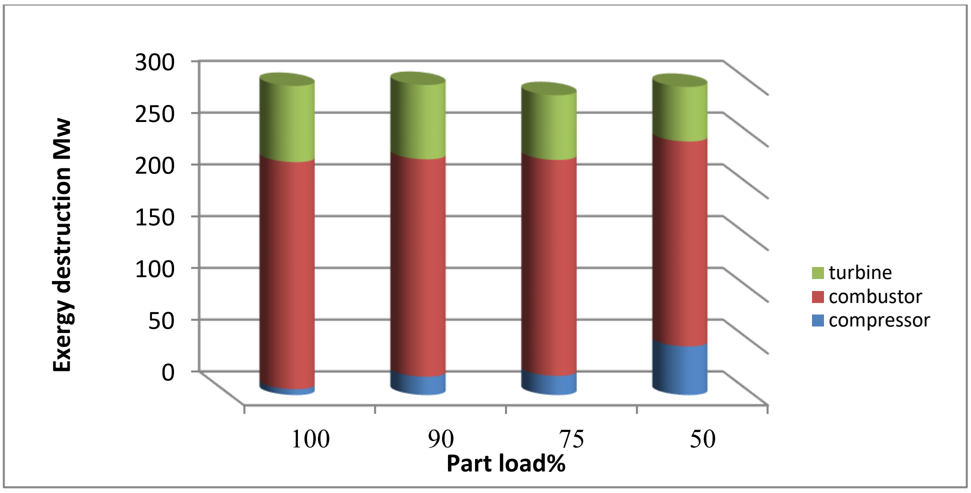

Figure 4. Exergy destruction various loads for light oil

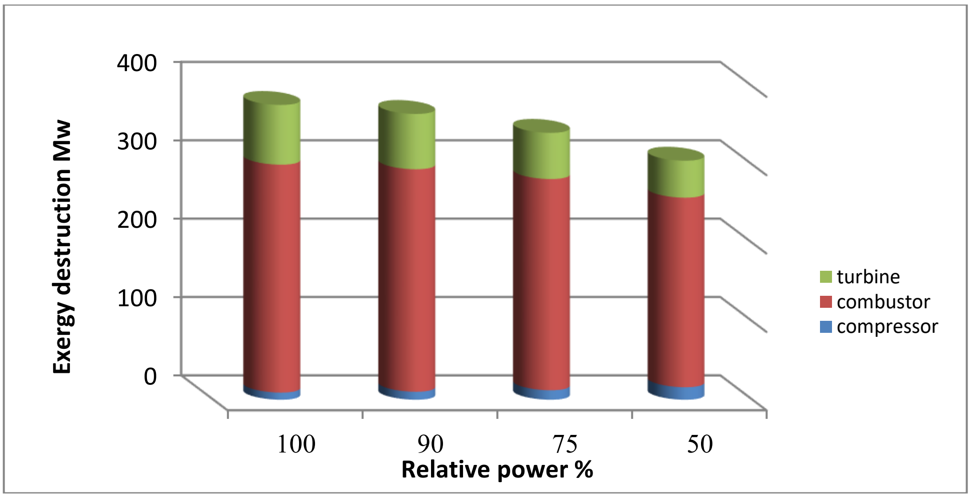

Figure 5. Exergy destruction various loads for natural gas

The thermal and exergetic efficiencies at different pressure ratios when natural gas used is shown in Figure 6. Within the given range of pressure ratio, it can be seen that thermal efficiency increases slightly with the increase of pressure ratio. The same behavior was observed for the exergetic efficiency. In all the investigated points of pressure ratio, the thermal efficiency is always higher than the exergetic efficiency. These results could be attributed to the enhancement of the combustion process via the increasing amount of air in the combustion chamber.

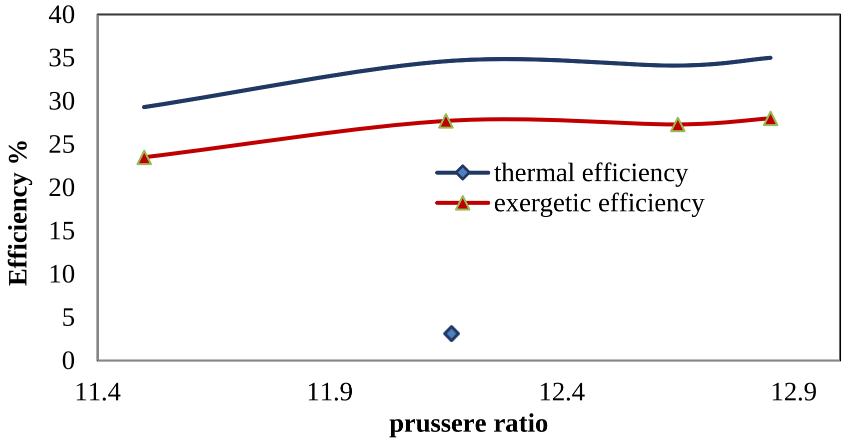

Figure 6. Exergetic and actual thermal efficiencies with variouse pressure ratio when natural gas is used

Figure 7 presents the exergetic and thermal efficiencies of the power plant at different pressure ratios when light fuel is used. As can be seen, within the given range of pressure ratio, both efficiencies increased slightly with the increase of pressure ratio. This increase in the overall efficiency could be interpreted as the increase of the efficiency of the combustion process due to the existence of an adequate amount of air in the combustion chamber.

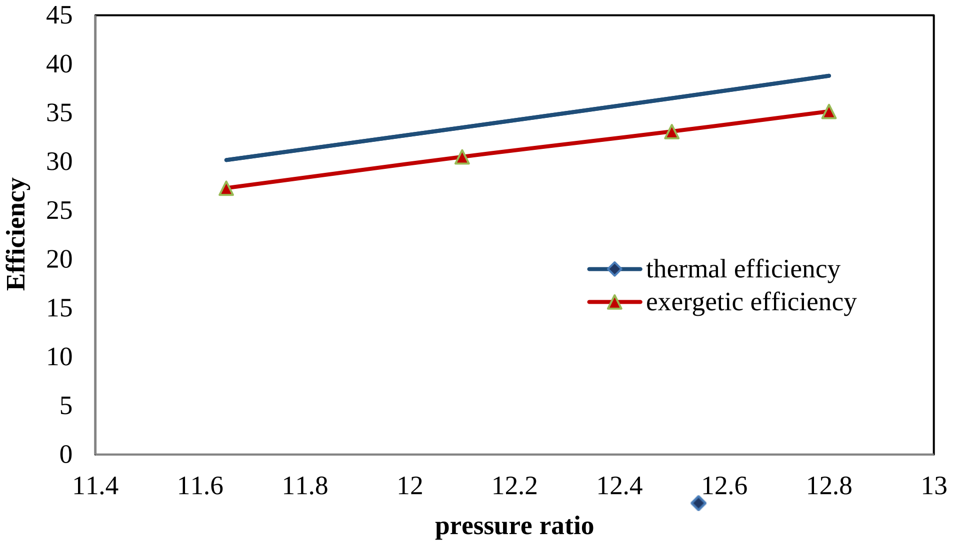

Figure 7. Exergetic and thermal efficiencies with various pressure ratio when light fuel is used

Table 5. Results Obtained from the Exergy Analysis of KGTPP

Component	ε_i**(MW)**	ε_e**(MW)**	**I(MW)**	**ε(%)**	**I(%)**
compressor	132.33	126.64	5.688	95.7	1.90
combustion chamber	534.18	314.85	219.33	58.9	73.44
Gas Turbine	314.85	241.22	73.63	76.6	24.65

Table 5 summarizes the inlet exergies, exit exergy rates, exergy destruction rates, and exergy destruction percentages for the tree main equipment in the power plant.

Finally, the exergy analysis is shown as a Grassmann diagram in Figure 8. It shows the exergy destruction and its percentages in each component of the plant based on the results obtained. As can be seen, 23.3% of the total turbine inlet exergy was destroyed and rejected in the exhaust to the surroundings.

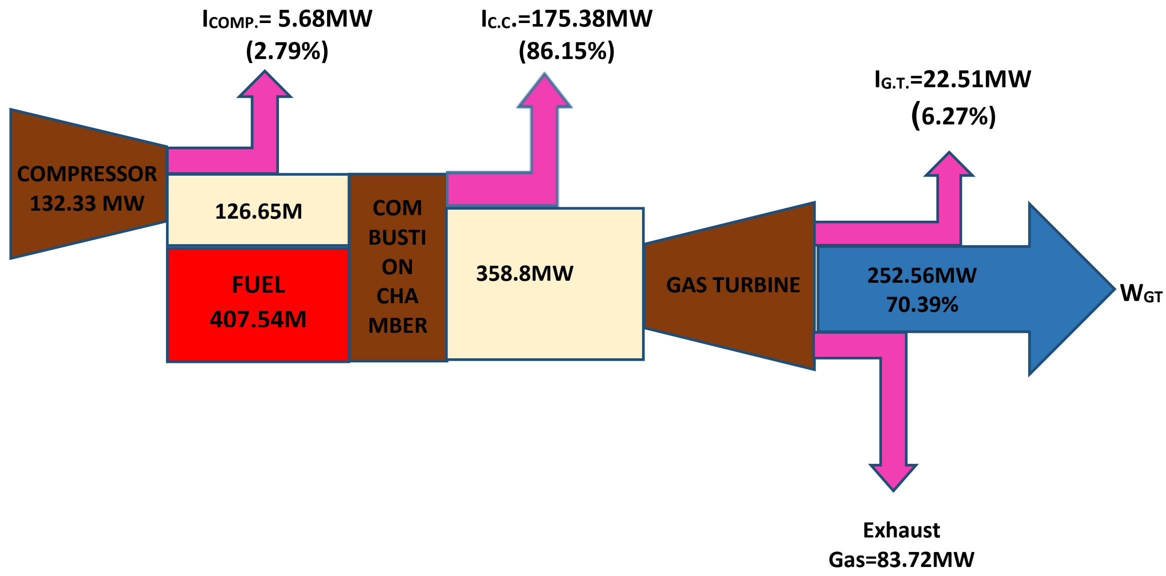

Figure 8. Grassmann Diagram of the Gas Turbine Power

Conclusion

The performance of a gas turbine power plant was investigated by using exergy analysis. In conclusion, the exergy method is an excellent tool for quantifying and localizing the energy losses at various plant components (exergy destruction), mainly by investigating behaviors of exergy relate parameters, such as exergy destruction "irreversibility" and second law efficiency under part-load condition. In the considered power cycle, the highest exergy destruction occurred in the combustion chamber followed by the gas turbine, whereas the smallest value occurred in the air compressor. It can be said that the highest exergy destruction in the combustion chamber occurred due to the large temperature differences between the combustion chamber and the working fluid. From this study, it can be concluded that the first law analysis of the thermal systems alone may lead to misinterpretations of the analysis results. Thus, the combined first and second law analysis are essential for understanding and explaining the effect of any parameter on the overall thermal system performance.

Compliance with ethical standards

Disclosure of conflict of interest

The authors declare that they have no conflict of interest.

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Nomenclature

E	*Exergy [kW]*
T	*Temperature [K]*
T_k	*Temperature of location k from which the heat $\dot{Q}_k$ is transfer to the working fluid [K]*
$\dot{Q}_k$	*Heat transfer through the boundary at temperature Tk at location k [KW]*
$\dot{w}$	*Work*
$\dot{m}$	*Mass flow rate [Kg/s]*
I	*Irreversibility or Exergy destruction [Mw]*
h	*Enthalpy [KJ/kg]*
s	*Entropy [KJ/Kg.K]*
R	*Universal gas constant*
P	*Pressure*
W	[5] *Power*
	[6]

Greek symbols

η	[7] *Efficiency*
η_{ex}	[8] *Exergetic efficiency*
$\mathcal{E}$	[9] *Specific exergy [Mw]*
Subscripts	[10]
ph	[11] *Physical*
ch	*Chemical*
ke	[12] *Kinetic*
p	[13] *Potential*
i	*Inlet*
e	[14] *Exit*
o	[15] *Standard conditions*
ex	[16] *Exergetic*
cv	*Control volume*
f	*Fuel*
fg	*Flue gases*
C	*Compressor*
GT	*Gas turbine*
th	[17] *Thermal*

