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Assessment of the gas turbine unit in Kirkuk gas power station using energy and exergy analysis

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Abstract: The present study aims to evaluate and analyze the gas turbine power generation unit (K3) in Kirkuk gas power station under actual weather conditions of (2017), which is located south of Kirkuk in Taza district. (K3) unit capacity production estimated at (255 MW) at maximum load The unit consists of a gas turbine and an air compressor connected to the generator via a common shaft and operates in the Joule-Brighton simple cycle. The current study dealt with the analysis of the energy and exergy for the mentioned unit based on the law of conservation of mass, the first law, the second law in terms of the influence of each of the factors (ambient temperature, compression ratio, and relative humidity). Two methods were used to simulate the working data; the first is simulation programming by Excel, and the other is by using CHEMCAD software. The average data have been used in the simulation process for each month in that year to evaluate the performance of the unit K3. The simulation results showed that the combustion chamber represents the most important part in the exergy destruction, the largest energy rate can be obtained from fuel as chemical, and the results of both methods showed maximum energy where energy efficiency was around (37.67%) and (36.89%), respectively.

Keywords: Simple cycle , Gas turbine ,Energy analysis, Exergy analysis, Irreversibility

Nomenclature

B.W.R	Brick work ratio	ηI	Energy efficiency
CO	carbon dioxide	ηII	Exergy efficiency
CP	Specific heat, J/kg.K	***Subscripts***	
Ex	Exergy, W	a	Ambient
h	Enthalpy, J/kg	c	Compressor
g	Acceleration, 9.81 m/s^2	CH	Chemical
JBC	Joule-Brighton cycle	cc	Combustion champer
KE	Kinetic energy, J	i	Inlet
LHV	Lower heating value, J	o	Out let
$\dot{m}$	Mass flow rate, kg/s	dec	Destructs
P	Pressure, Pa	KN	Kinetic
PE	Potential energy, J	PH	Physical
rp	Pressure ratio	PT	Potential
RH	Relative humidity	T	Turbine
s	Entropy,KJ/k	T_2	Compressor inlet temperature

T	Temperature, K	T_3	Turbine inlet temperature
W	Work, W	T_4	Turbine outlet temperature
V	Velocity, m/s		
z	Elevation, m		

1.Introduction

Because of the population growth that leas to the rising need for energy all over the world and the rising need for safe access to it in the future not only in domestic consumption but in industrial and commercial sectors and how to secure this energy which includes the economic aspects and its impact on the environment. The search for alternative energy sources and not relying on fossil fuels like natural gas, coal, oil, ...etc.) because they cause pollution to the environment, which lead to global warming caused by an increase (CO)has become necessary at present, especially after the damage that fossil energy causes to the environment and its rising price. Benefiting from renewable energy has become a necessity, as many studies have been conducted to make the most of these sources through the improvement and development of solar energy systems, wind energy, sub-earth energy ... etc., in order to improve their efficiency [1]. However, to this point, companies have not been able to provide renewable energy systems to generate electric power to meet the need of the world, only a very small percentage, not exceeding 10% of the total need. For this reason, the interest in improving the traditional electrical generating systems that operate on fossil fuels is significant.Relying only on thermal performance analysis is not an accurate measure to evaluate thermal systems because it accurately shows all the losses happened during power generation [2]. The second law determines the type and quantity of energy ; thus it determines the amount of losses that can be reduced, or the so-called exergy analysis through preserving mass and energy together and can improve design energy for all systems (ROSEN, n.d). The goal for developing systems using conventional fuels effectively, which are considered non-renewable resources, is to use available energy analysis because it is suitable to enhance the purpose. After all, it determines the actual size of losses, and this is useful in designing systems. It reduces the inefficiency of existing systems and thus reduces the cost [4], Because of the electrical energy crisis that the world is facing for several reasons including the lack of rationalization in the use of electrical energy and other reasonswhich prompted researchers to study generating stations in general and how to increase their efficiency and identify weaknesses that occur in the system that affects the performance and life of the unit and affects the environment. so it became necessary to improve the system by adding a specific part that was adding internal cooling, reheat, clamping CO2, hydro energ .... etc. or take advantage of SE to heat the air entering the combustion chamber [5] or take advantage of the heat of the exhaust gas [6] for gas The turbine cycle or the simple cycle called (JBC) is open, the fuel is an essential part of any energy-producing system that includes solid fuels, liquid fuel, and gaseous fuel.

The research aims to study ambient temperature, compression ratio, and relative humidity parameters on the thermal energy and exergy of a gas power plant in 2017 under actual weather conditions. Both theoretical and software methods are used to simulate power plant working data.

2. Modelling of the proposed system

The simulation was carried out on the working data of the gas turbine unit (K3) for 2017 based on the first and second laws of thermodynamics to determine the energy and exergy for each month, Fig. 1 shows the detailed sketch of the K3 unit and the arrangement of its components. First, started the energy simulation of the unit,

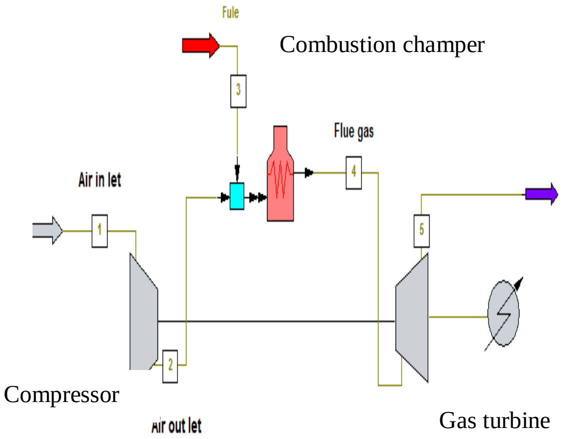

Fig. 1: Gas turbine schematic diagram

2.1 Energy Analysis

In order to analyze the energy of the gas turbine unit, a set of assumptions are introduced so that the resulting mathematical model is simple enough for analysis:

a. Gas turbine operated under steady-stated conditions.
b. KE and PE effects are deprecated.
c. Air and fuel are ideal gases. Fuel usage (dry gas), with LHV = (49383.738Kj/kg).
d. The efficiency of gas turbines and generators is 98% and 97%, respectively.

Gas turbine energy analysis is done for each component according to Fig. 2, this represents its real cycle on the T-s diagram. The energy analysis sequence is as follows:

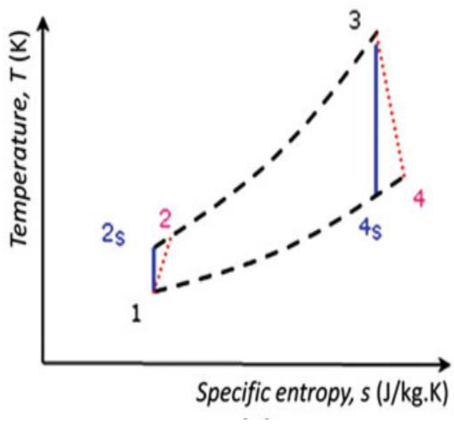

Fig. 2: Real cycle of gas turbine unit [7]

A. Compressor work (W_C)

The amount of compressor work is calculated as follows [9]:

$$W_C = \dot{m}\,\bar{C}_{p.i}\,(T_2 - T_1) \quad \dots\dots\dots\dots(1)$$

where

$$\dot{m}_a = \dot{m}_i = \dot{m}_e \quad \dots\dots\dots(2)$$

and

$\bar{C}_{p.i}$ is represented the products of a combustion chamber and is calculated as [9]:

$$\bar{C}_{p,i} = a_i + b_i T + c_i T^2 + d_i T^3 \quad \dots..(3)$$

The amount of heat added to the combustion chamber through the combustion of a mixture of fuel and air is calculated as [10]:

$$Q_{add} = \dot{m}_f \times LHV \quad \dots\dots..(4)$$

B. Turbine Work (WT)

Can express the work produced by the turbine as a result of the expansion of the combustion products through its blades [11]:

$$W_T = \dot{m}_g\, Cp_g\, (T_3 - T_4) \quad \dots\dots\dots(5)$$

where

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad \dots\dots.(6)$$

C. Thermal efficiency (ηI)

The thermal efficiency of the thermodynamic cycle is defined as the ratio of the network produced by the turbine to the heat energy generated by the combustion process in the combustion chamber. It is expressed [10 and 12]:

$$\eta_I = \frac{\dot{W}_{net}}{\dot{m}_f\,(LHV)} \quad \dots..(7)$$

D. Brake Work Ratio

The brake work ratio is defined as the ratio of thermal energy to the turbine work. It is expressed [13]:

$$B.W.R = \frac{W_C}{W_T} \times 100\% \quad \dots\dots.(8)$$

E. Specific Fuel Consumption

Specific fuel consumption (SFC) is one of the essential matters when analyzing the energy and exergy for gas power stations because it represents a measure of the fuel consumed to generate energy for a certain period by a gas turbine [14]. It is defined as a ratio of fuel mass consumption ($\dot{m}_f$) in one hour to a network ($\dot{W}_{net}$). The low values of (SFC) indicats that the turbine works well [15]. It is calculated as follows [16 and 17]:

$$SFC = \frac{3600 \times \dot{m}_f}{\dot{W}_{net}} \quad \dots\dots(9)$$

2.2 Exergy Analysis:

The exergy of the gas turbine power plant can be divided into four main types when the effect of each nuclear, electrical, surface tension and magnetic are neglected . The total exergy is summarized as follows, [17] [18] [19]:

$$\dot{E}\mathrm{x} = \dot{E}\mathrm{x_{CH}} + \dot{E}\mathrm{x_{PH}} + \dot{E}\mathrm{x_{KN}} + \dot{E}\mathrm{x_{PT}} \quad \ldots(10)$$

The amount of exergy to each part of the system the open cycle (JBC) is calculated as follows as [4] :

$$\mathrm{ex} = (\mathrm{h_i} - \mathrm{h_o}) - \mathrm{T_a}(\mathrm{s_i} - \mathrm{s_o}) + \frac{\mathrm{V^2}}{2} + \mathrm{g\,z} + \mathrm{ex_{CH}} \quad \ldots (11)$$

As for the rate of exergy for ṁ , it is calculated through the following law [20] :

$$\dot{E}\mathrm{x} = \dot{\mathrm{m}} \times \mathbf{ex} \quad \ldots..(12)$$

$\dot{\mathrm{E}}\mathrm{x_{PH}}$resulting from the change in the temperatures entering and exiting a system relative to the external environment, used for air, fuel and combustion gases, can be expressed in the following form [9] :

$$\mathrm{ex_{PH}} = (\mathrm{h_i} - \mathrm{h_i}) - \mathrm{T_a}(\mathrm{s_i} - \mathrm{s_o}) \ldots..(13)$$

To calculate the amount of ΔS between two states entrance and exit in each process when imposing the specific entropy of an ideal gas as follows as [7] :

$$\Delta\mathrm{S} = \mathrm{S}_{i+1} - \mathrm{S_o} = \mathrm{C_{P,i}} \ln\left(\frac{\mathrm{T_{i+1}}}{\mathrm{T_o}}\right) - \mathrm{R}\ln\left(\frac{\mathrm{P_{i+1}}}{\mathrm{P_o}}\right) \quad \ldots.(14)$$

The ηII for each component can be expressed as [21]:

$$\eta_{\mathrm{II,C}} = \frac{\mathrm{Ex_2} - \mathrm{Ex_1}}{\mathrm{W}} \times 100\% \quad \text{for AC} \ldots\ldots(15)$$

$$\eta_{\mathrm{II,CC}} = \frac{\mathrm{Ex_{flue\ gas}}}{\mathrm{Ex_{fule}} + \mathrm{Ex_{air}}} \times 100\% \quad \text{for CC} \quad \ldots\ldots\ldots..(16)$$

$$\eta_{\mathrm{II,T}} = \left(1 - \frac{\dot{\mathrm{E}}\mathrm{x_{des,GT}}}{\mathrm{Ex_3} - \mathrm{Ex_4}}\right) \times 100\% \quad \text{for GT} \quad \ldots\ldots\ldots.(17)$$

and $\eta_{\mathrm{II,cycle}}$ is calculated from the fllowing [22]:

$$\eta_{\mathrm{II,cycle}} = \frac{\mathrm{W_{net}}}{\mathrm{Ex_f}} \times 100\% \quad \ldots..(18)$$

where $\dot{\mathrm{E}}\mathrm{x_{des}}$ is determined as[23]:

$$\dot{\mathrm{E}}\mathrm{x_{des}} = \mathrm{T_a}\,\sigma_{\mathrm{cv}} \quad \ldots\ldots.(19)$$

and $\dot{\mathrm{W}}_{\mathrm{net,load}}$ is [24 and 25]:

$$\dot{\mathrm{W}}_{\mathrm{net,load}} = (\dot{\mathrm{W}}_{\mathrm{GT}} - \dot{\mathrm{W}}_{\mathrm{AC}}) * (\eta_{\mathrm{generator}} \times \eta_{\mathrm{mech}}) \quad \ldots..(20)$$

3. Result and discussion

The simulation of working data gas turbine (K3) was conducted for twelve months in 2017 with both methods. It includes energy and exergy changes by ambient temperature, compression ratio, and relative humidity. First, we discuss the effect of these parameters on the energy production behaviour:

3.1 Effect of change in (Ta , rp , RH) on η_I :

Figures 3 to 5 show the result of studying Ta, rp, and RH parameters on the η_{I}, respectively. It is based on average data for each month. Fig. 3 indicated that the change in a Ta has more influence on the performance of the unit; their maximum η_{I} obtained at Ta equal to 19.39 C, was around 37.67% where it was less than the design η_{I}, ranges between 38 to 40% as measured at a standard condition (1 bar and 15 oC) [26]. The value of T1 is affected by the fluctuation in Ta. from which the efficiency of the cycle is determined where T1 is inversely proportional to the η_{I}, but this does not mean that it must be at a freezing point or well below the design temperature (15 °C) because as we indicated earlier the best η_{I} of the unit was found when tested At standard temperatures, this result is in agreement with references [27, 28]. Also, Figure 2 indicates that rp and RH affect the η_{I} values, regardless of the air temperature, as shown in Figs 4 and 5, where the average temperature of March and December was equal and estimated at 14.34°C. Still, the efficiency was not identical because of the differences in rp and RH in each month. The value of rp and RH were 15.7 and 23%, respectively, while 16.5 and 30% in December. However, the efficiency of the unit in December was higher than in March.

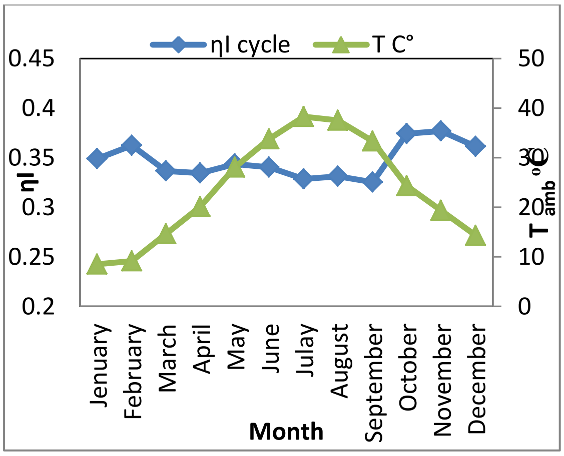

Fig.3: Effect of ambient temperature on thermal efficiency

Figure 4 shows the effect of rp on the overall thermal efficiency for one year of unit operation and indicates that rp is inversely related to unit efficiency, where an increase in rp leads to a decrease in η_I, until September, where after this month the η_I was increased with an increase in rp. Due to the rise in the relative humidity of the air, this result agrees with the results of references [29-32].

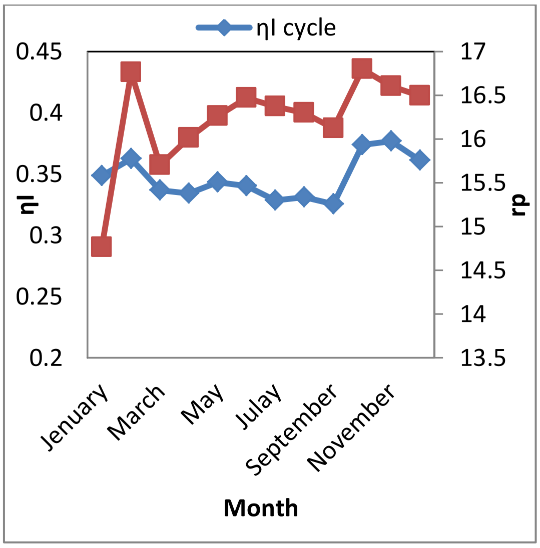

Fig. 4: Effect of compression ratio on thermal efficiency

The effect of change in RH on the compressor work was minute on the unit η_I, as shown in Fig.5 when it compared to other parameters. Stabilised almost the RH in the hot climate, where it has fluctuated in narrow ranges from May to September, but efficiency decreased in this period due to Ta rising. In the case of low RH, the gas turbine efficiency can be improved by adding a humidifier system to humidify the inlet air to the
compressor.

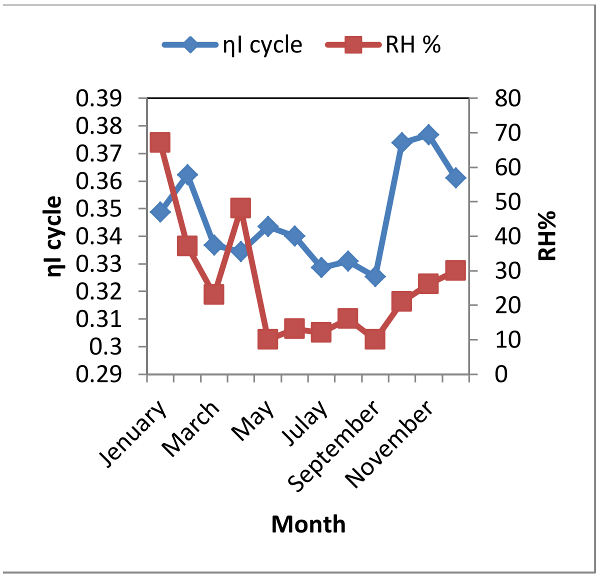

Fig.5: Effect of relative humidity on thermal efficiency

3.2 The effect of change in Ta and rp on the exergy efficiency :

The results of exergy analysis of the gas turbine unit K3 crossed Figs 6 to 8, which show the effect of both parameters Ta and rp on the efficiency of the η_{II}. Fig. 6 shows that the ambient temperature strongly impacts the η_{II}; Overall results showed that the maximum η_{II} was obtained in November; it was about 37% when Ta was 19.39°C. The minimum was obtained in September, ranging from 31 to 31.5%. Fig.6 indicates lower exergy efficiency obtained in higher Ta. Also, it gets less than the standard temperature of 15 °C. Fig.5 indicates lower η_{II} obtained in higher Ta. Also, it gets less than the standard temperature of 15 °C and this is consistent with most researchers [39] [40] [41]

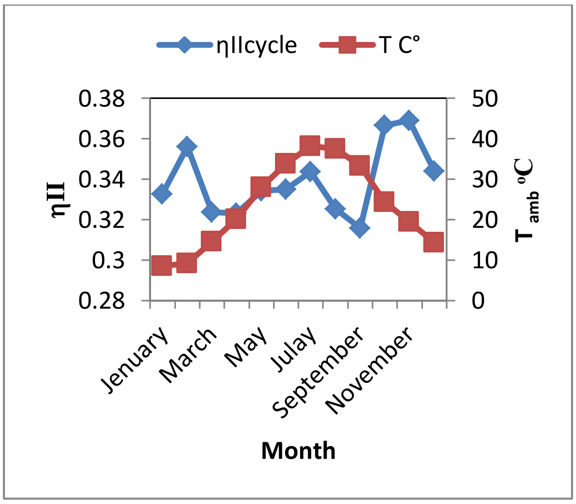

Fig..6: Effect of ambient temperature on exergy efficiency

The results showed the effect of (rp) for a year on η_{II}, as found that the maximum η_{II} was (36.89%) for November, where the (rp) was approximately (16.6), while the highest (rp) was at November (16.80) and (ηII) was (36.65%) with a difference of (0.24), while (ηII) was at the lowest value of (rp), which amounted to (14.76), and η_{II} was found to be (30.73%) for December as shown in Fig. 6, and this is consistent with most of the previous studies [42][43][44], who indicated that the (rp), the greater (ηII). As for the researcher, [25] showed that (ηII) increases by (6%) when (rp) increases by (5%) but starts to decline after that, as the efficiency decreases by (0.04) if (rp) increases by (7%), and η_{II} increases with (rp). The efficiency decreases again with a decrease in (rp).

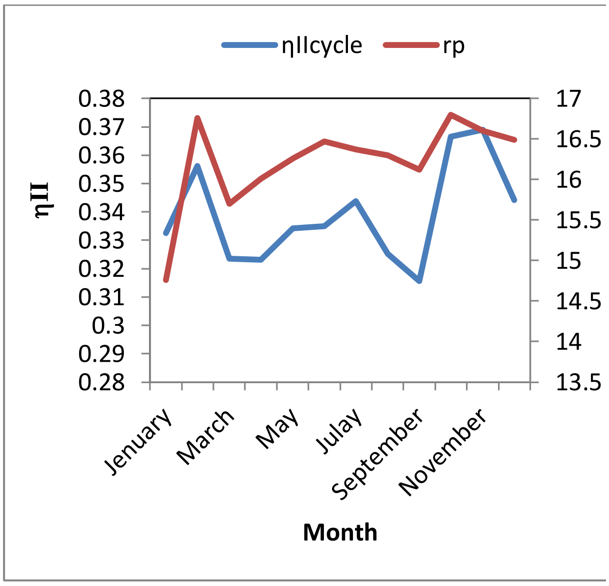

Fig.7: Effect of compression ratio on thermal efficiency

Figure 8 notes from the results that RH has an apparent effect on η_{II}, as the maximum exergy efficiency reached at November (36.89%) at (RH 16%), while the lowestη_{II} at the highest percentage of RH of the month was as (RH 67%) (30.73%), and η_{II} at the lowest value of RH for May and September was (RH 10%), where the efficiency in May was (33.42%), while it was (31.56%) in September. This confirms that η_{II} decreases with increasing RH.

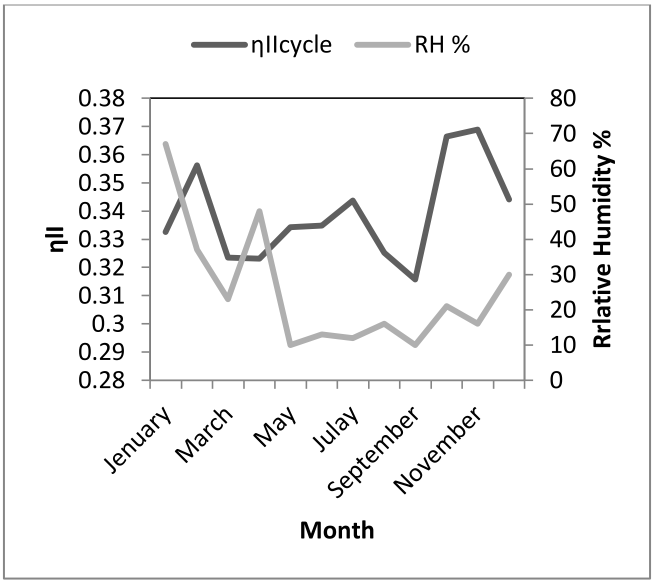

Fig.8: Effect of relative humidity on exergy efficiency

3.3 Results amount of exergy and exergy destroyed:

Through the results obtained, the highest rate of exergy that we can get from fuel because of the enormous energy stored in the fuel ($\dot{E}x_{CH}$) (Exfule (1003.598 MW)), followed by the gases coming out of (CC) (Ex3) (789.861MW) , (AC) ((Ex2) by (229.3101MW)), gases introduced to the atmospheric air from (GT) (Ex4 198.2518MW) (respectively, as shown in Fig .9.

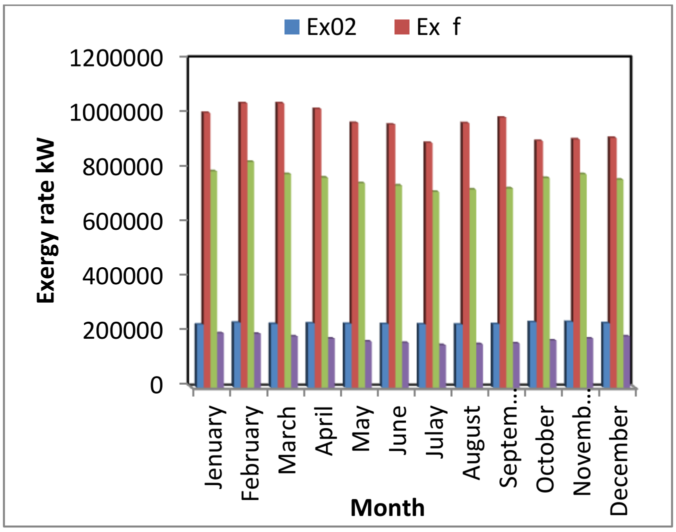

Fig.9: Exergy simulation of unit components over one year

As for the rate of (Ex,des), it was found that (CC) occupied the first place for the processes of destruction of exergy due to the irreversible processes that occur in the (CC) (chemical reactions) as indicated by most researchers [17] [45] [46], where Ex das, CC reached (443.0382 MW), While the two studies [47] [48] showed that

(Ex,des) occurs in the exhaust gas and then (CC), Then (AC) Ex des, AC (36.0847MW), then (GT) Ex des, GT, (7.6862MW) as shown in Fig.10.

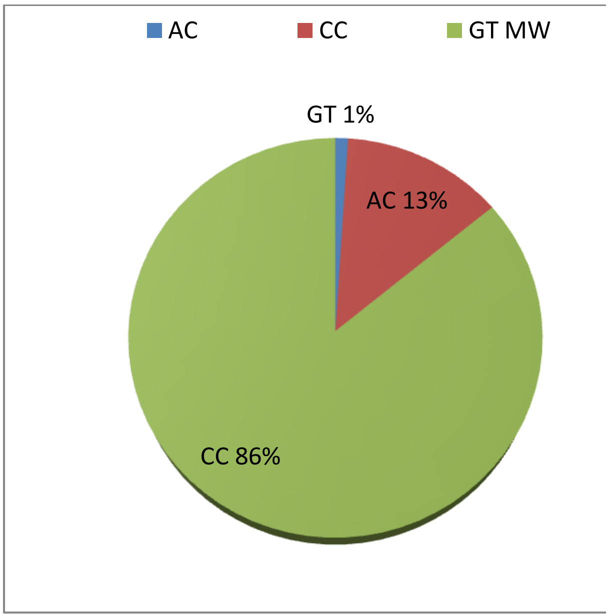

Fig. 10: The rate of exergy losses in the cycle process

4. Conclusion

Conducting energy and exergy analysis to the power plant in order to determine the losses in actual working conditions, in addition to studying the effect of external factors on the performance of the system and the specific rate of fuel consumption, which includes (Ta, RH, rp). The results of the operation of this system showed several conclusions, namely:

1. Ambient temperature has a noticeable effect on the unit's efficiency, as the efficiency decreases with increasing Ta.
2. Efficiency has increased with (rp) increase.
3. Maximum exergy distortion was obtained in the combustion chamber due to irreversible processes followed by compressor and gas turbine, respectively..
4. Higher ambient temperature increases specific fuel consumption. Thus it economically affects the price of power production.

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