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Comparative Exergo-Environmental Analysis of Simple and Regenerative Cycle Gas Turbine Plants

Ebigenibo Genuine Saturday[1*], Orok Anyama Etok Orok[2]
[1]Department of Mechanical Engineering, University of Port Harcourt, Nigeria
[2]Master's Degree Student, Master in Engineering Management Programme, University of Port Harcourt, Nigeria

Original Research Article

***Corresponding author**
Ebigenibo Genuine Saturday

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Abstract: Comparative exergo-environmental analysis of simple and regenerative cycle gas turbine plants was carried out in this work using PG 6581 B gas turbine engine operated by a company in the Niger Delta area of Nigeria as a case study. A gas turbine engine model was created for the analysis using in-house software and the first and second law analyses were applied to both plants. The environmental sustainability indicators of both plants (depletion factor for each component, environmental sustainability index, waste exergy ratio, and environmental effect factor) were investigated. The regenerative cycle plant was analyzed exploiting regenerative effectiveness values of the regenerator between 80% and 100%, but, all the results for the regenerative cycle presented here were obtained at 80% regenerator effectiveness, the lowest value used. The thermal efficiency of the simple cycle plant is 35.19% but that of the regenerative cycle plant is 43.65% at 80% regenerative effectiveness. The second law efficiency of the regenerative cycle plant is 40.85% as against that of the simple cycle plant which is 32.93%. The exergy destruction in the combustion chamber is the highest in both cycles but the value is lower in the regenerative cycle and it decreases with increase in the regenerative effectiveness. The environmental sustainability index which indicates how sustainable the environment is with respect to engine operation for the simple cycle is 1.32 while that of the regenerative cycle is 1.43. Smaller values of the waste exergy ratio and the environmental effect factor indicate safer environment from engine operation but both of these values are greater in the simple cycle plant. The regenerative cycle plant is thus more favourable to operate in both engineering performance terms and in terms of environmental sustainability.

Keywords: Exergo-environmental, Depletion factor, Environmental sustainability index, Waste exergy ratio, Environmental effect factor.

INTRODUCTION

The gas turbine as an internal combustion engine has three basic components which are turbine, compressor (C) and combustion chamber (CC). The gas turbine engine has different operating cycles. The simple open cycle engine consists of a compressor, combustion chamber and turbine only and the power produced is mainly extracted from the cold end of the engine. The combustion gases at the exit of the turbine are allowed to escape to the atmosphere and constitute pollution. It is possible to hit up the gases at the exit of the compressor with the exhaust gases before letting them into the CC. This gives rise to a modified cycle plant known as regenerative cycle gas turbine plant [1, 2]. Aside the pressure ratio and the ambient temperature, the level of performance of the regenerative plant depends basically on the effectiveness of the regenerator which is a heat exchanger. The gases at the exit of the regenerator are exhausted into the atmosphere like in the simple cycle plant, but in this case at much lower temperature. The impact of the exhaust gases on the environment will be lower in the regenerative cycle plant. In Nigeria, there are a lot of gas turbine power stations but all the turbines operate either on the simple cycle arrangement or aeroderivative engines are used with no regeneration. Incorporating a regenerator to existing plants is possible but cost must be borne. This paper compares the engineering performance and the environment impacts of simple cycle and regenerative cycle gas turbine plants. This exercise is carried out in this research using a frame 6 gas turbine engine (Model: PG 6581 B) by General Electric (GE) operated by a company in the Niger Delta area of Nigeria. The comparison of the economic indices of both plants is a subject for another paper. This study is thus centred on comparative exergo-environmental analysis of simple and regenerative cycle gas turbine plants.

Many researchers have looked at gas turbine performance and factors affecting engine performance [3–5]. Using a regenerative Brayton cycle model, the performance of a regenerative cycle gas turbine plant examining the effect of the heat exchanger (regenerator) on the system performance was analyzed [6]. The effect of regeneration on the thermal efficiency and the power output of a gas turbine plant was also studied [7]. The performance of a gas turbine plant fired by fuels other than natural gas at different conditions of engine operation to ascertain the technical feasibility of the usage of these other fuels has been evaluated in [8] while a parametric study of a gas turbine plant was carried out in [9]. The variation of engine operating conditions on performance parameters was also studied in [10] and results similar to those of related works were obtained. It observed that a gas turbine plant losses 0.1% of its thermal efficiency and 1.47MW of its total power output when ambient temperature increases by 1 K above ISO (International Standard Organization) condition [11]. Many other authors have carried out performance analysis of different configurations of gas turbine engine cycles [12–14].

Exergo-environmental analysis of a system entails finding the exergy destruction in each of the components and the estimation of the environmental sustainability indices from the plant. A gas turbine based combined cooling, heating and power (CCHP) system was modelled and analyzed in [15] through exergy method. The exergo-environmental analyses of non-gas turbine-based systems have been investigated by several other researchers [16–18]. The essence of some of these works is to optimize the performance of the system and evaluate the environmental impact rate. Although, there are several works on exergo-environmental analysis and parametric studies of gas turbine power plants, but such analysis has not been extended to the plant used as a case study viz-a-viz comparing the results with a regeneration plant derived from the simple cycle plant.

METHODOLOGY

An engine model will be created using in-house software. The engine model is a thermodynamic model which takes into account the various losses in a gas turbine system. Engineering performance analysis of the simple cycle plant and the regenerative cycle plant based on the first law of thermodynamics will be carried out. Also, exergy analysis in the various components in both engines will be carried out and the exergy destruction in each component will be estimated. The environmental sustainability indices in both systems will be presented and comparisons will be done.

Gas Turbine Engine Model Creation

The gas turbine model was created and used for the analysis in this work. The model was created using in-house software [19]. The procedure is to select different values of compressor isentropic efficiency, isentropic efficiency of the turbine, combustion efficiency, combustion pressure loss, and exhaust pressure loss. Figure 1 show an interface of the software where these parameters are inputted.

Fig-1: Interface of software for the creation of GT engine model

From this interface, these five parameters are varied until the power output and the turbine exit temperature from the simulations closely approximate those from the field. The isentropic efficiencies and the losses obtained are used in all further calculations.

Gas Turbine Performance Analysis

The energy approach will be applied to estimate the thermal efficiency of both the simple cycle and the regenerative cycle engines, where the turbine entry temperature (TET) estimated in the simple cycle will be transferred to the regenerate cycle engine. The fuel flow rate of the regenerative cycle engine will be estimated at different values of regenerative effectiveness of the regenerator. All basic losses in a gas turbine system as in [19] are considered in the analysis.

Performance Analysis of Simple Cycle Gas Turbine System

The performance analysis is simplified with the usage of the T-S diagram in Figure 2(a).

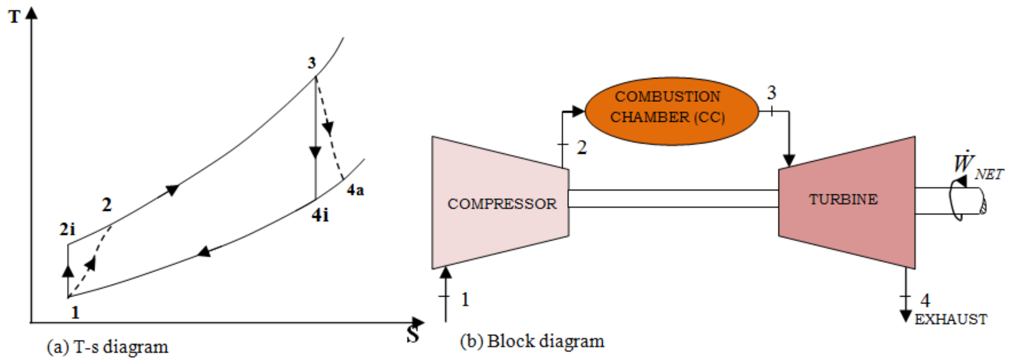

Fig-2: Temperature- entropy (T-s) and block diagrams of a real GT engine cycle

From the T-s diagram, process 1-2i is the isentropic compression, process 1-2a is the actual compression, and process 3-4i is isentropic expansion while process 3-4a is the actual expansion. The thermal efficiency η_{th} of the cycle is given as,

$$\eta_{th} = \frac{W_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_t - \dot{W}_c}{\dot{Q}_{in}} \qquad (1)$$

where W_{net} is the net power output of the cycle, $\dot{W}_t$ is the turbine power output, $\dot{W}_c$ is the power consumed by the compressor and $\dot{Q}_{in}$ is the rate of heat input into the cycle. The power consumed by the compressor is,

$$\dot{W}_c = \dot{m}_a c_{p,a} \left(T_{2a} - T_1\right) \qquad (2)$$

Where $\dot{m}_a$ the air flow rate in (kg/s) is, $c_{p,a}$ is the specific eat capacity of air, T_{2a} is the actual temperature at the compressor exit and T_1 is the temperature at the inlet of the compressor, assumed to be the ambient temperature. T_{2a} Relates with the ideal temperature at the compressor exit T_{2i} in the form,

$$T_{2a} = T_1 + \frac{\left(T_{2i} - T_1\right)}{\eta_{c,i}} \qquad (3)$$

Where $\eta_{c,i}$ is the isentropic efficiency of the compression process, accounting for the compression losses. The value of $\eta_{c,i}$ was obtained in creating an engine model. The ideal temperature at the compressor exit is given as,

$$T_{2i} = T_1 \left(r_p\right)^{\frac{\gamma-1}{\gamma}} \qquad (4)$$

Where r_p is the pressure ratio across the turbine and γ is the ratio of specific heat capacities. The power output from the turbine is given by Equation (5),

$$\dot{W}_t = \left(\dot{m}_a + m_f\right) c_{p,g} \left(T_3 - T_{4a}\right) \qquad (5)$$

Where $\dot{m}_f$ is the mass flow rate of the fuel, $c_{p,g}$ is the specific heat capacity of the flue gases and T_{4a} is the actual temperature of the gases at the turbine exit. T_{4a} relates with the ideal temperature at the turbine exit T_{4i} in the form,

$$T_{4a} = T_3 - \eta_{T,i}\left(T_3 - T_{4i}\right) \qquad (6)$$

where $\eta_{T,i}$ is the isentropic efficiency of the expansion process, obtained in the process of creating the engine model. The ideal temperature at the turbine exit is given as,

$$T_{4i} = T_3\left(r_p\right)^{\frac{1-\gamma_g}{\gamma_g}} \qquad (7)$$

where γ_g is the ratio of the specific heat capacity of the flue gases, taken as 1.33 in this work. The heat input into the cycle comes from the burning of the fuel, natural gas in this case. This is given as,

$$\dot{Q}_{in} = \dot{m}_f LCV_f \eta_{cc} \qquad (8)$$

Where LCV_f is the lower calorific value of the fuel and η_{cc} is the combustion efficiency. The value of the combustion efficiency was estimated in creating the engine model. At a given fuel flow rate, the TET (T_3) can be estimated by considering energy balance in the CC thus,

$$\dot{m}_a c_{p,a} T_{2a} + \dot{m}_f LCV_f \eta_{cc} = \left(\dot{m}_a + \dot{m}_f\right) c_{p,g} T_3 \qquad (9)$$

$$T_3 = \frac{\dot{m}_a c_{p,a} T_{2a} + \dot{m}_f LCV_f \eta_{cc}}{\left(\dot{m}_a + \dot{m}_f\right) c_{p,f}} \qquad (10)$$

This TET value obtained here is transferred to the regenerative cycle and the fuel flow rate at different values of the regenerative effectiveness will be determined. Pressure losses in the CC and exhaust are estimated in creating the engine model.

Performance Analysis of the Regenerative Cycle Gas Turbine System

The T-s diagram of the regenerative cycle GT system is shown in Figure 3(a) and it is exploited for the analysis here.

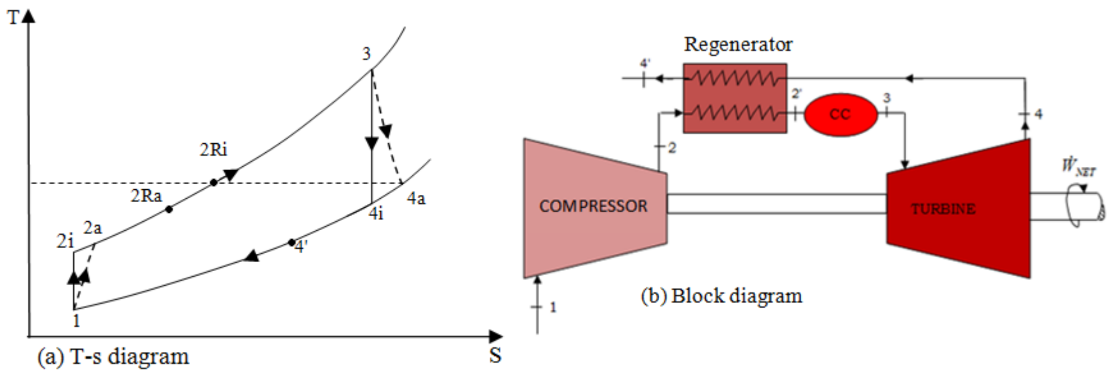

Fig-3: T-s and block diagrams of a regenerative gas turbine engine cycle

From Figure 3, the gases at the compressor exit are heated up from temperature T_{2a} to T_{2Ra}. In the ideal case, they are heated up to $T_{2Ri}=T_{4a}$. T_{2a}, T_{2Ra} and T_{2Ri} are related by the effectiveness of the regenerator ε given by Equation (11),

$$\varepsilon = \frac{T_{2Ra} - T_{2a}}{T_{2Ri} - T_{2a}} = \frac{T_{2Ra} - T_{2a}}{T_{4a} - T_{2a}} \qquad (11)$$

For a given value of ε , T_{2Ra} is obtained as,

$$T_{2Ra} = T_{2a} + \varepsilon\left(T_{4a} - T_{2a}\right) \qquad (12)$$

The thermal efficiency $\eta_{th,R}$ of the regenerative GT cycle is,

$$\eta_{th,R} = \frac{\dot{W}_{net,R}}{\dot{Q}_{in,R}} \qquad (13)$$

Where $\dot{W}_{net,R}$ is the net power output in the regenerative cycle, same as that of the simple cycle plant, and $\dot{Q}_{in,R}$ is the rate of heat input into the cycle. The latter can be estimated as,

$$\dot{Q}_{in,R} = \dot{m}_a c_{p,a}\left(T_3 - T_{2a} - \varepsilon\left[T_{4a} - T_{2a}\right]\right) \qquad (14)$$

The heat input depends on the regenerator effectiveness. The fuel flow rate can be expressed as,

$$\dot{m}_f = \frac{\dot{m}_a c_{p,a}\left(T_3 - T_{2a} - \varepsilon\left[T_{4a} - T_{2a}\right]\right)}{\eta_{cc} LCV_f} \qquad (15)$$

Exergy Analysis

Exergy analysis is carried out at the components level. The exergy at the inlet, exit, the exergy destruction and the exergetic efficiency of each component will be analyzed here. At a given state point defined by temperature T and pressure p, the specific exergy x (in KJ/kg) is defined as,

$$x = c_p(T - T_o) - T_o\left[c_p \ln\left(T/T_o\right) - R\ln\left(p/p_o\right)\right] \qquad (17)$$

where c_p is the specific heat capacity of the of the working fluid at that point, T_o and p_o are the environmental temperature and pressure respectively- referred to as dead state values, R is the gas constant of the working fluid. The exergy value in rate form $\dot{X}$ (in kJ/s = kW) relates with the specific parameter in the form

$$\dot{X} = \dot{m}x \qquad (18)$$

The exergy balance in a given system under steady state condition is generally given as,

$$\dot{X}_{heat} - \dot{X}_{work} + \dot{X}_{in} - \dot{X}_{out} - \dot{X}_D = 0 \qquad (19)$$

Where $\dot{X}_{heat}$ is the exergy associated with heat, $\dot{X}_{work}$ is the exergy associated with the work output of the system, $\dot{X}_{in}$ is the rate of exergy flow into the system, $\dot{X}_{out}$ is the rate of exergy exit from the system and $\dot{X}_D$ is the rate of exergy destruction. The exergy associated with a compression work is negative and it is equivalent to the actual compression work, while that associated with an expansion work is positive and is equivalent to the actual expansion work.

Exergy Analysis of the Simple Cycle GT Plant

The simple cycle plant consists of a compressor, combustion chamber and a turbine as in Figure 2(b). Exergy analysis is applied to each of these three components.

Exergy Analysis in the Compressor

The compression process is assumed to be adiabatic, hence $\dot{X}_{heat} = 0$. The exergy associated with the compression work is $\dot{X}_{work} = -\dot{W}_c$. The exergy in $\dot{X}_{in}$ is

$$\dot{X}_{in} = \dot{X}_1 = \dot{m}_a \{c_{p,a}(T_1 - T_o) - T_o[c_p \ln(T_1/T_o) - R\ln(p_1/p_o)]\} \quad (20)$$

The exergy out $\dot{X}_{out}$ equals $\dot{X}_2$ evaluated at T_{2a}. The exergy destruction rate in the compressor $\dot{X}_{D,C}$ and the exergetic efficiency $\eta_{II,C}$ are given in Equations (21) and (22) respectively as,

$$\dot{X}_{D,C} = \dot{W}_c + \dot{X}_1 - \dot{X}_2 \quad (21)$$

$$\eta_{II,C} = \frac{x_2 - x_1}{c_{p,a}(T_{2a} - T_1)} \quad (22)$$

Exergy Analysis in the Combustion Chamber

In the CC, fuel is supplied at temperature T_f and pressure p_f. Figure 4 is used in carrying out the energy balance in the combustion chamber.

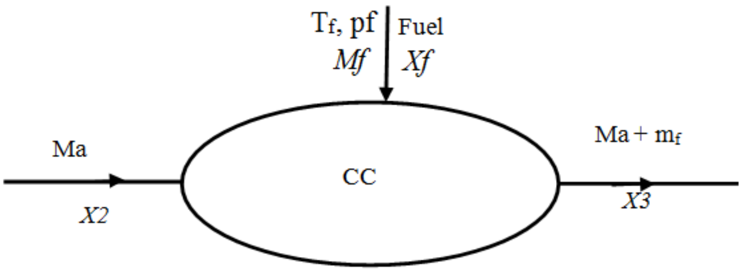

Fig-4: Exergy balance in the combustion chamber

In the CC, $\dot{X}_{heat} = 0$ and $\dot{X}_{work} = 0$.Thus,

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_D = 0 \quad (23)$$

$$\dot{X}_{in} = \dot{X}_2 + \dot{X}_f \quad (24)$$

$$\dot{X}_{out} = \dot{X}_3 \quad (25)$$

Where $\dot{X}_f$ is the exergy of the fuel which consists of two parts- physical exergy $\dot{X}_{ph}$ and chemical exergy $\dot{X}_{ch}$. The physical exergy is expressed as,

$$\dot{X}_{ph} = \dot{m}_f \{c_{p,f}(T_f - T_o) - T_o[c_{p,f} \ln(T_f/T_o) - R_f \ln(P_f/p_o)]\} \quad (26)$$

Where R_f is the gas constant of the fuel. The chemical exergy of the fuel is given as.

$$\dot{X}_{ch} = \xi_f LCV_f \quad (27)$$

where ξ_f is the exergy grade function defined for natural gas which contains mainly is methane (CH_4) in the form [20], [21],

$$\xi_f = 1.033 + 0.0169\left(\frac{b}{a}\right) - \frac{0.069}{a} \quad (28)$$

Where a is the number of carbon atoms and the b is the number of hydrogen atoms in the fuel.

The exergy destruction in the combustion chamber $\dot{X}_{D,CC}$ is given as,

$$\dot{X}_{D,CC} = \dot{X}_{in} - \dot{X}_{out} = \dot{X}_2 + \dot{X}_f - \dot{X}_3 \qquad (29)$$

The second law efficiency of the combustion process $\eta_{II,CC}$ is,

$$\eta_{II,CC} = \frac{\dot{X}_{out}}{\dot{X}_{in}} = \frac{\dot{X}_3}{\dot{X}_2 + \dot{X}_f} = 1 - \frac{\dot{X}_D}{\dot{X}_{in}} \qquad (30)$$

Exergy Analysis in the Turbine

Adiabatic expansion is assumed in the turbine hence, $\dot{X}_{heat} = 0$. Here, $\dot{X}_{work} = \dot{W}_t$,, $\dot{X}_{in} = \dot{X}_3$ and $\dot{X}_{out} = \dot{X}_4$. The exergy destruction can thus be expressed as,

$$\dot{X}_D = \dot{X}_3 - \dot{X}_4 - \dot{W}_t \qquad (31)$$

$\dot{X}_3$ And $\dot{X}_4$ can be computed as in Equation (20). The second law efficiency of the expansion process $\eta_{II,T}$ is,

$$\eta_{II,T} = \frac{c_{p,g}\left(T_3 - T_{4a}\right)}{x_3 - x_4} \qquad (32)$$

In the exhaust, no exergy is recovered; all the exergy is lost to the environment. The exhaust exergy loss is thus equivalent to $\dot{X}_4$. The second law efficiency η_{II} for the plant is,

$$\eta_{II} = \frac{\dot{W}_{net}}{\dot{X}_{in}} = \frac{\dot{W}_{net}}{\dot{X}_f} = \frac{\dot{W}_{net}}{\dot{X}_{ph} + \dot{X}_{ch}} \qquad (33)$$

Exergy Analysis of the Regenerative Cycle GT Plant

The regenerative cycle has one additional component (regenerator) in addition to the three basic components in the simple cycle system. The exergy analysis of the first three components in the regenerative cycle is the same as that of the simple cycle but the temperature at the inlet of the CC is T_{2Ra}. The exergy analysis here is thus limited to the regenerator.

Exergy Analysis in the Regenerator

Figure 5 shows exergy balance in the regenerator which will aid in the analysis. In the regenerator, there is no exergy associated with heat. Also, there is no work term. The exergy in and the exergy out from the regenerator are given respectively in Equations (34) and (35),

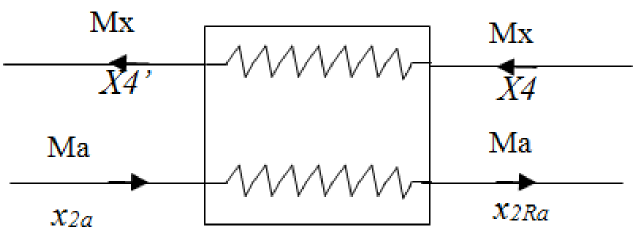

Fig-5: Exergy balance in the regenerator

$$\dot{X}_{in} = \dot{X}_2 + \dot{X}_4 = \dot{X}_{2a} + \dot{X}_{4a} \qquad (34)$$

$$\dot{X}_{out} = \dot{X}_{2Ra} + \dot{X}_{4'} \qquad (35)$$

The exergies at the respective state points are computed using the temperature values obtained. The only unknown temperature so far is $T_{4'}$ and this can be obtained from the energy balance of the regenerator and it is expressed as,

$$T_{4'} = \frac{\dot{m}_a c_{p,a} T_{2a} + \dot{m}_x c_{p,g} T_{4a} - \dot{m}_a c_{p,a} T_{2Ra}}{\dot{m}_x c_{p,g}} \qquad (36)$$

The exergy destruction in the regenerator $\dot{X}_{D,REG}$ and the second law efficiency of the regenerative plant $\eta_{II,REG}$ are expressed respectively in Equations (37) and (38),

$$\dot{X}_{D,REG} = \dot{X}_{in} - \dot{X}_{out} = \left(\dot{X}_{2a} + \dot{X}_{4a}\right) - \left(\dot{X}_{2Ra} + \dot{X}_{4'}\right) \qquad (37)$$

$$\eta_{II,REG} = \frac{\dot{X}_{2Ra} - \dot{X}_{2a}}{\dot{X}_{4a} - \dot{X}_{4'}} \qquad (38)$$

Like in the simple cycle plant, in the exhaust, no exergy is recovered. The exhaust exergy loss is equivalent to $\dot{X}_{4'}$ - evaluated at temperature $T_{4'}$.

Environmental Sustainability Indicators

The environmental sustainability indicators of both plants are compared. The environmental sustainability indicators are environmental sustainability index (ESI), depletion factor (DF), waste exergy ratio (WER) and environmental effect factor (EEF).

The ESI relates with the second law efficiency as [20],

$$ESI = \frac{1}{1 - \eta_{II}} \qquad (39)$$

Smaller values of ESI indicate greater negative impact of the operation of the gas turbine system on the environment. The DF applies to individual components in the system and is the ratio of the exergy destroyed in the system to the exergy input expressed as,

$$DF = \frac{\dot{X}_D}{\dot{X}_{in}} \qquad (40)$$

A given component in a system is adjudged to perform better if the DF is small. The waste exergy ratio (WER) is similar to the DF but it is applied to the entire system. The WER is defined as.

$$WER = \frac{\sum \dot{X}_D}{\dot{X}_{in}} = \frac{\sum \dot{X}_D}{\dot{X}_f} \qquad (41)$$

A small value of WER indicates better performance of the system. The EEF is the ratio of the waste exergy ratio to the second law efficiency of the system.

$$EEF = \frac{WER}{\eta_{II}} \qquad (42)$$

Smaller values of the EEF indicate lower impact of the plant operation on the environment.

RESULTS AND DISCUSSIONS

The basic features of the created engine model including isentropic efficiencies of the compressor and the turbine, combustion efficiency, comustion pressure loss and exhaust pressure loss are shown in Table 1.

Table-1: Results of field data in comparison with created engine model output

S/N	Parameter	Unit	Field Data	Engine Model Result	Percentage difference
1	Power Output	MW	35.5200	35.5185	0.004
2	Temperature of exhaust gas	K	822.77	825.41	0.321

The percentage difference between the power outputs is 0.004% while that for the exhaust gas temperatures is 0.321. Thus, the created engine model truly mimics the real engine in the field and the output from the model should be reliable. The created engine model has properties as presented in Table 2. These properties such as the compressor isentropic efficiency of 81.11% will definitely be different from the manufacturer's value (which is not available to the engine operators) because the compressor has been in use for several years and fouling and other degradation mechanisms might have set in and blade angles are different from the original values at the time of production.

Table-2: Basic features of the created engine model

S/N	Parameter	Symbol	Unit	Value
1	Isentropic Efficiency of the Compressor	η_{ci}	%	81.11
2	Isentropic Efficiency of the Turbine	η_{ti}	%	87.44
3	Combustion Efficiency	η_{cc}	%	97.00
4	Pressure Loss in Combustion Process	ΔP_{CC}	%	3.44
5	Pressure Loss in the Exhaust	ΔP_{EX}	%	5.00

Table-3: Thermal efficiencies of the simple and regenerative cycle plants

Thermal Efficiency (%)					
Simple Cycle	Regeneratve cycle				
	Regenerator Effectiveness (%)				
	80	85	90	95	100
35.19	43.65	44.09	44.45	45.01	45.48

The thermal efficiencies of the simple and regenerative cycle plants are presented in Table 3. The simple cycle thermal efficiency of 35.19% is lower than that of the regenerative cycle plant even at the lowest regenerative effectiveness value used.

Table-4: Fuel exergy parameters of the simple cycle

Parameter	Symbol	Unit	Value
Fuel parameter	ξ_f	-	1.03
Physical Exergy	$\dot{X}_{ph}$	kW	535.56
Chemical Exergy	$\dot{X}_{ch}$	kW	107338.50
Total Exergy	$\dot{X}_f$	kW	107874.05

Table-5: Exergy parameters of simple cycle

Component	Exergy Parameters			
	Exergy In (kW)	Exergy Out (kW)	Exergy Destruction(kW)	Exergetic Efficiency (%)
Compressor	0.00	49857.96	4725.09	91.34
CC	157732.01	119217.05	38514.96	75.58
Turbine	119217.05	31171.50	7304.03	91.70
Exhaust	31171.50	0.00	31171.50	0.00
2nd Law efficiency of the system	32.93			

The fuel exergy parameters are shown in Table 4 while other exergy terms are presented in Tables 5. The chemical exergy term dominates the physical exergy. The exergy destruction is highest in the CC, followed by the turbine. The exergy destruction is very high in the CC because of the high combustion temperatures and heat losses. The CC thus also has the lowest exergetic efficiency value compared to the compressor and the turbine. No exergy is recovered from the exhaust thus the exergetic efficiency is zero.

The exergy parameters of regenerative cycle system are shown in Table 6. The results are only shown for the combustion chamber, regenerator and the exhaust. Like in the simple cycle, the exergy destruction rate is highest in the CC and the value decreases with the regenerator effectiveness because more heating of the compressor exit gases occur with increase in the regenerator effectiveness. The lowest value of the exergetic efficiency obtained at 80% regenerator effectiveness is greater than that of the simple cycle plant. The exergy destruction rate in the regenerator decreases with increase in the regenerator effectiveness while the exergetic efficiency increases with increase in the regenerator effectiveness.

Table-6: Exergy parameters of regenerative cycle system

Exergy Parameter	Exergy Parameter Values				
	Regenerator Effectiveness (%)				
	80	85	90	95	100
COMBUSTION CHAMBER					
Fuel flow rate (kg/s)	1.72	1.70	1.68	1.66	1.65
Physical Exergy (kW)	431.54	427.21	422.87	418.54	414.21
Chemical Exergy (kW)	86490.29	85622.10	84753.92	83885.73	83017.55
Total Exergy (kW)	86921.83	86049.31	85176.79	84304.27	83431.76
Exergy In (kW)	144552.72	144187.37	143824.30	143463.49	143104.90
Exergy Out (kW)	119217.05	119217.05	119217.05	119217.05	119217.05
Exergy Destroyed	25335.68	24970.32	24607.25	24246.44	23887.85
Exergetic Efficiency	82.47	82.68	82.89	83.10	83.31
REGENERATOR					
Exergy In	81029.46	81029.46	81029.46	81029.46	81029.46
Exergy Out	80434.91	80475.21	80520.48	80570.73	80625.95
Exergy Destroyed	594.54	554.24	508.97	458.73	403.50
Exergetic Efficiency	92.89	93.73	94.53	95.30	96.05
EXHAUST					
Exergy In	22804.02	22337.16	21872.97	21411.51	20952.81
Exergy Out	0.00	0.00	0.00	0.00	0.00
Exergy Destroyed	22804.02	22337.16	21872.97	21411.51	20952.81
2^{nd} Law Efficiency of the system	40.86	41.28	41.70	42.13	42.57

The depletion factors of the simple cycle gas turbine components are presented in Figure 6 while those of the regenerative cycle components are shown in Figure 7.

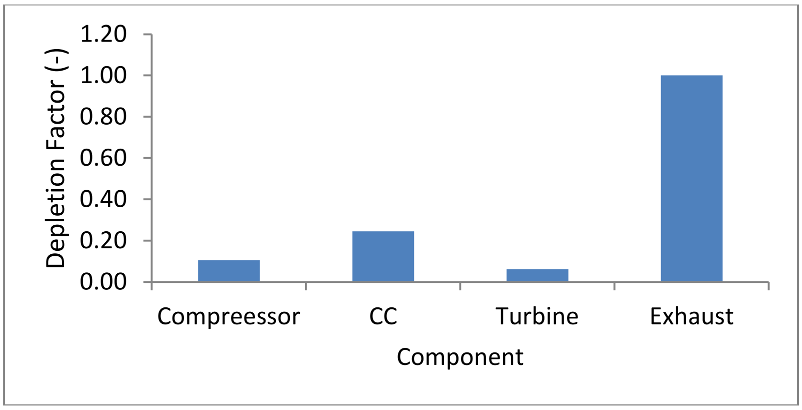

Fig-6: Depletion factors of simple cycle gas turbine components

The DF indicates the level of exergy destruction in relation to the exergy input into each component. Since in the exhaust all the exergy in is destroyed, the depletion factor is unity. The CC has the highest value of depletion factor compared to the compressor and the turbine. The depletion factors for the regenerative cycle are shown for the regenerator and the CC for different values of the regenerator effectiveness. The depletion factors of both components decreases with increase in the regenerator effectiveness as less exergy is available for destruction as the regenerator effectiveness increases. The depletion factors of the CC in the regenerative cycle plant are all lower than that obtained in the simple cycle. Thus, in terms of exergy destruction rate, the regenerative cycle is performing better than the simple cycle plant. The other eenvironmental sustainability indicators of both plants are presented in Table 7. The ESI in the simple cycle is smaller than those in the regenerative cycle while the WER and the EEF are greater in the simple cycle. Greater value of ESI indicates lower negative impact of the engine operation on the environment while higher values of WER and EEF indicates greater negative impact of engine operation on the environment. Thus, judging from the results obtained, the environment is safer with the operation of the regenerative cycle plant.

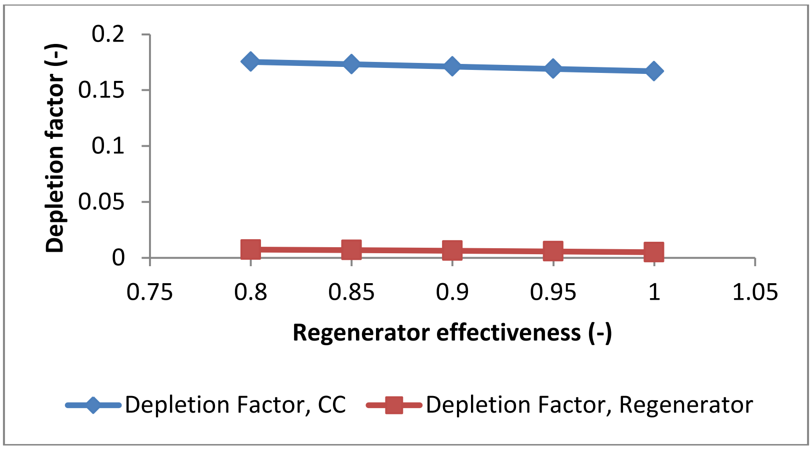

Fig-7: Depletion factor for regenerative cycle components at different values of regenerative effectiveness

Table-7: Environmental sustainability indicators of both plants

Environmental Sustainability Indicators	Cycle Type					
	Simple Cycle	Regenerative Cycle				
		Regenerative Effectiveness (-)				
		0.8	0.85	0.9	0.95	1
ESI (-)	1.3201	1.4305	1.4368	1.4432	1.4499	1.4567
WER (-)	0.7575	0.6991	0.6960	0.6929	0.6897	0.6865
EEF (-)	3.1239	2.3229	2.2895	2.2562	2.2228	2.1895

CONCLUSION

Comparative exergo-environmental analysis of simple and regenerative cycle gas turbine power plants was carried out in this work. The thermal efficiency of the simple cycle plant is lower than that of the regenerative cycle plant at all values of regenerative effectiveness. This implies less amount is spent on fuel in the regenerative cycle plant. Judging from the exergy destruction values at the components level using the depletion factor, the depletion factor of the combustion chamber at 80% regenerative effectiveness is lower than that of the simple cycle. The environmental sustainability indicators obtained for both plants indicate that the environment is safer with the operation of the regenerative cycle plant as against the simple cycle plant. For instance, the environmental sustainability index of the simple cycle plant is also lower than that of the regenerative cycle plant at 80% regenerative effectiveness. That is, the environment has less harmful effects from the operation of the regenerative cycle plant compared to the simple cycle. In essence, it was observed that every performance indicator is in favour of the regenerative cycle plant with more favourable results obtained as the regenerator effectiveness increases. Further analysis should take into account the economics of plant operation.

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