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Exergy Based Thermodynamic Performance Analysis of Gas Turbine Power Plant in Nigeria.

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ABSTRACT*: In this work a study of the exergy analysis of a gas turbine power plant was done, operational parameters from the gas turbine plant which is located at Onne , Rivers State, south – south region of Nigeria were used to evaluate the plant performance in terms of exergy inflow, exergy outflow, exergy losses, total exergy generated and the exergetic efficiency to each components of the gas turbine such as the compressor, combustion chamber, and turbine, Results obtained shows that exergy loss of 5.8211MW, 24.1185MW, 8.4069MW and 0.4410MW occurs in the compressor unit, combustion chamber, turbine section and the exhaust part of the gas turbine power plant respectively. Thus, the lost in the combustion chamber was found to be highest which is attributed to high irreversibilities due to fluctuation of gas pressure from the supply source. Also, the exergetic efficiency was found to be 43.32%, 74.83%, 74.43% for the compressor unit, combustion chamber, turbine section of the gas turbine power plant respectively. These values depict the percentage of maximum useful work that can be done by the system (components) during the operation of the plant in course of interacting with its environment which is at a constant pressure and temperature, Exergetic efficiencies were also gotten at varying ambient temperatures, and it was found out that the exergetic efficiencies in the respective gas turbine components decreases with increase in the ambient temperature. This is due to high irreversibilities in the combustion chamber and then the turbine. Therefore, in order to improve the performance of this existing gas turbine power plant, a proper maintenance of the filtration system (inlet Air Filters) as well as intercooling of compressor stage should be carried out periodically, also an effective and adequate lubrication plan should be put in place, while a combined cycle will adversely reduce the high exergy losses incurred in the exhaust pipe, by reducing the effect of thermal pollution.*

KEYWORDS:*Exergy, Performance, Destruction, Efficiency, Power Plant, Turbine, Combustion Chamber*

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I. INTRODUCTION

Energy is the ability to do work. Its availability is an important requirement for human existence. It is obvious that The social and economic development of a country is tied to energy availability and accessibility via its resource and generation mechanisms hence Because of the high increase in the demand and consumption of energy, it becomes necessary that power generating plant should be operated at a high performance level in terms of its efficiency and economic perspective.It is therefore necessary to understand the mechanism which degrade the quantity and quality of energy and to develop systematic approach as to evaluate the plant performance and optimization of the system. Over years, several principles and method has been adopted in analyzing the performance and the optimization of a system. Many researchers use the conservation of mass and the conservation of energy (first law of thermodynamic) principles in evaluating system performance. Similarly, exergy approach is a recent method of growing interest in evaluating the performance of thermodynamic processes in relation to energy generation and utilization. the exergy-based performance analysis is the performance analysis of a system based on the Second Law of thermodynamics.

The aim of this research work is to carry out exergy analysis of a gas turbine power plant inspired by Notore gas turbine power plant. And this is achieved through the following objectives:

i. To Identify where exergy destruction and losses occur in the system as well as to determine their level of loss significance.

ii. To evaluate the exegetic efficiency (2nd law efficiency) of each component of the gas turbine power plant.
iii. To recommend possible ways of system performance optimization.

[1] Hart (1999) used exergy-based method to analyze the performance of Afam (IV) thermal power plant in Rivers State. Hourly readings of parameters of the Afam (IV) power plant were collated for a period of one year (1992) from the control room log sheets/books. Average mean values of these readings were statistically analyzed and computed. From appropriate thermodynamic tables working fluid parameters were obtained. From standard thermodynamic equation the volume and mass flow rate of natural gas and air were computed.

The goal for plant performance analysis is to understand accurately the operation of the various units [2] (Rosen and Dincer, 2002). A gas turbine operates efficiently when working under its engine design points configuration and at various components optimum performance levels (3) (Sanjay and Rajav, 2009).
[4] Kwanbai (2005) conducted a research on exergy based method analysis of Olkaria I. Geothermal Power Plant in Kenya. The aim of the work was to analyze the plant overall efficiency and as well identify processes or points where exergy is lost or wasted.

[5] Ebadi and Gorji-Bandpy (2005), used exergy based method to analyzed the performance of a 116 MW gas turbine located in Mahshahr, Iran. Mass and energy conservation laws were applied to each component of the system. Quantitative exergy balance was done for the system on each component and for the whole system; the exergy of the system was decomposed into thermal, chemical, mechanical exergy and entropy – exergetic efficiency and exergy wasted in the plant was evaluated.

Thus, because of the limitations in energy reserves/resource it is of necessity that systems or plants of high efficiency operating performance be designed and developed. Hence, in the power generation industries the advent of gas turbines in recent decades attracted focus of most researcher. But according to Kelvin-Planck Statements of the second law of thermodynamics, which is expressed as follows: it is impossible for any devices that operate on a cycle to receive heat from a single reservoir and produce a net amount of work. That is, no heat engine can convert all the heat it receives to useful work but must exchange heat with a low-temperature sink as well as a high temperature source to keep operating. This means that no heat engine can have a thermal efficiency of 100 percent, or as for a power plant to operate, the working fluid must exchange heat with the environment as well as the furnace [6] (Walter et al, 2009).

II. METHODOLOGY

2.1.Data Collection

The data used for this study were obtained from an operational unit of a 25MW Industrial gas turbine plant located at Onne, Rivers State, south–south region of Nigeria. The values of the input parameters used for this study were obtained from the log sheet for an interval of twelve (12) months (September 2017- August 2018). The parameters were recorded at various average values daily, during the operation of the plant for a period of one (1) month.

2.2. General Overview of Notore's Gas Turbine Power Station

The power station is located in Onne close to Federal ocean terminal (FOT), Rivers State in south – south region of Nigeria. The plant has an installed generating capacity of 25 MW. It utilizes natural gas as its combustion fuel which it gets directly through piping networks from Nigeria Gas Company (NGC);The Gas power station consists of the following characteristics Turbine serial No (T – 206),Plant Model (MW – 251),Plant type(Open cycle single shaft),Power rating (25MW)Compressor stage(18)Turbine stages (3),AtmosphericAirtemperature(30^{o}C),Inlet air filter type single – stage, self-clearing ,Fuel gas strainers type (100A (4B) Y – Type)Strainer meshes(60)Fuel type (natural gas)Mass flow rate of air (142.6kg/s)Mass flow rate of exhaust gas (110.2 kg/s)Specific heat of exhaust gases (1.21kJ/kg.k)Specific heat of air of constant pressure (1.005kJ/kg.k).

2.3 Thermodynamic Operational Principle of the Plant

The gas turbine chosen for this study is said to operate on the principle of a sample Brayton System. It consists of an axial air compressor unit a combustion chamber, flow turbine and auxiliary equipment required to operate it. As show in figure 1,The T – s diagram is represented in Figure 1b showing the losses due to inefficiencies of the components of actual open cycle gas turbine plant.

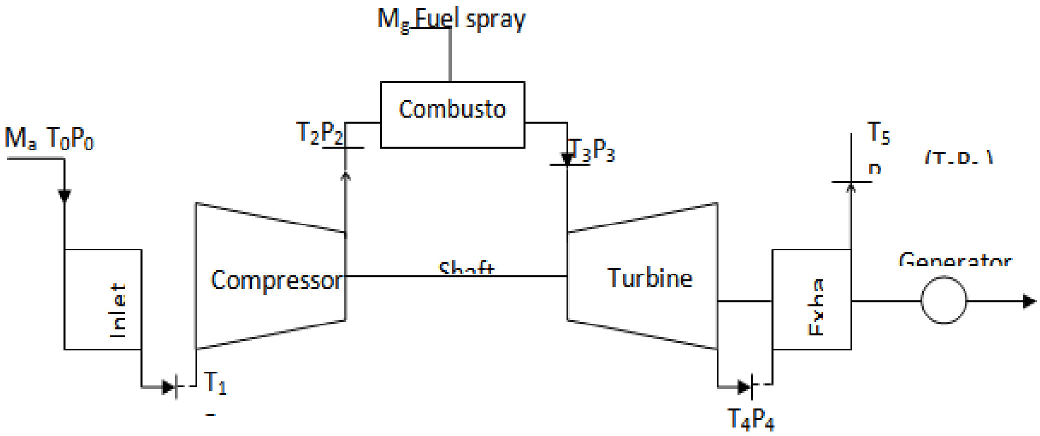

Fig. 1a: Schematic of the Gas Turbine Open Loop Components Arrangement

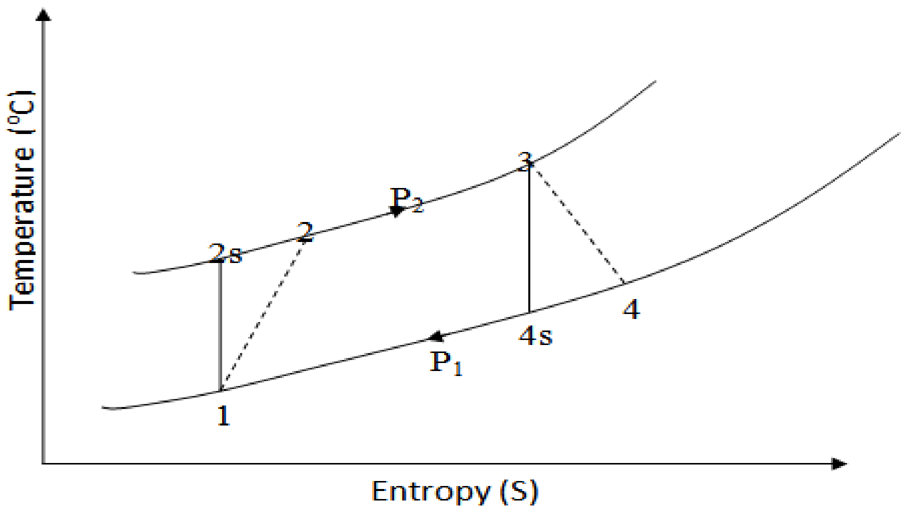

Fig.1b: Gas Turbine Cycle for Exergy Analysis

2.4. Formulation of Exergy Balance Equation (Component Exergy Analysis)

A general exergy balance equation, applicable to any component of a gas system can be formulated by utilizing the First and Second Laws of Thermodynamics. The exergy balance is given byAccording to [7] Wark and Richard (1988) an exergy balance equation for a control volume is given by;

$$\Sigma W = \Sigma Q + \Sigma \mathrm{E}_{x,out} - \Sigma E_{x,in} + \Sigma \mathrm{E}_{x,destroyed} \qquad (1)$$

2.5 Compressor Unit:

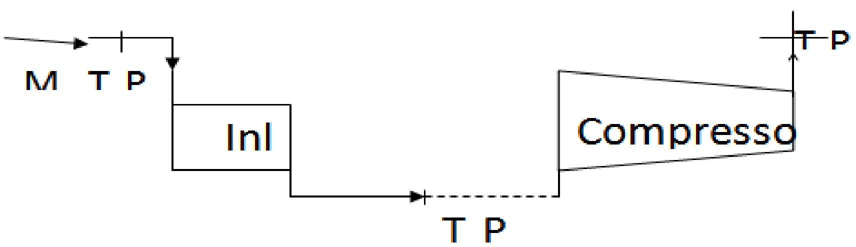

Fig. 2: Schematic of compressor unit and state points

Extracting from figure 2 the compressor unit and the state points as shown in figure 1b and applying equation 1 Implies that, -

$W_c = \sum E_{xin} - \sum E_{x,out} E_{x\ loss\ comp.irr.}$ (2)

Where;

W_c = Max work done in the compressor

$\sum E_{xin}$ = Exergy rate entering the compressor

$\sum E_{xout}$ = Exergy rate leaving the compressor

$E^{o}_{x,Losscomp\ .irr.}$ = Exergy loss due to irreversibilities in the compressor.

Then,

- Exergy inflow to the compressor is evaluated as:

$E_{xin} = Ma\,[(h_1 - h_0) - T_o(s_1 - s_o)]$ (3)

- Exergy outflow will be,

$E_{xout} = Ma\,[(h_2 - h_0) - T_o(s_2 - s_o)]$

- Exergy loss due to irreversibilities in the compressor will be

$E_{xlosscomp\ .} = T_o\Delta S = To(s2 - s1) =$

$M_a T_o \left[Cp_a In\frac{T_2}{T_1} - R_a In\left(\frac{P_2}{P_1}\right)\right]$

- Mechanical loss due to irreversibilities $M_{x,loss} = W_{c,Actual} - W_{c,ideal}$ (4)
- Isentropic efficiency of compressor $\eta_c = \frac{Idealwork}{Actualwork} = \frac{W_{c,ideal}}{W_{c,Actual}}$ (5)

Hence,

- Total exergy loss in the compressor will be

$$\sum E_{xloss} = \left(\frac{m_a Cp_a(T_2 - T_1)}{\eta_c} - m_a Cp_a(T_2 - T_1)\right) + m_a T_0\left[Cp_a In_0\frac{T_2}{T_1} - R_a In\left(\frac{P_2}{P_1}\right)\right] \quad (6)$$

Exegetic efficiency (of the compressor) $\eta_{e,comp\ .} = \frac{E_{x,output}}{Wc - (E_{x,out} + E_{x,loss})}$

2.6 Combustion Chamber:

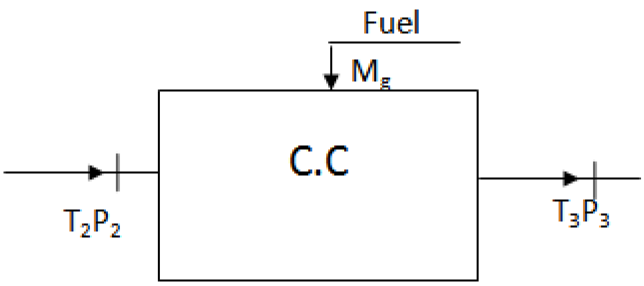

Fig. 3: Schematic of combustion chamber and state points

Considering fig. 3.4 Exergy balance equation will be

$\sum E_{xln} + \sum E_{xfuel} = \sum E_{xout} + \sum E_{x,loss}$ (7)

where

$\sum E_{xin}$ = Exergy of stream entering the Combustion Chamber = $E_{x,ouput}$ of compressor

$\sum E_{x,out}$ = Exergy rate of stream leaving the combustor

$\sum E_{x,fuel}$ = Exergy of fuel, $\sum E_{xloss}$ = Total exergy loss in the Combustion chamber

- Exergy inflow to combustion chamber is

$E_{x,in} = M_a[(h_2 - h_0) - T_0(s_2 - s_0)] = m_a\left[Cp_a(T_2 - T_0) - T_o\left(Cp_a In\left(\frac{T_2}{T_0}\right) - R_a In\left(\frac{P_2}{P_0}\right)\right]\right.$ (8)

- Exergy outflow from combustion chamber is 0

$E_{x,out} = M_g[(h_3 - h_0) - T_0(s_3 - s_0)] = m_g\left[Cp_g(T_3 - T_0) - T_o\left(Cp_g In\left(\frac{T_3}{T_0}\right) - R_g In\left(\frac{P_3}{P_0}\right)\right]\right.$ (9)

- Exergy loss in the combustor due to irreversibilities,

$E_{x,loss} = T_0\Delta S = T_0(s_3 - s_2) = m_g T_0\left[\left(Cp_g In\left(\frac{T_3}{T_2}\right) - R_g In\left(\frac{P_3}{P_2}\right)\right]\right.$ (10)

where

Combustor Exegetic efficiency $\eta_{\varepsilon,comb.} = \frac{E_{x,output}}{E_{xinput}} = \frac{E_{x,out}}{E_{x,in} + E_{x,fuel}}$

2.7 Turbine Unit

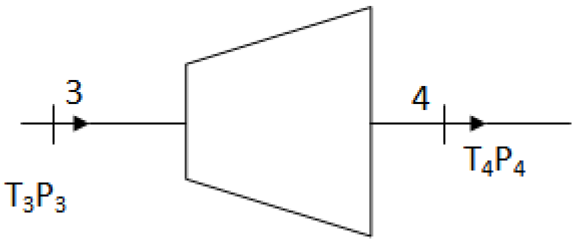

Fig. 4: Schematic of Turbine Unit and State Points

Applying the exergy balance equation.

$E_{x,in} = E_{x,out} + E_{xT} + E_{x,lossirr}$ (11).

where

$E_{x,in}$ = Exergy of stream entering the turbine unit = $E_{x,out}$ of Combustion Chamber

$E_{x,out}$ = Exergy rate of stream leaving the turbine unit

$E_{x,T}$ = Work done by the turbine (there is a mechanical loss attached)

$E_{x,loss,\ irr.}$ = Exergy loss due to irreversibilities

But,

- Exergy inflow to turbine is;

$E_{x,in} = m_g(h_3 - h_0) - To\ (s_3 - s_0) = M_g\left[Cp_g(T_3 - T_0) - T_0\ (Cp_g\left(\frac{T_3}{T_0}\right) - R_g In\left(\frac{P_3}{P_0}\right)\right]$ (12)

- Exergy outflow of turbine is

And $E_{x,out} = m_g(h_4 - h_0) - To\ (s_4 - s_0) = M_g\left[Cp_g(T_4 - T_0) - T_0\ (Cp_g In\left(\frac{T_4}{T_0}\right) - R_g In\left(\frac{P_4}{P_0}\right)\right]$

13)

- Exergy loss due to irreversibilities in the turbine is;

$E_{x,loss,irr.} = T_o\Delta S = T_0\ (s_4 - s_3) = M_g T_o\left[Cp_g In\left(\frac{T_4}{T_3}\right) - R_g In\left(\frac{P_4}{P_3}\right)\right]$ (14)

Rearranging equation 3.29, implies that

$E_{xT} = E_{x,in} - E_{x,out} - E_{x,loss}$ (15)

But,

- Total exergy loss in the turbine will be

Total $E_{x,loss} = E_{x,loss}$ + Mech. Exergy loss (16)

- Mechanical exergy loss in the turbine is

Mech. Exergy loss = $W_{T,Actual} - W_{T,Ideal}$ (17)

- Ideal work, $W_{T,Ideal,} = m_g\ (h_4 - h_3) = m_g Cp_a\ (T_3 - T_4)$ (18)
- Actual work, $W_{T,Actual} = \eta_T + W_{T,Ideal} = \eta_T m_g Cp_g (T_3 - T_4)$ (19)
- Thus, total exergy loss in the turbine will be

Total $E_{x,loss} = E_{x,lossirr.}$ + Mech.,Exergy loss (20)

- Turbine exergetic efficiency $\eta_{\varepsilon T} = \frac{E_{x,desired}}{E_{x,input}} = \frac{E_{x,out} + E_{xT}}{E_{x,in}}$ Nag (2013) (21)

2.8 Exhaust Port

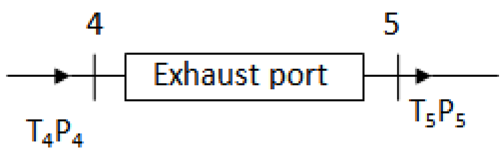

Fig. 5: Schematic of Exhaust port and state points

- Applying exergy balance equation to the exhaust port,

$E_{x,in} = E_{x,out} + E_{x,loss}$ (22)

But,

$Ex_{in} = Ex_{,out}$

- Exergy loss due to irreversibilities is

$E_{x,loss} = M_g Cp_a\left\{(T_{exht} - T_0) - T_0 In\left(\frac{T_{-EXht}}{T_0}\right)\right\}$

III. RESULTS AND DISCUSSION

3.1 Results

The parameters pertinent for this thermodynamics performance analysis (exergy based) are those measured values such as pressure, temperature,mass flow rates recorded at various state points during the operation of the gas turbine plant ascertained in the period under investigation , a summary of the operating conditions and parameters for this period is shown in Table 1a.

Table 1a Operating parameters of the gas turbine power plant (Sources:/offsite control room).

Months [20017/2018]	**T1T2 [K][K]**		**T3 [K]**	**T4P1P2 [K]**	**ma [bar]**	**mg [bar]**	**[kg/s]**	**[kg/s]**
Setember	292.76	548	1296.1	743	1.013	8.52	142.6	110.2
October	292.8	548.46	1296.1	743.15	1.013	8.52	142.6	110.2
November	293.5	549.49	1298	743.2	1.013	8.53	142.6	110.2
December	294	550.15	1298.2	743.3	1.013	8.54	142.6	110.2
January	294	550.8	1298.8	743.5	1.013	8.52	142.6	110.2
February	295.5	551.1	1299	743	1.013	8.53	142.6	110.2
March	296	551.2	1299.1	743.3	1.013	8.54	142.6	110.2
April	299	551.46	1299.1	743.55	1.013	8.52	142.6	110.2
May	300	553.49	1300	743.3	1.013	8.53	142.6	110.2
June	300.8	555.15	1300.7	743.5	1.013	8.54	142.6	110.2
July	301	555.7	1301.2	743.6	1.013	8.55	142.6	110.2
August	301.5	556.81	1301.9	743.8	1.013	8.56	142.6	110.2

A MATLAB code was written to generate the results for the analysis, hence The results presented in Figure5b shows the Exergy loss**,** Exergetic efficiency**,** of each component over the total exergy of the system .

Table 1b Summary of Results of Components Exergy Analysis Evaluated

Components	**Exergy loss**	**Exergetic efficiency**	**Total Exergy**
Compressor	5.8211	43.3194	12.3561
Combustion	24.1185	74.8291	58.073
Turbine	8.4069	74.4257	18.337
Exhaust	0.441		

3.2 Discussion

3.2.1 Variation of exergy loss in the gas turbine components

Figure 6 shows that exergy loss of 5.8211MW, 24.1185MW, 8.4069MW and 0.4410MW occurs in the compressor unit, combustion chamber, turbine section and the exhaust port of the gas turbine power plant respectively. Thus, the lost in the combustion chamber was found to be highest, which is attributed to high irreversibilities due to fluctuation of gas pressure from the supply source

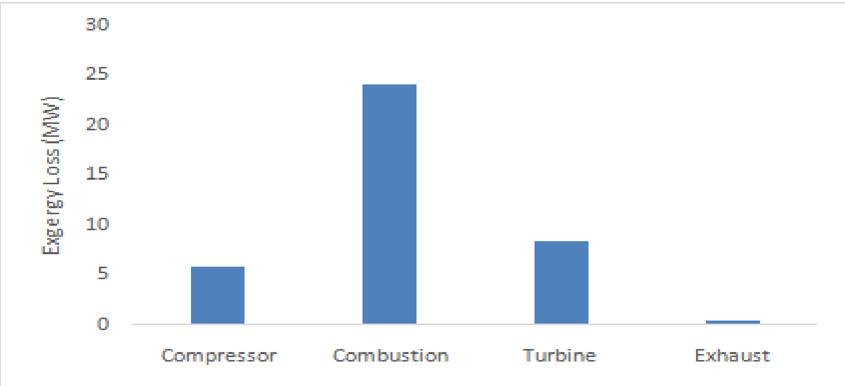

Fig. 6: Degree of Exergy Loss in the various Gas Turbine Components

3.2.2 Variation of exergetic efficiency in the gas turbine components

The exergetic efficiencies of the various components of the plant as summarized in Table 2 and shown in Figure 7 indicates that the compressor work ability to produce work was found to be 43.32%, that of the combustion process was 74.83% and for the turbine unit its exergetic efficiency was 74.43%. these values depict the percentage of maximum useful work that can be done by the system (components) during the operation of the plant in course of interacting with its environment which is at a constant pressure, P_0 and temperature, $T_{0.}$ for the exergetic efficiencies of the compressor, combustion, turbine respectively, as shown in Figure 7.

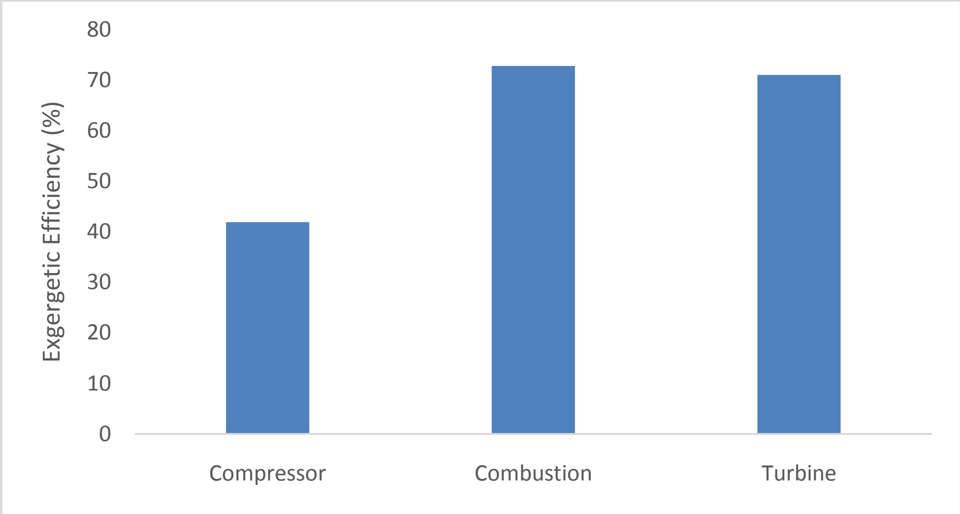

Fig. 7: Gas turbine components exergetic efficiencies

3.2.3 Variation of total exergy in the gas turbine components

Figure 8ashows the total exergy in the various components of the gas turbine plant such as the compressor, combustion and the turbine. 12.36MW, 58.07MW and 18.34MW are the results gotten from the matlab program for the compressor, combustion and the turbine sections of the gas turbine plant. The results show that the combustion have the highest total exergy followed by the turbine sections and then the compressor.

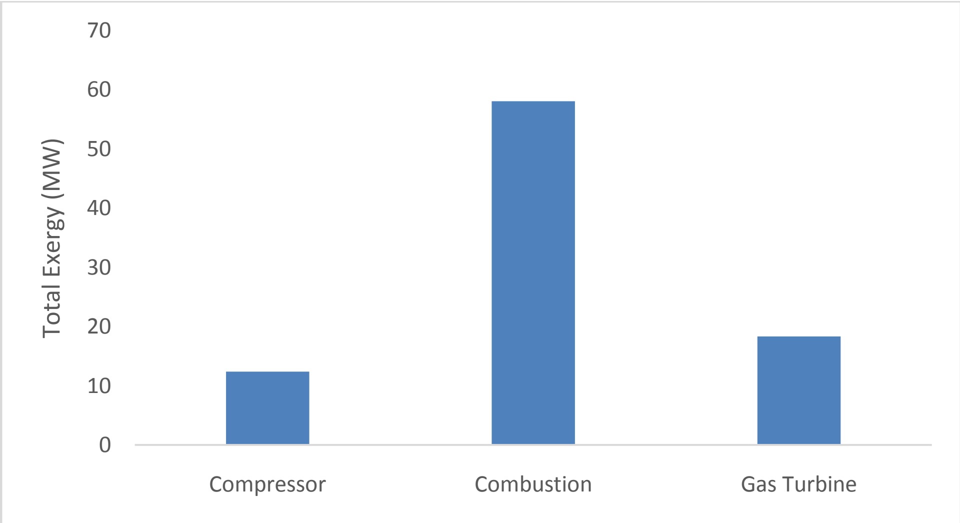

Fig. 8: Total exergy in the various components of the gas turbine plant

3.2.4 influence of ambient temperature on exergetic efficiency of the plant

Figure 9 compares the exergetic efficiencies of the different components of the gas turbine at different ambient temperatures. It shows that the exergetic efficiency in the respectictive gas turbine components decreases with increase in the ambient temperature. This is due to high irreversibilities in the combustion chamber and then the turbine.

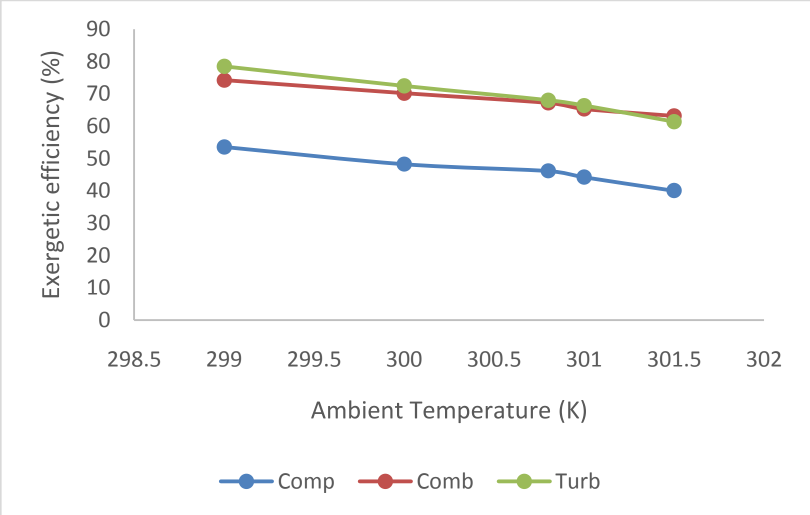

Fig. 9: Exergetic efficiency variation with ambient temperature

3.2.5 Flow exergy stream in the various gas turbine components

From the result of the exergy analysis of the plant and as shown in Appendix C, the exergy input due to compressor work (Actual compressor work) was 43.57MW, the exergy of stream leaving the compressor was 37.75MW and the exergy loss in the compressor was 5.82MW. The high exergy loss and compressor work input in the compression process is tied to inadequate mass flow rate of air intake to the compressor which resulted as from malfunctioning of the air filtration system due to blockages and collapse of filter elements as dirt might have impinched on the compressor blades and passages thus obstructing the compression process.

In the combustion process, the total exergy of the stream entering the combustion chamber was found to be 37.75MW of which 71.70MW is the flow exergy of stream out of the combustion chamber and 58.073MW was exergy of fuel for the burning process. The result also revealedthat24.12MW was found to be exergy lost in the combustion chamber, which is attributed to high irreversibilities due to fluctuation of the gas pressure from its primary source, inefficiency of operation and process control equipment and metallurgical limitations.

Similarly, the result of the turbine unit exergy analysis shows that 71.70MW was found to be the total exergy of stream entering the turbine, with a 23.84MWexergy drop in the turbine (i.e. Actual Mechanical Work Loss) and that the total exergy leaving the turbine field to frictional losses, heat generation and inter-stage seal leakages.

IV. CONCLUSION

An Exergy Based Thermodynamic Performance Analysis of a Gas Turbine Power Plant was carried out and the locations and quantities of exergy losses in the different processes were discovered hence The exergy balance applied to each of the major components of Plant and to the overall plant revealed the amount of the total exergy generated and exergy destruction in the station.. In addition, The results from the study also showed that considerable exergy destruction occurs in the combustion chamber; exergy efficiency, exergy destruction. power output depend on ambient temperature, i.e increase in ambient temperature decreases Exergetic efficiency.

Nomenclature

Symbol	Description	Unit
$\sum Q$	Sum of heat supplied	KJ/kg
$\sum w$	Sum of ideal work	MW
ΔS	Change in entropy	KJ/kgk
Cp_a	Specific heat of air at constant pressure	KJ/kg k
Cp_g	Specific heat of flue gas at constant pressure	KJ/kg k
E_{CHM}	Chemical exergy	MW
E_{KE}	Kinetic exergy	MW
E_{PE}	Potential exergy	MW
E_{PH}	Physical exergy	MW
E_{th}	Thermal exergy input	MW
E_{Total}	Total exergy of the system	MW
$E_{x,\ destroyed}$	Sum of exergy lost in the system due to irreversibilities	MW
$E_{x,desired}$	Sum of useful exergy outputs	MW

$E_{x,fuel}$	Exergy of fuel	MW
$E_{x,in}$	Sum of exergy inflow	MW
$E_{x,losscomb.irr.}$	Exergy due to irreversibility in the combustor	MW
$E_{x,out}$	Sum of exergy outflow	MW
$E_{x,T}$	Work done by the turbine	MW
h_0	Specific enthalpy of reference point	KJ/kg
h_1	Specific enthalpy of stream entering the compressor	KJ/kg
h_2	Specific enthalpy of stream leaving the compressor	KJ/kg
h_3	Specific enthalpy of stream entering the turbine	KJ/kg
h_4	Specific enthalpy of stream leaving the turbine	KJ/kg
h_5	Specific enthalpy of stream leaving the exhaust port	KJ/kg
m_a	Mass flow rate of air entering the compressor	Kg/s
$Mech_{exergyloss}$	Mechanical exergy loss	MW
m_g	Mass flow rate of exhaust flue gases	Kg/s
$M_{x,lossirr.}$	Mechanical loss due to irreversibilities	MW
P_0	Constant pressure of stream at references state	Bar
P_1	Pressure of air at compressor inlet	Bar
P_2	Pressure of compressed air at combustor inlet	Bar
P_3	Pressure of flue gases at turbine inlet	Bar
P_4	Pressure of flue gases at turbine outlet	Bar
P_5	Pressure of flue gas at exhaust port	Bar
R_a	Specific air constant	KJ/kg
R_g	Specific gas constant	KJ/kg
S_1	Specific entropy of stream entering the compressor	KJ/kgk
S_2	Specific entropy of stream leaving the compressor	KJ/kgk
S_3	Specific entropy of stream entering the turbine	KJ/kgk
S_4	Specific entropy of stream leaving the turbine	KJ/kgk
S_5	Specific entropy of stream of at the exhaust port	KJ/kgk
S_o	Specific entropy of stream at reference state	KJ/kgk
T_1	Temperature of stream entering the compressed	^{O}C/k
T_2	Temperature of stream leaving the compressor	^{O}C/k
T_3	Temperature of stream entering the turbine	^{O}C/k
T_4	Temperature of stream leaving the turbine	^{O}C/k
T_5	Temperature of stream at the exhaust port	^{O}C/k
T_o	Temperature of stream at the reference state	^{O}C/k
W_c	Maximum work done in the compressor	MW
$W_{c, Ideal}$	Ideal compressor work	MW
$W_{c,Actual}$	Actual compressor work	MW
$W_{T,Actual}$	Actual turbine work	MW
$W_{T,Ideal}$	Ideal turbine work	MW
Greek symbol	**Descriptive**	**Unit**
η_ε	Exergetic efficiency	%
$\eta_{\varepsilon c}$	Compressor exegetic efficiency	%
$\eta_{\varepsilon T}$	Turbine Exegetic Efficiency	%
η_{th}	Thermal Efficiency	%

Abbreviation

C.C combustion chamber
NPHR Net plant heat rate
TDI Thermal discharge index
T-S Temperature/Entropy
1-D 1-Dimensional
SFEE Steady flow exergy Equation
HMI Human machine interface
MW Mega Watt
Kg kilogram
Kg/s Kilogram per seconds
KJ/kg Kilojoules per Kilogram
% Percentage
^{o}C Degree Celsius
k Kelvin

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