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Gas Turbine Performance Improvement through Cycle Modification.

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ABSTRACT

The Gas Turbine is unquestionably one of the most important inventions of the 20th century, which has been utilized across major industries which includes the power industry, aircraft sector, and technological industry. But the Gas Turbine has its own limitation like deterioration in performance during operation leading to reduce capacity and thermal efficiency, this led to the modification of gas turbine increase power and thermal efficiency. Therefore this work was performed on a simple gas turbine "AES BARGES" using the regenerative method and excel software to calculate and compute results.

Result shows that after employing thermodynamics model to analyse the thermal efficiency, heat addition, specific fuel consumption and heat rate of a simple gas turbine, and further converting to simple gas turbine to regenerative with regenerative effectiveness of 0.8 to 1.0, the thermal efficiency was seen to be constant, the thermal efficiency for a regenerative cycle increase linearly while the heat addition, specific fuel consumption and heat rate reduces linearly as regenerative effectiveness increases. At regenerative effectiveness of 0.8 - 0.89, the thermal efficiency of the simple gas turbine is higher compare to the regenerative cycle, but from regenerative effectiveness of 0.90 – 1, the thermal efficiency of the regenerative cycle is higher compares to the simple gas turbine

Keywords: Gas Turbine, Power Plant, Thermal Analysis, Regeneration Effectiveness, Brayton Cycle.

1.0 INTRODUCTION

The gas turbine is an internal combustion engine that burns gaseous or liquid fuel. It delivers the energy of this combustion either by means of thrust, in an aircraft jet engine; or – by means of a rotating shaft – it provides electricity via a generator, or mechanical power for driving compressors or ship propulsion (Jansohn 2013).

The first gas turbine built in 1903 by Aegidius Elling, a Norwegian using rotary dynamic compressor and turbine marked the beginning of the gas turbine. This effort was credited with the building of the first gas turbine power plant that produced 8kw. (Jeffs 2008). This design was further improved by Elling in 1904 to operate at about 20000 rpm and achieve about 33kw with an exhaust gas temperature of 773K (Razak 2007). In 1949, the first electric utility gas turbine built by General Electric (GE) Company as a part of a combined cycle gas turbine plant was installed in Oklahoma (USA) with power rating of about 3.5MW.(Ragland 1998).Until the mid-70s, the efficiency and reliability of this system were consistently low and poor(Allouis et al 2016) In the 1990s, GE manufactured Gas turbines with a pressure ratio of 13.5, power rating of 135.7 MW, and thermal efficiency of 33 % under a normal cycle operation. Recently, GE manufactured a Gas turbine that produced power of up to 300 MW at a turbine inlet temperature of 1425℃ and a thermal efficiency 40 % under a normal cycle mode (Awad et al 2017)

The gas turbine is unquestionably one of the most important inventions of the 20th century, and it has changed our lives in many ways. The first important application of the gas turbine was the development of the military jet engine towards the end of the Second World War, when it provided a step change in speed from the existing propeller-driven aircraft. These early engines were fuel inefficient, unreliable and extremely noisy, but in less than 20 years they had matured to become the standard form of propulsion for civil aircraft. By the early 1970s, continuous development led to the development of the high bypass ratio turbo fan and the major improvement in fuel efficiency made the high-capacity wide-body airliner possible. The resulting gains in productivity and economics were remarkable and opened up air travel to the masses; the gas turbine has found increasing service in the past 40 years in the power industry both among utilities and merchant plants as well as the petrochemical industry, and utilities throughout the world. Its compactness, low weight, and multiple fuel application make it a natural power plant for offshore platforms. Today there are gas turbines, which run on natural gas, diesel fuel, naphtha, methane, crude, low-Btu gases, vaporized fuel oils, and biomass gases. The last 20 years has seen a large growth in Gas Turbine Technology. The growth is spearheaded by the growth of materials technology, new coatings, and new cooling schemes. This, with the conjunction of increase in compressor pressure ratio, has increased the gas turbine thermal efficiency from about 15% to over 45 (Saravanamuttoo et al, 2017).

In modern society there is still little appreciation of what a gas turbine is or does. This is in contrast to diesel or gasoline engines, the so-called reciprocating engines, since most people know that these are the motors which drive our cars and trucks. Despite this general lack of recognition, gas turbines are essential to everyday life. Modern air traffic would not exist without gas turbines, and we would also have serious shortages in electricity production without them. In the power generation sector, gas turbines have also become the main technology for conversion of fossil fuels into electricity. There are many applications for co-generation, for example, in the paper, process and food and beverage industries. Apart from these major areas, gas turbines are also used in a variety of other applications. In the oil and gas sector, gas turbines drive compressors or pumps for gas- and oil-processing and transport. Furthermore, gas turbines often power fast ferries and military vessels owing to their superior power/weight ratio. There is also a new trend to apply very small gas turbines for pre-firing in central heating systems.(Jansohn 2013).

Gas turbine power plant are usually built light and compact for performance improvement (Antonanzas et al 2014). The gas turbine power plants have usually been used as heavy-duty plants for electricity generation and have been presented as better power generation systems (Balevic et al 2010).There are methods of improving the performance of a gas turbine power plant. One of the commonly used methods is the intercooler. According to (Ibrahim et al 2015) this method is mainly used to reduce the air temperature between the compressor stages. This is aimed at reducing the compressors' power consumption while maintaining a high expelled air pressure from the compressor (Ibrahim et al 2010). The air between the compressor stages can be cooled by the compressor of the gas turbine power plants with a high pressure ratio to allow for the combustion of extra fuel and increase power generation using the intercoolers (Al-Doori et al 2011)

The performance improvement can also be achieved through the use of an additional combustion chamber (reheat) in the gas turbine cycle. There are two processes in the expansion process within the reheat gas turbine. An additional combustion chamber is positioned between the low-pressure turbines (Thamir et al 2019).

The combined cycle is another method of increasing performance. Firstly, the gas turbine drives a generator, delivering two thirds of the total power output. Secondly, the hot exhaust gas from the turbine passes through a boiler to produce steam that drives a steam turbine and a coupled generator. This secondary system delivers the remaining third of the total power output of the combined-cycle power plant (Jansohn 2013).

The exhaust gas expelled from the GT turbine is usually at a high temperature of about 5000C while air exiting the compressor is at a lower temperature of about 3000C (Ibrahim et al 2012). Hence, the hot exhaust gas serves as a medium of heat transfer to the air that is leaving the compressor through a heat exchanger in a process known as recuperation or regeneration (Dellenback 2002).

The gas turbine is the best suited prime mover when the needs at hand such as capital cost, time from planning to completion, maintenance costs, and fuel costs are considered. The gas turbine has the lowest maintenance and capital cost of any major prime mover. It also has the fastest completion time to full operation of any plant. Its disadvantage was its high heat rate but this has been addressed and the new turbines are among the most efficient types of prime movers. The design of any gas turbine must meet essential criteria based on operational considerations. These criteria are; High efficiency, High reliability, high availability, Ease of service, Ease of installation and commission, Conformance with environmental standards, Incorporation of auxiliary and control systems, which have a high degree of reliability, Flexibility to meet various service and fuel needs. The two factors, which most affect high turbine efficiencies, are pressure ratios and temperature (Jansohn, 2013).

The gas turbine just like any other internal combustion engines has its own limitation, starting with the deterioration in performance during operation, leading to reduced capacity and thermal efficiency. Loss of capacity result in lost production, affecting revenue. Loss in thermal efficiency increases fuel consumption. Performances deterioration generally results in increased emissions. This work is centered in increasing performance of the gas turbine and reducing emission rate of the gas turbine.

The importance of this work is to improve the efficiency of a simple gas turbine and reduce emission rate to the minimal. Simple gas turbines show performance characteristics that distinctly depend on ambient and operating conditions. Ensuring the environmental safety is an important task, this problem is becoming more urgent due to the increase in engine power, since an increase in power is achieved primarily by increasing the temperature in the combustion chamber, leading to an increase in NO_x emissions.

2.0 MATERIALS AND METHOD

2.1 Data Collection

In this work, AES barges gas turbine plant unit PB204 with installed capacity of 31.5 MW was selected for study. The plant is situated on the lagoon jetty, at the PHCN Egbin Thermal Station premise, in Ijede, a suburb of Ikorodu Town in Lagos, Nigeria. Operating data for the gas turbine unit were collected from the daily turbine control log sheet for a period of five years (2006-2010). The daily average operating variables were statistically analyzed and mean values were computed for the period of January to December, followed by an overall average. A summary of the operating parameters of the PB204 unit used for this study is presented in Table 2.1. The analysis of the plant was divided into different control volumes and performance of the plant was estimated using component-wise modeling. Mass, energy conservation laws and exergy balance were applied to each component and the performance of the plant was determined for the system.

Table 2.1. Average operating data for the selected gas turbine power plant

S/NO	**Operating Parameters**	**Unit**	**Values**
1	Ambient Temperature, T_1	K	303.63
2	Compressor Outlet Temperature, T_2	K	622.31

3	Turbine Inlet Temperature, T_3	K	1218.62
4	Turbine Outlet Temperature, T_4	K	750.00
5	Exhaust gas temperature, T_{exh}	K	715.00
6	Compressor inlet pressure, P_1	Bar	1.013
7	Compressor outlet pressure, P_2	Bar	9.80
8	Pressure ratio	__	9.67
9	Mass flow rate of fuel	Kg/s	2.58
10	Inlet mass flow rate of air	Kg/s	125.16
11	Thermal Efficiency	(%)	39.01
12	Power output	MW	29.89
13	LHV of fuel	Kj/kg	47,541.57

2.2 Thermodynamic Modelling of simple Gas Turbine Engine

The major components of simple gas turbine (GT) are compressor, combustion chamber and the turbine. The compressor takes in air from the atmosphere, compresses it to a higher temperature and pressure which is then sent to the combustor and the products of combustion are expanded in the turbine.

2.2.1. Energy (First Law of Thermodynamics)

Analysis Using the first law of thermodynamics for a thermal system, it is possible to calculate the cycle thermal efficiency, which is the ratio of the work output to the heat input. Also, the energy loss in each component and the entire plant can be computed using energy balance. For any control volume at steady state with negligible potential and kinetic energy changes, energy balance reduces to (Barzegar et al., 2011)

$$\dot{Q} - \dot{W} = \sum \dot{m}\, ehe - \sum \dot{m}\, ihi \qquad (1)$$

The energy balance equations for various components of the gas turbine plant shown in figure above are as follows:

2.2.2. Air Compressor Model

The compression ratio (r_p) can be defined as:

$$r_p = \frac{p_2}{p_1}, \qquad (2)$$

Where p_1 and p_2 is the compressor inlet and outlet air pressure, respectively the isentropic efficiency for the compressor is expressed as:

$$\eta_c = \frac{\left[(r_p)^{\frac{\gamma_a-1}{\gamma_a}} - 1\right]}{\left(\frac{T_2}{T_1} - 1\right)} \qquad (3)$$

Where T_1 and T_2 is the compressor inlet and outlet air temperatures respectively, Compressed air temperature can be written in terms of the pressure ratio and the inlet compressor temperature as:

$$T_2 = T_1\left[1 + \frac{(r_p)^{\frac{\gamma_a-1}{\gamma_a}} - 1}{\eta_c}\right] \qquad (4)$$

Where T_2, is the temperature in K of the compressed air entering combustion chamber and ηc, is the compressor's isentropic efficiency. At full load, the compressor work rate, W_c can be written in terms of the pressure ratio and the inlet compressor temperature as:

$$\dot{W}_c = \frac{m_a c_{pa} T_1}{\eta_c}\left((r_p)^{\frac{\gamma_a-1}{\gamma_a}} - 1\right) \qquad (5)$$

where c_{pa} is the specific heat capacity of air which is considered in this study as a function of temperature and can be fitted by Equation (6) for temperature in the range of 200K<T<800K (Rahman et al., 2011; Kurt et al ., 2009):

$$c_{pa}(T) = 1.04841 - \left(\frac{3.8371T}{10^4}\right) + \left(\frac{9.4537T^2}{10^7}\right) - \left(\frac{5.49031T^3}{10^{10}}\right) + \left(\frac{7.9298T^4}{10^{14}}\right) \qquad (6)$$

2.2.3 Combustion Chamber Model

The energy balance in the combustion chamber ($\dot{Q}_{in}$) is given

$$\dot{Q}_{in} = \dot{m}_g C_{pg}(T_3 - T_2) \quad (7)$$

Where $c_{pg} = 1.11$

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad (8)$$

2.2.4 Gas Turbine Model

The isentropic efficiency for turbine can be written in terms of the turbine pressure ratio, the turbine inlet temperature and turbine exit temperature as:

$$\eta_T = \frac{1-\left(\frac{T_4}{T_3}\right)}{1-(r_T)^{\frac{1-\gamma_g}{\gamma_g}}} \quad (9)$$

Where r_T is the turbine pressure ratio: $r_T = P3/P4$

The exhaust gases temperature from the gas turbine is given as:

$$T_4 = T_3\left\{1 - \eta_T\left[1-\left(\frac{P_3}{P_4}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right]\right\} \quad (10)$$

The shaft work rate of the turbine is written in terms of the pressure ratio and the turbine inlet temperature as:

$$\dot{W}_T = \dot{m}_g c_{pgT_3}\left[1-(r_T)^{\frac{1-\gamma_g}{\gamma_g}}\right] \quad (11)$$

The network rate of the gas turbine is given in terms of the pressure ratio, compressor inlet temperature and turbine inlet temperature as:

$$\dot{W}_n = \dot{m}_g c_{pgT_3}\left[1-(r_T)^{\frac{1-\gamma_g}{\gamma_g}}\right] - \frac{ma\dot{C}_{pgT_1}}{\eta_c}\left[(r_p)^{\frac{\gamma_{a-1}}{\gamma_a}} - 1\right] \quad (12)$$

C_{pg} is the specific heat capacity of combustion product (gas). The power output is expressed in terms of the pressure ratio, compressor inlet temperature and turbine inlet temperature as:

$$P = \dot{m}_g\left[c_{pg}T_3\eta_T\left(1-(r_p)^{\frac{1-\gamma_g}{\gamma_g}}\right) - \frac{C_{pa}T_1}{\eta_C}\left((r_p)^{\frac{\gamma_{a-1}}{\gamma_a}} - 1\right)\right] \quad (13)$$

The gas turbine thermal efficiency (ηth) can be determined by Equation (14):

$$n_{th} = \frac{\dot{W}n}{\dot{m}_f LHV} \quad (14)$$

The thermal efficiency can also be determined by:

$$n_{th} = \frac{\dot{W}n}{\dot{Q}in} \quad (15)$$

The specific fuel consumption (SFC) is determined by:

$$SFC = \frac{3600}{W_n} f,$$

Where f (fuel –air mass ratio) is given by:

2.3 Thermodynamic Modelling of a Regenerative Gas Turbine Cycle

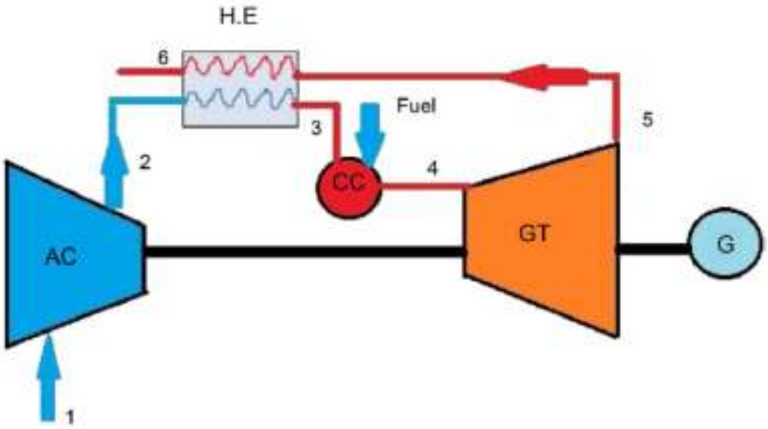

Figure 2.1: Regenerative Gas Turbine Cycle diagram

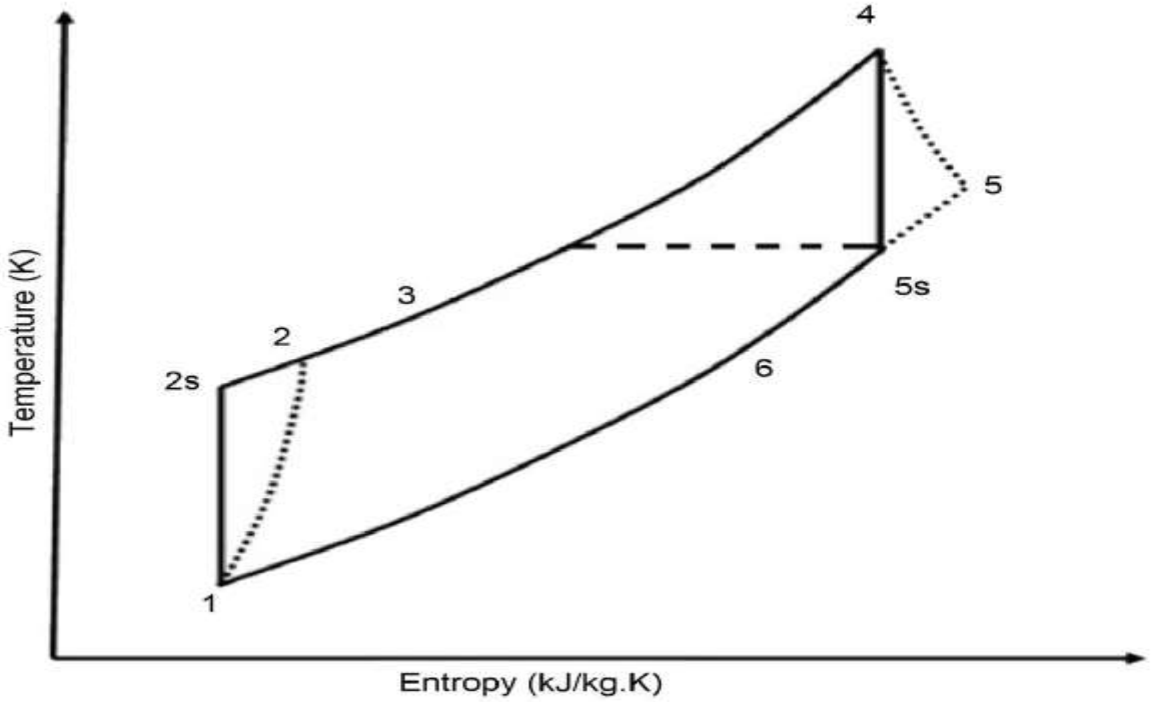

Figure 2.2: T-S Diagram of Regenerative Gas Turbine Cycle

Using the first law of thermodynamics and the intake pressure drop (ΔP intake)

is taken to be 0.004 bar, the intake temperature is the same as the ambient temperature.

$$P_1 = P_{atm} - \Delta P_{intake} \tag{21}$$

However, to simplify the thermodynamic analysis of the regenerative gas turbine cycle the following assumptions were considered:

1. Regenerative effectiveness of 0.8 -1.0 was considered.

2. Turbine Inlet Temperature is same as the one in Simple Gas Turbine.

3. Maximum heat transferable in the heat exchanger ($T_2 = T_6$)

2.3.1 Air Compressor Model

The compressor pressure ratio (rp) can be defined as Equation (20):

$$r_p = \frac{P_2}{P_3} \tag{22}$$

where P_1 and P_2 are the compressor inlet and outlet air pressures, respectively. The isentropic outlet temperature leaving the compressor is determined by Equation (23).

Take specific heat ratio for air $\gamma\, a = 1.4$,

$$\frac{T_1}{T_{2s}} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma a - 1}{\gamma a}} \tag{23}$$

The isentropic efficiency of the compressor expressed as Equation (24):

$$\eta_{is)c} = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{24}$$

where T_1 and T_2 are the compressor inlet and outlet air temperatures, respectively. The work of the compressor (Wc) when blade cooling is not taken into account can be calculated as Equation (25):

$$\dot{W}_C = \dot{m}_a Cp_a (T_2 - T_1) \tag{25}$$

where the specific heat of air is *Cpa* = 1.005 kJ kg K

2.3.2 Combustion Chamber Model

From the energy balance in the combustion chamber:

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f \text{ x } CV + \dot{m}_f C_{pf} T_f = (\dot{m}_a + \dot{m}_f) C_{pg} \text{X } T_4 \qquad (26)$$

where m_f is the fuel mass flow rate (kg/s), m_a is the air mass flow rate (kg/s), C.V is the calorific value, T_4 is turbine inlet temperature, Cp_f is the specific heat of fuel, and T_f is the temperature of the fuel. The specific heat of flue gas is Cpg = 1.07 kJ kg K, efficiency is η_{CC}

$$\eta_{C.C} = \frac{\dot{m}_g Cp_g T_4 - \dot{m}_a Cp_a T_2}{\dot{m}_f \text{ x C.V}} \qquad (27)$$

where natural gas high heating value is assumed and air/fuel ratio (A/F) is determined from the following equation:

$$\frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} \qquad (28)$$

2.3.3 Gas Turbine Model

The isentropic outlet temperature leaving the turbine is determined by Equation (29).

Take specific heat ratio for gases $\gamma_{g\,=}$ 1.3,

$$\frac{T_4}{T_{5s}} = \left(\frac{P_4}{P_5}\right)^{\frac{\gamma g - 1}{\gamma g}} \qquad (29)$$

The actual temperature drop is obtained from the definition of turbine's isentropic

efficiency:

$$\eta_{is)g.T} = \frac{T_4 - T_5}{T_4 - T_{5s}} \qquad (30)$$

The effectiveness of regenerator (heat exchanger) (ε) is considered in this study.

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} \qquad (31)$$

The total mass flow rate is given by:

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \qquad (32)$$

The work produced from the turbine is determined by the following equation:

$$\dot{W}_{g.T} = \dot{m}_g \; Cp_g (T_5 - T_4) \qquad (33)$$

The network of the GT (WGnet) is calculated by Equation (34):

$$\dot{W}_{Gnet} = \dot{W}_c + \dot{W}_{g.T} \qquad (34)$$

The output power from the gas turbine (*G*)*net*) is expressed as:

$$P_{G)net} = 2 \text{ x } \left[(\dot{W}_c + \dot{W}_{g.T})\eta_{M.GT}\right] \text{ x } \eta_{G.GT} \qquad (35)$$

The specific fuel consumption (SFC) is determined by

$$S.F.C = \frac{3600}{AFR \; X \; \dot{W}_{Gnet}} \qquad (36)$$

The heat supplied is also expressed as:

$$\dot{Q}_{add} = \dot{m}_g Cp_g T_4 - \dot{m}_a Cp_a T_2 \qquad (37)$$

The GT efficiency (*th*)*GT* η) can be determined by:

$$\eta_{th)GT} = \frac{-\dot{W}_{Gnet}}{\dot{Q}_{add}} \qquad (38)$$

The heat rate (HR) can be expressed as Equation:

$$HR = \frac{3600}{\eta_{th)GT}} \qquad (39)$$

The rate of emission can be calculated by the expression:

Emission Rate = Emission Factor * SFC (40)

3.0 RESULTS AND DISCUSSION

3.1 Effect of Regenerative Effectiveness on the Combustion Chamber Inlet Temperature

From Figure 3.1 it was observed that for every 0.01 regenerative effectiveness increase, there was an increase of 12°K, of the combustion inlet temperature of the regenerative cycle while for the simple gas turbine it was constant for all the cases. And when the regenerative effectiveness is unity (1), the inlet temperature of the combustion chamber which is 763°K for the regenerative cycle was same as the exit or exhaust temperature of the simple gas turbine.

And from figure 3.1 it was observed that as the regenerative effectiveness values was increasing, there was an increase in the inlet temperature of the combustion chamber of the regenerative cycle, thereby leading to reduction in heat addition in the regenerative cycle and also reducing specific fuel consumption and emission rate.

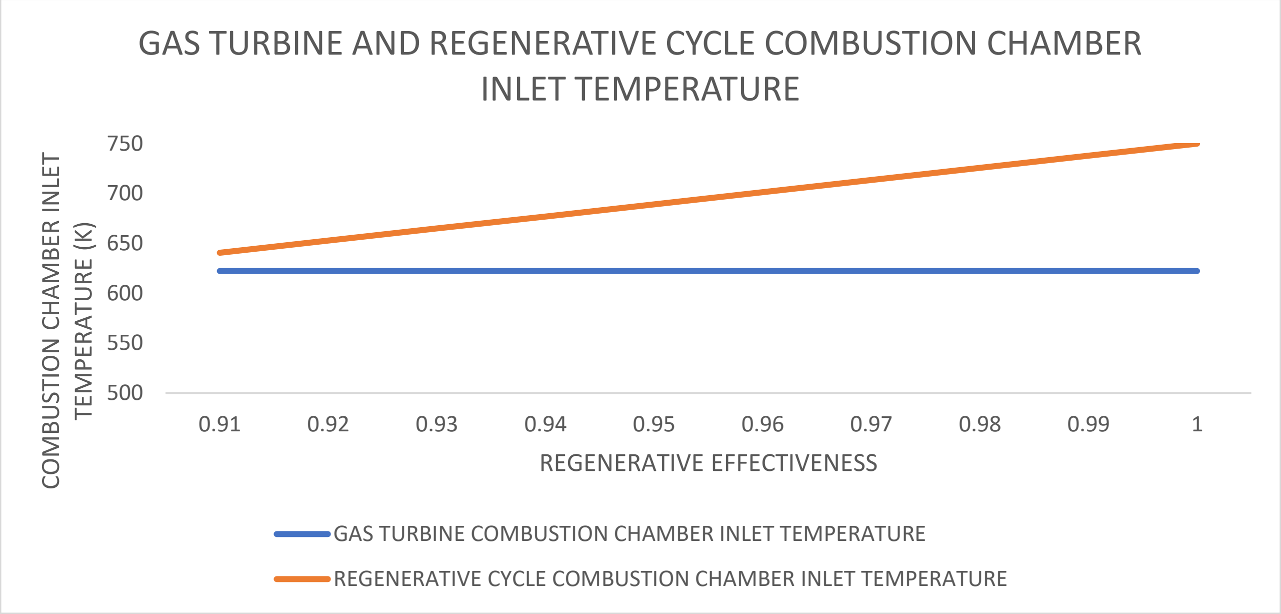

Figure 3.1: Combustion Chamber inlet temperature of a Simple Gas turbine and Regenerative Cycle

3.2 Effect of Regenerative Effectiveness on Heat Addition

From Figure 3.2 it was observed that for every 0.01 regenerative effectiveness increase, there was a decrease of 731,206W, of heat addition in the regenerative cycle while for the simple gas turbine it has a constant heat addition of 44,237,812.5W. And from figure 3.2 it was observed that as the regenerative effectiveness values was increasing, there was a decrease in the heat addition in the regenerative cycle while for the simple gas turbine it was constant. And when the regenerative effectiveness is unity (1), it has the least heat addition of 25,158,375W thereby having higher thermal efficiency of the regenerative cycle and low specific fuel consumption and emission rate.

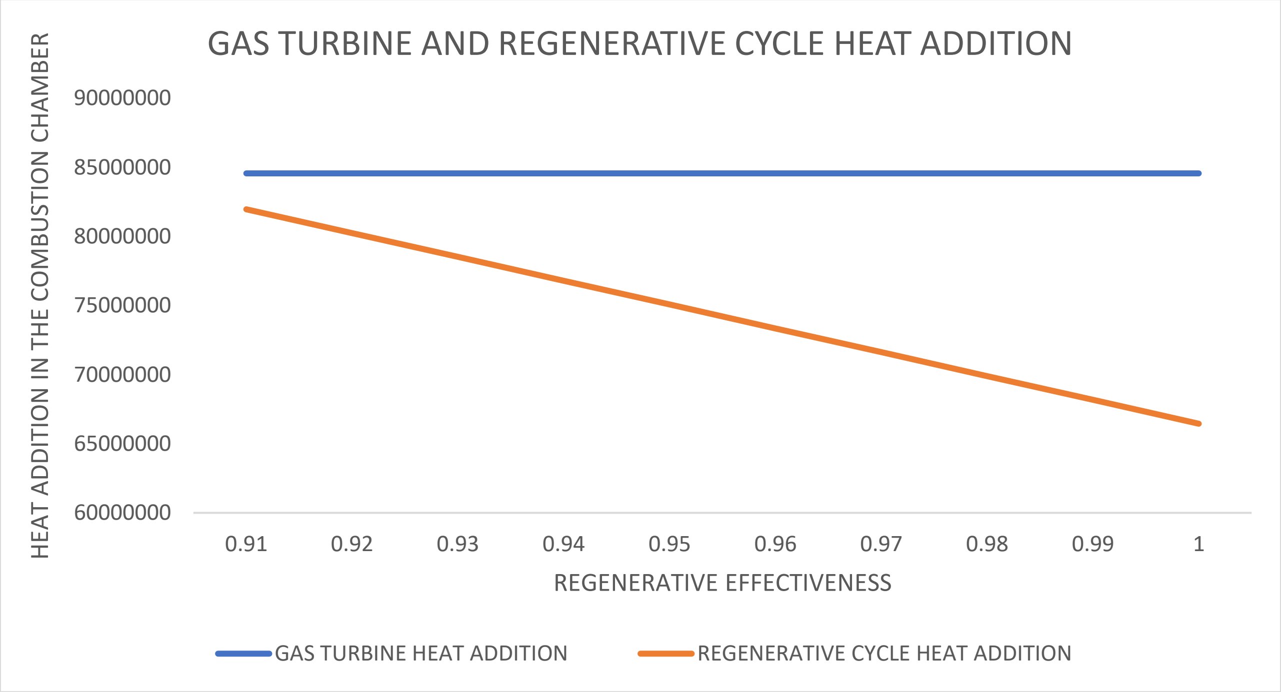

Figure: 3.2: Heat Addition in a Simple Gas turbine and Regenerative Cycle

3.3 Effect of Regenerative Effectiveness on the Thermal Efficiency

From Figure 3.3 it was observed that for the least regenerative effectiveness value of 0.91, the thermal efficiency (41.85%) of the regenerative cycle is higher than the thermal efficiency (40.57%) of the simple gas turbine. And also, for regenerative effectiveness of 1, the thermal efficiency (65.03%) of a regenerative cycle is 60.29% greater than the thermal efficiency (40.57%) of the simple gas turbine, thereby resulting to reduction to specific fuel consumption and emission rate. From figure 3.3 it was seen that the thermal efficiency of a simple gas turbine was constant and the thermal efficiency of a regenerative cycle increases as regenerative effective increases.

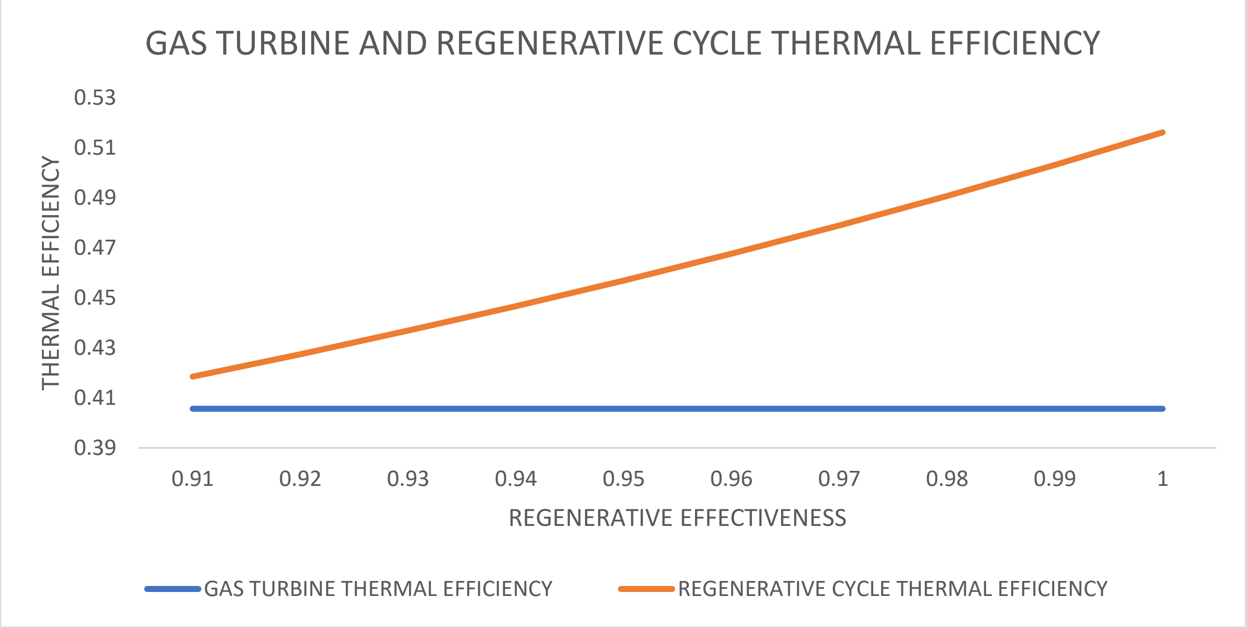

Figure: 3.3: Thermal Efficiency of a Simple Gas turbine and Regenerative Cycle

3.4 Effect of Regenerative Effectiveness on the Specific Fuel Consumption

From figure 3.4 it was observed that for every 0.01 regenerative effectiveness increase, there is a decrease of 0.00458(Kg/Kw.h), of fuel consumption in the regenerative cycle while for the simple gas turbine it has constant fuel consumption of 0.276824936 (Kg/Kw.h). And from figure 3.4 it was observed that as the regenerative effectiveness values increases, there is a decrease in the specific fuel consumption in the regenerative cycle, thereby leading to increase in the thermal efficiency of the regenerative cycle and emission rate.

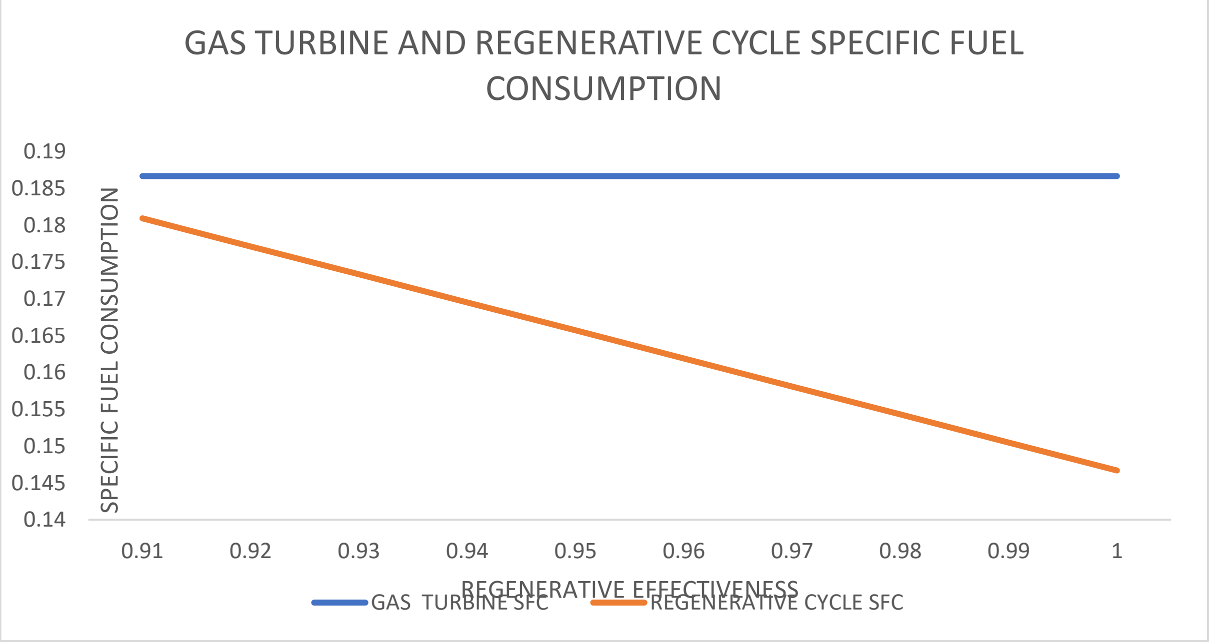

Figure: 3.4: Specific Fuel Consumption of a Simple Gas turbine and Regenerative Cycle

3.5 Effect of Regenerative Effectiveness on the Heat Rate

From figure 3.5 it was observed that for every 0.01 regenerative effectiveness increase, there was a decrease of 161(Btu/Kwh) of the heat rate in the regenerative cycle while for the simple gas turbine it has a constant heat rate. And from figure 3.5 it was observed that as the regenerative effectiveness values was increasing, there was a decrease in the heat rate in the regenerative cycle while for the simple gas turbine it was constant. Since the heat rate of the regenerative cycle is lower compare to the simple gas turbine, that means it has higher thermal efficiency, low specific fuel consumption and emission rate.

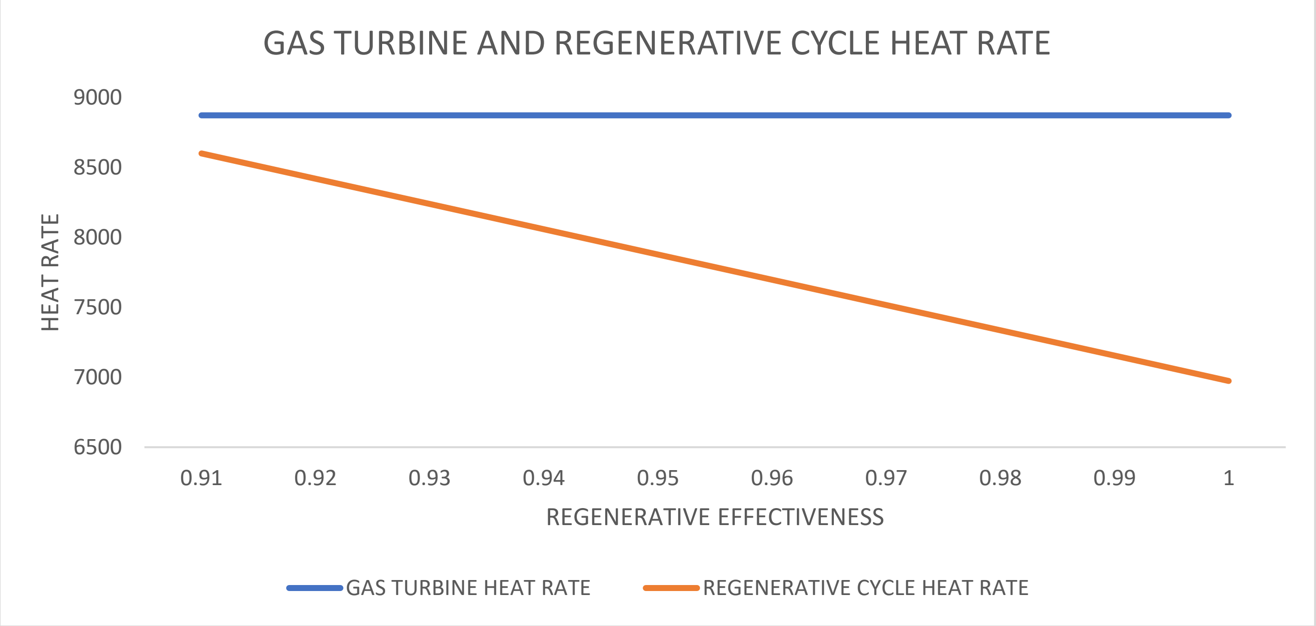

Figure: 3.5: Heat Rate of a Simple Gas turbine and Regenerative Cycle

3.6 Validation of Computer Analyzed data

The model developed in this study is validated by the actual data that were taken from the operational data of an existing gas turbine power plant in Nigeria. The operational parameters of the simple gas turbine are set as base line for comparison with the calculated results. The parameters considered in this study in gas turbine engine during simulation are the thermal efficiency, Heat rate and Power output. The results of thermodynamic properties of the cycle form the modeling part and the operational data are illustrated in Table 3.1. The comparison of simulation results and the actual data from the power plant show that the difference in the simulation results and the actual data varies from 1.1 to 4.5 %. The maximum difference is about 4.5 % for the thermal efficiency while the minimum difference is about 1.1 % for the Heat Rate. This validates the correct performance of the developed simulation values to model the selected gas turbine power plant, as the results of the simulation values are close to the actual design data of the plant considered in this study.

Table: 3.1 Result between design data and simulated data

Parameters	Operational data	Simulated data	Differences	Percentage increase (%)	Percentage decrease (%)
THERMAL EFFICIENCY (%)	39.01	40.57	1.58	4.5	-
HEAT RATE (Btu/Kwh)	8776	8874	98	1.1	-

CONCLUSION

In this study, comprehensive thermodynamic models were used to analyze the thermal efficiency, Heat addition, specific fuel consumption and heat rate of a simple gas turbine simple gas turbine, the gas turbine furtherly was converted to regenerative cycle using regenerative effectiveness of 0.91 – 1.0. And the thermal efficiency, Heat addition, specific fuel consumption and heat rate of the regenerative was calculated and was compare with that of the simple gas turbine.

To achieve this aim, Excel 2010 was used to carry out the simulation of the thermal efficiency, heat rate and power output of the simple gas turbine and the result was compared to the operation parameters, the results showed a reasonably good agreement between the simulated results and design data. The thermodynamic model reveals that the influences of specific fuel consumption, heat rate and heat addition have significant effect on the thermal efficiency of a simple gas turbine and regenerative cycle. The thermodynamic simulation results are summarized as follows.

- The thermal efficiency is constant for the simple gas turbine
- The thermal efficiency for a regenerative cycle increases linearly while the heat addition, specific fuel consumption and heat rate reduces linearly as regenerative effectiveness increases.
- At regenerative effectiveness of 0.91 – 1, the thermal efficiency of the regenerative cycle is higher compares to the simple gas turbine.
- In general, it was seen that the thermal efficiency of the modified regenerative cycle is higher than the simple gas turbine.

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