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EFFECT OF REGENERATIVE EFFECTIVENESS ON THE PERFORMANCE OF SIMPLE GAS TURBINE

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ABSTRACT

The deregulation of the power and energy sector introduced a strong element of competition. Power plant operators try to develop techniques to maximize profits in the dynamic power industry. New methods of improving and optimizing simple gas turbine power plants are needed to enhance operational decision making and therefore to maximize power plant profitability by reducing operational and maintenance cost and increasing revenue. In this work, the thermodynamic model equations of both a simple gas turbine and regenerative gas turbine cycle were used to analyses the effect of increase in the regenerative effectiveness to the performance of the simple gas turbine. Microsoft Excel 2010 software was used to carried out the analysis. It was observed that for the simple gas turbine the thermal efficiency, heat rate, specific fuel consumption and heat addition have a constant value of 35.4%, 9630Btu/Kwh, 0.2768kg/Kwh, 44237.8Kw, all through, while for the regenerative gas turbine cycle as the value of the regenerative effectiveness increases from 0.8 to 1.0, the thermal efficiency increases from 41.1% to 65.0%, the heat rate decreases from 8753Btu/Kwh to 5536 Btu/Kwh , specific fuel consumption decreases from 0.248945122Kg/Kw.h to 0.157432413 Kg/Kw.h and heat addition reduces from 39782.5Kw to 25158.4Kw. It was seen clearly that the regenerative gas turbine cycle has higher thermal efficiency compare to the simple gas turbine, and as the regenerative effectiveness of the regenerative gas turbine cycle increases the performance of the regenerative gas turbine increases.

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

INTRODUCTION

Gas turbine was dated back to 1791, the idea of open and closed cycles is contained in the patent of John Barber, on the basis on how to implement the gas turbine power plant. The present gas turbine power plant used today was developed by Franz Stole in 1873, the power plant operates on an open circuit and using internal combustion. The thermodynamic principle of any gas turbine power plant operates on Brayton/Joule cycle.

In the last five decays, gas turbine has been used greatly because of it low initial cost and this made it among the top choice. Gas turbine industry has made great success because of the low cost of natural gas used to fuel the engines.

Most researchers have carried out experimental, analytical and numerical work on the optimization and improvement of the performance of a simple gas turbine. Further improvement in the performance of the gas turbine involves utilization and employing of different strategies and techniques like regeneration, inter-cooling, reheating and intercooled reheated.

For a regenerative gas turbine cycle, the air from the exit of the turbine is usually higher than the ambient temperature, therefore a regenerative technique will be employed to recover the heat from the exhaust of the gas turbine. Extra heat exchanger (regenerator) will be employed so as to be used to help increase the temperature of the air prior its entry into the combustion chamber. The efficiency of the regenerator depends on ratio of the heat recovers by the regenerator and the maximum heat that can be recovered.

Bassily (2001), presented six gas turbine configurations with reheat, intercooled and regenerative before proposing a parametric analysis for the influence of relative humidity, turbine inlet temperature and ambient temperature on the performance of the system. It was observed that the optimal pressure ratio increased by about 1.5 after increasing the turbine inlet temperature. The simulation model which studies the influence of turbine inlet temperature, ambient temperature, pressure ratio, relative humidity and the regenerative effectiveness of the heat exchanger on the performance of all configurations was introduced by Bassily (2004). The study reviewed that after increasing a turbine inlet temperature there was an increase in the regenerative heat exchanger capacity, and this increase resulted to an increase in the effectiveness of the regenerated heat exchanger and a gain in the gas turbine cycle thermal efficiency. Nishida et al (2005), analyzed two configuration of regenerative steam-injection gas turbine systems, they concluded that the performance, based on the thermal efficiency of regenerative steam-injection gas turbine systems is higher than those of regenerative, water injected and steam injection gas turbine systems. Cohen. C et al (1996), has describe the use of the regenerator in a traditional way, whereby the exhaust gases leaving the final stage turbine is the source of heat. The recovered heat is used usually to pre-heat the compressed air before entering the combustion chamber. Several other methods, including cogeneration applications and combined cycle are suggested by Bathe (1996) and Khartchenko (1998).

Dellenback (2002), clearly observed that locating a regenerator after the power turbine is inefficient, thus suggested the location of the heat exchanger between the two turbines, which will substantially improve the cycle efficiency.

In this work thermodynamic model equations for both a simple and regenerative gas turbine were used to determine the effect of the regenerative effectiveness on the simple gas turbine considering regenerative effectiveness of 0.8 – 1.0

MATERIALS AND METHOD

Data Collection

In this work, Titan 130S gas turbine plant used in one of the oil and gas platform in Nigeria, with installed capacity of 16.53MW was selected for the study.

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Designed parameters for the gas turbine unit were collected. A summary of the designed parameters of the gas turbine unit used for this study is presented in Table 1. The analysis of the plant was divided into different control volumes and performance of the plant were estimated using component-wise modeling, Mass and energy conservation laws.

Table 1: Design Parameter of Titan 130S Simple Gas turbine Engine

S/No	Design Parameters	Unit	Value
1	Ambient temperature, T_1	K	288
2	Compressor outlet temperature, T_2	K	446
3	Turbine inlet temperature, T_3	K	1181
4	Turbine outlet temperature, T_4	K	763
5	Exhaust gas temperature, Texh	K	763
6	Compressor inlet pressure, P_1	Bar	1
7	Compressor outlet pressure, P_2	Bar	19.1
8	Pressure ratio	-	19.1
9	Mass flow rate of fuel	Kg/s	0.85
10	Mass flow rate of air	Kg/s	55.4
11	Power output	MW	16.53
12	LHV of fuel	KJ/Kg	35,161.73
13	Heat Rate	Btu/Kwh	9630
14	Thermal Efficiency	%	35.4

Gas Turbine Plant Simulation with Excel

Gas turbine engine (Figure 2.1) was simulated using Excel 2010 from the thermodynamic model equations. The effect of variation of operating conditions on the performance of gas turbine engine was investigated. Gas turbine cycle (Figure 2.2) was modelled by using each component (compressor, combustion chamber, and turbine). Performance indices such as power output, specific fuel consumption, thermal efficiency, heat supplied and network output were calculated.

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

1. The air and gases in gas turbine cycle were perfect gases.

2. The specific heat capacities during cycles' processes were constant.

3. The drop in the pressure during combustion is constant and it represents a percentage of the pressure at the inlet of the combustion chamber.

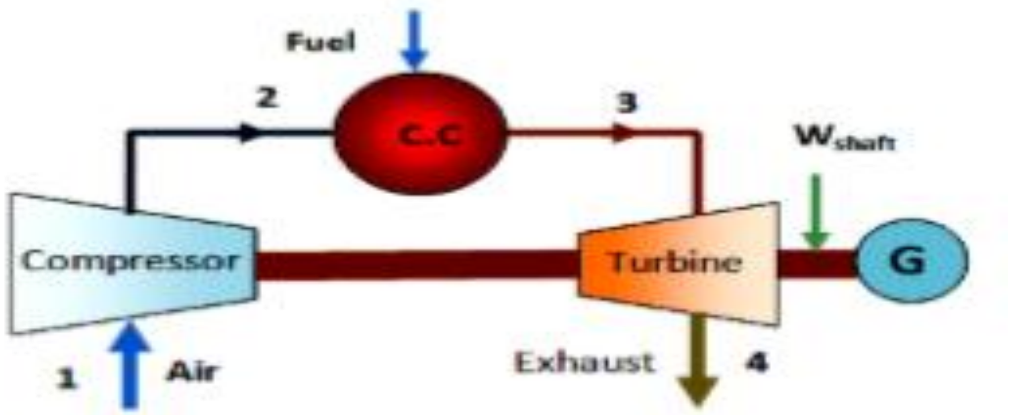

Figure 2.1: diagram of a simple gas turbine

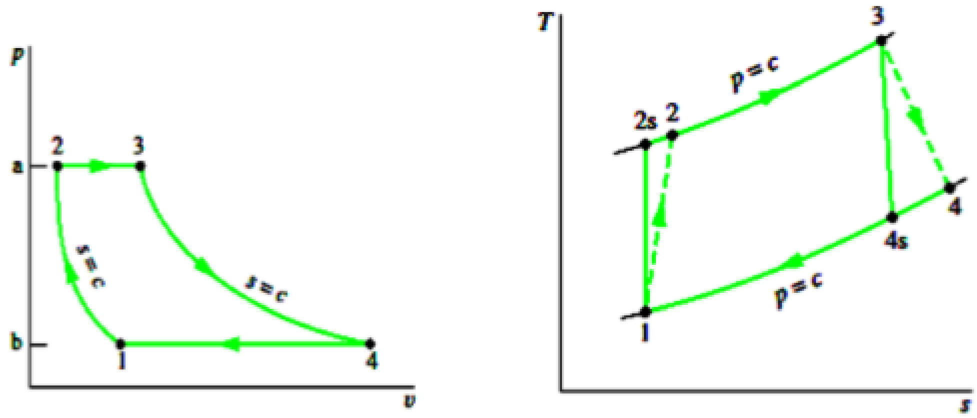

Figure 2.2: P-V and T-S Diagram of a simple gas turbine

Thermodynamic Modelling of Simple Gas Turbine Engine

The three major component of a simple gas turbine are considered in carrying out the governing thermodynamics models equations (1- 19). In this study, energy models are considered for gas turbine performance assessment.

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. For any control volume at steady state with negligible potential and kinetic energy changes, energy balance reduces to (Barzegar et al., 2011):

$$\dot{Q} - \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \qquad (1)$$

The energy balance equations for various components of the gas turbine plant shown in Figure 2.1 are as follows:

Air Compressor Model

The compression ratio (r_p) can be defined as:

$$r_p = \frac{P_2}{P_1}, \qquad (2)$$

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where P_1 and P_2 are the compressor inlet and outlet air pressure, respectively

The isentropic efficiency for the compressor is expressed as:

$$\eta_c = \frac{\left[(r_p)^{\frac{\gamma a-1}{\gamma a}} - 1\right]}{\left(\frac{T_2}{T_1} - 1\right)}, \tag{3}$$

where T_1 and T_2 are 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

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

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

$$\dot{W}_C = \frac{\dot{m}_a C_{pa} T_1}{\eta_c}\left((r_p)^{\frac{\gamma a-1}{\gamma a}} - 1\right), \tag{5}$$

where *cpa* 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 (Kurt et al., 2009):

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

Combustion Chamber Model

The energy balance in the combustion chamber is given by (Rahman, et.al, 2011):

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f\left(LHV + C_{pf} T_f\right) = (\dot{m}_a + \dot{m}_f) C_{pg} T_3 \tag{7}$$

where $\dot{m}_f$, is fuel mass flow rate (kg/s), $\dot{m}a$ is air mass flow rate (kg/s), LHV is low heating value, T_3 is turbine inlet temperature (K) Cp_f is specific heat of fuel and T_f is temperature of fuel (K). Cpg is the specific heat capacity of combustion product (gas) which is considered in this work to be a temperature variable function and can be fitted by Equation (13) for temperature in the range of 1000K <T< 1500K (Tahouni et al., 2012; Kurt et al., 2009):

$$C_{pg}(T) = 0.991615 + \left(\frac{6.99703T}{10^5}\right) + \left(\frac{2.7129T^2}{10^7}\right) - \left(\frac{1.22442T^3}{10^{10}}\right) \tag{8}$$

The fuel – air ratio (f) is expressed as:

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg}T_3 - C_{pa}T_1(1+r_{pg})}{LHV + C_{pf}T_f - C_{pg}T_3}, \tag{9}$$

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:

$$\eta_T = \frac{1-\left(\frac{T_4}{T_3}\right)}{1-(r_T)^{\frac{1-\gamma g}{\gamma g}}}, \tag{10}$$

where r_T is the turbine pressure ratio: $r_T = P_3/P_4$.

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

$$T_4 = T_3 \left\{1 - \eta_T \left[1 - \left(\frac{P_3}{P4}\right)^{\frac{1-\gamma g}{\gamma g}}\right]\right\} \tag{11}$$

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

$$\dot{W}_T = \dot{m}_g c_{pg} T_3 \eta_T \left[1 - (r_T)^{\frac{1-\gamma g}{\gamma g}}\right] \tag{12}$$

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

$$\dot{W}_n = \dot{m}_g c_{pg} T_3 \eta_T \left[1 - (r_T)^{\frac{1-\gamma g}{\gamma g}}\right] - \frac{\dot{m}_a c_{pa} T_1}{\eta_c} \left[(r_p)^{\frac{\gamma a-1}{\gamma a}} - 1\right] \tag{13}$$

Where

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{14}$$

cpg is the specific heat capacity of combustion product (gas) and it is given as in (13).

The power output is expressed in terms of the pressure ratio, compressor inlet temperature and turbine inlet temperature as:

$$P = m_g \left[C_{pg} T_3 \eta_T \left(1 - (r_p)^{\frac{1-\gamma g}{\gamma g}}\right) - \frac{C_{pa} T_1}{\eta_c}\left((r_p)^{\frac{\gamma a-1}{\gamma a}} - 1\right)\right] \tag{15}$$

Energy input in the turbine is given as:

$$\dot{Q}_T = \dot{m}_g C_{pg} T_3 \tag{16}$$

The gas turbine thermal efficiency (ηth) can be determined by Equation (27):

$$\eta_{th} = \frac{\dot{W}_n}{\dot{m}_f LHV} \tag{17}$$

The specific fuel consumption (*SFC*) is determined by:

$$SFC = \frac{3600}{w_n} f, \tag{18}$$

where f (fuel –air mass ratio) is given by Equation (9).

The heat rate (HR) (i.e., the consumed thermal energy to generate unit energy of electrical energy) can be expressed as:

$$HR = \frac{3600}{\eta_{th}} \tag{19}$$

Thermodynamic Modelling of a Regenerative Gas Turbine Cycle

Regenerative Gas turbine engine (Figure 2.3) was simulated using Microsoft Excel 2010 from the thermodynamic model equations. The effect of variation of operating conditions on the performance of regenerative gas turbine engine was investigated. regenerative gas turbine cycle (Figure 2.4) was modelled by using each component (compressor, combustion chamber, regenerator and turbine).

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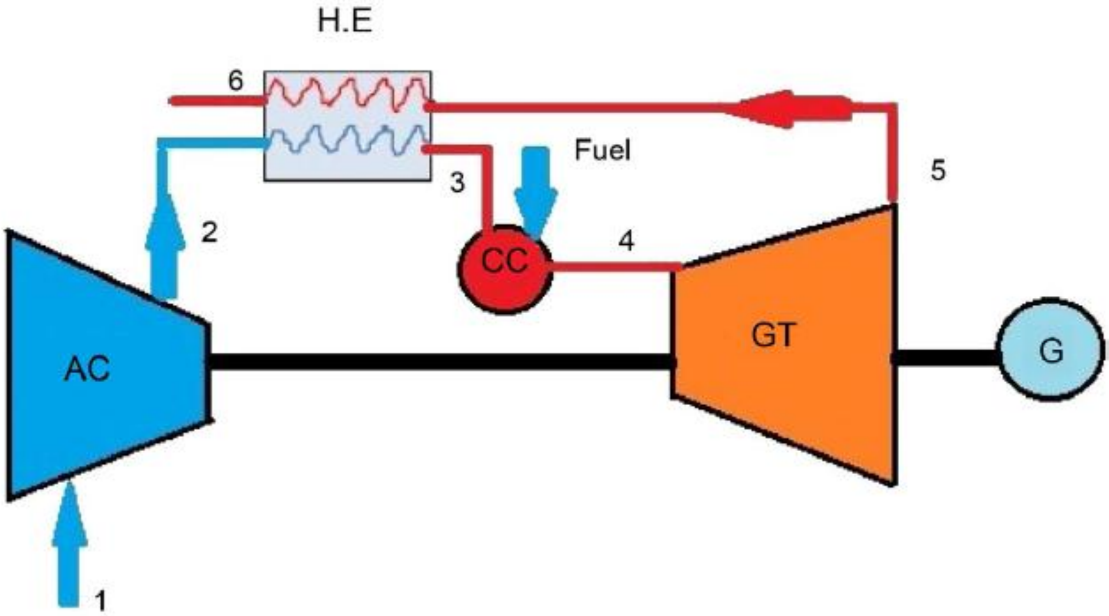

Figure 2.3: Regenerative Gas Turbine Cycle diagram

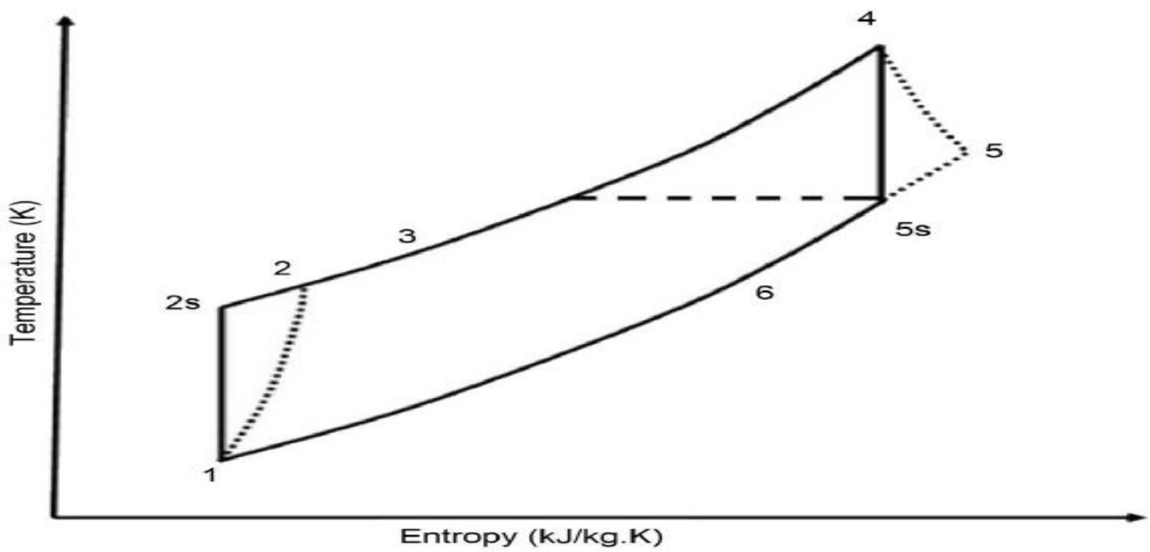

Figure 2.4: T-S Diagram of Regenerative Gas Turbine Cycle

Using the first law of thermodynamics and the intake pressure drop.

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

$$P_1 = P_{atm} - \Delta P_{intake} \qquad (20)$$

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 ($T_2 = T_6$)

Air Compressor Model

The compressor pressure ratio (rp) can be defined as Equation (21):

$$r_p = \frac{P_2}{P_3} \tag{21}$$

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 (22).

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

$$\frac{T_1}{T_{2s}} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma a-1}{\gamma a}} \tag{22}$$

The isentropic efficiency of the compressor expressed as Equation (23):

$$\eta_{is)c} = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{23}$$

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 (24):

$$\dot{W}_C = \dot{m}_a Cp_a (T_2 - T_1) \tag{24}$$

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

Combustion Chamber Model

From the energy balance in the combustion chamber:

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f \text{ x } CV + \dot{m}_f C_{pf} T_f = (\dot{m}_a + \dot{m}_f) C_{pg} \text{X } \mathrm{T}_4 \tag{25}$$

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}

$$\eta_{C.C} = \frac{\dot{m}_g Cp_g T_4 - \dot{m}_a Cp_a T_2}{\dot{m}_f \text{ x C.V}} \tag{26}$$

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

$$\frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} \tag{27}$$

Gas Turbine Model

The isentropic outlet temperature leaving the turbine is determined by Equation (28).

Take specific heat ratio for gases $\gamma_g = 1.3$,

$$\frac{T_4}{T_{5s}} = \left(\frac{P_4}{P_5}\right)^{\frac{\gamma g-1}{\gamma g}} \tag{28}$$

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

$$\eta_{is)g.T} = \frac{T_4 - T_5}{T_4 - T_{5s}} \tag{29}$$

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

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$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} \tag{30}$$

The total mass flow rate is given by:

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{31}$$

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

$$\dot{W}_{g.T} = \dot{m}_g \, Cp_g (T_5 - T_4) \tag{32}$$

The network of the GT (WGnet) is calculated by Equation (33):

$$\dot{W}_{Gnet} = \dot{W}_c + \dot{W}_{g.T} \tag{33}$$

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

$$P_{G)net} = 2 \text{ x } \left[\left(\dot{\mathrm{W}}_{\mathrm{c}} + \dot{\mathrm{W}}_{\mathrm{g.T}}\right)\eta_{\mathrm{M.GT}}\right] \text{ x } \eta_{\mathrm{G.GT}} \tag{34}$$

The specific fuel consumption (SFC) is determined by

$$S.F.C = \frac{3600}{AFR \; X \; \dot{W}_{Gnet}} \tag{35}$$

The heat supplied is also expressed as:

$$\dot{Q}_{add} = \dot{m}_g Cp_g T_4 - \dot{m}_a Cp_a T_2 \tag{36}$$

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

$$\eta_{th)GT} = \frac{-\dot{W}_{Gnet}}{\dot{Q}_{add}} \tag{37}$$

The heat rate (HR) can be expressed as Equation 38:

$$HR = \frac{3600}{\eta_{th)GT}} \tag{38}$$

RESULTS AND DISCUSSIONS

From Figure 3.1 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 3.1 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.

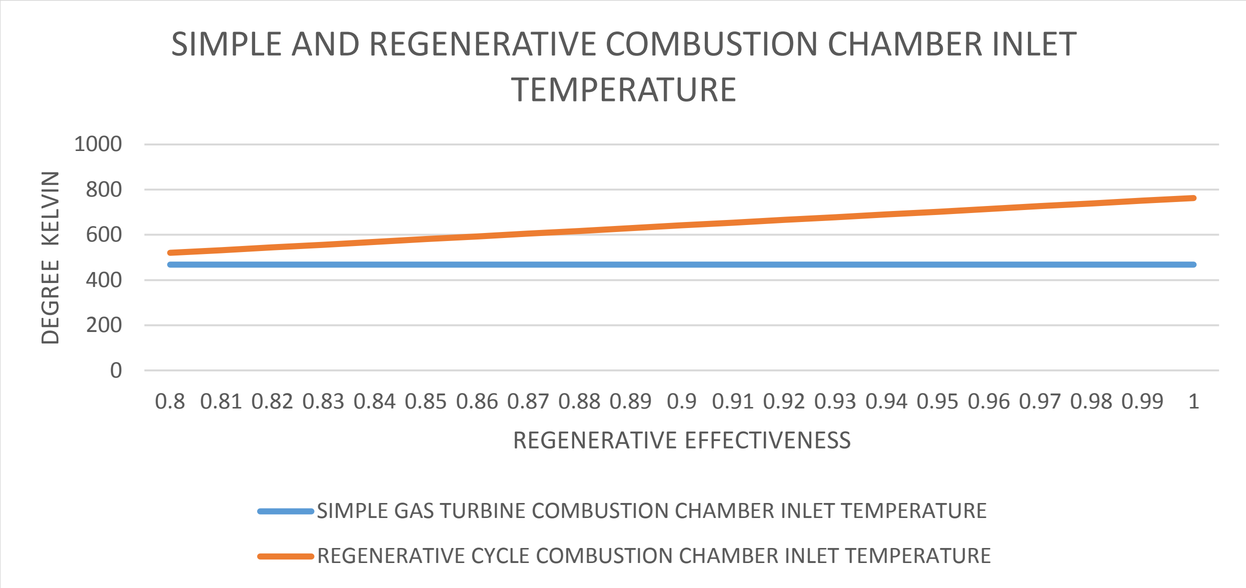

Figure: 3.1: Heat Addition in a Simple Gas turbine and Regenerative Cycle

From Figure 3.2 it was observed that for the least regenerative effectiveness value of 0.8, the thermal efficiency (41.12%) of the regenerative cycle is higher than the thermal efficiency (35.4%) of the simple gas turbine. And also, for regenerative effectiveness of 1, the thermal efficiency (65.03%) of a regenerative cycle is 83.7% greater than the thermal efficiency (35.4%) of the simple gas turbine, thereby resulting to reduction to specific fuel consumption and emission rate. From figure 3.2 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 effectiveness increases.

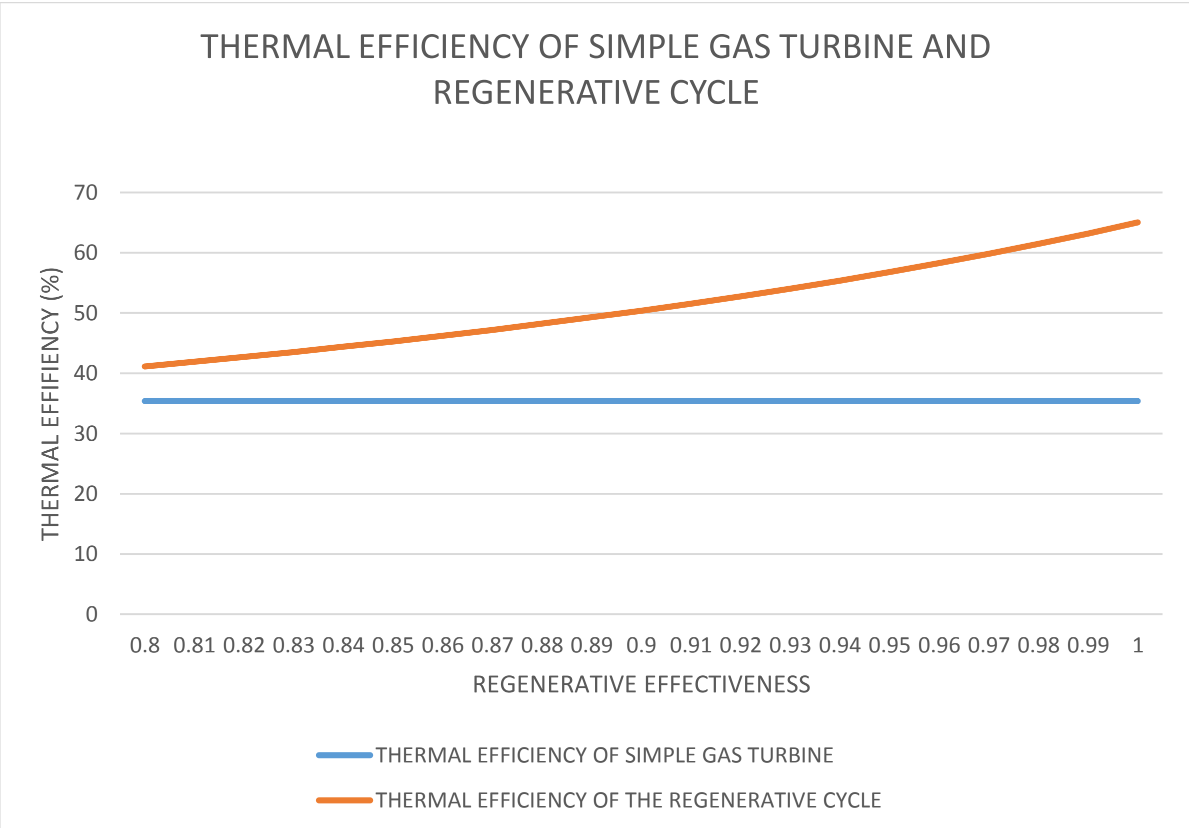

Figure 3.2: Thermal Efficiency of a Simple Gas turbine and Regenerative Cycle

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From figure 3.3 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 3.3 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.

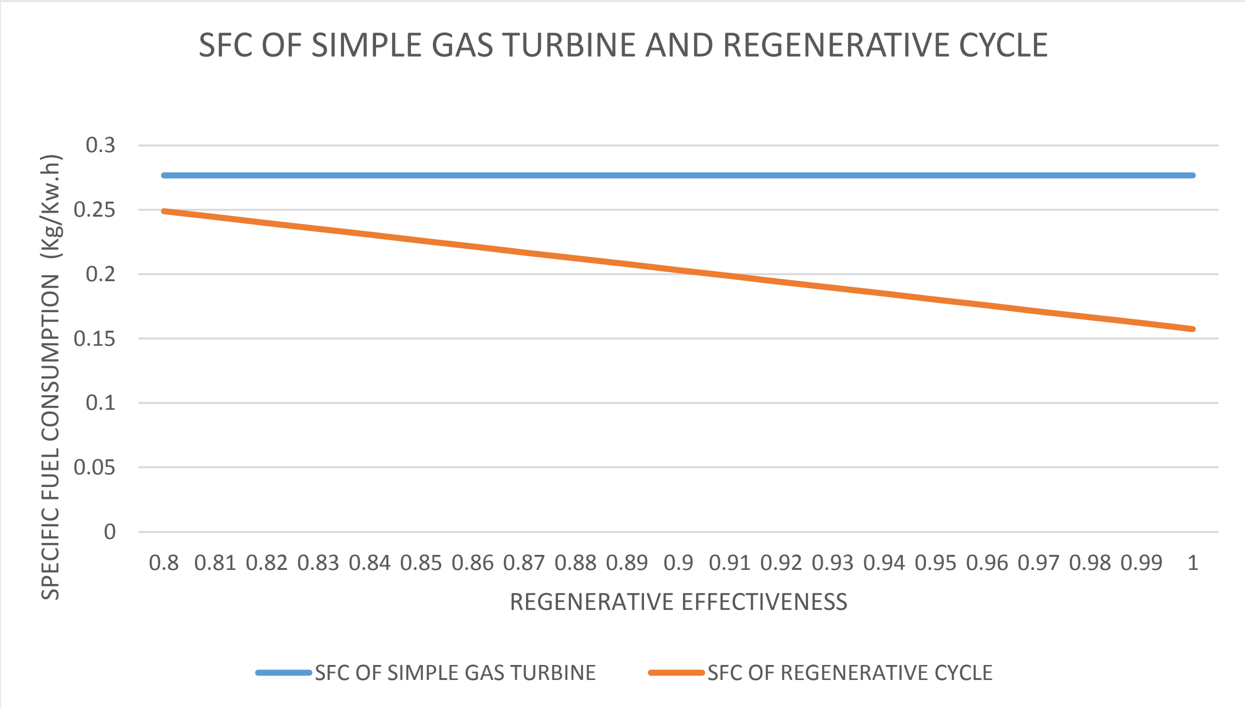

Figure 3.3: Specific Fuel Consumption of a Simple Gas turbine and Regenerative Cycle

From figure 3.4, 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 3.4 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.

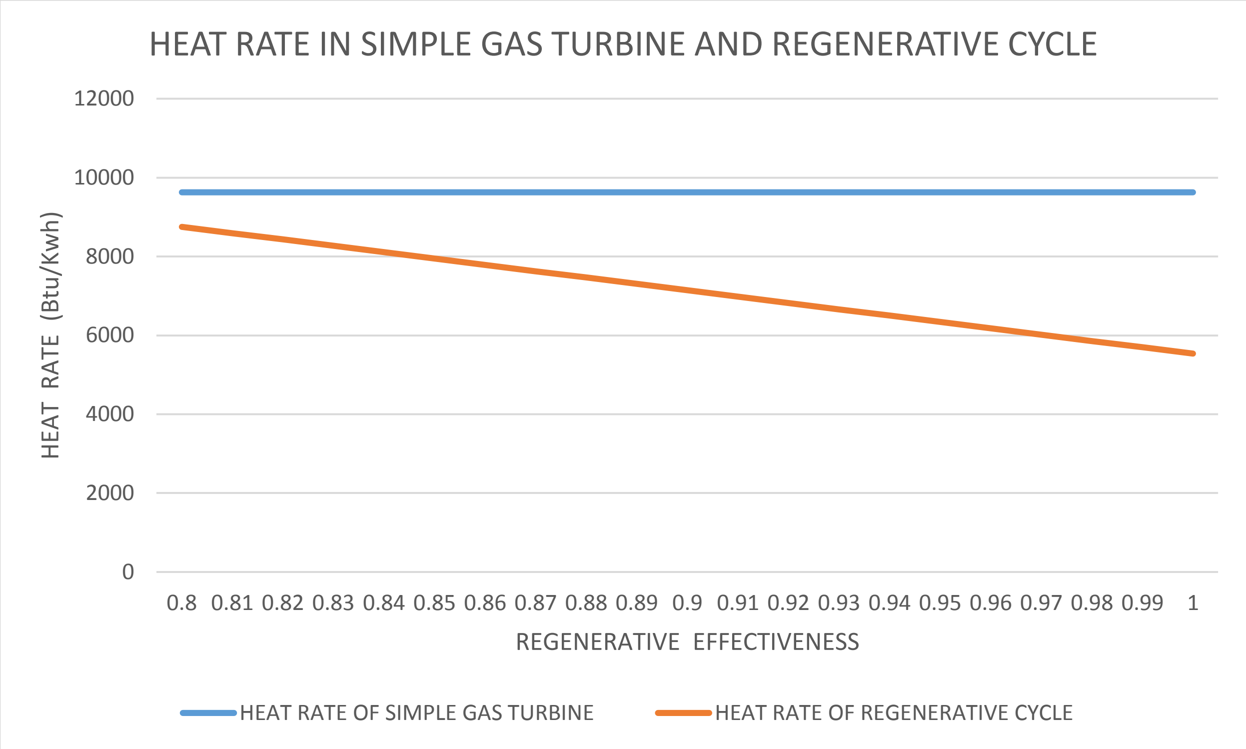

Figure 3.4: Heat Rate of a Simple Gas turbine and Regenerative Cycle

Validation of Computer Analyzed data

The model developed in this study is validated by the actual data that were taken from the design data of an existing gas turbine power plant in Nigeria. The design 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 Design data are illustrated in Table 4.1. 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: 4.1 Result between design data and simulated data

Parameters	Design data	Simulated data	Differences	Percentage increase (%)	Percentage decrease (%)
Thermal Efficiency (%)	35.4	36.98	1.58	4.5	-
Heat Rate (Btu/Kwh)	9630	9734	104	1.1	-
Power Output (MW)	16.53	16.22	0.31	-	1.9

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 (Titan 130S Simple gas turbine plant

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used with installed capacity of 16.53MW), Titan 130s simple gas turbine was further converted to regenerative cycle using regenerative effectiveness of 0.8 – 1.0. And the thermal efficiency, Heat addition, specific fuel consumption and heat rate of the regenerative cycle were calculated and compared with that of the simple gas turbine (Titan 130S). 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 (Titan 130S), and the result was compared to the design 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 and emission rate of a simple gas turbine and regenerative cycle. The thermodynamic simulation results are summarized as follows.

- The thermal efficiencies are 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.
- In general, it was seen that the thermal efficiency of the modified regenerative cycle is higher than the simple gas turbine (Titan 130S)

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