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Udeh, G.T. and Udeh, P.O. (2019) Comparative thermo-economic analysis of multi-fuel fired gas turbine power plant. Renewable Energy, 133. pp. 295-306. ISSN 0960-1481

https://doi.org/10.1016/j.renene.2018.10.036

Original article

Comparative thermo-economic analysis of multi-fuel fired gas turbine power plant

GodfreyT. Udeh[a,]* and Patrick O. Udeh[b]

a. Lecturer, University of Port Harcourt, Port Harcourt (Rivers State) P.M.B 5323, Nigeria.
b. Researcher, University of Port Harcourt, Port Harcourt (Rivers State) P.M.B 5323, Nigeria.

Abstract: This study compares the performance of a thermal power plant fired by natural gas to that fired by biodiesel blend, from exergetic and economic perspectives. A thermodynamic model has been developed to predict the performance of a running plant and was used to conduct the comparative study. Plant life of 25 years has been used to assess the viability of the gas turbine power plant by analyzing the net present cost and the break-even point for both fuels. The plant specific fuel consumption for natural gas fired and biodiesel blend fired are 0.3151[kg/kWh] and 0.3884 [kg/kWh] respectively. The system fired by natural gas only, has a payback period of 1.9 years, internal rate of return of 52% and exhaust temperature of 915.74 [K], while that fired by the biodiesel blend has a payback period of 2.4 years, internal rate of return of 60% and exhaust temperature of 858.50 [K]. Nevertheless, biodiesel blend is preferable because it is biodegradable, produces less emissions, and as a consequence, environmentally benign. Biodiesel blend would be more suitable for firing gas turbine engines, if the combustor is redesigned to improve its efficiency. Thermo-economic analysis of gas turbine power plants is essential to improve its thermodynamic and economic performance.

Keywords: Gas turbine power plant; Thermodynamic and Economic analysis; Biodiesel blend-fired gas turbine; Bio-energy; Electricity generation; Exergy rate.

1 Introduction

The world's energy demand is projected to grow significantly over the next 20 years. This increase stemmed from economic growth, industrial expansion, high population growth, and urbanization, and has become a big issue especially in the developing countries. Most of the rising energy demand is to be met by using non-renewable fossil fuels with a limited supply and very negative environmental impacts [1]. Energy demand in Nigeria has been rising and only about 50% of the total population have access to grid electricity [2]. Meanwhile, most of the energy demand are currently being met by peak load generation plants such as gas and combined gas/steam turbine power plants, which are contributing approximately 60.7% of the total installed capacity of the nation, as at 2012 [3]. Over the past three decades, gas turbines have become dominant sources of power for large scale power generation and for mechanical drives application. Factors that resulted in its increased utilization include: improved thermal efficiency; availability and the ability to operate on a wide spectrum of gaseous and liquid fuels [4]–[6].

Whilst natural gas has been the fuel of choice, there are large populations of engines worldwide in which liquid fuels are used. Diesel (No.2 Diesel) has been one of the standard liquid fuels used to power gas turbine engines over the years [4]. The natural fuel fired gas turbines, release its exhaust gas containing several harmful gases such as CO, SOx and even un-burnt hydrocarbons to the environment. Generally speaking, these gases are contributing significantly to the climate change crisis, being experienced globally. As a consequence, the scientific community has been focusing on studies, to corroborate the viability of deploying bio-fuels and ethanol, as alternative sources of fuel for firing these thermal power plants. Bio diesels, however, are basically fatty acids, ethyl or methyl ester, made from virgin or used vegetable oils (both edible and non-edible) such as Jatropha Curcas and Pongamia Pinnata [7], that have similar characteristics as the diesel fuels, although with slightly higher viscosity [8]. These fuels are reported to produce low carbon emissions, and have no sulfur content compared to the conventional fuels, making them environmentally benign [9].

In the literature, several studies comparing the performance of fossil fuel fired gas turbine plants to bio-derivative fuels have been undertaken [10]–[13]. Kunte [11] investigated the opportunity of deploying either of biogas or syngas to fire a 200kW micro-gas turbine with a heat recovery steam generator, intended to reduce the carbon emissions from the thermal power plant. He reported that the efficiency of the thermal power plant using natural gas, biogas or syngas were 50.9%, 48.61% and 47.9% respectively. Escudero et al. [10] in their study compared the performance of pure methane fired gas turbine with that of biogas, ethanol and synthesis gas fired gas turbine. The energy and exergy audit they performed suggested that the efficiency of the plant with these fuel sources were comparable, however, the ethanol fired gas turbine produced the least value of exergetic efficiency.

Unfortunately, there are some limitations with using pure bio fuels to fire gas turbines, such as the poor energy density of the fuels, due to their characteristic low calorific values. Consequently, efforts are being advance to overcome this

Toochukwu.udeh@uniport.edu.ng
Department of Mechanical Engineering, Faculty of Engineering, University of Port Harcourt, PMB 5323, Port Harcourt, Nigeria.

weakness mainly by blending it with conventional fuels, although most of the successful attempts were largely on internal combustion engines [14]–[16]. For instance, Navindgi, Dutta and Kumar [17] reported a comparable performance in the diesel engine fired by biodiesel blend comprising mahua methyl ester (MME) and diesel fuel, to that fired by only diesel fuel. However, the authors reported a reduction in the IC engine's power output with an increase in the concentration of the MME in the diesel fuel. While in the experimental investigation conducted by Vijajakumar et al. [18], a reduction in the carbon emissions of the diesel engine was reported when using the MME diesel fuel blend to fire the IC engine. On the other hand, the authors concluded from the results of the brake specific fuel consumption (Bsfc) that the engine would consume more volumes of the biodiesel blend fuel to produce a kW of power compared to the pure diesel fuel.

In another study, Badami et al. [19] compared the performance of three fuels – Jet-A kerosene, a synthetic gas to liquid (SGL) fuel and a blend of 30% Jatropha methyl ester (JME) and 70% Jet-A kerosene – when used to fire a small-scale turbojet engine. The authors reported higher specific fuel consumption for the bio-diesel blend fuel compared to the other fuels, although the thermodynamic performance of all the fuels was comparable. On the other hand, the experimental results suggested that the SGL and JME – Jet-A fuels produced lower NOx and CO emissions compared to the traditional Jet-A-kerosene, whilst the bio-diesel blend yielded less unburnt hydro carbon emissions among all the fuels. Somorin and Kolios [20] investigated the techno-economic viability of replacing natural gas with Jatropha bio-diesel to fire an open cycle gas turbine engine. They reported a marginal difference in the engine's power output and efficiency of about 2% and 1% respectively, when using Jatropha bio-diesel compared to the conventional fuel. However, the results of the economic analysis revealed that the biodiesel blend fuel is unviable to be deployed to fire the thermal power plant. One more advantage of blending bio-fuels with conventional fuels has been revealed in [13]. Here, the authors investigated the effects of bio-fuel mixing ratio (mixing ratio of Jatropha pure oil or Jatropha methyl ester to diesel blend) on the combustion characteristics of a combustor, using air-assisted pressure swirl atomiser experiments. They concluded that mixing bio-fuel with liquid fossil fuels such as diesel fuel or heavy oil may extend the life of the combustor, by reducing the flame radiation intensity. The flame radiation intensity is responsible for the soot volume fractions, and high radiation heat transfer – capable of damaging the combustor.

Another problem that may arise from using pure bio-fuel in firing gas turbine engines is the formation of gum on the fuel injector. This problem can be overcome by the redesign of the fuel injector [21]. There are some new designs of gas turbine combustors with atomisers that are capable of handling fuels of different nature. For instance, according to LBNL [27], their low swirl injector (LSI) Gas turbine has a unique fuel flexibility capability, meaning that the combustion turbines running on natural gas now would run on carbon-neutral, bio- or waste gases. The LSI produces low emissions with no cost premium; no need for substantial redesign of the basic gas turbine, and no need for expensive materials such as catalysts. Whilst for some improved combustor design like the LSI, there is no need to modify the combustor to use bio-diesel (mostly the case for internal combustion engines), in the traditional combustors, the redesign of the fuel injector and retrofitting of a pre-vaporisation premixing tube (PP tube) to the combustor is essential for evaporation of the fuel spray [12].

From the foregoing discussions, it is evident that the bio-fuel blends have similar performance characteristics as the conventional fuels, when employed to fire thermal power plants. However, they are more attractive from the stand point of reduced emissions as has been revealed in the literature. Howbeit, despite the promising future of biodiesel blend in firing gas turbine engines as has been reported by the past investigators, there is a paucity of information on the quality of the energy, i.e. the exergy destroyed in the components using both fuels. Most of the previous studies have been limited to the comparative energetic and economic analysis of the gas turbine engine, using various fuels. Traditionally, exergetic performance analysis is often conducted to reveal the thermodynamic imperfections (irreversibilities) in a power plant. This has been pivotal to the design improvement of the components with the greatest proportion of exergy destruction, so as to enhance the overall performance of the system.

The present study is intended to fill this gap, comparing the quality of energy in the major components of the Brayton cycle power plant, using both fuels. An attempt will be made to ascertain the economic potentials of using bio-fuel blend to fire the power plant, as opposed to using the natural fuel. Finally, results will be simulated to show the effect of the ambient condition, and other key parameters on the plant's performance. This study has been undertaken with the aid of MATLAB which was employed to model the plant performance, using statistically analyzed daily operating data, collected from a GE frame 52G9E gas turbine plant. The grid connected power plant is a 125MW barge open cycle gas turbine power plant, the Ogorode Generation Company, located in Southern Nigeria. Gas turbines in Nigeria use natural gas as their primary fuel. Any shortage in natural gas supply to these plants will result in the plant lying idle, worsening the poor state of electricity generation in the nation. Since gas turbines can be installed to run on liquid fuel such as biodiesel, there is need to ascertain the viability of adopting the biodiesel blend fired or to incorporate both in the system to meet the future power requirement while making the plant greener and minimizing the running cost. The difference in the performance of the power plant fired by the two fuels is studied from two perspectives:

- Thermodynamic – By applying the First and Second Laws of Thermodynamics and
- Economic view points

The study will be limited to:

- The major components of an open cycle Gas Turbine power plant;

- The major components will be treated as control volumes and only the energy and mass flow at inlet and exit from the components will be considered and
- It will not include the mechanical, thermal and hydrodynamic design of the components.

2 Problem formulation and solution methods

This study of the gas turbine thermal power plant, involves the thermodynamic and economic analyses of the system which would be the crux of the following sub-sections.

2.1 System Configuration

The open gas turbine power plant comprises three major components including the compressor (TC), combustion chamber (CC) and turbine (GT). The schematic diagram of a simple gas turbine is shown in Figure 1a. Fresh atmospheric air is drawn into the TC continuously, compressed and supplied to the CC. Energy is added in the CC by the combustion of the fuel which serves as the working fluid. The products of combustion are expanded through the GT producing the useful work, and are finally discharged to the atmosphere via the stack. The ideal and actual thermodynamic processes involved in the well-known Brayton cycle, are represented in full and dashed lines respectively, on the temperature-entropy cycle diagram shown in Figure 1b.

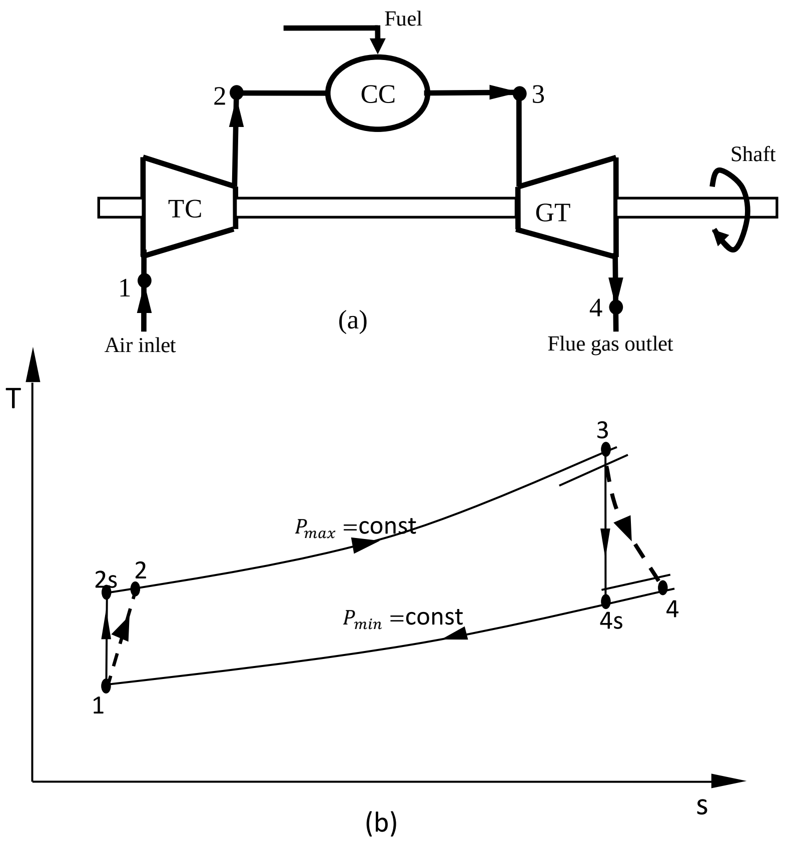

Figure 1(a) Schematic diagram of a simple open cycle Gas Turbine power plant and (b) Ts diagram of a Gas Turbine cycle

2.2 Formulation of Energy Audit Models

Applying the First Law of Thermodynamics, the model for predicting the thermodynamic performance of the thermal power plant is presented to evoke the subsequent techno-economic analysis.

$$\dot{W}_c = \dot{m}_a C_{pav}(T_2 - T_1) \tag{1}$$

where C_{pav} $(\frac{kJ}{kgK})$ *is* specific heat capacity of air, $\dot{m}_a$ $(\frac{kg}{s})$ is mass flow rate of air, $\dot{W}_c$ (W) is power consumed by the compressor, $T_{1,2}$ (K) are the stream temperatures at inlet and exit respectively.

According to Rahman et al. [22], the average specific heat capacity of compressed air is fitted as:

$$C_{pav} = 1.0189 \times 10^3 - 0.13784(T_a) + 1.9843 \times 10^{-4}(T_a^2) + 4.2399 \times 10^{-7}(T_a^3) - 3.7632 \times 10^{-10}(T_a^4) \tag{2}$$

The turbine power output is expressed below.

$$W_t = \dot{m}_T C_{pg}(T_3 - T_4) \quad (3)$$

with $\dot{m}_T = \dot{m}_f + \dot{m}_a$; where $\dot{m}_f$ is the fuel flow rate and C_{pg} is the specific heat capacity of the flue gas.

The net power output of the power plant is then computed by applying the First Law of thermodynamics and is expressed as

$$\dot{W}_N = \dot{W}_t - \frac{\dot{W}_c}{\eta_m} \quad (4)$$

where η_m, the mechanical efficiency of the drive has been taken to be 100%. This choice of mechanical efficiency stems from the fact that the impact of the mechanical efficiency on the net power output is expected to be uniform, for both fuels. However, in practice, the mechanical efficiency of the drive shaft which accounts for the frictional losses in the drive is usually less than 100%. In the literature, a mechanical efficiency of 98% has been adopted by Somorine and Kolios [20] in their study.

2.3 Formulation of Fuel Consumption Models

To determine the fuel efficiency of the gas turbine power plant using these fuels, the specific fuel consumption (sfc) is used to model the plant. Specific fuel consumption is determined from the following expression;

$$sfc = \frac{3600}{\dot{W}_N}\dot{m}_f \quad (5)$$

2.4 Heat Rate and Efficiency Models

The heat rate (HR) is a measure used to determine how efficiently a generator uses heat energy. It can be expressed as:

$$HR = \frac{Heat\ Supplied}{\dot{W}_N} \quad (6)$$

The heat rate in terms of sfc and lower heating value of fuel (kJ/kg), LHV is determined as;

$$HR = sfc \times LHV \quad (7)$$

The gas turbine power plant thermal efficiency (η_{th}) is the percentage of the total fuel energy input that appears as the net work output of the cycle. It is a measure of the thermodynamic perfection of a system, from the quantity view-point.

$$\eta_{th} = \frac{\dot{W}_N}{\dot{Q}_{in}} \quad (8)$$

In terms of sfc and LHV, the thermal efficiency is given as;

$$\eta_{th} = \frac{3600}{sfc.LHV} \quad (9)$$

2.5 Formulation of Models for Exergy Analysis

Exergy analysis is necessary to determine the quantity of useful energy available to a system, and the efficiency of the system with respect to converting the useful energy to useful work [29]. Equation models for determining the exergy balance of the major components of the Gas turbine power plant were formulated, based on the Second Law of Thermodynamics. These equation models which neglected the kinetic and potential exergy parameters are presented for the major components of the plant.

The reference (dead) state temperature and pressure for the exergy analysis is defined as: $T_0 = 298.16$ K and $P_0 = 0.101325$MPa respectively.

2.5.1 The Compressor Inlet

The exergy of compressor inlet, $\dot{E}x_1(kW)$ is calculated using the following relation:

$$\dot{E}x_1 = \dot{m}_a C_p^h(T_1 - T_0) - \dot{m}_a T_0(S_1 - S_0) \quad (10)$$

$$S_1 - S_0 = C_p^s In\left(\frac{T_1}{T_0}\right) - RIn\left(\frac{P_1}{P_0}\right) \quad (11)$$

To account for the effect of the relative humidity on the exergy destroyed in the power plant, a more detailed expression incorporating the specific humidity of the inlet air has been formulated below, to determine the exergy of the inlet air.

$$\dot{E}x_1 = \dot{m}_a\left[\left(C_{pa} + \omega C_{pv}\right)\left(T_1 - T_0 - T_0\left(\log\left(\frac{T_1}{T_0}\right)\right)\right) + (1+\omega_0)R_aT_0\left(\log\left(\frac{P_1}{P_0}\right)\right) + R_aT_0\left((1+\omega_0)\left(\log\left(\frac{(1+\omega_1)}{(1+\omega_0)}\right)\right) + \omega_0\left(\log\left(\frac{\omega_0}{\omega_1}\right)\right)\right)\right] \quad (12)$$

where $\omega_0(kg\ w.v/kg\ d.a)$ is the humidity ratio of the air at the dead state and $\omega_1(kg\ w.v/kg\ d.a)$ is the humidity ratio of the air at the ambient condition.

For the compressor; $C_{pi} = C_{pa,v}$

For the combustion chamber and turbine; $C_{pi} = C_{pg}$

C_p^h is the isobaric heat capacity for evaluating enthalpy, C_p^s is the isobaric heat capacity for evaluating entropy.

2.5.2 The Compressor outlet

The exergy rate at the compressor outlet, $\dot{E}x_2(kW)$ outlet is given as;

$$\dot{E}x_2 = \dot{m}_aC_p^h(T_2 - T_0) - \dot{m}_aT_0\left[C_p^s In\left(\frac{T_2}{T_0}\right) - RIn\left(\frac{P_2}{P_0}\right)\right] \quad (13)$$

where T_2 is the compressor outlet temperature, P_2 is the pressure at compressor exit.

The total exergy destruction rate (irreversibility), $\dot{I}_c(kW)$ in the compressor is given by

$$\dot{I}_c = \dot{E}x_1 - \dot{E}x_2 + \dot{W}_c \quad (14)$$

The exergy (second law) efficiency of the compressor, $\eta_{\mathrm{II,c}}(-)$ process is given as;

$$\eta_{\mathrm{II,c}} = \frac{\dot{E}x_2}{\dot{E}x_1 + \dot{W}_c} \quad (15)$$

2.5.3 The Turbine Inlet

The exergy at the turbine inlet $\dot{E}x_3(kW)$ is expressed as;

$$\dot{E}x_3 = \dot{m}_tC_p^h(T_3 - T_0) - \dot{m}_tT_0\left[C_p^s In\left(\frac{T_3}{T_0}\right) - RIn\left(\frac{P_3}{P_0}\right)\right] \quad (16)$$

$T_3(K)$ is the Turbine inlet temperature, $P_3(kPa)$ is the Turbine inlet pressure.

2.5.4 The Turbine outlet

The exergy rate at the turbine outlet $\dot{E}x_4(kW)$ is defined as;

$$\dot{E}x_4 = \dot{m}_tC_p^h(T_4 - T_0) - \dot{m}_tT_0\left[C_p^s In\left(\frac{T_4}{T_0}\right) - RIn\left(\frac{P_4}{P_0}\right)\right] \quad (17)$$

$T_4(K)$ is the Turbine outlet temperature and $P_4(kPa)$ is the Turbine outlet pressure.

The total exergy destruction rate (irreversibility) in the turbine, $\dot{I}_T(kW)$ is given by

$$\dot{I}_T = \dot{E}x_3 - \dot{E}x_4 - \dot{W}_T \quad (18)$$

The exergy (second law) efficiency of the turbine expansion process, $\eta_{\mathrm{II,t}}(-)$ is given as;

$$\eta_{\mathrm{II,t}} = \frac{\dot{E}x_4 + \dot{W}_T}{\dot{E}x_3} \quad (19)$$

2.5.5 The Combustion Chamber (CC)

The exergy flow rate in the CC is obtained as;

$$\dot{E}x_f = \dot{m}_fC_p^h\left(T_f - T_0\right) - \dot{m}_fT_0\left[C_p^s In\left(\frac{T_f}{T_0}\right) - RIn\left(\frac{P_f}{P_0}\right)\right] + \dot{m}_f(LHV) \quad (20)$$

where, $\dot{E}x_f(kW)$ is the exergy of fuel, $T_f(K)$ is the temperature of fuel, and $\dot{m}_f\left(\frac{kg}{s}\right)$ is the mass flow rate of fuel.

The total exergy destruction rate (irreversibility) in the combustion chamber $\dot{I}_{cc}$ is given as;

$$\dot{I}_{cc} = \dot{E}x_f + \dot{E}x_2 - \dot{E}x_3 \quad (21)$$

The exergy (second law) efficiency of the combustor processes $\eta_{\mathrm{II,cc}}$ is given as;

$$\eta_{\mathrm{II,cc}} = \frac{\dot{Ex}_3}{\dot{Ex}_f + \dot{Ex}_2} \quad (22)$$

2.6 Performance Analysis

The gas turbine (GT) performance is affected by component efficiencies and turbine working temperatures. The overall efficiency of the gas turbine cycle depends primarily upon the pressure ratio of the compressor. The performance of the plant can be qualified with respect to its efficiency, power output, and specific fuel consumption as well as work ratio. There are several parameters that affect its performance including the compression ratio of the compressor, compressor inlet temperature and turbine inlet temperature (TIT). Results were simulated to show how these parameters affect the performance of the gas turbine power plant.

2.7 Economic Analysis

The department of energy and climate change (DECC), UK outlines some parameters used in the economic appraisal of a project as: Simple Payback Period (SPBP); Net Present Value (NPV); and Internal Rate of Return (IRR) [33]. These capabilities are captured as contents of the economic method developed in this work. A discounting technique based on NPV method is presented to allow an assessment of the economy of Natural gas powered and biodiesel blend powered combustion turbine.

2.7.1 Cost of Fuelling the Plant

To accurately determine the cost of operating the plant, the cost of fuel (blended biodiesel and Natural gas) required for a given period of operation has to be accurately determined. In this study, the volume of fuel consumed for a given period was determined from the thermodynamic analysis. The cost of fuelling the plant was then calculated by simply multiplying the volume of fuel by the cost per unit volume of each fuel.

The cost of fuelling the plants is given by:

$$C_f = V_f \times C_{fv} \quad (23)$$

where, C_f(₦) is the Cost of fuelling the plant, $V_f(\mathrm{m}^3)$ is the Volume of fuel used for N(yrs) and C_{fv}(₦$/m^3$) is the Cost per unit volume of fuel.

$$V_f = V_{fh} \times E_p \times h \quad (24)$$

where, V_{fh} is the Volume of fuel consumed per MWh

2.7.2 Revenue from Selling Electricity

The revenue obtained from selling electricity, R_e(₦) is calculated from the relation:

$$R_e = C_E \times E_p \times h \quad (25)$$

Where C_E(₦/MWh) is the Electricity tariff, $E_p(\mathrm{MW})$ is the Electric power generated and h is the Hours of operation.

2.7.3 Net Cash Flow (C_t)

The net cash flow generated by the plant is calculated as:

$$C_t = R_e - C_f - C_{o/m} \quad (26)$$

where, $C_{\frac{o}{m}}$(₦) is the Operation/maintenance cost.

$$C_{o/m} = \left[\left(V_{o/m} \times E_p \times h/yr\right) + \left(F_{o/m} \times E_p/yr\right)\right] \times N \quad (27)$$

where, N is the Numbers of years, $V_{o/m}$ is the variable operation/maintenance cost and $F_{o/m}$ is the fixed operation/maintenance cost

2.7.4 Present Value (PV)

The calculation involving the annual net cash flows and discounting using an estimated appropriate rate of interest gives the present value (PV) of the cash flow.

The PV is calculated as:

$$PV = \frac{C_t}{(1+r)^t} \tag{28}$$

where, r (%) is the market interest rate (discount rate), and t is the time of operation in years.

2.7.5 Net Present Value (NPV)

NPV is a Capital Investment (CI) appraisal which measures the cash in-flow, discounting it over the life span of the project and gives the present worth. The NPV method shows the importance of the cash received now over cash received in the future. Basically, NPV is a mathematical calculation which involves calculating the annual net cash flows and discounting using an estimated appropriate rate of interest, thus giving the present values of the cash flow, which are added together to obtain the NPV [6]. The NPV capital investment appraisal method was applied in the present study. According to Nkoi et al. [23], the NPV is given as:

$$NPV = -C_o + \sum_{t=1}^{N} \frac{C_t}{(1+r)^t} \tag{29}$$

where, C_o(₦) is the Capital cost of installation of the plant.

2.7.6 Internal Rate of Return (IRR)

The IRR is concerned with the rate of return which gives the total NPV equal to the total initial cost. IRR presents the efficiency of the capital investment. The internal rate of return is the interest rate at which the net present value (NPV) equals zero.

If at a certain value of r, NPV = 0, then r = IRR.

2.7.7 Payback Period (PBP)

The payback method of appraisal is a technique that estimates the time needed for a project to recover the initial investment and afterwards, starts to yield some profits. After investing in the power plant, the NPV is expected to be negative, and would gradually begin to tend towards a positive value as the power plant becomes operational and starts to yield some revenue. The payback period is simply the time period taken for the NPV to acquire a positive value. This method was used in the present study to obtain the payback period for each of the fuels.

3 Results and Discussion

The input data for the analysis of multi fuel fired gas turbine power plant under study are presented below.

Table 1 Input data for Thermodynamic Analysis. Ogorode Generation Company, CHP [28]

S/N	**PARAMETER**	**SYMBOL**	**UNIT**	**VALUE**
1	Ambient Temperature	T_o	K	302.15
2	Ambient Pressure	P_o	MPa	0.1003
3	Compression ratio	r	-	8.5
4	Operating efficiency of turbine	η_{th}	-	0.3
5	Specific heat ratio for air	γ_a	-	1.4
6	Specific heat ratio for Natural gas fuel	γ_g	-	1.33
7	Pressure drop in the Combustor	ΔP_c	%	2.0
8	Low Heating Value of Natural gas	LHVg	MJ/kg	47.54
9	Low Heating Value of biodiesel blend	LHVd	MJ/kg	42.51
10	Fuel -Air ratio at full load	f	-	0.036
11	Compressor efficiency	η_c	-	0.85
12	Turbine efficiency	η_t	-	0.87
13	Combustion efficiency	η_{comb}	-	0.95
14	Plant size under consideration	WN	MW	160
15	Years of Operation	N	Yrs	25
16	Density of Natural gas at STP	ρ_f	kg/m^3	450
17	Density of biodiesel blend at STP	ρ_{f1}	kg/m^3	900

The plant data from the Ogorode Generation Company has been used to validate the model built in MATLAB, to ascertain the accuracy of the program in predicting the actual plant performance. The percentage error computed reveals that the program can predict the actual performance with about 7% error, as depicted in Table 2.

Table 2 Validation of the MATLAB program

S/No	Quantity	Symbol	Unit	Value		Error
				Plant	Program	(-)
1	Compressor outlet temperature	T_2	K	613.15	582.08	0.05
2	Flue gas maximum temperature	T_4	K	863.15	915.74	0.061

3	Net power delivered by plant	W_{net}	MW	93.86	86.93	0.071

Table 3 Input data for Economic Analysis. Samuel et al. [30], PHED [31]

S/No	Parameter	Symbol	Unit	Value
1	Cost of Natural gas	N_c	₦/ m³	17.66
2	Cost of biodiesel blend	L_c	₦/ m³	73000
3	Electricity Tariff	C_E	₦ /kWh	15.9
4	Average None fuel Fixed O&M cost for 125MW gas fired Plant	F_{o/m^g}	₦/kW/Year	2400
5	Average None fuel variable O&M cost for 125MW gas fired Plant	V_{o/m^g}	₦/kW/Year	1100
6	Average None fuel Fixed O&M cost for biodiesel fired Plant	F_{o/m^d}	₦/kW/Year	2400
7	Average None fuel variable O&M cost for biodiesel fired Plant	V_{o/m^d}	₦/kW/Year	1200
8	Capital Cost of Plant (Same for both natural gas and biodiesel fired plant for Siemens SGT5 2000E)	C_o	₦/kW	157400

3.1 Result of Thermodynamic Analysis

Table 4 Comparing the Plant's energy performance using both fuels

S/No	Parameter	Symbol	Unit	Value	
				Natural fuel fired	Blended fuel fired
1	Specific Fuel Consumption	SFC	kg/kWh	0.3151	0.3884
2	Mass flow rate of Fuel	$\dot{m}_f$	kg/s	7.3512	7.3512
3	Mass flow rate of air	$\dot{m}_a$	kg/s	233.72	233.72
4	Combustion Pressure	P_3	MPa	0.8440	0.8440
5	Compressor Power	$\dot{W}_c$	MW	169.70	169.70
6	Turbine Inlet Temperature	TIT	K	1468.20	1376.40
7	Exhaust Temperature	T_4	K	915.74	858.50
8	Heat Addition in the Combustion	Qin	MW	357.98	312.49
9	Turbine Power	$\dot{W}_t$	MW	253.74	237.88
10	Heat Rate	HR	-	2.3368	2.3277
11	Volume of Fuel used	Vf	m³/MWh	0.4207	0.2352

3.2 Result of Exergy Analysis

Table 5 Comparing the Plant's exergy destruction for both fuels

S/No	Parameter	Symbol	Unit	Value	
				Natural fuel fired	Blended fuel fired
1	Exergy at compressor inlet	$\dot{E}x_1$	MW	0.000041	0.000041
2	Exergy at compressor exit	$\dot{E}x_2$	MW	127	127
3	Irreversibility in the compressor	$\dot{I}_c$	MW	41.82	41.82
4	Exergy efficiency of compressor	η_{xc}	-	0.7536	0.7536
5	Exergy at turbine inlet	$\dot{E}x_3$	MW	397.93	364.61
6	Exergy at turbine exit	$\dot{E}x_4$	MW	129.99	112.54
7	Irreversibility in the turbine	$\dot{I}_T$	MW	14.1953	14.1953
8	Exergy efficiency of turbine	η_{xt}	-	0.9470	0.9437
9	Exergy of fuel	$\dot{E}x_f$	MW	363.46	337.50
10	Irreversibility in the combustor	$\dot{I}_{cc}$	MW	93.4493	100.81
11	Exergy efficiency of combustor	η_{xcc}	-	0.7429	0.7013
12	Exergetic efficiency of the plant	η_{II}	-	0.4330	0.4647

From the results presented in the table (Table 4), the biodiesel blend fired power plant consumed more fuel per MW of power produced compared to the natural gas fired gas turbine plant, as expected, owing to the low calorific value of biodiesel as well as its slightly higher viscosity. Some variations were recorded also in the net power and efficiency of the plant using both fuels, although it was not very significant [6, 19, 14].The exergy destruction rates are higher for biodiesel blend than for natural gas (see Table 5), because of the slightly high viscosity of bio diesel blend and the consequent poor mixing of the fuel and air, resulting to incomplete combustion of the mixture. These values would improve if the atomizers are optimized [16].

The MATLAB results reveal that more than 60% of the exergy destructions occurred in the combustor, with the compressor and turbine processes contributing the remaining [8]. In this study, a combustor efficiency of 95% has been used for both fuels [20]. This is necessary to ensure uniformity in the parameters of the engine used to compare the performance of both fuels. In practice, the combustor efficiency when employed to fire biofuel, would be less than the assumed value. And if this is the case, the exergy destruction rate in the combustor would be more, further deteriorating the performance of the engine, whilst running on the biodiesel blend. This therefore, elicits further research on pathways to improve the efficiency of the combustor of the conventional Brayton cycle, so as to accommodate other green fuels in the future. Exergy rate at the turbine outlet is higher for the natural gas than the biodiesel blend, showing that the low grade energy from the natural gas exhaust would be more suitable to produce additional power from other devices like the Rankine cycles (organic and steam), than that of the biodiesel blend. However, in all, the exergetic efficiency of the biodiesel blend gas turbine is higher than that of the natural gas fired turbine.

3.3 Result of Performance Analysis

The performance of the plant was investigated for the two fuels: Natural gas and biodiesel blend. The data generated were plotted on the spread sheet using MATLAB. The plots of the simulations for the biodiesel blend fired and natural gas fired combustion turbine are presented and analyzed here.

3.3.1 Effect of TIT on Thermal Efficiency

The effect of the turbine inlet temperature (TIT) on the thermal efficiency of the plant was investigated for the TIT values range of 1200K – 1800K, at a step increase of 100K. The results show that the thermal efficiency increased with the increase in the turbine inlet temperature as it is seen in Figure 2, for both fuels. Interestingly, the plant's thermal efficiency started to decline after attaining a peak value at around 1480K (1375K for biodiesel blend), which coincides with the plants operating TIT. This is expected, as further increases in the TIT, would result in additional compressor work offsetting the initial gain. The trend of the graph, which appears to align with that of the measured data from the running plant, further supports the claim that the model is capable of predicting the actual plant's performance to a tolerable limit. It is remarkable to note the point of intersection of the two curves (~1375K), as it may be the suitable temperature for introducing biodiesel blend in firing a gas turbine (at this pressure ratio), if maximum benefits would be derived. Plotting the TIT against the Fuel-air ratio (Figure 3) confirms the earlier trend in the graph. The TIT of the biodiesel blend is lower because of the low calorific values of biodiesel.

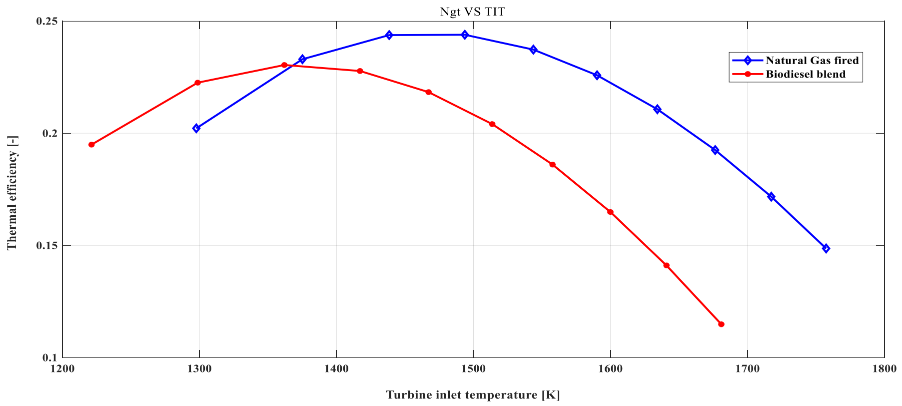

Figure 2 Effect of TIT on the thermal efficiency of the gas turbine engine

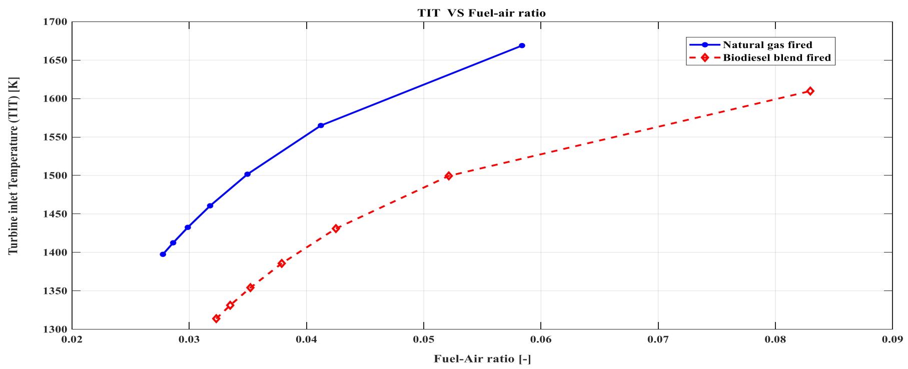

Figure 3 Effect of Fuel-air ratio on the TIT of the Brayton cycle engine.

3.3.2 Effect of Pressure Ratio on Thermal Efficiency

The pressure ratio has been varied from 4 – 22, and the corresponding thermal efficiency of the plant computed. Similar trends were observed for both fuels as it is seen from the presented data in Figure 4. The results show that at low and high pressure ratios (above or below the design condition), the thermal efficiency of the gas turbine plant would deteriorate

(Figure 4). The optimum pressure ratio for operating this plant is 8-10 for natural gas fuel, and just around 8 for biodiesel blend. This suggests that natural gas could offer more flexibility in running a gas turbine plant compared to the biodiesel blend.

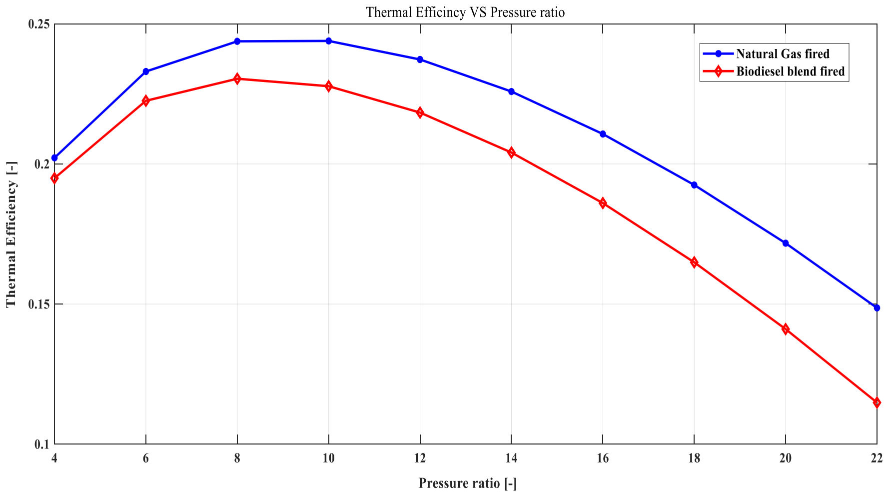

Figure 4 Effect of pressure ratio on the engine's thermal efficiency

3.3.3 Effect of Fuel-air ratio on Thermal Efficiency

The effect of fuel-air ratio on the thermal efficiency of the plant was investigated by varying the fuel-air ratio from 0.02 to 0.05 which represents both the lean and rich mixtures ratio. Efficiency versus fuel-air ratio graph was plotted and the result has been presented in Figure 5. The result shows that the thermal efficiency increased with increase in the fuel-air ratio, as it would be expected for both fuels. This simply buttresses the traditional significant role oxygen plays in the combustion of fuels. As expected, a fuel rich in oxygen will support the rise in the input energy of the plant. Similar trends were observed for both fuels, although slightly higher values of efficiency were produced by the natural gas fired plants as compared to the biodiesel blend. Furthermore, this result is a pointer to the fact that more biodiesel fuel would be required to attain similar thermodynamic performance, as can be obtained while using natural gas to fire the plant.

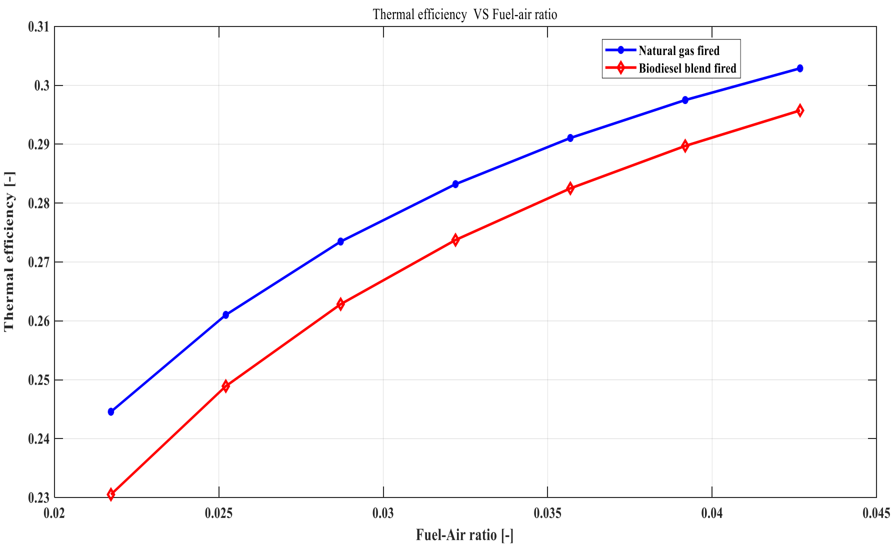

Figure 5 Effect of Fuel-air ratio on the engine's thermal efficiency

3.3.4 Effect of Ambient Condition on Turbine Performance

To investigate the effect of the ambient condition on the performance of the combustion turbine, the ambient temperature and the relative humidity were varied from 293K (20^{0}C) to 305K (32^{0}C) and 50% to 90% respectively. These ranges of values were selected based on the daily ambient condition in southern Nigeria, where the gas turbine plant is installed. The results of the simulations show that the turbine efficiency decreased with increase in the ambient temperature for both fuels. Figure 6 and Figure 7 show the effect of the ambient temperature on the thermal efficiency, and the plant's power output. It can be seen that increasing the ambient temperature would impact the power output, negatively [24], [25], [32]. Impacts of the ambient temperature and the relative humidity, on the gas turbine plant's First and Second Laws efficiencies have been simulated and presented in Figures 8 and 9 respectively. The results suggest that low ambient temperature favors the performance of the gas turbine engine as it is seen in the high thermal and exergetic efficiencies recorded at those conditions.

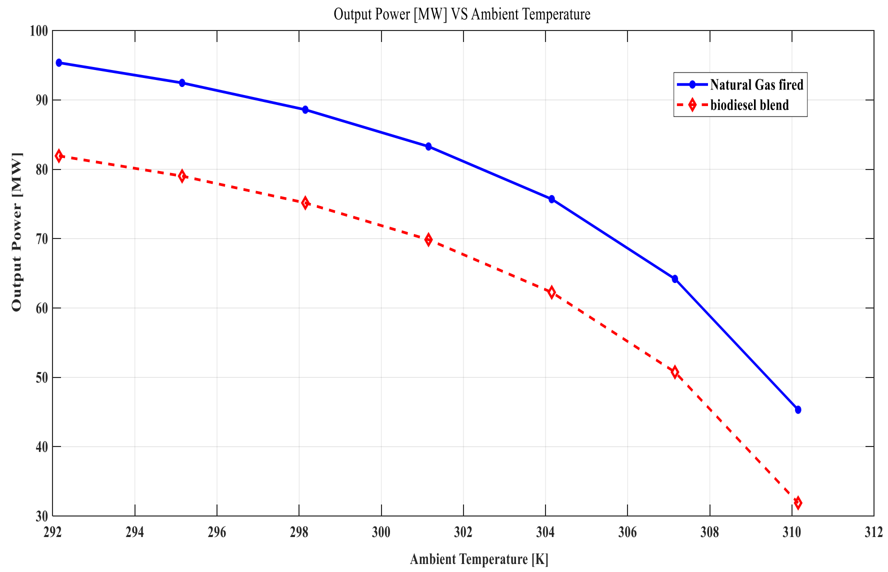

Figure 6 Effect of the ambient air temperature on turbine output power

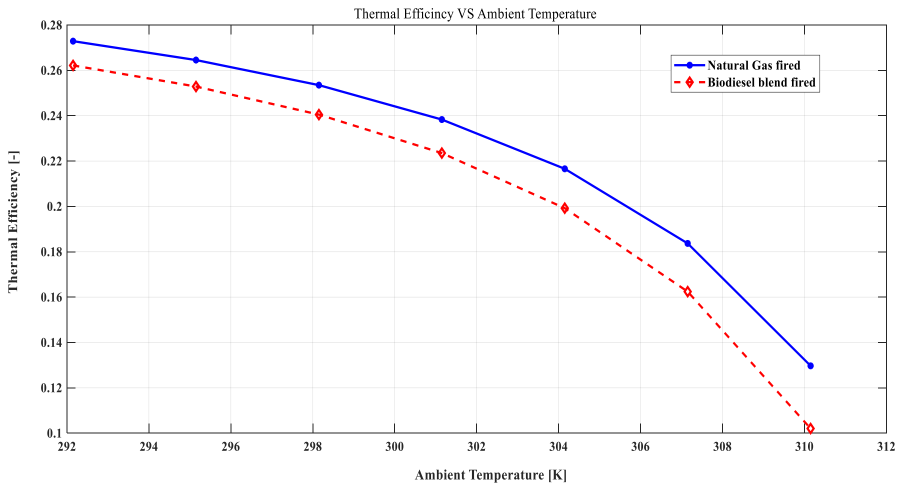

Figure 7 Effect of the ambient temperature on the engine's thermal efficiency

However, the impact of the relative humidity on the plant's performance appears to be more pronounced at higher ambient temperature, with the plant suffering more as the relative humidity increased at elevated ambient temperature. On the other hand, as the ambient temperature of the air sucked into the compressor began to decrease, the effect of the relative humidity on the plant's performance became less pronounced. In particular, at much lower ambient temperature, the effect of the relative humidity may be insignificant as can be seen from the narrow band of the thermal efficiency readings of 0.27 – 0.282, recorded at the ambient temperature of 292K, for the relative humidity of 80, 70, 60 and 50%, respectively. This is in tandem with the ISO optimum conditions for the operation of a gas turbine – ambient temperature (15^{0}C) and relative humidity (60%). Operating a gas turbine plant outside this condition, would bring about huge loss in both the energy and exergy efficiency as it has been revealed from the data presented in the graphs.

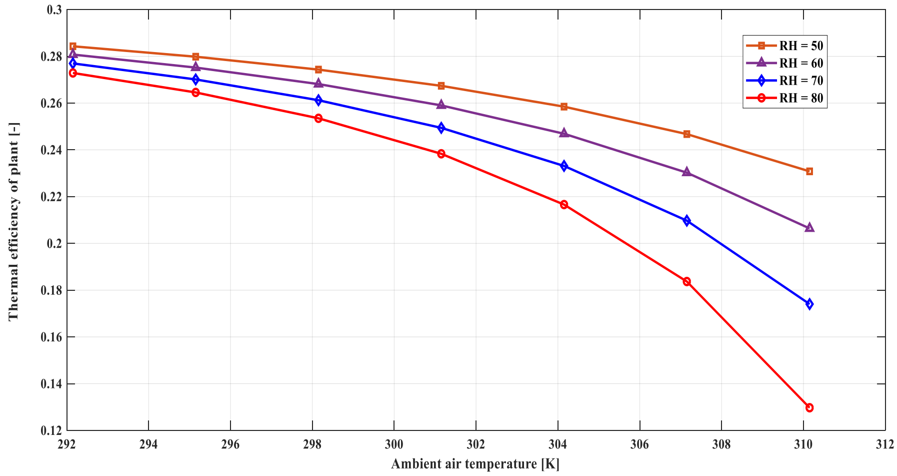

Figure 8 Variation of plant's thermal efficiency with the ambient condition

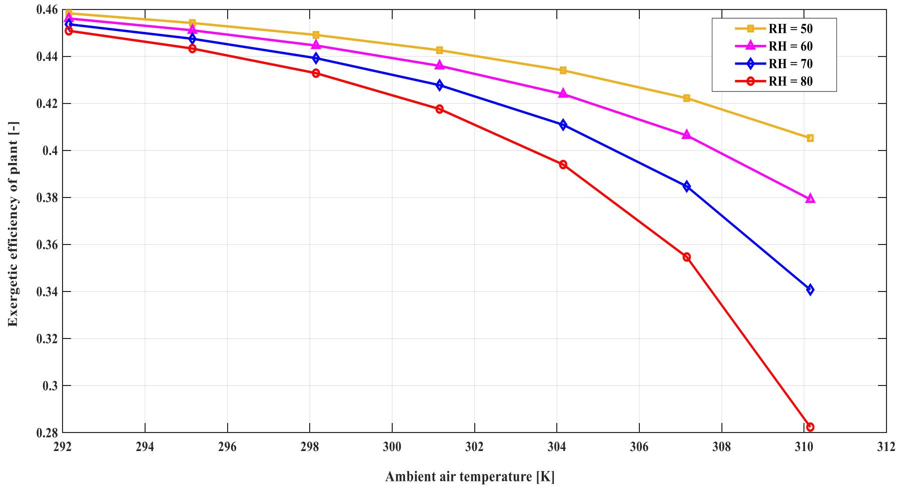

Figure 9 Impact of the ambient air temperature and relative humidity on the gas turbine plant's exergetic efficiency

3.3.5 Effect of sfc on the Power Generation

From Figure 10, increasing the Specific fuel consumption would impact the output power for both fuels, negatively. Of the two fuels, the biodiesel blend fired gas turbine power plant consumed more fuel per MW of the power output than the natural gas fired power plant. The high fuel consumption rate recorded for the biodiesel blend is as a consequence of its low calorific value.

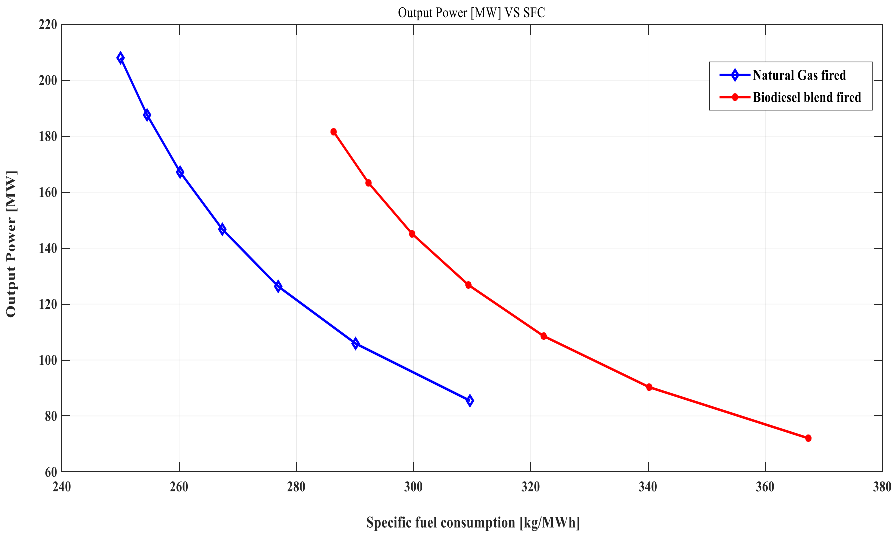

Figure 10 Effect of the engine's specific fuel consumption on the output power

3.4 Result of Economic Analysis

3.4.1 Net Present Value (NPV) and Payback Period

The results of the economic analysis show that the natural gas fired combustion turbine power plant is more economical than the biodiesel blend fired plant [20]. From Figure 12, the NPV for the Natural gas fired plant and the biodiesel blend fired plant are ₦70 Billion and ₦58 Billion respectively for 25 years of continuous operation. The payback period for the Natural gas fired plant is 1.9 years while that for the biodiesel blend fired power plant is 2.4years. This shows that the capital invested is recovered at a shorter time for natural gas than that of biodiesel blend.

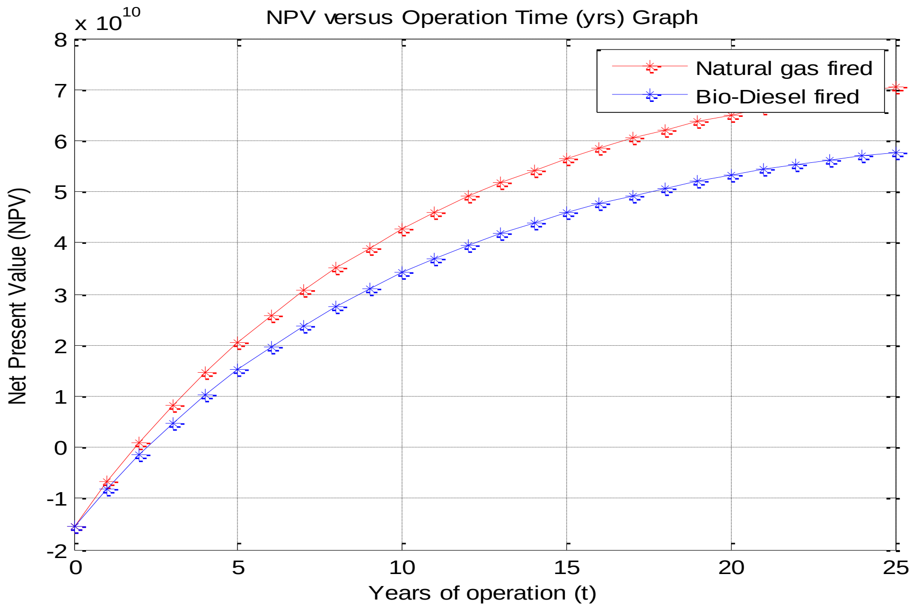

Figure 12 Determination of NPV and payback period of the gas turbine

3.4.2 Internal Rate of Return (IRR)

The internal rate of return is the value of market interest rate at which the NPV is zero. It provides essential information pertaining to the optimum interest rate for obtaining loans from the bank, so as to ensure that the project would pay-back the capital invested within the plant's operating life. In principle, if the computed IRR is less than the interest rate, loss would be incurred in running the plant over its estimated life span. It also serves as a second degree check, for the viability of a plant a prior. From Figure 13, the IRR for Natural gas fired turbine and biodiesel blend fired plant is 52% and 60% respectively. The high value of IRR obtained confirms the initial result suggesting that the plant would pay-back the capital on investment, within a short time in the plant life. Firing the plant using either of both fuels is therefore, viable. <u>It is important to state that the very high values of the IRR recorded may be as a result of the difficulties in estimating the various costs associated with running the power plant.</u>

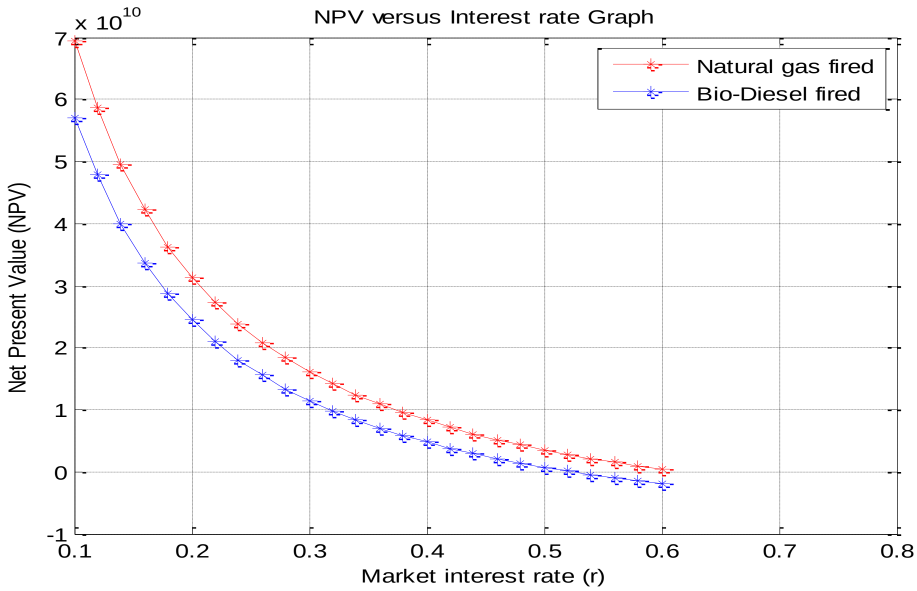

Figure 13 Determination of the IRR of the thermal power plant

4 Conclusion

The use of biodiesel blend for firing gas turbines in power generation is viable. The efficiencies of the gas turbine fired by Natural gas and biodiesel blend were comparable at low TIT values, pressure ratios and ambient condition. However, there is a significant difference in the two at very high operating conditions. The biodiesel blend fired Gas turbine plant could serve as a substitute to natural gas fired plants at low operating conditions. In terms of fuel consumption, natural gas fired Gas turbine plants consumed less fuel, as a result of their high heating values as well as a lower burn out time compared to the bio diesel blend. In particular, results from the exergetic study has identified the combustor to be the most inefficient component in the gas turbine engine. It is crucial to focus on the redesign of the combustor so as to improve the performance of the engine.

The results of sensitivity analysis conducted on the plant have revealed the important role of the plant's parameters in its performance. These results support the need to operate a thermal power plant within the quoted test conditions, so as to derive maximum benefits from its operation. It is essential to retrofit the plant with devices that will help improve its performance by conditioning the ambient air sucked into the compressor at a reduced cost.

The economic analysis of the plant using both fuels, gave high NPV and IRR and low payback period for natural gas fired than for the biodiesel blend fired power plant. Natural gas fired plant has NPV of ₦70 Billion, payback period of 1.9 years and IRR of 52% while biodiesel blend fired Plant has NPV of ₦58Billion, IRR of 60% and payback period of 2.4 years. The values corroborate the plants viability.

Concluding, it is evident that Natural gas fired power plant is a more viable option for commercial power plants especially from economic viewpoint, whereas both fuels show a similar trend from results of thermodynamic analysis. However, the biodiesel blend fired combustion turbine would offer more benefits compared to the natural gas fired, in terms of the impact of the emission to the environment, as seen from the lower value of the stack temperature produced noting also, that the former produces less emissions. Finally, natural gas fired gas turbine power plants will become more beneficial if combined with a steam turbine power plant.

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