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Engineering and Economic Implications of Conversion of a Simple Cycle Gas Turbine Plant to a Combined Cycle Plant

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Abstract

This work investigated the engineering and economic implication of converting simple cycle gas turbine power plants to combined cycle power plants using a gas turbine power plant in Rivers state as a case study. A gas turbine model which behaves like the real engine in the field was created and the performance parameters were estimated. Steam turbine plant operating on the superheat Rankine cycle fired by the exhaust gases of the gas turbine through a heat recovery steam generator was simulated and incorporated in the gas turbine cycle to form a combined cycle plant. Engineering and economic performance parameters such thermal efficiency, specific fuel consumption (SFC), heat rate (HR), payback period, net present value (NPV) and levelized cost of electricity (LCOE) of the gas turbine plant and the simulated combined cycle plant were estimated. The economic parameters were estimated for project lifetime of 20 years, discount rate of 10% and electricity price of N75 per kWh. The thermal efficiencies of the gas turbine cycle steam turbine plant and the combined cycle plant are 33.66%, 27.42% and 50.20% respectively. The SFC and HR of the three systems are 0.0616 kgs-1/MW, 10,695 kJ/kWh; 0.1254 kgs-1/MW, 13127 kJ/kWh and 0.0413 kgs-1/MW, 7171kJ/kWh respectively. The payback, the NPV and the LCOE of the gas turbine plant are 5.32yrs, \$25.87 million and N69.94 per kWh respectively. The respective values of the combined cycle plant are 4.36yrs, \$63.80 million and N66.64. The study shows that it is possible to obtain additional power of over 49% the installed capacity from the existing simple cycle gas turbine power plants.

Keywords: Gas Turbine; Power Plants; Engineering; Economic Implication

I. INTRODUCTION

Currently, the gas turbine is the most adaptable piece of turbo machinery available. Several distinct modes of operation are available in vital sectors such as power generation, oil and gas extraction, process plants, aviation and other smaller linked industries, in addition to household and smaller associated businesses (Alkumait et al,2018). A gas turbine is simply a device that brings together air, which it compresses in its compressor module, and fuel, which is then ignited. The gases that are produced are expanded by the use of a

turbine. Despite this, the turbine's shaft continues to revolve and drive the compressor, which is also on the same shaft, and the machine continues to operate normally. A separate starting unit is utilized to generate the initial rotor motion until the turbine's rotation speed reaches the design speed and is capable of maintaining the complete unit's operational speed. It is possible to generate mechanical power using a gas turbine, provided that there is a constant supply of gaseous working fluid. It makes use of the energy of burning gases and air that is at high temperature and pressure by expanding via a series of fixed and moveable blades in a ring-like configuration (Khaljani *et al.*, 2013).

A centrifugal or axial compressor is necessary to achieve the high pressure, volume, and speed of the working fluid that is required for the expansion process. Axial compressors are recommended for gas turbines that are utilized in the production of electricity. After that, fuel is introduced to the compressed air, which is mixed, and the combustion chamber is lit. Hot exhaust gases are released, which cause the turbine wheels to spin, resulting in the generation of electricity. The gas turbine engine has been proven to be useful in four key applications. Military aviation, civil aviation, oil and gas processing, and power generating are some of the industries involved. It is possible to use a number of fuels in the gas turbine, such as natural gas, diesel fuel, low-Btu gases, vaporized fuel oils, and biomass gas. One key benefit of the gas turbine engine over many other types of heat engines is that it delivers a significant amount of power in relation to its size and weight. Increasing the quantity of the working fluid at a higher volume and constant pressure, and vice versa, may result in a larger amount of power being generated by the turbine. Ahmarinezhad *et al.* (2013).

Global demand for electricity is growing every year. Gas turbines are indicated to be one of the cleanest options running on fossil fuels to meet this growing energy demand. Gas turbines find its massive utilization in power generation, aviation industry and maritime industry where they are used as prime movers due to its peculiar advantages such as high power/weight ratio, large operational adaptability, ease of maintenance, long operating hours and high dependability and reliability (Mouloud *et al.*, 2016). Research has shown that gas turbines will dominate the Nigeria power production industry in the near future (Saturday and Ebieto 2020).

Shi & Che (2007) proposed a liquefied natural gas fuelled combined cycle power plant with a waste heat recovery. Parametric analyses are performed for the proposed combined cycle to evaluate the effects of several factors, such as the gas TIT, the condenser pressure, fuel gas heating temperature on the performance of the proposed cycle. When the condenser pressure is decreased, the net electrical efficiency of proposed cycle increased. The net electrical efficiency increases as the condenser pressure decreases for a given TIT. Heating fuel results in higher gas turbine efficiency due to the reduced fuel flow. The power output and thermal efficiency of system chosen as the performance criteria, the effects of compressor PR, TIT and isentropic efficiencies on the turbine performance analyzed in the model.

Ibrahim *et al.* (2016) discussed new strategies designed for GT plants to enhance their overall performance based on operational modeling and optimization. They analyzed the effect of various cycle configurations (intercooler, two-shaft, reheat, and regenerative) on the performance of GT plants based on real power plant data. A thermodynamic analysis was carried out to evaluate the enhancements in cycle configurations.

Ibrahim *et al.* (2017) presented a comprehensive review of the modelling analysis for enhancing the performance and optimization conditions of Combined Cycle Gas Turbine (CCGT) power plants. The study aimed to identify the characteristics of CCGT systems and locate the optimum operation conditions via simulation models. The developed simulation models encapsulate thermal analysis according to thermodynamics principles, optimization techniques, error analysis, and performance metrics assessment of various CCGT system configurations using MATLAB 10A software. They used ANOVA as a statistical tool to assess the results and develop correlations between the power outputs and performance metrics of CCGT plants. The developed correlations are of acceptable accuracy for all the considered range of simulation data with a coefficient of determination (R^2) of 0.985.

Rahman *et al.*, (2010) The results of the investigations revealed that all of the modified GT plants had

improved performance, with a cycle efficiency improvement ranging from 9 percent to 26 percent when compared to their original values.

Omar *et al.* (2017) went on to propose a parametric analysis to determine how the temperature of the turbine intake, the ambient temperature, and relative humidity affect the systems' performance. According to the findings, raising the turbine intake temperature by around 1.5 degrees Celsius increases the ideal pressure ratio by approximately 1.5 degrees Celsius. Several factors (including turbine intake temperatures, combustion efficiency, pressure ratio, and isentropic turbine and compressor efficiencies) have been studied in detail to determine their influence on GTPPS performance. The values of the performance parameters were determined using the fundamental cycle equations and under the assumption that the thermodynamic characteristics remained constant during the cycle.

Shi & Che (2007) proposed that a combined-cycle gas turbine (CCGT) plant be built that uses Liquefied Natural Gas as a fuel. Aiming to determine the impact of the turbine intake temperature, pinch point, fuel heating temperature, and condenser pressure on the performance of the systems, the research evaluated the effects of the operating parameters on them. Increasing the total thermal efficiency of the system will result from lowering the condenser pressure at any given turbine intake temperature. The increased thermal efficiency of GTPPs is a result of the lower fuel consumption related with the raising of the temperature of the fuel gas. At higher turbine intake temperatures, however, the occurrence of a higher exhaust temperature of an HRSG system is more frequent.

Gülen (2017) investigated the performance of a GT system, with particular attention on the modeling and analysis of how the pressure ratio, isentropic compressor and turbine efficiency, and turbine intake temperature affect the performance of GTPPs. The thermal efficiency and power output of the system were used as performance measures to evaluate the system's overall performance. The maximum cycle temperature was chosen based on the strength of the material used in the GTPP under full load, as well as on ISO standards for maximum cycle temperature. Kurt and colleagues have suggested a thermodynamic model for evaluating the performance of an ideal open GT cycle.

II. METHODS

2.1 Gas turbine model creation

A gas turbine model was created in this work using in-house software. It is a computer model that was mimics the operation of the real engine in the field. The interface of the software used for creating the gas turbine model is shown in Figure 1. From the interface it is clear the input data required in creating and engine model are the air flow rate, pressure ratio, ambient temperature, fuel flow rate, heating value of the fuel, turbine exit temperature and the power output. These are obtained from the field. Other input data are which are inputted in a range are the isentropic efficiency of the compressor, isentropic efficiency of the turbine, combustion efficiency the combustion pressure loss, the exhaust pressure loss, the net power output of the turbine and the exhaust gas temperature. The software uses the turbine exit temperature and the power output in fixing the values of the last first five parameters provided in ranges.

frmRealCycleSimulations

Performance Estimation Exit

SELECT SIMULATION TYPE

☐ Simulate with TET

☒ Simulate with Fuel Flow

Provide General Input Data

Air Flow Rate , ma (kg/s)

Pressure Ratio (PR)

Ambient Temperature , T1 (K)

Fuel Flow Rate , mf (kg/s)

Heating Value of Fuel (MJ/kg)

Input Data For Results Comparison

Turbine Exit Temp._Field Value, T4 (K)

Net Work_Field Value (kW)

☐ Write Data To Excel for Given TET

☒ Write Data To Excel for Given Fuel Flow

☒ Pressure Losses Included

 Provide Specific Input Data

	1st Val	2nd Val
Compressor Isentropic Efficiency (%)		
Turbine Isentropic Efficiency (%)		
Combustion Efficiency (%)		
Combustion Pressure Loss (%)		
Exhaust Pressure Loss (%)		

Figure 1: Software interface for creating engine model

2.2 The Combined Gas and Steam Turbine Cycle Engine

The thermodynamic cycle of the basic combined cycle is made up of two power plant cycles that are interconnected. These are the Joule, which is a gas turbine cycle, and the Rankine cycle, which is a steam turbine cycle. The gas turbine cycle is known as the topping cycle while the steam turbine cycle is the bottoming cycle. Figure 2 shows the schematic diagram of the combined cycle plant while Figure 3 shows the T-s diagram of the plant.

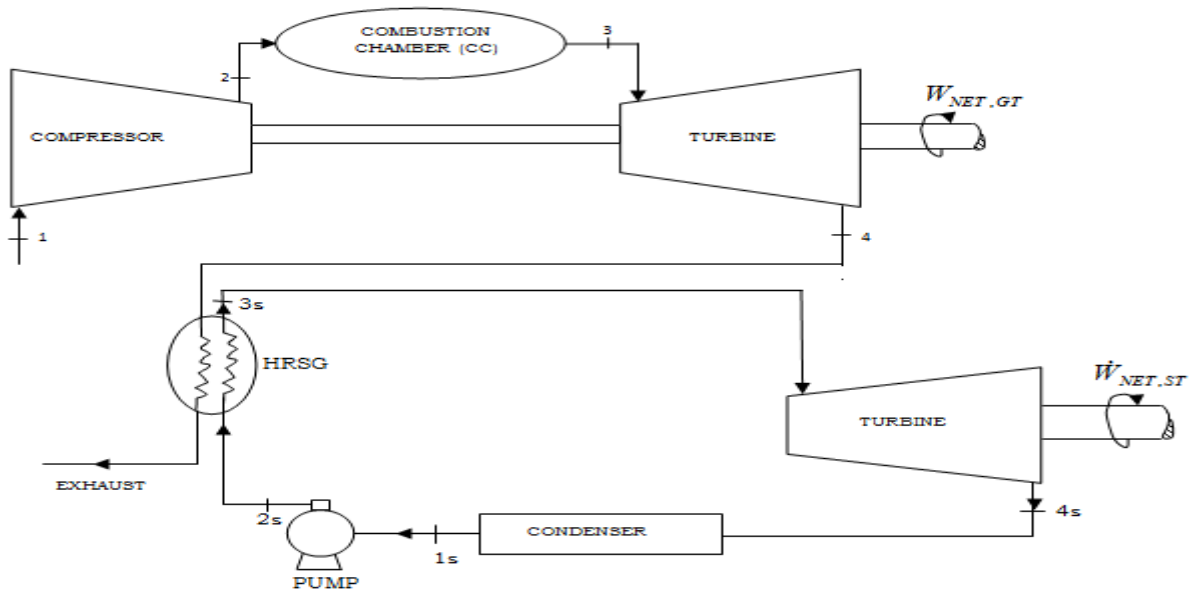


Figure 2: Schematic diagram of the combined cycle plant

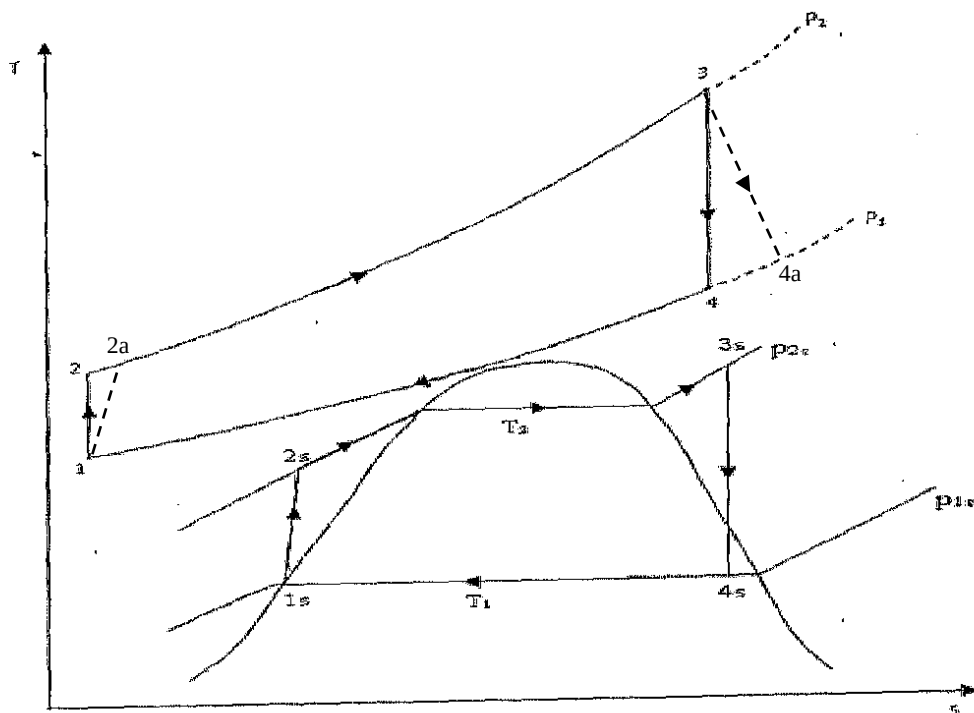


Figure 3: T-s diagram of combined gas and steam turbine power plants

2.3 Performance Simulation of the Steam Turbine Plant

A steam turbine power plant comprises four basic components- pump, boiler, turbine and condenser. The boiler in the combined cycle arrangement is replaced by the HRSG. The working fluid is usually superheated in practical plants. Steam turbines operate on the Rankine cycle. Superheat Rankine cycle is assumed in this work. The temperature at which the working fluid is to be superheated is limited by the exhaust gas temperature of the gas turbine. Figure 4 shows the arrangement of the basic components of the steam turbine part of the combined cycle plant. The T-s diagram is shown in Figure 5. The heat source is from the exhaust of the gas turbine which is harnessed for usage via the HRSG. The effectiveness of the HRSG was assumed in this work. The assumed value here should be conservative so as that obtain lower heat input into the steam turbine plant hence obtain a low power output from the plant. This is to ensure that the power output obtained is not higher than what is practicable feasible.

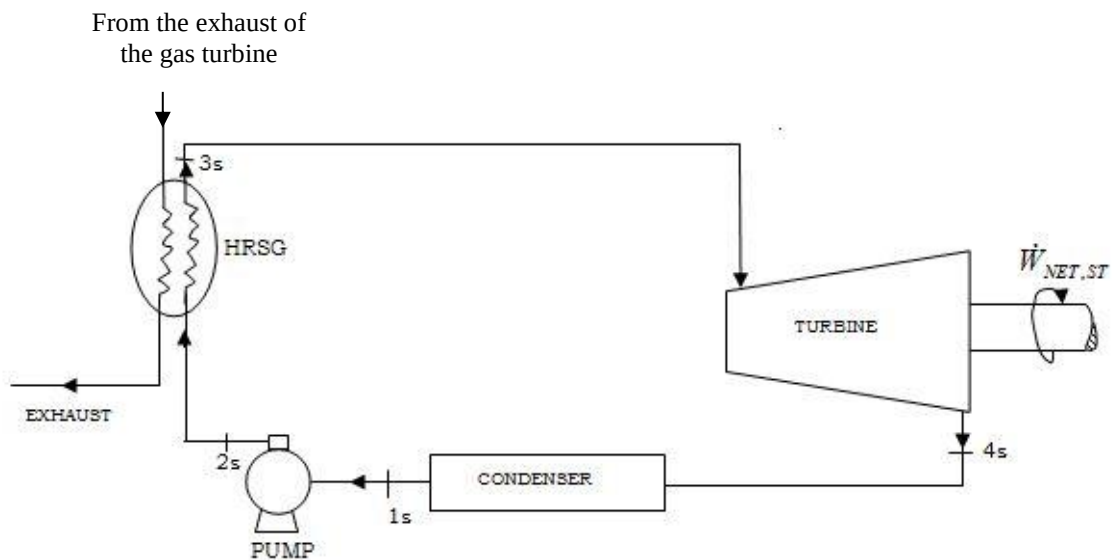


Figure 4: Block diagram of the steam turbine part of the combined cycle plant

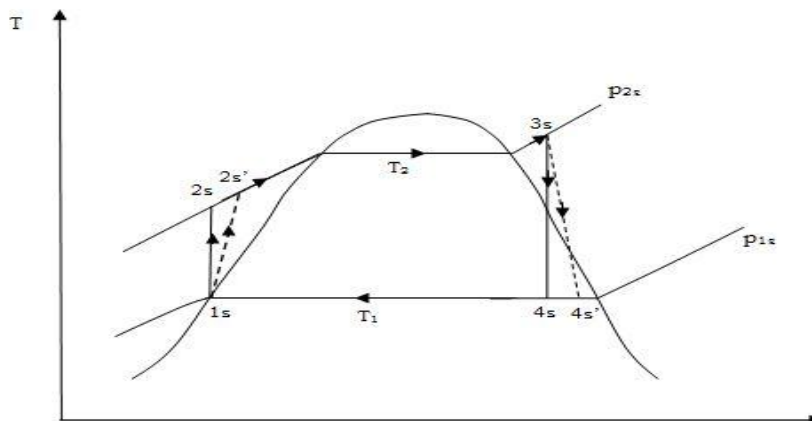


Figure 5: T-s diagram of the steam turbine cycle

Aside the effectiveness of the HRSG, the following parameters have to be assumed for the analysis:

- i. The condenser pressure level,
- ii. The boiler pressure level,
- iii. The superheat temperature value,
- iv. The isentropic efficiency of the pump, and
- v. The isentropic efficiency of the turbine.

With these assumptions, the mass flow rate of the steam will be estimated and hence the power output from the plant and other parameters of the plant can be estimated. The superheat temperature T_{3s} should be selected such that the value is much less than the exhaust gas temperature of the gas turbine.

2.4 The Power Output and Thermal Efficiency of Steam Turbine Plant

The thermal efficiency of the steam turbine plant is given by the Equation below;

$$\eta_{ST} = \frac{W_{NET,ST}}{Q_{IN,ST}} \quad (1)$$

But, $Q_{IN,ST} = \varepsilon Q_{OUT,GT}$. The exhaust gas from the gas turbine cycle enters the HRSG at temperature T_4 . Assuming at the exit of the HRSG the gases are reduced to temperature T_5 , where $T_5 < T_4$, the heat extracted can be calculated as,

$$Q_{EX} = mc_p(T_4 - T_5) \quad (2)$$

Where; m is the mass flow rate of the exhaust gases from the gas turbine cycle and c_p is the specific heat capacity of the exhaust gases. The value of T_5 is taken as 100°C in this work. The heat input into the steam turbine cycle will be,

$$Q_{IN,ST} = \varepsilon Q_{EX} \quad (3)$$

The net work out of the cycle is the difference between the work output of the turbine and the work consumed by the pump:

$$W_{NET,ST} = W_{ST} - W_p \quad (4)$$

where W_{ST} is the work output of the steam turbine while W_p is the work consumed by the pump. The work output of the turbine is,

$$W_{ST} = \dot{m}_s(h_{3s} - h_{4s'}) \quad (5)$$

where $\dot{m}_s$ is the mass flow rate of steam while h_{3s} and $h_{4s'}$ indicates the enthalpy of steam at stage 3s and 4s' respectively. The enthalpy value h_{3s} is read from steam tables based on the pressure value and the superheat temperature assumed. The actual enthalpy at the turbine exit is calculated using the assumed isentropic efficiency of the turbine $\eta_{i,st}$ and the ideal enthalpy at the turbine exit h_{4s} . It is given by Equation (6),

$$h_{4s'} = h_{3s} - \eta_{i,st}(h_{3s} - h_{4s}) \quad (6)$$

Alternatively the turbine work output is given by Equation (7),

$$W_{ST} = \dot{m}_s \times \eta_{i,ST} (h_{3s} - h_{4s}) \quad (7)$$

The ideal enthalpy at the exit of the turbine is given by Equation (8),

$$h_{4s} = h_{f1} + x_{4s} h_{fg@p1} \quad (8)$$

where h_{f1} is the enthalpy of the saturated liquid at state point 1s, is the latent heat of vapourization at the condenser pressure level while x_{4s} is the dryness fraction at state point 4s. The first two parameters are read from the steam tables. The last parameter is estimated from the entropy values and it is given by Equation (9)

$$x_{4s} = \frac{s_{4s} - s_{f1}}{s_{fg@p1}} \quad (9)$$

Where; s_{4s} is the entropy at state point 4s ($s_{4s} = s_{3s}$), s_{f1} is the entropy at state point 1s and $s_{fg@p1}$ is the difference in entropy between the saturated vapour and the saturated liquid at the condenser pressure level. All these entropy values are read from the steam tables.

The work consumed by the pump is given by Equation (10)

$$W_p = \frac{\dot{m}_w \times v_{f1} (p_{2s} - p_{1s})}{\eta_{i,p}} \quad (10)$$

where $\dot{m}_w$, v_{f1} , $\eta_{i,p}$, p_{2s} and p_{1s} are the mass flow rate of the feed water (to be calculated), the specific volume of water at state point 1, the isentropic efficiency of the pump, the boiler pressure and the condenser pressure level respectively.

2.5 Determination of the Flow Rate of the Feed Water

The flow rate of the feed water $\dot{m}_w$ which is taken to be the same as the flow rate of the steam is obtained from Equation (11),

$$\dot{m}_w = \frac{Q_{IN,ST}}{h_{3s} - h_{2s'}} \quad (11)$$

where $h_{2s'}$ is the actual enthalpy of the working fluid at the exit of the pump. It can be obtained by considering the specific work consumed by the pump (w_p).

$$w_p = \frac{W_p}{\dot{m}_w} = \frac{v_{f1} (p_{2s} - p_{1s})}{\eta_{i,p}} = h_{2s'} - h_{f1} \quad h_{2s'} = w_p + h_{f1} = \frac{v_{f1} (p_{2s} - p_{1s})}{\eta_{i,p}} + h_{f1} \quad (12)$$

2.6 Economic Analysis of Real Cycle Gas Turbine Plant and the Combine Cycle Plant

It costs money to produce energy using turbines; the cost of production is a crucial factor in determining whether or not the energy cycle technology is economically feasible. In order to compare the economic

viability of the simple open cycle gas turbine plant and the proposed combined cycle plant, three economic parameters were employed. These are:

- i. Payback period (PBP),
- ii. Net present value (NPV) and
- iii. Levelized cost of electricity (LCOE).

i. Payback Period

The payback period represents the length of time in years taken for the net annual income cumulated over time equals the initial investment value. It is given simply by Equation (13),

$$PBP = \frac{IC}{NACF} \quad (13)$$

Where; IC is initial cost or cost of installation of the power plant and $NACF$ is the net annual cash flow. The $NACF$ is the difference between the annual revenue AR and the annual cost AC of operating the power station, given by Equation (14),

$$NACF = AR - AC \quad (14)$$

Both the annual revenue and the annual cost of operation are projected values. The annual cost of operating a gas turbine plant consists of three basic parts. These are the operation of maintenance cost $C_{O\&M}$, personnel cost C_p and cost of fuel C_f . For the combined cycle plant, the installation cost of the gas turbine part and the steam turbine part (expressed in monetary value per unit power output) will be taken separately since the idea is to incorporate a steam turbine component into the existing gas turbine plant. The revenue comes from the sale of electricity. The price of electricity varies with the users. An average value was used in this work.

ii. Net Present Value

The NPV accounts for the time value of money. It represents the net total cash flow over the lifetime of a given project brought to the present time. It is given by Equation (15)

$$NPV = \sum_{i=1}^n \frac{AR_i}{(1+r)^i} - \sum_{i=1}^n \frac{AC_i}{(1+r)^i} - IC \quad (15)$$

where r is the interest rate, n is the length of time (in years) of the project, AR_i and AC_i are the annual revenue and the annual cost respectively in year i . Project lifetime of 20 years was used in this work.

iii. Levelized Cost of Electricity

The LCOE represents the total cost of producing a unit amount of electricity over the lifetime of the project (brought to the present time). It is given by Equation (3.16),

$$LCOE = \frac{IC + \frac{AC_i}{(1+r)^i}}{E} \quad (16)$$

Where, E is the total amount of energy produced over the lifetime of the power plant. The yearly annual cost comprises the operation and maintenance cost personnel cost and cost of fuel.

The three economic parameters were applied to the gas turbine plant used as a case study in this work and the simulated combined cycle power plant.

III. RESULTS

3.1 Results

Before presenting the various results obtained in this work information about the turbine used as a case study and the relevant input data used are first presented. Table 1 shows some data about the design point and field operation of the turbine used as a case study. It also contains other data for the performance analysis of the gas turbine plant. Table 2 shows some input data for the steam turbine performance simulation while table 3 shows some data used for the economic analysis. Table 4 shows the results of the performance analysis from the gas turbine plant.

The results of the performance analysis of the gas turbine are shown in Table 5 while the simulated results of the steam turbine power plant are shown in Table 6. In Figure 6, the variations of both the feed water flow rate and the net power output of the steam turbine unit with the effectiveness of the HRSG are presented. The results of the combined cycle power plant are shown in Table 7. The effect of the effectiveness of the HRSG on the efficiency of the combined cycle plant is shown in Figure 7. A comparison of the efficiency, specific consumption and the heat rate of the gas turbine plant, the simulated steam turbine plant and the simulated combined cycle plant are presented in Figure 8, Figure 9 and Figure 10 respectively. A comparison of their respective net power outputs is provided in Figure 11.

Table 1: Input parameters for the gas turbine plant used as case study

Parameters	Symbols	Units	Design data	Average field data
Mass flow rate of fuel	$\dot{m}_f$	kg/s	3.65	3.54
Mass flow of air	$\dot{m}_a$	kg/s	247.0	247.0
Lower heating values of fuel	LHV	MJ/kg	48.95	48.95
Ambient air temperature	T_0	K	288	301
Ambient air pressure	P_0	kPa	101.32	101.32
Pressure ratio	r_c	—	9.67	9.67
Power Output	W_{net}	kW	65000	57450

Exhaust Gas Temperature	T_4	K	758	762
Specific heat capacity of air	c_{pa}	kJ/kgK		1.005
Ratio of specific heat capacities of air	γ_a	-		1.4
Specific heat capacity of flue gases	C_{pg}	-		1.2
Ratio of specific heat capacities of flue gases	γ_g	-		1.33

Table 2: Input data for steam turbine performance analysis

Parameter	Symbols	Units	Value
Condenser pressure	p_{1s}	bar	1.2
Boiler pressure	p_{2s}	bar	120
Superheat temperature	T_{3s}	$^{\circ}C$	500
The effectiveness of the HRSG	ε	%	90
Isentropic efficiency of the pump	$\eta_{i,p}$	%	85
Isentropic efficiency of the turbine	$\eta_{i,t}$	%	90

Table 3: Input data for power pants economic analysis

Parameter	Symbols	Units	Value
Project lifetime	n	yr	20
Discount rate	r	%	10
Cost of natural gas	C_F	\$/1000ft ³	2.6
Price of electricity	C_e	N/kWh	75
Operation and maintenance cost of gas turbine plant (personnel cost inclusive)	$C_{O\&M_GT}$	\$/kWh	0.048

Operation and maintenance cost of combined cycle plant (personnel cost inclusive)	$C_{O\&M_CC}$	\$/kWh	0.052
Installation cost of gas turbine plant	IC_{GT}	\$/kW	750
Installation cost of steam turbine plant	IC_{ST}	\$/kW	850

Table 4: Results of the performance analysis of the gas turbine plant

Parameter	Symbols	Units	Value
Power Output	W_{net}	kW	57452
Exhaust Gas Temperature	T_4	K	759
Thermal efficiency	η_{GT}	%	33.66
Specific fuel consumption	sfc	kgs ⁻¹ /(MW)	0.06162
Heat rate	HR	kW/kWh	10,695

Table 5: Simulated Results of the steam turbine plant

Parameter	Symbols	Units	Value
Feed water flow rate	$\dot{m}_w$	kg/s	39.63
Power consumed by the pump	W_p	kW	579.34
Turbine power output	W_{ST}	kW	28817.69
Net power output of the plant	$W_{NET,ST}$	kW	28238.35
Heat input into the plant	$Q_{IN,ST}$	kW	102969.36
Thermal efficiency of the plant	η_{ST}	%	27.42
Specific fuel consumption	sfc	kgs ⁻¹ /(MW)	0.1254
Heat Rate	HR	kJ/kWh	13127

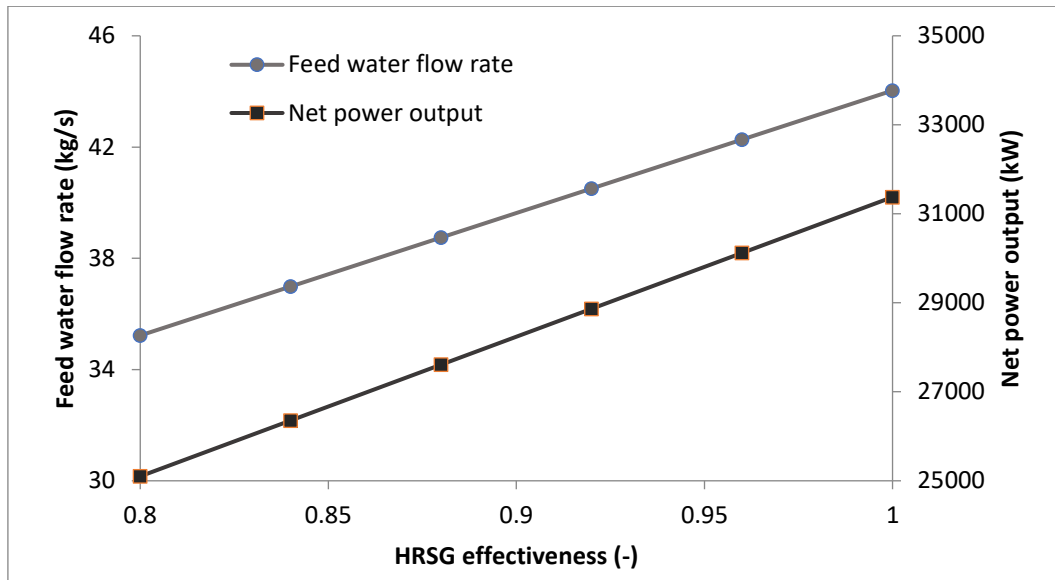


Figure 6: Variation of feed water flow rate and net power output with the effectiveness of the HRSG

Table 6: Simulated Results of the combined cycle turbine plant

Parameter	Symbols	Units	Value
Temperature of exhaust gases at stack exit	T_5	K	373
Heat input into the cycle	$Q_{IN,CC}$	kW	170,683
Net power output	$W_{NET,CC}$	kW	85,688.35
Thermal efficiency	$\eta_{th,CC}$	%	50.20
Specific fuel consumption	sfc	kgs ⁻¹ /(MW)	0.04131
Heat Rate	HR	kJ/kWh	7,170.87

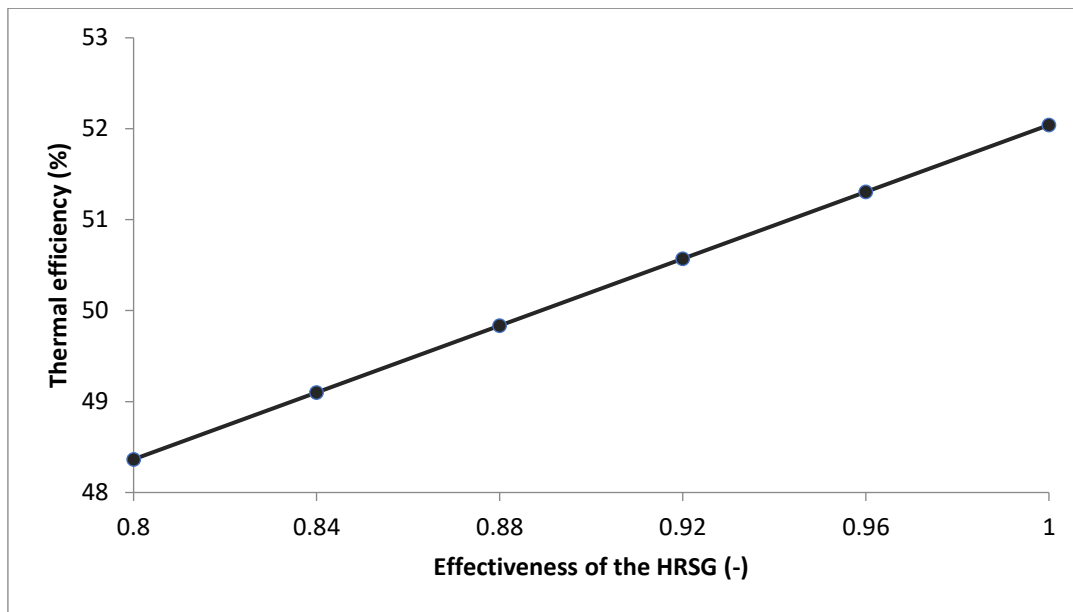


Figure 7: Effect of the effectiveness of the HRSG on the combined cycle thermal efficiency

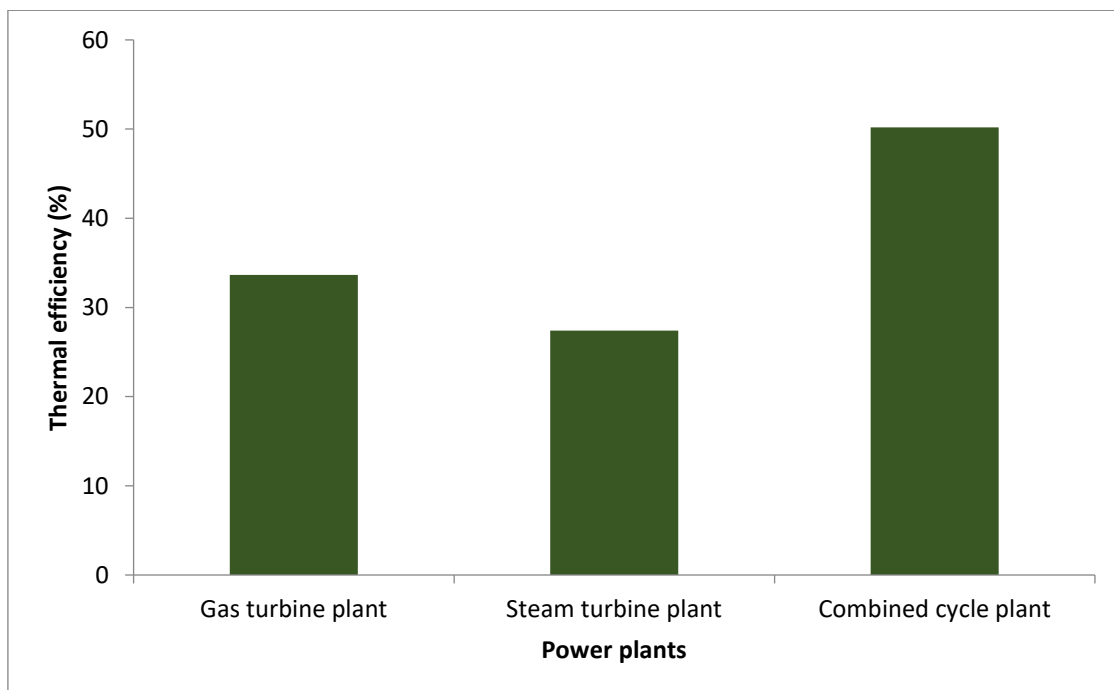


Figure 8: Comparison of the thermal efficiencies of the plants

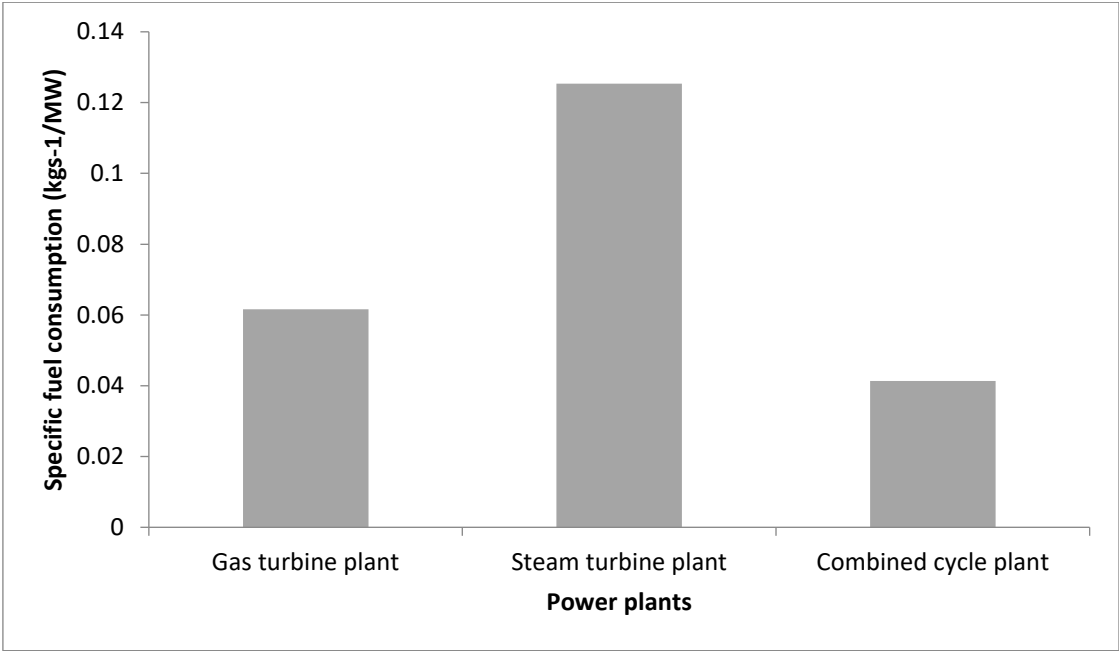


Figure 9: Comparison of the specific fuel consumptions of the plants

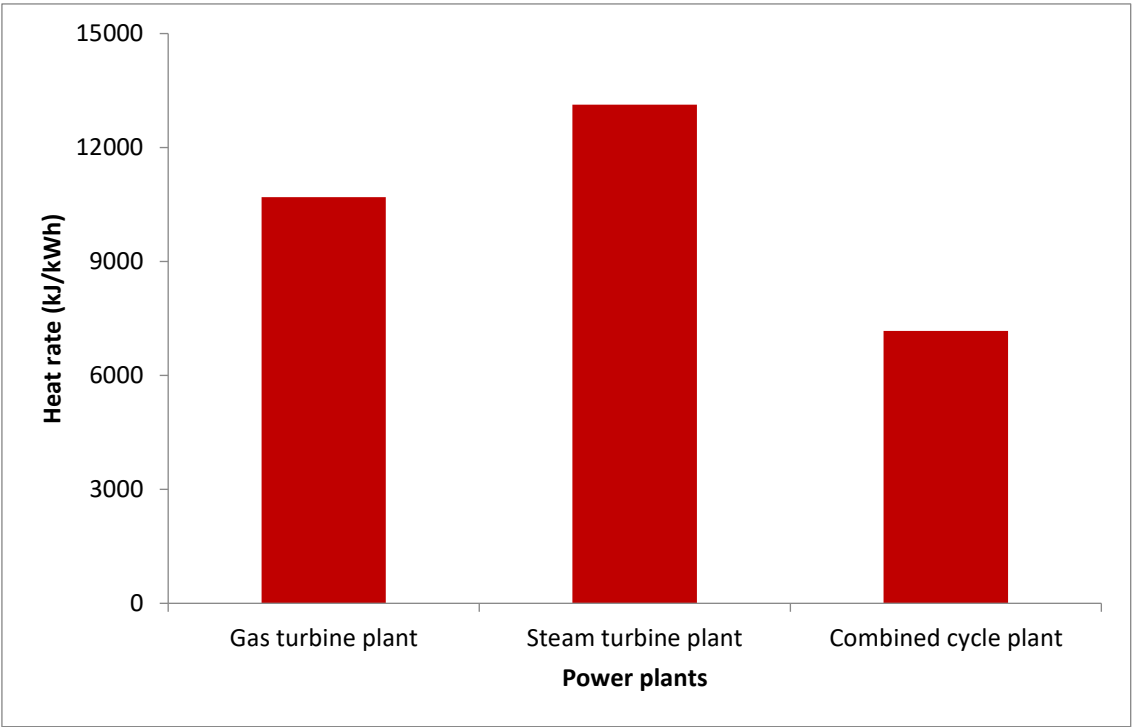


Figure 10: Comparison of the heat rates of the plants

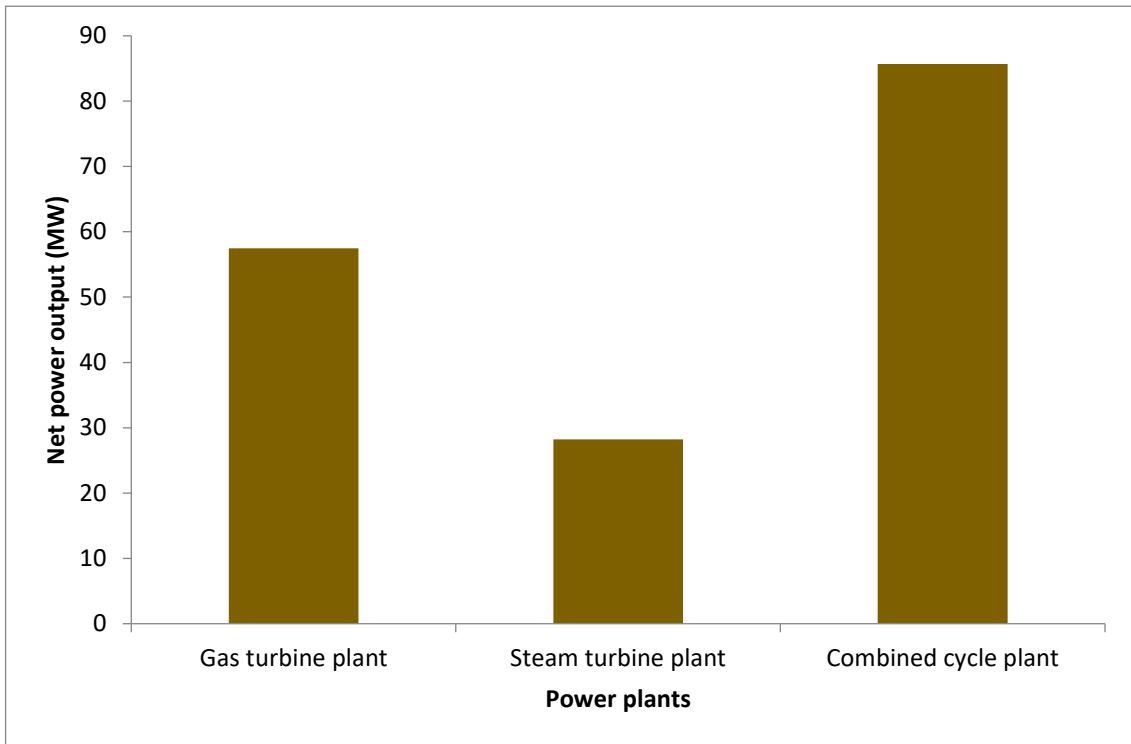


Figure 11: Comparison of the net power outputs of the plants

The performance analysis of the gas turbine power plant shows that the power output from the plant and the exhaust gas temperature obtained are very close to those from the field (see Table 1 and 4). The simulated turbine exit temperature was used as the temperature of the flue gases powering the heat recovery steam generator to provide steam for the steam turbine plant. The simulated parameters of the steam turbine plant shown in Table 5 depend largely on the effectiveness of the heat recovery steam generator. Figure 6 shows that both the feed water flow rate and the net power output from the steam plant increases with increase in the effectiveness of the HRSG.

For the combined cycle plant, the simulated results in Table 6 shows that the power output of the combined cycle plant is the sum of those of the gas turbine plant and the steam turbine plant. Since the power output of the steam turbine plant increases with the effectiveness of the HRSG, the power output of the combined cycle plant also increases with increase in the effectiveness of the HRSG. Thus the thermal efficiency of the combined cycle plant increases when the effectiveness of the HRSG increases. This is shown in Figure 7.

Basic performance parameters of thermal power plants worth comparing are the thermal efficiency, specific fuel combustion and heat rate. These comparisons made in Figures 8 to 9 indicate that the thermal efficiency of the steam turbine plant is the lowest. The thermal efficiency of the combined cycle plant is much higher than that of the gas turbine power plant. This is because additional power of over 28MW was produced from the steam turbine plant without burning any additional fuel aside that used in the gas turbine plant. Since the steam turbine cycle does. Although the steam turbine does not consume any fuel directly, it enjoys the fuel input from the gas turbine plant to produce power. The fuel utilized in the gas turbine cycle was used in calculating the SFC in the steam turbine plant. The steam turbine plant has the highest SFC, followed by the gas turbine plant as shown in Figure 9. The heat rate values shown in Figure 10 indicates that the combined cycle plants consumes far less heat compared to both the steam turbine plant and the gas turbine plant to produce it amount of electrical energy. This also translates to lower thermal destruction of the environment

resulting from the operation of power plants.

The power outputs from the power plants are shown in Figure 11. The comparison is made here to see if the contributions from the gas turbine and the steam turbine parts forming the combined cycle plant follow the proposal by Ragland & Stenzel (2000). The power output from the gas turbine plant is 57.45 MW and that from the steam turbine plant is 28.24 MW. The contribution from the steam turbine plant to the combined cycle plant in this work is 67.05 % which is very close to what Ragland & Stenzel (2000) proposed. This 66 % contribution of the gas turbine proposal was applied to the NIPP power plants. Table 7 shows the installed power capacities of the gas turbines in the NIPP plants and the additional power obtained if converted to combined cycle power plants. Of the ten NIPP gas turbine-based power stations, two are combined cycle power plants (CCPP), seven operate simple cycle gas turbines (SCGTs) while one (Geregu II power plant) currently operate SCGTs with intention to be converted to CCPP.

The results of economic analysis of the gas turbine plant and the combined cycle plant are shown in Tables 8 and 9 respectively.

Table 7: The NIPP plants and additional power obtainable when converted to CCPP

Power plant	State located	Cycle type	Capacity, MW	Additional power via ST, MW
Gbarain Power Plant	Bayelsa	SCGTs,	225	116
Odukpani Power Plant	Cross River	SCGTs	562	290
Sapele II Power Plant	Delta	SCGTs	451	232
Ihovbor PP	Edo	SCGTs	451	232
Omoku II Power Plant	Rivers	SCGTs	225	116
Alaoji Power Plant	Abia	CCPP	1076	
Egbema II Power Plant	Imo	SCGTs	338	174
Olorunsogo II PS	Ogun	CCPP	676	
Omotosho II PS	Ondo	SCGTs	451	232
Geregu II PP	Kogi	SCGTs	434	224
Total additional power from steam turbines				1616

Table 8: Cash flow and economic parameters of the gas turbine power plant

Parameters	Year																				Total	NPV
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20		
Installed Cost (Million \$)	-43.09	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-43.09	
NACF (Million \$)	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	161.99	
Discount factor at 10 %	0.909	0.826	0.751	0.683	0.620	0.565	0.513	0.466	0.424	0.386	0.351	0.319	0.290	0.263	0.239	0.218	0.198	0.180	0.1635	0.1486		
Discounted NACF (M\$)	7.36	6.69	6.09	5.53	5.03	4.57	4.16	3.78	3.43	3.12	2.84	2.58	2.35	2.13	1.94	1.76	1.60	1.46	1.32	1.20	68.96	25.87
Cumulative NACF	7.36	14.06	20.14	25.67	30.70	35.28	39.43	43.21	46.64	49.77	52.61	55.19	57.53	59.67	61.61	63.37	64.97	66.43	67.75	68.96		
Annual Cost (AC) M \$	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97	36.97		
Discounted AC	33.61	30.55	27.78	25.25	22.96	20.87	18.97	17.25	15.68	14.25	12.96	11.78	10.71	9.74	8.85	8.05	7.31	6.65	6.04	5.50	314.75	
Energy produced per year (GW-hr)	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6	402.6		
Energy prod. Discounted (GW-hr)	366.0	332.8	302.5	275.0	250.0	227.3	206.6	187.8	170.8	155.2	141.1	128.3	116.6	106.0	96.6	87.6	79.66	72.42	65.83	59.84	34.28	
Total energy Produced (GW-hr)	3427.76																					
Total life cycle cost (Million \$)	357.84																					
LCOE (\$/kW-hr)	0.1044																					
LCOE (N/kW-hr)	69.94																					

Table 9: Cash flow and economic parameters of the combined cycle plant

Parameters	Year																				Total	NPV
	1	2	3	4	5	6	7	8	9	10	11	12	13		15	16	17	18	19	20		
Installed Cost (Million \$)	-67.09	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		0.00	0.00	0.00	0.00	0.00	0.00	-67.09	
NACF (Million \$)	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37		15.37	15.37	15.37	15.37	15.37	15.37	307.5	
Discount factor at 10 %	0.909	0.826	0.751	0.683	0.621	0.564	0.513	0.467	0.424	0.386	0.350	0.319	0.290		0.239	0.218	0.198	0.180	0.164	0.149		
Discounted NACF (M\$)	13.98	12.71	11.55	10.50	9.55	8.68	7.89	7.17	6.52	5.93	5.39	4.90	4.45		3.68	3.35	3.04	2.77	2.51	2.29	130.9	63.8
Cumulative NACF	14.0	26.7	38.2	48.7	58.3	67.0	74.9	82.0	88.5	94.5	99.9	104.8	109.2		116.9	120.3	123.3	126.1	128.6	130.9		
Annual Cost (AC) M \$	51.85	51.85	51.85	51.85	51.85	51.85	51.85	51.85	51.85	51.85	51.85	51.85	51.85		51.85	51.85	51.85	51.85	51.85	51.85		
Discounted AC	47.13	42.85	38.95	35.41	32.19	29.27	26.61	24.19	21.99	19.99	18.17	16.52	15.02		12.41	11.28	10.26	9.32	8.48	7.71	441.4	
Energy produced per year (GW-hr)	600.5	600.5	600.5	600.5	600.5	600.5	600.5	600.5	600.5	600.5	600.5	600.5	600.5		600.5	600.5	600.5	600.5	600.5	600.5		
Energy prod. Discounted (GW-hr)	545.9	496.3	451.2	410.2	372.9	339.0	308.2	280.1	254.7	231.5	210.5	191.3	173.9		143.8	130.7	118.8	108.0	98.2	89.3	5112.4	
Total energy Produced (GW-hr)	5112.4																					
Total life cycle cost (Million \$)	508.48																					
LCOE (\$/kW-hr)	0.0995																					
LCOE (N/kW-hr)	66.64																					

At 10% discount rate, 20 years project lifetime and electricity price at N 75 /kWh the net present value of the gas turbine plant is 25.87 million US dollars while that of the combined cycle plant is 63.80 million US dollars. The combined cycle plant has a shorter payback period (4.36yrs) as against that of the gas turbine plant (5.32yrs). The combined cycle power plant is thus economically more viable. The levelized cost of electricity for the combined cycle plant is lower than that of the gas turbine plant (0.1044 \$/kwh for the gas turbine plant and 0.0995 \$/kWh for the combined cycle plant). These values, in terms of Nigeria with exchange rate of N 670 to a dollar, are N 69.94 per kWh and N 66.64 per kWh respectively.

IV. DISCUSSION

The possibility of converting simple cycle gas turbine plants to combined cycle power plants where the exhaust of the gas turbine is used to fire a heat recovery steam generator to run a steam turbine plant was investigated in this work. The engineering as well as the economic implications of obtaining combined cycle plants from simple gas turbine plants was thoroughly investigated. A gas turbine power plant operating in Rivers state was used as a case study. A model of the gas turbine was created using an in-house software. The performance of the gas turbine model created closely matched the results obtained from the field. The exhaust gas temperature of the gas turbine model was used as source of heat in the HRSG. Selecting practicable feasible condenser pressure condenser pressure and superheat temperature for the steam turbine power plant operating on the superheat Rankine cycle, a combined cycle power plant.

The thermal efficiency of the combined cycle plant increases with the effectiveness of the heat recovery steam generator. The thermal efficiency of the combined cycle plant is much higher than that of the gas turbine plant. The steam turbine plant has the lowest thermal efficiency and has the highest specific fuel consumption. The steam turbine plant also has the highest heat rate (consumes more heat than the gas turbine plant to produce unit amount of electricity). The combined cycle plant has the lowest specific fuel consumption as well as the lowest heat rate.

The total power obtainable from the combined cycle plant was compared with that of previous work where it was stated that the gas turbine contributes 66% to the total power output of the combined cycle. It was observed that for the case study, the gas turbine plant contributes 67.05% to the total power output of the combined cycle plant. This translates to about 49% increase in power from the gas turbine. This value is very close that that proposed the previous study. The additional power output when the open cycle NIPP plants are converted to combined cycle power plants was investigated noting that the gas turbine contributes some 66% to the total power output of the combined cycle plant. It was observed that additional 1.616GW of electricity could be obtained from the NIPP power plants.

The economics of operating the gas turbine power plant and the combined cycle power plant for project lifetime of 20 years was investigated. The net present value and the levelized cost of electricity were investigated in the economic analysis. It was observed that the NPV of operating the gas turbine plant for 20 years at discount rate of 10% is 25.87 million US dollars. That of the combined cycle plant is 63.80 million US dollars. The levelized cost of electricity obtained for the simple cycle plat and the combined cycle plant are 0.1044 \$/kwh and 0.0995 \$/kWh respectively. The combined cycle power plant also has shorter payback period compared to the gas turbine power plant. It was thus observed in this study that it is possible to obtain additional power from the existing open cycle gas turbines by incorporating a steam turbine to the existing plant and by selecting appropriate operating data of the steam turbine, additional power output could be around 34% of the total power output of the combined cycle plant.

V. CONCLUSION

Looking at the objectives set out at the onset of this study and the various results obtained the following conclusions can be made:

- i. An engine performance model was developed in this study to aid the performance analysis of the simple cycle gas turbine power. Performance parameters such as thermal efficiency, heat rate and specific fuel consumption were estimated for the simple cycle plant,
- ii. Combined cycle power plant was derived from the simple cycle plant by incorporating a steam turbine plant. The performance parameters such as thermal efficiency, heat rate and sfc were calculated for the combined cycle plant;
- iii. Aside the engineering performance parameters economic performance parameters such as payback period, net present value and levelized cost of electricity were estimated for both the simple cycle plant and the combined cycle plant.
- iv. The combined cycle plant performs better than the simple cycle plant using any of the engineering as well as the economic performance indicators,
- v. In comparing the power outputs from the gas turbine and the combined cycle plants, the power output of the later is about 50% higher than that of the former.

CONTRIBUTION TO KNOWLEGDE

The study provided performance results of both simple cycle and combined cycle gas turbine power plants and thus created enough ground for the need to convert simple cycle gas turbine power plants to combined cycle gas turbine power plants.

Declaration of Conflicting Interests

The authors declare no potential conflicts of interest with respect to the research, authorship and publication of this article.

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