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Thermodynamic and Economic Evaluation of a Simple Cycle Gas Turbine Plant Conversion 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 ₦75 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 the 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 ₦69.94 per kWh respectively. The respective values of the combined cycle plant are 4.36yrs, $63.80 million and ₦66.64. 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.

I. INTRODUCTION

The gas turbine engine has wide applications including power generation, oil and gas extraction, process plants, aviation and other smaller linked industries, in addition to household and smaller associated businesses [1]. The efficiency of gas turbines has increased dramatically over the years. Improvements to the basic cycle, as well as the inclusion of steam turbine bottoming cycles, have the potential to increase the efficiency even more [2]. As of now, a combined gas turbine and steam turbine cycle may achieve efficiency levels approaching 60 percent. This timeline depicts the advancement of power producing technologies across time. Historically, advancements to the gas turbine cycle have been focused on boosting efficiency, cutting investment costs, and minimizing environmental pollutants, among other goals. Turbine designers have been working to raise the firing temperatures of their turbines without causing damage to them in order to boost efficiency. However, operating turbines at temperatures over the critical operating temperatures of its components endangers their structural integrity and dependability (Rahmanet al.,2011[3]). The development of better cooling systems and the improvement of materials are two important options for addressing this issue. Combined cycle configuration where the exhaust of the gas turbine is used to generate steam for a steam turbine plant is a very viable means of producing additional power and reducing the pollutants.

In Nigeria, gas turbine is widely used for power generation. Also, research has shown that gas turbines will dominate the Nigeria power production industry in the near future [4].The open cycle engine, although goes with minimal pollution issues, the cost of producing unit power is high. Also, there is need to reduce the emissions as much

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as possible. Majority of the gas turbine power generation plants operate on the simple open cycle basis. Many of those power plants, at the time of conception, were meant to be combined cycle power plants. There a lot of benefits with the combined cycle power plant. The overall electrical efficiency of a combined-cycle power system is typically in the range of 50–60% a substantial improvement over the efficiency of a simple, open-cycle application of around 33% [5]. Several studies have been carried out concerning the simple cycle as well as the combined cycle power plants [6-10].

Rai [11] discussed the fundamental idea behind the combined cycle while a complete thermodynamic modelling of a combined cycle power plant is covered in [12]. Combined cycle configuration of the Omoku power plant with a view to improving the efficiency is presented in [13]. A number of other studies on the gas turbine focused the gas turbine as a topping cycle plant and other performance improvement techniques while some dwells on review of different research activities involving gas turbines [14-17]. Studies aimed at converting existing simple cycle gas turbine power plants to combined cycle power plants where steam turbine power plant is the bottoming cycle plant and estimating the additional power derivable are not very common. That is the focus of this study. In this study, an existing simple cycle gas turbine power plant was analyzed to obtain the power obtainable. A combined cycle configuration was derived from the simple cycle plant and the additional power obtainable from the steam turbine plant was thus estimated. The economics of operating the simple cycle plant and the combined cycle plant were analyzed for a period of 20 years operation of both plants.

II. MATERIALS AND METHODS

2.1Performance Analysis of the Simple Cycle Plant

The simple cycle gas turbine engine is made up of three components: the compressor, combustion chamber turbine, leading to the rotation of a shaft a generator to generate electricity. The performance analysis of the gas turbine system requires its presentation in a temperature-entropy (T-s) diagram as show in Figure 1. The thermal efficiency of the plant needs to be estimated. Other performance parameters to be estimated are the specific fuel consumption (sfc) and the heat rate.

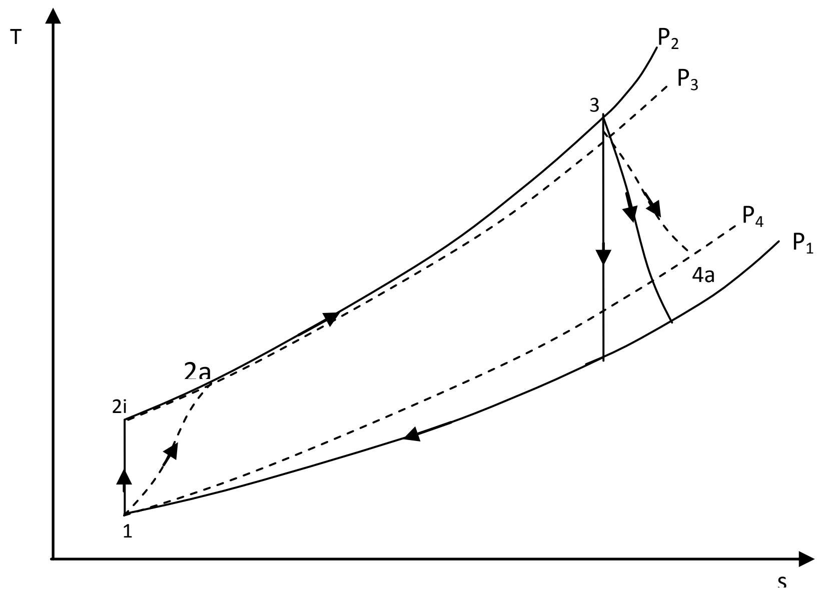

Figure 1:The Temperature –Entropy (T-S) of the Real Cycle GT Engine

The thermal efficiency of the plant is given by Equation (1),

$$\eta_{TH} = \frac{W_{NET}}{Q_{IN}} = \frac{W_{TA} - W_{CA}}{Q_{IN}} \quad (1)$$

Where W_{TA} , W_{CA} W_{NET} and Q_{IN} are the actual turbine work output, the actual compressor work input, the net work output of the plant and the heat input into the plant. The actual compressor work W_{CA} is given by Equation (2),

$$W_{CA} = \dot{m}_a (h_{2a} - h_1) \quad = \dot{m}_a c_{pa} (T_{2a} - T_1) \tag{2}$$

Where $\dot{m}_a$ is the air flow rate through the plant, c_{pa} is the specific heat capacity of air, T_{2a} is the temperature at the compressor exit and T_1 is the ambient temperature. The actual compression work relates with the ideal compression work W_{CI} via the isentropic efficiency of the compression process η_{CA} . This is given as,

$$\eta_{CA} = \frac{W_{CI}}{W_{CA}} = \frac{h_2 - h_1}{h_{2a} - h_1} = \frac{T_2 - T_1}{T_{2a} - T_1} \tag{3}$$

$$W_{CA} = \frac{W_{CI}}{\eta_{CA}} = \frac{\dot{m} c_p (T_2 - T_1)}{\eta_{CA}} \tag{4}$$

The isentropic efficiency of the compressor η_{CA} was obtained via engine model creation, presented hereafter. The actual turbine work (W_{TA}) is given as,

$$W_{TA} = \dot{m}_x c_{Pg} (T_3 - T_{4a}) \tag{5}$$

Where $\dot{m}_x$ is the mass flow rate of the working fluid after combustion, c_{Pg} is the specific heat capacity of the combustion products (flue gases), T_3 is the maximum cycle temperature while T_{4a} is the actual temperature at the turbine exit. The isentropic efficiency of the actual expansion process η_{TI} connects the actual turbine work output W_{TA} and the ideal turbine work output W_{T1}, ptresented as

$$\eta_{T1} = \frac{W_{TA}}{W_{T1}} = \frac{T_{3a} - T_{4a}}{T_3 - T_4} \tag{6}$$

The isentropic efficiency of the actual expansion process η_{TI} was obtaied in engine model creation process.

Heat Addition to the Cycle

Heat addition ($\dot{Q}_{IN}$) is as a result of the combustion of the fuel in the combustion chamber. It is given as,

$$\dot{Q}_{IN} = \dot{m}_f \ \text{LHV} \ \eta_{cc} = \dot{m}_x c_{pg} (T_{3a} - T_{2a}) \tag{7}$$

where $\dot{m}_f$ is the mass flow rate of fuel in kg/s, LHV is the lower heating value of the fuel (in J/kg), η_{cc} is the combustion efficiency, $\dot{m}_x$ is the mass flow rate of the working fluid after combustion and c_{pg} is the specific heat capacity of flue gases.Fuel is introduced into the combustion chamber and combustion takes place producing hugh temperature flue gases. The mass flow rate of the working fluid after combustion $\dot{m}_x$, is given by Equation (8),

$$\dot{m}_x = \dot{m}_a + \dot{m}_f \tag{8}$$

The specific fuel consumption (sfc)

The specific fuel consumption is the amount of fuel consumed to produce unit amount of power. It is given by Equation (9)

$$sfc = \frac{\dot{m}_f}{W_{NET}} \tag{9}$$

The Heat Rate (HR) of the Cycle (HR)

The heat rate is the amount of heat consumed to generate one kilo-Watt-hour (kWh) of electricity. It is expressed as,

$$\text{HR} = \frac{3600}{\eta_{TH}} \tag{10}$$

Engine Model Creation

A engine model was created using an in-house software via adaptation [18]. The model creation process leads to obtaiig five efficiency parameters- isentropic efficiecy of the compression process, isentropic efficiency of the expasion process, the combustion efficiency, the pressure losses in the combustion chamber and the pressure loss in the expansion process. The gas turbie enngie model is a computer model that 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 2.

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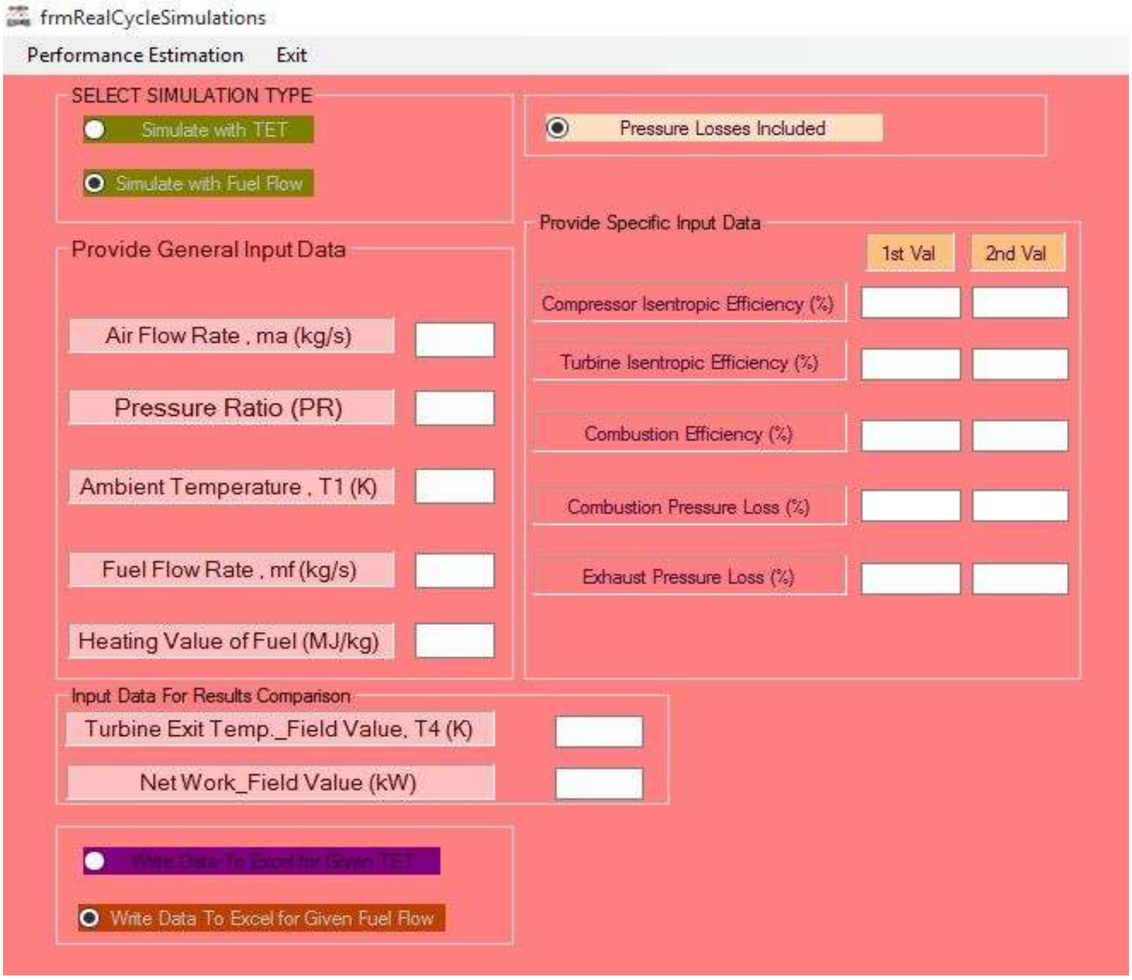

Figure 2: Software interface for creating engine model

2.2 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 steam turbine from the combined cycle arrangement is replaced by a heat recovery steam generator (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 3 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 4. 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 tot 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 practically feasible.

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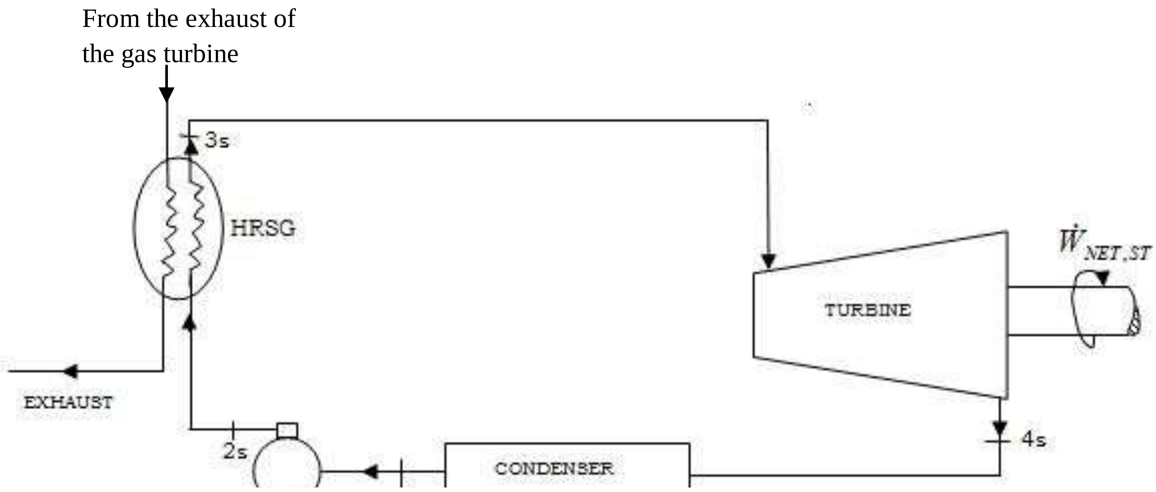

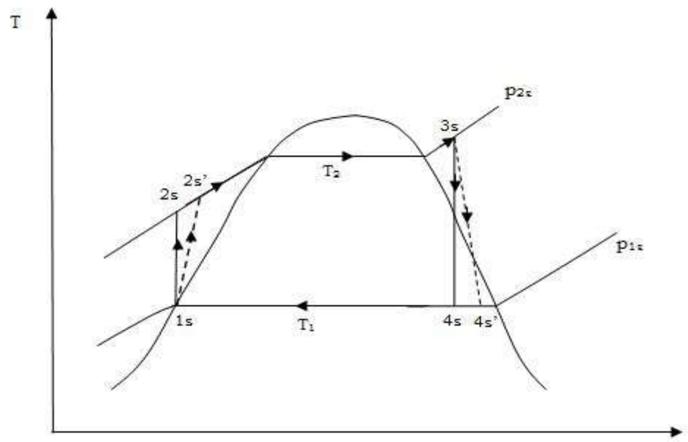

Figure 4: T-s diagram of the steam turbine cycle

Aside the effectiveness of the HRSG, the following parameters were 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 was estimated and hence the power output from the plant and other parameters of the plant were also 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.

The Power Output and Thermal Efficiency of Steam Turbine Plant

The thermal efficiency of the steam turbine pant is given by Equation (11),

$$\eta_{ST} = \frac{W_{NET,ST}}{Q_{IN,ST}} \quad (11)$$

Where $W_{NET,ST}$ is the net power output from the steam plant and $Q_{IN,ST}$ is the heat input into the plant. The exhaust gases 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,

$$Q_{EX} = mc_p(T_4 - T_5) \quad (12)$$

wherem 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

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is taken as 100°C in this work. The heat input into the steam turbine plant will be,

$$Q_{IN,ST} = \varepsilon Q_{EX} \quad (13)$$

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

$$W_{NET,ST} = W_{ST} - W_p \quad (14)$$

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,

$$W_{ST} = \dot{m}_s(h_{3s} - h_{4s'}) \quad (15)$$

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 $\eta_{\mathbf{i,st}}$ and the ideal enthalpy at the turbine exit h_{4s}. It is given by Equation (16),

$$h_{4s'} = h_{3s} - \eta_{\mathbf{i,st}}(h_{3s} - h_{4s}) \quad (16)$$

Alternatively the turbine work output is given by Equation (17),

$$W_{ST} = \dot{m}_{s\times}\eta_{\mathbf{i,st}}(h_{3s} - h_{4s}) \quad (17)$$

The ideal enthalpy at the exit of the turbine is given by Equation (18),

$$h_{4s} = h_{f1} + x_{4s}h_{fg@p1} \quad (18)$$

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 (19)

$$x_{4s} = \frac{s_{4s} - s_{f1}}{s_{fg@p1}} \quad (19)$$

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 were read from the steam tables.The work consumed by the pump is given by Equation (20)

$$W_p = \frac{\dot{m}_w \times v_{f1}(p_{2s} - p_{1s})}{\eta_{\mathbf{i,p}}} \quad (20)$$

where $\dot{m}_w$, v_{f1}, $\eta_{\mathbf{i,p}}$, 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.

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 (21),

$$\dot{m}_w = \frac{Q_{IN,ST}}{h_{3s} - h_{2s'}} \quad (21)$$

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.

$$w_p = \frac{\dot{W}_p}{\dot{m}_w} = \frac{v_{f1}(p_{2s} - p_{1s})}{\eta_{\mathbf{i,p}}} = h_{2s'} - h_{f1}$$

$$h_{2s'} = w_p + h_{f1} = \frac{v_{f1}(p_{2s} - p_{1s})}{\eta_{\mathbf{i,p}}} + h_{f1} \quad (22)$$

2.3The Combined Gas and Steam Turbine Cycle Plant

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 5 shows the schematic diagram of the combined cycle plant while Figure 6 shows the T-s diagram of the plant. Exhaust gases from the gas turbine are made to pass through a heat recovery steam generator (HRSG) to produce steam for the steam turbine. The HRSG takes the place of a boiler in the traditional steam turbine cycle.

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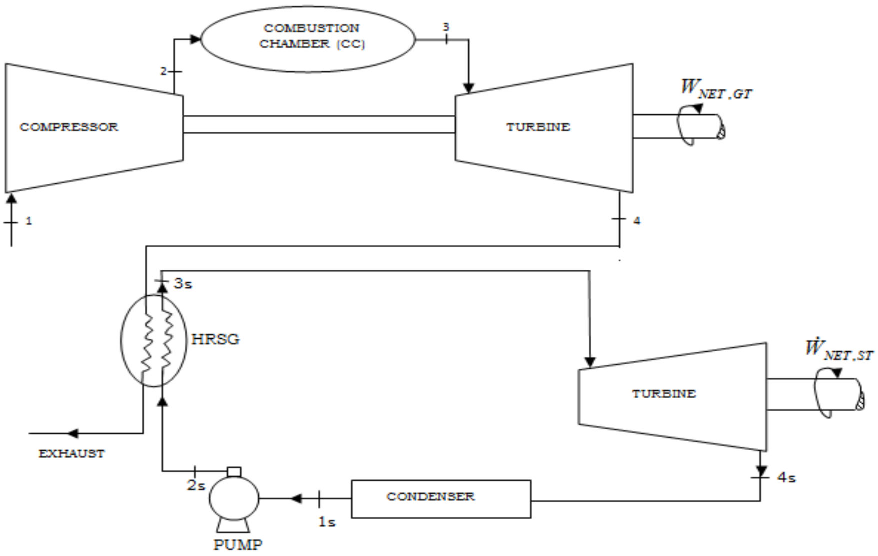

Figure 5:Schematic diagram of the combined cycle plant

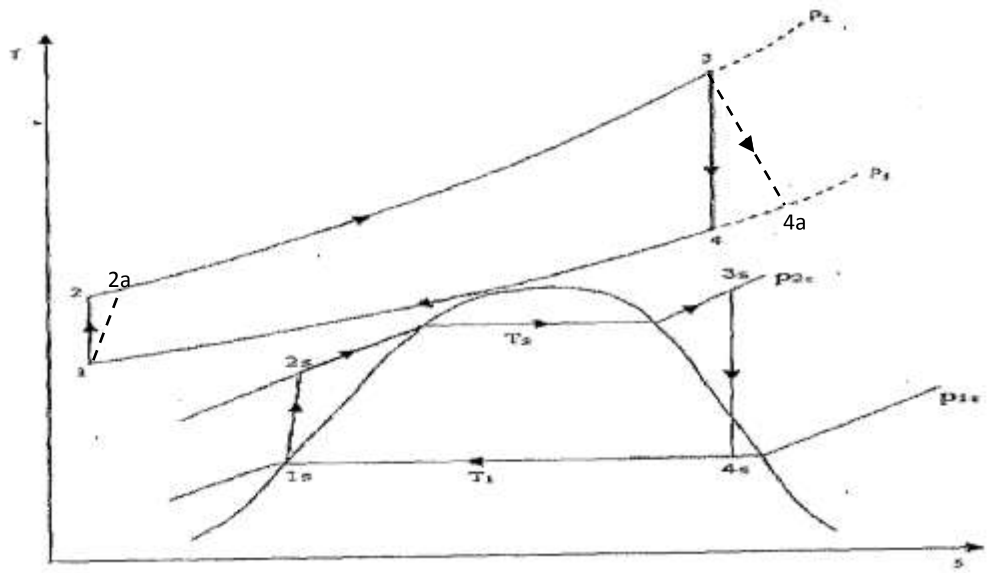

Figure 6: T-s diagram of the combined cycle plant

The thermal efficiency of the combined cycle plant $\eta_{TH,CC}$ is,

$$\eta_{TH,CC} = \frac{\dot{W}_{NET.GT} + \dot{W}_{NET.ST}}{\dot{Q}_{IN.GT}}$$

$$= \frac{\dot{W}_{NET.GT}}{\dot{Q}_{IN.GT}} + \frac{\dot{W}_{NET.ST}}{\dot{Q}_{IN.GT}} \qquad (23)$$

It can be easily show that the thermal efficiency of the combined cycle plant is as give by Equation (24) [19]

$$\eta_{TH,CC} = \eta_{GT} + \eta_{ST} \times \varepsilon \left(1 - \eta_{GT}\right) \qquad (24)$$

Where ε is the effectiveness of the HRSG.

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To apply Equation (24) to estimate the thermal efficiency of the combined cycle plant the following three parameters must be known:

i. The thermal efficiency of the gas turbine cycle,
ii. The thermal efficiency of the steam turbine cycle, and
iii. The effectiveness of the HRSG

Alternatively, if the amount of heat input into the gas turbine plant is known together with the power outputs from the gas turbine and the steam turbine plants respectively, then, the combined cycle plant efficiency can be calculated. Using a gas turbine power plant in Rivers state as a case study, the performance of the steam turbine part to be incorporated can be simulated.

2.3.1 Alternative Approach to Power Output from Combined Cycle Plant

Considering the work by Ragland &Stenzel (2000)[20]where the power output from the gas turbine plant represents 66% of the total power output of the combined cycle plant, the total power output from the combined cycle plant can be easily obtained. If the power output from the gas turbine plant is given as $W_{NET,GT}$, the power output of the combined cycle plant and that from the steam cycle plant are given by Equations (25) and (26) respectively:

$$W_{NET,CC} = \frac{W_{NET,GT}}{0.66} \tag{25}$$

$$W_{NET,ST} = 0.34 \times W_{NET,CC} \tag{26}$$

The results of the simulated combined cycle power plant were compared with the proposed percentage values provided by Ragland &Stenzel (2000). Also, this proposed percentage values were applied to the NIPP power plants to obtain the additional power obtainable from each power plant.

3.4 Economic Analysis

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 operating the simple 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).

Payback Period

The payback period represemts 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 (27),

$$PBP = \frac{IC}{NACF} \tag{27}$$

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 (28),

$$NACF = AR - AC \tag{28}$$

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 lectricity varies with the users. An average value was used in this work.

Net Pesent 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 (29)

$$NPV = \sum_{i=1}^{n} \frac{AR_i}{(1+r)^i} - \sum_{i=1}^{n} \frac{AC_i}{(1+r)^i} - IC \tag{29}$$

wherer 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.

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 (30),

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

(3.52)

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 econopic 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 AND DISCUSSION

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 while the simulated results of the steam turbine power plant are shown in Table 5. 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 in Figure 7. The results of the combined cycle power plant are shown in Table 6 while in Figure 8, the effect of the effectiveness of the HRSG on the efficeincy of the combined cycle plat is shown is presented.A comparison of the efficiency, specific comsumption and the heat rate of the gas turbine plant, the simulated steam turbine plant and the simulated combined cycle plant are presented in Figure 9, Figure 10 and Figure 11 respectively. A comparison of their respective net power outputs is provided in Figure 12.

Table1: 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 value 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	Wnet	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}	^oC	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

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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	₦/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	Wnet	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

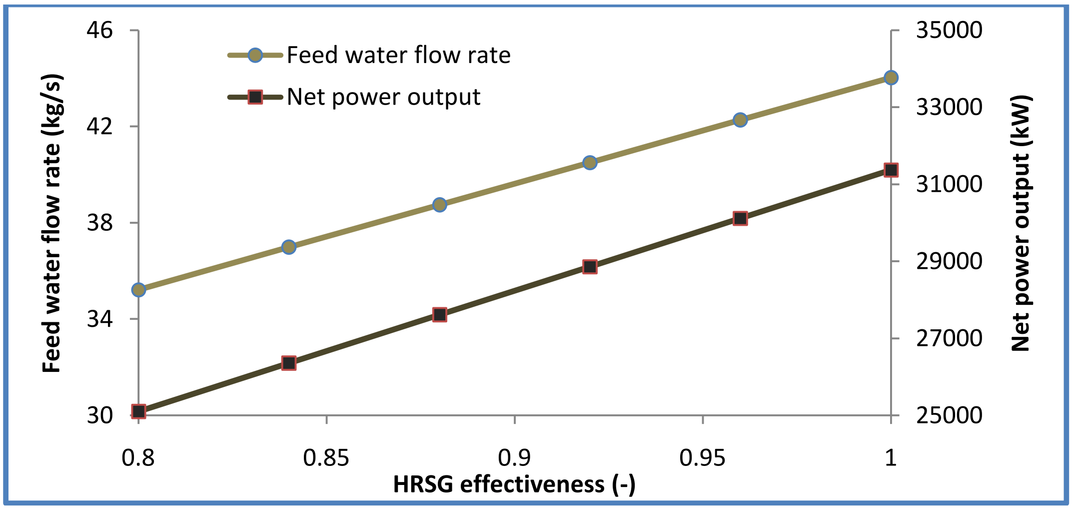

Figure 7: Variation of feed water flow rate and net power output with the effectiveness of the HRSG

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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

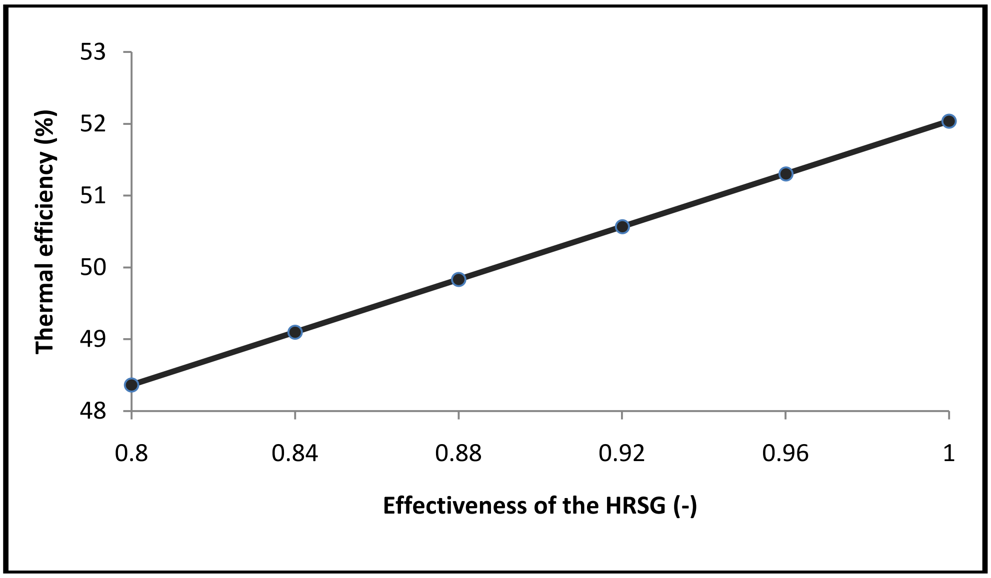

Figure 8: Effect of the effectiveness of the HRSG on the combined cycle thermal efficiency

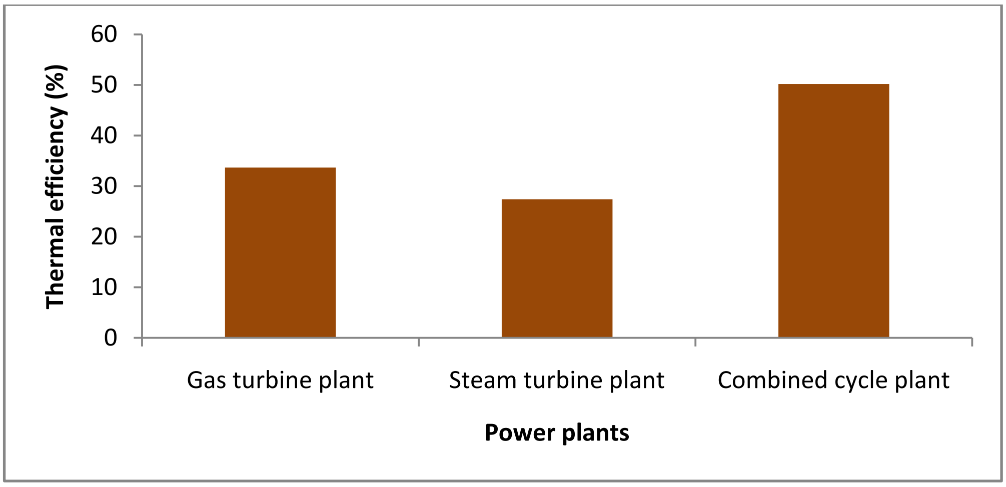

Figure 9: Comparison of the thermal efficiencies of the plants

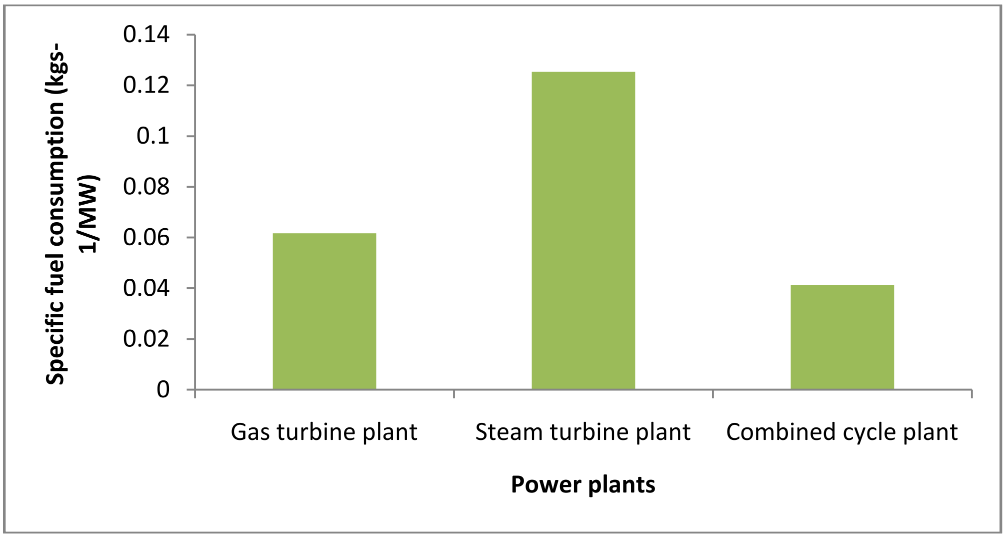

Figure 10: Comparison of the specific fuel consumptions of the plants

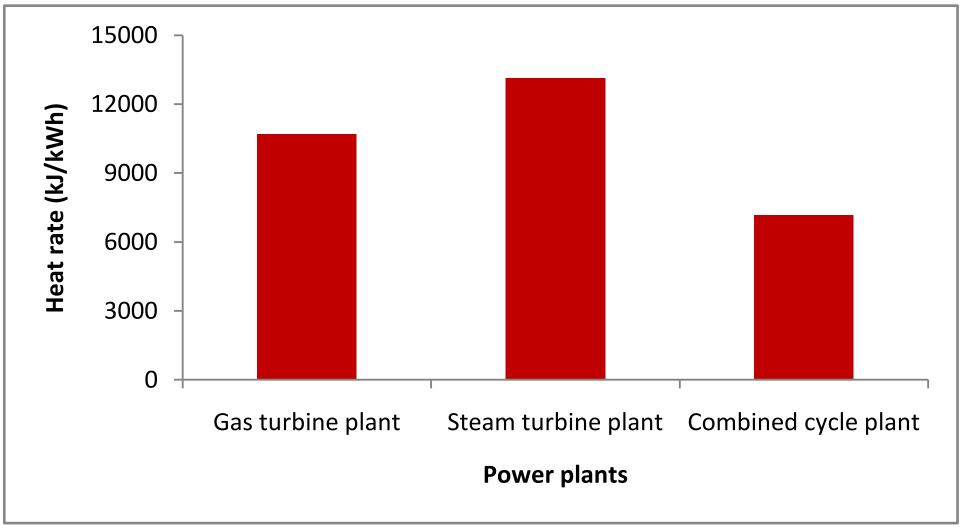

Figure 11: Comparison of the heat rates of the plants

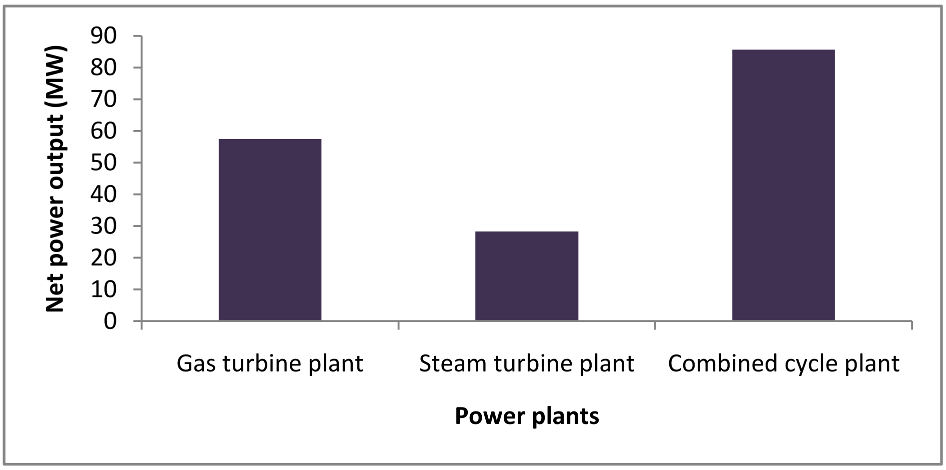

Figure 12: Comparison of the net power outputs of the plants

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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 Tables 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 HRSG. Figure 7 shows that both the feed water fow 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 therma efficiency of the combined cyce plant increses when the effectiveness of the HRSG increases. This is shown in Figure 8.

The thermal efficiency, specific fuel combustion and heat rate of the plants presented in Figures 9 to 11 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.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 10. The heat rate values shown in Figure 11 indicate that the combined cycle plant consumes far less heat compared to both the steam tiurbine plant and the gas turbine plant to produce uiit amout 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 12. The comparison is made here to see if the contribitions from the gas turbine and the steam turbine parts forming the combined cycle plant follows 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. The 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 obtainable if converted to combined cycle power plants

The results of economic analysis of the gas turbine plant and the combined cycle plant are shown in Tables 8 and 9 respectively. The analysis was done for 20 years but only values for 11 years are shown in the Tables for conveniece.

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	3	5	7	9	11	13	15	17	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	-43.09	
NACF (Million $)	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	8.10	161.99	
Discount factor at 10 %	0.909	0.751	0.620	0.513	0.424	0.351	0.290	0.239	0.198	0.1635	0.1486		
Discounted NACF (M$)	7.36	6.09	5.03	4.16	3.43	2.84	2.35	1.94	1.60	1.32	1.20	68.96	25.87
Cumulative NACF	7.36	20.14	30.70	39.43	46.64	52.61	57.53	61.61	64.97	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		
Discounted AC	33.61	27.78	22.96	18.97	15.68	12.96	10.71	8.85	7.31	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		
Energy prod. Discounted (GW-hr)	366.0	302.5	250.0	206.6	170.8	141.1	116.6	96.6	79.66	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	3	5	7	9	11	13	15	17	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	-67.09	
NACF (Million $)	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	15.37	307.5	
Discount factor at 10 %	0.909	0.751	0.621	0.513	0.424	0.350	0.290	0.239	0.198	0.164	0.149		
Discounted NACF (M$)	13.98	11.55	9.55	7.89	6.52	5.39	4.45	3.68	3.04	2.51	2.29	130.9	63.8
Cumulative NACF	14.0	38.2	58.3	74.9	88.5	99.9	109.2	116.9	123.3	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		
Discounted AC	47.13	38.95	32.19	26.61	21.99	18.17	15.02	12.41	10.26	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		
Energy prod. Discounted (GW-hr)	545.9	451.2	372.9	308.2	254.7	210.5	173.9	143.8	118.8	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 ₦ 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)..

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IV. CONCLUSIONS

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. The efficiency of the steam turbine plant is alwayslower than the efficiency of the gas turbine plant. The additional power obtainable from the steam turbine plant is about 33% of the total power obtainable from the combined cycle. This value is very close to that obtained in previous studies. Thus the thermal efficiency of the combined cycle plant is about 33% higher than that of the simple cycle gas turbine plant. All performance parameters of the combined cycle plant (both engineering and economic) are better than those of the simple cycle gas turbine plant, andWhen the gas turbine power plants in the NIPP are converted to combined cycle power plants additional power output of about 1.6 GW can be obtained.

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