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Performance Improvement of Industrial Gas Turbine for Power Augmentation in the Nigeria Oil and Gas Sector

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ABSTRACT: This paper provides a comprehensive overview of the electricity supply situation in Nigeria, focusing specifically on enhancing the performance of industrial gas turbines, GE GT13E2 (Afam Power Plant owned by FIPL) in Port Harcourt. The study encompasses two primary aspects. Firstly, it involves an examination of the nominal operating conditions of a combined cycle system and explores their dynamic performance through advanced simulations. Secondly, the research introduces a supplementary burner into the combined system and conducts a comparative analysis, considering factors such as system efficiency and power output. The power plant configurations are simulated using Gasturb14. The analysis is carried out by establishing the maximum attainable temperature in the Heat Recovery Steam Generator (HRSG), which determines the number of stages of supplementary firing that can be implemented. The results presented in this paper unveil the substantial potential for power generation with the addition of supplementary burners, indicating a remarkable increase in power output, yielding approximately 187.444 MW. However, this significant boost in power generation comes with an associated trade-off, wherein the overall system efficiency experiences a reduction. From an economic perspective, the additional power generated by the afterburner translates directly into increased revenue from electricity sales. Considering the average residential electricity rate in the U.S. as of February 2023, the calculated additional revenue from the afterburner amounted to approximately $25,641.58 per hour.

KEYWORDS: Combined Cycle, Efficiency, Performance, Supplementary Burner, Power Generation, Heat Recovery Steam Generator.

I. INTRODUCTION

There is an increased emphasis on increased power output of industrial gas turbines in Nigeria's oil and gas sector. Gas turbine power plants dominate the Nigerian power sector, accounting for over 80% of the total grid-based installed capacity, and this is primarily due to the availability of natural gas at relatively lower costs compared to distillate fuels. The power that is currently being generated from these gas turbines is not enough to meet the power demand in Nigeria and going for further installations would incur a huge cost to the country's suffering economy; hence, it is more sustainable to improve the performance of Industrial Gas Turbines for improved power generation.

Gas turbines are typically designed to operate under International Organization for Standardization (ISO) conditions, with intake air at 15°C and 60% relative humidity. However, in practice, they often operate under off-design conditions, which leads to a loss of power production potential. The turbine performance is affected by several environmental factors such as relative humidity, ambient pressure, and temperature. When the air density at the turbine inlet is increased, this improves the performance of gas turbines. This is because higher air density means more air is available for combustion, which results in enhanced power. Cooler air is denser and contains more oxygen, which improves air density. Most of the studies on improving the performance of a simple cycle gas turbine focused on implementing a combined cycle system by coupling the gas turbine with a steam turbine to extract additional power from the waste heat in the exhaust gases. Capturing waste heat from the turbine's exhaust and using it for combined heat and power (CHP) applications or preheating combustion air can significantly reduce energy wastage and enhance overall efficiency. It was concluded that the adoption of combined cycle power plants with low specific fuel consumption (SFC) in the Nigerian oil and gas sector could yield significant results [1]. Specifically, they found that utilizing 50% of the available natural gas reserves could generate approximately 35 GW of electricity for over 50 years [1]. It's noteworthy that the existing rate of natural gas production currently enables the generation of 27.06 GW of electricity while maintaining an SFC of 0.06 kg/s per MW.

Given Nigeria's abundant natural gas reserves, the oil and gas sector is dominated by the use of Industrial Gas Turbines (IGTs) for power generation. These IGTs are designed to operate under ISO conditions, with intake air at 15°C and 60% relative humidity [2]. However, in practice, they often operate under off-design conditions due to the challenging environmental conditions in Nigeria, such as high ambient temperatures and humidity, which can significantly affect their performance. The performance of these IGTs often falls short of the increasing power demand in Nigeria, resulting in operational inefficiencies and increased costs.

Given these challenges, there is a pressing need for performance improvement of IGTs for power augmentation in the Nigerian oil and gas sector. This research aims to address this need by focusing on the GE GT13E2 turbine at the Afam Power Plant [3]. We shall calculate the gas turbine cycle and design a parametric study using the simulation software, Gasturb14. By examining its nominal operating conditions and exploring its dynamic performance through advanced simulations, the study seeks to enhance its performance and, consequently, augment power generation in the Nigerian oil and gas sector.

Although previous work has been done to improve the performance of industrial gas turbines, the objective of this paper is to further investigate supplementary firing by adding further heat to the exhaust gases after they pass through the gas turbine. Thereafter, we shall determine the gas turbine cycle and design a parametric study using the simulation software, Gasturb14. The primary aim of this research is to enhance the overall power output of the gas turbine through the implementation of supplementary firing in a combined cycle system. To achieve the aim, the following objectives were targeted:

- Conducting a dynamic performance analysis of the gas turbine under nominal operating conditions.
- Introducing a supplementary burner into the combined system and analysing its impact on system efficiency and power output.
- Investigating the potential trade-offs between increased power generation and overall system efficiency.

Waste heat recovery systems generate power from low-temperature heat, and as a result, a relatively small amount of electricity is generated. To increase power generation, a Sequential Supplementary Firing Combined Cycle (SSFCC) is employed to effectively boost the exit temperature and pressure of the gases [4]. The heat recovery steam generator (HRSG) extracts this heat energy from the exhaust gases and uses it to generate steam. The heat from the exhaust gases is then transferred to a heat transfer fluid, often water, which results in the generation of high-pressure and high-temperature steam used to drive a steam turbine, thus producing additional electrical power. The enthalpy of steam is directly proportional to its temperature and pressure. Therefore, increasing the heat supplied to the HRSG increases the temperature and pressure of the steam produced. This results in an increase in enthalpy at both the inlet and outlet of the turbine, which increases its power output 1. In addition to increasing power output, higher heat input can also lead to higher thermal efficiency and lower emissions 1. However, it’s important to note that there are limits to how much heat can be supplied to an HRSG before it becomes inefficient or unsafe.

As gas turbines have become more advanced and complex, the cost of hardware testing has risen. Consequently, gas turbine simulations have become invaluable tools throughout the engine life cycle. Accurate cycle calculation programs rely on a good understanding of gas properties. In GasTurb, the working fluid is assumed to behave like a half-ideal gas, where gas properties such as specific heat, enthalpy, entropy, and gas constant are functions of temperature and gas composition but not dependent on pressure. It's important to note that gas turbines typically use hydrocarbons as fuel, with specific characteristics and heating values. In GasTurb 14, the primary fuel type is known as Generic Fuel, but the software allows for flexibility in selecting other fuels like diesel, natural gas, or hydrogen for cycle calculations.

II. SEQUENTIAL SUPPLEMENTARY FIRING COMBINED CYCLE (SSFCC)

This study looks into the performance analysis of supplementary firing in combined cycle systems, comparing it with a base-case scenario that lacks supplementary firing. The incorporation of supercritical steam generation in the HRSG has the potential to marginally improve the thermal efficiency of natural gas firing, thereby reducing the efficiency penalty to 5.7% points low heating value (LHV), as noted by [4]. While the efficiency remains somewhat lower than in the conventional configuration, the increased power output in the combined steam cycle can lead to a reduction in the number of gas turbines required, achieving a power output similar to that of a conventional natural gas combined cycle. This, in turn, positively affects the number of absorbers and capital costs in post-combustion capture plants by halving the total flue gas volume on a normalized basis. It is important to note that the utilization of SSFCC results in a significant boost in power output. However, it comes at the cost of a reduction in the overall efficiency of the gas turbine cycle due to increased fuel consumption and emissions.

III. SUPPLEMENTARY FIRING IN GTS

In the pursuit of the most cost-effective electricity generation methods, utilities face various constraints, including fuel availability, the peak-to-base demand ratio, and adherence to national environmental standards. To comprehensively evaluate how a power plant performs across its anticipated operating ranges, mathematical models have been developed. These models can predict performance under various conditions, encompassing both design-point and off-design or part-load operational scenarios.

The exhaust from a gas turbine holds significant energy that can be harnessed through an HRSG to generate steam. This steam serves various purposes, including industrial processes, heating, driving turbines, and even reducing nitrogen oxide (NOx) emissions and enhancing power output when injected into the gas turbine's burner. Furthermore, steam finds utility in cooling the gas turbine's blades and vanes. The use of steam for power generation in a combined cycle augments both the total electrical power output and the overall electric efficiency of the plant, a concept known as Combined Cycle. When steam is utilized for industrial processes in the chemical industry, it falls under the category of Cogeneration.

Existing literature extensively explores gas turbine performance, primarily focusing on varying thermodynamic parameters and analysing performance within the framework of Combined Cycle or Cogeneration applications. However, not many related open literature have studied SSFCC. Non-sequential, single-stage, supplementary firing is typically used in Natural Gas Combined Cycle (NGCC) power plants to increase power output by around 30% during times of peak demand for electricity and high electricity selling prices [5]. Burning supplementary fuel in consecutive stages increases the heat available in the HRSG and leads to a larger combined cycle power output and a reduction of the number of the GT trains, at constant power output.

Studies by [6] have proposed the implementation of supplementary firing in gas-fired power plants with carbon capture. This approach has been observed to significantly impact the composition of exhaust gases, leading to alterations in temperature differentials within the HRSG and higher gas temperatures, albeit accompanied by increased heat transfer irreversibility. Notably, supplementary firing introduces challenges related to the use of advanced alloys to withstand elevated temperatures. Supplementary firing efficiently utilizes excess air, which is employed to moderate flame temperatures in the gas turbine's combustor. Additional fuel, burned in supplementary burners typically located at the inlet duct of the HRSG, capitalizes on the excess oxygen levels in exhaust gases, resulting in higher steam production compared to an unfired unit. While this practice has historically been employed to respond to peak electricity demand and increase revenue, it can impact the thermal efficiency of the cycle.

A three-stage supplementary firing approach using a single-pressure HRSG and a supercritical steam turbine to enhance cycle efficiency was proposed by [7]. This method involves firing natural gas at three points in the primary heat exchange section to mitigate peak HRSG temperatures during supplementary power generation. Maximum temperatures reached can be as high as 760°C. Conversely, it was suggested by [8] that the possibility of achieving even higher maximum temperatures is with modifications to the HRSG. These enhancements include insulated casings and water-cooled furnaces, pushing temperatures up to 1316°C. To maintain reasonable costs and avoid the use of advanced alloys, it's common to limit exhaust gas temperatures to around 820°C. With sequential supplementary firing, additional fuel is burned in multiple stages throughout the HRSG, with each stage constrained by temperature limitations on heat exchangers. This method offers the advantage of significantly increasing the steam cycle's output, thus reducing the need for multiple gas turbine trains. Additionally, transitioning to advanced supercritical steam conditions can maintain acceptable overall efficiency by enhancing the steam cycle's efficiency, despite a substantial portion of the total output being generated by the combined cycle.

The configuration proposed by [9] combines sequential supplementary firing with a double reheat supercritical steam turbine in a combined cycle (SSFCC). This configuration achieves the same power output as a reference setup consisting of a single GE 937 IFB gas turbine, a single HRSG, and a single-pressure supercritical steam cycle with three turbines. The HRSG in this configuration employs a Once Through Steam Generator (OTSG), burning supplementary gas in five stages to maximize the use of excess O_2, while maintaining it at 1% v/v concentration. The temperature peaks after each of the first three stages of additional gas combustion, reaching 820°C, followed by around 790°C in the fourth stage and 700°C in the last stage. This configuration optimally balances in-duct burners and their respective heat recovery sections to minimize irreversibility and maximize steam production. CO_2 concentration in the exhaust gas closely approaches stoichiometric limits, measuring twice as large as that in a conventional unfired NGCC. While subcritical combined cycles are typically applied in NGCC plants, supercritical cycles with double reheat are now increasingly used in coal-fired steam plants and can effectively leverage sequential supplementary firing in the HRSG to substantially improve performance [10].

IV. DETERIORATION IN GAS TURBINE

It's widely acknowledged that gas turbines experience deterioration over time during operation. Failure to account for this deterioration can result in significant economic losses for gas turbine operators or owners [11]. With the advent of high-performance gas turbines and rising fuel costs, maintaining optimal efficiency in gas turbine systems has become a necessity. For instance, a large-scale 934 MW combined cycle plant can incur an annual fuel bill of 203 million dollars. Over a life cycle of 20 years, the fuel cost could exceed four billion dollars, assuming a flat fuel cost of $4/MM BTU. With a moderate escalation in fuel cost, this could easily total six or seven billion dollars [12]. Because of severe environmental conditions, industrial gas turbines experience fouling while in operation, leading to a decrease in overall output. Fouling takes place at the compressor due to the ingestion of particles from the operating environment during intake, ultimately diminishing engine performance and consequently reducing power output [13]. In a deregulated electric utility market that fosters competition among power producers and in an environment where fuel costs have dramatically escalated, plant operators must understand and control all forms of gas turbine performance deterioration. This deterioration can affect the performance, efficiency, and lifespan of the turbine. Gas turbine operators often look for means to reduce and manage performance degradation since it has a direct impact on their profitability [14].

Thermal degradation occurs due to the high operating temperatures in the turbine, which can lead to material weakening and eventual failure. At temperatures exceeding 1000°F (538°C), nickel-based superalloys are susceptible to high-temperature oxidation. This process involves the reaction of oxygen in the gas stream with the nickel alloy, resulting in the formation of a nickel-oxide layer on the airfoil surface [12]. When exposed to operational conditions involving vibration and start/stop thermal cycles, this nickel oxide layer tends to develop cracks and spalling. The phenomenon may extend to the inner surfaces of blade cooling passages, leading to potential blade failure. Protective coatings are available to alleviate this effect, applicable to both the external blade surface and internal cooling holes [15].

V. METHODOLOGY

A standard gas turbine cycle is considered for the present analysis to make it easier for results comparison. In this study, the selected actual engine, GE GT13E2 (Afam Power Plant owned by FIPL) from General Electric Power Systems is made up of 2 phases. At present, Phase 1 is operational with an installed GE (formerly Alstom) GT13E2 gas turbine of 180MW capacity, exporting an average of 3500MWH per day into the national grid. The expectation is that phase 2 of the same type of unit would be installed and come online before the end of 2019. Natural gas for the operation of the Afam turbine is supplied by Shell Nigeria. The GT13E2 can help meet over 38% simple-cycle efficiency and over 55% combined-cycle efficiency. The GT13E2 model can operate with up to 30% hydrogen and has features that allow it to adjust to fluctuations in demand and fuel prices. It's also capable of fast starts and has a low part load capability, which makes it suitable for various applications. To undertake a design-point analysis for the chosen GT engine, a set of available practical data was used, as summarized in Table 1.

In the combined cycle system depicted in Figure 1, the process begins with ambient air at Stage 1 being compressed isentropically (Stage 2) and then actually (Stage 2') in the compressor. The compressed air then undergoes heat addition in the combustion chamber at Stage 3, resulting in high-temperature, high-pressure gases. These gases expand isentropically in the gas turbine at Stage 4, performing work, but then the actual expansion accounts for inefficiencies at Stage 4'. A supplementary burner further heats the exhaust gases and transfers this heat to the HRSG at Stage 5, generating steam while cooling down. The HRSG transfers heat to water, creating high-pressure steam at Stage 6, which expands isentropically in the steam turbine at Stage 7 but actually expands with some inefficiencies at Stage 7'. The exhaust steam is then condensed in the condenser at Stage 8 and pumped back into the HRSG Stage 9 to restart the cycle.

Table 1: Technical Specification of Studied Combine Cycle Engine

Component Parameters	**Value**	
Ambient Conditions	Total Temperature T1	300 K
	Total Pressure P1	101.325 kpa
	Ambient Pressure Pamb	101.325 kpa
	Relative Humidity	60 %
	Inlet pressure	93 kPa
	Airflow rate	70 kg/s
	Power output	20 MW
	Altitude	735 m
	Pressure ratio, P2/P1	20
Compressor	Stages	17
	Speed	5100 rpm
	Type	axial flow, heavy duty
	Air compressor efficiency, ηc	0.88
	Rel. Enthalpy of Interst Bleed	0.712
	Compr Interstage Bleed/W2	0.0485
Combustor	Fuel net calorific value, NCV	49.74 Mj/Kg
	Fuel	Natural gas
	Combustion efficiency, η_{comb}	0.9999
	Mechanical efficiency	0.9999
	Burner Pressure Ratio	0.97
	Burner Exit Temperature	1502 K
Turbine	Stages	2
	Speed	5100 rpm
	Inlet turbine temperature	1350 K
	Turbine efficiency, ηt	0.885
	Turbine Exit Duct pressure ratio	0.98
	Exhaust Pressure ratio/Pamb	1.03
	NGV 1 cooling air/W2	0.09426
	Rotor 1 cooling air/W2	0.145
HRSG	Design Steam Temperature	800k
	Steam Exit Pressure	10000 Kpa
	Econ and Evap H20 pressure ratio	0.980769
	Superheater H2O pressure ratio	0.980392
	Rel. heat to ambient	0.01
	Feed water Temperature	381.483 K
	Pinch Delta T at Evaporator	10 K
	Evaporator Approach Delta T	10 K
	Rel. Steam Bleed Evaporator	0.01

	Duct burner exit temperature	1300 K
	Duct burner efficiency	0.9
	Design Pressure ratio	0.98
	Design inlet Mach No	0.1

The research design for this study is a combination of theoretical analysis and simulation modeling. The theoretical analysis involves a detailed review of the existing literature on gas turbine performance, degradation, and improvement techniques. The simulation modeling involves the use of GasTurb14. GasTurb14 is equipped with features for detailed and integrated design of electric propulsion systems. The electric system can be designed, and optimized, and its off-design performance calculated in varying levels of detail - just as required. The software simplifies the tasks that performance engineers encounter most frequently with a user-friendly interface that presents information in both easy-to-understand lists and graphically. It also provides versatile graphical output and adaptable models. This software is used to simulate the performance of the GE GT13E2 under nominal operating conditions and with the introduction of a supplementary burner. The simulation model is calibrated using the collected operational data to ensure its accuracy.

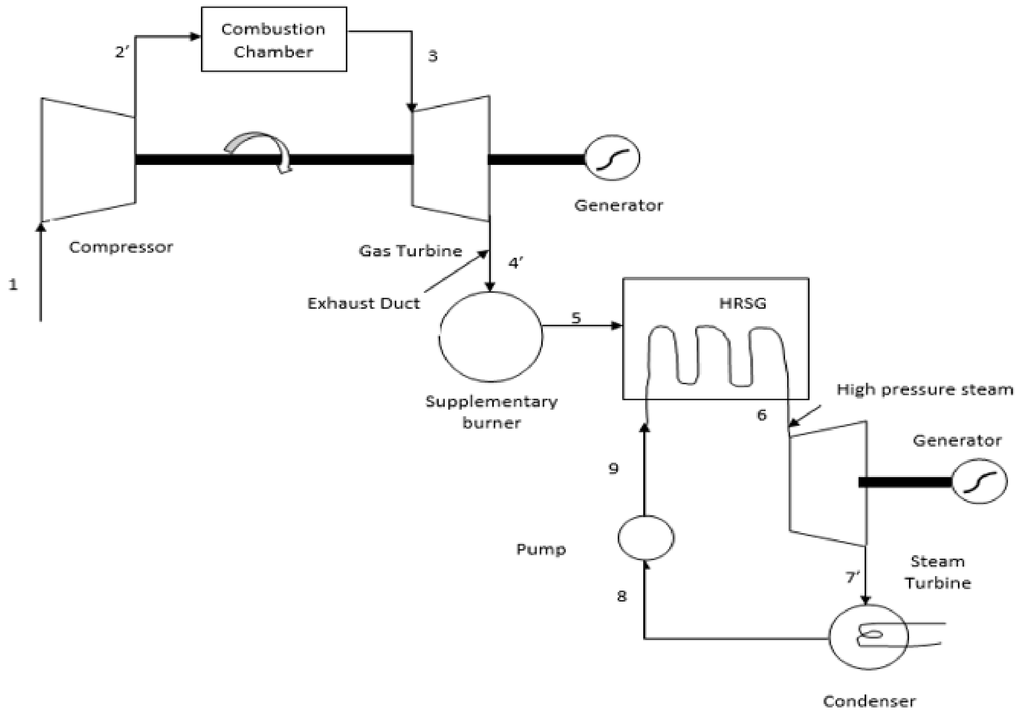

Figure 1: Simple Sketch of a Combine Cycle System

Energy transfer in a combined cycle power plant involves a complex series of thermodynamic processes designed to maximize efficiency and power output. The cycle begins with ambient air entering the compressor at Stage 1. During the isentropic compression process (Stage 1-2), the air is compressed adiabatically, meaning there is no heat exchange with the surroundings. The compression increases the air's pressure and temperature without a change in entropy, thereby preparing it for efficient combustion. The specific work required for this compression is supplied by the shaft work from the turbine.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad (1)$$

where T is temperature, P is pressure, and γ is the specific heat ratio. The actual work done on the air during compression is given by:

$$W_c = \frac{C_p(T_2' - T_1)}{\eta_c} \quad (2)$$

where W_c is the compressor work, C_p is the specific heat at constant pressure, (η_c) is the compressor efficiency, and T represents temperatures at the respective stages. The compressed air at Stage 2' is mixed with fuel and ignited in the combustion chamber (Stage 3). The combustion process significantly increases the temperature and pressure of the gases, converting chemical energy from the fuel into thermal energy. This stage is crucial as it determines the total energy available for the subsequent expansion processes. The high-temperature, high-pressure gases are now ready to perform work in the turbine. The heat added in the combustion chamber can be expressed as:

$$Q_{in} = m \cdot C_p(T_3 - T_2') \quad (3)$$

where (Q_{in}) is the heat added, m is the mass flow rate, and T represents temperatures at the respective stages. The high-energy gases expand isentropically through the gas turbine (Stage 3-4), converting thermal energy into mechanical work. This expansion drives the turbine blades, which in turn drives the compressor and the generator, producing electrical power. The efficiency of this process is critical as it directly influences the overall performance of the power plant. Ideally, the expansion should be adiabatic, with no heat loss and constant entropy. The isentropic relationship for expansion can be expressed as:

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} \quad (4)$$

The actual work output from the turbine can be expressed as:

$$W_t = \eta_t \cdot C_p(T_3 - T_4') \quad (5)$$

where (η_t) is the turbine efficiency. After expanding through the gas turbine, the exhaust gases still contain substantial thermal energy. In Stage 5, a supplementary burner can be used to add more fuel to the exhaust gases, increasing their temperature before entering the Heat Recovery Steam Generator (HRSG). The HRSG plays a vital role by capturing this waste heat and transferring it to water, converting it into high-pressure steam. The energy added by the supplementary burner is:

$$Q_{supp} = m \cdot C_p(T_5 - T_4') \quad (6)$$

The water in the HRSG absorbs heat from the exhaust gases, reaching high temperatures and pressures at Stage 6. This steam generation process is critical as it recovers energy that would otherwise be lost, enhancing the overall efficiency of the combined cycle system. The steam produced is now ready to perform work in the steam turbine. The heat transferred in the HRSG can be expressed as:

$$Q_{HRSG} = m \cdot h_{fg} \quad (7)$$

where (h_{fg}) is the enthalpy change during phase change from liquid to vapour. The high-pressure steam expands isentropically through the steam turbine (Stage 6-7), similar to the gas turbine expansion process. This expansion converts thermal energy from the steam into mechanical work, which drives another generator to produce additional electrical power. The isentropic expansion ensures that the process is adiabatic, maintaining constant entropy. The isentropic relationship for steam expansion can be expressed as:

$$\frac{T_7}{T_6} = \left(\frac{P_7}{P_6}\right)^{\frac{\gamma-1}{\gamma}} \quad (8)$$

The actual work output from the steam turbine can be expressed as:

$$W_{st} = \eta_{st} \cdot (h_6 - h_{7'}) \quad (9)$$

where h represents enthalpy at the respective stages and (η_{st}) is the steam turbine efficiency. The exhaust steam from the steam turbine enters the condenser at Stage 7, where it is cooled and condensed back into water. This condensation process releases latent heat, which is typically rejected to a cooling medium such as air or water. The condensed water is then pumped back to the HRSG at Stage 9, completing the cycle. The pump work required to return the water to high pressure is relatively small compared to the work extracted by the turbines, making this process energy-efficient. The work required for the pump can be expressed as:

$$W_p = \frac{v \cdot (P_9 - P_8)}{\eta_p} \quad (10)$$

wher v is the specific volume of the fluid, P is pressure, and (η_p) is the pump efficiency. GasTurb14 is equipped with a variety of thermodynamic equations that facilitate the analysis and optimization of gas turbines and combined cycle systems. Some of the key equations include continuity equation, which ensures mass flow rate consistency throughout the system.

$$\dot{m}_1 = \dot{m}_2 \quad (11)$$

The energy balance equation accounts for the energy added or removed in each component of the cycle.

$$\dot{Q} - \dot{W} = \dot{m}(h_2 - h_1) \quad (12)$$

Isentropic efficiency measures the efficiency of compression and expansion processes relative to their isentropic counterparts.

$$\eta_s = \frac{T_{2s} - T_1}{T_2 - T_1} \quad (13)$$

Polytropic efficiency evaluates the performance of multi-stage compression and expansion processes.

$$\eta_p = \frac{\ln(P_2/P_1)}{\ln(T_2/T_1)} \quad (14)$$

Heat transfer equations calculates the amount of heat transferred in the HRSG, combustion chamber, and supplementary burner.

$$Q = m \cdot C_p \cdot (T_{out} - T_{in}) \quad (15)$$

VI. ASSUMPTIONS

- The supplementary burner efficiency is assumed as 0.99.
- The turbines, compressor, pump, and interconnecting heat exchanger operate adiabatically.
- The turbines and compressors efficiencies are isentropic.
- There are no pressure drops for flow through the combustor, interconnecting heat exchanger, and condenser.
- An air-standard analysis is used for the gas turbine. (i.e T_0 = 300 K, p_0 = 100 kPa)

VII. RESULTS AND DISCUSSION

The gas turbine power improvement by using a supplementary system has been studied on a 180 MW gas turbine in Port Harcourt, located in Trans Amadi. The evaluation of the gas turbine's performance involves the assessment of various parameters such as inlet temperature, pressure ratios, and exhaust gas characteristics. Through detailed data analysis, the study aims to quantify the efficiency gains achieved with the supplementary burner and compare them against the baseline performance. To facilitate a systematic comparison, the study begins by assessing the combined cycle's performance without the supplementary burner under specific operational and design conditions. The integration of the supplementary burner necessitates a re-evaluation of the HRSG performance, considering factors such as steam generation efficiency and heat transfer rates. It's important to note that this analysis primarily focuses on the GT engine's design point performance, and off-design operations are not within the scope of this study.

Using GasTurb14, the author simulated the gas turbine's performance under nominal and off-design conditions. The analysis of the design-point performance of the gas turbine under study involves evaluating key performance metrics such as power output and efficiency of the combined system. The fuel chosen for this analysis is natural gas. Power output is a key performance metric, with the system generating a substantial output of 193,657.2 kW and an overall thermal efficiency of 35.39%. The HRSG component of the system also performs efficiently, with high effectiveness in its superheater (95.03%), evaporator (92.10%), and economizer (90.66%). The HRSG manages steam generation effectively, contributing to the overall system performance with an electrical power output of 45,303.0 kW and a total efficiency of 43.15%. The author ran additional simulations for off-design conditions, varying only the ambient temperature to match the climatic conditions in Nigeria. The result is summarised in Table 2. The gas turbine system produced a substantial power output of 180,092.8 kW, which matched the operating off-design condition of the GT13E2 gas turbine. The overall thermal efficiency stands at 34.47%, with the compressor and turbine showing isentropic efficiencies of 88% and 88.5%, respectively. Additional metrics include a total Heat Recovery Steam Generator (HRSG) electrical power output of 45,303.0 kW and an overall system efficiency of 43.15%. Most stations show a small deviation (around 2-3%), indicating stable performance across different stages, while the cooling flow shows a deviation of 3.6%, which suggests a slight reduction in cooling requirements under off-design conditions. Power output shows a significant decrease of 7%, which tells us that there is a reduced overall power generation capability under off-design conditions. Power Specific Fuel Consumption (PSFC) and Heat Rate show an increase of 2.7%, while Thermal Efficiency shows a small decrease of 2.6%. Fuel Flow decreased by 4.3%, which correlates with the reduced power output. We notice higher nitrogen oxide emissions under off-design conditions. Small deviations in mass flow rates suggest stable component performance, while larger deviations in emissions and power indicate areas for potential optimization.

Table 2: Comparison between Nominal Performance Results and Off-Design Performance Results

Parameter	Nominal Conditions	Off-Design Conditions	Deviation (%)
Station 1 W (kg/s)	634.165	620.229	0.022
Station 2 W (kg/s)	634.165	620.229	0.022
Station 3 W (kg/s)	645.674	632.183	0.021
Station 31 W (kg/s)	534.436	522.237	0.023
Station 4 W (kg/s)	534.436	522.237	0.023
Station 41 W (kg/s)	645.674	632.183	0.021
Station 45 W (kg/s)	645.167	632.132	0.020
Station 5 W (kg/s)	645.167	632.132	0.020
Station 6 W (kg/s)	645.167	632.132	0.020
Cool W (kg/s)	30.757	29.665	0.036
Turbine RNI	2.98	2.975	0.002
PWSD (kW)	193657.2	180092.8	0.070
PSFC (kg/(kW*h))	0.2045	0.21	-0.027
Heat Rate (kJ/(kW*h))	10172.1	10444.2	-0.027
Therm Eff	0.3539	0.3447	0.026
WF (kg/s)	11.00186	10.53427	0.043
s NOx	0.5398	0.60898	-0.128
XM8	0.2103	0.2103	0.000
A8 (m^2)	12.4433	11.2143	0.099
war0	0.00637	0.01336	-1.097
FHV	49.736	49.736	0.000

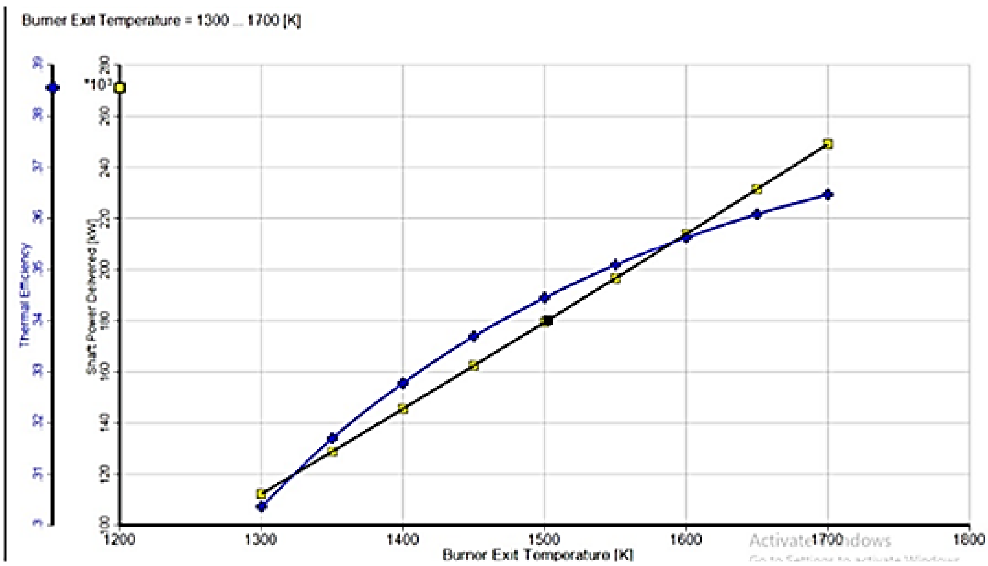

Figure 2 (a): Nominal Condition Plot

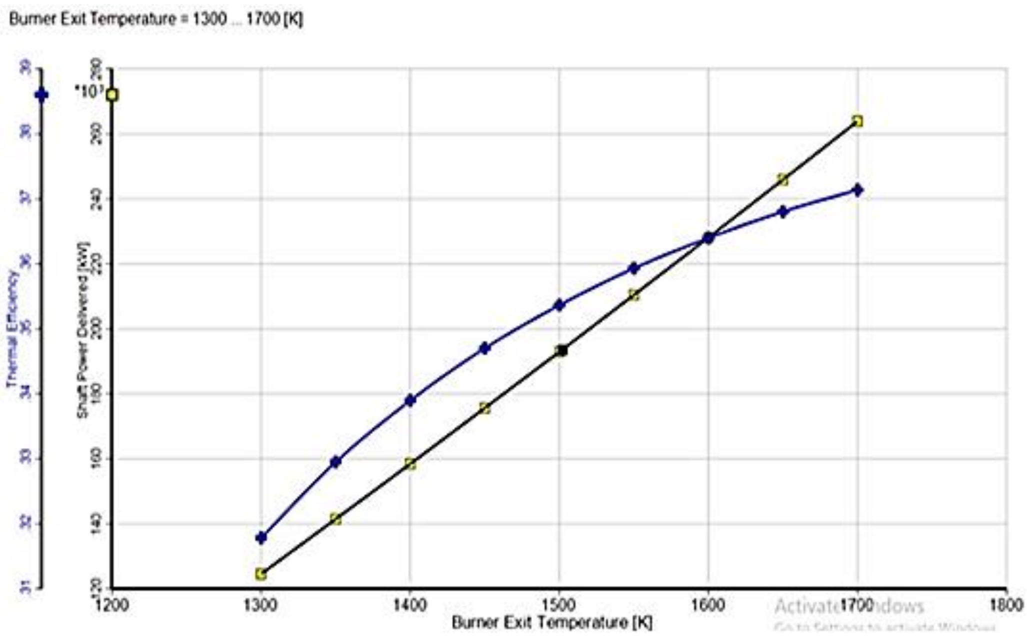

Figure 2 (b): Off-Design Condition Plot

Figures 2 (a) (b) provide plots of the gas turbine's performance under nominal and off-design conditions. In the nominal condition plot shown in Figure 4, the thermal efficiency and shaft power delivered both increase steadily as the burner exit temperature increases from 1300 K to 1700 K. The efficiency starts at approximately 0.31 and rises to around 0.37, while the power output begins near 120 kW and reaches close to 260 kW. This shows a direct and proportional relationship between burner exit temperature and both thermal efficiency and power output under nominal conditions. In contrast, the off-design condition plot in Figure 4 shows a similar trend but with slightly lower values. The efficiency under off-design conditions starts at around 0.305 and reaches approximately 0.365, while the power output ranges from about 110 kW to 250 kW as the burner exit temperature increases. This indicates that the gas turbine operates less efficiently under off-design conditions compared to nominal conditions.

VIII. THE IMPACT OF THE SUPPLEMENTARY BURNER ON THE OVERALL EFFICIENCY

The addition of a supplementary burner to a gas turbine engine significantly enhances power output by burning additional fuel in the exhaust stream. This process elevates the exhaust gas temperature beyond standard operating levels, leading to notable performance improvements. In the off-design condition, the exhaust gas temperature increased from 783.60 K to 900 K which is approximately a 12.93% rise. This higher temperature provides additional heat to the Heat Recovery Steam Generator (HRSG), boosting its efficiency. The increased exhaust gas temperature results in a higher mass flow rate of gas. Without the supplementary burner, the mass flow rate was 645.167 kg/s, which decreased to 630.740 kg/s with the burner, a reduction of approximately 2.24%. This reflects enhanced fuel combustion and greater energy input into the system.

Additionally, the mass flow rate of net steam generation rose from 65.984 kg/s to 93.838 kg/s, and the steam flow rate increased from 66.657 kg/s to 94.795 kg/s, representing increases of 42.21% for both parameters.

One of the most notable outcomes of adding the supplementary burner is the substantial increase in the electrical power output of the steam turbine. The was an output power of 75941.5 KW compared to the scenario without the burner which gave only 45233.1 KW as output, indicating a remarkable increase in power output by 67.89%. This substantial rise in power output demonstrates that the supplementary burner significantly contributes to enhanced power generation capabilities within the steam turbine system.

The graph in Figure 3 shows the performance of a combined cycle power plant and helps us better understand how the power output of the steam turbine is affected as the supplementary burner temperature is increased. As can be seen, between the temperature range of 700K to 800K, there were no significant changes in power output. But then, when the burner exit temperature was increased from 800K, the steam turbine's power output increased significantly. This phenomenon can be attributed to the fundamental principle of thermodynamics, wherein an increase in temperature differential between the hot gases and the working fluid in the steam turbine results in enhanced thermal energy conversion and subsequent power generation. The net power output in Table 3, which is the sum of the steam turbine and gas turbine power outputs, rose from 225,325.90 kW to 256,034.30 kW. This increase of 30,708.40 kW represents a percentage increase of 13.63%.

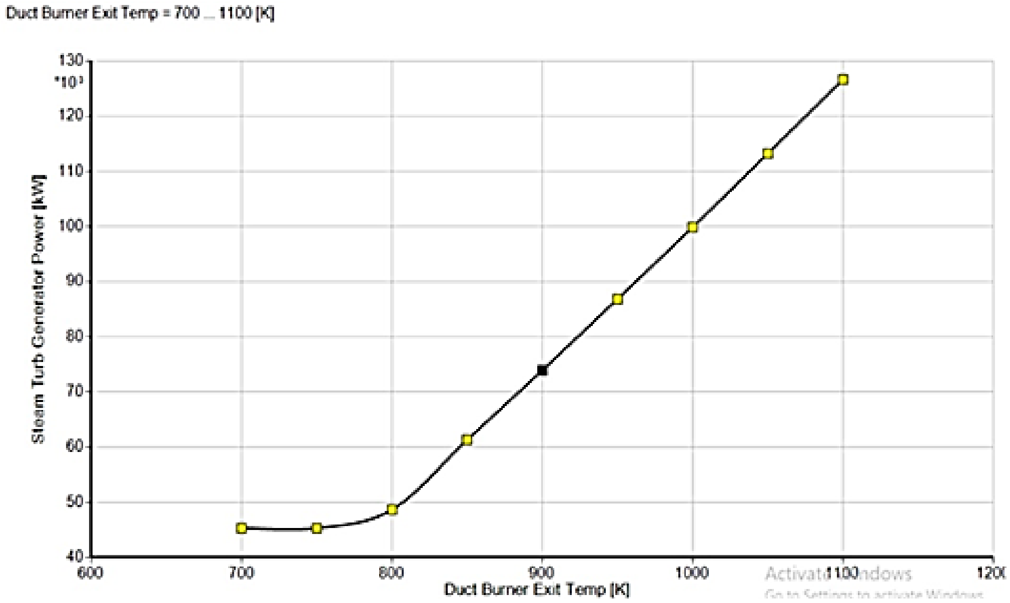

Figure 3: Power Output VS Burner Temperature

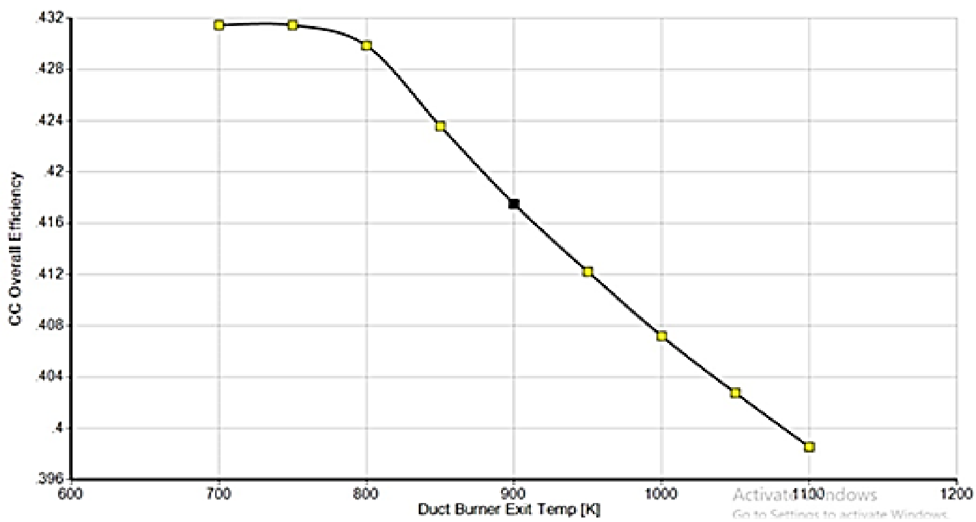

Figure 4: Impact of the Burner on the Overall Efficiency of the Combined Cycle System

As shown in Figure 4, the efficiency of the system varies with the duct burner exit temperature, ranging from 700 K to 1100 K. At lower burner exit temperatures (i.e., 700 K to around 800 K), the overall efficiency of the combined cycle system remains relatively stable, hovering around 43.2%. But then, as the burner exit temperature increases beyond 800 K, a noticeable decline in efficiency begins. The efficiency drops to around 41.8% when the temperature reaches 900 K. Between 900 K and 1100 K, the efficiency continues to decrease steadily. At the highest measured temperature of 1100 K, the efficiency falls to approximately 39.8%.

The SFC can be influenced by several factors, including burner exit temperature, fuel type, and overall system configuration. The addition of a supplementary burner can significantly impact SFC by altering the combustion process and energy output.

Figure 5 shows a clear downward trend, indicating that as the burner exit temperature increases, the PSFC decreases. At the lower end of the temperature range of around 1300 K, the PSFC is relatively high, and this suggests that more fuel is required to produce each unit of power when the burner exit temperature is lower. But then, as the burner exit temperature increases, the PSFC steadily decreases and this reduction continues consistently across the temperature range.

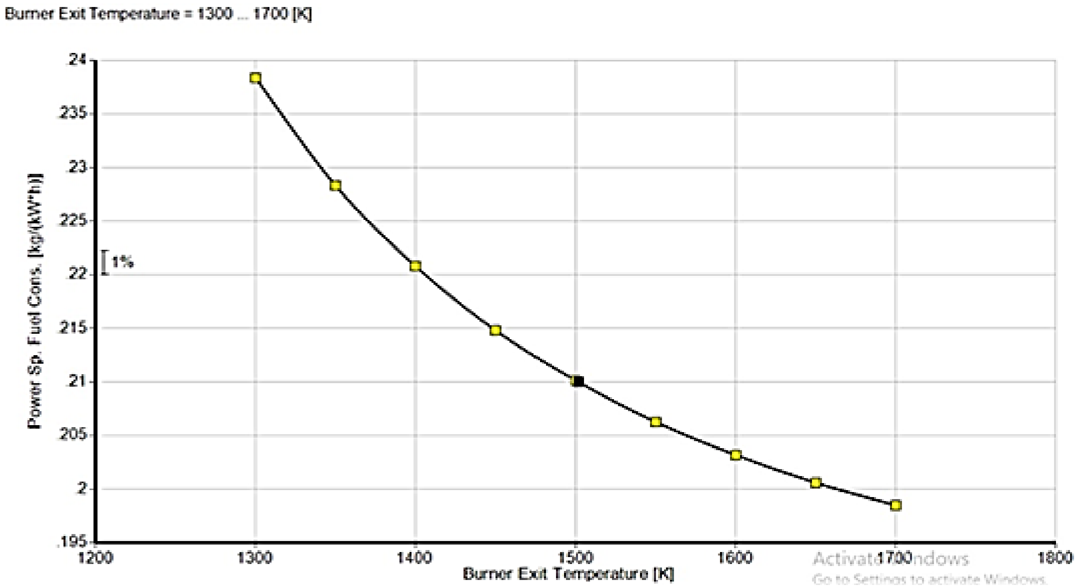

Figure 5: Plot of Power Specific Fuel Consumption vs Burner Exit Temperature

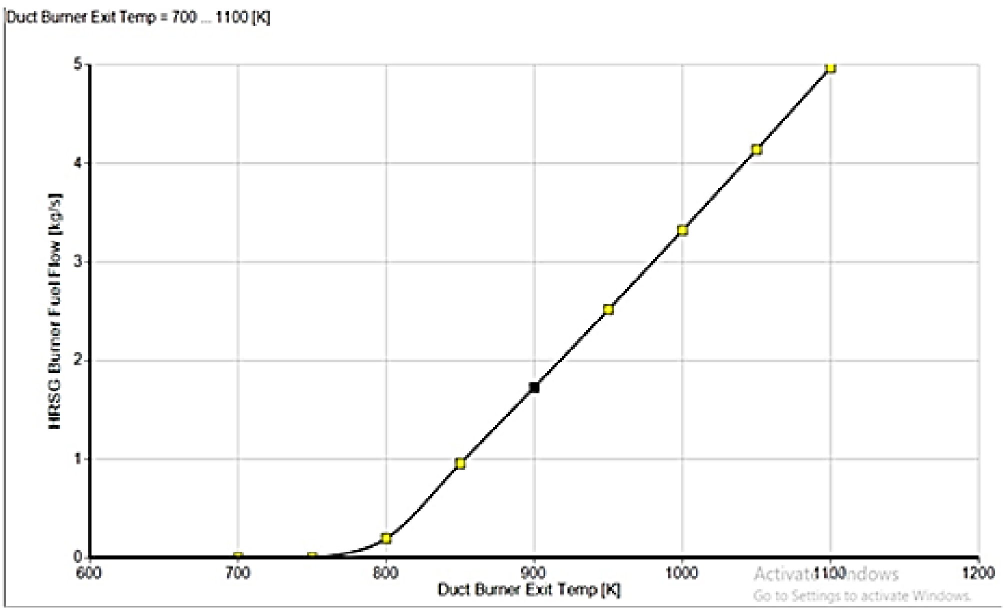

Figure 6: The Plot of HRSG Burner Fuel Flow and the Duct Burner Exit Temperature

In Figure 6, the HRSG burner fuel flow increases significantly with rising duct burner exit temperatures. At temperatures below 800 K, the HRSG burner fuel flow remains close to zero, indicating minimal fuel consumption. However, as the duct burner exit temperature surpasses 800 K, there is a noticeable increase in fuel flow, which becomes more pronounced as the temperature continues to rise. This trend suggests that higher exit temperatures necessitate greater fuel flow to maintain the desired energy output and efficiency of the system. Hence, we can infer that an increase in HRSG burner fuel flow corresponds to an increase in PSFC. Higher temperatures and fuel flow rates indicate that more fuel is being consumed to generate additional power, thus affecting the PSFC.

Table 3 presents a comparative analysis of the performance data obtained with and without the supplementary burner to help us understand the efficacy of the supplementary system in enhancing gas turbine power output and overall system performance. Key performance metrics such as thermal efficiency, net power output, and steam generation rates are compared to quantify the impact of the supplementary burner. The percentage increase in power output is also calculated to quantify the impact of the supplementary burner.

Table 3: Comparison between Performance Results with and Without Supplementary Burner

Parameter	Without Supplementary Burner	With Supplementary Burner	Percentage Increase (%)
Thermal Efficiency (%)	43.66	41.43	-5.11%
Steam Turbine Power Output (kW)	45233.1	75941.5	67.89%
Steam Generation Rate (kg/s)	65.984	93.838	42.21%
Steam Flow Rate (kg/s)	66.657	94.795	42.21%
Mass Flow Rate of Gas (kg/s)	645.167	630.74	-2.24%
Gas Turbine Power Output (kW)	180,092.80	180,092.80	0.00%
Net Power Output (kW)	225,325.90	256,034.30	13.63%

This comparative analysis highlights the substantial improvements in power output and steam generation with the use of the supplementary burner, despite the minor reduction in thermal efficiency. As already discussed in section 4.3, the net power output increased significantly by approximately 67.89% with the addition of the supplementary burner, while the thermal efficiency showed a slight decrease of 5.11%. Steam generation rates and steam flow rates both increased by about 42.22%, indicating enhanced steam production and utilization. The mass flow rate of gas showed a small decrease of 2.23%, suggesting that the increased power output is primarily due to the increased efficiency of fuel combustion and steam generation rather than an increase in gas flow.

IX. ECONOMIC ANALYSIS

The use of the supplementary burner results in a substantial increase in electrical power output, leading to additional revenue from electricity sales. However, it's essential to consider the additional costs associated with fuel consumption and maintenance to determine the overall cost-effectiveness of implementing the afterburner. The net financial gain is assessed to make informed decisions regarding its implementation in gas turbine systems for power augmentation, considering that the average residential electricity rate in the U.S. as of February 2023, is approximately 23 cents per kWh (EnergySage, 2023).

Hence, since the afterburner increases power output by 30,708.4 kW and the electricity rate is $0.23/kWh, the additional revenue per hour would be 30,708.4 kW x $0.23/kWh = $7,062.93 per hour. Cost-benefit analysis helped us evaluate whether the potential benefits of the increased power outweigh the costs of operating the afterburner. To estimate the increased fuel consumption associated with the operation of the afterburner in this research, we can use the Power Specific Fuel Consumption (PSFC) value of approximately 0.1789 kg/(kW•h) from the GasTurb14 output. The additional fuel consumption due to afterburner operation is estimated at 0.1789 kg/(kW•h) x 30,708.4 kW = 5493.73 kilograms per hour based on the specific fuel consumption rate of 0.1789 kg/(kWh) and the increased power output of 30,708.4 kW. According to GlobalAir.com (2023), typical Jet Fuel Price ranges between $0.75 - $1.00 per kilogram. Assuming a conservative fuel price of $0.75 per kilogram, the fuel cost per hour for the afterburner operation would be 5493.73 kg/h × $0.75/kg = $4,120.2995.

X. CONCLUSION

This publication undertakes a rigorous technical assessment aimed at enhancing the performance of the GE GT13E2 industrial gas turbine. The analysis encompasses two distinct scenarios: one involving the GT13E2 operating as a combined cycle system with a supplementary burner and the other as a standalone Combined cycle system. The specifications of the GT plant are comprehensively detailed in Table 1. The electrical power output increased by 67.89%, and the net power output of the system rose by 13.63%. This increase in power output was accompanied by an enhanced steam generation rate, which improved by 42.21%. Despite the notable improvements in power output and steam generation, the CC overall efficiency experienced a slight decrease from 43.66% to 41.43%. However, the decrease in efficiency is offset by the significant gains in power output and steam generation, making the trade-off acceptable from an operational perspective.

The specific fuel consumption (PSFC) demonstrated a positive trend, decreasing as the supplementary burner exit temperature increased. This indicates better fuel efficiency and improved overall system performance. The HRSG burner fuel flow also increased significantly with rising burner exit temperatures, highlighting the supplementary burner's contribution to the system's enhanced performance. The economic analysis underscored the financial viability of implementing the supplementary burner. The increased electrical power output translated into higher revenue generation, with an additional hourly revenue of $7,062.93. Although the increased fuel consumption for the supplementary burner operation incurred additional costs, the net financial gain remained substantial, supporting the economic justification for the integration of the supplementary burner.

ABBREVIATIONS:

BTU British Thermal Unit

CHP Combine Heat and Power

GasTurb Gas Turbine

HRSG Heat Recovery Steam Generator

ISO International Organization for Standardization

OPR Engine overall pressure ratio

OTSG Once Through Steam Generator

PSFC Power Specific Fuel Consumption

SFC Specific fuel consumption

SSFCC Sequential Supplementary Firing Combine Cycle

LHV Fuel heating value (KJ/Kg)

LCV low fuel calorific value (KJ/Kg)

NGCC Natural Gas Combine Cycle

SYMBOLS:

rp Pressure ratio

η Efficiency

W gas Mass flow rate of gas

h1 Heat content at compressor inlet

h2 Heat content at compressor outlet

h3 Heat content at the gas turbine inlet

h4 Heat content at the gas turbine exit

h5 Heat supplied to the Heat Recovery Steam Generator (HRSG)

h6 Heat content of the exhaust gas

h7 Heat value provided by the pump

h8 Heat supplied to the steam turbine

h9 Heat content at the steam turbine outle

h10 Heat content at the condenser outlet.

T1 Compressor inlet temperature (k)

T2 Compressor exit temperature (k)

T3 Turbine entry temperature (k)

T4 Turbine exit temperature (k)

CONFLICTS OF INTEREST

The authors declare that they have no conflicts of interest between them and with any third party.

REFERENCES

1. S. Ebigenibo and E. Ebieto, "Nigerian power sector: Why gas turbines will be relevant for the next 50 years," *Global J. Eng. Technol. Adv.*, vol. 5, 2020. Available from: https://doi.org/10.30574/gjeta.2020.5.1.0078
2. Baakeem, J. Orfi, S. Alaqel, and H. Al-Ansary, "Impact of ambient conditions of Arab Gulf countries on the performance of gas turbines using energy and exergy analysis," *Entropy*, vol. 19, no. 1, p. 32, 2017. Available from: https://doi.org/10.3390/e19010032
3. First Independent Power Ltd., "Plants." Available from: https://fipl-ng.com/plants/
4. A. González Díaz, E. Sánchez Fernández, J. Gibbins, and M. Lucquiaud, "Sequential supplementary firing in natural gas combined cycle with carbon capture: A technology option for Mexico for low-carbon electricity generation and CO2 enhanced oil recovery," *Int. J. Greenhouse Gas Control*, vol. 51, pp. 330-345, 2016. Available from: https://doi.org/10.1016/j.ijggc.2016.06.007
5. P. Kiameh, *Power Generation Handbook: Selection, Applications, Operation, and Maintenance*. New York, NY, USA: McGraw-Hill, 2003. Available from: https://cir.nii.ac.jp/crid/1130000794537154560
6. H. Li, M. Ditaranto, and J. Yan, "Carbon capture with low energy penalty: Supplementary fired natural gas combined cycles," *Appl. Energy*, vol. 97, pp. 164-169, 2012. Available from: https://doi.org/10.1016/j.apenergy.2011.12.034
7. R. Wylie, "Supercritical combined cycle for generating electric power," U.S. Patent 20040148941 A1, 2004. Available from: https://10.3390/en14112985
8. V. Ganapathy, *Heat-Recovery Steam Generators: Understand the Basics*. ABCO Industries, 1996. Available from: https://www.angelfire.com/md3/vganapathy/hrsgcep.pdf
9. A. Gonzalez Diaz, E. Sanchez, J. M. Gonzalez Santalo, J. Gibbins, and M. Lucquiaud, "On the integration of sequential supplementary firing in natural gas combined cycle for CO2-enhanced oil recovery: A techno-economic analysis for Mexico," *Energy Procedia*, vol. 63, pp. 7558–7567, 2014. Available from: https://doi.org/10.1016/j.egypro.2014.11.791
10. Siemens, "Introduction to the Complementary Fired Combined Cycle Power Plant," in *POWER-GEN International*, Orlando, FL, Nov. 28-30, 2006.
11. J. D. Enyia, A. E. Archibong, D. I. Igbong, and U. E. Eyod, "Economic viability of compressor washing methods for maximum power output," *Int. J. Mech. Eng. Appl.*, vol. 5, no. 4, pp. 227-238, 2017. Available from: http://dx.doi.org/10.1115/GT2017-64950
12. C. B. Meher-Homji, M. A. Chaker, and H. M. Motiwala, "Gas turbine performance deterioration,". Available from: https://oaktrust.library.tamu.edu/bitstream/handle/1969.1/163330/t30pg139.pdf
13. J. D. Enyia, D. I. Igbong, Y. Li, and I. Thank-God, "Industrial gas turbine on-line compressor washing for power generation," *Int. J. Eng. Res. Technol.*, vol. 4, no. 8, pp. 607-614, 2015. Available from: https://doi.org/10.1016/j.apenergy.2012.02.016
14. A. Fentaye, A. Baheta, and S. Gilani, "Effects of performance deterioration on gas path measurements in an industrial gas turbine," *ARPN J. Eng. Appl. Sci.*, vol. 11, pp. 14202-14207, 2016. Available from: https://doi.org/10.1016/j.applthermaleng.2022.119907
15. I. S. Diakunchak, "Performance deterioration in industrial gas turbines," *J. Eng. Gas Turbines Power*, vol. 114, no. 2, pp. 161-168, 1992. Available from: https://doi.org/10.1115/1.2906565
16. EnergySage, "How much does electricity cost in 2023?". Available from: https://www.energysage.com/local-data/electricity-cost/

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