

An Inquisition on the Combined Effects of Ambient Temperature and Relative Humidity on The Performance of a Uniform Speed Single Shaft Gas Turbine in Tropical Monsoon Climate, using GPAL

E. A. Abhulimen, M. U. Bonet., S. O. Oyekunle, N. Achara, W. C. Solomon

Abstract—This paper investigates the combined effects of Ambient Temperature and Relative Humidity on the performance of a uniform speed single shaft Gas Turbine, sited in Tropical Monsoon climate. A single shaft gas turbine simulator (known as GPAL) from Gas Path Analysis Ltd was employed. The City of Portharcourt, Nigeria, was chosen to represent the tropical monsoon climate, with its climatic data of monthly ambient temperature and relative humidity obtained from Koppen. With parameters like speed, reference power, inlet and exhaust losses kept constant, the ambient temperature and relative humidity were continually varied according to their climatic values. Each time, the performance of the gas turbine was simulated and parameters such as; Efficiency, Turbine power and Net power output, Turbine inlet Temperature and Exhaust Gas Temperature, as well as Specific fuel consumption were monitored. The environmental impact of the gas turbine was equally assessed in terms of Carbon (IV) Oxide (CO₂) emission in Tonnes/day and in Kg/MW hr, NOX emission and Carbon Monoxide (CO) emission. The results of the study indicate that it is most efficient and productive to operate the gas turbine in Portharcourt in the months of January and December whereas it is least efficient in the month of April. Whereas CO emission was relatively low and uniform throughout the year, the highest specific fuel consumption was recorded in April.

Index Terms—Efficiency, Emission, Gas Turbine, Humidity, Temperature.

I. INTRODUCTION

The gas turbine is widely used in industrial applications that require power. This power is employed in driving equipment such as pumps and process compressors or for electricity generation [1]. It is a continuous flow non-reciprocating internal combustion engine whose initial concept was meant for aircraft propulsion so as to overcome the drawbacks of its reciprocating piston counterpart in the aerospace industry. The development of gas turbine began not long before the Second World War with shaft power in mind, and later transferred to the turbojet engine for aircraft propulsion. Producing great amounts of energy for its size and weight, the gas turbine has found increasing service in more than 40 years and in conjunction with growth in materials technology and increase in compressor pressure

ratio, thermal efficiency has increased from 15% to 45% [2]. The gas turbine began to compete successfully in other fields only in the mid-1950s, but since then it has made a progressively greater impact in an increasing variety of applications [3]. The basic operation of gas turbine is a Brayton cycle with air as the working fluid. Fresh atmospheric air flows through the compressor that brings it to higher pressure. Energy is then added by spraying fuel into the air and igniting it so the combustion generates a high-temperature flow. This high temperature high-pressure gas enters a turbine, where it expands down to the exhaust pressure, producing a shaft work output in the process. Gas turbines are used to power aircraft, trains, ships, electrical generators, pumps, gas compressors and tanks. In an ideal gas turbine, gases undergo four thermodynamic processes: an adiabatic compression, isobaric combustion, adiabatic expansion and isobaric heat rejection. Together, these make up the Brayton cycle [4]. The gas turbine or combustion turbine burns a very lean mixture of fuel and compressed air. The gas passing through the turbine consists mainly of hot air plus some products of combustion. The gas turbine comprises three main components namely; the compressor, the combustion chamber (combustor) and the turbine. These three components are illustrated in Fig. 1. A gas turbine comprising these components is often referred to as a simple cycle gas turbine. Clearly, the power output from a gas turbine depends on the efficiency of these three components. The higher the efficiency of the components, the better will be the performance of the gas turbine, resulting in increased power output and thermal efficiency.

To begin the cycle, air is drawn in from atmosphere through an air filter into the compressor. As the air passes through the compressor, the pressure is raised. Where the pressure ratio is 10: 1, the pressure of the air as it leaves the compressor is 10Atmospheres, which is about 147 psia, 1035 kilopascals or 10.35bar. The temperature of the air will also rise as it is compressed [5]. If energy is added into the compressor discharge air, corresponding to the losses in the compressor and turbine, then the system will run but will not produce any net power output. To obtain net power from the gas turbine, there is need for additional supply of energy into the compressor discharge air. This is achieved by burning fuel in the compressor discharge air, in a combustion chamber or combustor. The combustor is located between the compressor and the turbine.

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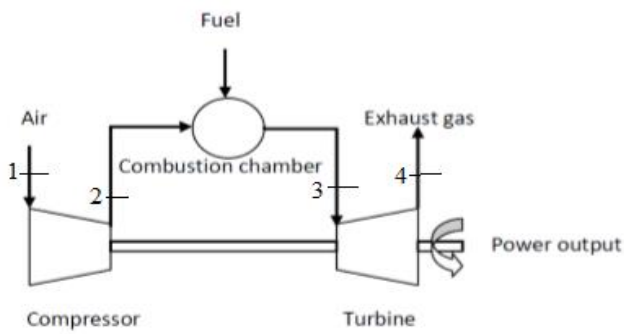


Fig. 1. Single shaft, simple cycle gas turbine

The turbine is the component that produces work to the surrounding. The turbine produces power by converting heat into work. Some fraction of that work is used by the compressor while the balance work is considered as the net work [6].

Various arrangements of the gas turbine components have evolved over the years. Whereas some are better suited for applications such as power generation, other layouts are more suited to mechanical drive applications.

A. The Simple Cycle Gas Turbine

The Brayton or the Joule cycle is the thermodynamic cycle governing the operation of the gas turbine. The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today, it is used for gas turbines where both the compression and expansion processes take place in rotating machinery [7]. The Temperature-Entropy (TS) diagram representation of Fig. 1 is shown in Fig. 2. Point 1 to point 2 begins the process, where air is drawn in and isentropically compressed. The compressed air is heated at constant pressure from point 2 to point 3. The flue gas is isentropically expanded from point 3 to point 4. Finally, the working fluid is exhausted at constant pressure to the atmosphere from point 4 back to point 1.

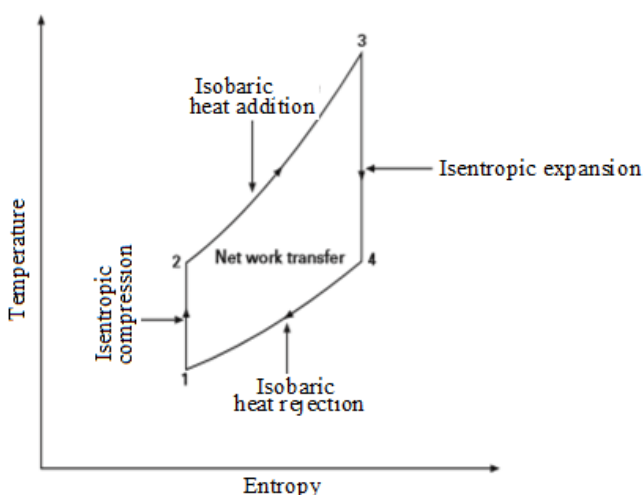


Fig. 2. T-S Diagram of an ideal simple cycle

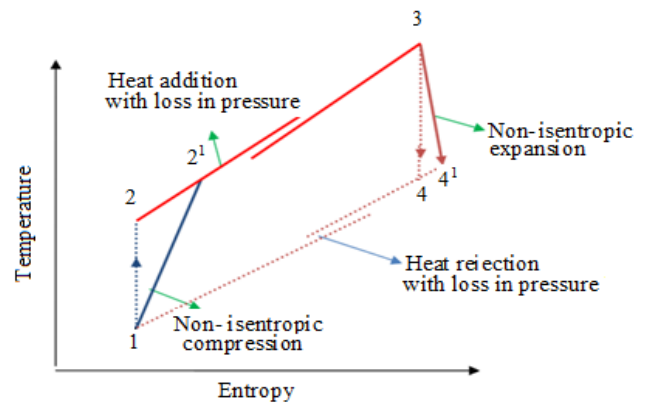


Fig. 3. T-S Diagram of a real simple cycle

In the ideal case (Fig. 2), the turbine work is assumed equal to the compressor work. Also, the processes in the compressor and the turbine are assumed to be isentropic. In practice however, the compression process and the expansion process always increase their entropy along the flow path due to the various losses inside the machines. Also, the processes from point 2 to point 3 and from point 4 to point 1 above, experience pressure drops along the flow path due to losses. In the analysis of actual cycle therefore, the irreversibility associated with these devices is considered. The effect of the irreversibility is a reduction in the power that can be obtained from the engine. Hence, the overall performance of the gas turbine highly deviates from the ideal cycle [8] as in Fig. 3.

B. Ambient Conditions

Ambient conditions refer to a set of parameters used by designers. These conditions are also considered when systems are commissioned and tested. A key factor that affects the performance of the gas turbine is the condition of inlet air, typically air temperature, humidity and pressure since the density of air depends on its temperature, its pressure and how much water vapour it contains [9].

C. Ambient Pressure

The ambient pressure on an object is the pressure of the surrounding medium, such as a gas or liquid, in contact with the object [10]. The ambient pressure may change quite significantly at a given elevation. Within the atmosphere, the ambient pressure decreases as height increases [11]. At an elevation of 1000 metres for instance, the ambient pressure would be about 0.9 Bar on an International Standard Atmosphere (ISA) day. However, the ambient temperature at this altitude will be lower, thus partly compensating for the effect of this pressure variation. To investigate the impact of ambient pressure changes on engine performance, Razak A.M.Y., argued that the ambient temperature will be assumed to remain constant at 15 degrees Celsius. This will result in the engine power output being limited by the exhaust gas temperature (EGT) limit rather than by the speed limits from the gas turbine. The temperature rises across the combustion system and the combustion inlet temperature would also remain constant during this ambient pressure transient. The compressor and turbine pressure ratios would be constant during the decrease in the ambient pressure, and any fall in ambient pressure will result in a decrease in compressor discharge

pressure. The decrease in the compressor discharge pressure will be directly proportional to the ambient pressure [6].

In this research, attention was rather given to the variations in ambient temperature and relative humidity.

D. Ambient Temperature

The gas turbine performance is affected by a variation in the density and or mass flow of the air intake to the compressor. The air density is a function of ambient temperature, pressure and humidity [12]. Ambient temperature refers to the temperature of any object or environment where equipment is situated. It relates to the immediate surroundings, sometimes referred to as the ordinary temperature or the baseline temperature. Its value is important for system design and thermal analysis. In the gas turbine, ambient temperature represents the compressor inlet temperature. If the temperature drops, the air density will increase & hence heavier air will be compressed by the compressor which will increase mass flow rate thereby increasing the compressor work requirements. As heavy compressed air enters the turbine section, it creates extra expansion & more work from turbine will be produced.

As compressor inlet temperature drops, air density increases. This will cause the compressor work as well as fuel mass flow to the turbine to increase to handle the increase in air mass flow, resulting in extra turbine work.[13]

E. Relative Humidity

The humidity of air is the amount of water vapour needed to saturate the air and is referred to as relative humidity. The relative humidity is related to specific humidity in that, an increase in relative humidity brings about increase in specific humidity. However, the increase in specific humidity and the change in the mass of vapour in the air are small at low ambient temperature. At high ambient temperatures, relative humidity will have a noticeable effect on the thermodynamic properties of air and products of combustion. At any given ambient temperature and pressure, an increase in relative humidity increases the specific humidity thus resulting in an increase in the gas constant, R , and specific heat at constant pressure, C_p , for air while decreasing its isentropic index, γ [6]. Hence influencing the engine performance [14].

It is most times hard to believe that humid air is lighter, or less dense, than dry air. One would understandably wonder how air becomes lighter if water vapour is added to it. According to one of the laws of nature by the Italian

physicist Amadeo Avogadro in the early 1800s, a fixed volume of gas, say one cubic meter, at the same temperature and pressure, would always have the same number of molecules no matter what gas is in the container. So, in a cubic meter of perfectly dry air, there is about 78% nitrogen molecules, with a molecular weight of 28 (2 atoms with atomic weight 14) each. Another 21% of the air is oxygen, with each molecule having a weight of 32 (2 atoms with atomic weight 16). The final one percent is a mixture of other gases [15]–[17].

Since molecules are free to move in and out of the cubic meter of air, it is logical to conclude that if water vapour molecules are added, some of the nitrogen and oxygen molecules would leave. Remember that the total number of molecules in the cubic meter of air remains the same. The water molecules, which replace nitrogen or oxygen, have a molecular weight of 18 (One oxygen atom with atomic weight of 16, and two hydrogen atoms each with atomic weight of 1). This is lighter than both nitrogen and oxygen. Therefore, replacing nitrogen and oxygen with water vapour, decreases the weight of the air in the cubic meter; bringing about a reduction in density. However, liquid water is heavier or denser, than air. But, the water that makes the air humid is not liquid but water vapor, which is gaseous and lighter than nitrogen or oxygen.

As air density decreases with increase in humidity, compressor mass flow also decreases. But an increase in compressor mass flow due to a decrease in relative humidity, leads to increase in turbine power output.

II. METHODOLOGY

The methodology of this research involves the collection of climatic data on tropical monsoon climate, represented by Portharcourt, Nigeria. The primary parameters considered were the monthly ambient temperature and relative humidity. The data were obtained from the Köppen climate classification which is one of the most widely used climate classification systems [18]. It was first published by the German-Russian climatologist Wladimir Köppen in 1884, with several later modifications by Köppen, notably in 1918 and 1936. Later, the climatologist Rudolf Geiger introduced some changes to the classification system, which is sometimes called the Köppen–Geiger climate classification system [19], [20]. The monthly ambient temperature and relative humidity for Portharcourt are shown in Table I and Fig. 4 respectively.

TABLE I: MONTHLY AMBIENT TEMPERATURE IN PORTHARCOURT [21]

| | January | February | March | April | May | June | July | August | September | October | November | December |
|------------------------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|-------------|
| Min. Temperature (°C) | 21.6 | 23 | 22.8 | 23.1 | 22.5 | 22.1 | 21.8 | 21.9 | 21.9 | 22.1 | 22.5 | 22.1 |
| Max. Temperature (°C) | 31.3 | 32.3 | 31.5 | 31.9 | 31.2 | 30.3 | 29 | 28.5 | 29.4 | 29.7 | 25.6 | 30.8 |
| Avg. Temperature (°C) | 26.4 | 27.6 | 27.1 | 27.5 | 26.8 | 26.2 | 25.4 | 25.2 | 25.6 | 25.9 | 26.6 | 26.4 |

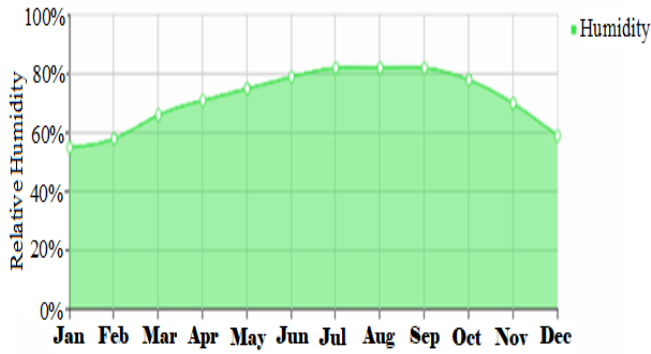


Fig. 4. Monthly Relative Humidity [22]

With the above information as input into the Gas Path Analysis software (GPAL), the performance of a single shaft, uniform speed gas turbine was analyzed in terms of efficiency, turbine power as well as net power output. The turbine inlet temperature, the exhaust Gas temperature and the Specific fuel consumption were modeled. Emissions of Carbon IV Oxide (CO₂), in Tonnes/day and in Kg/MW_{hr}, Oxides of Nitrogen (NO_x) and Carbon Monoxide (CO) were equally simulated.

III. GAS PATH ANALYSIS SIMULATION

The simulator was used to investigate the combined effects of changes in ambient temperature and relative humidity on engine performance. These values were continually varied in line with the monthly climatic data. The simulation was based on a quasi-steady-state model, making it possible to subject the model to significant changes in ambient conditions. The strategy changes the fuel flow to control the power output while the VIGV is modulated. The exhaust gas temperature (EGT) limit which could be fixed or varied with the change in ambient temperature and increased linearly with ambient temperature was set for variable temperature control. The Maximum Power Limit, representing the maximum power output from the gas turbine and ranges between 1 and 50 MW, was set to 45 MW. The ambient pressure was given a default value of 1.013 bar (14.696 PSI).

Other inputs include compressor and turbine fouling fault indices and efficiency fault indices. These indices represent changes in the components' flow characteristic as well as efficiency characteristics as a percentage of the design characteristics. These were set to "no fault". The inlet and exhaust losses which range from 0 to 200 mm water gauge (between 0 and 7 inches' water gauge), were set to 100mm water gauge (4 inches' water gauge) respectively. The simulator with a provision for Fuel type selection corresponding to Natural gas, Methane and Diesel (liquid fuel), was set to Natural gas. The Ramp time, which is the specification of the period for the above inputs to take place (with the exception of fuel gas selection, which takes place instantly), was given a value of 30 seconds.

IV. GAS TURBINE SYSTEMS THEORY

The gas turbine cycle is best depicted by the Brayton Cycle, with characteristics of the operating cycle shown in Fig. 2 above. It is a continuous flow machine, best described

by the first law of thermodynamics as;

$$W ({}_1Q_2) \equiv W (h_2 - h_1 + KE_2 - KE_1 + PE_2 - PE_1 + \frac{{}_1W_2}{J}) \quad (1)$$

Where;

W* = Mass flow rate, lbm/sec

{}_1Q_2 = Heat transferred to or from the system, Btu/lbm

h₂ = enthalpy of the fluid leaving, Btu/lbm

h₁ = enthalpy of the fluid entering, Btu/lbm

KE₂ = kinetic energy of the fluid leaving, Btu/lbm

KE₁ = kinetic energy of the fluid entering, Btu/lbm

PE₂ = potential energy of the fluid leaving, Btu/lbm

PE₁ = potential energy of the fluid entering, Btu/lbm

{}_1W_2 = Work per unit mass on or by the system, ft-lbf/lbm

J = Ratio of work unit to heat unit, 778.2 ft lbf/Btu

{}_1W_2/J = Work, Btu/lbm

In most gas turbine applications, the numerical magnitude of the difference in potential energy is so small, relative to the other values in (1) above, and can hence be disregarded. Therefore, (1) can be rewritten as:

$$W ({}_1Q_2) \equiv W (h_2 - h_1 + KE_2 - KE_1 + {}_1W_2/J) \quad (2)$$

For adiabatic processes, (no heat transfer).

$$\therefore {}_1Q_2 = 0. \quad (3)$$

$$W (\Delta h + \Delta KE + {}_1W_2/J) = 0 \quad (4)$$

$${}_1W_2/J = W (h_2 - h_1 + KE_2 - KE_1) = Power \quad (5)$$

Ideally, both the compressor and the turbine are isentropic devices. Hence we consider the isentropic relations of these components developed by ideal gas;

For the Compressor:

$$Q - W_c = \Delta h + \Delta KE + \Delta PE \quad (6)$$

But, Q, ΔKE and ΔPE = 0 (no heat transferred and no change in either the kinetic or potential energy)

$$\therefore -W_c = \Delta h \quad (7)$$

But Δh = mC_pΔT = C_p(T₂-T₁), with mass flow (m) being constant.

$$\therefore W_{1,2} = C_p(T_2 - T_1) \quad (8)$$

For Isentropic compression;

$$\Delta S = 0 \equiv T_2 = T_1 \left[\frac{p_2}{p_1} \right]^{\frac{\gamma-1}{\gamma}} \quad (9)$$

For the turbine:

Applying (7) and (8) to Fig. 2;

$$-W_t = \Delta h = C_p \Delta T = C_p(T_4 - T_3)$$

$$\therefore W_{3,4} = C_p(T_3 - T_4) \quad (10)$$

Similarly, from (9);

$$\frac{T_4}{T_3} = \left[\frac{p_4}{p_3} \right]^{\frac{\gamma-1}{\gamma}} \equiv P_4 = P_3 \left[\frac{T_4}{T_3} \right]^{\frac{\gamma}{\gamma-1}} \quad (11)$$

Where;

γ is the ratio of specific heats of the gas (C_p/C_v) and it is known as the isentropic index,

C_p is the specific heat of the gas at constant pressure

T_1 to T_4 and P_1 to P_4 represent the temperature and pressure at various points in Fig. 1. Since the work done in the combustor is zero; (6) can be rewritten as;

$$Q = \Delta h \tag{12}$$

$$\therefore Q_{2,3} = C_p(T_3 - T_2) \tag{13}$$

The net work done by the cycle per unit mass flow rate (specific work, W_{net}) is the difference between the expansion and compression work.

$$\therefore W_{net} = C_p(T_3 - T_4) - C_p(T_2 - T_1) \tag{14}$$

A. Components Efficiency

The efficiency of individual components contributes to the overall efficiency of the gas turbine. Examining the efficiency of each component is a necessary tool in isolating engine problems

For the compressor; the efficiency (η_c) is directly proportional to the compressor pressure ratio and inversely proportional to the compressor discharge temperature as in (15) below.

$$\eta_c = \frac{\frac{R_c^\sigma - 1}{T_2}}{\frac{R_c^\sigma - 1}{T_1} - 1} \tag{15}$$

Where;

$$\sigma = \frac{k-1}{k}, \tag{16}$$

R_c = Compressor pressure ratio, P_2/P_1

P_2 = Compressor total discharge pressure, psia

P_1 = Compressor total inlet pressure, psia

k = Ratio of specific heats, C_p/C_v

C_p = Specific heat at constant pressure, Btu/lb

C_v = Specific heat at constant volume, Btu/lb

T_2 = Compressor total discharge temperature, °C

T_1 = Compressor total inlet temperature, °C

For the turbine;

$$\text{Efficiency } \eta_t = \frac{1 - \frac{T_4}{T_3}}{1 - \frac{T_4}{T_3^c}} \tag{17}$$

Where;

T_4 = Turbine exhaust temperature, °C

T_3 = Turbine inlet temperature, TIT °C

R_t^c = Turbine total inlet pressure/Turbine total exhaust pressure

Similarly, The cycle thermal efficiency, η_{th} , is defined as the ratio of the net work done and the heat input. Hence the thermal efficiency is given by (18).

$$\eta_{th} = \frac{W_{net}}{Q_{2,3}} \tag{18}$$

Substituting (13) and (14) into (18);

$$\eta_{th} = \frac{C_p(T_3 - T_4) - C_p(T_2 - T_1)}{C_p(T_3 - T_2)} \tag{19}$$

$$\therefore \eta_{th} = \frac{(T_3 - T_2) - (T_4 - T_1)}{(T_3 - T_2)} \tag{20}$$

$$\eta_{th} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

By substituting (9) and (11) into (20);

$$\eta_{th} = 1 - \frac{T_1}{T_2} \tag{21}$$

Hence, the ideal gas turbine thermal efficiency is dependent only on the compressor pressure ratio but less than the Carnot efficiency, since T_2 is less than T_3 .

Expressing (21) in terms of compressor pressure ratio using (9) gives;

$$\eta_{th} = 1 - \frac{1}{c} \tag{22}$$

Where $c = \left[\frac{P_2}{P_1} \right]^\frac{\gamma-1}{\gamma}$

This implies that thermal efficiency will increase with increase in pressure ratio and maximum possible thermal efficiency is achieved when T_2 tends to T_3 and will be zero as the pressure ratio tends to 1. The specific work given by (14) can then be rewritten as;

$$W_{net} = C_p T_1 \left(\frac{c-1}{c} \right) \left(\frac{T_3}{T_1} - c \right) \tag{23}$$

Thus, for a given gas, the specific work of the ideal gas turbine cycle depends on the compressor pressure ratio P_2/P_1 , the maximum and minimum temperature ratio T_3/T_1 and the compressor inlet temperature T_1 . As such, increasing the ratio T_3/T_1 will increase the specific work, whereas increasing pressure will lead to an increase in the specific work initially but will later decrease at high pressure ratios [23].

V. RESULTS AND DISCUSSION

The gas path analysis (gpal) software was used in harmony with (1) to (23) to establish the behaviours of thermal efficiency, power output, air flow and fuel flow with variation in pressure ratio. It was equally used to model the combined effects of variation in ambient temperature and relative humidity with inputs from Table I and Fig. 4 on; thermal efficiency, turbine power, net power output, the specific fuel consumption and emissions.

A. Pressure ratio variation

The performance of the gas turbine using a maximum cycle temperature of 1395k was analyzed. It was established that the thermal efficiency of a single shaft uniform speed, simple cycle gas turbine is dependent on the pressure ratio as shown in Fig. 5. The thermal efficiency increased with increase in pressure ratio. At maximum possible thermal efficiency, when t_2 tends to t_3 , the thermal efficiency would be zero as the pressure ratio tends to 1. This however was not achieved based on the turbine inlet temperature involved. According to Razak A. M. Y[6], the maximum thermal efficiency, when t_3 equals 900k, occurs at a pressure ratio of

about 8 and at a maximum cycle temperature of about 1400k, the maximum thermal efficiency would occur at a pressure ratio greater than 14.

The specific work of the single shaft uniform speed, simple cycle gas turbine was found to depend on the compressor pressure ratio, p_2/p_1 as shown in Fig. 6. the increase in compressor pressure ratio caused an increase in power output provided the design compressor pressure ratio was below the maximum cycle specific work. when the compressor pressure ratio equals unity, the specific work, w_{net} , would be zero. when the compressor pressure ratio is increased such that $c = (p_2/p_1)(\gamma - 1)/\gamma$, which is equal to t_3/t_1 from (22), the specific work would again reduce to zero. thus, the maximum specific work occurs at some pressure ratio between these values, and this optimum pressure ratio would depend on γ , t_1 and t_3/t_1 .

The non-dimensional mass flow and speeds are relative to design. the non-dimensional mass flows increased with pressure ratio shown in Fig. 7 and beyond a certain pressure ratio the mach number inside the aerofoil would reach unity and this would restrict the amount of non-dimensional flow that could pass through the turbine. increase in pressure ratio was found to increase the air flow and correspondingly, the fuel flow.

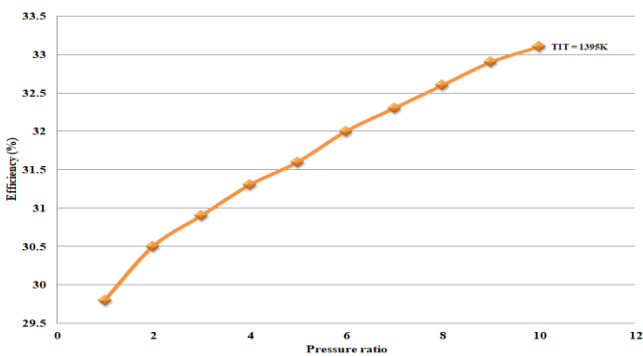


Fig. 5. Effects of Pressure ratio on Efficiency

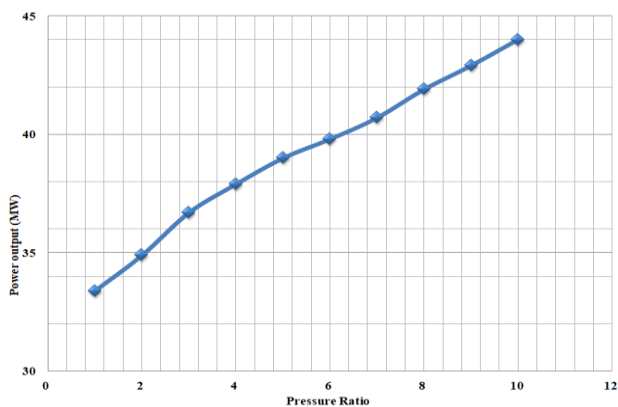


Fig. 6. Effects of Pressure ratio on Power output

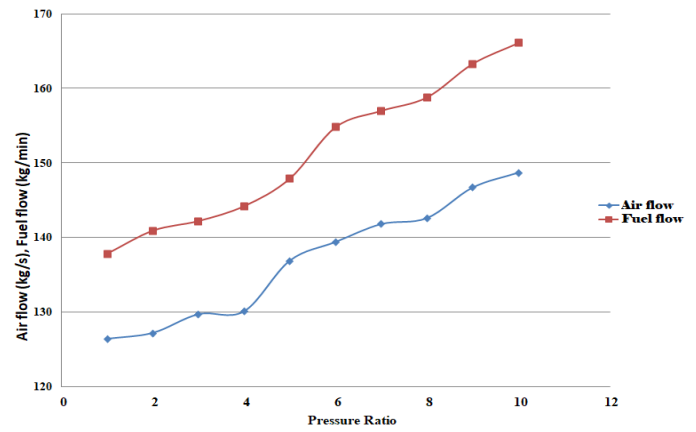


Fig. 7. Effects of Pressure ratio on Air flow and Fuel flow

B. Ambient Conditions Variation

The monthly performance evaluation of a single shaft uniform speed gas turbine sited in Portharcourt, monsoon climatic zone is presented in Fig. 8 to Fig 15.

C. Efficiency

Fig. 8 shows the monthly variation in efficiency as a result of variations in the ambient temperature and relative humidity. It was found that the gas turbine fared best in terms of efficiency, in the month of January. A better efficiency was recorded in the months of December, August, June, March May and September. The study also revealed that it is least efficient to operate a single shaft uniform speed gas turbine in Portharcourt, in the month of April.

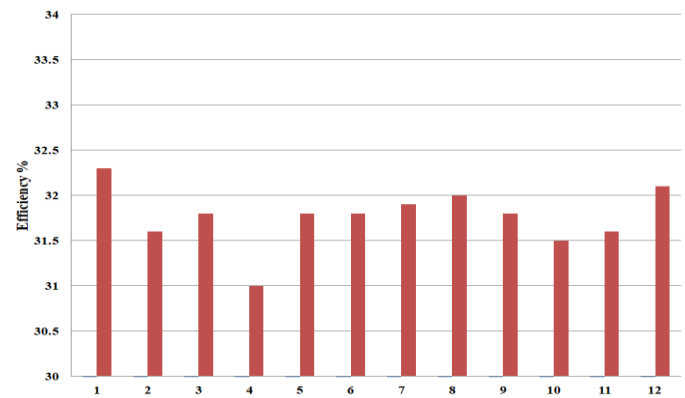


Fig. 8. Effects of ambient temperature and humidity on Thermal Efficiency

D. Power Output

The power output from the turbine as well as the net power output is presented in Fig. 9.

The difference between the turbine work output and the net work output suggests the work done on the compressor. For a uniform monthly load setting of 45MW, the unit was only able to deliver an average of 35MW in the month of April, based on the conditions under investigation. The months of January, December, August, June, July, March and September respectively have relatively high power outputs. The decline in output was found to be in the order of April, October and November respectively

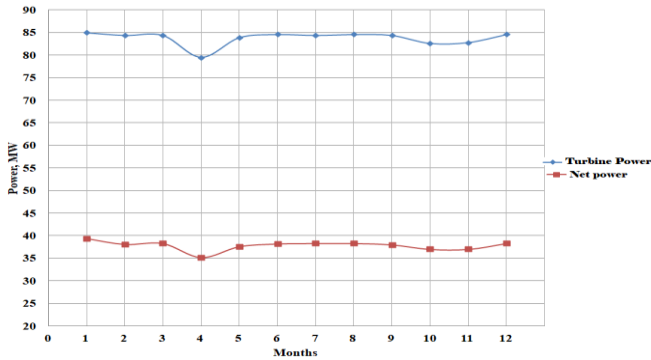


Fig. 9. Effects of ambient temperature and humidity on Power Output.

E. Specific Fuel Consumption

The specific fuel consumption (SFC) is an alternative means of determining the heat input per unit of work done. In line with the conditions investigated, the result of the specific fuel consumption is displayed in Fig. 10. This was found to be relatively high in the months of April, July, October and November respectively. Whereas the months of August, March, May and June had a relatively low SFC, the month of January and December were most favoured.

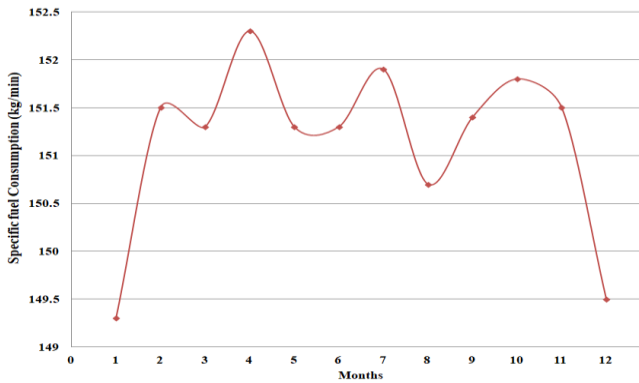


Fig. 10. Effects of ambient temperature and humidity on SFC

F. Temperature

Fig. 11 shows the result of the turbine inlet temperature and the exhaust gas temperature. Based on the conditions under investigation, the turbine inlet temperature was found to be lowest in the month of June and highest in the months of March, April, October and November respectively at about 1400k. The exhaust gas temperature was found to be fairly uniform with slight drop in the month of June.

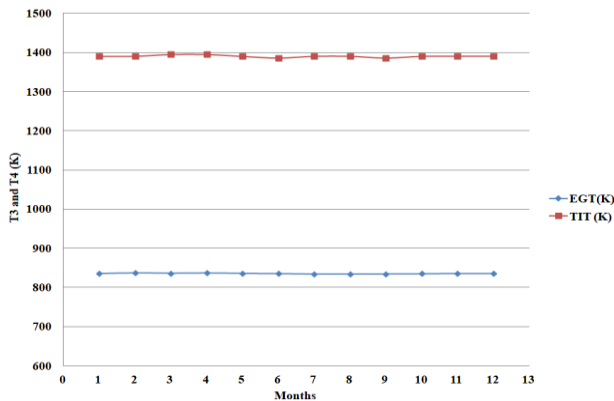


Fig. 11. Effects of ambient temperature and humidity on TIT and EGT

G. Emissions

The combustion system that uses hydrocarbon fuel, produces carbon IV oxide (CO₂) and water vapour (H₂O) as a result of carbon and hydrogen oxidation. CO₂ and H₂O may be considered as non-toxic but they are greenhouse gases and have been associated with global warming. All combustion systems including those in the gas turbine produce pollutants like oxides of nitrogen (NO_x) and carbon monoxide (CO) as well. In this research, the environmental impact of variation in ambient temperature and relative humidity on the gas turbine was investigated. From the results, efficiency has an inverse bearing on CO₂ emissions as the increase in thermal efficiency results in a decrease in the CO₂ emission in Tonnes/day and was found to be highest in the month of April (with the lowest thermal efficiency) as shown in Fig. 12. Fig. 13 shows the trend in CO₂ emission in kg/MWh, being highest in the months of April, October and July respectively whereas the emission was low in the months of January, December and September respectively. This is not unconnected with the specific fuel consumption as CO₂ emission increases with increase in the specific fuel consumption.

The trend in NO_x emissions is in line with the TIT. At higher turbine inlet temperatures, being in the months of March and April, NO_x emission increased significantly as shown in Fig. 14.

The formation of CO is generally due to poor combustion efficiencies. When an engine operates at a lower load than the design value, it is possible to increase the combustion temperature helping to maintain CO emissions, which tend to increase due to lower combustion pressure. The research was conducted at 45MW for a gas turbine rated 50MW and the CO emission was found to be fairly low and uniform as shown in Fig. 15.

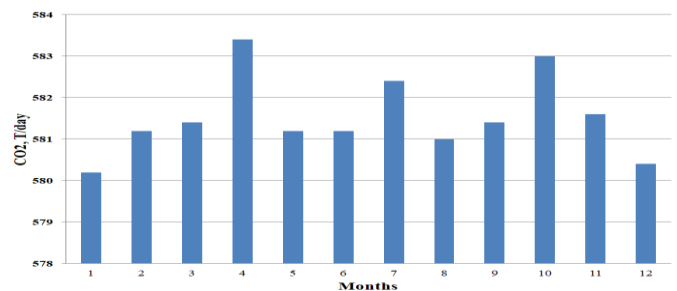


Fig. 12. Effects of ambient temperature and humidity on CO₂ emission in Tonnes/day

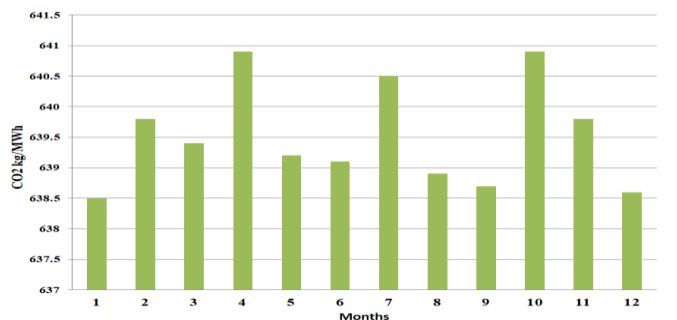


Fig. 13. Effects of ambient temperature and humidity on CO₂ emission in Kg/MWh

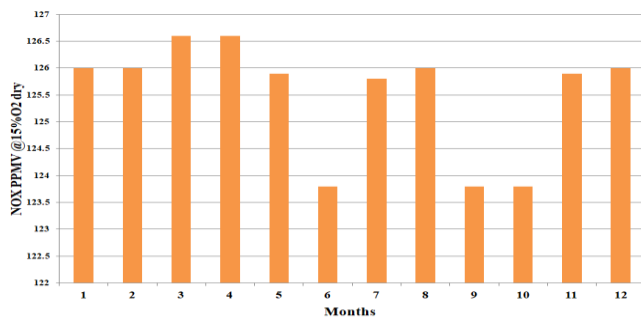


Fig. 14. Effects of ambient temperature and humidity on NO_x emission

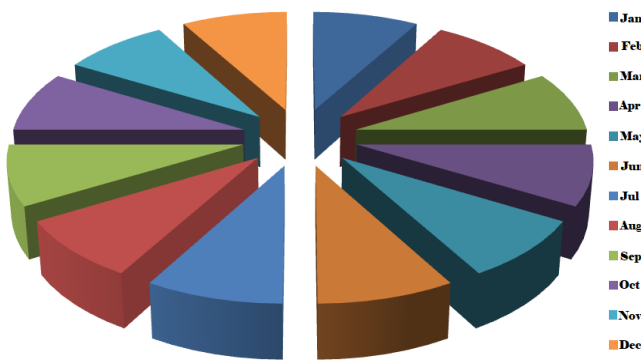


Fig. 15. Effects of ambient temperature and humidity on CO emission

VI. CONCLUSION

The result shows that GPAL could successfully be used to model the operation of a gas turbine, having established the power output and efficiency relationship with the pressure ratio in line with known theories. It equally reveals that, though variation in ambient temperature has a varying impact on the Performance of the gas turbine, the trend could be uttered by the position of ambient relative humidity.

The months of January and December were found to be the most efficient and productive months in the life of a single shaft uniform speed gas turbine sited in Portharcourt whereas the month of April was found to be least efficient and productive. It is therefore advisable that a plant manager of a single shaft uniform speed gas turbine, sited in Portharcourt, defers annual planned maintenance to the month of April, especially if emissions are taxed as this will further reduce the operating costs.

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