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# Techno-economic and Performance Analysis of Associated Gas Utilization in Gas Turbines

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## ABSTRACT

*This study focuses on the economic viability of using associated gas (A-gas) as a fuel in gas turbines for power generation applications. GASTURB simulation software was employed in modelling the natural gas and associated gas. Also, it was used for simulating the performance of the gas turbine using the two fuels. When natural gas and A-gas were compared for clean condition, the heat rate of natural gas was found to be 9923 kJ/kWh as against 9974kJ /kWh for an A-gas Fuel. Also, the plots of clean and degraded conditions for natural gas showed that heat rate increased from a clean case of 9923 kJ/kWh to 10178 kJ/kWh for a degraded condition. Techno-economic analysis conducted showed that the annual cost saving for utilizing the A-gas, when obtained at no cost is about $14.1million over the annual cost of natural gas.*

**KEYWORDS:** Gas turbine, Performance analysis, Associated gas, Natural gas, Techno-economic analysis.

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## 1. INTRODUCTION

In the past, gas flaring was used to routinely dispose of flammable gases that were either unusable or uneconomical to recover. However, modern technology has introduced ways and means of harnessing associated gas (A-gas) for very productive uses. It is estimated that about 168 BCM (Billion Cubic Meters) of natural gas (NG) is flared yearly worldwide (equivalent to about 400 million tons of carbon dioxide). Nigeria accounts for 23 BCM, the biggest after Russia; about 13% of global flaring is attributed to originate from Nigeria (Anosike, 2010). About 1000 standard cubic feet (SCF) of A-gas is produced in Nigeria with each barrel of oil. Hence oil production of 2.5 million bpd amounts to about 2.5 billion SCF of A-gas produced daily (Igbatayo, 2007). This amounts to an annual financial loss of about $2.5 billion (Ogbe *et al*., 2011). Sonibare and Akeredolu (2007) showed that, of the total NG production in Nigeria, about 17% is re-injected, 33% used commercially and 50% flared (equivalent to about 75% of total A-gas produced.

A study carried out for the Bureau of Public Enterprises of Nigeria estimated that each year the country loses between US$500 million and US$2.5 billion to gas flaring. Experts believe Nigeria is burning billions of Dollars from its oil wells and letting potential profits go up in flames. The massive amount of NG flared annually is an enormous economic waste and gives off greenhouse gas emissions, causes air pollution, have health implications and results in acid rain. By using the gas for energy, instead of flaring, much of the acute power needs in Nigeria would be fulfilled. Nigeria is in need of extra power generation and the gas that is being burned could go a long way towards providing the electricity that the country so desperately needs.

Gas turbines (GTs) burn NG, whether clean or impure, to produce power. Impurities have effects and cause the LHV (Lower Heating Value) of one fuel to differ from that of another. The impurities initiate the process of degradation of the GT or components along the hot gas path. The peak energy demand forecast for Nigeria is 10200 MW, but the current generation capability is 5157 MW. The highest generation recorded as at April 2012 stood at 3462 MW while the lowest generation recorded was 2444 MW (Allison, 2014). Allison (2014) revealed that though the generation capability of most Power plants in Nigeria is much more than the actual generation; unutilized generation capability is almost equal to the actual generation. The unutilized electricity generation capability of existing gas stations was attributed to gas shortages. This underscores the need for efforts to harness A-gas so as to achieve the full generation capability. However, there performance and degradation of hot components implications of utilization Associated gas in gas turbines for power generation purposes. This study aims to present the economic benefits of utilizing the associated gas in gas turbines.

## 2. MATERIALS AND METHODS

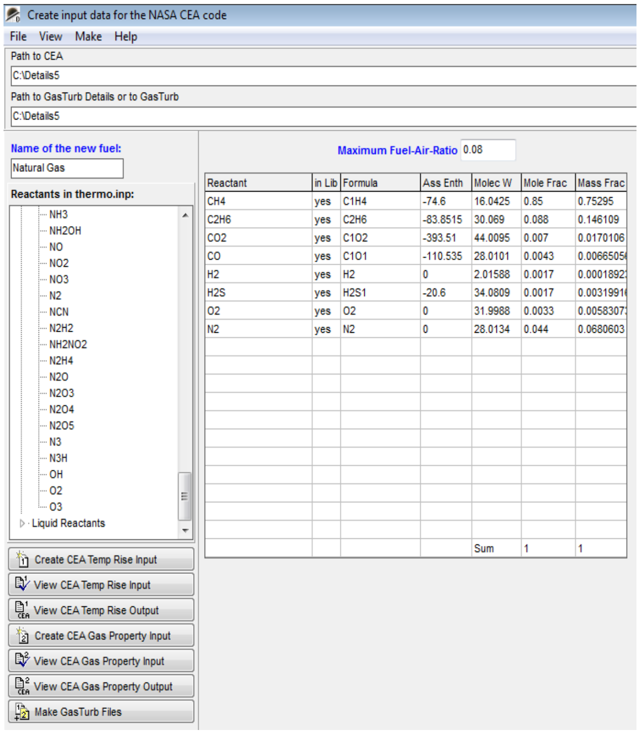
In modelling the various fuels, the GASTURB details 5.1 software was employed. This software is capable of modelling the different kinds of fuels ranging from gaseous to liquid fuels. Data obtained from field observation were employed, to ascertain various chemical composition and volume of compounds which constitute the natural gas and associated gas (A-gas) fuels under investigation. Tables 1 and 2 show the various chemical compositions and volumes of compound, which constitute the natural gas and A-gas fuels. These data were utilized to model the corresponding fuels using the GASTURB details. Figures 1 and 2 show the implanted field data on the GASTURB 5.1 details simulation software interface when modelling the natural gas and A-gas fuels respectively. Consequent upon obtaining the fuel composition/mixture from field data presented in Table 1, the steps taken to model the fuels in the GASTURB details 5.1 interface (see Figure 1) are provided:

1- Enter a name for the new fuel
2 - Enter the fuel composition
3 - Enter the path to FCEA2.exe
4 - Enter the path to GASTURB
5 - Create CEA temp rise input
6 - Run FCEA2 with that input
7 - Create CEA gas prop input
8 - Run FCEA2 with that input
9 - Make GASTURB files

Consequently, the fuel is created in the GASTURB details 5.1 and this is then exported to the GASTURB 11, to run the performance simulations.

**Table 1: Natural gas fuel composition**

| Compound | Formula | Vol. % |
|---|---|---|
| Methane | $CH_4$ | 85 |
| Ethane | $C_2H_6$ | 8.8 |
| Carbon dioxide | $CO_2$ | 0,7 |
| Carbon monoxide | $CO$ | 0.43 |
| Hydrogen | $H$ | 0.17 |
| Hydrogen Sulphide | $H_2S$ | 0.17 |
| Oxygen | $O_2$ | 0.33 |
| Nitrogen | $N_2$ | 4.4 |



Path to CEA: C:\Details5

Path to GasTurb Details or to GasTurb: C:\Details5

Name of the new fuel: Natural Gas

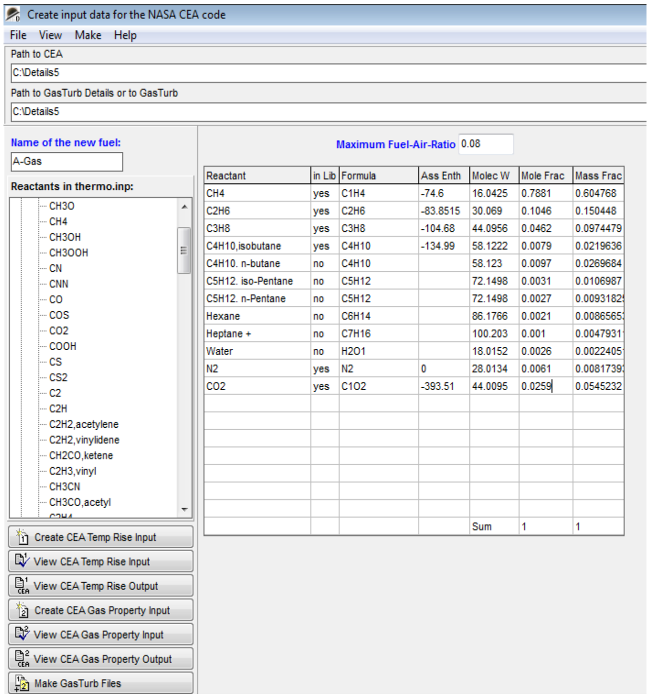
Maximum Fuel-Air-Ratio: 0.08

| Reactant | in Lib | Formula | Ass Enth | Molec W | Mole Frac | Mass Frac |
|---|---|---|---|---|---|---|
| CH4 | yes | C1H4 | -74.6 | 16.0425 | 0.85 | 0.75295 |
| C2H6 | yes | C2H6 | -83.8515 | 30.069 | 0.088 | 0.146109 |
| CO2 | yes | C1O2 | -393.51 | 44.0095 | 0.007 | 0.0170106 |
| CO | yes | C1O1 | -110.535 | 28.0101 | 0.0043 | 0.0066505 |
| H2 | yes | H2 | 0 | 2.01588 | 0.0017 | 0.0001892 |
| H2S | yes | H2S1 | -20.6 | 34.0809 | 0.0017 | 0.0031991 |
| O2 | yes | O2 | 0 | 31.9988 | 0.0033 | 0.0058307 |
| N2 | yes | N2 | 0 | 28.0134 | 0.044 | 0.0680603 |
|  |  |  |  | Sum | 1 | 1 |

Fig 1 Screen shot for modelled natural gas fuel

### Table 2: A-gas fuel composition

| Component | Vol. % |
|---|---|
| Water | 0.26 |
| Nitrogen | 0.61 |
| Carbon dioxide | 2.59 |
| Hydrogen Sulphide | 0.001 |
| Methane | 78.81 |
| Ethane | 10.46 |
| Propane | 4.62 |
| Iso-buthane | 0.79 |
| N-butane | 0.97 |
| Iso-pentane | 0.31 |
| N-pentane | 0.27 |
| N-hexane | 0.21 |
| N-heptane | 0.10 |



Path to CEA: C:\Details5

Path to GasTurb Details or to GasTurb: C:\Details5

Name of the new fuel: A-Gas

Maximum Fuel-Air-Ratio: 0.08

| Reactant | in Lib | Formula | Ass Enth | Molec W | Mole Frac | Mass Frac |
|---|---|---|---|---|---|---|
| CH4 | yes | C1H4 | -74.6 | 16.0425 | 0.7881 | 0.604768 |
| C2H6 | yes | C2H6 | -83.8515 | 30.069 | 0.1046 | 0.150448 |
| C3H8 | yes | C3H8 | -104.68 | 44.0956 | 0.0462 | 0.0974479 |
| C4H10,isobutane | yes | C4H10 | -134.99 | 58.1222 | 0.0079 | 0.0219636 |
| C4H10. n-butane | no | C4H10 |  | 58.123 | 0.0097 | 0.0269684 |
| C5H12. iso-Pentane | no | C5H12 |  | 72.1498 | 0.0031 | 0.0106987 |
| C5H12. n-Pentane | no | C5H12 |  | 72.1498 | 0.0027 | 0.0093182 |
| Hexane | no | C6H14 |  | 86.1766 | 0.0021 | 0.0086565 |
| Heptane + | no | C7H16 |  | 100.203 | 0.001 | 0.0047931 |
| Water | no | H2O1 |  | 18.0152 | 0.0026 | 0.0022405 |
| N2 | yes | N2 | 0 | 28.0134 | 0.0061 | 0.0081739 |
| CO2 | yes | C1O2 | -393.51 | 44.0095 | 0.0259 | 0.0545232 |
|  |  |  |  | Sum | 1 | 1 |

Fig 2 Screen shot for modelled A-gas Fuel

## 2.1 Engine Performance Simulations

Following the completion of the modelling of the various fuels, an engine configuration was adopted to investigate the performance. This engine configuration was selected based on intended application, which is power generation. The engine adopted and modelled in the simulation software was inspired from the LM2500 class of GE gas turbines. Figures 3 and 4 show the Twin Shaft engine configuration and schematic employed for the investigations, while Table 3 depicts the engine design specifications.

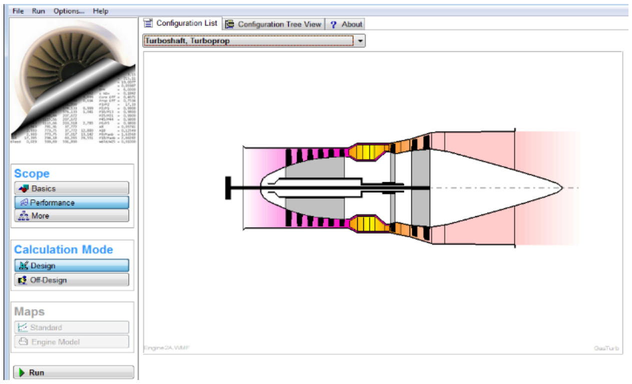


Fig 3 Twin shaft aero-derivative gas turbine engine configuration

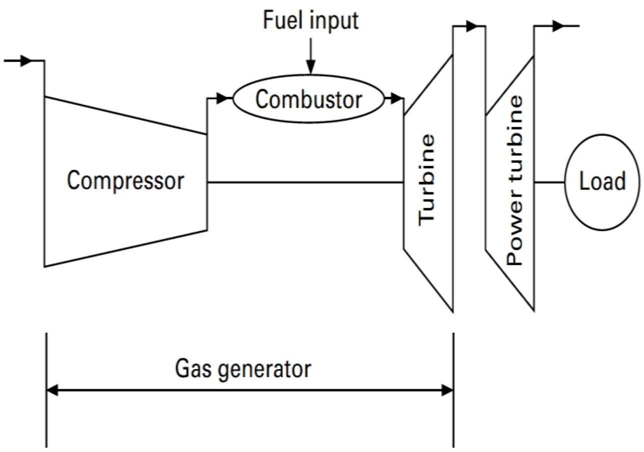


Fig 4 Twin shaft aero-derivative engine schematic

Table 3: Engine design specifications

| Design Parameters | Values |
|---|---|
| Power Output | 25 MW |
| Pressure Ratio | 18 |
| Thermal Efficiency | 34% |
| Mass Flow Rate | 70kg/s |

Consequent upon modelling natural gas and A-gas Fuels in the GASTURB 5.1 details, the two modelled fuels were imported into the GASTURB 11 version (see Figures 5 and 6 respectively), to simulate the overall performance of the different fuels in the gas turbine. Table 4 shows the fuel heating value of the natural gas and A-gas fuels extracted from the design point simulation interface (see Figures 5 and 6 respectively).

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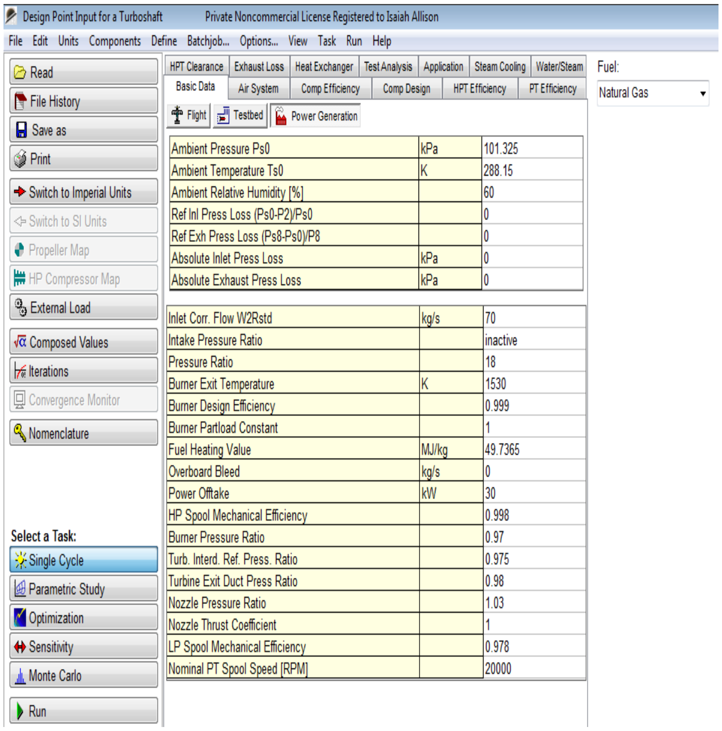
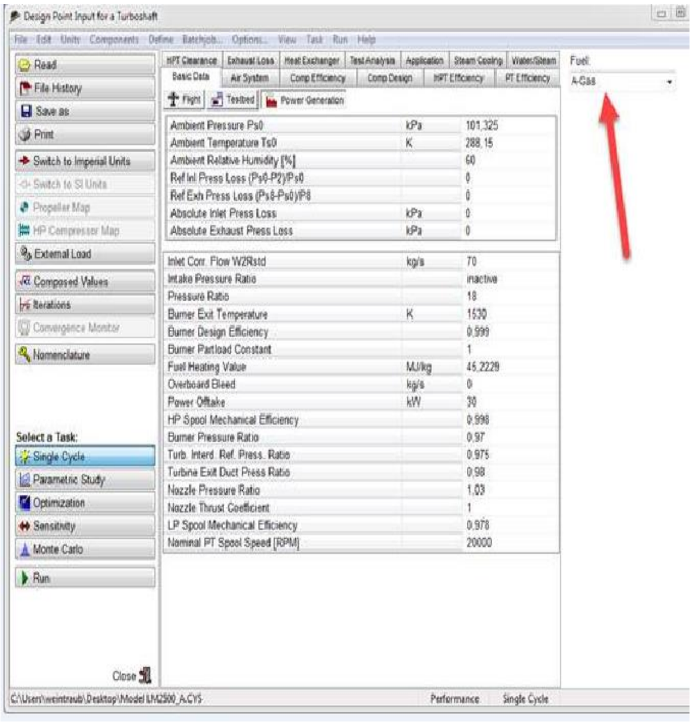


Fuel: Natural Gas

| | | |
|---|---|---|
| Ambient Pressure Ps0 | kPa | 101.325 |
| Ambient Temperature Ts0 | K | 288.15 |
| Ambient Relative Humidity [%] | | 60 |
| Ref Inl Press Loss (Ps0-P2)/Ps0 | | 0 |
| Ref Exh Press Loss (Ps8-Ps0)/P8 | | 0 |
| Absolute Inlet Press Loss | kPa | 0 |
| Absolute Exhaust Press Loss | kPa | 0 |

| | | |
|---|---|---|
| Inlet Corr. Flow W2Rstd | kg/s | 70 |
| Intake Pressure Ratio | | inactive |
| Pressure Ratio | | 18 |
| Burner Exit Temperature | K | 1530 |
| Burner Design Efficiency | | 0.999 |
| Burner Partload Constant | | 1 |
| Fuel Heating Value | MJ/kg | 49.7365 |
| Overboard Bleed | kg/s | 0 |
| Power Offtake | kW | 30 |
| HP Spool Mechanical Efficiency | | 0.998 |
| Burner Pressure Ratio | | 0.97 |
| Turb. Interd. Ref. Press. Ratio | | 0.975 |
| Turbine Exit Duct Press Ratio | | 0.98 |
| Nozzle Pressure Ratio | | 1.03 |
| Nozzle Thrust Coefficient | | 1 |
| LP Spool Mechanical Efficiency | | 0.978 |
| Nominal PT Spool Speed [RPM] | | 20000 |

Fig 5 Screen shot of modelled natural gas fuel on GASTURB 11 interface



Fuel: A-Gas

| | | |
|---|---|---|
| Ambient Pressure Ps0 | kPa | 101,325 |
| Ambient Temperature Ts0 | K | 288,15 |
| Ambient Relative Humidity [%] | | 60 |
| Ref Inl Press Loss (Ps0-P2)/Ps0 | | 0 |
| Ref Exh Press Loss (Ps8-Ps0)/P8 | | 0 |
| Absolute Inlet Press Loss | kPa | 0 |
| Absolute Exhaust Press Loss | kPa | 0 |
| Inlet Corr. Flow W2Rstd | kg/s | 70 |
| Intake Pressure Ratio | | inactive |
| Pressure Ratio | | 18 |
| Burner Exit Temperature | K | 1530 |
| Burner Design Efficiency | | 0,999 |
| Burner Partload Constant | | 1 |
| Fuel Heating Value | MJ/kg | 45,2229 |
| Overboard Bleed | kg/s | 0 |
| Power Offtake | kW | 30 |
| HP Spool Mechanical Efficiency | | 0,998 |
| Burner Pressure Ratio | | 0,97 |
| Turb. Interd. Ref. Press. Ratio | | 0,975 |
| Turbine Exit Duct Press Ratio | | 0,98 |
| Nozzle Pressure Ratio | | 1,03 |
| Nozzle Thrust Coefficient | | 1 |
| LP Spool Mechanical Efficiency | | 0,978 |
| Nominal PT Spool Speed [RPM] | | 20000 |

**Fig 6 Screen shot of modelled A-gas fuel on GASTURB 11 interface**

Table 4: Fuel heating value

| Fuel Type | Fuel Heating Values (MJ/kg) |
|---|---|
| Natural gas | 49.7365 |
| A-gas | 45.2229 |

To conduct a comparative performance analysis on the two fuels, a twin shaft engine shown in Figure 3 was modelled. The fuels modelled were then simulated in the GASTURB 11 simulation software, to ascertain the performances of the various fuels. In simulating the performance of the two fuels, namely natural gas and A-gas, the clean and degraded operating conditions for both fuels were considered. The degradation simulations were also considered because in real life scenario, the degradation in gas turbines performance is unavoidable even when operated under the best possible conditions due to several degradation mechanisms. One of the key factors that lead to compressor performance degradation during plant operation is compressor fouling. This is the adherence of particles and small droplets to the blading surface (Leusden *et al.*, 2004). Also, degradation simulations are considered because of the A-gas which contains impurities that deposit and degrade the blade performance.

It is well-known fact that during operation, gas turbine components deteriorate in performance. This is because gas turbines being air breathing machines ingest large amount air flow, which contains contaminants ranging from dust particles to soot, from salt to oil etc. that deposit on the surface of the compressor blades, thereby resulting in performance deterioration. Apart these factors mentioned above, there is also degradation associated with aging of the gas turbine components, which is wear and tear. Hence, this underscores the relevance of considering degradation investigation in this study.

Stalder (2001) in his experimental study, observed 10% degradation in power output for a power plant running over 4000 operating hours without any form of compressor cleaning. Also, Lakshminarasimha and Saravanamutto (1986) found from open literature that a reduction in 5% inlet mass flow will result in a compressor efficiency drop of about 2.5%. This would translate to a power output reduction of about 10%.
Based on the deductions from these literatures, a flow reduction in inlet mass flow and efficiency of 5 and 2.5% respectively, were adopted and implanted to simulate the effects of degradation in this study. In addition, it also assumed that the 5% reduction in mass flow and 2.5% occurred over 4000 operating hours without any form of compressor cleaning or maintenance activity.

Two different scenarios, namely comparison of natural gas and A-gas fuels and clean and degraded conditions were investigated. It is worthy to note that the simulations were conducted under a constant load condition. Also, the annual fuel costs for utilizing natural gas and associated gas were calculated using equation (1):

$$\mathit{Annual\ fuel\ cost} = \mathit{Engine\ power\ output} \times \mathit{Heat\ rate} \times \mathit{Operating\ hours\ per\ year} \times \mathit{Fuel\ cost} \qquad (1)$$

(Ganapathy, 1993)

## 3. RESULTS AND DISCUSSION

The bar chart plots of Figures 7 to 10 show the comparison of natural gas and A-gas fuels for clean and degradation operating conditions obtained from the simulation results. As can be seen in Figure 7, when Natural Gas and A-gas fuels were compared for clean condition, the heat rate of Natural Gas is 9923 kJ/kWh as against 9974kJ /kWh for an A-gas Fuel, which translates to 0.5% change.

Also, in Figure 7, when the plots of clean and degraded conditions were compared for natural gas, the heat rate of clean is 9923 kJ/kWh as against 10178 kJ /kWh for a degraded condition, which translates to 2.5 % change.

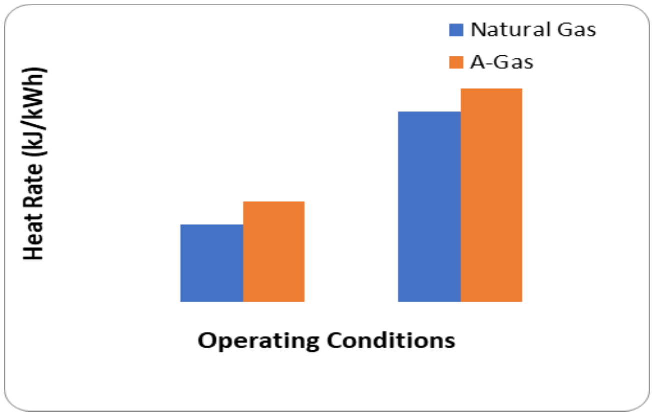


**Fig 7 Heat rate against operating conditions**

As can be seen from Figure 8, when Equivalent SFC of natural gas and A-gas were compared at clean condition, the equivalent specific fuel consumption of natural gas is 0.19953 kg/kWh as against 0.231298 kg/kWh for A-gas fuel, which is approximately 13.7% change between the two fuels. This can be attributed to the higher Fuel Heating Value of Natural Gas, which is 49.7365MJ/kg as against A-Gas of 45.2229MJ/kg.

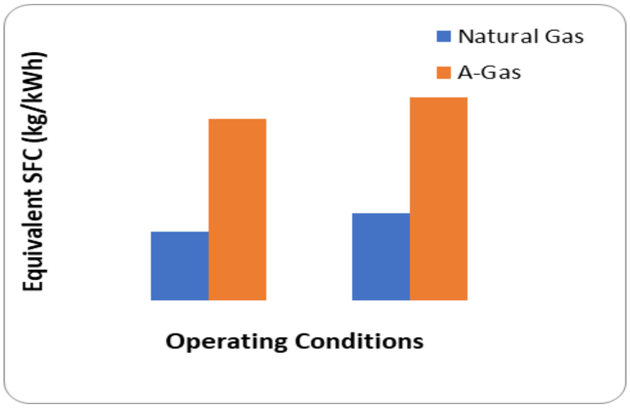


**Fig 8 Equivalent specific fuel consumption against operating conditions**

Figure 9 shows plots thermal efficiency for natural gas and A-gas. As expected, the thermal efficiency of the natural gas is higher than that of the A-gas. This is because of the lower heat rate of the natural gas. Hence, resulting in higher thermal efficiency because it is the inverse of heat rate. Also, from the figure, the clean case produced higher thermal efficiency than the degraded condition. The reduced thermal of degraded condition is as a result of lower pressure ratio arising from the degraded condition.

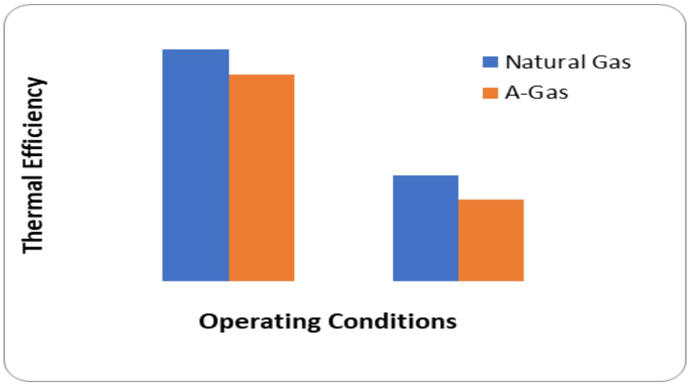


Fig 9 Thermal efficiency against operating conditions

The plot of fuel flow in Figure 10 is similar to that of Equivalent Specific Fuel Consumption in Figure 8. At clean condition, the percentage change in fuel flow between A-Gas and Natural Gas is approximately 13.7%. The higher fuel flow of the A-Gas can be attributed to its lower Fuel Heating Value. Hence, demanding higher amount of fuel to maintain the required power setting.

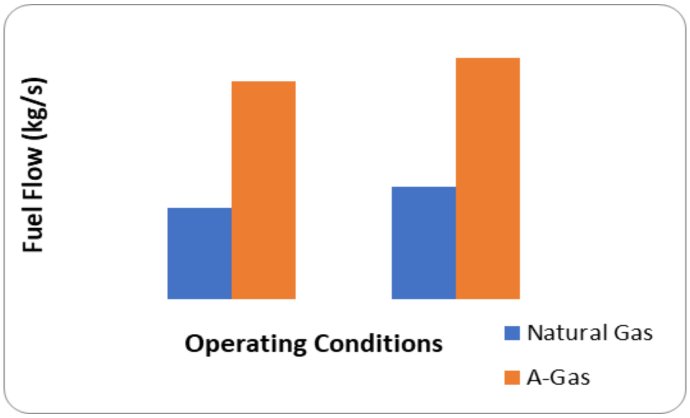


Fig 10 Fuel flow against operating conditions

The investigation of the influence of different fuel composition and degradation have presented a good understanding of how both scenarios affect the overall performance of the gas turbine; in particular, increased fuel consumption when the engine is running at constant load condition. It is worth mentioning that the focus of this study is on economic viability of harnessing/utilizing associated gas, to prevent energy wastage and other environmental hazards associated with gas flaring.

According to Boyce (2002), the operating cost which essentially is the cost of energy account for about 70-80% percent of the life cycle cost of any power plant. The remaining percentage is shared between the costs of a new power plant and maintenance costs. About 7-10% and 15-20% are the costs of a new power plant and maintenance costs respectively. This underscore the relevance of this study especially in today's world where prices of fuel are so high and environmental issues are a major concern. It is assumed that the gas turbine under investigation operated 4000 hours per year. The operating hours of the gas turbine depends on the energy requirements from the flow station and the neighbouring communities and the availability of fuel (gas). It is therefore presumed that the gas turbine operated for an average of twelve hours daily, which translates to approximately over 4000hours for one year.

### 3.1 Comparing cost of A-gas and NG fuels

It is worthy to note that in conducting the economic analysis, the production cost of the two fuels and capital cost of gas turbine were not considered. In addition, the details regarding redesigning of the gas turbine combustor so as to accommodate the A-gas fuel was not taken into account. Hence, the gas turbine combustor adopted in the case, is that which utilizes natural gas as the fuel, and it was adopted for the A-gas fuel investigation. This procedure can be considered acceptable.

However, using hydrogen on a gas turbine combustor designed for natural gas application may have some negative implications, especially the degradation in performance of the turbine blades due impurities of the A-gas. Ganapathy (1993) method of estimating the annual cost of fuel is adopted to conduct the economic analysis in this study. According to Ganapathy (1993), annual fuel cost = engine power output x heat rate x operating hours per year x fuel cost.

### 3.1.1 Natural Gas Cost Analysis

Annual fuel cost = engine power output x heat rate x operating hours per year x fuel cost (Ganapathy,1993).

**Table 5: Natural gas cost analysis**

| Operating Parameters | Values |
|---|---|
| Power Output | 25492kW |
| Heat Rate | 9929.9kJ/kWh |
| Operating Hours | 4000hrs/year |
| Fuel Cost | $14.1/MBtu |

Therefore, utilizing the data of Table 5, annual cost of natural gas fuel=25,492kW x 9406 Btu/ kWh x 4000 hr x $14.71 /MBtu= $14.1Million

### 3.1.2 A-gas cost analysis

The cost of A-Gas is assumed to be zero because it is an impure gas and usually flared or wasted.
Therefore, it is assumed the A-Gas is obtained at no cost. Although in real scenario may be cost implications; however, in this study, it was not considered.

**Table 6: A-gas cost analysis**

| Operating Parameters | Values |
|---|---|
| Power Output | 25492kW |
| Heat Rate | 9974.47kJ/kWh |
| Operating Hours | 4000hrs/year |
| Fuel Cost | $0/MBtu |

Annual Fuel Cost for A-Gas: 25,492kW x 9453Btu / kWh x 4000 hr x $0 /MBtu= $0

## 4. CONCLUSION

This study examines the economic viability of using associated gas as a fuel in gas turbines for power generation applications. GASTURB simulation software was employed in modelling the natural gas and associated gas (A-gas) fuels and simulating the performance of the two fuels. When the equivalent specific fuel consumption of natural gas and A-gas were compared at clean condition, the Equivalent specific fuel consumption of natural gas is 0.19953 kg/kWh as against 0.231298 kg/kWh for A-gas fuel, which is approximately 13.7% change between the two fuels. Similarly, the fuel flow follows same trend as the equivalent SFC.

Also, from the economic analysis, when A-gas is obtained at no cost, the annual cost saving for utilizing the A-gas is about $14.1million.
Although, all the gas turbine performance parameters investigated favour the utilization of natural gas as a fuel in the twin shaft gas turbine as against A-gas. However, as mentioned above, if the A-gas which usually flared or wasted can be obtained at no cost, the annual cost saving for utilizing the A-gas is about $14.1million.

## 5. ACKNOWLEDGEMENT

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## NOMENCLATURE

| | |
|---|---|
| *A-gas* | Associated gas |
| *BCM* | Billion cubic meters |
| *BTU* | British thermal unit |
| *GT* | Gas turbine |
| *PHCN* | Power Holding Company of Nigeria |
| *PR* | Pressure ratio |
| *SCF* | Standard cubic feet |
| *SFC* | Specific fuel consumption |
| *T* | Total temperature |
| *TET* | Turbine entry temperature |