Nigerian Research Journal of Engineering and Environmental Sciences
Journal homepage: www.rjees.com

Original Research Article

Performance Analysis of Industrial Gas Turbines Fueled with Hydrogen

***[1]Agbadede, R., [2]Nkoi, B. and [3]Kainga, B.**

[1]Department of Electrical Engineering, Nigeria Maritime University, Okerenkoko, Warri, Delta State, Nigeria.
[2] Depart of Mechanical Engineering, Rivers State University, Nkpolu, Port Harcourt, Rivers State, Nigeria.
[3]Department of Mechanical Engineering, Nigeria Maritime University, Okerenkoko, Warri, Delta State, Nigeria.
*roupa.agbadede@nmu.edu.ng

ARTICLE INFORMATION

Article history:
Received 29 May, 2020
Revised 14 Jun, 2020
Accepted 17 Jun, 2020
Available online 30 June, 2020

Keywords:
Hydrogen
Gas turbines
Performance analysis
Natural gas
Creep life

ABSTRACT

Hydrogen fuel is a promising alternative to fossil fuels because of its numerous performance advantages and clean combustion. However, there are economic and technical concerns which need to be addressed for its application to be feasible. This study aims to present the performance analysis of industrial gas turbines fueled with hydrogen. Default fuels of hydrogen and natural gas provided in the GasTurb performance simulation software were adopted and consequently simulated to ascertain the performances of the various fuels under investigation in the industrial gas turbine model. When the plots of natural gas and hydrogen were compared for clean condition, the thermal efficiency for the engine run on natural gas was 0.363 as against 0.374 for hydrogen fuel. This translates to 2.9% increase in thermal efficiency for hydrogen fuel. Also, when the clean and degraded conditions were compared for natural gas, it shows that thermal efficiency reduced from 0.363 to 0.352. *Time to rupture for the engine run on natural gas is 77,652 hours as against 222,050 hours for hydrogen.*

1. INTRODUCTION

With the continuous demand for clean fuel and stringent environmental regulations regarding emission of greenhouse gases, coupled with the fact that the depletion of fossil fuels is on the increase, experts in the energy sector are of the view that hydrogen may likely be the alternative to fossil fuels. This is because of the numerous performance advantages hydrogen offers over other sources of fuels. These attractive performance advantages of hydrogen include high specific energy, clean combustion, wide flammable range permitting weak operating mixtures etc. In addition, hydrogen is globally available, and causes less noise, longer engine life and lower maintenance when used in engines (Najjar et al., 1990). However, there are economic and technical concerns which need to be overcome for its application to be feasible.

Hawksworth *et al.* (2015) performed a study on safe operations of combine cycle gas turbine and gas engines using hydrogen fuel. Flammable mixtures together with make-up oxygen were injected into the tube close to the Rolls Royce Viper engine outlet and then ignited. The authors reported deflagration to detonation transition under certain conditions with flame velocities of 200 m/s and overpressures in excess of 8 bar.An experimental campaign was carried out by varying mixtures of hydrogen and natural gas mixtures on a Dry Low NOx (DRN) combustor usually employed for a 70 MWe gas turbine (Giancarlo et al., 1999). The results show that hydrogen enrichment of the natural gas can promote a wide flame stability range together with NOx reduction. Wu *et al.* (2007) conducted a detailed combustor system testing on hydrogen rich syngas fuels to demonstrate low emissions capability. The study revealed that the system can be used to meet low emission requirements. Anderson *et al.* (2013) conducted a test using the 3rd generation DLE (dry low emission) burner with hydrogen feed to one burner in a gas turbine. The study highlights the capability of the concept burner to operate with at least 45% volume of hydrogen in natural gas. The authors reported that NOX emissions could be kept below 24 parts per million @ 15% O_2 under 70-100 % gas turbine load with the hydrogen rich fuel. Chiesa et al. (2005) investigated the possibility of burning hydrogen in large size and heavy-duty gas turbines designed to run on natural gas. This was done to ascertain the possibility of reducing green- house emission on short term basis in the power industry. A new combustor capable of suppressing the generation of the low flow velocity in the swirling center region was developed (Inoue et al., 2018). According to the authors, this was done so as to prevent the occurrence of the flashback phenomenon caused by hydrogen co-firing. The study reported that the prospect for gas turbine operation under 30% vol. hydrogen co-firing conditions was obtained.

From literature review conducted, it is obvious that most of the publications available in the public domain on hydrogen fuel utilizing are focused on the improvement of the combustion system, to accommodate the effects of flashback and other factors experienced switching from conventional gas turbines designed to run on natural gas. There is no available study which has holistically considered the performance of an engine fueled with hydrogen and its impact on the turbine blade creep life. This study thus presents the performance analysis of industrial gas turbines fueled with hydrogen and natural gas. Also, the impact of hydrogen utilization on the turbine blades creep life was investigated.

2. MATERIALS AND METHODS

2.1. Gas Turbine Performance Analysis

GasTurb simulation Software employed in this study comprised of the GasTurb Details 5.1 and GasTurb 11. The GasTurb program utilizes predefined engine configurations, thus allowing for an immediate start of calculations. Problems are solved without the need for setting up an engine configuration. Also, many standard tasks are prepared in such a way that one gets answers quickly. With the Gasturb details 5.1, the major components such as compressor, Turbine, mixers, nozzles, fuel modeling etc can be studied independently from the main GasTurb software. It uses some basic calculations with the same procedures as employed in GasTurb 11, to execute the analysis of the individual components. GasTurb 11 is designed for easy evaluation of the thermodynamic cycle both for design and off-design performances of the GT. GasTurb is a user-friendly performance program for cycle design and off-design simulations. It also covers the preliminary geometrical design of engines including disk stress calculations. Based on the capabilities and features of the GasTurb software, it adopted in this study to simulate the performance of engines fueled hydrogen and natural gas.

To conduct a comparative performance analysis of the hydrogen and natural gas fuels, a twin shaft aero derivative industrial gas turbine inspired from LM2500 engine was modelled. The model was achieved by selecting a twin shaft engine configuration from the software interface and consequently implanting design

R. Agbadede et al. / Nigerian Research Journal of Engineering and Environmental Sciences
5(1) 2020 pp. 178-189

specification data obtained from open domain, which is similar to the proposed engine model. Few of the data such as component efficiencies and turbine entry temperature were altered to arrive at the expected design engine specifications. Default fuels of natural gas and hydrogen provided in the software were adopted and then simulated in the Gasturb software, to ascertain their performances. Figure 1 shows the industrial gas turbine engine configuration model adopted for the investigations, while Table 1 presents the design point performance specifications. Modelled design point simulation interface is presented in Figure 2.

The gas turbine combustor adopted in the performance analysis for both fuels, is the one which utilizes natural gas as the fuel, and it was employed for the hydrogen fuel performance simulations.

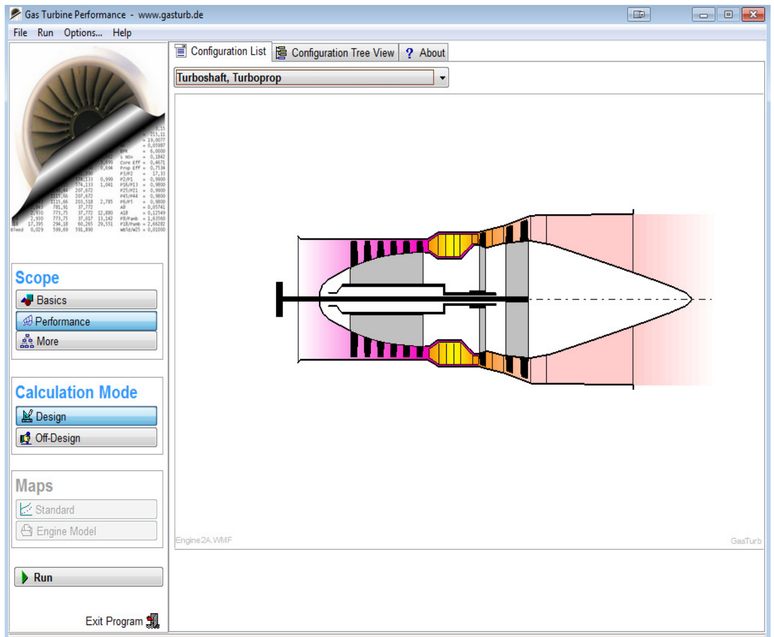

Figure 1: Industrial gas turbine engine configuration

Table 1: Engine design specifications (Courtesy of General Electric)

Design parameters	Units
Power output	25 MW
Thermal efficiency	36
PR	18
Exhaust temperature	839 K
Exhaust flow	70.5 kg/s
Heat rate	9705 kJ/kWh

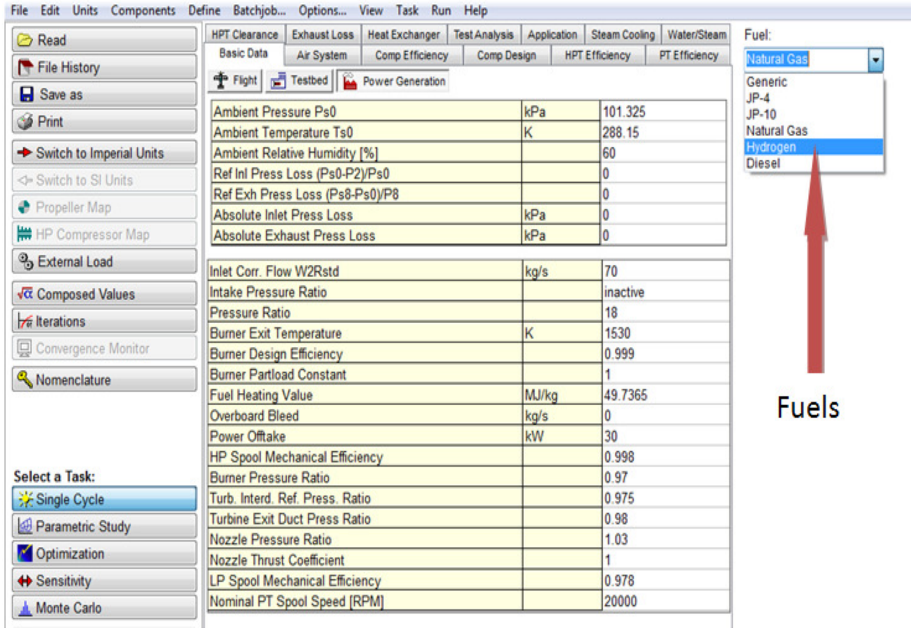

Figure 2: Modelled design point with various fuels

In simulating the performance analysis of the two fuels, degradation values were also incorporated. Degradation simulations were considered along with the comparative performance analysis of the fuels in this study because gas turbines in operation always degrade in performance even when operated under the best possible conditions due to several degradation mechanisms (Leusden et al., 2004). Based on the deductions from Lakshminarasimha and Saravanamutto (1986), a flow reduction in inlet mass flow and efficiency of 5 and 2.5% respectively, were adopted in this study. It is worthy to mention that the simulations were conducted under a constant load condition.

2.2. Blade Creep Life Estimation

To have a better understanding of the impact of hydrogen fuel utilization on the life of the gas turbine under a given load requirement, it is imperative to conduct a creep life estimation on the high pressure turbine (HPT) blades for the two cases: engine run on natural and on hydrogen fuel). To achieve this objective, the HPT turbine blade was sized. The initial design data were selected based on data from Jane's aero engines (Gunston, 1996) to scale the HPT blade coupled with a GE publication on the gas turbine. Consequently, the gas path inlet and outlet parameters at design point were obtained using GasTurB simulation software. Constant nozzle inlet angle design approach was adopted. This method was used in order to satisfy the radial equilibrium condition, which will in addition take care of the constant mass flow per unit area requirement of all the blade radii (Ramsden, 2009).

Adopting the blade design process outlined by Ramsden (2009), the inlet and outlet geometry were sized, followed by predicting the stage efficiency using Smith's correlation and calculating the rotor inlet velocity. Table 2 shows the blade design dimensions at the root, mean and tip obtained.

Table 2: Blade design specification (Allison, 2010)

Inlet annulus geometry	
Mean diameter (m)	0.9525
Height(m)	0.0469
Tip diameter(m)	0.9950
Hub diameter(m)	0.9056

In designing the blade, the following values were adopted for the blade aerodynamic parameters (Allison, 2010):

- The inlet Mach number of the HP turbine is constant at 0.3
- The axial velocity is constant at 226 m/s
- The HPT speed of 8000 RPM obtained from the performance simulations was adopted.
- Cooling effectiveness of 0.4 was adopted.

Equation 1 was used to calculate the cooling effectiveness (ε), and it also shows how the gas stream temperature T_g, blade metal temperature (T_b) and the coolant temperature are related to the cooling effectiveness.

$$\varepsilon = \frac{T_g - T_b}{T_g - T_{c1}} \tag{1}$$

ε is the cooling effectiveness, T_g is the gas stream temperature, T_b is the blade metal temperature and T_{c1} is the coolant temperature.

The multi-dimensional stresses and strains arising from load-imposed degradation and the non-uniform temperature distribution exiting from the combustor have made the prediction of the turbine blade creep life difficult. Assuming a uniform stress and temperature could prevent these problems when estimating the creep life of turbine blades. In order to achieve a more accurate result, radial temperature distribution factor (RTDF) is employed to estimate the blade creep life in this study. Turbine blades, due to their rotation, experience average circumferential temperatures in a given radial plane. The RTDF which affects the rotor blade life when employing measured circumferential values is used to calculate the temperature variation at each section of the blade. According to Walsh and Fletcher, (2004) the controlled value of RTDF should be less than 20%. Equation 2 was used for the RTDF calculation.

$$\text{RTDF} = T_{max} - \text{TET}/ (\text{TET} - T_3) \tag{2}$$

T_{max}= Circumferentially mean outlet peak temperature, TET = Mean temperature and TET – T_3=Mean Combustor temperature rise.

Also, based on the assumption that the maximum temperature will occur at the 75% blade height, the following formulae were derived. Calculations of maximum and minimum temperatures were thus given below (Eshati et al., 2010):

$$T_{max} = T_{RI} + (\Delta T_{burner}\, X\, RTDF) \tag{3}$$

$$T_{min} = {(5T_{RI} - 2T_{max})}/{3} \tag{4}$$

T_{RI}= Rotor inlet relative gas temperature (K) and ΔT_{burner}= Temperature rise at burner (K)

The assumptions which were adopted in this study are (Eshati et al., 2010):

- The minimum gas temperature occurs at the root and tip of the blade
- Linear rise of gas temperature from root to 75% of the blade span
- Linear reduction in gas temperature from maximum to the blade tip
- The average of the root datum section temperature (T_{RDS}), temperature at 75% of blade height ($T_{75\%}$) and top datum section temperature (T_{TDS}) equal the turbine inlet temperature
- The temperature leaving the HP compressor was assumed to be the cooling air temperature in this study. This is because the cooling air is bled from the compressor

It was also assumed that the cooling air temperature will remain the same at all sections of the blade. The metal temperature, T_b, is given in Equation (5). T_b as a function of the cooling effectiveness, gas and coolant temperatures as highlighted in the equation.

$$T_b = T_g - \varepsilon (T_g - T_c) \tag{5}$$

In estimating the blade section metal temperatures, each section was treated as an individual blade, where the metal temperature was assumed to be constant. In addition, the cooling effectiveness and cooling air temperatures were assumed to be same at all the blade sections. In reality, this cannot be completely correct because there would be temperature variation from the entrance of the blade to the exit (Laskaridis, 2009). However, for the purpose of this study, it is assumed that the coolant temperature is the same for all the blade sections.

According to Laskaridis, (2009) the gas temperature equals the difference between the turbine inlet temperature and the temperature drop due to thermal barrier coating (TBC).

$$T_g = TET - \Delta T_{tbc} \tag{6}$$

According to Blackie (2008), centrifugal stress only contributes about 70-90% of the total blade stress. Hence, it is only centrifugal force that was considered for calculating the stress model in this study. Also, in considering the stress analysis, the blade was divided into four sections: stress acting at root datum section (RDS), 25%, 50% and 75% of blade height. Centrifugal Force experienced by the blade was calculated using Equation (7).

Therefore:

$$\text{Centrifugal force, } CF = mr\omega^2 \tag{7}$$

Where m= mass of blade in kg, r = radius of centre of gravity, ω = rotational speed in rads/sec

Mass of blade = density of blade x volume of blade

Substituting mass of blade into Equation (7) gives Equation (8).

$$CF = Ah\rho r \left(\frac{2\pi N}{60}\right)^2 \tag{8}$$

$$\sigma CF = \frac{CF}{A} \tag{9}$$

As mentioned earlier, the blade height was divided into four equal sections and the distances from the centre line (CL) to the RDS is taken into consideration to compute the height at various sections of the blade. This approach adopted is similar to the method employed by Allison (2010), to calculate the centrifugal force at different sections of the blade.

Radius from CL to centre of gravity (CoG) of first section, at 25% blade height = 0.4586475 m
Radius from CL to CoG of second section, 25 % blade height - mid blade height = 0.4703925 m
Radius from CL to CoG of third section, mid blade height – 75% blade height= 0.4821375 m
Radius from CL to CoG of forth section, TDS – 75% blade height = 0.4938825 m
From the material properties of René 80, the blade density, ρ = 8194.5 kg/m^3
Also, the rotational speed of the blade, N = 8000 RPM

Therefore, the stresses at different blade sections were calculated using Equation (9).

The stresses at the various blade sections are computed hereunder.

<u>TDS – 75% blade height</u>

Stress at 75% Height, using Equation 7:

σCF= 0.011725 x 8194.5 x 0.4938825 x$(2\pi x8000/60)^2$ = 33.3 MPa

<u>Mid blade height – 75% blade height</u>

Stress developed at second section is given by:

σCF= 0.011725 x 8194.5 x 0.4821375 x$(2\pi x8000/60)^2$ = 32.5 MPa

Total stress acting at 50% height is:

Total σCF = 32.5 + 33.3 = 65.8MPa

<u>25% blade height - mid blade height</u>

Stress developed at third section is given by:

σCF= 0.011725 x 8194.5 x 0.4703925 x$(2\pi x8000/60)^2$ = 31.7 MPa

Total stress acting at 25% height is:

Total σCF = 31.7 + 65.8 = 97.5 MPa

HPT blade creep life was estimated for the two cases: when the gas turbine was run on natural gas and on hydrogen fuels. The stresses calculated at the various sections of the blade were used to obtain LMP (Larson Miller parameter) from Master curve of the blade material. Consequently, the LMP and blade metal temperature at the blade sections were used to estimate the rupture time and the blade creep life for the calculated blade metal temperature.

Table 3: Stress at various sections of the blade

Blade section	Stress introduced	Total stress in section
Top datum section (TDS)-75% Blade span	σCF=33.3 MPa	Total stress acting at TDS-75% blades span, σCF=33.3 MPa
Mid blade span -75% blade span	σCF=32.5 MPa	Total stress acting at 25 % blades span, σCF=97.5 MPa
25% blade span –Mid blade span	σCF=31.7 MPa	Total stress acting at 50 % blades span, σCF=65.8 MPa
Root datum section – 25% blade span	σCF=30.9 MPa	Total stress acting at RDS, σCF=128.5 MPa

3. RESULTS AND DISCUSSION

The outcome of the analysis of engine run on hydrogen and on natural gas are presented using performance parameters such as thermal efficiency, ESFC, fuel flow and heat rate. These parameters were used because they provide information about the work done per unit heat input.

Figure 3 shows the plot of thermal efficiency at different conditions. As can be seen from the figure, the thermal efficiency of the engine run on natural was 36.27%, while that for the engine run on hydrogen was 37.37%. The higher thermal efficiency produced by the engine run on hydrogen can be attributed to the higher lower heating value (LHV) of hydrogen when compared to natural gas fuel. This trend is in agreement with the study of Razak (2007), where diesel and natural fuels were compared. In the study, the author reveals that 1-2% increase in thermal efficiency and power was achieved as a result of switching from diesel to natural gas fuel. The author stated that the increase in performance is as a result of the higher LHV of natural gas as against diesel. Also, when the clean and degraded conditions were for natural gas in the figure, it shows that thermal efficiency reduced from 0.36 to 0.35. This reduction in thermal can be caused by drop in component isentropic efficiency, thereby resulting in overall reduction of the thermal efficiency.

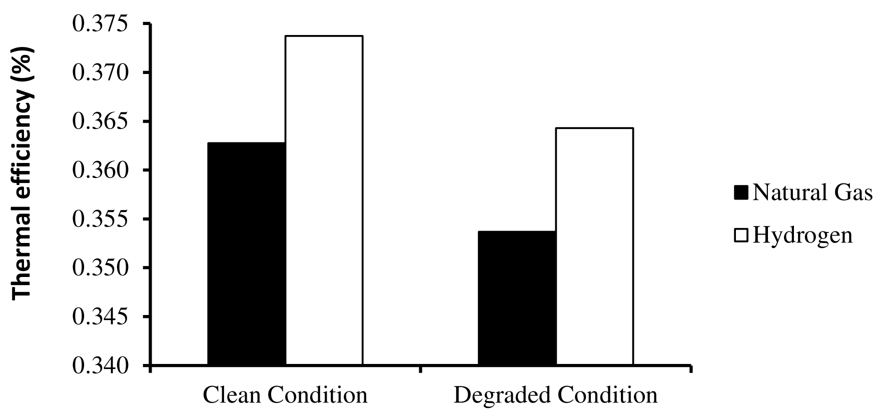

Figure 3: Plot of thermal efficiency at different conditions

Figure 4 shows the plot of heat rate at different conditions. When the plot of engine run on natural gas and on hydrogen were compared for the clean condition, the heat rate for the engine run on natural gas is 9923.9

kJ/kWh as against 9632.7 kJ/kWh for hydrogen fuel. The lower heat rate of hydrogen can be attributed to its higher FHV when compared to natural gas fuel. This signifies higher work done at a given heat input than the natural gas.

Figures 5 and 6 show the plots of equivalent specific fuel consumption (ESFC) and fuel flow respectively. The performance plots of ESFC and fuel flow followed a similar pattern to the heat rate plots when natural gas and hydrogen fuels were compared. ESFC for natural and hydrogen are 0.1995 kg/kWh and 0.0834 kg/kWh respectively. As mentioned earlier, the lower ESFC and fuel flow experienced for the hydrogen fuel is also as a result of the higher fuel heating value. Razak (2007) reported an increased fuel flow when the fuel was switched from natural gas to diesel. The author stated that the increased fuel flow for diesel was caused by its low LHV. Similarly, Najjar et al. (1990) reported improved fuel saving for hydrogen fuel when compared to LPG. Also, Figure 6 shows that fuel flow increased from a clean condition at 1.41 kg/s to 1.45 kg/s for the degraded condition in the gas turbine fueled with natural gas. This is due to the drop in component isentropic efficiency which results a reduction in output power. Therefore, fuel flow is increased to maintain the load demand (base load condition).

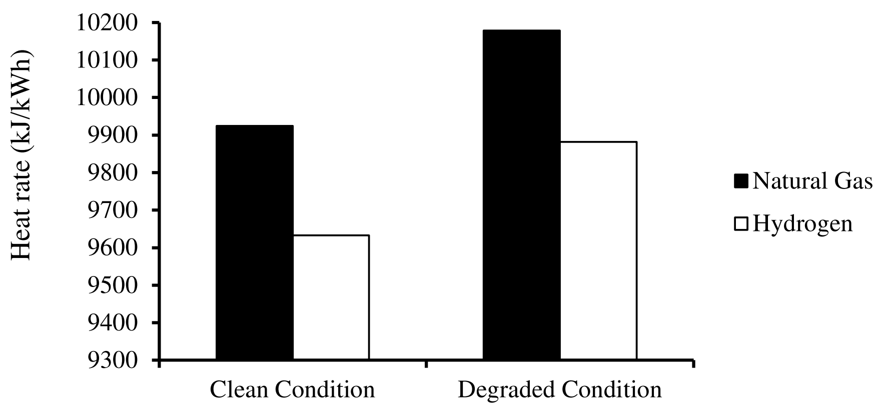

Figure 4: Plot of heat rate at different conditions

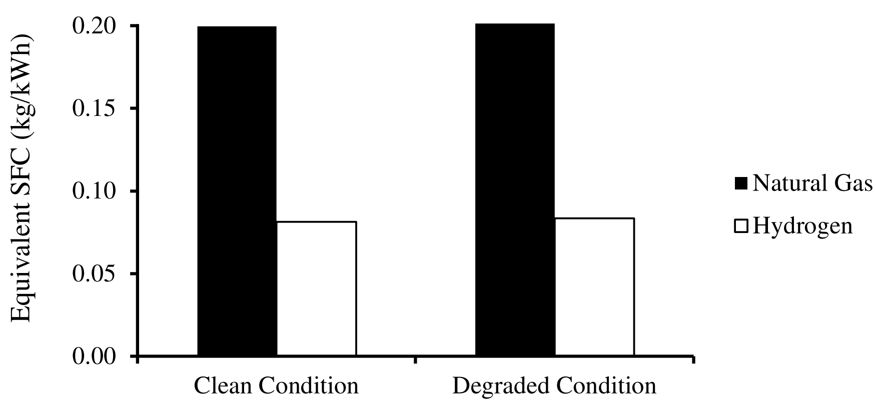

Figure 5: Plot of ESFC at different conditions

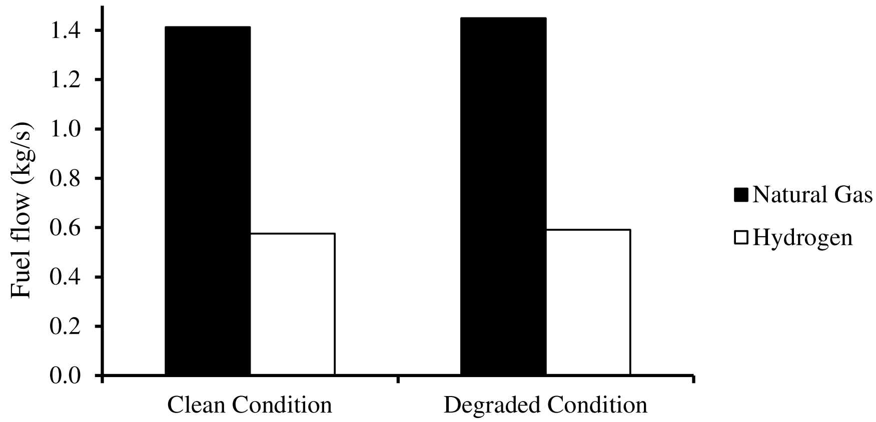

Figure 6: Plot fuel flow at different conditions

Figure 7 shows the bar chart plots of time to rupture of the blade estimated for engines operated on natural and of hydrogen fuel. When the time to rupture of the turbine blades for engine run on hydrogen and natural gas fuels was compared, the figure shows that the time to rupture for engine run on natural gas is 77,652 hours as against 222,050 hours for hydrogen. Converting the time to rupture of the blade from hours to years shows that engine run on natural gas is about 8.8 years as against 25 years for hydrogen fuel (See Figure 8). The time to rupture obtained for engine run on natural seems reasonable. This is because Weber et al. (2005) predicted hot section repair and retirement interval for Rene' 80 direction alloy at 25,000 and 50,000 hours respectively. Saravanamuttoo et al. (2009) stated that the nominal operating lives for gas turbines are in the range of 10,000 to 100,000 hours. In addition, the blade material considered in this study was the Rene' 80 materials. The wide variation in time to rupture experienced for the hydrogen fuel could be attributed to its better lower fuel flow/ consumption than the natural gas. Hence, the engine is operated at lower turbine entry temperatures to meet required load demand. Also, the time to rupture of the blades predicted by Weber et al. (2005) was for engines operated on natural gas fuel.

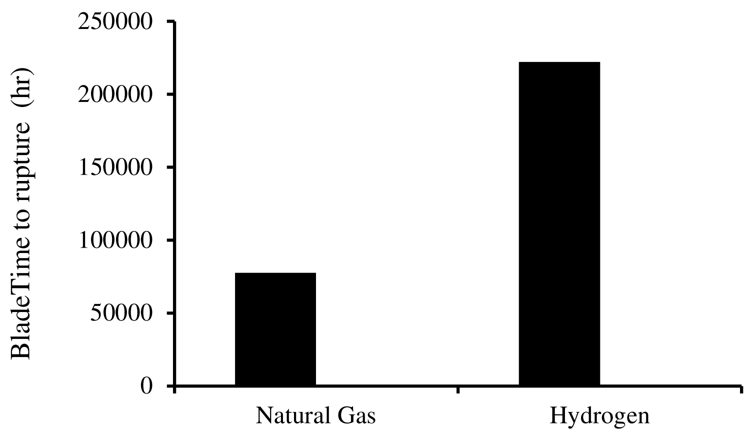

Figure 7: Turbine blade time to rupture for engines fueled with different fuels

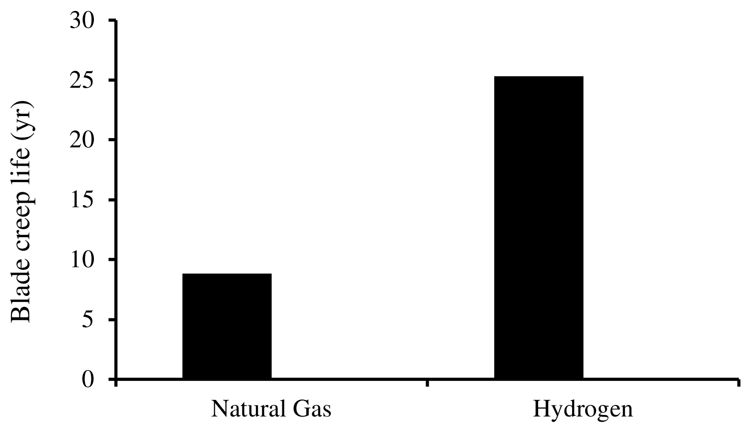

Figure 8: Turbine blade creep life for engines fueled with different fuels

4. CONCLUSION

This study presents the performance analysis of industrial gas turbines fueled with hydrogen and natural gas. The study demonstrates that gas turbine engine run on hydrogen fuel produced a better thermal efficiency than the engine fueled with natural gas. In addition, engine fueled with hydrogen produced lower heat rate, equivalent specific consumption and fuel flow than the engine run on natural gas. Also, the study shows that the time to rupture for engine run on natural gas is 77,652 hours as against 222,050 hours for hydrogen. Finally, the study reveals that the overall performance of engine fueled with hydrogen better than the engine run on natural gas.

5. ACKNOWLEDGMENT

The authors wish to express their gratitude to the Faculty of Engineering, Nigeria Maritime University for providing all the technical assistance required to conduct this study.

6. CONFLICT OF INTEREST

There is no conflict of interest associated with this work.

REFERENCES

Allison, I. (2010). Development of a Preliminary Creep Life Estimation Model for High Pressure Turbine Blade of Stationary Gas Turbines. MSc thesis, Cranfield university, Cranfield.

Andersson, M., Larfeldt, J. and Larson, A. (2013). Co-firing with hydrogen in industrial gas turbines. SGC report 256, 2013. Available electronically http://www.sgc.se/ckfinder/userfiles/files/SGC256.pdf. Accessed on March, 2020

Blackie, J. (2008). Engine Lifing Analysis-An Assessment of Creep, Diffusion and Cyclic Oxidation of HPT Blades Comprised of CMSX-4 Base Material and NiAl Coating. MSc thesis, Cranfield University, UK.

Chiesa, P., Lozza, G. and Mazzochi, L. (2005). Using Hydrogen as Gas Turbine Fuel. *Journal of Engineering for Gas Turbine and Power*, 127, pp. 73-80.

Eshati, E., Abdul Ghafir, M. F., Laskaridis, P. and Li, Y. G. (2010). Impact of Operating Conditions and Design Parameters on Gas Turbine Hot Section Creep Life. *Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air,* June 14-18, 2010, Glasgow, UK, ASME.

Giancarlo, B., Giordano, T. and Valerio, C. (1999). ENEL's Hydrogen and natural Gas Turbine Combustion. Firenze (Italy). Available electronically at https://www.cti2000.it/utils/downloadfile.php?table=pubblicazioni&id=19848f. Accessed on May, 2020

Gunston, B. (1996). *Jane's Aero-Engines*

Hawksworth, S. J., Modie, K., Gummer, J., Ewan, B., Michels, H., Linsteldt, P. Winstanley, P. and Pekalski, A. (2007). Safe Operation of Combustion Cycle Gas Turbines and Gas Engine Systems Using Hydrogen Rich Fuels. Available electronically at https://h2tools.org/sites/default/files/2019-08/paper_245.pdf Accessed on March, 2020

Lakshminarasimha, A. N. and Saravanamutto, H. I. H. (1986). Prediction of fouled compressor performance using stage techniques. *American Society of Mechanical Engineers, Fluids Engineering Division*, 37, pp. 59.

Laskaridis, P. (2009). Gas Turbine Lifing. Unpublished Fatigue and Fractures Course Notes, Department of Thermal Power, School of Engineering, Cranfield University, UK.

Leusden, C. P., Sorgenfrey, C. and Dümmel, L. (2004). Performance benefits using Siemens advanced compressor cleaning system. *Journal of Engineering for Gas Turbines and Power*, 126(4), pp. 763-769

Najjar, Y.S.H., Zahed, A. H., Bashir, M. D. and Alp, T. (1990). Feasibilty of hydrogen utilization in gas turbines. *Journal of Energy and Environment*, 1 (3), pp. 240-251.

Ramsden, K.W. (2009). Gas Turbine Fundamentals and Turbomachinery. MSc Lecture Notes, Department of Thermal Power, School of Engineering, Cranfield University, UK.

Razak, A.M.Y. (2007). *Industrial Gas Turbines Performance and Operability*. Woodhead Publishing Limited, Abington Hall, Abington Cambridge, England.

Saravanamutto, H.I.H., Rogers, G.F.C., Cohen, H. and Straznicky P. V. (2009). Gas *Turbine Theory*. 6th Ed. Pearson Education Limited, Edinburgh Gate Harlow, Essex, England.

Walsh, P. P. and Fletcher, P. (2004). *Gas turbine performance*. 2nd Ed, Blackwell Science, Limited, Oxford, UK.

Weber, B., Jin, H. and Pistor, R. and Lowden, P. (2005). Application of an Integrated Engineering Approach for LM1600 Blade Life on-line Assessment. Proceedings of the 16th Symposium on Industrial Application of Gas Turbines, October 12-14, 2005, Banff, Alberta, Canada.

Wu, J., Brown, P., Diakunchak, I. and Gulati, A. (2007). Advanced Gas Turbine Combustion Development for High Hydrogen Fuels. *Proceedings of ASME Turbo Expo 2007*: Power for Land, Sea and Air, May 14-17, 2007, Montreal, Canada.