International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 08, August-2015

Enhancing Performance and Creep Life of a Gas Turbine Engine Operating in a Load-Following Mode

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Abstract* - An assessment of the impact of compressor fouling on the creep life of gas turbine engines operating in a load- following mode is presented. Most base load plants are now been cycled or used for load-following purposes due to economic reasons such as minimizing operating cost (fuel) by cycling their plants for load-following operations. Some power plant operators and utilities are also forced to cycle aging fossil units that were originally designed for base load operations. However, due to frequent starts and stops associated with gas turbines used for load-following mode operations, the high pressure turbine (HPT) blades experience a high degree of centrifugal and thermal stresses. The situation becomes worse when such power plants are operated in locations where airborne impurities are prevalent. This is because the ingestion of airborne impurities such as sand or dusts, salt spray and hydrocarbon soot causes engine performance deterioration. The impact of performance deterioration results in loss of power output, increased fuel consumption, loss of revenues and increase in overall equipment lifecycle cost. In order to recover lost power, the engines are operated at higher firing temperatures resulting in decreased component creep life for a given power demand. In addition, compressor washing techniques have been introduced to recover lost power due to fouling. Nevertheless, there is no clear cut recommended compressor washing intervals proposed in open literature.This paper presents a performance based creep life model capable of quantifying the impact of fouling on the creep behavior of an engine operating in a load following mode. The model consist of a performance, stress, heat transfer and life estimation models. HPT blades are selected as the life limiting component of the gas turbine; therefore the model is employed to investigate the effect of several operating conditions on the creep life of a model two shaft aero-derivative gas turbine engine. An algorithm has been developed to estimate the compressor washing intervals that gives the best economic benefits. The developed algorithm could be used to select an appropriate compressor washing interval which may improve the performance, fuel consumption, creep life and the overall life cycle costs.

Keywords— Fouling; lifecycle cost; Creep life; Load-following;

I. INTRODUCTION

Gas turbines have been used extensively for power generation, aircraft propulsion, mechanical drive applications etc. Their ability to operate over a wide range of conditions makes them suitable for both design and off design conditions [1]. In recent times, gas turbines are now been cycled for load-following purposes due to the use of renewable energy sources (RES) which could make new designs, operation regimes, regulations to comply with this development [2]. For engines operating in load-following modes, Low cycle fatigue (LCF) has been identified as the predominant failure mechanism [3] due to the nature of the operating mode. So, most gas turbine life estimation of engines operating on load-following mode has been solely based on LCF. However, due to the high thermal loads and stresses at which these engines are being operated, they also suffer from creep deformation. Therefore, creep deformation is also an important and critical aspect of life prediction of gas turbine engines used in load-following operations. Furthermore, increasing engine operating temperatures to compensate for the loss in performance results in increasing fatigue and creep damage to the hot section components and increases the engine overall costs of operation and maintenance. More so, it could be argued that component deterioration such as fouling could also contribute significantly to loss in creep life of gas turbine engines. Hence, this paper aims to consider the impact of fouling on the creep life of gas turbines operated in a load-following mode. Such investigation will help gas turbine users to select the best operating regime that will help to minimize fouling thereby improving the creep life of the engine. However, from the economic and operation viewpoint, it is desirable to obtain the maximum amount of useful life when operating in harsh environments. In a bid to enhance gas turbine engine performance, compressor washing was introduced and has been successfully used over the years

[4]. The algorithm introduced estimates the compressor washing interval based on operating conditions.

A. Load-following Operations

During load-following operations, the blades are subjected to stresses which over time results in low cycle fatigue because of the frequent starts/stops operation. Load-following power plants have the capability to adjust their power output as the demand for electricity fluctuates during the day. The variation in power output is dependent on factors such as the demand, the location of the load-following plants and the season. Load-following plants are associated with frequent starts and stops which over time impairs on the engine life. This operation contributes some of the largest costs incurred by a plant which has consequences on the economics of the plant because it reduces the load factor; which is the ratio of the power output over the peak power over a period of time. When operating less than the full load, the implication is the likelihood of incurring additional cost on components which includes capital costs, fuel cost, operating and maintenance cost. Cycling related failures aren't noted immediately, but eventually critical components will start to fail over time. Several studies have been carried out on increased plant cycling due to renewables [5].

B. Fouling In Gas Turbine Compressors

The efficiency of an axial compressor depends on, among other considerations, the smoothness of the rotating and stationary blades surfaces. These surfaces can be roughened by erosion, but more frequently by tiny particles of aerosols contained in the air consumed by gas turbine, that stick on the blades of the compressor, causing the phenomenon of fouling [6; 7]. In addition, internal gas turbine oil leaks from axial compressor front bearing, when combined with dirt can cause serious fouling problems [8]. It has been well documented for many years that compressor fouling can greatly affect the performance of gas turbine engines [9]. The output of a gas turbine can be reduced as much as 20 per cent in cases of extreme compressor fouling [10]. The size of airborne particles causing fouling is usually 5µm or less and that particles greater than 5µm causes erosion [4].
Fig. 1 below shows black deposits caused by hydrocarbon contaminants in the vicinity of a gas turbine plant.

Fig. 1. Black deposits on compressor blades caused by hydrocarbon contaminants [11]

C. Operating Environment

Gas turbine compressors foul in most operating environments. Land-based gas turbines usually operate in a diversified environment which include desert, off-shore, onshore, industrial and marine as shown in *Fig. 2.* The range of contaminants that are contained in the air stream entering the gas turbine can range from gases, liquids and solid particles such as sand/dust, to particles such as chemicals, factory discharge gases, fuel vapours and a variety of industrial by-products. The location where the gas turbine operates and the surrounding activities define the type of contaminants present in the environment. Each environment presents different challenges to the filtration of the gas turbine inlet air. Some of these contaminants are influenced by the climatic conditions and can vary seasonally or even daily.
In the desert environment, the main concern is the dry sand/dust particles ingestion which can be large enough in size to cause compressor blade erosion, resulting to the permanent damage of engine components due to the abrasive removal of blade surface material.

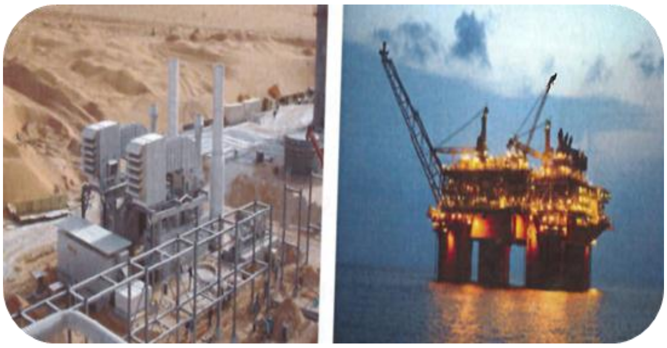

Fig. 2. Gas turbines operating in a desert (left) and offshore (right) environment [12].

D. Effect of Fouling on Overall Gas Turbine Performance

The issue of fouling and its effect on gas turbine performance has been discussed widely by many authors [13-16; 16]**.** According to [13]**,** compressor fouling affects both compressor flow capacity and efficiency. The deposition of particles in the gas path reduces the airflow and inhibits the performance of the gas path components [17]. According to [15]**,** the build-up of material on aerofoil surfaces leads to increased surface roughness and to some degree, changes the shape of the aerofoil especially if the material build up forms thicker layers of deposits as shown in
Fig. 1. Gulen et al [18] notes that the deposits can result in a reduction of compressor mass flow rate, pressure ratio and overall cycle efficiency which in turn causes a drop in gas turbine power output and an increase in heat rate as shown in *Fig. 3*.

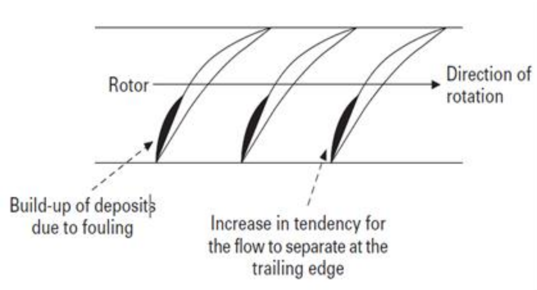

Fig. 3. Build-up of deposits on compressor blade profile during fouling [13]

E. Performance Recovery

Gas turbine performance optimization has being an important issue in recent years due to increased focus on emission and operating costs. The power demand across sectors where gas turbines are used (industrial, oil and gas, power generation and aerospace) has increased over the years. The increase in power demand forces gas turbines to operate at higher power levels [13]. Fabbri et al [19] investigated compressor performance recovery system. Their work concludes that the main solutions to reduce compressor fouling are the use of inlet air filtration systems, compressor washing and compressor blade coating. This paper will look at compressor washing as means of performance recovery.

F. Compressor Washing

In gas turbine engines, the compressors consume approximately 50 to 60 per cent of the overall cycle energy produced in the turbine [16]. Each cycle consumes very large quantities of air which usually contain small quantities of dust, aerosol and water that pass through the filters and deposit on the blades to cause fouling. To recover performance loss due to fouling, compressors are cleaned to remove the deposited particles [4; 28]. When gas turbine compressors are washed, the efficiency improves, thereby reducing the maintenance cost.

1) Methods Of Compressor Cleaning

Several approaches have been employed over the years to clean fouled compressors. The three main approaches to compressor cleaning are abrasion, hand cleaning and solvent cleaning [20-22]. Currently, the two methods of solvent cleaning are online and off-line cleaning [23]. These two methods of compressor washing will be discussed briefly in the following paragraphs.

a) Off-Line Cleaning

Off-line washing also known as soak or crank wash is done when the gas turbine is shut down or the engine is run at sub-idle shaft speeds, with only the starter motor turning the engine. The process starts by injecting the washing fluid into the engine and leaving the fluid to soak on the blades for a short period of time during which the solution can work on dissolving the contaminants. Afterwards, several rinse cycles (with demineralized water) may follow. Each rinse cycle involves the acceleration of the machine to approximately 20–50 per cent of the starting speed, after which the machine is allowed to coast to a stop [4].

These compressor washing procedures are generated by the engine manufacturer and are included in the Engine Maintenance/Service Manuals. For most turbine engines these procedures are similar in concept and practice. However, the mode of engine application determines choice of solvents and many other service features can vary from one engine manufacturer to the other. It may even vary within the range of engine models produced by the same manufacturer [32]. The effect of off-line cleaning can better be appreciated by comparing a fouled and a washed compressor as shown in
Fig. 4.

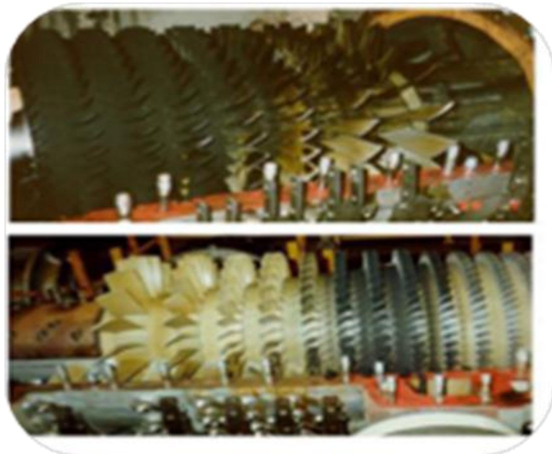

Fig. 4. A fouled versus washed engine [24]

b) On-line Cleaning

An on-line cleaning is a solvent cleaning method that can be performed without shutting down the gas turbine plant by injecting the cleaning solution into the air-intake of the compressor, hence avoiding the associated downtime cost [19; 25]. For industrial gas turbines, on-line cleaning is carried out with the variable inlet guide vanes (VIGVs) close to a fully opened position [21]. This method is considered to be the most advanced because it is performed when the engine is under full load without any significant reduction in the engine capacity and speed [4]. It should be noted that online washing is only used as a means to control fouling from developing. The on-line wash system is therefore intended as a supplement to the off-line crank wash system and not as a substitute [26]. Both compressor on-line and off-line washing systems are usually installed at the air inlet of the gas turbine facing the compressor front blading [21].

II. METHODOLOGY

The idea of this research is to assess the influence of fouling on the creep life and performance of a gas turbine operating in a load following mode. This is with a view to using appropriate compressor washing techniques to recover and enhance the performance and the creep life of gas turbines. To achieve this, an engine performance model was created using TURBOMATCH [27] . Thereafter, a physics-based creep life model was developed and the data from the performance model was used for the assessment of the creep life. HPT blade of a two-shaft aero derivative model gas turbine engine is selected as the creep life limiting component of the gas turbine. Two types of compressor washing techniques are employed to assess their impact on the performance and creep life recovery on a deteriorated gas turbine. Consequently, an algorithm was developed which could be used to estimate the compressor washing interval that gives the best operational and economic benefit.

A. Performance Model And Blade Geometry

TURBOMATCH [27] was used to create a performance model based on the engine configuration shown in *Fig. 5*. The performance simulations provided the basic parameters of the thermodynamic cycle for the model engine under investigation.

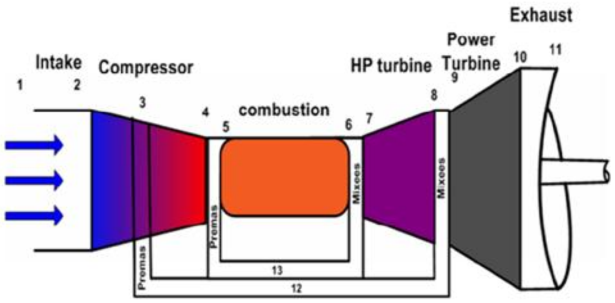

Fig. 5. Layout of a 2 shaft aero-derivative engine

The engine performance parameters used for this study are listed in Table I [28]. Using the results from the performance simulations and available information from open literature, the first stage of the HPT blade was sized using the constant mean diameter method [29].

Table I. Engine Performance Parameters

Parameter	Value	Unit
(a)	(b)	(c)
Pressure ratio	23.1	m
Exhaust gas flow rate	84.2	Kg/s
Power Output	27.6	MW
Thermal Efficiency	28-42	%

This method was adopted in order to satisfy the condition of constant mass flow per unit area at all blade radii [29]. The blade geometry specifications at the mid-height are presented in Table II.

Table II. Parameters at blade mid height

Parameter	**Value**	**Unit**
(a)	**(b)**	**(c)**
Spacing	0.0234	m
Optimum axial solidity	1.00	
Axial chord	0.0234	m
Axial aspect ratio	1.4	
Stagger angle	35	degrees

Using the information from blade sizing model, CATIA which is computer aided design (CAD) software was used to obtain the blade shape as shown in *Fig. 6*. Cooling passages are introduced to depict a cooled blade.

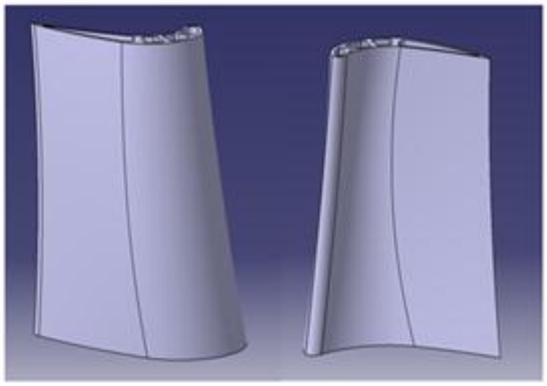

Fig. 6. Three- dimensional blade model showing pressure and suction side.

B. Blade Creep Life Estimation Model

To enhance the performance and creep life of a gas turbine operating in a load following mode, it was imperative to develop a creep life assessment model that could be used for the analysis. Consequently, a physics based creep life assessment model was developed and applied on the HPT first stage rotor blade shown in *Fig. 6*. The creep life assessment model was simplified using an analytical approach. The model consists of three main sub models which include stress, thermal and life estimation models. *Fig. 7* shows the methodology used for the creep life estimation model.

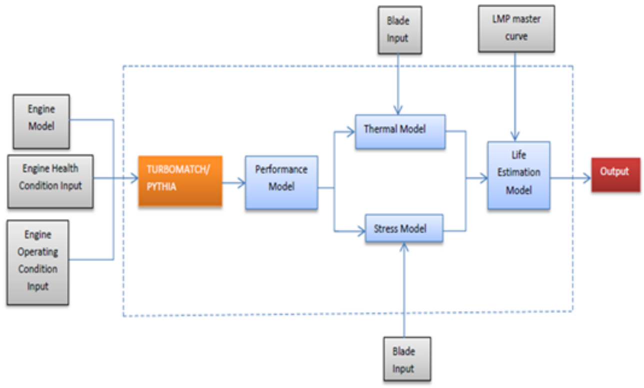

Fig. 7. Blade Life Estimation Model Methodology

C. Stress Model

The details of the stress model are presented in [30]. The stress model calculates the total stress acting on the blade. The two main sources of stress considered are (i) stresses due to centrifugal load caused by engine rotation and (ii) stresses due to gas bending momentum. The thermodynamic data such as rotational speed, temperature and pressures were calculated by TURBOMATCH. The variations of the blade stresses are predicted at locations along the blade span and chord.

D. Thermal Model

The thermal model was developed and used to estimate the blade temperature. The blade is regarded as a heat exchanger which is subjected to a mainstream of hot gas flow from the burner. The main parameters of the model are the cooling methods, blade geometry, TBC thickness, heat transfer coefficients, gas properties, radial temperature distribution factor (RTDF), blade material etc. Similar to the stress model, the thermal model uses a two-dimensional approach to evaluate the temperature variation at each blade section. Full details of the thermal model used in this paper are presented in [30].

E. *Creep Model*

The Larson Miller Parameter [31] is used to evaluate the creep life. The blade creep life varied at each blade section due to changes in blade temperature and stresses across the blade span. The minimum blade life is the lowest creep life of any individual blade section considered. The LMP equation is expressed as:

$$\mathrm{LMP} = \frac{T}{1000}(\mathrm{Log}\,t_f + \mathrm{C}) \qquad (1)$$

where T is the temperature of the material, t_f is the time to failure and C is a constant which is often generalized as 20 in most industrial applications. In this work, 20 was used for C. When equation 1 is re-arranged:

$$t_f = 10\,(\frac{1000\,LMP}{C} - \mathrm{C}) \qquad (2)$$

The creep life is different for the different blade sections due to the fact that the stress varies along the different blade sections. The blade's remaining life will be the calculated minimum creep life. The creep process is a continuous deformation process that often occurs at high temperature due to the stresses that are induced in the hot gas components over a period of time.

III. RESULTS/DISCUSSION

In order to study the impact compressor fouling on the performance and creep life of a gas turbine operating in a load following mode, the model aero derivative gas turbine engine shown in *Fig. 5* was used for all the cases considered. Also, different degradation indexes ranging from 1% to 4% as explained in Table III are used to examine the impact of various levels of compressor fouling on HP turbine blade creep life and other important GT performance parameters. The Foul Index values were chosen based on typical values used to represent compressor fouling [29; 32].

Table III. Classification of fouling index

Fouling Index (%)	Change in mass flow capacity (%)	Change in isentropic efficiency (%)
1	-0.5	-1
2	-1	-2
3	-1.5	-3
4	-2	-4

A. *Impact Of Compressor Fouling On Gas Turbine Performance Parameters*

The effect of compressor fouling on the performance of the model gas turbine engine was studied and the results are presented in *Fig. 8*. The results illustrate the percentage change of the monitored parameters from the clean engine.

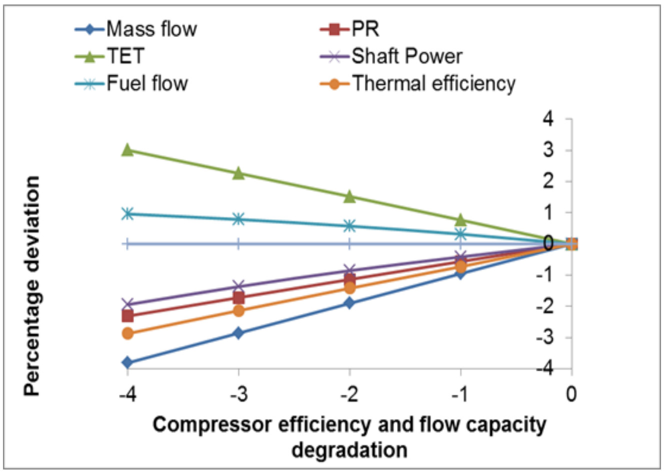

Fig. 8. Effect of compressor degradation on gas turbine performance parameter

It could be observed from the results that compressor fouling reduces the shaft power, thermal efficiency, mass flow and pressure ratio whereas the TET and fuel flow increased significantly. The highest deviations were the mass flow and TET. At 4% FI, the results show a reduction in mass flow by 3.9%, thermal efficiency by 3%, shaft power by 2% whereas the TET and fuel flow increased by 3% and 1% respectively.

B. *Effect of Compressor Washing on Performance Recovery*

As earlier explained, compressor washing is one of the best ways of recovering performance in a deteriorated gas turbine. Therefore in this section, the model gas turbine was deteriorated using TURBOMATCH and the two most common types of compressor washing techniques (online and offline) were employed on the deteriorated engine with a view to recovering performance parameters. The results are presented in *Fig. 9* to *Fig. 13* showing the effect of compressor washing on the performance recovery of deteriorated gas turbines.

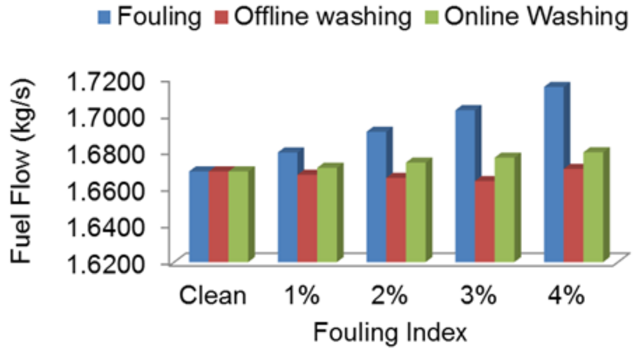

Fig. 9. Compressor against fuel flow

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 08, August-2015

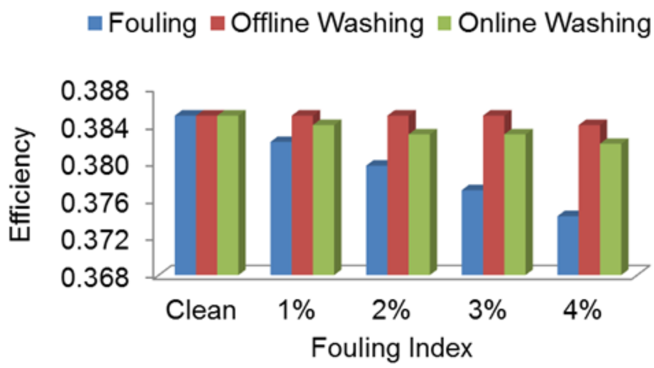

Fig. 10. Compressor washing against thermal efficiency

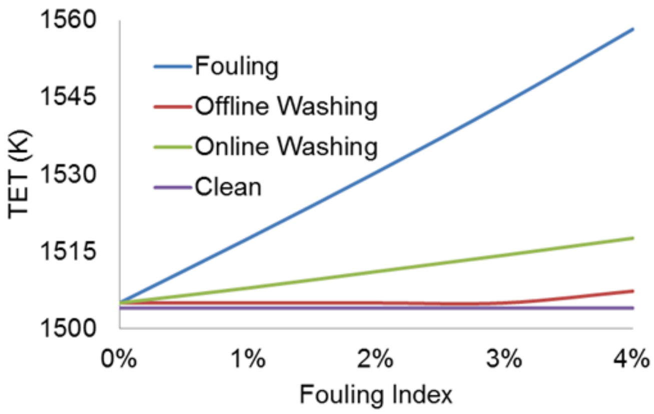

Fig. 11. Compressor fouling against TET

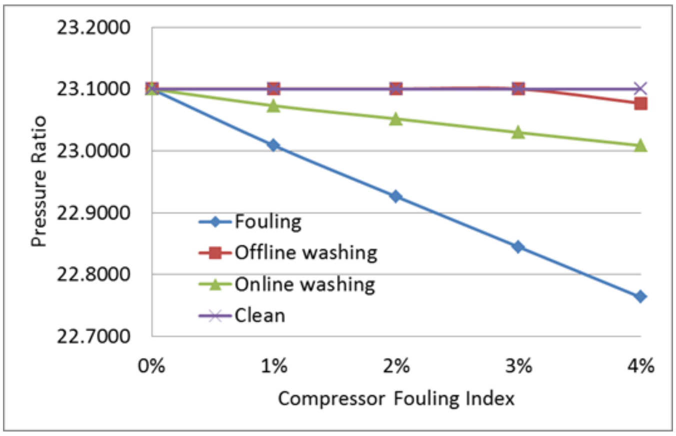

Fig. 12. Compressor washing against pressure ratio

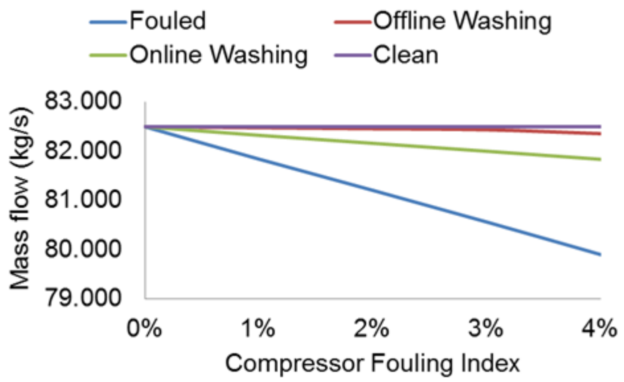

Fig. 13. Compressor washing against mass flow

It can be seen in the figures above that Compressor washing has the ability to recover lost engine performance parameter. The offline washing method showed a higher recovery rate than the online washing. For instance in *Fig. 11*, it is observed that at FI of 4%, compressor fouling increased the TET from 1505K to 1558K representing an increase of about 3.5%. Offline compressor washing reduced the TET to 1507K which represents about 95% recovery whereas online washing reduced the TET to 1518 representing about 75% recovery. Similarly, in Fig. 9, the result shows that for a FI of 3%, fouling increased the fuel flow from 1.669kg/s to 1.702kg/s. Online compressor washing reduced the fuel flow to 1.68kg/s whereas offline reduced the fuel flow by 1.66kg/s. As shown above in the results, washing offline shows a great improvement in performance recovery than online washing.

C. Effect of Compressor Fouling on Blade Metal Temperature

The impact of compressor fouling on the blade metal temperature of the model gas turbine was conducted and the results are presented in *Fig. 14*. In comparison with the reference condition (DP), compressor fouling increases the blade metal temperature. The results were consistent for all the different points on the blade span and the severity increased as the Fouling Index was increased. The results show that compressor washing reduced the blade metal temperature with offline washing showing better performance than online washing.

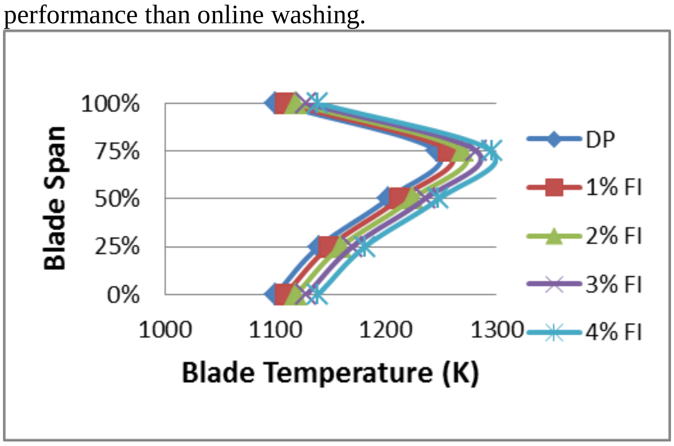

Fig. 14. Effect of compressor fouling on blade temperature

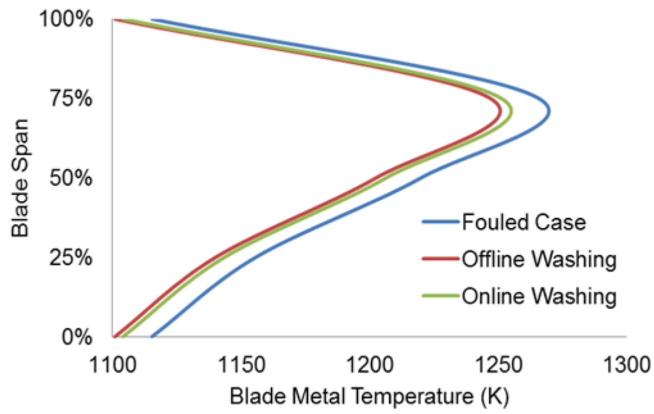

Fig. 15. Blade metal temperature as a function of blade aerofoil span

D. *Effect of Compressor Washing on Creep Life*

The effect of compressor fouling on the creep life of the model gas turbine engine was investigated and the results are presented in *Fig. 16.* It can be seen that compressor fouling reduced the creep life of the blade. At FI of 2%, the creep factor is 0.45 and as the FI increases to 3%, the creep factor reduced to 0.29. The results show that as compressor fouling increases, the creep life consumption also increases.

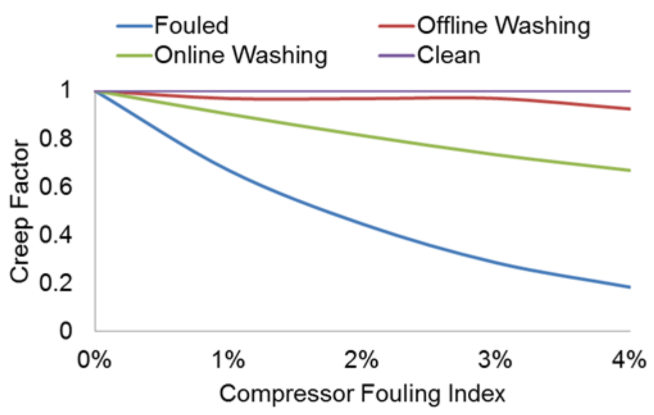

Fig. 16. Compressor fouling on creep life

The results clearly show how compressor washing positively affects the performance. When washed at the appropriate interval, the performance also improves. The offline washing is more effective in performance recovery when compared to the online washing as shown in *Fig. 16*
.

E. *Optimum Time for an Off-Line Compressor Cleaning*

To achieve optimum benefits from compressor washing, it is imperative to select the best washing interval that gives the optimum economic benefit. Hence in this section, an algorithm is presented that may help gas turbine users to understand the beat time to conduct compressor washing.
The effect of non-recoverable degradation is introduced by using a user-specified *recovery factor*, which is given by [18].

$$P_{lost} \;\; \Phi^n P_x - P_a \tag{3}$$

where ϕ is the recovery factor for power output, P_a is the actual measured power output, and n is the number of the off-line compressor wash since on-line monitoring started. Thus, we account for the fact that each off-line compressor wash restores only a fraction (equal to ϕ) of the power output previously restored by the preceding crank wash, and n^{th} off-line compressor wash restores only Φ^n fraction of the original (baseline) power output. A similar consideration applies to the heat rate; i.e. each off-line wash restores the heat rate to a level slightly higher than that restored by the preceding wash. In that case, extra fuel burned can be expressed as:

$$F_{xtra} = f_a - \Phi^n \Phi'^n f_x \tag{4}$$

where ϕ' is the recovery factor for the heat rate, f_a and f_x are actual and expected fuel mass flow rates, respectively. The values of ϕ and ϕ' *as* used by [18] are 0.998 and 1.005, respectively. The average linear rate of change of power lost due to compressor fouling at a given time t can be found from the relation:

$$\Pi = \frac{2}{t^2}\int_0^t P_{lost}(t)dt \tag{5}$$

Similarly, for the average linear rate of change of the extra fuel burned due to compressor fouling is given by:

$$\Phi = \frac{2}{t^2}\int_0^t F_{extra}(t)dt \tag{6}$$

It can then be shown that the optimum time to do an off-line compressor wash is given by:

$$\tau^* = \sqrt{\frac{2C_m}{\Pi C_p + \Phi C_f}} \tag{7}$$

Where C_m, C_p and C_f are cost of maintenance, power sale price and fuel purchase price, respectively, and τ^* is the time from the last compressor off-line wash. These algorithms can be used to continuously update the estimated off-line compressor wash dates as new data is received and processed.

IV. CONCLUSION

In this paper, the impact of fouling on the performance and creep life of a model two-shaft aero-derivative gas turbine operating in a harsh environment and also used for load-following operations was investigated. For this purpose, the environmental effect caused by sand, dust particles, etc. which causes fouling on the compressor blades, as a common cause of gas turbine performance deterioration, on the compressor and power turbine were studied. Results show that fouling had a significant effect on the performance and creep life of gas turbines used in load-following operations. The results obtained were in acceptable agreement with public data when validated. In order to achieve optimum benefits from compressor washing, it is imperative to select the best washing interval that gives the optimum economic benefit. Therefore, an algorithm which aids gas turbine users to understand the best washing interval was presented.

ACKNOWLEDGMENT

The authors will like to thank the Bayelsa state scholarship board (BSSB) and the petroleum technology development fund (PTDF) Nigeria for their total support in sponsoring this research. Also, we wish to thank Prof Pericles Pilidis and Dr Y. Li for their continued support.

NOMENCLATURE

CAD	Computer Aided Design
C_f	Fuel Purchase Price
C_m	Cost of Maintenance
C_p	Power Sale Price
EGT	Exhaust Gas Temperature
f_a	Actual Fuel Mass Flow Rate
FI	Fouling Index
f_x	Expected Mass Flow Rate
GT	Gas Turbine
HPT	High Pressure Turbine
LMP	Larson-Miller Parameter
P_{lost}	Lost Power
RTDF	Radial Temperature Distribution Factor
TBC	Thermal Barrier Coating
TET	Turbine Entry Temperature
VIGV	Variable Inlet Guide Vanes
ϕ	Recovery Factor For Power Input
ϕ'	Recovery Factor For The Heat Rate
P_a	Actual Measured Power Output
P_x	Expected Measured Power Output

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