International Journal of Maritime and Interdisciplinary Research
©2025 IJMIR/Volume 7(3) SE (1) | ISSN: 2734-3286.

Journal homepage: www.ijmir.edu.ng

INVESTIGATION OF THE EFFECT OF COMPRESSOR FOULING ON THE PERFORMANCE AND CREEP LIFE OF A TWIN-SHAFT INDUSTRIAL GAS TURBINE

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ABSTRACT

This study investigates the effect of compressor fouling on the performance and creep life of a twin-shaft industrial gas turbine. Data obtained from the public domain were used to model the twin-shaft engine. The modeled twin-shaft engine was derived from the LM2500 class of gas turbines. Flow capacity and compressor isentropic efficiency reductions of 5% and 3%, respectively, representing typical compressor fouling scenarios, were obtained and used to simulate the effect of compressor fouling on the engine's performance. GasTurb simulation software was employed to analyze the performance of the twin-shaft engine under both clean and fouled conditions. The results show that thermal efficiency dropped from 36.3% in the clean case to 35.4% in the fouled case, representing a 2.5% reduction. Also, mass flow rate and surge margin decreased by 2.6% and 16%, respectively from the clean to the fouled condition. A comparison of turbine blade creep life revealed that the clean engine had a time-to-rupture of 77,681 hours, while the compressor fouled engine had only 13,130 hours.

Keywords: Compressor Fouling, Creep Life, Blade Metal Temperature, Thermal Model, Stress Model

1.0 INTRODUCTION

Compressor fouling is one of the key factors that contribute to degradation in performance of gas turbine plant during operation. The phenomenon of fouling is caused by foreign particles contained in air that find their way through filter systems and stick to the surface of the compressor blades. These particles, depending on the environment, may range from dust and soot to water droplets, hydrocarbon aerosols, pollen and salt. The deposition these foreign particles result in the development of surface roughness on the compressor blades, changes to the shape and/or inlet angles of the aerofoil, and decrease of throat area [1].

This further leads to a decrease in power output when operated under fixed exhaust temperature or increased fuel consumption when operated at a given load requirement as a result of performance deterioration. Loss in power output in a power plant can lead to a reduction in revenue for the plant owner, and can also increase equipment lifecycle cost. The effect of compressor fouling, therefore, is very important when it comes to the overall GT performance. This is because the compressor uses nearly 60% of the work produced by the gas turbine [2].

Meher-Homji and Bromley [3] stated that to sustain high revenue streams in power plants, it is essential for the compressor to be highly efficient. This is because approximately 50 to 60% of the work produced by the turbine is consumed by the axial compressor. Similarly, Diakunchak [4] opined that about 70 to 85% of all gas turbine engine performance loss accumulated during operation is attributable to compressor fouling.

According to Mezheritsky and Sudarev [5] the effect of fouling is not just all about mass flow reduction and efficiency but it also causes a reduction in compressor surge margin. Lakshminarasimha and Saravanamutto [6] 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%. Fouling has a more significant effect on the mass flow than the compressor efficiency. This was shown by Diakunchak [4] in an experimental study of the effect of fouling on a small turboprop engine. Further to this, the author states that the decrease in mass flow changes with the engine operating speed.

The effect of fouling on GTs such as those used for pipe line is mainly on fuel economy unlike those used for hovercrafts or helicopters where the major concern is on power output [6]. This is because the power will not be sufficient for the purpose it is meant. Fouling results in an increased cost of fuel consumption in pipeline operations when running the GT at higher power settings for a very long period of time. These findings were also in agreement with that of Scott [7] in a single Rolls Royce Avon engine used on pipeline system, which states that due to small variations in heat rate; about two million dollars was spent on fuel annually.

From an economic point of view, Meher-Homji and Bromley[3] conducted a study on the impact of fouling degradation on two GTs, frame 5 and 9, which have power output 26 & 225MW respectively that are being operated on natural gas under base load condition. They observed an annual average of 3% reduction in power output and increased heat rate of 1%, assuming it was operated for 8000hrs per year. This translates to a cost of power of five hundred thousand U.S dollars ($500,000) for the 26MW engine and above five million U.S dollars for the 225MW engine per year.

The degradation of the compressor as a result of fouling usually occurs on the surface of the annulus of the compressor blades. The build-up of dirt results in reduction of the throat area which further causes a decrease in Pressure Ratio (PR), isentropic efficiency of the component and mass flow [8]. The effect of fouling on the GT performance can be considered in two different perspectives. Firstly, considering a situation where fouling causes a decrease in power due to mass flow reduction when it is operating at constant Turbine Entry Temperature (TET). The second aspect is where TET is being increased as result of fouling in order to maintain a constant power. This would result in an increase in fuel flow, emissions and surge margin. The increase in TET would definitely result in a shortening of the component creep life.

Syverud et al. [9], in their study on the deterioration caused by injecting saltwater into an axial compressor, reported that the salt-induced deterioration led to an increase in the sidewall boundary layer and significant deterioration in the stage characteristics of the first four stages.

Agbadede et al.[10] employed an accelerated method of creating compressor fouling by uniformly applying a sticky agent to the blades, followed by ingesting flour particles into a compressor

cascade rig to investigate the efficacy of different wash agents in cleaning fouled compressor blades. The study showed that compressor fouling on the blades causes an increase in the percentage drop in thermal efficiency. In addition, the authors reported that the percentage increase in fuel flow and TET correlated with the extent of compressor fouling.

Fouflias et al. [11] investigated the effect of fouling on two-dimensional compressor blades. After attaching sticky tape to the three middle blades, Carborundum particles ranging in size from 63 to 254 μm were applied to create a uniformly distributed roughness, thereby degrading the performance of the compressor in terms of the total pressure rise coefficient. The study established that increased roughness correlated with an increase in the loss coefficient, which was attributed to the reduction in blade-to-blade pitch size, resulting in a reduction in mass flow rate. The authors also revealed that when the roughness level on the compressor blades was increased from a clean case of 0 μm to 254 μm, the compressor efficiency dropped from 0.892 to 0.823, representing a percentage decrease of 7.7%. From the foregoing, it is clearly evident that many studies have been conducted on compressor fouling. However, there are few available that have investigated the effects of fouling on creep life. This study investigates the effect of compressor fouling on the performance and creep life of a twin-shaft industrial gas turbine

2.0 MATERIALS AND METHODS

2.1 Engine Performance Simulations

GasTurb simulation software was employed to simulate the effect fouling on the overall performance of the gas turbine. GasTurb simulation Software employed in this study utilizes predefined engine configurations, thus allowing for an immediate start of calculations. GasTurb is developed to simplify the analysis of the thermodynamic cycle for both design and off design performance of gas turbines. Moreover, the software applies fundamental gas dynamic principles, integrating thermodynamic equations such gas laws, mass conservation, and momentum conservation to assess gas turbine performance [12].

To examine the effect of compressor fouling on the overall performance of a gas turbine engine, a twin-shaft gas turbine derived from the LM 2500 was selected for the study. The GasTurb simulation software was utilized to model and simulate the effects of compressor fouling on the twin-shaft engine's performance. The LM 2500 engine data obtained from the open domain were used to model the twin-shaft engine under fouled condition investigation. The modelling process involved adjusting component efficiencies, turbine entry temperature, and other engine parameters until key performance parameters such as power output, thermal efficiency, fuel flow, exhaust gas temperature, and exhaust gas flow closely matched those of the original engine.

Consequently, compressor fouling degradation of flow capacity and isentropic efficiency reductions were imposed into the twin shaft engine, to simulate the effect of compressor fouling on the overall performance of the engine. Typical values flow capacity and efficiency reduction, which are indicative of compressor fouling, were imposed on twin shaft industrial gas turbine. Figure 1 shows the twin shaft gas turbine engine configuration adopted for the investigation, while Table 1 presents the design point performance specifications.

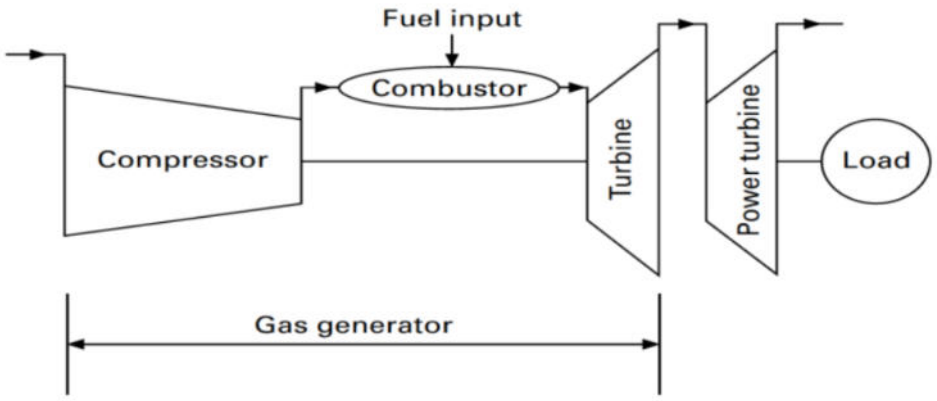


Figure 1: Twin shaft aero-derivative gas turbine engine configuration

Table 1 Engine design specifications

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

2.2 Creep analysis

The approach employed by Agbadede et al.[13] to estimate creep life of turbines was adopted in this study. Five distinct models such as engine performance, blade sizing, stress, thermal, and creep were developed and utilized to predict the creep life of a gas turbine operating under fouled conditions. These models were built in Microsoft Excel, incorporating relevant formulas and assumptions. Estimating the lifespan of gas turbine blades using engine performance, stress, thermal, and creep models is a well-established approach, as numerous researchers have previously employed similar methodologies for turbine life assessment.

Saturday et al. [14] introduced a comprehensive system for estimating creep life that combines various models, including engine performance, blade thermal and stress, creep-life, and data acquisition and preprocessing models. According to the authors, the system was designed to assess and monitor the first-stage turbine rotor blades of aero-derivative engines. In a related study, Eshati [15] also utilized engine performance, blade thermal and stress, and creep-life models to predict the creep life of high-pressure turbine blades.

For brevity, since the engine employed to investigate the effect of compressor fouling on creep life in this study is similar to the one used in Agbadede [13], to avoid repetition, the authors have deliberately chosen only present the equations used to create the above mention models, as the detailed procedures can be found in Agbadede [13]. The equations employed to create the thermal, stress and creep models are presented below:

Thermal Model

Equation (1) was applied to determine the cooling effectiveness (ε), establishing the relationship between gas stream temperature (Tg), blade metal temperature (Tb), and coolant temperature (Tc).

$$\varepsilon = \frac{T_g - T_b}{T_g - T_{c1}} \quad (1)$$

Where ε, is the cooling effectiveness,

Tg is the gas stream temperature

Tb is the blade metal temperature

Tc1 is the coolant temperature

In turbine blades, rotational forces cause average circumferential temperatures to be experienced at a specific radial plane. The Rotor Temperature Distribution Factor (RTDF) measures temperature variation across blade sections and is critical to rotor blade life. Walsh and Fletcher [16] recommended that RTDF values be maintained below 20%. Equation (2) provides the formula for calculating RTDF:

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

Tmax= Circumferentially mean outlet peak temperature

TET = Mean temperature

TET – T3=Mean Combustor temperature rise

Assuming that the maximum temperature occurs at 75% of the blade height, Equations (3) and (4) from Eshati et al. [17], were utilized to determine the maximum and minimum temperatures, respectively:

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

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

The cooling air temperature was considered uniform across all sections of the blade. Blade metal temperature, Tb, was determined using Eq. (5).

$$T_b = T_g - \varepsilon\,(T_g - T_c) \quad (5)$$

The gas temperature, as shown in Eq. (6), is obtained by subtracting the temperature drop caused by the thermal barrier coating (TBC) from the turbine inlet temperature [18].

$$T_g = TET - \Delta T_{tbc} \quad (6)$$

Stress Model

This study focused solely on centrifugal force for the stress model, as it accounts for approximately 70–90% of the total blade stress [19]. For the stress analysis, the blade was segmented into four sections: the stress at the root datum section (RDS), as well as at 25%, 50%, and 75% of the blade height. The centrifugal force acting on the blade was determined using Eq. (7):

Therefore:

$$\text{Centrifugal force, } \quad CF = mr\omega^2 \quad (7)$$

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

Mass of blade = density of blade x volume of blade

Substitute mass of blade into Equation (7) gives Equation (8)

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

$$\sigma CF = \frac{CF}{A} \quad (9)$$

HP Turbine Blade's Creep Life Estimation

To estimate the creep life of the HPT blade for clean and fouled conditions, the calculated stresses at different blade sections were used to derive the Larson-Miller Parameter (LMP) from the master curve of the blade material. Subsequently, the LMP, along with the blade metal temperature at each section, was utilized to determine the rupture time for the various blade sections. For clarity, the blade metal temperature and TET, rupture time and the turbine blade creep life of the clean and fouled cases were considered for the (75 -50% Blade Height) only, which suffers the highest creep life reduction. The estimated time to rupture of HP turbine blades dropped from a clean case of 77,681 hours to 13,130 hours for the fouled condition.

The stresses obtained for the various blade sections were used to derive the Larson Miller Parameter (LMP) from the blade material's Master curve. The LMP and blade metal temperatures were then utilized to estimate the rupture time and creep life under clean and fouled conditions. Equation (10) was used to estimate the time to failure life for the two conditions [18].

$$P = \frac{T}{1000}\left(log_{10}t_f + C\right) \quad (10)$$

Where

T – Temperature K

t_f - Time to failure

P – Larson Miller Parameter

C – Constant.

3.0 RESULTS AND DISCUSSION

3.1 Engine Performance Analysis

Figure 2 shows the plots of thermal efficiency for different operating conditions. From the figure, it can be observed that the engine operated under clean conditions produced a thermal efficiency of 36.3% compared to 35.4% under fouled conditions. This represents a reduction in thermal efficiency of 2.51% from clean to fouled conditions. The reduction in thermal efficiency is attributed to the fouling of compressor blades, which increases the boundary layer thickness due to surface roughness. This roughness results in increased drag and a decreased lift-to-drag ratio on the compressor blades. Consequently, the compressor must consume more power to compress the same amount of air, leading to a drop in the gas turbine engine's thermal efficiency.

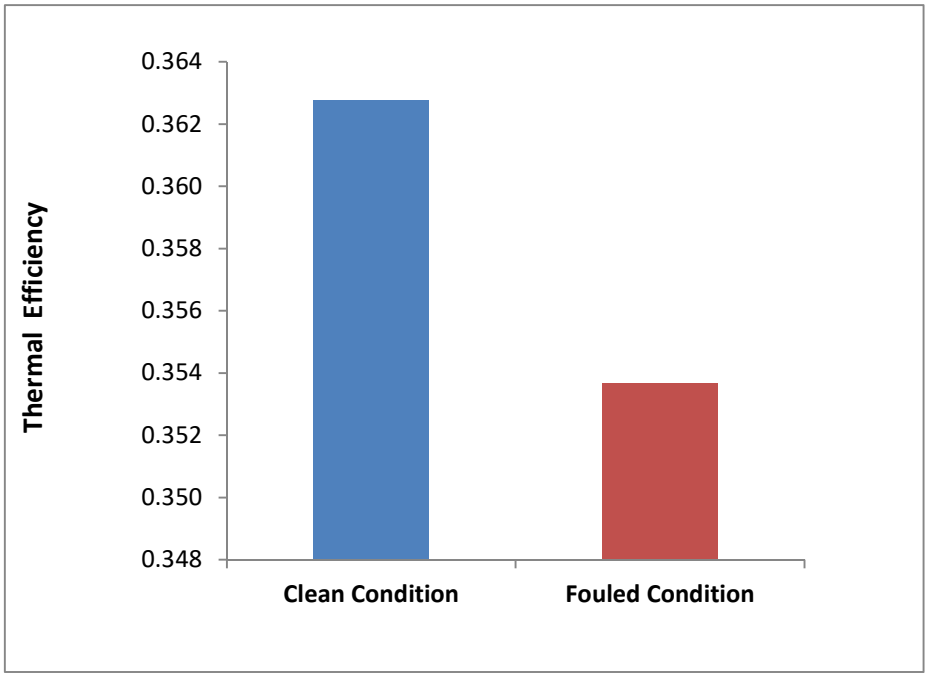


Figure 2: Thermal efficiency against different operating conditions

For the engine flow rate plot shown in Figure 3, it can be observed that the mass flow rate for the engine operated under fouled conditions is lower compared to the clean condition. This reduction in mass flow rate is attributed to fouling, which narrows the flow passages and reduces the compressor's ability to draw in and handle large volumes of air.

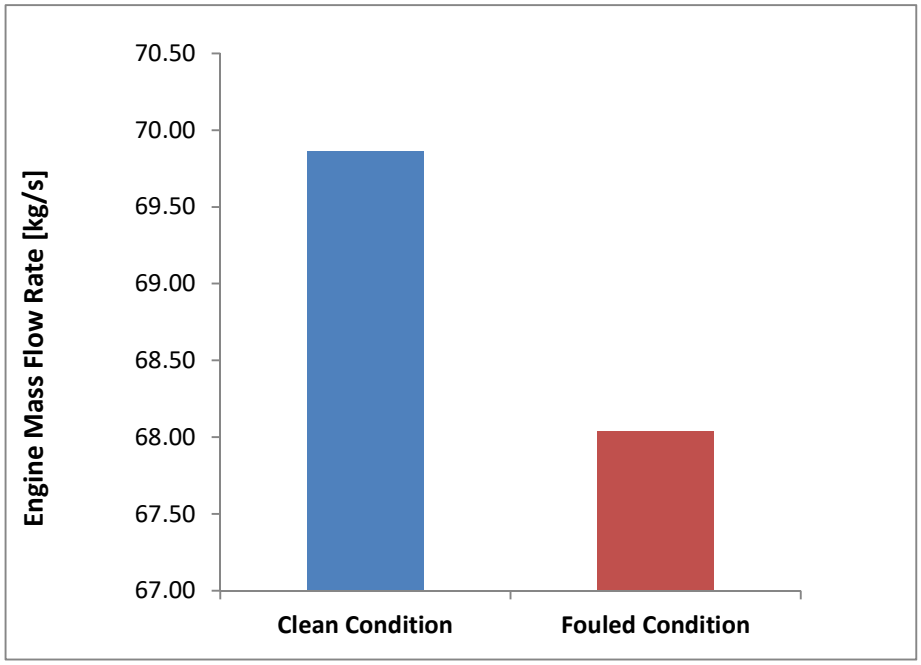


Figure 3: Engine mass flow rate against different operating conditions

Figure 4 shows the plot of compressor isentropic efficiency for different operating conditions. From the figure, it is evident that the compressor isentropic efficiency decreased from a clean condition of 87% to a fouled condition of 83%. This drop in compressor isentropic efficiency can be attributed to compressor fouling, which alters the blade profile and causes increased aerodynamic losses, reduced effective flow area, and flow misalignment, among other factors. These changes contribute to a reduction in compressor isentropic efficiency. These findings align with the study by Fouflias et al.[11], in which the authors reported that compressor efficiency decreased with increased roughness of the blade surfaces, indicating increased fouling.

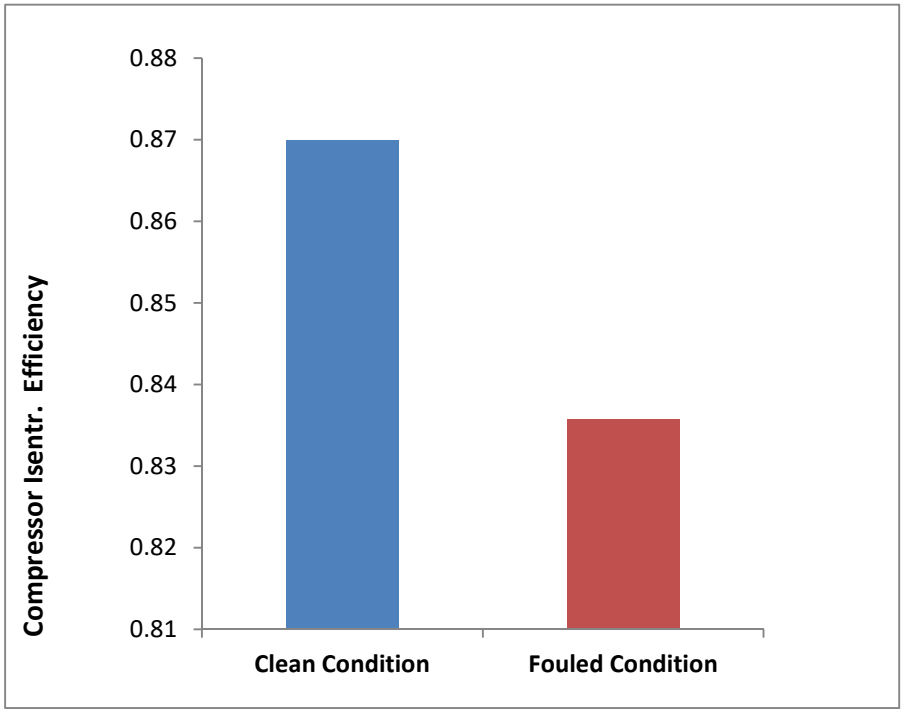


Figure 4: Compressor isentropic efficiency against different operating conditions

Figure 5 shows the plot of Turbine Entry Temperature (TET) for clean and fouled conditions. From the figure, it is obvious that the TET increased from a clean value of 1530K to 1579K for the fouled condition, which translates to an approximate 3.2% increase in TET from clean to fouled condition. This increase in TET is a result of the reduction in mass flow rate caused by increased roughness due to fouling, where fuel flow is increased to maintain the required power output. As shown in Figure 6, when the engine is operated under fouled conditions, the fuel flow increased from a clean condition value of 1.41 kg/s to 1.44 kg/s for the fouled condition, representing approximately a 2.6% increase in fuel flow.

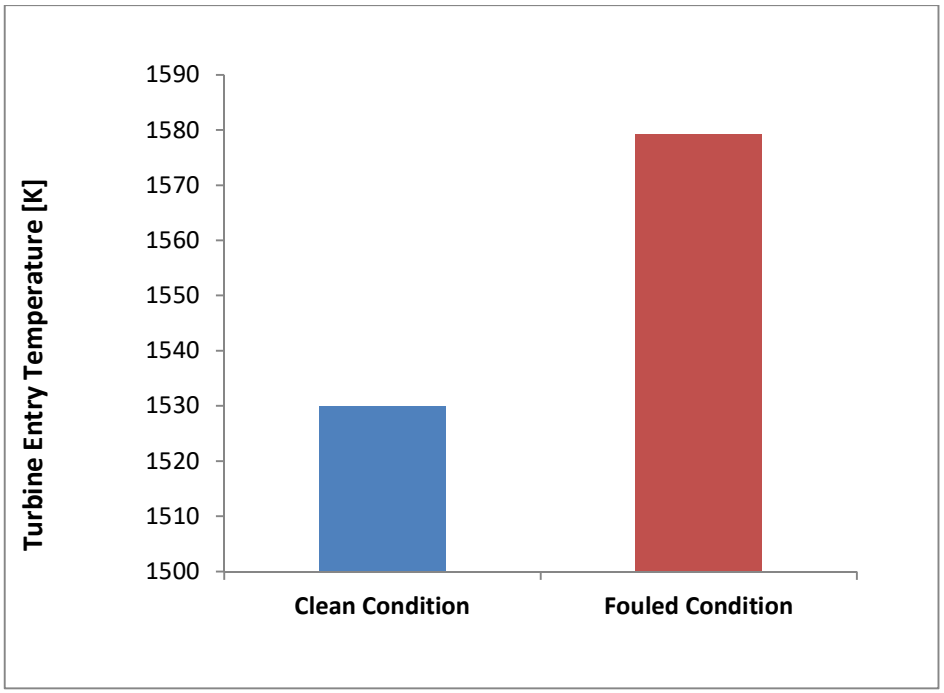


Figure 5: Turbine entry temperature against different operating conditions

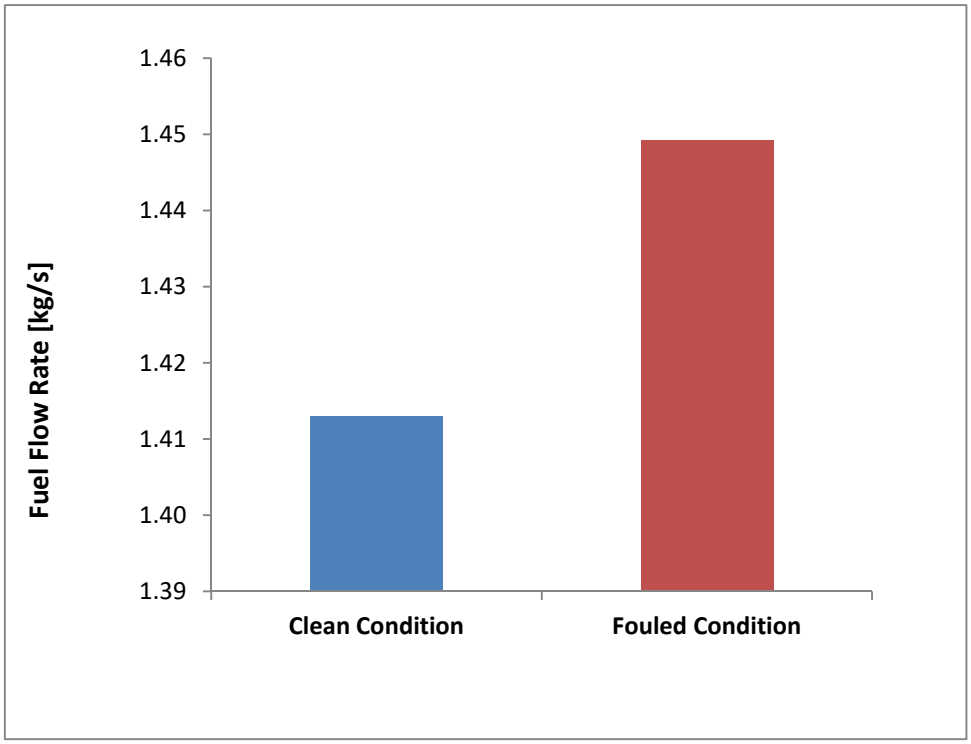


Figure 6: Fuel flow rate against different operating conditions

Figure 7 shows the plot of surge margin against different operating conditions. As can be seen in the figure, when the compressor is operating under fouled conditions, the surge margin is reduced from a clean condition of 24.1% to 20%, which is approximately a 16.1% reduction. This reduction in surge margin from the clean to the fouled condition can be attributed to a reduction in mass flow, which is caused by a decrease in the airflow passage resulting from compressor fouling.

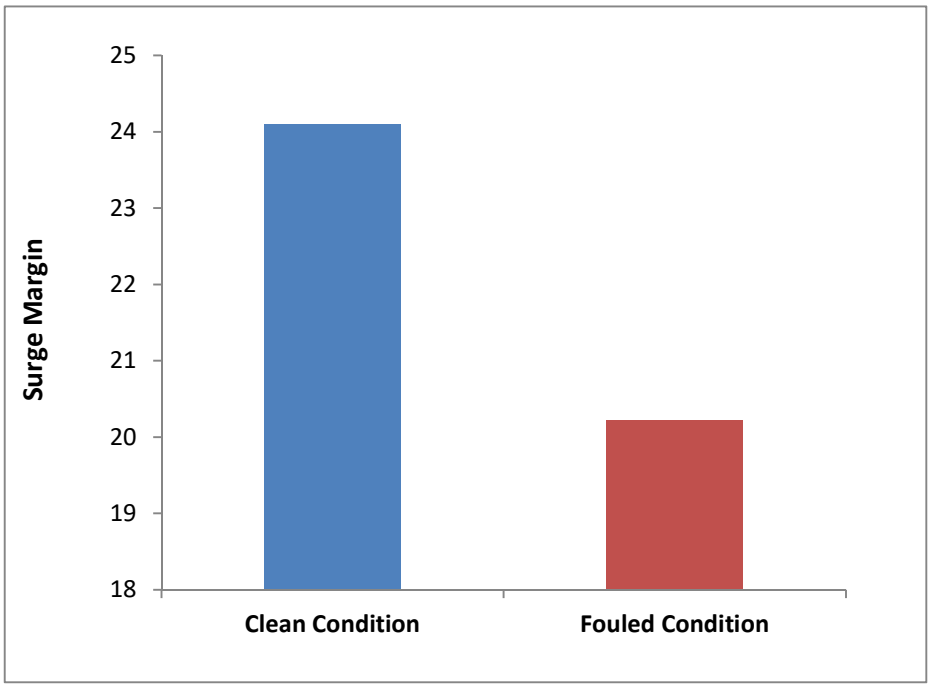


Figure 7: Surge margin against varying different operating conditions

3.2 HPT Blade Creep Life Estimation

Figure 8 illustrates the estimated time to failure of the HPT blade for engines operated under clean and fouled compressor conditions. The results show that the HPT blade has an estimated failure time of 77,681 hours when operated under clean conditions, compared to 33,158 hours under fouled compressor conditions. The shorter creep life of the engine operated under fouled conditions is linked to the higher TET generated due to increased fuel flow resulting from compressor fouling. These findings align with Weber et al. [20], who predicted hot section repair and retirement intervals of 25,000 and 50,000 hours, respectively, for Rene' 80 directionally solidified alloy blades in engines using natural gas. Additionally, the estimated failure times in this

study fall within the 10,000 to 100,000-hour operating lifespan range for gas turbines noted by Saravanamuttoo et al. [21].

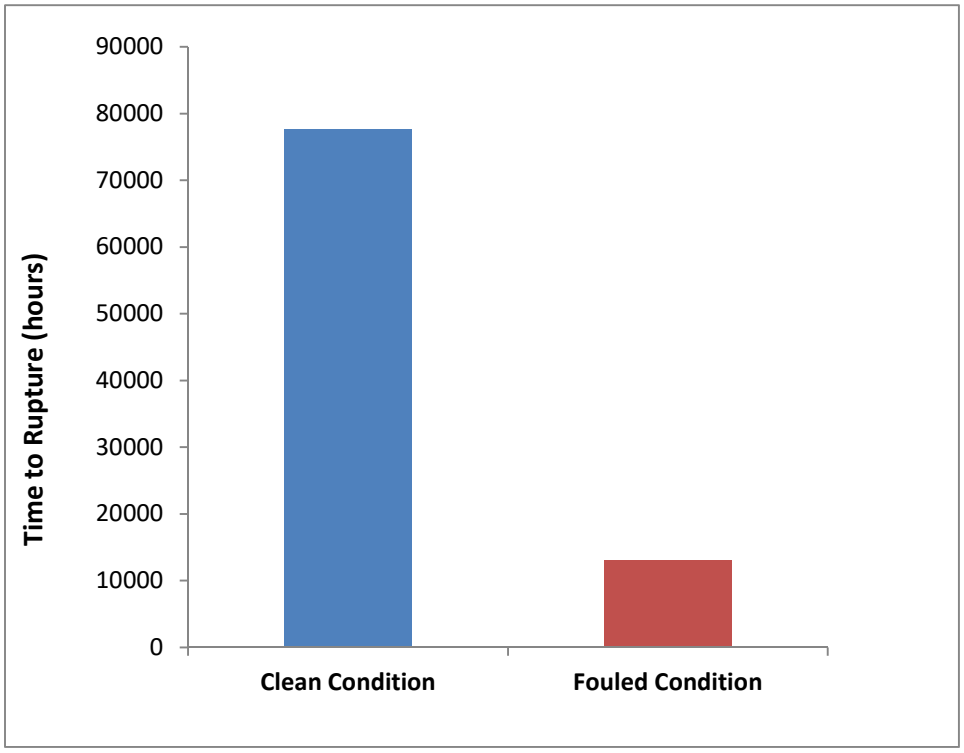


Figure 8: HPT blade time to failure under different operating conditions

4.0 CONCLUSIONS

This study has successfully investigated the effect of compressor fouling on the performance and creep life of a twin-shaft industrial gas turbine. By employing GasTurb simulation software and adopting modeling techniques from previous studies, the investigation demonstrated that compressor fouling significantly impacts engine performance parameters, including thermal efficiency, mass flow rate, turbine entry temperature, and surge margin.

The results revealed a 2.51% reduction in thermal efficiency and a notable increase in turbine entry temperature by 3.2% under fouled conditions, indicating the additional strain placed on the engine. Furthermore, compressor fouling led to a reduction in isentropic efficiency and surge margin of

3.9% and 16.1%, respectively. The subsequent creep life estimation highlighted that fouling conditions notably reduce the high-pressure turbine (HPT) blade's lifespan, with a decline from 77,681 hours under clean condition to 13,130 hours for fouled condition.

These findings underscore the importance of maintaining compressor cleanliness to preserve engine performance and extend the operational life of turbine components. The study demonstrates the efficacy of combining performance, thermal, stress, and creep models for predicting turbine blade life, providing valuable insights for gas turbine maintenance and operational strategies.

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