CRANFIELD UNIVERSITY

JAMES DIWA ENYIA

GAS TURBINE PERFORMANCE ENHANCEMENT THROUGH FAULT DIAGNOSTICS AND COMPRESSOR CLEANING

SCHOOL OF AEROSPACE TRANSPORT AND MANUFACTURING

PhD
Academic Year: 2015-2016

Supervisor: Dr. I. Y. Li and Prof. P. Pilidis

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This thesis is submitted in partial fulfilment of the requirements for the degree of PhD

ABSTRACT

Due to harsh environmental conditions, industrial gas turbines are subjected to fouling during operation, and the total output is thereby affected as a result of this fouled component parts. The fouling considered in this research is at the compressor as a result of ingested particles from the operating environment. It has become imperative to develop effective models for optimisation of power plant in this environment, especially operational strategies for off-design conditions specifically for gas turbine engines. Optimisation of the engine profit is necessary for adequate and integrated evaluation of many important constraints such as; engine performance and efficiency, present and future revenue generated, availability and fuel price, selling price of electricity, engine life cycle cost, and future dynamics of the market.

The state-of-the-art contribution of this research is the application of the method to both maximise total profit and usage availability of a particular gas turbine engine to generate power. In this method, the user is allowed the opportunity to select compressor washing duration and intervals considering the ambient operating condition of the engine. The optimiser employed in this research is a linear programming model that is powerful and capable of determining the maximum and minimum (optimal) values of the net profit of the project in question.

Performance simulation was carried out with the help of Pythia and Turbomatch software of Cranfield University, and the output result appears almost very accurate since the deviations from the values gotten from public domain are very close. As such, the engine was validated and further used for the rest of the research. Because the number of variables involved in carrying out the optimisation process is very large, it becomes very difficult to carry out the optimal configuration. For this reason, the research is divided into phases to ascertain that all necessary factors are critically considered before the optimal value is attained. The engine performance simulation was first achieved, so as to determine the major engine parameters mostly affecting off-design performance. The second phase was the presentation of a mathematical

correlation known as the Larson-Millar Parameter (LMP), which was used to determine the creep life of the turbine blade as a result of the effect of very hot air hitting the blades from combustion chamber. In this regards, the LMP was used to determine the life of the blade along the section. Sub-models known as the stress and thermal models were also incorporated to determine the stress along the different sections of the blade. The remaining creep life investigated was later converted to creep cost based on the equivalent hour of the lost life of the high pressure turbine (HPT) blade. Next was the emission model, which was applied to determine the amount of emission index exerted to the environment during the engine operation. The emission investigated was converted into cost based on emission tax rate.

Hence, the three models mentioned above were incorporated into the economic model for gas turbine off-design performance analysis. The model includes life cycle cost assessment such as; maintenance and operating cost, fuel cost, and creep cost, emission and other taxes. Including total revenue which makes it possible to develop a model which will enable maximisation of total profit under variable operating conditions. Afterwards, the optimiser was introduced, and the optimiser was linked with Turbomatch library output result for the particular engine in use. The tool uses an evaluation of the fitness value of the objective function and takes into account the optimisation constraints.

A sensitivity analysis was introduced lastly to investigate the gravity of each of the effect of the necessary factors mitigating the successful output and net profit of the engine performance. It was observed that the total profit is affected by; rate of dirt deposit on the turbine blades, price of fuel, price of electricity, and number of hours spend during shut-down for off-line compressor water washing.

Keywords: Compressor Washing, Creep Life, Emission, Fouling, Optimisation, Performance

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Above all, I dedicate this research success to the almighty God, without whom all effort would have been abortive. Thank you lord.

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LIST OF ABBREVIATIONS

ABC	Aqueous Based Cleaning
AFT	Adiabatic Flame Temperature
APS	American Physical Society
BMT	Blade Metal Temperature
BTU	British Thermal Unit
C	Capital and Cost of operations
CDP	Compressor Delivery Pressure
CF	Centrifugal Force
CFD	Computational Fluid Dynamics
CG	Centre of Gravity
CHP	Combine Heat and Power
CO	Carbon Monoxide
CO_2	Carbon Dioxides
DLE	Dry Low Emission
DLN	Dry Low NOx
DOD	Domestic Object Damage
DP	Design Point
EGT	Exhaust Gas Temperature
EI	Emissions Index
EINOx	Emissions Index for NOx
EIUHC	Emission Index for Unburned Hydrocarbon
$EICO_2$	Emissions Index for Carbon Dioxide
EICO	Emissions Index for Carbon Monoxides
FOD	Foreign Object Damage
GE	General Electric
GGT	Gas Generator Turbine

GPA	Gas Path Analysis
GT	Gas Turbine
HPC	High Pressure Compressor
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator
IGT	Industrial Gas Turbine
KWH	Kilowatt Hour
LCC	Life Cycle Cost
LM	Land Marine
LMP	Larson-Miller Temperature
LNG	Liquefied Natural Gas
MSEK	Million Swedish Krona
MW	Mega Watt
MWH	Mega Watt Hour
NASA	National Aeronautics and Space Administration
NDMF	Non Dimensional Mass Flow
NGV	Nozzle Guide Vein
NOx	Oxides of Nitrogen
NPV	Net Present Value
ODP	Off Design Point
OEM	Original Equipment Manufacturer
O & M	Operations Maintenance
P_{amb}	Ambient pressure
Ppmv	part per million by volume
RDS	Root Datum Section
RTDF	Radial Temperature Distribution Factor
SCC	Society of Cosmetics Chemist
SCR	Selective Catalytic Reduction

SEK	Swedish Krona
SFC	Specific Fuel Consumption
SHP	Shaft Horse Power
SOx	Oxides of Sulphur
SPS	Shaft Power Speed
T_{amb}	Ambient Temperature
TBC	Thermal Barrier Coating
TET	Turbine Entry Temperature
TERA	Techno-Economic Environmental Risk Analysis
TDS	Top Datum Section
TP	Total Profit
TR	Total Revenue
UHC	Unburned Hydrocarbon
US	United States

1 INTRODUCTION

Gas turbines (GT's) are used for various applications including aeroplane, ship propulsion, power generation and its likes. In all the aforementioned, industrial gas turbines have found applications in various industries world-wide and the demand for this engine is growing, as it is very fast at start-up and able to react quickly to electricity demand fluctuations. These engines have been ensured from research and development to have an economically viable life [1]. The oil and gas industry is a sector where GT's have found wide and increasing application for generating power, and this increasing popularity has triggered the necessity to investigate the means by which these engines can be operated at the least possible cost, as it is liable to deterioration due to several inevitable problems surrounding it successful operation.

The GT industry for power generation has been improved geometrically due to advances in science and technology into a complex environment that no longer operates from the conventional philosophy of demand and supply. The deterioration characteristics of this engine due to environmental and operating conditions, together with the increasing fuel costs globally, has further call for the need to operate the systems more economically without compromising on the supply stability and reliability. It is crucial to consider the safety of the operator and the equipment, as failure of the engine component parts could lead to severe damage of lives and engine components. In addition, costs accrue as a result of component deterioration can be minimised by adopting efficient compressor water washing method.

Compressor fouling is inevitable for GT engines as the environment where it is being operated have made it susceptible to degradation since they ingest some particles alongside air during suction. These particles adhere to the compressor blades and causes fouling. Gas turbine component can also be damaged as a result of contaminants such as; foreign object damage (FOD) and domestic object damage (DOD), particles of sands present in atmospheric air, abrasion, corrosion, and erosion. Amongst all, compressor fouling has been found to constitute the highest amount of loss to the engine power output because it occurs on the surface of the component, system or plant performing a defined and useful function and the fouling process impedes or interferes with the function. The maintenance practice and location of the engine determines the rate at which the engine can be fouled.

Fouling is also caused by contaminant deposits in the presence of moisture and/ or oil on compressor blades and annulus surfaces. This in turn reduces the compressor efficiency and flow capacity, and causes GT engine overall performance reduction in the form of power loss. Increasing TET to regain lost power will reduce blade creep life, increase fuel consumption, increase emission and maintenance costs, and there will be vibration problems and a tendency towards compressor surge, which will all lead to economic problem.

The operating environment such as ambient temperature, altitude and humidity also affect the performance output of the engine and influences fouling to a noticeable extent.

This research finding encompasses various causes of fouling and novel approach in preventing the problems. The HPT blade creep life was investigated by integrating the stress and thermal, and the emission cost were also investigated with result derived from the in-house performance model known as Pythia and Turbomatch. The economic module was integrated to determine the washing methods with the least cost suitable for carrying out GT maintenance.

1.1 Background

For economic viability of industrial gas turbine (IGT) power output, it is necessary to understand the performance under varying operating conditions and other constraints that might affect power generation. This will help identify the best practice and advice for the GT owner or operator, as it reduces operating cost to optimum and ensure availability of the (GT), thereby maximising utmost satisfaction for the user. For example, as an IGT operator, it is necessary to carry out compressor washing either online or offline. The frequency (how often) and effectiveness of the wash that reduces operating cost and yield maximum power output will be adopted as most viable. There are some constraints that are likely to affect the smooth operation of such an engine as stated below, and thus need to be investigated.

1.1.1 Technical Constraint

Improving the efficiency of GTs requires running the engine at high turbine entry temperatures (TET). This high TET in turn subject the hot section of the engine to higher temperatures and stresses, thereby exposing the turbine blades to several failure mechanisms such as low and high cycle fatigue, thermal stress and creep life deformation [2].

Creep is among the most common failure mechanism leading to the reduction of high pressure turbine (HPT) blade life in industrial gas turbines (IGTs). The operating condition, mode of operation, design parameters and hot section components determines the creep failure mechanisms [3] . The aerodynamic and structural requirements determine the design parameters of such components, while life assessment has always been vital to GT owners and users for economic and safety reasons. This is because underestimating the life of the blade will mean changing the component before it due date, thereby wasting money, and overestimating the real blade life can also cause accidents and economical losses. The thermal and mechanical stresses and design measures of the foreseen base load are the basis used by the original equipment manufacturer (OEM) to calculate the life limits. These are functions of the operating environment and potentiality of the material within the given conditions, but the OEM advice will not specifically state the operating environment and requirement of each operator. To this regard, it has become imperative to know the operating and health condition of an engine, its performance parameters, and its effects on creep life [4]. Incorporating the lifing module based on the thermal and stress variations using the Larson-Miller Parameter (LMP), has made it possible to determine the required time to failure of the given engine HPT blade life [4-8].

1.1.2 Environmental Constraint

All combustions systems including those in GT produce pollutants such as oxides of nitrogen (NO_x), carbon monoxide (CO) and unburned hydrocarbon (UHC). NO_x formation occurs due to the high combustion pressure and temperatures that prevail, resulting in the oxidation of atmospheric nitrogen. The formation of CO and UHC is generally due to poor combustion efficiencies. NO_x has been associated with the formation of acid rain and smog, and it has been associated with the depletion of the ozone layer. CO is a poisonous gas and UHC is not only toxic but also combine with NO_x to produce smog. Combustion systems that use hydrocarbon fuels produce carbon dioxide (CO_2) and water vapour (H_2O) due to the oxidation of carbon and hydrogen. Although CO_2 and H_2O are considered non-toxic, they are greenhouse gases and have been associated with global warming [152].

The need to reduce emissions is now of paramount importance in protecting health and the environment. The last decade has seen a rapid change in regulations for controlling gas turbine emissions. Such regulations have resulted in the development of dry low emission (DLE) combustion systems and today, many gas turbines operate using such combustors.

The power output from the GT is controlled primarily by the amount of fuel that is burnt in the combustion system. Excess or uncontrolled fuel addition results in overheating of the turbine and over-speeding, which can seriously damage the engine. It is the responsibility of the engine control system to prevent any engine operating limits from being exceeded. However, in the process it should not compromise the performance of the gas turbine.

The more fuels consumed leads to economic disaster. As such, since the fouling at the compressor is constituted mainly from operating environment, it has become imperative to carry out washing of the compressor as this is the component part of the engine that is most prone to fouling.

To arbitrate the best approach to carry out the most economically viable compressor washing methods, which will meet a wide range of emission legislation scenarios, and mission taxation policies with minimum possible cost, a reliable economic model is required.

The effects of changes in operating and maintenance cost at varying operating conditions, based on electricity cost for power generation are considered. This research purpose encompasses comparison of compressor online wash, offline wash, and a combination of both online and offline wash at varying wash effectiveness and frequencies to determine the most viable economical approach to be adopted for operation and maintenance practices in power generation and supply.

The GT performance modelling tool known as Pythia/Turbomatch was implemented for the simulation of the thermodynamic models of the engine to be investigated [9, 10]. Generated data from the simulation was used as input to the economic model, which is similar to that of [11] and include wide descriptions of the incorporated models. The various wash methods and frequencies (how often) were also incorporated.

1.1.3 Economic Constraint

The rate of electricity demand is higher than economic growth globally, and almost about two times the demand for primary energy sources in most countries. Demand for electricity appears set for continuous growth and higher share of the energy market with the current technologies and its applications as described by [12].

It is very important to carry out economic evaluations in IGTs. GT engines which are mostly used for power generation and every aspects ranging from performance to economics including emissions and creep life need to be greatly analysed [12]. The economic issues associated with GT operations have triggered the need for research project to investigate means of minimizing economic life cycle cost (LCC) which encompass both operating and maintenance cost. Integrated systems such as engineering, financial and management, features the wash regimes have been developed, with scenario based studies to quantify potential advantages and problems.

Over the last thirty years, the economics of power generation from GT has developed from the conventional calculation of generating cost from kwh to the enormously intricate analysis of entire cost of operation of the engine. Individual costs within large systems of generating units require that the economic analysis represents both the real operations, technical and financial involvement [13]. There has been development in both increase reliability and durability of power systems. It is expensive to acquire new turbines and the reason is to minimise operating cost for maximum profits. The purpose of research and development for power generation market is to design power plant that will be operated throughout its service life profitably and efficiently [14]. Just like any other investment, the major consideration in GT power plant is the economic benefits and hence there will be comparisons of different wash methods and frequencies to obtain the most economically viable one out.

The initial capital cost in power generation is usually the highest expenditure but the highest operational running cost is usually on fuel. There will be changes in cost of fuel since there will be changes later under different conditions time in operation than during evaluation of the project. It is very important to incorporate the time value of money for such long term projects that is capital intensive with high cost of operation as in the case of electric power generation and supply.

1.2 Research Aim

The research aim is to investigate the most economically viable maintenance approach in terms of compressor fluid wash methods, considering the technical, environmental and economic analysis for gas turbine performance enhancement.

1.3 Research Objectives

- Build engine model using PYTHIA software: show design point (DP) and off design point (ODP) performance considering the variation in operating and ambient conditions.

- Present a diagnostic model (PYTHIA) to examine the engine output at different cases (clean, degraded, and washed engine behaviour).
- Determine the parameter that is most affected by engine fouling.
- To examine the mechanism associated with GT performance deterioration.
- To investigate the effect of compressor fouling on GT performance and high power turbine (HPT) blade creep life.
- Investigate the significance of thermal barrier coating (TBC), root temperature distribution factor (RTDF) and cooling effectiveness on the (HPT) blade life.
- Investigate the emission index ejected and at what level as a result of fouling.
- To determine the effectiveness and benefits of compressor washing.
- Improvise an economic model to investigate the various cost involve in terms of compressor wash frequencies.
- Economic analysis and comparison of the importance of the various compressor fluid wash scenarios on GT performance.
- Optimisation of economic importance of gas turbine operation and maintenance.
- Sensitivity analysis of the various factors mitigating the effective operation of the gas turbine engine for power generation.

1.4 Contribution to Knowledge

The following are contributions to knowledge:

- Present an optimal economic approach in GT maintenance plant layout from engine diagnostics via various wash methods of the component.
- Present a means to determine various cost associated with different engine degradation and emission level.
- Develop a financial optimisation of GT operation and maintenance.

1.5 Thesis Structure

This thesis encompasses several areas regarding GT power enhancement, and for a better understanding, each aspect has been discussed in each chapter for clarity to the reader.

Chapter one is about general introduction on the subject area, key issues about the thesis topic to be discussed, aims and objectives and the contribution to knowledge.

Chapter two is on literature review governing the focus of the thesis. Related works were identified and discussed leading to knowledge gap identification. The key areas such as IGT simulation and performance, creep life, emission, compressor wash techniques, GT economics, and financial optimisation were all discussed.

Chapter three is the methodology applied in this research work. The model used for the GT performance is presented and explained. The creep, emission and economic models and all their sub-models were all presented and analysed showing how they are all linked and incorporated to achieve the thesis aim and objectives.

Chapter four is the full explanation of the GT performance and simulation both at design point and off design performance. Presenting the operating ambient condition and its effects on the various engine parameters such as; thermal efficiency, pressure ratio, mass flow, fuel flow, shaft power, and exhaust gas temperature for the clean, fouled, and different wash frequencies & cases.

Chapter five discusses the creep model and sub-models including the properties of the blade material/ geometry. Effects of RTDF, TBC, BMT, cooling effectiveness on each other, effects of fouling on blade creep life, effect of washing on creep life, comparison of the creep life for clean, fouled and washed engine condition and the estimated creep life of the engine hot section.

Chapter six comprises the emission model with empirical correlations for CO, NOX, and CO_2. Effects of fouling and washing on GT emissions, comparison of clean, fouled and washed effects on engine performance, CO_2 tax and revenue loss from fouling and gain from washing.

Chapter seven involves the economic analysis of various costs and incorporation of all the models. The combined technical and economic analysis of the clean, fouled and washed engine for various wash frequencies & cases is discussed to get the most economically viable means of wash, Including losses and recovery rates of power.

Chapter eight analyses the economic optimisation of all models which include performance, creep life, emissions and the combine technical and economic effects, showing how it is being incorporated and integrated to yield the optimal value.

Chapter nine gives a summary of the entire research and presents the thesis conclusion and recommendations for future work in a similar research area.

2 LITERATURE REVIEW

2.1 Introduction

The performance of GT engines is dependent on the thermodynamics and aerodynamics design of all the component parts as a result of the complexity of the entire frame, operating condition and environment [15-17]. The required standard day conditions for most GT's is 15^0C and 101kpa, which is the design point. The engine efficiency is basically defined by the heat rate which is a parameter used only in the power generation industry, and is the rate of fuel energy input divide by the useful power output. Hence, it is comparable to specific fuel consumption (SFC) but is dependent on fuel calorific value. The high cost of fuel has brought about the need for high engine efficiency.

The compressor, the burner and turbine which are the gas path components, are very reliable. Operating such an engine in an off-design environment with varying temperature and speed, conditions of load, altitude, and sensitivity of the cycle to degradation of engine component, could result in deterioration in engine performance and possibly breakdown [18-23]. GT performance is affected by several factors which include; the mass flow rates of air as it dictate engine performance [24], because obstruction in the smooth flow of this air along the engine will result in engine performance degradation. The compressor pressure ratio (PR_c), the operating temperature of the engine (Turbine Entry Temperature) TET, and the efficiency of the different components of the entire engine affects the performance as well [15]. These are key factors considered during GT engine design. To obtain the most efficient performance needed for a GT, the best suited; air mass flow, pressure ratio and TET is selected.

Gas turbines in operation take in large amount of air from the atmosphere, which enables the production of power output. This air contains contaminants such as soot, sand, pollen and salt [25]. There are usually inlet filtration systems to prevent these contaminants from getting into the engine, but some of it still penetrates into the engine and form deposits, causing wear and also form chemical reaction with the material of the blades and other engine components.

The extent of these interactions depends on the distribution and size of the particles, hardness and the composition of the contaminants [26, 27]. These interactions could cause a reduction in mass flow, airfoils' geometry, and reduced pressure ratio and compressor efficiency which adversely affects the output power, heat rate and thermal efficiency of the GT. One month in GT continuous operation, could possibly result in 5% reduction in power output as presented in [28, 29]. Hence, it is very important to consider the effect of fouling in determining the GT overall performance, since it occupies very high percentage of the entire work produced by the GT [25].

The presence of impurities in the fuel, combustion derived contaminants (ash), quality of water (steam injection), and cooling air contaminants, all contributes to the degradation of the hot end of the engine performance [30, 31]. The reduction in the nozzle guide vain (NGV) area is as a result of the ash deposition, and these affects the matching point of the compressor-turbine, which deviate it from the design point and hence performance loss [32]. Performance loss is known as degradation, which takes place in GT main components. Compressor fouling is said to be more severe though [33, 34]. These components; individually or collectively contributes to the engine degradation as a result of change in flow capacity and efficiency, which in turn leads to increase in engine cost of operation, and reduction in power output. Increasing the shaft speed, and or TET in order to retain the power output as that of the clean engine will shorten the life span of the component parts and possibly lead to GT breakdown.

Considering the high cost and time involved in carrying out breakdown maintenance, it is wise to have vast knowledge about GT inspection as this help the operator in taking necessary maintenance actions in reducing downtime and increase engine availability. The ability to sustain the power output of a GT as that of its design point of reliability, availability and safety is known as maintenance. Examples of the GT condition monitoring system could be found in [35, 36], periodic data gathering measurements from the engine instruments in use and process the information that can improve the GT operation as well as maintain, repair and overhaul [37]. Identifying degradation at the module level requires analysing the entire gas path (compressor, burner, and turbine), excluding the burner as it involve very longer time in service for degradation to be observed. Gas Path Analysis also helps in identifying the tendency of the degradation during engine operation, and also forecast the maintenance action. Hence GPA helps in predicting, identifying, and quantifying engine fault to enable maintenance response before engine breakdown. These in turn extend engine life, reliability, and minimise maintenance cost.

2.2 Physical Fault in Gas Turbine

The individual components and overall performance of a GT can be affected by some number of physical faults as shown in the table 2-1:

Table 2-1: Effect of physical faults on components performance [38].

Faults	**Characteristic Representations**	**Range**
Compressor Fouling	Drop in flow capacity (Γ)	0.0 - (-5.0%)
	Drop in isentropic efficiency (η_c)	0.0 - (-2.5%)
Compressor Erosion	Drop in flow capacity (Γ)	0.0 - (-5.0%)
	Drop in isentropic efficiency (η_c)	0.0 - (-2.5%)
Turbine Fouling	Drop in flow capacity (Γ)	0.0 - (-5.0%)
	Drop in isentropic efficiency (η_t)	0.0 - (-2.5%)
Turbine Erosion	Drop in flow capacity (Γ)	0.0 - (+5.0%)
	Drop in isentropic efficiency (η_t)	0.0 - (-2.5%)
FOD	Drop in η_c and η_t	0.0 - (-5.0%)

2.3 Compressor Degradation

The deposition of ingested dust particles mixed with air causes the compressor to foul, and in turn decrease the isentropic efficiency of the compressor. It has been found that the engine power output and thermal efficiency are greatly affected by fouling [39]. Reference [40], the geometry of the airfoils can be changed by deposition of particles in critical areas and then the flow condition is modified. Similarly, dust accumulation reduces tip clearance and increases surface roughness [41].

The impact of compressor performance degradation mechanisms based on engine power output has been calculated [42, 43]. Reference [44] reported that the reduction in mass flow decreases power output by 5%.

2.4 Fouling

The degradation in flow capacity and efficiency caused by contaminants adhered in GT annulus surfaces and airfoil, is known as fouling [45]. Fouling occurs both in the compressor and compressor-turbine, but research has shown that compressor is more prone to fouling as compared to the compressor-turbine and performance degradation of the engine [38]. The large quantities of air ingested by the engine makes it susceptible to fouling, because the ingested air contains particles of various sizes and shapes. Hard particles like sand, ash, dirt, dust, and soft particles such as unburned hydrocarbons, air-borne industrial chemicals, herbicides, fertilizers and so on causes fouling. Fouling issues could be increased if oil leaks and the oil could glue at the later stages of the compressor. The high temperature can also bake the oil on the blade. The accumulation of material caused by fouling, changes the shape and inlet angle of the airfoil, reduces the airfoil throat opening, and increases roughness on the surfaces [46].

In cases where fouling does not cause destruction to surfaces of the flow path, performance could be recovered through cleaning or washing. Monitoring or quantifying fouling could help in achieving economical and safety benefits of the GT, and hence a washing plan can be determined. It allows the engine operator to shift from conservative periods of washing to on-condition maintenance strategies [32]. This is because too frequent compressor water washing will result in excessive expenses as in down time, increased maintenance cost, and early blade surface erosion. Also, lengthy periods between washing could result in improper performance recovery. A 3% increase in fouling is enough signals for compressor cleaning or washing [47, 48]. On the other hand, turbine fouling is mainly caused by type of fuel used or water injection method. Running a GT with natural gas which is a clean fuel slows the rate of degradation, but if the engine is running on heavy fuel such as crude oil, the degradation of the turbine will be faster. Fouling of the turbine is more expensive because of the quality of the technology involve in the material used for the hot gas path session.

Below is a view of a fouled compressor and turbine;

Figure 2-1: A fouled compressor [49]; Right: a fouled turbine NGV [30]

2.5 Compressor Fouling

No matter how efficient the GT inlet air filter is, there will be some penetration of particulate matter like fine dust particles and aerosols, which later deposits build ups on the compressor blades. This calls for the need of an efficiently designed inlet air filtration systems together with a proper compressor cleaning technique to enhance performance recovery or retention of the GT. While compressor cleaning is seen as corrective method of GT maintenance in performance loss recovery, inlet air filtration is considered as preventive measures. A compressor is said to be due for cleaning when there is about 2% drop in pressure ratio and about 3% drop in mass flow [50].

The various techniques in compressor cleaning are as listed below [50]:

- Manual cleaning method
- Abrasive cleaning method
- Liquid injection cleaning method (comprising both offline and online cleaning)

2.5.1 Compressor Fouling Mechanism

Flows in axial compressor are complex and in three-dimension. Reference [51] suggested that the three-dimensional flow field should be regarded as two-dimensional for a better understanding of the phenomenal since it is less complicated. Reference [52] described the surface of a body situated in the stream of the air-aerosol mixture which entrapped mechanism of particles. The action of inertia forces on the particles deposited on the surface of the blade forces them to move across the curved stream lines making the trajectory of the particles to deviate from the stream lines as could be seen in figure 2-2 [53]. Collision of dust particles can stick to the surface of the blade and cause fouling.

Similarly, movement of the particles along the compressor flow path causes the centrifugal inertia forces move to the compressor passage periphery. The separation factor **E** or the coefficient of entrapment according to reference [52] is given as:

$$\boldsymbol{E} = \frac{\boldsymbol{h}}{\boldsymbol{L}} \qquad 2-1$$

Where: h is the amount of particles colliding with the surface of the body.

L is the amount of particles which could fall on the surface of the body assuming the stream lines were not deviated by the body.

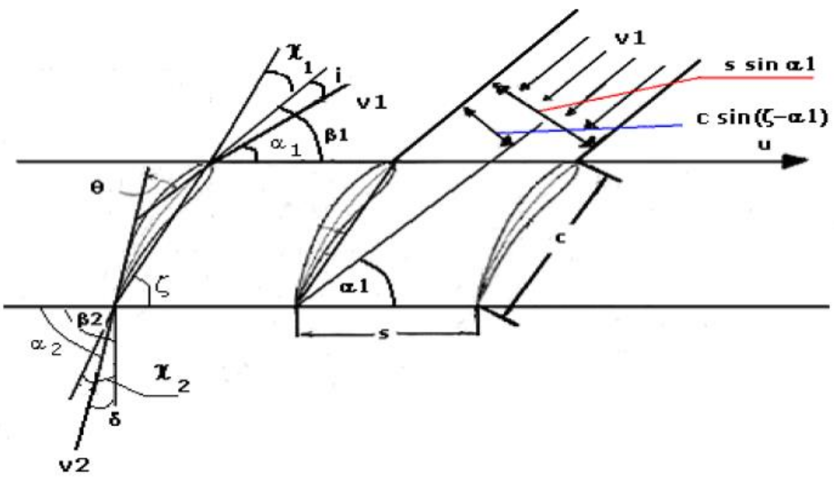


Figure 2-2: Axial compressor cascade profile [53]

$$E_c = \frac{h}{s \sin \alpha_1} = \frac{h}{c \sin(c - \alpha_1)} X \frac{c}{s} X \frac{\sin(c - \alpha_1)}{\sin \alpha_1} \qquad 2-2$$

The flow turning angle increases with increase in pressure ratio of the compressor stage. Hence, the high-head stage is more sensitive than the low-head stage.

2.5.2 Effects of Compressor Fouling on GT Performance

The rate of degradation of individual components of GT is always different, thus component mismatch and overall GT performance loss. Effects of GT performance loss as a result of fouling are generally classified as follows;

- Overall thermal efficiency reduction
- Reduction in power output resulting from degradation in inlet mass flow.
- Reduction in performance as a result of compressor fouling.
- Increase in fuel consumption and reduction in HPT blade life resulting from increased TET to regain lost power or keep power constant.
- The aforementioned consequences will lead to economic losses emanating from more fuel burnt, maintenance (repair) and frequent component parts replacement (hot section, which is very expensive) and the need to employ more labourers.
- Reduction in power supply will also cause loss in production.
- Heavy fouling could lead to compressor surge or possibly damage the engine.
- Fine particles entering the turbine could partially or completely block the cooling air passage thereby causing hot section overheating.
- Corrosion and fouling of component could lead to unstable and critical engine operation and even vibration breakup.

As could be deduced from figure 2-3, as presented by [54, 55], degradation increases by 1.5% from every 1% reduction in compressor efficiency. Fouling is said to have more impact in axial compressors than in centrifugal compressors in aspects of compressor configuration [55, 56], this is because axial compressors are more sensitive to dirt and to foreign objects which enters the intake.

Fouling causes reduction in surge margin as could be seen in the axial compressor map deduced from [55, 57]. Engine speed could hide GT fouling effect as well because a shift from the design point indicated in the compressor map could be easily confused with a different compressor line of operation [55]. It is illustrated when the design point of the compressor is located in a higher shaft speed point and lower efficiency.

Compressor efficiency increases with reduction in shaft power as a result of compressor fouling causing difficulty in detecting fouling [58]. This is because loss of material, increased surface roughness, and alternation of aerodynamic profile, resulting from deterioration, manifest itself in terms of reduced stall margin, increased operating temperatures, increased fuel consumption and inability to meet power requirements. Surge or rotating stall might occur as a result of reduced stall margin, which might have a dangerous effect on blades and overall engine integrity. Component life will be reduced resulting from increased operating temperatures.

The figure 2-3 as curled from [59] shows the effects of engine deterioration on efficiency for various components and engine parameters. It can be observed that deterioration decreases engine components efficiency.

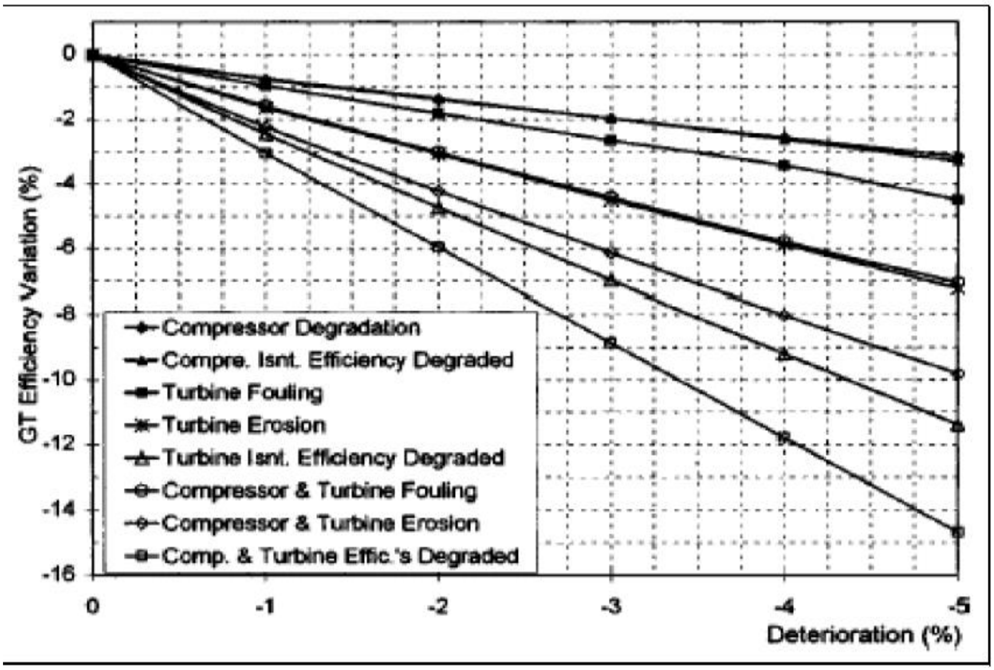


Figure 2-3: Gas turbine efficiency degradation in specific sections [59].

Research carried out by [60], analyses the merit of engine configuration to estimate the impact of fouling. A similar engine to that of GE LM2500 two shaft engine configurations as shown in the figure 2-4 was first used, and a specific value of compressor delivery pressure (CDP) was applied in the normal operation condition. Afterwards, a specific value of shaft power speed (SPS), which is the revolution per minute of the engine shaft that was applied to the gas generator turbine (GGT).

The turbine entry pressure (TEP) reduces with reduction in CDP as a result of fouling. The GGT was affected, thus reduction in SPS. The fuel consumption automatically increased to regain lost power from the GGT and to obtain the correct better SPS. Here, compressor fouling could be detected as a result of the speed and fuel consumed by the engine.

Another case by the same authors [60] was the use of Allison 501K single shaft engine as could be seen in the figure 2-5. The SPS decreases with decrease in CDP resulting from fouling of the compressor. The TET was increased to retain the lost SHP by an automatic increase in SFC. Because it is required to check the engine performance with two different loads, it becomes difficult to detect the problem, as it is not possible to change the load in real life situation.

It is also difficult to detect fouling in multiple axial compressors, as the rear stage is modified by the early stage. Based on research by [61] blade aerodynamics to detect the stage affected is complex to analyse. Thus, it becomes difficult to detect fouling inside the compressor and generally based on experience of operation [62- 64].

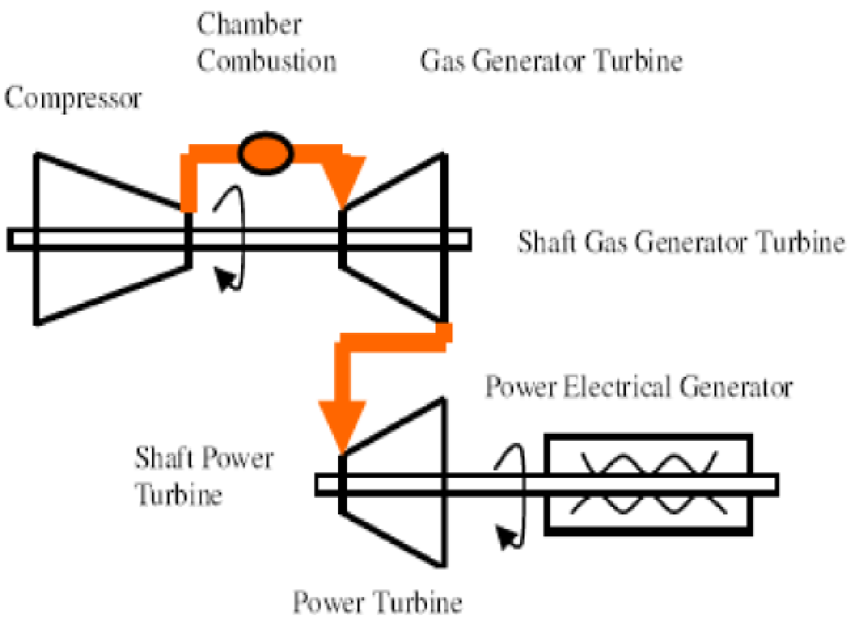


Figure 2-4: Two shaft configuration Gas Turbine

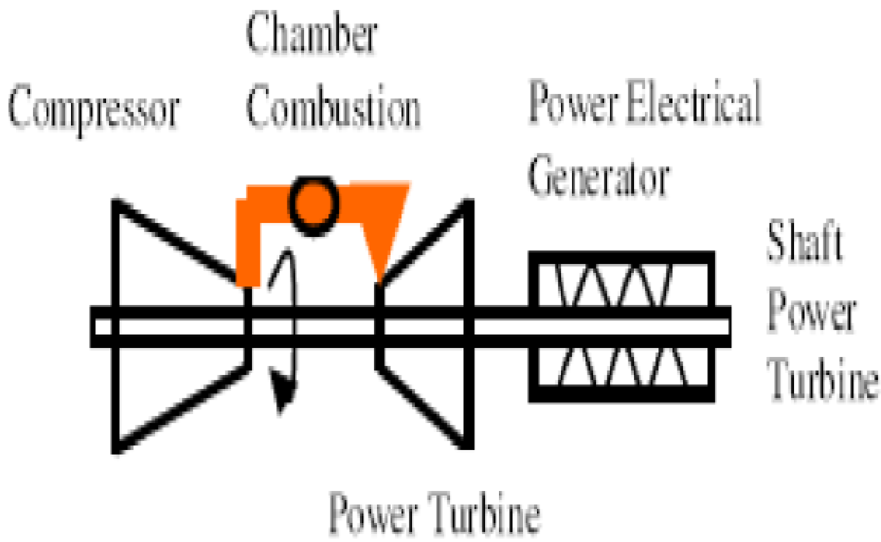


Figure 2-5: Single shaft configuration Gas Turbine

2.5.3 Effects of Fouling on air Distortion and Compressor Surge

A major consequence of compressor fouling is the reduction in the inlet mass flow, and the effects and problems will accumulate. The reduction in air mass flow decreases the efficiency and the stall region margin from the compressor [65, 66], and it is as a result of increase in surface roughness that changes the thickness of the boundary layer and reduction of the aerodynamic properties from the blade [41].

The compressor map has this changes being represented when the operating point is moving close to the surge [67, 68, 56]. It is very dangerous to have the operation close to surge line for compressor [63].

2.5.4 Factors Influencing Compressor Fouling

Some amounts of contaminants are ingested alongside atmospheric air into the GT compressor during intake, such as soft aerosols resulting from pollen, oil water, sticky industrial chemicals, particles of dirt, soot particles, dust, un-burn hydrocarbons, insects etc [69, 70]. These particles can cause temporal or permanent problem as in the case of fouling and erosion to the blade respectively, which in turn result in GT performance deterioration [21]. Compressor fouling originally emanated from the mixture of particles and atmospheric air, which is about 80% of the dust found on the filters formed by the fouling layers on the blade surface [71]. Many cases of combination of contaminants with residue oil or water mist are sources of layers of fouling [41]. Under harsh conditions such as chemical polluted storm or sand storms, there is an increase in the concentration of particles and acceleration in the mechanism of fouling [72]. In this case, it is imperative to consider the environmental condition, maintenance plan and plant layout in order to reduce the possibility of compressor fouling [73].

References [41, 74, 75] shows that the deposited particles on the compressor blade surface area are in the range of micrometres. Figure 2-6 below is an x-ray analysis indicating fouling sample showing that the layer of fouling is a mix of different components [76].

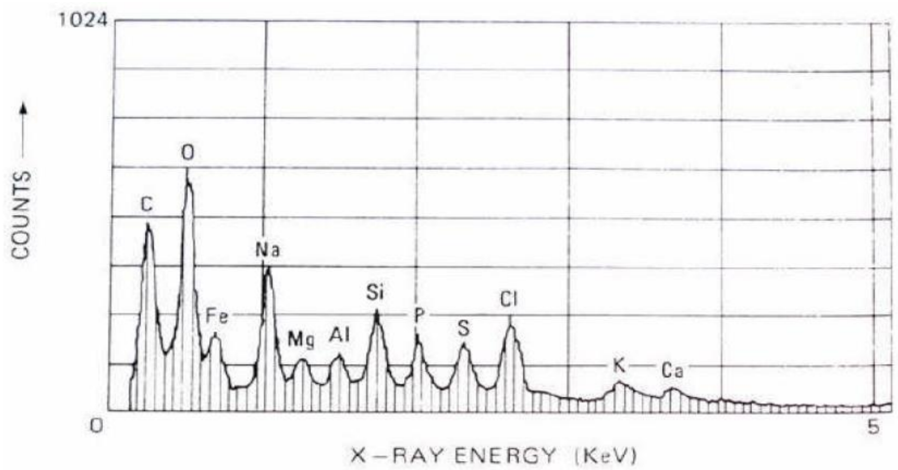


Figure 2-6: EDX Spectrum of layers deposit on the surface of compressor blade [76]

The two groups of components identified from this analysis comprised of water-insoluble solids represented by the presence of silicon, and the organic materials were represented by the presence of carbon and oxygen in the first group, and water soluble substance that causes corrosion in the second group. They were hydroscopic and their chloride content promotes corrosion.

- ***Sources of internal contaminants***

Non-maintenance or incorrect operation of auxiliary systems could result in internal losses that cause fouling [77]. The presence of oil residues due to leak seals is an example of fouling commonly found in compressor. Salt and or water impurities can be deposited on the compressor via the cooling unit by fog in the inlet plenum of the engine [40, 74]. Erosion and corrosion affects the filter panel and thus increases the presence of FOD inside the engine [74]. The presence of steel and aluminium particles was found from components rubbing of brush seals and bearing resulting from fouling [66]. The demonstration of wear of graphite brushing from the compressor guide vanes have also shown to be source of compressor fouling [78].

- ***Sources of external contaminants***

A general classification of the contaminants according to geographical location is as presented in table 2-2 [79]. It has also been discussed by [79, 80, 81] that salt is a major source of compressor fouling.

Table 2-2: Typical contaminants in gas turbine location [79].

LOCATION	MAIN CONTAMINANT
Industrial	Dust and hydrocarbon aerosols
Rural	Pollen
Costal	Salt

Table 2-3: Concentration and particles sizes in atmospheric air [82, 83].

TYPE	TYPE OF PARTICLE	SIZE (μM)
F1	Sand	20 ~ 2,000
F2	Ground-Dust	1 ~ 300
F3	Oil Smokes (oil & gas plants)	0.02 ~ 1
F4	Fly Ash	1 ~ 200
F5	Salt Particles in Mist	Less than 10
F6	Salt Particles on Spray	More than 10
F7	Insects Swarms	More than1,000
F8	Smog*	Less than 2
F9	Clouds & Fog	2 ~ 60
F10	Rain	More than 60
F11	Fume*	Less than 1
F12	Clay	Less than 2
F13	Rosin smoke	0.01 ~ 1
F14	Fertilizer	10 ~ 1,000
F15	Coal Dust	1 ~ 100
F16	Metallurgical Dusts and Fumes (welding smoke)	0.001 ~ 100
F17	Ammonium	0.1 ~ 3
F18	Cement Dust	3 ~ 100
F19	Carbon Black	0.01 ~ 0.3
F20	Contact Sulphuric Mist	0.3 ~ 3
F21	Pulverized Coal	3 ~ 600
F22	Paint Pigments	0.1 ~ 5
F23	Plant Spores	10 ~ 30
F24	Pollens	10 ~ 100
F25	Snow & Hail	More than1×10^4

Modern filtration technologies have proven to reduce the inlet particles from the air stream, but it effectiveness still fluctuates because of the environmental conditions. Possible particles trapped by the filters are as shown in table 2-3.

The pressure, temperature, humidity, and ambient conditions, play an important role in the performance of the engine and are very necessary to be considered in the study of fouling. The ambient temperature is classified into three variations: Hot (50 – 30^0C), Warm (29 -15^0C), and Cold (15 – (-20^0C)) as could be seen in table 2-4.

Table 2-4: Industrial gas turbine operation environmental scenario.

CASE	CONDITION	LOCATION EXAMPLES:	FOULING EXAMPLES:
1	Hot+Dry	Desert locations	F1,F2,F3
2	Hot+Medium	Jungle and Marshes locations	F7, F9, F10, F14, F23, F24
3	Hot+Wet	Coast & offshore locations	F1, F3, F4, F5, F6, F7, F9, F10, F14,
4	Warm+Dry	Barred locations	F2, F3, F4, F11, F15, F16, F18
5	Warm+Medium	Central locations	F2, F3, F4, F7, F8, F14, F15, F23, F24
6	Warm+Wet	Raining or Coast locations	F1, F3, F4, F9, F10, F12
7	Cold+Dry	Central Artic Locations	F3, F4, F17, F25
8	Cold+Medium	High Sea Level locations	F2, F3, F4, F5, F8, F9, F12, F25
9	Cold+Wet	Artic Coasts locations	F1,F2, F3, F5, F6, F16, F25

The humidity is classified into three categories: Dry (less 10%), Medium (about 50%), and Wet (above 75%) [84]. There are nine cases presented for possible environmental scenarios as could be seen in table 2-4 between the temperature and humidity results.

Fouling in a particular location could vary due to seasonal change in ambient conditions and particles concentration. A typical example is the behaviour of ambient condition in the northern hemisphere. The altitude also modifies the performance of the engine, but does not have significant effect on the mechanism of fouling.

- ***Fouling from steam and vapour***

Oil and vapour inside the compressor triggers the adherence of particles on the surface of the blade [85], and the oil vapours emanated from the oil leaks in the internal components of the engine [40]. A hard layer is formed from the deposition of oil and particles on the rear compressor blades as a result of the temperature. In the general engine maintenance, it is only possible to remove the layer by hand [86], and the chemical vapours are mixed with air which pollutes the environments. The diesel vapour produced by auxiliary engines in marine applications close to the engine inlet [86] is a typical example, and the natural ambient agents accelerates the adhesion process as such as heavy fog, rain and excessive humidity [87].

2.5.5 Preventing Compressor Fouling

Some of the measures to put in place in other to prevent compressor fouling are as stated below;

- A scheduled maintenance programme
- Frequent maintenance of intake air filtration
- Regular component wash
- Proper fuel treatment to abate fouling in the hot section
- Adherence to manufacturers maintenance and operation manual

2.6 Gas Turbine Performance Degradation

Mechanisms responsible for GT degradation include abrasion, particle fusion, erosion, corrosion, foreign object damage (FOD), domestic objects damage (DOD), Turbine blade cooling passage plugging, and fouling. These could be classified furthermore [88, 89] as:

- Recoverable degradation
- Non-recoverable degradation and

The basis of this classification is on whether the performance loss as a result of these mechanism is recoverable or not, when washed or cleaned.

2.6.1 Recoverable Degradation

It is the type of performance loss that can be restored when a proper washing or cleaning is carried out on the GT component. Regular overhauls are carried out between short intervals with the engine still running at crank, while major overhauls are done with engine shutdown involving major parts cleaning or component change where required.

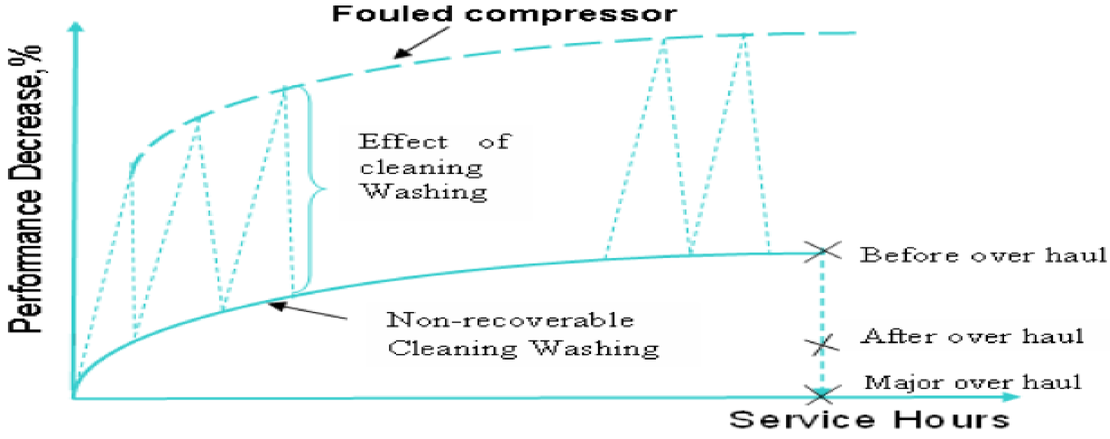


Figure 2-7: Typical performance deterioration of a compressor

2.6.2 Non Recoverable Degradation

The loss in performance can only be restored in this type of degradation if the entire engine is shut down for major overhaul, and the component parts being repaired or replaced. That means the component parts will only need to be changed to get the entire engine function again, and it normally as a result of component wear [45].

2.7 Methods of Compressor Washing

Compressor fouling reduces the air flow rate, pressure ratio and efficiency, which causes drop in power output, and can result in surge problems in some cases [47]. Combining two methods will best control fouling; using high quality filtration system or regular washing. Applying the offline and online techniques of compressor washing, and a combination of frequent online washing plus periodic offline washing will optimize the compressor cleaning [47].

As the vanadium contents in fuels increases, magnesium salt is added to the fuels to encounter the corrosive action of the vanadium. The ash which deposits on the blades and causes degradation in flow capacity of the turbine is created by the magnesium salt. Restoring power and efficiency requires the turbine section to be blasted with steam while running on turning gear at slow speed.

The engine will be accelerated to it normal speed as soon as the turbine is dried, and the entire process could take about 20 hours [90, 49, 91].

2.7.1 Online Compressor Washing

It is a method of compressor cleaning also known as fired or hot washing and it is performed with the engine running load at little or no reduction in capacity or speed of the GT. It is aimed at keeping the compressor clean through frequent wash in about 5 to 30 minutes. The injection of demineralized water and or cleaning solution takes place at the inlet. Compared to offline wash, it is less destructive and intensive [92]. It is not as effective as offline washing because it cannot recover lost power to a greater percentage when compared. Offline washing is highly recommended when the compressor is heavily fouled [50, 93]. Frequent online washing has a demerit in that, adding water for wash increases pressure ratio of the compressor which in turn reduces the surge margin [93].

The online compressor wash is more effective at the first few stages of the compressor, because the washing solutions are being evaporated by high temperatures from the compression process before getting to the latter stages. It thereby pushed the contaminants from the front to the rear compressor stages [94]. This method is effective and reduces rate of fouling, but cannot be compared to offline method because in the online regime, the rear stages of the compressor are always not properly cleaned.

Fouling of the first stage guide vanes is the root cause of compressor flow capacity reduction [47]. Online cleaning can remove the deposits and restore the air flow rate and power output of the GT. Regularly cleaning the engine will help keep it free from deposit build-ups.

Performance recovery through both online, offline and combined washing methods is presented in figure 2-8, and it illustrates the importance of the combined methods as it maintains high GT efficiency with time. The red line indicates the rate of compressor degradation when no wash is performed at all. It can be observed that the engine life would have been very short if this happens in real life scenario. In a similar way, the blue line shows the rate of degradation when the engine compressor wash is done only for the online case, this show that the engine would have had a longer life compared to the no wash case. The green line shows the combination of both online and crank wash. This can be seen obviously that the life at this stage is far longer compared to other two cases, as such explains that a reasonable combination of both online and offline compressor water washing would yield better savings for the GT owner.

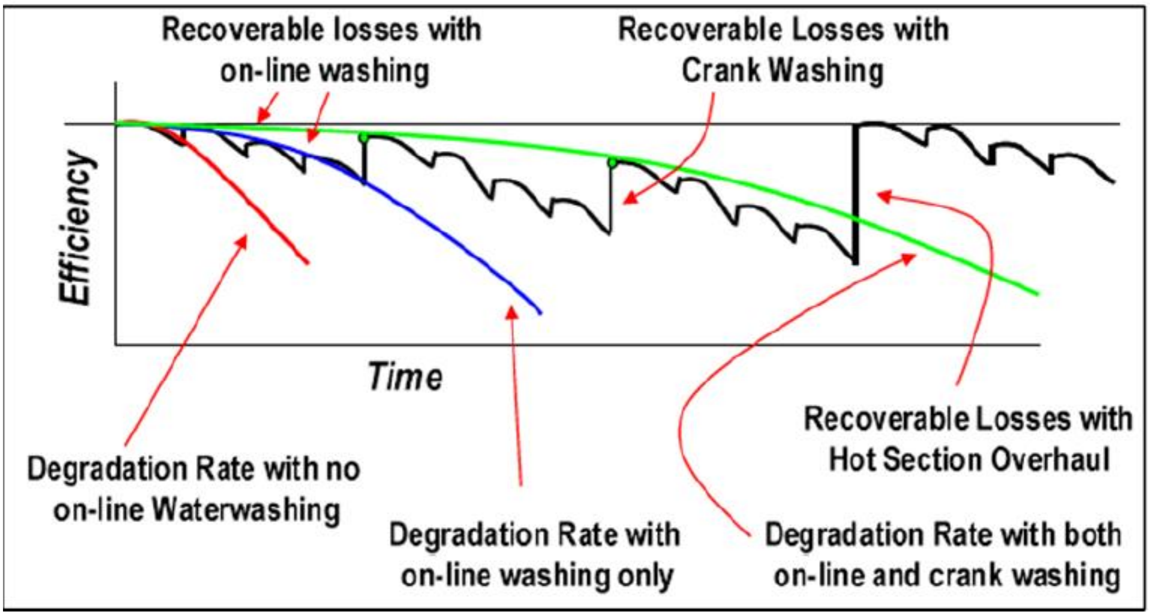


Figure 2-8: Effects of washing towards compressor efficiency [58].

The main reasons for online washing could be summarized as stated below;

- To retain the compressor effectiveness after major offline washing.
- Minimizing on-going losses to maintain power and efficiency
- Reduces deposits build up in the compressor by washing, which help to extend time between offline wash and in turn reduces on going incremental losses in power.

Table 2-5 below shows a summary of the merits and demerits of compressor cleaning methods. It can be seen that the offline is very effective, but have the demerit of shutdown which will lead to loss of production hours and economic loss as well.

Table 2-5: Merits and demerits of compressor cleaning methods [50].

Method	Advantages	Disadvantages
Manual Cleaning (brushes, washing agent)	Very effective	Shut down of engine Laborious
Grit blasting (Charcoal, rice, nut, synthetic resin particles)	Simple and fast. No engine downtime. Effective in cold environments.	Less effective at rear stages and for oil deposits. Clogging of internal cooling passages. Erosion. Increased surface roughness. Damage of blade coating.
Soak, Crank, offline washing (demineralised water, washing agents)	Very effective	Shutdown of engine
Fired, Online washing (demineralised water, washing agents)	No interference with load profile. Extends intervals of crank washes.	Less effective. Cannot replace offline washing (complementary).

2.7.2 Offline Compressor Washing

Also known as crank or soaks washing. It is fundamentally meant for cleaning fouled compressor to enable restoration of lost power and efficiency of the engine. A proper performance of this method on the engine could restore about 100% of the lost power and efficiency [47]. It requires shutting down the turbine and allowed to cool to avoid thermal stresses of the component; while the compressor is being rotated on crank that is 20% or 30% of its normal operating speed [31] as cleaning fluid is being injected through the nozzles. The operator will experience loss in revenue due to this downtime [50]. Allowing large disparity between the temperature of the washing detergent and the turbine component will cause the occurrence of thermal stress. The method is best practiced on scheduled outage, because it requires long downtime. It is very effective though. It is usual to carry out this wash method alongside other maintenance practice on the GT.

The two means below can be used to perform washing:

- A fixed system-containing nozzles already mounted on the plenum and or
- With a manual system-containing hand hose and spray nozzle [93].

It is advisable to adhere to the OEM specifications in terms of rates of injection, detergents and other recommended directives. The entire circumference of the bell-mouth needs to be covered by the spray. To ensure deposits are properly trapped from entering the engine during wash, the bell-mouth and the plenum are adequately cleaned. The engine is kept to run at crank speed while water or solvent based detergent is being applied into the compressor inlet continuously. Time is allowed for the compressor to get soaked by the detergent or water based solvent to enable the removal of salts and grease deposits from the compressor casing, while it is later rinsed off properly with clean water. To achieve effective result of this wash and rinse period, it is required that the effluent water in the drain be visually inspected to check for cleanliness. Electrolytic conductivity of the runoff water is an alternate method of confirming this effectiveness, as low conductivity value indicates an effective wash; and the engine wash-rinse is continued otherwise.

A blow-run followed by a dry-run is carried out after the final rinse. The blow-run is carried out at crank speed and it helps to drain all the trapped water in the GT internal piping systems. It is followed by a dry run at no load to establish that the engine is ready for start [50].

Figure 2-9: Before and after offline wash [95]

Figure 2-9 indicates the appearance of a compressor blade before and after an offline wash has been performed. A relationship needs to be carried out between the amount of revenue and the availability of the power unit in other to determine the amount of revenue loss due to downtime associated with offline washing.

As earlier discussed, the availability of a power plant is the percentage of time the engine is available to produce required power at any given time as it load acceptance [96]. Worthy to know is that offline compressor cleaning reduces the availability of a unit.

2.8 Compressor Cleaning Fluids

The major types of compressor cleaning fluids are classified into three group viz [86]:

- Solvents
- Aqueous based cleaning fluid (ABC)
- High quality de-mineralised water

The available detergent fluids present for use nowadays are mostly non-ionic and mainly designed to meet the online and offline gas turbine washing requirements which include onsite handling, storage, environmental impacts and draining of effluents. The surfactant that acts like a surface active agent to reduce the surface tension of the solution so as to wet, penetrate and disperse the foulant deposits are the main content of the cleaning agent.

2.9 Compressor Washing Schedule

According to [86], the frequency or periodicity of compressor wash or performance recovery is as stated below;

- The rate of fouling
- Amount of accumulated fouling
- The fouled component involved
- Critical depth of the fouling

The factors stated above determine the intervals between compressor washing for effective performance, extraction of maximum possible power recovery, and revenue savings. These are the major reasons for compressor washing. In this research work, the on-line and off-line compressor washing are applied in other to extend the life span of GT in service. Frequent on-line washing is carried out before a major off-line wash takes place.

2.10 Failure Mechanism on Gas Turbine Hot Section

Operating gas turbine at extreme temperature causes severe damage mechanism to the hot section such as high temperature oxidation/corrosion, fatigue, and creep deformation. With the emergence of such mechanism, the component will lose its ability to sustain its intended function, reduced life span and reasonably causes premature failure to the component. The extent of the failure is complex and depends on the rate at which strain is applied, temperature of the material and deformation [97]. The mode of failure and origin of damage are the two major questions involved when a metal component fails. Thus, examining the 'how' is very important to understand the deterioration phenomena [98].

2.11 Creep Deformation

Creep stretches or elongates the GT hot section components and it is a thermally assisted deformation at very high operating temperature over a period of time with constant mechanical loading below the material yield stress. For instance, in the event of severe creep deformation, there will be change in the physical shape of the blades and therefore cause it to malfunction. More so, the blade will be in contact with the casing due to elongation, thereby leading to blade fracture and engine failure as could be noticed in figure 2-10.

Figure 2-10: Deformed Turbine Blade Due to Creep [99]

It can be observed that the original tip features have been lost due to creep attack. Creep on a general note becomes significant when the ratio of the material temperature and its melting temperature is more than 0.5, but can be in the range of 0.4 to 0.6 [100-102]

2.12 Stress Rupture and Creep Curve

Creep behaviour is determined through laboratory test where a constant uniaxial load σ is applied at a constant temperature, resulting to a creep strain ϵ_c recorded as a function of time t. the creep curve of ϵ_c against t is as shown in figure 2-11.

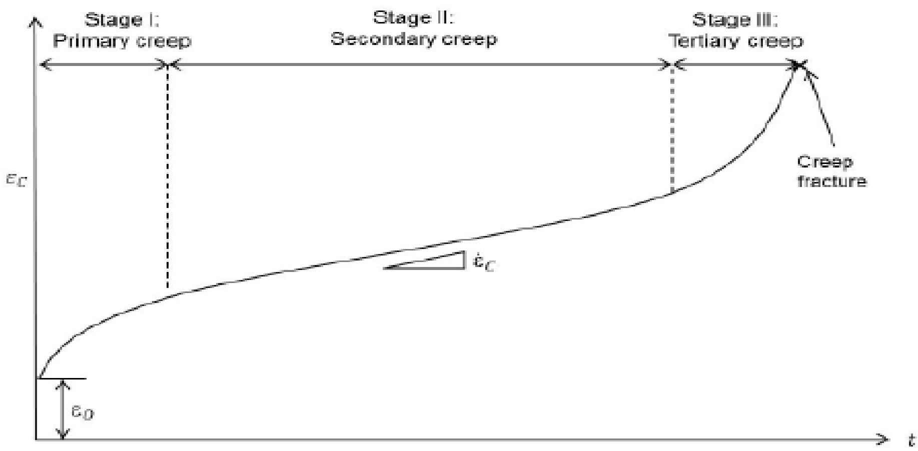


Figure 2-11: Creep curve

At the primary creep which is the first stage after the instantaneous plastic strain ϵ_o caused by load application ϵ_c which increases with time, the creep rate/ plastic strain rate will increase initially as a result of higher dislocation motion of the material. The higher the rate of material dislocation, the higher the rate of density dislocation until it becomes saturated, thus preventing further dislocation and creating strain hardening that reduce the creep rate until it reaches a constant rate.

The steady-state creep also known as the secondary creep will possess a fairly constant creep rate and the strain hardening rate will become proportional to the deformation rate which provides the balance between both, causing the rate to become steady as could be seen in the figure 2-11.

The tertiary creep is the final creep and a period that will lead to fracture which can be caused by some factors as stated below [100]:

- Microstructural instability and grain growth or re-crystallisation with single-phase material or the gradual loss of creep strength as over-ageing occurs during creep of precipitation-hardened alloys.
- Mechanical instability like occurrence of necking leading to localised reduction in cross sectional area and/ or
- Nucleation and growth of internal micro cracks which develops until the number and sizes of the micro cracks are sufficient to cause the creep rate to increase.

2.12.1 Causes of Creep Deformation

Creep is a time dependent and thermally assisted deformation, and the more the component is exposed to high temperature, the more deformed it will become. Several factors such as period of exposure, metal temperature and material properties determines creep characteristics [103-105].

The creep resistance vary from one material to another as a result of varying microstructural arrangement, grain size, activation energy, and vacancy concentration within the material. The material variations creep resistance at 100 hours of creep rupture as could be seen in figure 2-12 shows for instance, that the temperature of Tungsten and Niobium alloys is higher than that of Nickel alloy but show lower stress resistance.

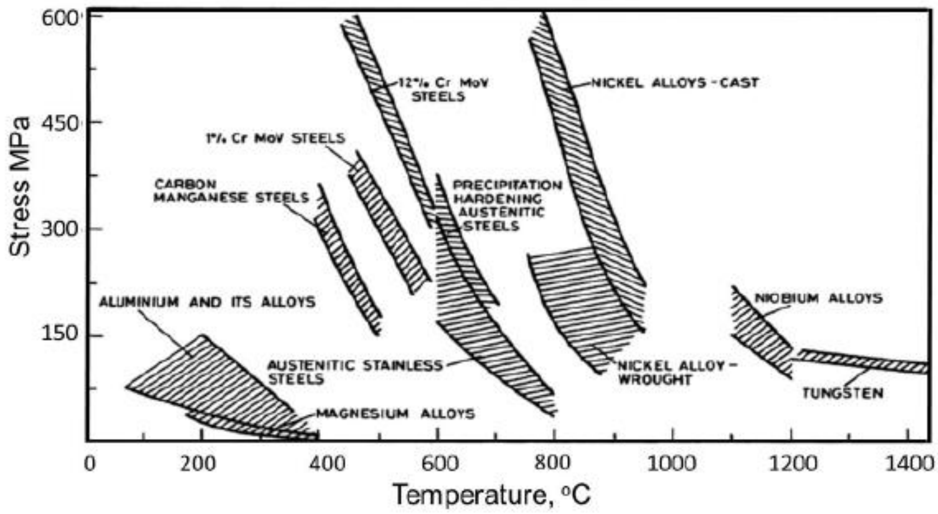


Figure 2-12: Stress and Temperature to produce creep rupture in 100 hours in various alloys [102].

Higher temperature weakens the material since creep is a thermally activated process, resulting in dislocation increase, creep cavity nucleation, grain boundary sliding etc [106], thus increase in creep rate will shorten the time to failure t_f of the material. When the material temperature is increased at a constant applied load, the t_f is shortened and the secondary creep rate also increases as could be seen in figure 2-13.

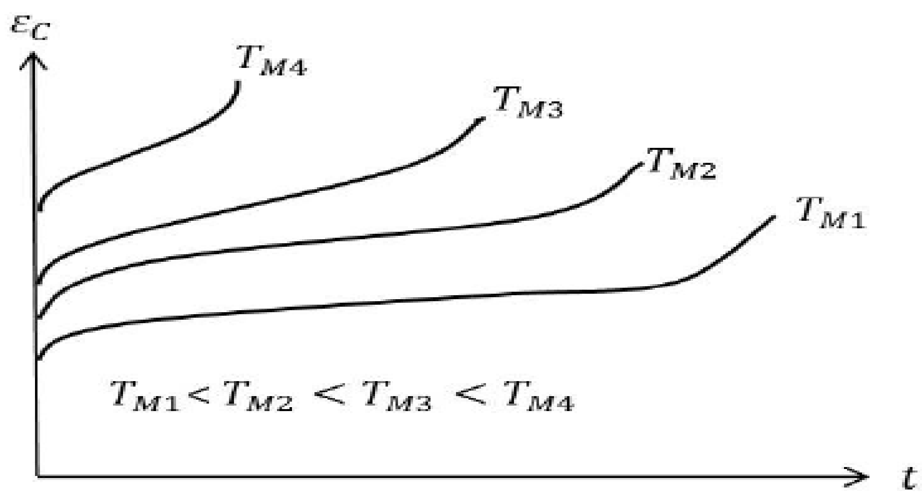


Figure 2-13: Creep curve at different temperatures

More so, at higher metal temperature, the material elongation is also higher, but the exposure time is shortened.

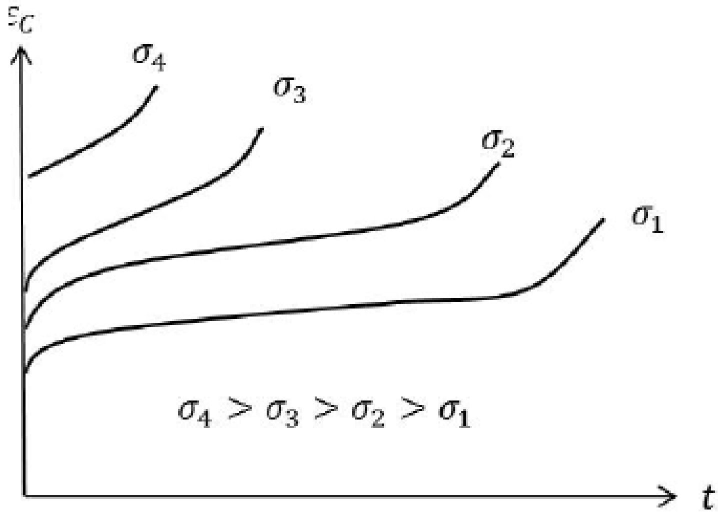


Figure 2-14: Creep curve at different level of applied stress

As could be seen in figure 2-14, creep rate is sensitive to the level of applied stress σ at constant temperature exposure. It can be deduced from the figure that the increase in σ shortened the primary and secondary creep stages or even eliminates it, and hence reducedt_f. Similarly, at higher σ level, there is higher elongation of the material, although time exposure to the applied σ is shortened.

2.12.2 Effects of Compressor Fouling on Blade Integrity

Mechanisms such as erosion, corrosion, and FOD's have effect on blade failure, while fouling has direct effect on component performance but not the major reason for blade failure. Rotating stall and compressor surge are direct effect of fouling as it can affect blade integrity. Imbalances and vibration are often caused by heavy fouling on moving surfaces. In the presence of excess humidity, fouling layers adheres to each other and build up, corrosive contaminants such as salts, acids, and aggressive gases like NOx and SOx could adhere and lead to corrosion and blade pitting. This could result to stress concentration and reduction in the fatigue life of the blade.

Dirt and particles could also penetrate the small clearances like seals and bearings, which could build up and lead to unstable disc and blade operation.

2.13 Determining Time to Failure

There is need to carry out the extrapolation and interpolation to the required σ and metal temperature because the experimental method is mostly not sufficient to cover various σ and metal temperature. This is because either by experiments or using recorded data from creep curve, it will not necessarily provide the most appropriate forms of creep estimation.

An alternative solution is the time temperature parameters which are correlated parameters and allows results obtained from the stress rupture test over a range of temperatures to be superimposed unto a single master curve. The widely applied out of the various time temperature parameters include: Orr-Sherby-Dom parameter (OSD), Manson-Succop parameter (MSP), Manson-Haferd parameter (MHP), and Larson Miller Parameter (LMP). The LMP will be explained further since it is the choice of parameter to be applied in the research work as it has proven to be more accurate when compared to others [4].

2.14 Methods of Determining Creep Life

The relationship between the level of load and life of the component is defined by the lifing model. This model estimates the total time to failure, and the main division in lifing models is between total life models and crack growth models [107]. Since deformation and fraction are time-dependent, the useful life of hot section components in practice decreases progressively due to creep deformation as could be seen in figure 2-15. The ability of the material to withstand creep deformation and the operating conditions of the actual GT engine is a factor to determine the rate at which useful life is consumed. With extreme operating condition, there will be faster degradation in the material and thus consumes useful life faster. The crack deformation which begins from the primary stage is as a result of the progressive degradation of the material.

Reference [108] shows how micro-cracks start on the surface and propagate until link into macro-cracks and become visible at the tertiary stage of creep.

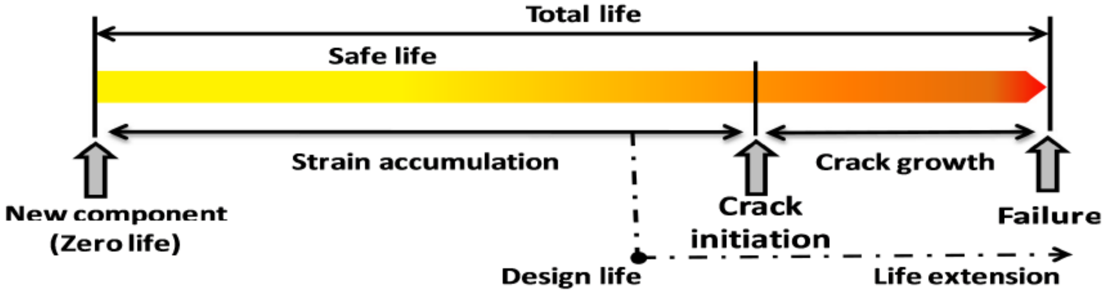


Figure 2-15: Deformation of creep life span for hot section components [109, 110]

The life methods amongst the several for determining creep life could be grouped into three classes as stated in figure 2-16 below [6, 7];

- Design approach
- Post-service approach
- Statistical/probabilistic approach

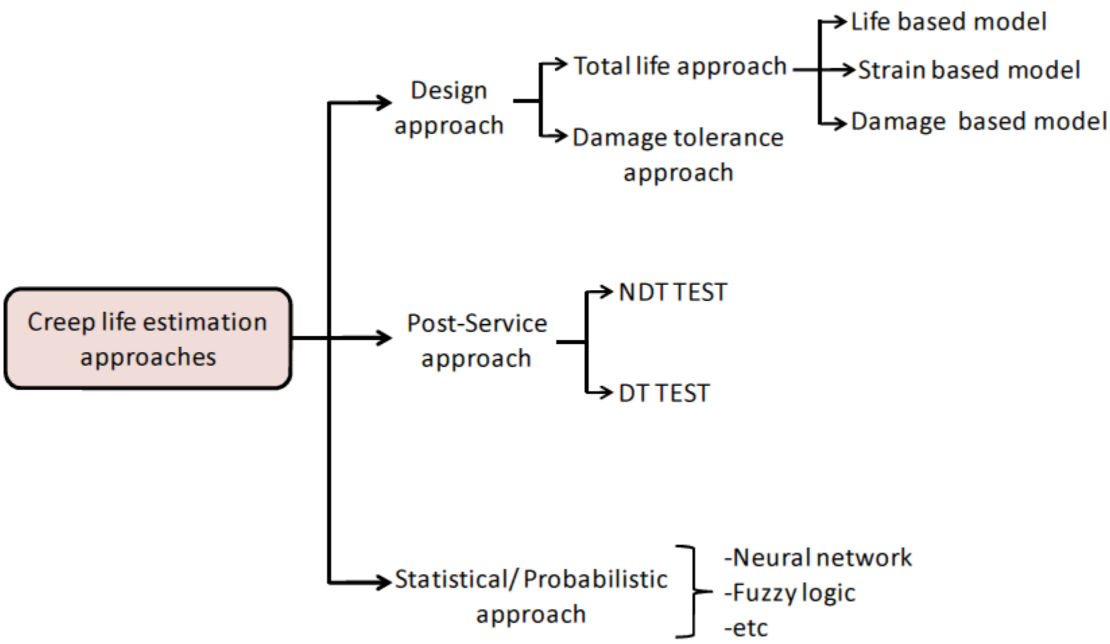


Figure 2-16: Methodologies for creep life estimation [6, 7]

Table 2-6: Lifing approach [5-7]

	Design approach	Post-Service approach	Statistical approach
Methods	✓Analytical method ✓Empirical method ✓Numerical method	✓Non-destructive inspection ✓Destructive test	✓Statistical tools ✓Probabilistic tools ✓Artificial intelligent methods
Advantages	✓Life estimation can be performed at design stage. ✓Low cost if low fidelity model is used.	✓Identify the components that need to be monitored. ✓Offers different techniques for both destructive and non-destructive test.	✓Reduce complexity ✓Fast computing ✓Identify driving factors ✓Tackle the uncertainty
Disadvantages	✓Based on data to build the model. ✓High complexity to achieve. ✓High cost.	✓Need prior techniques to estimate life at design stage. ✓Time consume. ✓ Perform during maintenance. ✓Inaccurate if uncelebrated device used.	✓Requires prior models ✓The factors may vary according to the nature of the research.

The methods shown here are applied for different sections and at different stages of failure of the components, which means there is no comparable accuracy and reliability. The advantages and disadvantages of each approach for the three lifing approach are as shown in table 2-6.

2.14.1 Larson-Miller Parameter (LMP)

The equation for LMP is as shown below;

$$logt_f = \frac{LMP}{T_M} - C_{LMP} \qquad 2-3$$

Here, C_{LMP} is the LMP constant. As could be seen in figure 2-17, plotting $logt_f$ against $^1/_{T_M}$ will create iso-stress lines which will converge to a point C_{LMP} on the $logt_f$ axis. In addition, the master curve for different values of σ can be generated.

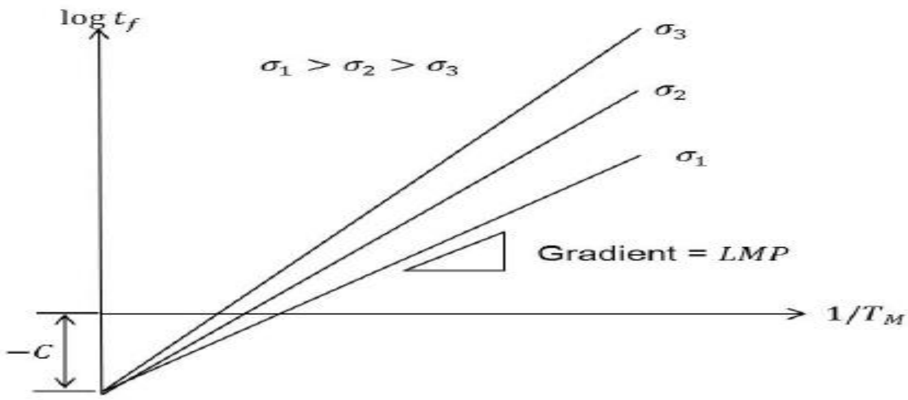


Figure 2-17: Plot of log t_f versus $^1/_{T_M}$ using LMP

2.15 Economic Consequences of Compressor Fouling

There are several ways by which fouling is associated with cost of maintenance for a GT;

- **Daily Operation Cost**

It comprises cost of additional fuel consumed in other to maintain lost power. The TET increase is limited by a control system mounted during manufacture, which has a major effect on power loss and thus production loss. The power generation and Oil & Gas industries will find this noting worthwhile.

- **Maintenance and Repair Costs**

Increasing TET to retain lost power as a result of fouling will definitely reduce turbine blade creep life. The cooling passages of the blade could be blocked by fouling, thereby causing the turbine hot section to overheat and increase metal temperature of the blades. Increasing TET by 22K increases the HPT blade by a factor of 3.2 or 69% as estimated by [111]. There is huge economic loss in replacing hot section component too often or early. The early replacement of compressor components could also be as a result of fouling which could lead to corrosion of the compressor blade. Increasing heat rate by 1% and reducing power output by 3% for a 46.5MW engine could cause about $1.5 million dollars (US) in three years period [33].

Another research by [71] also showed that; total loss incurred from fouling for two engines of 26MW and 225MW, each operating at 8000h yearly were \$500,000 (US) and \$ 5,000,000 (US) respectively. The cost associated with fouling could be reduced drastically by introducing a suitable and properly designed compressor cleaning operation method. This could be achieved for an operator who uses proper washing systems such as the required nozzle & injection system, correct cleaning detergent choice, ideal washing frequency & procedure, and the washing skid [112].

2.16 Environmental Impact of Gas Turbine Emission

The Kyoto Protocol [113] and the Stern Report [114] has brought about reasons for carrying out research on emissions control and reduction. The economics of climate change and highlights for growing needs to take action against greenhouse gases were discussed in the Stern report, while government targets for 2012 to 2020 for levels of emission reduction targets were sets by the Kyoto Protocol [113]. It is shown in the figure below that power generation (electricity and heat) occupies 25% of the emissions.

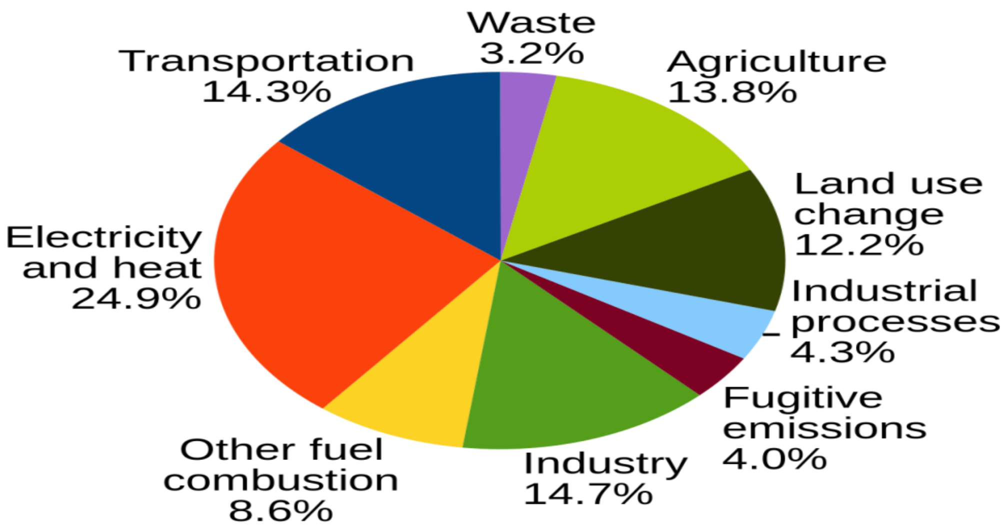


Figure 2-18: Annual world greenhouse gas emissions for industrial sector 2005. [114].

2.16.1 Emissions of Carbon and Nitrogen Oxides

These are the two main pollutants from gas turbines. Most of the global warming is contributed by Carbon dioxide, while petrochemical smog is promoted by Nitrogen oxides which are very harmful to the health. The NOx problem can be solved by reducing the temperature of the combustion zone and the firing temperature, which is associated with reduction in power, but with an increase in production of carbon dioxide and unburned hydrocarbons (UHC) such as carbon monoxide.

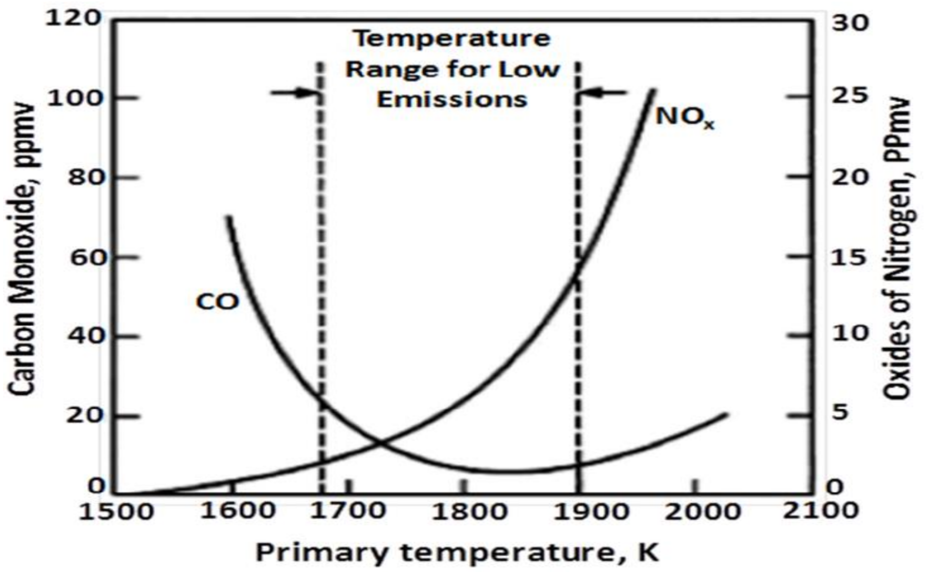


Figure 2-19: NOx and CO emission against temperature [115].

Although high temperatures could improve engine efficiency and reduce UHC, but will exponentially increase NOx production [116, 117]. The trace is as shown in the figure 2-19.

It is shown in the figure 2-19 that NOx emission predominate at high combustion temperature, and this high temperatures produces more power outputs, and because compressor drives work most times at base load, the EINOx has more importance than other emissions. The Dry Low Emission (DLE) methods have decreased the amount of NOx emissions but the difference in engine firing temperature between various emissions such as NOx, CO_2, CO and UHC in IGT will be the study of this part of the research.

CO_2 can be modelled with the help of stoichiometric calculations assuming incomplete combustion; the amount of CO_2 produced can be given by the balanced equations of the combustion reaction. More specialist treatment is required for the CO, UHC and NOx as will be seen later in this work.

2.16.2 Empirical and Semi-Empirical Methods

This is the method applied here in this research which makes use of established trends in emissions from known empirical equations and combustor types that relates these emissions to the engine thermodynamic conditions.

Semi-empirical techniques has been analysed by [118] as calculation approach that simulates the combustion process by global expressions to account for reaction temperature, evaporation, system pressure, and mixing which are used to provide details regarding blowout, ignition and emission indices. More so, the model include variables such as; inlet and outlet temperatures and pressures of the combustor, liner walls and combustor pressure loss, adiabatic flame temperature, fuel flow, combustor volume, equivalent ratio, time required for fuel evaporation, fuel-air ratio, volume occupied by evaporated fuel, and mass flow etc. These are what make this type of model more within reach of analytical tool like TERA.

2.16.3 Control Measures for Emissions

Other prime movers emit more pollutants than the GT's because the air supplied in GT's is enough to complete the combustion process, but more fuel is injected at part load, mostly when the load condition is minimum, thereby increasing the equivalent ratio. The policies controlling GT emission and it release to the environment has generated reasonable research on emission estimation methods and control measures.

The table 2-7 below shows the emission species for NOx, CO, UHC, and CO_2 and their modes of control [119];

Table 2-7: Sources of emissions and their control measures [119].

	Control Technique
NOx	a. Lean-Head-End liner.
	b. Water & Steam Injection
	c. Dry-Low-Nox
CO	a. Combustor design
	b. Catalytic reduction
UHC & VOC	Combustor design
CO2	a. Combustor design
	b. Fuel choice
Smoke / particulate matter	a. Fuel atomization
	b. Air atomization
	c. Ash
Sox	Sulphur control in fuel.

2.16.4 Emissions Indices

The emission index of component emitted during combustion process of fuel-air mixtures is defined as the ratio of the mass in grams of the emitted component to the mass in kilograms of fuel:

$$EI_{pollutant} = \frac{grams_{pollutant}}{kg_{fuel}} \qquad 2-4$$

On the other hand, finding the amount of pollutant in grams requires the product of the specific fuel consumed, time spent during that particular power setting and the emissions index;

$$grams_{pollutant} = EI_{pollutant}(time)(Fuel\ consumed) \qquad 2-5$$

Soot and SOx emissions are not considered in this research because its effect is insignificant as the fuel used is natural gas, whose quality is very high and the sulphur content in the fuel is assumed to be negligible, and the weight and size of the GT is considerably greater than those of aviation GT, which indicates that a considerable increase in the size of combustion chamber is possible, thereby giving enough time for the fuel-air mixture and the soot to react and form CO_2 particles.

With some of the key parameters for this engine, the Lefebvre correlation was used, as it was defined for a fixed combustor with known geometry and the author felt this could be useful [120]. The correlation is set to match the correlations presented by the OEM.

The Lefebvre correlation was also used for predicting the other emissions, which details can be found in Courtinho [121].

2.16.5 Calculations of Emissions

The carbon tax as presented by the Society of Cosmetics Chemists (SCC) varies by source of fuel, the CO_2 produced by this source per unit mass or volume is multiplied by this SCC to achieve the tax. As a result on the mean peer reviewed value ($43/tC or $ 12/tCO_2), the tax for different fuels is as shown in the table below [122, 123].

Table 2-8: CO_2 Tax Estimation [122, 123].

Fuel	Tax (per fuel unit)	CO2 Emissions (mass of CO2 produced)	Tax per kWh of electricity
gasoline	$0.11/US gal ($0.028/L)	n/a	n/a
diesel fuel	$0.12/US gal ($0.032/L)	n/a	n/a
jet fuel	$0.12/US gal ($0.032/L)	n/a	n/a
natural gas	$0.00066/cu ft ($0.023/m3)	117 lb/MBTU (181 g/kWh)	$0.0066
coal (lignite)	n/a	215 lb/MBTU (333 g/kWh)	$0.0121
coal (subbituminous)	n/a	213 lb/MBTU (330 g/kWh)	$0.0119
coal (bituminous)	n/a	205 lb/MBTU (317 g/kWh)	$0.0115
coal (anthracite)	n/a	227 lb/MBTU (351 g/kWh)	$0.0127

It should be noted that the tax of electricity in kWh depends on the thermal efficiency of the generating power plant, and it varies from plant to plant. The American Physical Society (APS) estimate of 10.3 BTU/wh (33%) as shown in table 2-8 [124]. It was noted by the APS that “it is expected that future plants, especially those based on gas turbine systems, will always have higher efficiencies exceeding 50% in some cases". A conversion rate of 100% is 3.412 BTU/wh is theoretical, and a more practical limit for thermal power plants is Carnot's theorem.

2.17 Economic Estimation of Gas Turbine Engine

It is difficult to measure the value of a product or service. This is the reason why people value shares differently in the stock market. It is difficult to identify when to value, what to value or who values. In a likely manner, profit does not equal cash since cash has different values at different times. It is the major reason cost is used in accounting and finance. Risk is another important implication in the sense that the future is uncertain and there is no guarantee that a particular rate of return can be ascertained. It can then be deduced that value is driven by factors such as time, risk, and returns on the initial investment. Returns can be evaluated in terms of profits and cash, whilst risk is a measure of uncertainty and volatility of returns. The time factor means if there is sustainability in an investment and involves the total investment cycle.

Projects involving IGT's, especially in the power generation sector could last for a very long time because it has a well-designed plant life of over 20 years. Therefore, it is very necessary to consider all the three factors which include; returns on profits, risk taken when investing and the time value of money and resulting profits [125].

For over thirty years now, there has been economic improvement in the field of gas turbine power generation ranging from simple unit generating cost calculation (in millions per kWh) to the complex operation analysis and complete systems generating unit cost.

It is obvious that this idea has brought about the improvement of techniques of reliable and economic power systems operation, as it is a function of economic analysis to increase the total profit through an extended time interval, which demands that the economic analysis represents the technical, financial, and the actual operation [126].

A study by Gay et al [127] was published on plant playback, and it described the application of plant optimisation software to find the merits of upgrading plants analysis for a combined cycle power plant. The basis was on the characteristics of the power plant and its operational profile, which encompasses variation in loads and ambient conditions, contract stipulation and so on, sales of electricity and fuel cost, and engine configuration and performance.

2.17.1 Discount Cash Flow Rate of Return

This is the discount rate which results in the future cash flow equalling the initial investment. The project with the highest discounted cash flow rate of return will be the best choice. Projects with a discounted cash flow rate of return that exceed the cost of money are considered worth-while in a case where the investments risks involve are ignored [128].

This method is a broadly used economic tool by competitive business enterprises which examines every cash flow alternatives for the duration of the evaluation. Various worth rate are used to discount each cash flow alternatives.

2.17.2 Investment Pay-back Period

It is used by the utilities and free market enterprise for scoping analysis. It is calculated as the number of years required for the net benefit equal the initial investment. This is used as screening tool to examine a variety of alternatives by business enterprise in a free market system. The enterprise conduct discounted cash flow rate of return on the most promising candidate after narrowing down from the group of alternatives to a manageable size of about 5 to 10. The pay back method is mostly applicable on small discretionary investments, particularly for spare parts or retrofit activities of utility in the regulated utility industry [128].

2.17.3 Minimum Revenue Requirement Method

This is an economic evaluation method that is widely used by regulated utilities, as the rate of return of any investment is based on what is allowed by the regulators. The return herein is the weighted average return on bonds, the bond rating is used to determine the calculated interest, and the regulating commission allows the equity return.

The return to bond holders and return on equity, depreciation and taxes incurred for a number of alternative strategies could be calculated. The one that necessitate the lowest revenue requirements is term the best alternative. Hence, the most preferred project is that with the lowest present worth revenue requirements [127].

2.17.4 Electricity Cost

It is important to estimate the absolute and comparative cost of generating electricity since GT's are broadly used in the electricity generation industries [125].

$$COE\left(\frac{\$}{kwh}\right) = capital\ cost + fuel\ cost + O\ \&\ M\ cost + emission\ cost \qquad 2-6$$

Capital recovery factor (CRF) is used in net present value (NPV), whereas initial investment is changed into a sequence of equal annual payments. CRF is a simple formula which converts present value into equal annual payments over a definite time, at a particular rate of discount [129] and it is presented as [125]:

$$CRF = \frac{1}{\left[\frac{1}{r} - \frac{1}{r(1+r)^n}\right]} \qquad 2-7$$

References [130-133] presented necessary information to evaluate the capital cost of GT plant, annual updated price list of commercial GT packages from different manufacturers, and other important cost data. The 2011 list installed in kilowatt hour (kWh) was transferred into the economic module. The output from the GT in kilowatt (kW) is inputted, the likely price is interpolated from the list, and fuel data, O&M, and emission cost calculations are obtained.

2.17.5 Levelized Cost of electricity (LCOE)

This is another technic for determining economic value of electricity. If every unit of electricity produced by the GT engine all through the period were sold at the LCOE, the income would be equal to the total life cycle cost (TLCC) discounted back to the base year [130], which indicates that if every unit of electricity produced were sold at the LCOE, the project would break even and the NPV would be zero. The formula below indicating emission cost, maintenance cost, fuel cost and capital cost, could be used to determine the LCOE.

$$LCOE = \frac{Capital\_C + \sum_{i=1}^{n} \frac{Fuel\_C}{(1+r_i)^n} + \sum_{i=1}^{n} \frac{M\_C}{(1+r_i)^n} + \sum_{i=1}^{n} \frac{E\_C}{(1+r_i)^n}}{\sum_{i=1}^{n} Power * Operating\ hours\ year} \qquad 2-8$$

The two assumptions stated below could be used to asses LCOE;

- Electricity price does not change all through the period of the project.
- The interest rate, r, used for discounting both costs and revenues does not change along the progress of the project.

LCOE is broadly applied for comparing the costs of different power generation methods, or different operating scenarios using the same technology, despite the problems involved.

2.17.6 Fixed Charge Rate

The utility industries mostly used this concept of fixed charged rates, which is the annual owning cost of an investment as the percent of the investment. A generally used value for the power generating industry is 20%/year. The owning cost to the utility when an investment in utility plant is made and placed into service could include the under listed;

- Income tax
- Charge of depreciation on the investment
- Interest on bond used to partially finance the project
- Insurance and property taxes
- Equity return requirements of the stockholders

2.17.7 Annual Expenses

The annual costs of any system greatly rely on the decision at the design and construction stages. Fuel price greatly affect the operating and maintenance costs to about 70%. Reducing capital and installation costs may cause increase in O&M costs, which will negatively affect the economic performance of the project. For steam cycles, gas turbines , and reciprocating engines, a typical cost of O&M for a co-generation plant as trace in 2004 prices are 0.0035£/kwh, 0.005-0.0115£/kwh and 0.008-0.016£/kwh [134-136].

2.18 Chapter Summary

In the last three decades, the application of industrial gas turbine for power generation has significantly inclined. Market deregulation, increase in fuel price, privatization of industries, and environmental protection are been emphasised as it calls for higher operating efficiencies and reduced level of emissions. One of the most important aspects of gas turbine performance degradation is the compressor fouling and its control, which is emphasised in details. Gas turbine axial compressor fouling and washing was done by [47], where he concluded that 'a judicious combination of online and offline cleaning usually provides the best results in helping operators fight this common and insidious operating problem', close monitoring of compressor performance can help optimize compressor-washing regimes and improve plant profitability. GT off-line optimisation approach for power generation was also carried out by [31], where he concluded that the optimum off-line water washing interval and degradation limits are not fixed, and can vary for each installation depending on these important factors of selling price of electricity, rate of degradation, fuel price and shutdown time. Compressor hot section fouling was also emphasised by [90]. Fouling degradation overview was presented by [138], and GT performance deterioration and other sources has been dealt with by [139], while an insight contribution of GT performance degradation was done by [140], also, non-recoverable deterioration was presented by [141]. The author of this research looked at bridging the gap between the works done by these other immediate authors presented above.

The literature review encompasses series of technical, environmental, and economic considerations in connection with GT power plant selection, performance degradation and also optimisation means by component washing methods and frequency. The optimisation is to ascertain which wash frequency will most reduce maintenance and operating cost, thereby yielding maximum profit for the GT owners and users.

The components of the GTs undergo different types of degradations during operation due to temperature and mechanical loads. These harsh conditions may subject the component to failure which includes environmental attack, thermal fatigue, high cycle fatigue, and creep. Creep is one of the major failure mechanisms for stationary GTs under high stresses and temperatures that reduces the component life significantly, and the effect of creep mostly depends on the engines operating conditions.

Understanding the lifing factors of the failure mechanisms and their interactions will assist the designer to know the trade-off between different options in designs and to also make best decisions by the operators in terms of maintenance procedures.

The economic method is introduced as it is used to carry out the financial calculations. The method of Net Present Value assesses the time value of money and all other modules input into the final modules. Detailed procedures and techniques applied and how all cost involved will be determined.

It can be deduced that component washing could revive the engine health, optimise performance and reduce maintenance cost and increase profit to a reasonable level. The wash frequencies would help the GT user to determine the most effective frequency at which wash maintenance should take place.

The literature review has given insight of what the entire research entails curling different aspect involved and inter-linking them together from the performance, to the creep, emissions and economic benefits.

With all the vital information gathered from literature, the various parameters and their effects on GT will be determined to know which is most affected by the ambient temperature and fouling, as such, there is need to start with the engine modelling and simulations as could be found in a later chapter herein. But before then, a broad analysis of the research methodology will be described and links between each model will be shown in the next chapter.

3 RESEARCH METHODOLOGY

3.1 Introduction

The challenging demand of operating a gas turbine efficiently to meet external needs, satisfying environmental conditions and economic stability for optimal benefit has called for the application of a technique which in this case involved online and offline compressor washing. The technique applied here involves washing the compressor online using intervals of 7days wash frequencies. The compressor offline water wash is assumed to be carried out ones every three months, which makes a total of four offline washes in a year and a combination of both online & offline washing was also applied. In case of the combination, the offline wash was assumed to be done as a major wash after equal interval of online wash, which enables more power augmentation. The reason is to make comparisons of these wash intervals at the most reasonable wash effectiveness to obtain the most economically beneficial means of compressor washing, to help minimise operating and maintenance cost so as to optimise profits in all possible aspects of the operations.

The wash technique applied here incorporated some models and sub-models of the governing equations from different models of the operating environments such as performance, emissions, lifing and economics. The models used in the successful achievement of the research aim which is the operation of IGT's to obtain the required power output at the barest minimum operational cost is as listed below:

- Gas turbine performance simulations
- Turbine blade creep life
- Gas turbine emissions
- Economic modules
- Sensitivity analysis

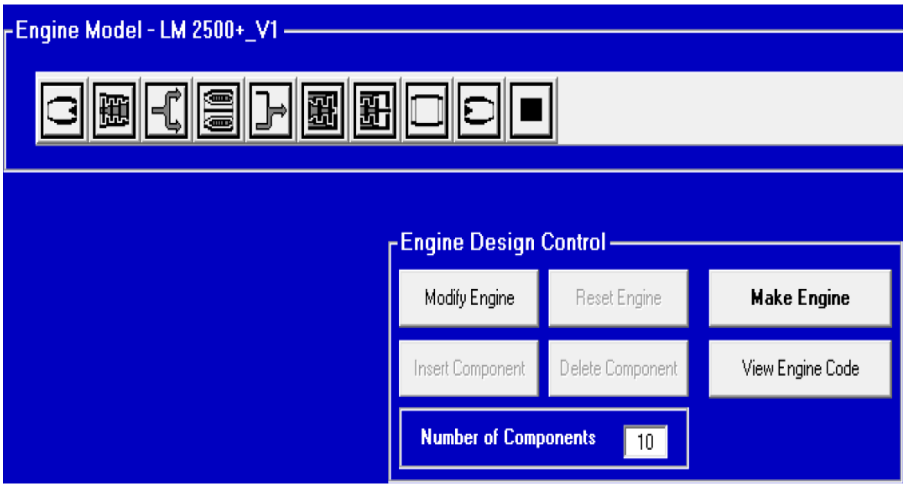


Figure 3-1: Performance simulation engine model

The assembling of each bricks that made up the entire performance model is as shown in Figure 3-1. The function of each bricks and how it links with each other are described in details in the next chapter of this research. The model inputs and outputs are described standards taken as reference from the operating environment for the operation of the engine, and the results are described thereafter.

3.2 Engine Operating Data and Environment

The engine operating data for the performance simulation of this research work was the average monthly ambient condition of a particular city in the western part of Africa (Maiduguri, Nigeria), taken over a year period in 2012. The data gotten was inputted into the engine and simulated, and the output result was used as the baseline engine and hence served as reference to the fouled and washed engine results.

Nigeria climate is tropical and are of wide range in various part of the country. Temperatures around the coast mostly exceed 32 degrees Celsius (90 degrees Fahrenheit), with very high humidity and hot nights. There are two different seasons known as the wet and dry season. The monthly temperatures varies all through the year with the month of July having the highest temperature rising above 40 degree Celsius (100 degree Fahrenheit), with relatively cool nights dropping to as low as 12 degree Celsius (54 degree Fahrenheit) [142].

Figure 3-2: Nigeria map [142]

Along the coast, average rainfall varies from about 180cm (70 inch) in the West to about 430cm (170 inch) in some part of the East. It decreases to about 130cm (50 inch) over most of the central Nigeria and only about 50cm (20 inch) in the extreme North. The hot Harmattan wind from the Sahara sweeps across the North-eastern areas which the Harmattan dry, and carries a reddish dust from the desert, while the Southwest wind brings cloudy and rainy weather. Figure 3-2 is a typical map of Nigeria showing different states in the country. The average monthly temperatures are as will be seen in table 4-1 in chapter four of this thesis work.

3.3 Choice of Baseline Engine

Highly recognized IGT in the market today similar to that of the General Electric (GE's), which could operate in severe climate conditions and remote locations, high efficient inlet air filters, and better materials and could withstand all possible operating environmental conditions were considered. Power output ranging from 13- 100MW for small engines and11- 340MW for heavy duty engines with the use of different fuels, are types of engines one can find for IGT. Its reliability in maintaining power output is unbeatable and it comes in a modular package giving room for shorter manufacturing cycles and faster installation cost with less cost involve. After assembly, a proper test is being carried to ensure its readiness for operation. Additional reasons for its worthiness include; modern manufacturing technology, advanced design procedure, low emission and availability with fast maintenance plan. Not less than 7,000 heavy duty engines are in operation across the globe today with over 200 million fired hours of working experience and wide range of technology option required to meet challenging energy standard.

The LM2500+ is employed here in this research because of it track record based on industrial standard, and could withstand the operating environment employed in this research effectively. It is a GE engine developed from the LM2500 which was metamorphosed from the then GE CF6-6 aircraft engine. It is a single spool GT with an aerodynamically coupled free power turbine. It compression section is derived from the former LM2500 with a redesign of additional zero stage to increase the air mass flow rate and pressure ratio. To improve the GT performance, an upgrade in design is also made in the materials of the hot gas path section.

Table 3-1: Performance information for LM2500+ design point [146]

COMPONENT PARAMETERS	**VALUES**
Power Output	29MW
Mass flow	80kg/s
Compressor Pressure Ratio	23.1
Turbine Entry Temperature	1505K
Exhaust Gas Temperature	518^{0}C (791K)
Turbine speed	3600 rpm
Thermal Efficiency	38%
Combustor efficiency	99%

The compression ratio of the LM2500+ gas generator is 23:1 as compared to that of LM2500 of 18:1 [143]. The engine performance have improved as a result of these modifications, producing more power of about 25% more than the LM2500, with a simple cycle thermal efficiency of about 39% [144, 145]. The engine has over 15 years application experience ranging from LNG plants to cruise ships all over the world, with unsurpassed reliability and availability. This makes it one of the best GT in oil and gas, power generation, combine heat and power (CHP) for independent and commercial power supplies. Its specifications are as outlined in table 3-1.

3.4 Engine Model Incorporation

As could be seen in figure 3-1, and as earlier explained in section 3.2, the engine data was inputted into the engine as will be explained in detail later in this research work. Performance simulations were carried out for the baseline engine; afterwards, the engine was degraded. The result for the degraded engine was also recorded, and later a wash in the compressor was applied at various intervals as explained in session 3.1. These results were also recorded as could be described in the next session.

3.4.1 Gas Turbine Performance Simulation and Modelling

The performance simulations were made with the help of Turbomatch and Pythia software which is an in-house software for gas turbine engine simulation in Cranfield University. Using a hot climate condition as described earlier in session 3.2. The GT performance parameters under investigation were predicted for the design point (DP) and off design point (ODP) possible for the operating conditions. Firstly, the data were gathered and inputted into the performance simulation, the results were validated, and the output results were assumed to be the DP performance of the GT as would be found in the next session of this thesis, which is referred to as baseline performance. The ODP performance for the GT was also investigated for the various cases involved. The parameters considered during the investigation were ambient temperature

(T_{amb}), ambient pressure (P_{amb}) and turbine entry temperature (TET). TET and (T_{amb}) were varied while (P_{amb}) was kept constant in the investigation.

The ODP performance parameters were determined with the help of pythia software which produce the performance parameters for corresponding values of TET and (T_{amb}). These are to obtain the predicted ODP parameters for the GT in investigation, and were further tabulated in dimensional look-up tables where mass flow, fuel flow, compressor temperature, power output etc, were being obtained and inputted into other sub-models to achieve the required engine parameter output.

The reason is to find out the behaviour of a single engine of 29MW industrial gas turbine in three different health conditions of its clean, fouled and washed conditions throughout a year period. The temperature variation for this chosen part of the country ranges between 20^0C and 40^0C as will be explained in details in the next chapter. These was absorbed and incorporated into the off design engine while the behaviours were carefully studied in different cases of:

- Clean behaviour
- Fouled behaviour
- Compressor washed behaviour.

The given temperatures were the maximum experienced temperatures throughout a year period in the region.

The clean engine is assumed to be the baseline engine behaviour, and the fouled engine is the maximum power achievable without wash administered. The online wash was taken at various intervals of 7days throughout the one year period. The offline wash was administered ones in every three months in the year, making a total of four times offline compressor water washes per year, and lastly a combination of both online and offline wash was administered. In all, the maximum recoverable energy was recorded for each case and combination of cases as well. The recoverable power (MW) achieved was converted into energy (kwh), since this energy will be sold as electricity for consumption. The area under the wash curves of the graphs that will be produced later during performance is determined by the application of integration rule for area of a triangle as shown below:

$$Energy\ gained = \int_{Fouled}^{Clean} (All\ intervals)dt \tag{3-1}$$

The cost of power saved, cost of fuel saved, cost of electricity sold, maintenance cost, and every other cost associated with performance for each of the wash case was determined, computed and compared for economic benefits. The TET was varied to retain power during the simulation; as such, the exhaust gas temperature was increased thereby affecting the turbine blade, which has now called for the HPT blade creep life assessments as could be seen in the next session.

3.4.2 Creep Life Estimation Model

The GT creep life is assumed to be determined from the HPT blade. Creep is a main failure mode assumed in this research with the incorporation of physics based model which includes mechanical and thermal stress as could be seen in figure 3.3.

The engine data gathered from public domain were used as input data for the performance simulation. The results will be seen later in the next chapter of this research, and some of the output parameters were later fed into lifing model. The results from the engine performance thermodynamic parameters include; combustor inlet and outlet temperature and pressure, firing temperature, and the rotational speed. The performance output results were fed into the sub-models known as stress and thermal model determination, having the blade material temperature and geometry for the stress model, and the cooling effectiveness ε for the thermal model. Both models were later integrated to perform a parametric study based on the Larson-Miller Parameter (LMP). From here, the creep life and time to failure of the first stage HPT blade of the stationary GT were determined. The LMP and stress values are sets of LMP creep generated from standard tests where the material is tested under specified temperature and loadings as in the Rene 80 shown in Appendix A.

The summary is to develop a creep model which is incorporated with sub-models such as thermal and stress analysis, with pythia output results. The output of the thermal and stress model were later integrated into the creep model (LMP) to determine the remaining creep life of the blade.

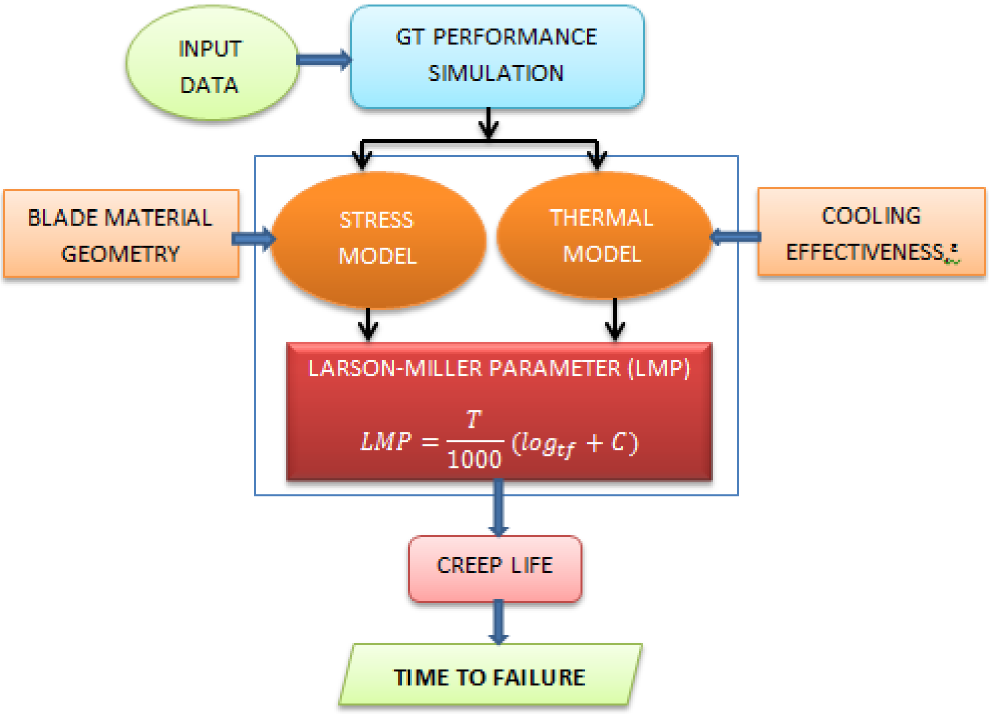


Figure 3.3: Creep Life assessment model

- **Thermal model**

It is used to calculate the blade metal temperature (BMT) for the first stage of the turbine. Based on the blade's cooling technology, the value of the overall effectiveness is first calculated. The cooling air (T_{cin}) entering the blades, and the gas stream temperature (T_g) surrounding the blades are both determined from the pythia simulation output result. The blade technology and outlet temperature of the nozzle guide vane (NGV), which is determined with the output result from Pythia for gas stream temperature and inlet cooling temperature, with the use of equation 3-3, as could be seen in figure 3-4, determines it overall cooling effectiveness [147].

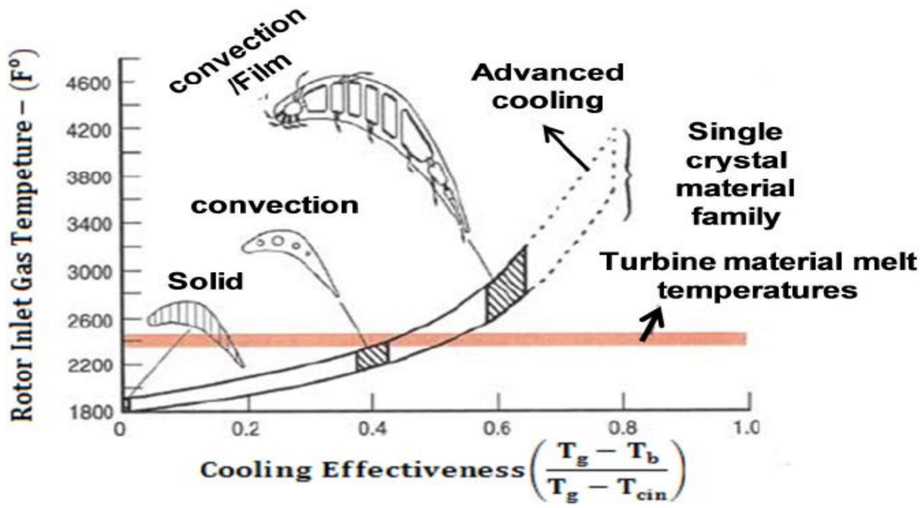


Figure 3-4: Turbine blades cooling technology [147].

The blade metal temperature T_b is calculated with this model, and the model is updated continuously as the temperature changes resulting from change in operating conditions. The component creep life is being calculated according to T_b [148].

$$\varepsilon = \left(\frac{Tg_{-}T_b}{T_{g-}T_{cin}}\right) \qquad 3-2$$

Here, ε is known as the cooling effectiveness, T_g is the gas stream temperature while T_{cin} is the inlet cooling temperature. The blade metal temperature is obtained when equation 3-2 is being re-arranged, as could be seen in equation 3-3:

$$T_b = T_g - \varepsilon\left(T_g - T_{cin}\right) \qquad 3-3$$

The high demanding rate of higher efficiency in GT calls for higher firing temperature, but this firing temperature increases turbine blade metal temperature which reduces blade life. To stabilize the metal temperature, internal and film cooling are used together, but the metal temperature is still above 950^0C on the surface [149]. The blade cooling reduces the temperature when increasing TET.

- **Stress model**

Amongst the various types of stresses in turbine blade, direct centrifugal stress is considered due to the mass of the blade. Between 50 to 80% of the blade material strength is used to overcome this stress operating in an inertia field [150].

The blade material properties/ geometry were gathered and applied accordingly with the use of Rene 80 master curve, creep factor was determined, and all necessary steps were taken to determine the remaining creep life of the blade.

The centrifugal stress on the blade was evaluated from root to tip, to enable creep life calculation. Results generated by pythia were used as input into this model. The blade was sub-divided into several sections as could be seen in figure 3-5. The axial velocity was assumed to be constant along the blade span, with centrifugal force acting on the centre of gravity (CG) of the blade section. On the rotational section, the centrifugal force is given as [150]:

$$CF_{sec} = mass\ x\ \omega^2 x\ d_{cg} \qquad 3-4$$

Here, $mass$=mass of the component, ω =the angular speed of the component, d_{cg} =distance between the rotational axis and the section CG.

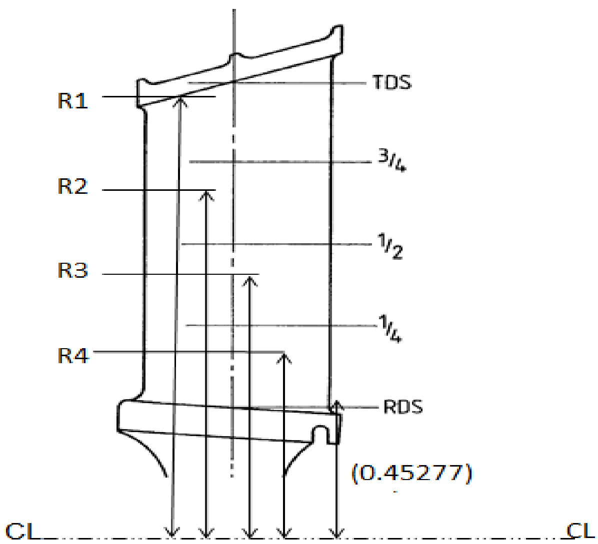


Figure 3-5: Blade showing Radii to Centres of Gravity of Section

If the blade is assumed to be rectangular in shape, its mass will be equal to density*average cross-sectional area*height and the centrifugal force could be determined by the use of equation 3-5:

$$CF_{sec} = \rho \; x \; A_{AVCS} \; x \; h_{sec} x \; \omega^2 x \; d_{cg} \qquad 3-5$$

Here, ρ = material density, A_{AVCS} =average cross-sectional area between the top and bottom of the section, h_{sec} =section height, ω^2 = angular speed, and d_{cg} =distance between the rotating axis and the section CG. In determining the centrifugal stress acting on a blade of constant cross-sectional area, the following equation is suitable:

$$\sigma_{sec} = \rho \; x \; h_{sec} \; x \; \omega^2 \; x \; d_{cg} \qquad 3-6$$

The sources of stress that could be examined when building the stress model is as stated below:

- Stress due to axial gas bending moment as a result of static pressure change.
- Stress due to axial gas bending moment resulting from change in momentum
- Stress due to tangential gas bending moment as a result of momentum change
- Stress due to centrifugal loading

This thesis considered centrifugal loading due to rotation of the blade.

- **Creep life**

The LMP approach is used to obtain reasonable creep life estimation, both at the current and reference operating conditions. According to Arrhenius law [150] the equation can be expressed as shown in 3-7:

$$LMP = \frac{T}{1000}\left(Log_{tf} + C\right) \qquad 3-7$$

We can obtain t_f by re-arranging the equation as 3-8:

$$t_f = 10^{\left(\frac{1000LMP}{T} - C\right)} \qquad 3-8$$

Here, T = absolute temperature of the material in Rankine, tf = time of failure in hours, and C = a constant which is usually assumed to be 20 for industrial applications, though vary between 13 and 27 depending on the material in use [151]. The creep life vary along the blade sections since the temperature is varied, thereby varying the stress. The minimum calculated creep life will be taken as the value which represents the blade's remnant life.

The number of cycles to failure accumulatively can be estimated using Miner's rule mathematically expressed as;

$$Fraction\ of\ Life\ Consumed = \sum \frac{t_i}{t_{fi}} \qquad 3-9$$

Where t_i is time spent at a particular stress/ temperature combination, and t_{fi} is the time to failure.

The time to failure is as expressed in equation 3-10 considering a factor of safety of 60%.

$$Time\ to\ failure = \frac{8760}{Fraction\ of\ Life\ Consumed} \qquad 3-10$$

- **Model layout for lifing**

As could be seen in figure 3-3, the lifing model is divided into three major parts which include;

Part one is defined by the user and it comprised of blade material, blade material geometry, number of segments into which the blade is divided, and the cooling effectiveness of the blade which is used for thermal analysis.

Part two is curled from thermodynamic performance, and includes; compressor exit temperature, RPM, and firing temperature at each operating condition.

Part three is from the data source of the material in use, and it includes; LMP constant (C), blade material density, and mainly test house data for the LMP data material.

Before the creep is determined, the temperatures along the HPT blade sections were first determined for five sections, and the averages were taken to attain the gas stream temperatures of the equal four sections of the blade. Afterwards, the TBC temperature was subtracted, and the blade metal temperature for each blade sections was determined by the application of equation 3-3. Thereafter, equation 3-8 was applied to determine the creep life of the blade at the various sections. Details will be seen in the creep determination session later in this research work.

3.5 Exhaust Emissions Model

A simple and reliable semi-empirical method was applied in this research for the determination of GT exhaust emissions. The inputs for the semi-empirical model provides estimate for the aero-derivative combustor where readily available. Figure 3-6 shows a schematic view of the emissions model employed in this research work.

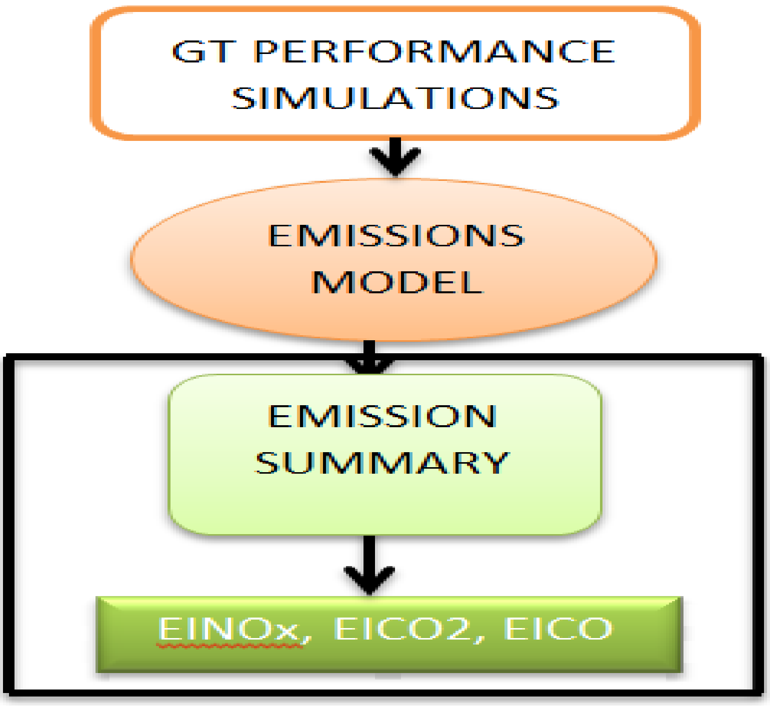


Figure 3-6: Schematic diagram for emission model

The output result from GT performance was fed into the emission model with the help of an empirical model from the Lefebvre correlation for the NO_X and CO_2. The CO_2 is considered because of the exhaust emission from the GT since it is a natural consequence for combustion, and the NO_X is also considered because of the variation in firing temperature to augment power. This increased firing temperature or fuel-to-air ratio will increase the amount of NO_X emission to the operating environment, NO_x increases with increase in residence time in the flame zone; it increases with the square root of the combustor inlet pressure, and increases exponentially with combustor inlet air temperature. The UHC and CO were not considered in the economic loss since the engine is assumed to be operated with natural gas and there is assumed to be complete combustion, though CO was assumed when comparing it effect with high temperature to that of NO_2. The emissions model is divided into three sections as could be seen below;

Section one: In this case the user defined the values for the combustor such as; residence time, combustor efficiency, altitude, switch between aero and industrial frame combustor types input, type of fuel i.e natural gas, kerosene etc, primary zone evaporation fuel volume, combustor mode i.e constant volume or constant pressure, combustor volume, number of combustor chamber, fraction of primary zone occupied by air.

Section two: Performance result thermodynamic parameters such as; firing temperature, mass flow, pressure before and after combustor chamber, compressor outlet temperature, and fuel flow.

Section three: No set input but rather a known combustor characteristics correlations, the known characteristics is mapped as an empirical equation in spreadsheet, and each emission has its own correlation.

Simple combustor geometry is presented with inputs set one such as primary zone information, combustor efficiency, fuel type, and residence time. The performance output results are been fed as input set two which includes fuel flow, core mass flow, temperature and pressure. The heat enthalpy of fuel, T3, P3, and proportions of air in the primary zone which are all thermodynamic performance parameters help in estimating the adiabatic flame temperature (AFT). The combustion volume, AFT, core mass flow and pressure (P3) which are entered into the turbine used in estimating the emission indices. The coefficients of the emission index equations are revised to fit the known trends for a given type of combustor and the correlations are then computed as in section three above.

The empirical correlations employed for NO_X determination in this research is that of Lefebvre as could be seen below;

$$NOx = \frac{9x10^{-8}P^{1.25}V_c\ \exp(0.01T_{st})}{m_A T_{pz}}\ \frac{g}{kg}fuel \qquad 3-11$$

Where P_3 is the combustor inlet temperature, T_{st} is the stoichiometric gas temperature, T_{pz} is the primary zone temperature, V_c is the combustor volume, and m_A is the air mass flow.

For a combustion chamber assumed to have complete combustion in the presence of excess air; the quality, quantity and type of fuel used will determine the amount of CO_2 produce. The stoichiometric equation 3-12 is used to calculate the amount of CO_2 emitted. The fuel analysis approach for estimating CO_2 emission is as shown in equation 3-12 [152];

$$C_xH_y + nO_2 \rightarrow n_1CO_2 + n_2H_2O \qquad 3-12$$

In the case of carbon tax, the equation 3-13 is employed to determine the cost of carbon emitted to the environment [153];

$$Carbon\,Tax = mCO_2\ \ x\,Tax\,Rate\,\frac{\$}{Kwhr} \qquad 3-13$$

Assumptions

- The combustor geometry used is the simplest, and the combustor efficiency, fuel type, primary zone and residence time information is inputted in part one.
- The performance model results gives information about the operating conditions such as pressures, mass flow, temperatures and fuel flow which are inputted in part two.

- The thermodynamic performance parameters T3, P3, the heat enthalpy of fuel and proportion of fuel in the primary zone estimates the adiabatic flame temperature AFT.
- The combustor volume, AFT, mass flow, and pressures (P3) which enters the turbine are used to estimate the emissions indices.
- The coefficients of the emission index equations are revised to fit known trends for this particular type of combustor.
- The correlations are later inputted to part three.
- The emission NOX is calculated based on known correlations, while the emission CO_2 is based on stoichiometry.

3.6 Economic Model

The economic model comprised of the integration of several models in this research work, which include the performance, lifing and emission as could be seen in figure 3-7. The effects of the ambient/operating conditions were considered during the performance simulation, and the output results were inputted in the lifing and emissions model to determine the creep life and emission index respectively as described earlier. The output results were later used for the determination of the economic implications for each compressor wash case and frequencies.

- **Economic Optimization Model**

The main aim of this research is to improvise an optimiser which is capable of integrating the whole models so as to be able to achieve reliable economic measures of the system such as maintenance, fuel consumption, operating cost, and other cost and cumulative revenues that can be evaluated with time along the operating horizon. This system level economic metrics provide a basis from which to evaluate long and short term plant profitability when optimizing operational performance [154]. Integrating local economic metrics along the entire operating time horizon is one of the approaches, and such an approach requires the accuracy and efficiency to be balanced evaluating long term economic measures and inter-relationships. The optimization method applied herein is compressor water wash. The economic approach applied in this study is as summarised in figure 3-7.

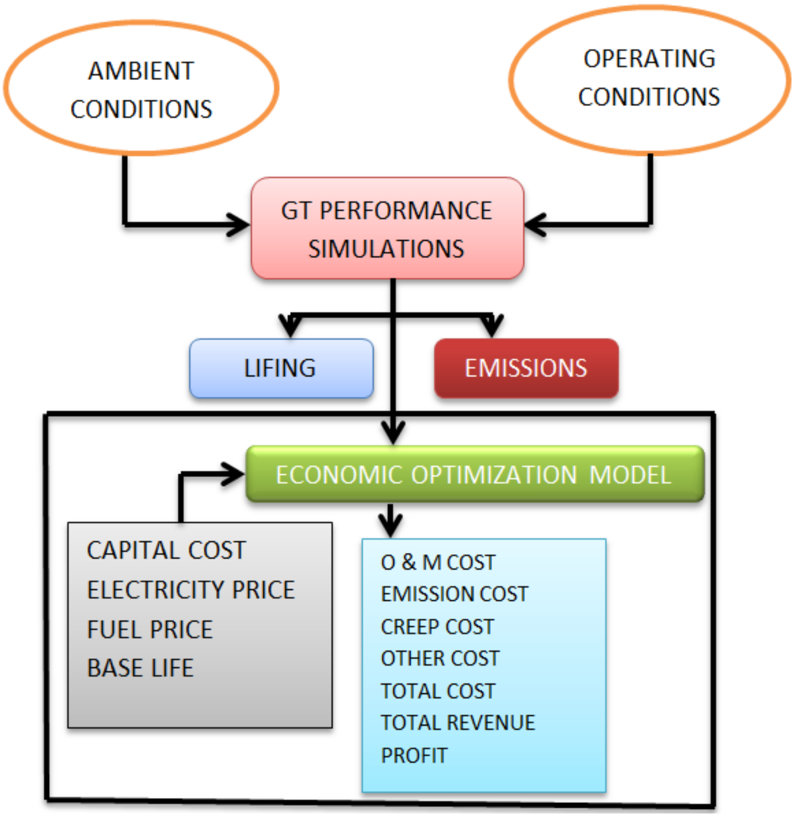


Figure 3-7: Economic Model for GT Plant

- **TOTAL REVENUE**

Revenue is all gain made. Including total sales, exchange of assets, increase in owners' equity and interest. Calculated before any expense is subtracted.

$$TR_T = \int EP(t) * P(t)dt \qquad 3-14$$

Where: TR_T is the total revenue over time T ($).

$EP_{(t)}$ the projected price of electricity at time t ($/kwh).

$P_{(t)}$ is the electricity power output (Energy gained) of the plant at time t (MW) (kwh).

- **ELECTRICITY PRODUCTION AND PRICE**

Energy produced is source of income to GT owners. For each wash case calculation, the total electricity produced is taken to be the sum of energy gained multiply by the cost of electricity in kwh for the period of operation.

$$Annual\ EP = \sum EP * Energy\ gained\ \ from\ washing \qquad 3-15$$

- **LIFE CYCLE COST**

This is the sum of the total fixed cost and the total variable cost. It is also known as the ownership cost. The capital cost which comprises of the cost of equipment purchased, installation and transportation, and fixed operation & maintenance cost are known as the fixed cost. While the fuel cost, operation & maintenance and repair cost are classified as variable cost. This is known as operating cost.

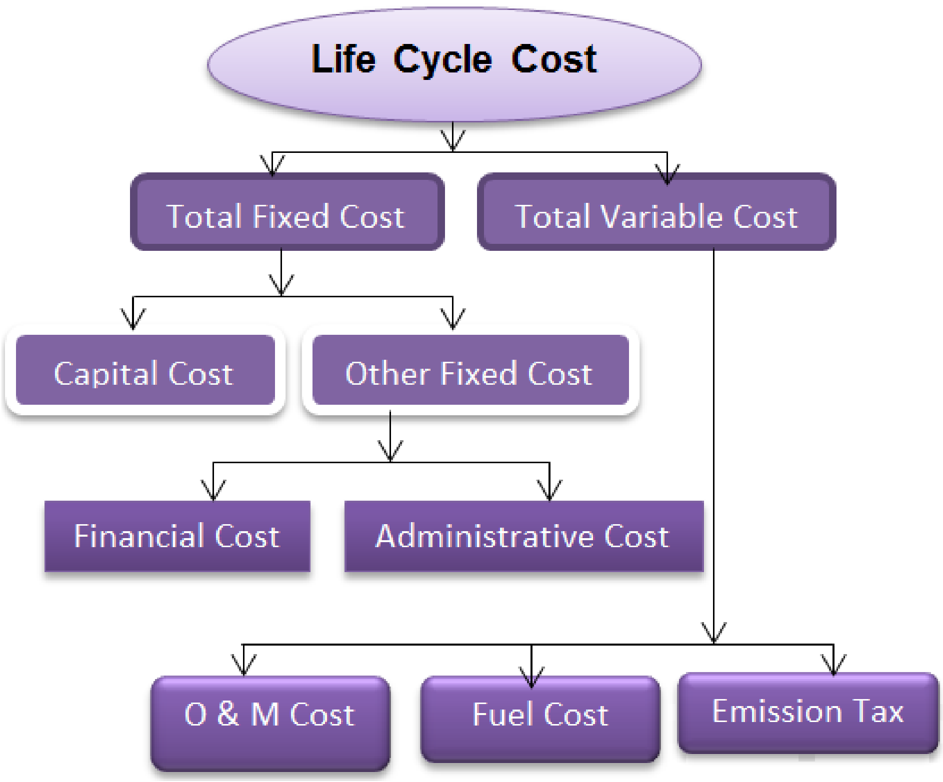


Figure 3-8: Schematic layout for Life Cycle Cost

Life Cycle Cost = Capital Cost + Fuel Cost + Operation and Maintenance Cost + Emission Tax + Financial cost

$$LCC\ (pa) = CC_{pa} + FC_{pa} + O\&MC_{pa} + ETC_{pa} + FinC_{pa} \qquad 3-16$$

Where: CC is the capital cost per annum.

FC is the fuel cost per annum.

O&MC is the operating and maintenance cost per annum.

ETC is the emission tax cost per annum.

FinC is the financial cost per annum

- **TOTAL PROFIT**

This is the total gain in each wash case.

$$TP = TR - LCC \qquad 3-17$$

Considerations are capital cost, maintenance cost and fuel cost.

For a given time (T), the TP is:

$$TP_{pa} = TR_{pa} - LCC_{pa} \qquad 3-18$$

Where: TR_{pa} = No of electricity sold x mean price per unit for a year. TR is the total revenue obtained from the sale of electricity.

LCC_{pa} is the life cycle cost per annum which includes cost of wash fluids, operation and maintenance etc.

TP is total profit.

- **FUEL COST AND CONSUMPTION**

$$Fuel\ Consumed = Fuel\ Flow * 3600 * Operating\ hours \qquad 3-19$$

This is converted to energy units called Threm. It is the unit for the natural gas price. The mass of fuel can be converted to Threm and the price can be determined when the lower heating value (LHV) of natural gas is known.

$$C_{fuel}(T) = F_p * F_c = \left[\frac{LHV * F_p}{105506}\right] * F_c \qquad 3-20$$

Where: $C_{fuel}(T) = Cost\ of\ fuel\ for\ the\ period\ of\ time\ (T) \left(\frac{£}{MJ}\right) \qquad 3-21$

$$F_p = Fuel\ price\ (\frac{p}{threm})$$

$$F_c = Fuel\ consumed\ (\frac{kg}{s})$$

- **OPERATING AND MAINTENANCE COST**

It is dependent on factors like fuel type, operation cycle and environment, and operation and maintenance quality. It is divided into fixed cost FC =staff, insurance etc and variable cost VC= wash fluids etc.

- **CO_2 EMISSION CALCULATION**

The fuel type used for this research is assumed to be natural gas and the equation governing the formation of CO_2 during the burning of hydrocarbon fuels is as shown in equation 3-12 ;

$$C_xH_y + nO_2 \rightarrow n_1CO_2 + n_2H_2O$$

Where x/y is the carbon-hydrogen atomic ratio of the fuel, and the mass of fuel is presented as;

$$m_f = \frac{E}{Efficiency * FCV} \qquad 3-22$$

$$eCO_2 = \frac{44}{12} * C \qquad 3-23$$

$$mCO_2 = eCO_2 * m_f \qquad 3-24$$

$$Carbon\ Tax = mCO_2 * Tax\ Rate \frac{\$}{MWhr} \qquad 3-25$$

Where: Mf is the mass fuel consumption

C is the mass content of carbon in fuel e.g kg C/kg fuel

mCO_2 is the mass of CO_2 emitted

E is the useful energy produced by the system

eCO_2 is the emission CO_2 per unit mass of fuel e.g kg CO_2/kg fuel η is the efficiency of the system based on the lower heating value of fuel

FCV is the lower fuel calorific value

- **OBJECTIVE FUNCTION OF TOTAL PROFIT**

Based on the ambient and operating conditions, and the economic aspects discussed earlier, the profit equation for one year operation can be as stated below for the various compressor water wash optimisation cases:

$$\left[\sum_{t=1}^{t=12}\Big(\big(E * EP - (CC + FC + O\&MC + ETC + FinC)\big)\Big)\right] \qquad 3-26$$

For fouled:

$$\left[\sum_{t=1}^{t=12}\Big(\Big(E_{f(t)} * EP_{f(t)} - \big(CC_{f(t)} + FC_{f(t)} + O\&MC_{f(t)} + ETC_{f(t)} + FinC_{f(t)}\big)\Big)\Big)\right]$$

For online wash:

$$\left[\sum_{t=1}^{t=12}\Big(\Big(E_{on(t)} * EP_{on(t)} - \big(CC_{on(t)} + FC_{on(t)} + O\&MC_{on(t)} + ETC_{on(t)} + FinC_{on(t)}\big)\Big)\Big)\right]$$

For offline wash:

$$\left[\sum_{t=1}^{t=12}\Big(\Big(E_{of(t)} * EP_{of(t)} - \big(CC_{of(t)} + FC_{of(t)} + O\&MC_{of(t)} + ETC_{of(t)} + FinC_{of(t)}\big)\Big)\Big)\right]$$

For combine wash:

$$\left[\sum_{t=1}^{t=12}\Big(\Big(E_{co(t)} * EP_{co(t)} - \big(CC_{co(t)} + FC_{co(t)} + O\&MC_{co(t)} + ETC_{co(t)} + FinC_{co(t)}\big)\Big)\Big)\right]$$

The optimum wash case is identified, presented and advised accordingly, as it will help the engine operator in determining the best cost effective means of carrying out GT maintenance in terms of compressor water wash.

3.7 Wash Methods, frequencies and effectiveness for performance enhancement & economic viability.

From the performance simulation, the engine was assumed to be working on base load at varying TET to augment power reduction resulting from fouling.

The optimisation procedure was said to be compressor water washing for various cases which includes; online water wash cases for 7days intervals between each wash, an offline water wash was also administered at equal intervals four times all through the year, which is a major overhaul involving engine shutdown. The third case was the combination of both online & offline compressor water washes. The online water wash was assumed to be carried out immediately after each offline washing for the combine case. As such, the offline water wash was assumed for two times per year since there is an existing online water wash before each offline wash.

The cost involved in the various cases were computed and compared, considering, the cost incurred from frequent online wash, cost of engine shutdown from offline wash, operation and maintenance cost, wash fluid cost, cost incurred from emission tax, life cost, and all other associated cost; were all analysed and summed for each case and compared, afterwards the optimum wash case was presented. A sensitivity analysis was made afterwards to show the effect of engine shutdown, electricity price, fuel price and degradation on net profit on the total performance of the engine.

3.8 Summary

The step-by-step approach to this research success is as stated in this chapter. The function of the individual model and the link between the various models have been identified and explained both analytically and schematically. The input data and operating environment of the GT, the performance simulation output results and how it is been fed into the emissions, lifing model, and the incorporation of the creep life result, and emissions output to the economic model in determining the economic output results at the various washes for online, offline, and combination of both. The results from this approach will be shown later in this research work.

4. GAS TURBINE SIMULATION AND PERFORMANCE MODEL

4.1 Introduction

There is bound to be failure in IGT performance if kept in use for a prolonged period without proper adherence to maintenance procedure and safety rules. These will in turn result in economic set back to the GT users or owners as the case may be.

The reason for the choice of engine and it specifications are as discussed earlier in session 3.3 and table 3-1 respectively. It has high availability and reliability, high efficiency, simple modular design for easy maintenance, features a split compressor casing, and in-place hot section maintenance and external fuel nozzle which is suitable for the intended environment like Nigeria.

The analysis of the comparisons between clean and degraded engine performance at measured operating conditions, and at varying wash effectiveness will also be discussed as would be seen later on in this chapter.

The content of this research is summarily about improvising models capable of enhancing decision making in power generating project involving the use of gas turbine engines. The various models have been divided into four;

- Performance model
- Creep life model
- Emissions Model, and
- Economic model.
- Sensitivity analysis

The incorporation of these models introduces economic frame work for GTs, which is then connected with the component wash frequency that serves as an optimiser for power generation.

The performance simulations were made with the help of PYTHIA/TURBOMATCH software, for a gas turbine engine operating under a hot climate condition of the Northern part of Nigeria, with the hottest temperature variation in the country as stated in session 3.2. The reason is to find out the behaviour of a single engine of 29MW industrial gas turbine in different health conditions of its clean, fouled, online, offline, and combine washed situations throughout a year period. The temperature variation for this chosen part of the country ranges between 20^0C and 40^0C as could be seen in table 4.1 below. These were absorbed and incorporated into the off design engine while the behaviours were carefully studied in different cases of:

- Clean Engine
- Fouled Engine and
- Compressor washed

The given temperatures are the maximum experienced temperatures throughout a year period in the region.

Table 4-1: Weather condition for Maiduguri, Nigeria [142]

Climate Data for Maiduguri in Nigeria		
Months	Temperature (Celsius)	Temperature (K)
January	20	293
February	21	294
March	25	298
April	30	303
May	36	309
June	39	312
July	40	313
August	39	312
September	37	310
October	33	306
November	28	301
December	22	295

The results from the simulation performance are as described in the following subheadings for various engine parameters and their effects as a result of the monthly temperature variation.

Studying the behaviour of the engine at off-design conditions such as varying ambient temperatures and power settings is a key technical objective. This will enable the determination of the parameter that has the highest effects on the operational strategies, in order to maximise the total profit for the gas turbine operator, while satisfying other constraints which will be the total demand in this study. It also involves the sensitivity of each of the parameters to the GT, since the parameters are required as variable parameters by the optimisation tool. The different components of the engine are as described below:

- **Inlet**

It consists of the bellmouth and the bullet nose. Provision is made on the bellmouth for a spray manifold with the reason to inject water and or detergent during compressor washing.

- **Compressor**

The compressor of the engine employed in this research is a single rotor, variable stator 16 stage axial flow device with overall pressure ratio ranges between 18-24 depending on the series. Titanium and nickel based alloys are the materials used for the production of the stators and rotors. Some of the compressed air is extracted for cooling the hot section of the engine.

- **Combustor**

The dome, cowl assembly, inner and outer skirt are the four major components riveted together for the annular combustor. It is fitted with 30 square nozzles in the individual swirl chambers. The inner walls are film cooled by air introduced through small holes on the combustor walls. The ignition system comprised of two ignition units and is used only at start-ups since combustor is sustained afterwards.

- **High Pressure Turbine**

There are two axial flow stages that drive the compressor. The discharged air from the compressor cools the blade in these stages. The cooling is by internal convection and external film cooling. Convection and impingement-air cooled are the two stages of the HP nozzle assemblies. They are also coated to improve corrosion, erosion and oxidation resistance.

- **Power Turbine**

This is also an axial flow turbine and consists of 6 stages. The stators contain two casing halves and the blades contain interlocking tip shrouds for low vibration levels, and the rear frame of the power turbine forms the exhaust flow path.

- **Accessory Drive Section**

It consists of an inlet gearbox in the hub of the front frame, a radial shaft and a transfer gearbox fastened underneath the front frame. The starter, fuel pump and air/oil separator are mounted on the transfer gearbox.

- **Fuel/Control Systems**

This system comprised of centrifugal and displacement pumps, high pressure fuel filters, fuel control, fuel shut off and drain valves, fuel manifold and fuel nozzles. The system is operated hydro-mechanically in principle and it uses fuel as the servo fluid. The control is such that excess fuel flow is by-passed to the high pressure pump. The generated speed of the power turbine, compressor discharged pressure, compressor inlet temperature are governed by the system. The movement of the compressor variable stator vane is scheduled by fuel control as a function of the gas generator speed and compressor inlet air temperature to maintain compressor efficiency and stall margin for all speeds.

4.2 Engine Performance Simulation Model/ Description

The Cranfield University in-house simulation software known as pythia [58] was employed for the simulation with engine performance parameters as could be seen in table 3-1. It can be used to determine gas turbine behaviours and performance estimation at design and off design point. The

TURBOMATCH/PYTHIA output results include power, fuel flow, and fuel consumption [10]. Turbomatch is built based on pre-programmed routines called “bricks” each of which simulates the performance of one of the component parts of the GT. it is necessary to determine the engine configuration in other to model the GT’s off-design point performance, and to build it from the component parameters. Station numbers are used to connect components. The gas state is described by a station vector at each gas state which contains eight quantities. BRICK requires is done by the use of “codewords”.

Inputs:

The inlet station vector

BRICK DATA=BD (K) to be input (Efficiencies, Pressure ratio, pressure loss factor, etc

Engine Vector Data (Compressor work, generated in COMPRE and be used in TURBIN) [10].

Outputs:

The outlet Station Vector

Engine Vector Results (Thrust, Power, Compressor work),

Inputting experimental results in a Turbomatch input file will require making a model of the engine and building it in a modular fashion by using various pre-programmed units called bricks.

INTAKE: intake

COMPRE: Compressor

PREMAS: Splitter, Bleed, By-pass duct

DUCTER: Mixing of two flows

BURNER: Combustor

TURBIN: Turbine

NOZCON: Convergent nozzle

PERFOR: Final calculation of performance.

The simple cycle LM2500+ [10], an industrial and co-generation two shaft GT have been selected for this study. It is used in a wide range of power generation as simple or combined cycle but can also be used as mechanical drive. The simple cycle power generation is the application herein as could be seen in table 3-1. The Turbomatch model is built for the engine as shown schematically in figure 4-1, and the engine blocks are as set-up in figure 3-1. The simulation procedures in Turbomatch have been followed to carry out the engine modelling. The objective was to simulate the engine behaviour at varying ambient conditions all through the twelve months of a particular year. The monthly temperatures are as shown in table 4-1. The output result was taken to be that of the clean (baseline engine), and afterwards, the degradation resulting from fouling was inputted. Thereafter, compressor water wash cases were administered to show the effects of washing on power output and other parameters affected.

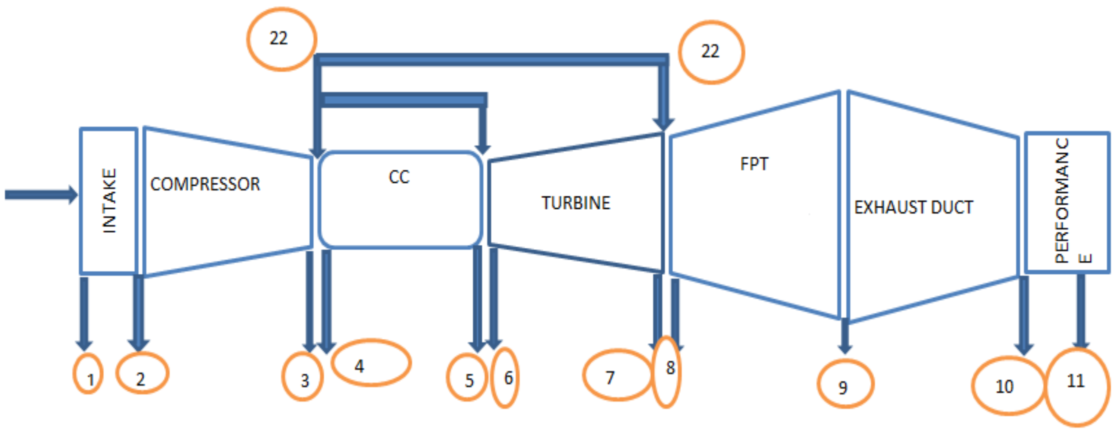


Figure 4-1: Schematic engine layout for GE LM2500+

4.2.1 Design Point Performance

The design point of a GT engine could be defined as the actual point in the operating range of an engine where it is running at a particular speed, mass flow, power output, and pressure ratio for which it is designed [10]. The engine design point performance employed in this research is obtained from the Pythia output file of a simulated engine similar to that of GE's LM2500+ engine in the sense of it features. After simulation, considering the stipulated operating environment, the output results compared reasonably with that of the real engine, as such, it was adopted and assumed to be the reference point (baseline engine) for further discussions in this work.

For an optimised design in GT engine operation, the most important requirements are low fuel consumption with high power output. These can be obtained by employing high pressure ratios and high TET. Adding different components such as intercooler (decreases the work of the compression for the high pressure compressor, which allows for higher pressure ratio, thus increasing overall efficiency), heat exchanger or combination of the two to the initial cycle could improve the GT efficiency, but it will be an additional cost as well. As such, some other economic considerations could be made in this research work for the optimisation, such as; the initial cost, life cycle cost, operating and maintenance cost, fuel cost, price of electricity, emission tax and other cost associated with the progress of engine optimisation as will be discussed later in details. The specific power increases with increase in TET and also with increase in pressure ratio. Selecting GT parameters that most affect the optimisation process is very vital because it is key sign for a good optimisation. It is very important to note that TET cannot be increased indefinitely since there are temperature limitations on the material and also the cooling system for the turbine blades.

In the GT performance, pressure and TET strongly affects thermal efficiency and specific power. But the maximum specific power and the maximum thermal efficiency do not occur at the same pressure ratio. This means, when designing a GT, the design pressure ratio should be a compromise between the maximum specific power and the maximum thermal efficiency which means minimum specific fuel consumption so as to decide whether to employ a small engine with high specific power or an efficient engine with low specific fuel consumption and this directly relates to the capital cost and annual cost for the engine as will be seen later in details herein [155].

4.2.2 Off Design Performance

The ambient conditions effects on the engine performance are the key considerations employed for the off-design point of this research, following the effect of fouling to compressor degradation on engine performance. The performance is generally given based on the operating environment of the engine, as could be seen in table 4-1. The change in steady state performance is investigated. Since the operational conditions have been determined, it is possible to investigate the various performance behaviours for the GT, such as power output, thermal efficiency, fuel flow, mass flow etc.

4.3 Effects of Ambient Conditions on GT Performance

The thermodynamic performance simulation of the given engine is based on a particular site location with specific ambient conditions. The variations in ambient conditions for the engine operation is as explained in section 3.2 in this research. The data is a yearly occurrence of ambient temperature variations, and a particular year is selected for the performance simulation to show the effects of ambient conditions on the engine performance parameters at each month all through the twelve months of the year, afterwards, the yearly average temperature is used for the performance simulation in this research. Controlling the humidity and temperature of air entering the combustion inlet is one of the most important, as it has a direct effect on the turbine efficiency, emission, and operational reliability.

The figures (4-2 to 4-7) herein are plots from data simulated from tables 4.1, representing the engine behaviour at varying ambient conditions for clean and degraded comparisons. The clean, degraded and washed engine behaviour with different wash cases is represented as shown herein. All the engine parameters are decreasing as a result of increase in ambient temperature, except (exhaust gas temperature) EGT which is increasing due to component fouling. If the degradation continues in this manner, the power output will decrease and more fuel will be burnt, making the engine life shorter, emitting more gases, and thereby yielding excessive maintenance cost.

There is variation in thermal efficiency as a result of ambient temperature variation. The thermal efficiency reduces with increase in ambient temperature and increases with reduction in ambient temperature. It could be seen from figure 4-2 that between the first and the seventh months, there is decrease in thermal efficiency which is as a result of increase in ambient temperature, and between the seventh month and the twelfth month, there is an increase in thermal efficiency due to decrease in ambient temperature, leaving the seventh month of the year as least efficient since it has the highest ambient temperature of 40^0C. It is investigated that the thermal efficiency for the online washed engine is better than the fouled engine. On average through the whole year (twelve months period), the thermal efficiency experienced a drop of 1.32% for the fouled engine, 0.40% was recovered from the online compressor washed, while the offline and combine offline and online compressor washed yielded 1.06% and 1.19% respectively.

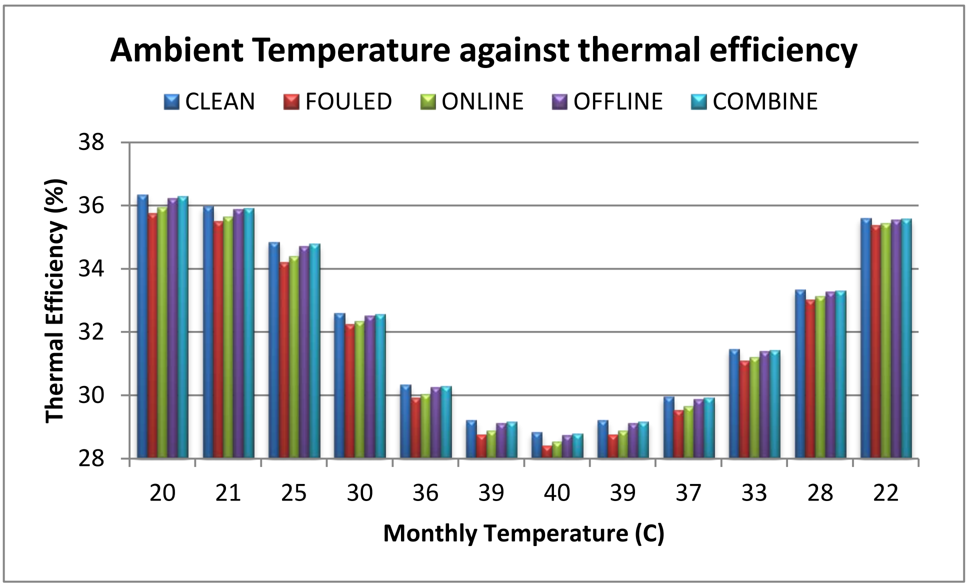


Figure 4-2: Effects of Ambient Temperature on Thermal Efficiency

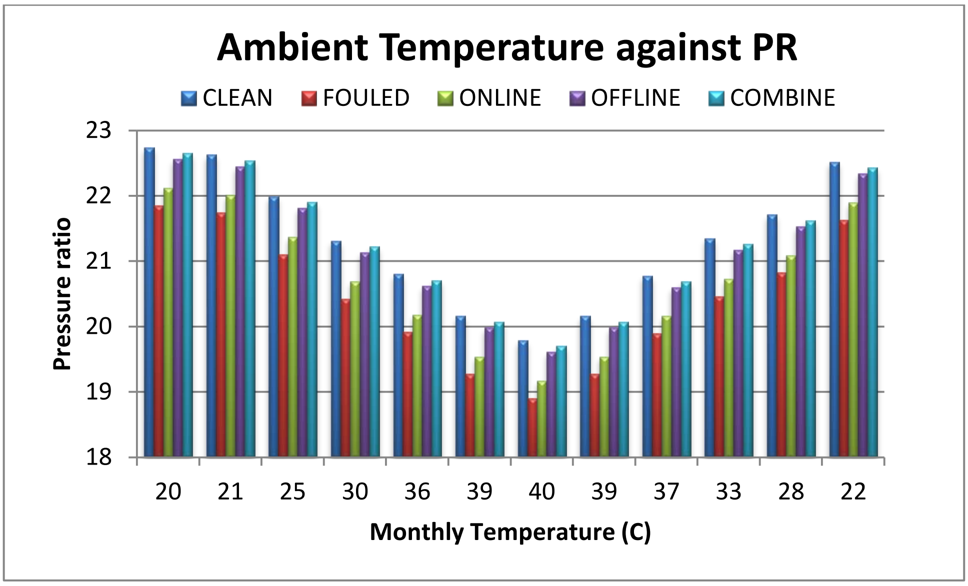


Figure 4-3: Effects of Ambient Temperature on GT Pressure Ratio

The pressure ratio experienced a drop of 4.15% from the fouled behaviour and after the online compressor water wash a recovery of 1.28% was observed, in a similar way, the offline and the combination of offline and online water experienced a hooping recovery of 3.34% and 3.75% respectively as could be seen in figure 4-4. The pressure ratio has a similar trend as the thermal efficiency, as it reduces with increase in ambient temperature and increases with decrease in ambient temperature. Months one to seven shows decreases, as seventh to twelfth month shows increase in PR with the seventh month being the least resulting from the highest ambient temperature. Increase in PR result in increase in the thermal efficiency and vice versa.

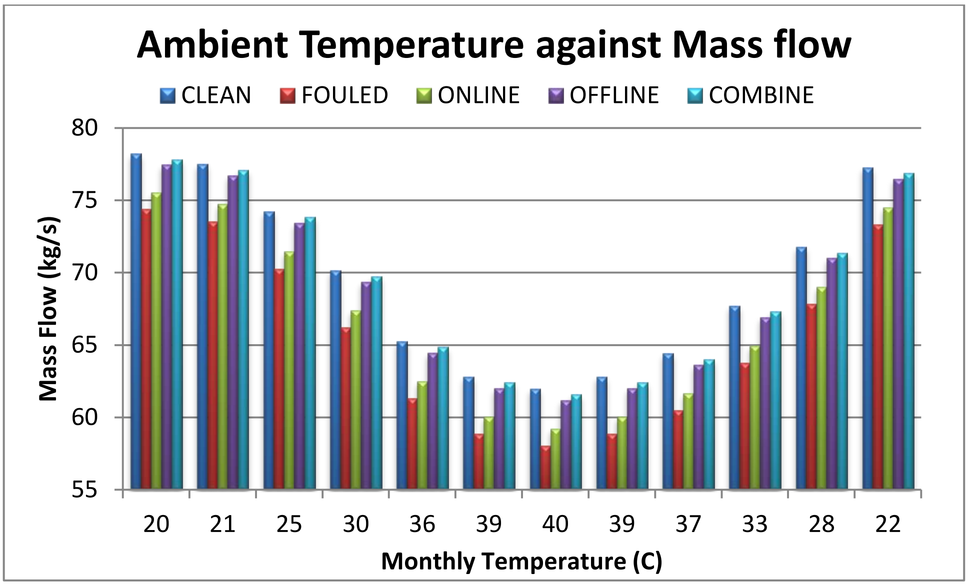


Figure 4-4: Effects of Ambient Temperature on GT Mass Flow

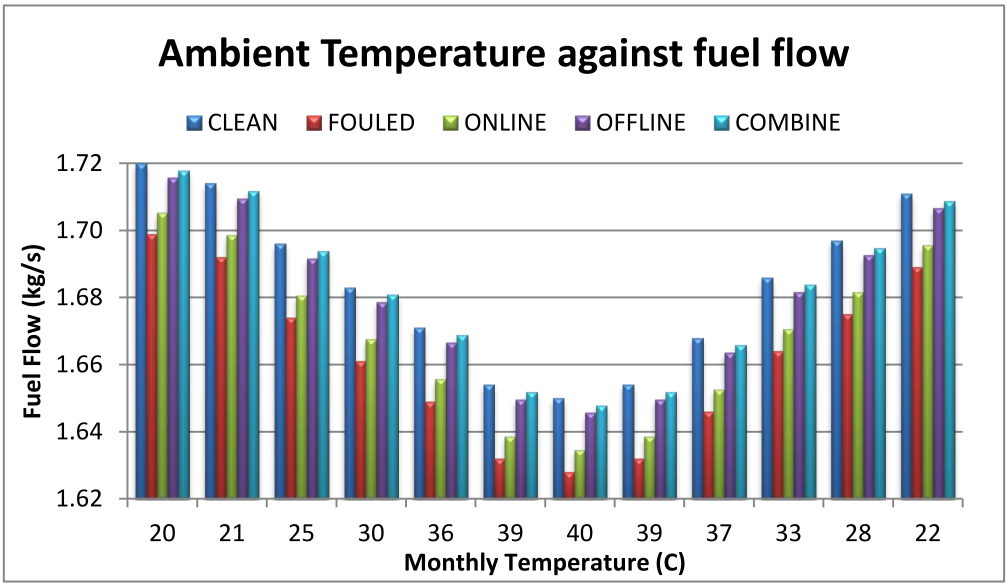


Figure 4-5: Effects of Ambient Temperature on GT Fuel Flow

Similarly, figure 4-5 represents the behaviour of the engine conditions with mass flow against ambient temperature. On average, the fouled condition deviated by about 5.67%, and the online compressor water wash was able to recover about 1.77%, while the offline and combination of offline and online yield about 4.58% and 5.13% respectively. The mass flow is less dense with hot air, as such reduces power output of the gas turbine. It is observed that the percentage deviation is high with mass flow. This is because gas turbines are air breathing engines and as such the quality of air intake matters a whole lot to the output performance of the engine health. The cleaner and colder the air, the better the output power and vice versa.

In the same way, the fuel flow is also affected by ambient conditions, because high temperature reduces fuel flow, while lower temperatures increases fuel flow and keep the engine more efficient. Figure 4-5 shows the engine behaviour for the various cases. The fouled being the highest deviation with a reduction of about 1.30%, with the online compressor wash yielding about 0.39%, while the offline and combined offline & online wash yielded a recovery of about 1.04% and 1.17% respectively leaving the remaining unrecoverable.

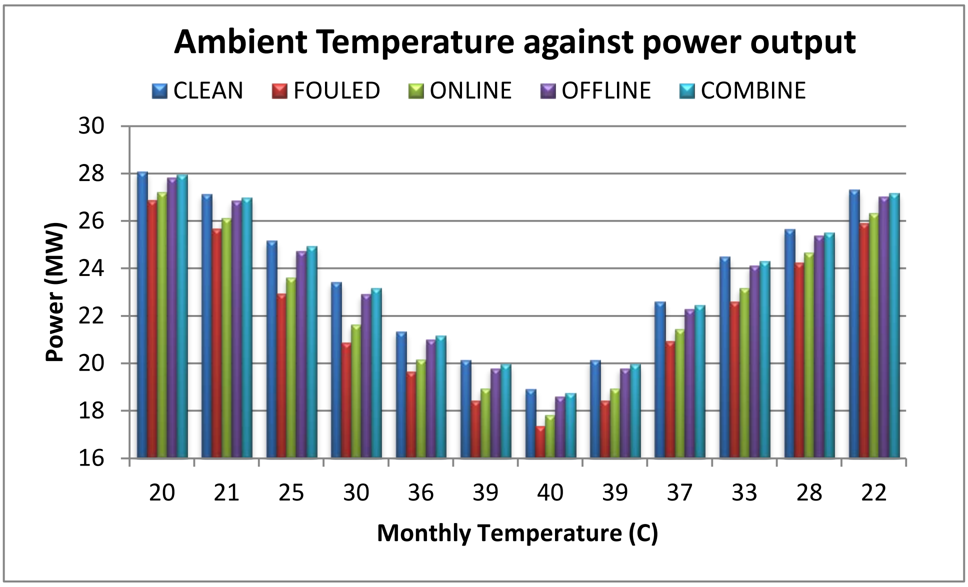


Figure 4-6: Effects of Ambient Temperature on GT Shaft Power

It can also be observed from figure 4-6 the power output has the highest deviation of about 7.20% from the fouled case, and about 2.28% was recovered from the online compressor water wash. While the offline and combination of offline & online was 5.85% and 6.50% respectively. The lost power of a gas turbine engine could be recovered in various ways which include; increase in fuel flow & TET, water injection, supercharging, inlet fogging etc.

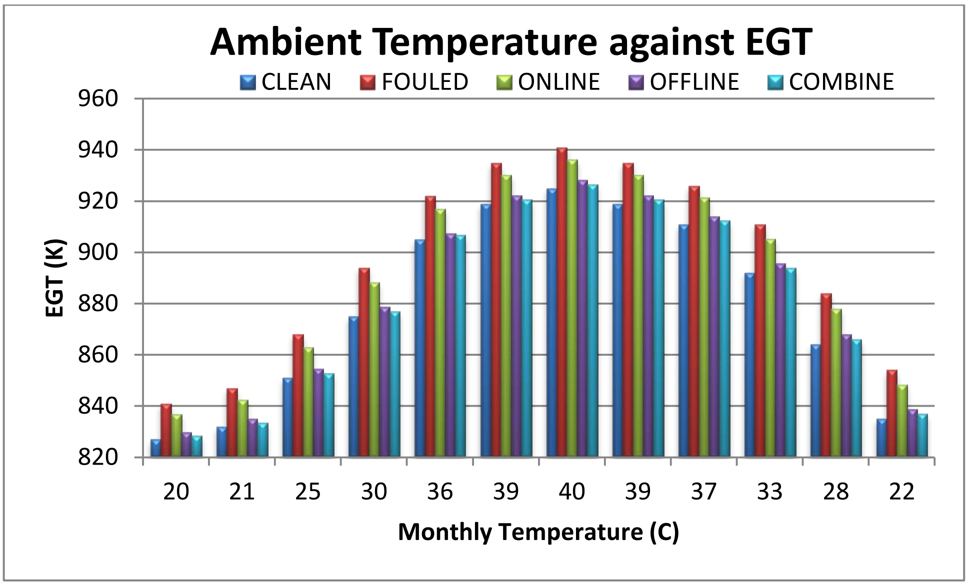


Figure 4-7: Effects of Ambient Temperature on GT EGT

Figure 4-7 is unique compared to the others, because the fouled engine has the highest value while the clean engine has the lowest value. This indicates that exhaust gas temperature (EGT) increases with increase in ambient temperature and decreases with decrease in ambient temperature. Higher EGT relates to lower fuel efficiency which in turn reduces power output from the gas turbine. The clean engine has better EGT compared to the fouled engine as the fouled engine has lower fuel efficiency as could be seen here, increase in ambient temperature by 6.8% increases EGT by 1.92%, while the online compressor wash was able to reduce the temperature by 0.57% after the wash at 30% effectiveness. The offline and combination of offline & online compressor water wash was able to recover 1.54% and 1.73% respectively.

4.4 Effects of Compressor Fouling on GT Performance

Compressor fouling is the most common cause of GT performance degradation. A very small sand particles which passes through the inlet filters and mixes with oil deposits, contributes significantly to performance degradation of the engine caused by compressor fouling in dusty environment. The increasing blockage of the gas path by fouling reduces the compressor efficiency and air-flow capacity. The effect of fouling on GT efficiency reduction is as shown in figure 4-7.

The continuous accumulation of deposits over a period of one year accompany by schedule online compressor water wash, necessitates a continuous increase in firing temperature to maintain constant output power. This will in turn lead to increase in EGT, higher fuel consumption as shown in figure 4-8, and as such seriously reduce hot section turbine blade life due to the increasing firing temperature.

More so, when the fouling is allowed to build up in the compressor for a prolonged time, there will be changes in the natural frequency of the blades. These combine effect can increase vibration of the blade, reduce surge margin in the compressor and subject the compressor to more failure modes. The effect of degradation in GT power output and thermal efficiency is as shown in figure 4-8. It can be seen that the compressor degradation increase from 1% to 5%, reduces thermal efficiency and power output. The reduction in power output and thermal efficiency is because increase in degradation leads to an increase in rate of reduction in pressure ratio. In this situation, the GT might require an increase in TET to augment its required power output.

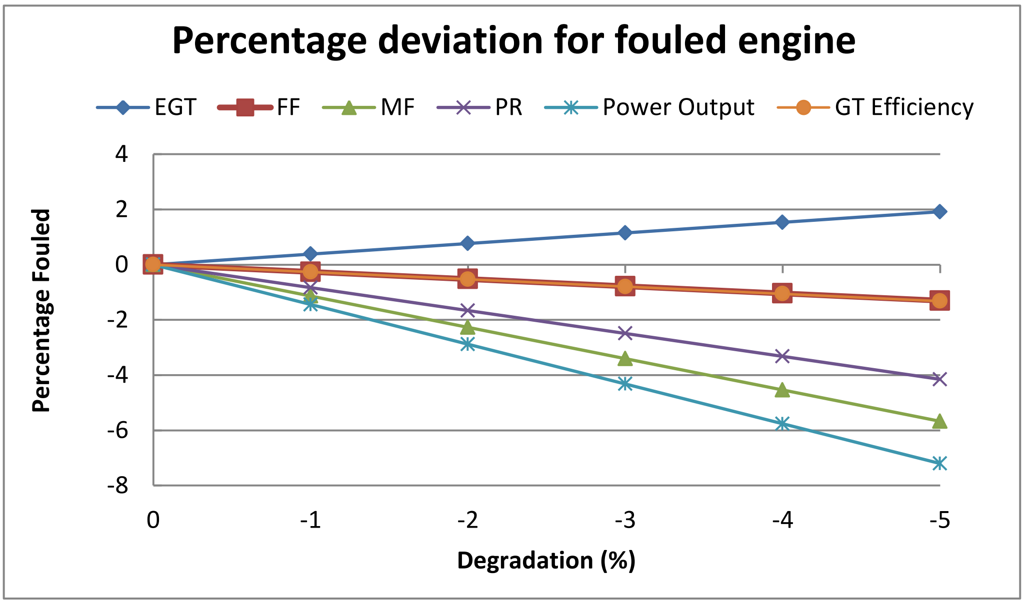


Figure 4-8: Effect of compressor fouling on engine parameters

Table 4-2: Implanted degradation

Steps	Degradation
1	$\Delta\eta$=0.5%, $\Delta\Gamma$=1.0%
2	$\Delta\eta$=1.0%, $\Delta\Gamma$=2.0%
3	$\Delta\eta$=1.5%, $\Delta\Gamma$=3.0%
4	$\Delta\eta$=2.0%, $\Delta\Gamma$=4.0%
5	$\Delta\eta$=2.5%, $\Delta\Gamma$=5.0%

Investigating the effects of compressor degradation on engine performance, the degradation is represented by a decrease in flow capacity and compressor efficiency in the ratio 2:1. The degradation steps are as presented in table 4-2. The effect of compressor fouling on engine performance due to this degradation is as shown in figure 4-8. The plots described the nature of the implanted degradation and its effect on the various engine parameters. It can be seen that the parameters deviated in varying percentage for the same percentage fault implantation. This shows that the degradation affects the engine parameters differently and in other words, some parameters are more sensitive to degradation than the other. On applying the different compressor wash cases employed in this research work, there happens to be significant change in performance recovery when compared with the fouled case as presented in figure 4-7. It was observed that the combine online & offline wash cases yielded better performance results, but how economically viable these will be on the entire operation will be seen later in the economic optimisation in this research. Other cases can be found in Appendix B.

4.5 Effect of compressor washing on engine performance

Though compressor fouling does not cause total engine breakdown, it reduces the compressor efficiency and power. As such, it has become imperative to keep the GT operating clean by regular washing of the compressor to enable maximum production and availability. Too frequent washing will result in increased downtime and cost which will adversely affect production as a result of engine unavailability. Similarly, irregular compressor washing will also emanate to loss in profit due to poor production resulting from decrease in GT performance caused by fouling. Therefore, it is important to optimize the frequency of wash so as to minimize the loss in profit or revenue resulting from fouling [156]. The two widely known methods of cleaning the GT compressor to recover performance are online and offline [143,157].

The offline washing, sometimes called the soak or crank wash is carried out when the engine is shut down. Wash fluids are injected onto the compressor blades and allowed to soak for a while, at this period the contaminants are dissolved, after which the compressor can be rinsed, often with demineralized water. The downtime has direct revenue effect on the engine washing, and the cost of the wash needs to be added to this lost revenue to determine the total cost resulting from compressor washing. On the other hand, the online compressor washing is carried out while the engine is still in operation. This takes place at regular intervals, and the solution is being injected directly into the engine through the intake without engine shut down, thereby preventing downtime cost [158]. Online wash frequencies would tend to be high. When the engine performance loss due to residual fouling is greater than the performance improvement due to an offline wash, then this will be an indication as to when an offline wash will be beneficial [159]. There are several factors influencing the frequency of compressor wash such as: price of product (unit price of electricity), fuel cost, emission tax, production, downtime, and washing cost which include cost of disposal and wash material such as detergents.

The IGT investigated for this study is a 29MW single shaft engine with a free power turbine. It is assumed to operate for a period of 8760 continuous operating hours with no washing implemented over the year, and the output trend is as shown in the figure 4-8. The number of hours is assumed to enable a complete cycle for each wash interval. It is interesting to see that comparison of wash frequencies of 3, 7, 14 and 21 days for the same wash effectiveness shows that the trends are very close to each other. This implies that if the wash effectiveness is constant for each wash for different intervals, the total power recovered for the various intervals is very close. Finding the integral area under the curve with use of trapezoidal rule for each trend is the applicable approach towards investigating whether the difference is significant.

The engine was initially assumed to be operated at full load 100% power output. The fouled engine trend is as shown in figure 4-9 after 8760 operating hours without washing, leaving the engine with a power output indicated by the red line. This means it will be very disastrous to operate a GT over the year without implementing compressor washing. The difference between the area under the curves of trends of the gas turbine engine running at perfectly clean condition with no losses for 8760 operating hours and fouled trend is known as the energy lost resulting from compressor fouling which amount into loss in revenue, degradation of the engine components, thereby leading to unfavourable GT performance characteristics.

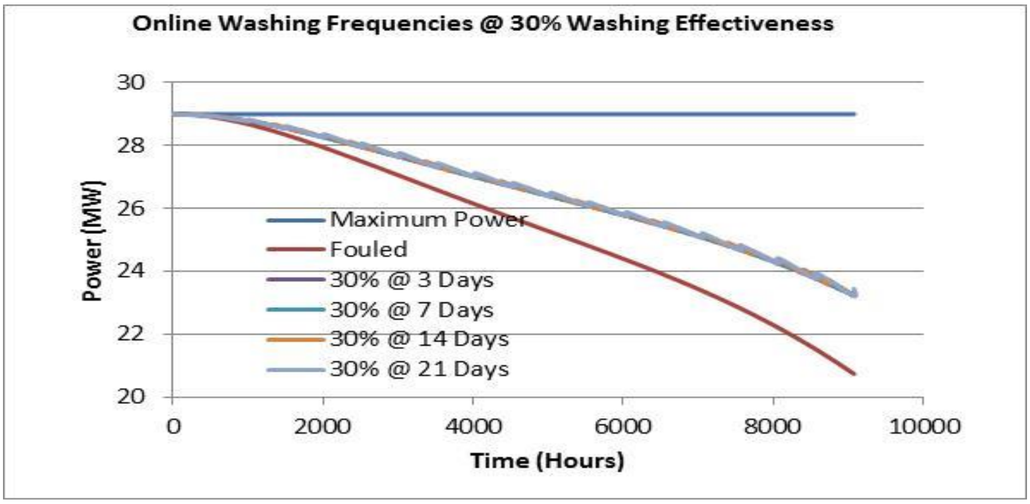


Figure 4-9: Online compressor washing at varying intervals

It is investigated that the 3day wash interval recovered 12.34% power loss, while the 7, 14 and 21 day interval recovered 10.60%, 10.22% and 10.09%. This implies that the 3 day wash interval is more effective as it recovers more power from engine fouling than every other frequencies applied herein. Other cases can be found in Appendix C.

The offline compressor water wash was carried out four times in a year with equal intervals, which makes it ones in every three months. Figure 4-10 shows the behaviour of the trend for offline washing, and the energy loss is the integration of the zigzag curve of the areas from 29 to 26MW, which can be determined as shown in the equation below;

$$Eneregy\ Loss = \int_{fouled}^{clean}(1st + 2nd + 3rd + 4th\ interval) * dt \qquad 4-1$$

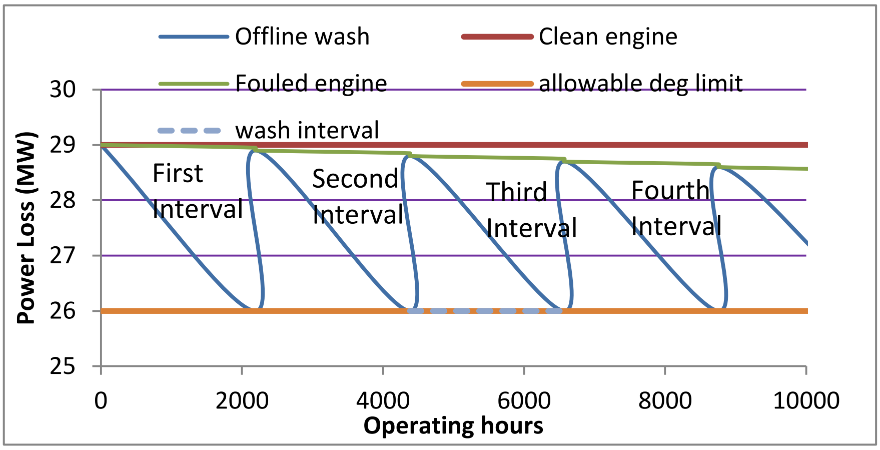


Figure 4-10: Effects of offline compressor washing on power output

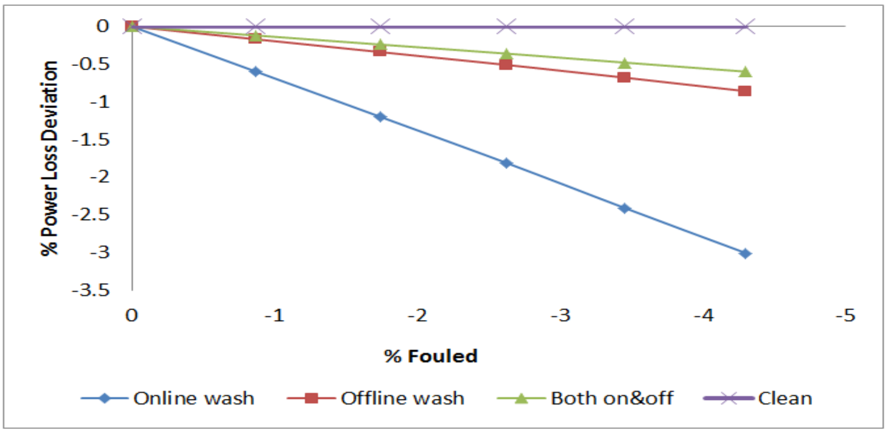


Figure 4-11: Percentage wash deviation on power output

Details will be given in the economic analysis section where the wash cases will be incorporated. Figure 4-11 is a percentage deviation of power loss due to fouling, and it rate of power recovery from the fouled stage. The four zigzag curves in figure 4-10 indicate the number of offline washes for the entire year. It is obvious that the offline wash recovered more power than the online wash at each wash interval, but the economic viability for the entire cost accrue all through the entire year will be demonstrated in the financial terms later in this research work. The gap between the green and red line is an indication of the non-recoverable loss from the power due to fouling, and the maximum limit allowable for degradation before offline washing can be administered as indicated at 10 points in figure 4-11.

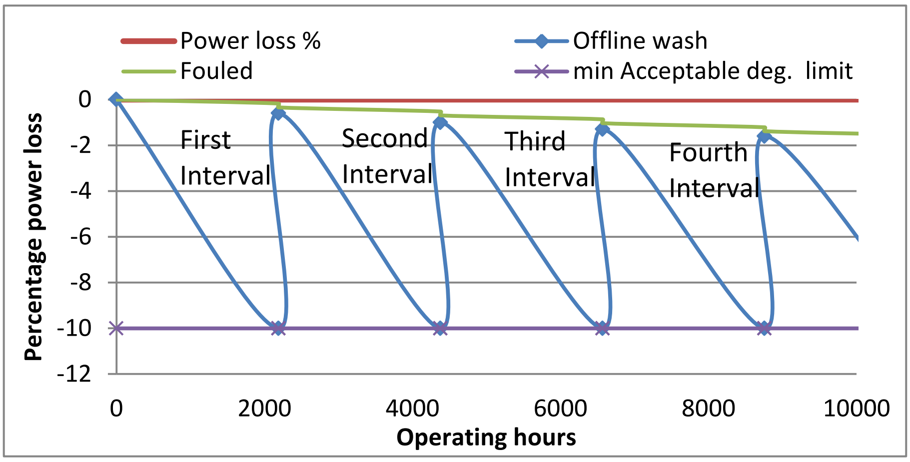


Figure 4-12: Percentage power loss from engine degradation

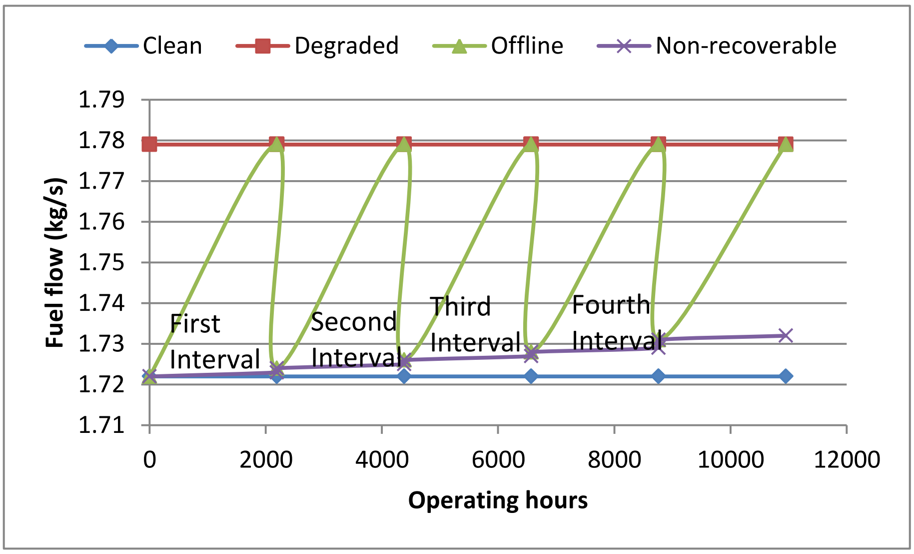


Figure 4-13: Effect of offline compressor washing on fuel flow

In a similar way, it is observed that the fuel flow for the offline wash is as shown in figure 4-12. The curve is similar to that of power loss, and the maximum acceptable degradation limit is as indicated at point 1.78. The non-recoverable degradation limit is as indicated by the blue and purple curves in figure 4-13. It can be seen that the compressor wash made a reasonable recovery rate deviated from the fouled case as could be seen in figure 4-14 below.

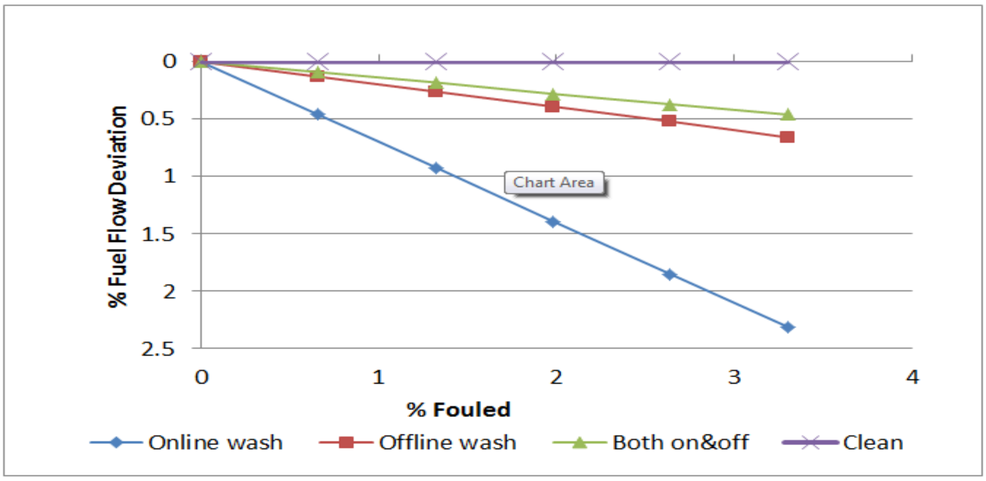


Figure 4-14: Percentage wash deviation on fuel flow

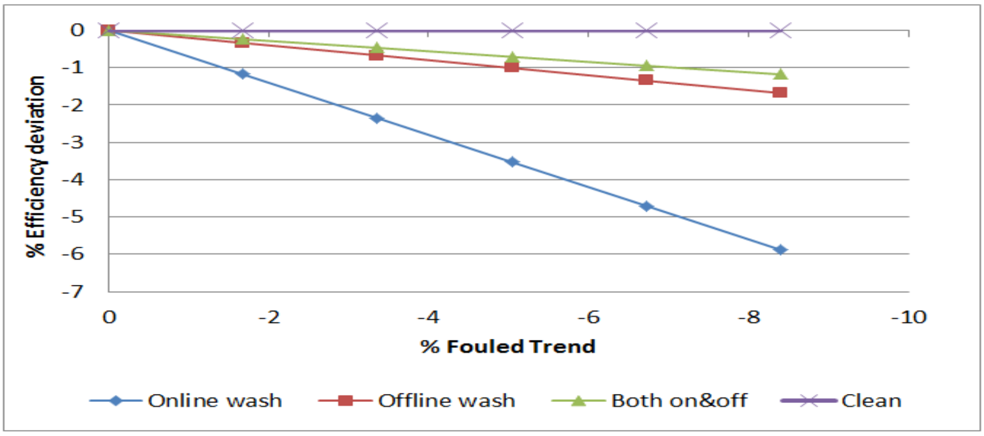


Figure 4-15: Percentage wash deviation on compressor efficiency

The effect of online washing is shown in figure 4-8, and percentage variation for both power loss and fuel flow is as shown in figure 4-10 & 4-13 respectively. While a wider view of the offline wash percentage deviation is as shown in figure 4-11 & 4-12 with intervals to enable calculation of area under the curve at each wash interval. Figure 4-14 is in a similar form but indicating the efficiency and its reaction to fouling. Figure 4-15 is an indication of the deviation in rate of recovery for the efficiency deviation of the GT.

4.6 Rate of engine performance recovery

It has been investigated earlier in this research work that fouling increases faster at the beginning of service life, and later, the rate of debris accumulation reduced to a more or less constant rate. Some considerations made in minimising or protecting fouling include; a frequent maintenance strategy, frequent compressor water wash, inlet air filtration system is set at its best point, thorough treatment of fuel to reduce hot section fouling, adhering to manufacturers instruction and recommendation regarding operations and maintenance procedures.

The compressor water wash cases in this research as stated earlier include; online wash only for 7 days intervals. Offline wash taking place at equal intervals and four times per annum, making it at intervals of three months at each offline compressor water wash. A combination of both online & offline wash, in this case the online is carried out at the regular intervals while the offline is done two times a year at equal intervals. Other cases can be located in Appendix D.

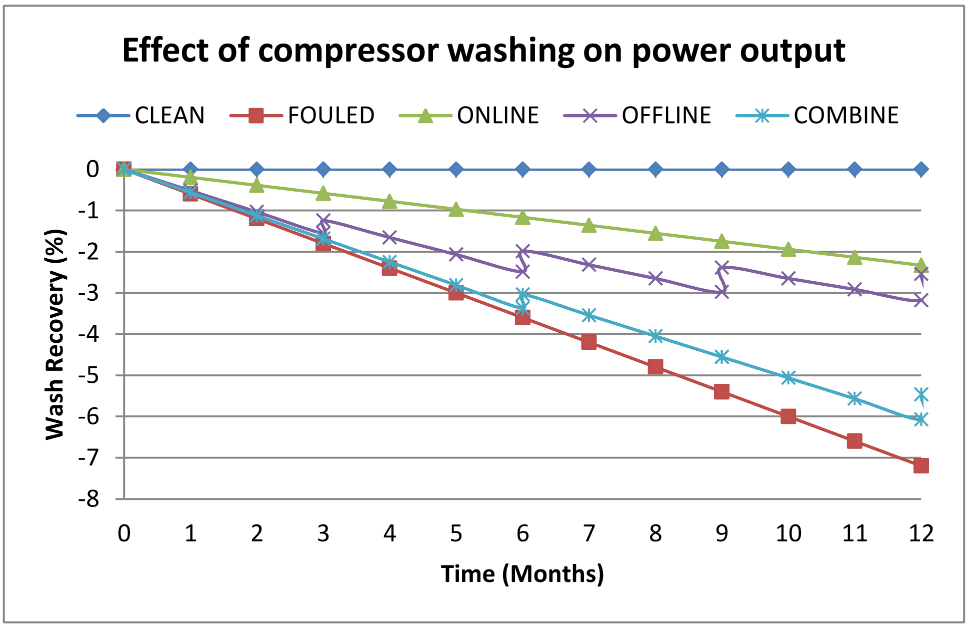


Figure 4-16: Effect of compressor washing on power recovery

Figure 4-15 and 4-16 represents the cumulative percentage deviation of all the wash case for power and fuel flow respectively. It can be observed that the combinations of both offline and online compressor water wash deviated more, but it economic viability will be shown in financial terms later in this research when all cost associated with the process at each wash case most have been collated.

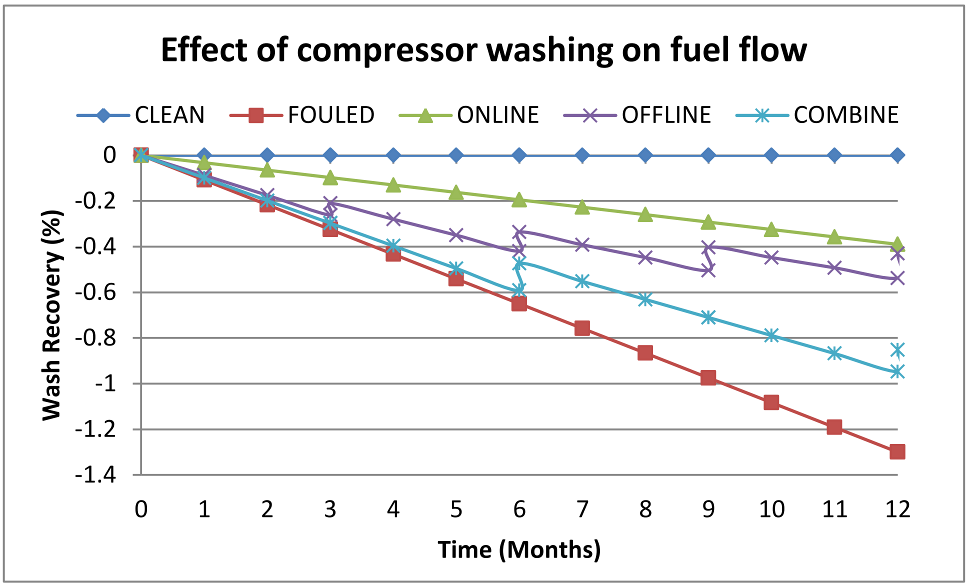


Figure 4-17: Effects of compressor washing on fuel flow

4.7 Summary

It can be seen from figures 4-2 to 4-7 how the various engine parameters are been affected by ambient temperature and to what magnitude. The efficiency as could be seen in figure 4-2 shows the effect of ambient increase on thermal efficiency reduction. This is as a result of change in the inlet air density and compressor work, because hotter air reduces inlet air density, and it is this air density that drives the compressor work which determines the power output. This is why the thermal efficiencies for the three engine cases decrease as the ambient temperature increases; as such air mass flow rate will increase if fuel/air ratio is constant throughout the year. In such case the power increase will be less than inlet mass flow rate at the compressor, which will in turn increase the EGT as the ambient temperature increases. As could be seen in figure 4-7, increase in EGT is as a result of incurred losses from high quantity of flue gasses.

An industrial gas turbine for power generation is operated at constant speed; hence increase in ambient temperature will lead to decrease in mass flow which can be seen in figure 4-4. When TET is kept constant as the fuel flow is being varied, it will result in increase in inlet temperature, which reduces temperature ratio (TET/T_1) which will further lead to reduction in compressor pressure ratio as a result of decrease in specific work of the engine, and then reduces the shaft power output as could be seen in figure 4-6.

GT engines are greatly influenced by the environmental conditions like; the quality of air they ingest. High ambient temperatures and compressor fouling affects the engine performance greatly. This high temperature increase has significant effect on the HPT blade creep life, as such there is need to determine the creep life of the HPT blade as could be seen in the next chapter of this research.

5. CREEP LIFE ESTIMATION MODEL

5.1 Introduction

Gas turbine blades operating above a certain temperature under centrifugal stress will deform with time. This phenomenon is commonly known as creep and it is measured by the rate of strain per hour for a particular stress and temperature [104]. Creep strain increases with increase in stress and temperature, and creep usually occurs at temperature above the absolute melting temperature of turbine materials [160, 161]. The deformation of creep will eventually lead to the turbine blade material being fractured.

It has earlier been stated that high TET is very important for GT performance enhancement. As such, modern GT operates at very high temperature making creep a serious problem. Special nickel-based alloys have been developed to resist creep. GT components undergo different types of degradation resulting from high temperature and mechanical loading, even with the advanced cooling technology the performance degradation results to an increase in TET and the metal temperature as well, since they are directly proportional to each other. Industrial gas turbine can operate continuously with blade metal temperature at about 1000K in the recent day. Creep is still a major factor that limits the allowable TET.

The well-known parameter that has found widespread use in determining creep life is the Larson-Miller parameter. This parameter combines temperature and creep life data and is a useful analytical method for assessing the effects of stress on creep life over a range of temperatures. The parameter is presented as;

$$LMP = 1.8T\big(20 + In(t)\big) \qquad 5-1$$

Where T is the metal temperature in K and t is the creep life in hours.

GT Performance relies greatly on the gas temperature entering the turbine. In the absence of turbine blade cooling, the gas temperature and the turbine blade temperature will be the same. Significant increase in gas temperature can be attained by cooling the turbine blade so as to reduce the blade metal temperature at an acceptable value, thereby achieving the necessary creep life. The benefits in increased engine performance due to the high gas temperature is still substantial, even after accounting for any additional losses in the turbine due to the effects of employing cooling methods [156].

Liquid or air can be used as cooling medium for turbine blade. Air is mostly used as cooling medium nowadays since the liquid cooling using water have proved unreliable [156]. Steam and mist (wet steam) cooling have been investigated and are currently used in GT for combine cycle plants. It is normally applied by bleeding air from the compressor discharge and channelled it into the turbine nozzle and the rotor internal passages. The bleeding of air for cooling purposes has an impact on engine performance, and the cooling air is usually re-introduced into the gas stream after carrying out the cooling function, to reduce the loss resulting from these bleeds.

The four methods of blade cooling include; convection, impingement, film, and transpiration cooling processes. The blade cooling technology can be applied to reduce the turbine blade metal temperature (BMT) below that of the gas stream temperature. For air-cooled turbines, the BMT can be calculated by using the cooling effectiveness parameter as shown in equation 3-2. The value of ε will vary depending on the cooling technology applied and this value could range between 0.4, 0.5 and 0.6 for film cooling, or higher for advanced cooling concept [156].

The increase in mass flow rate through the engine as discussed earlier will means the gas generator turbine power output is increasing and it is necessary to satisfy the increased power demand from the compressor as the ambient temperature decreases. This implies that the torque on the turbine rotor blade will also increase. The effects of the increased speed and torque will also increase the stress on the rotor blades, thereby having an adverse effect on the gas generator turbine creep life usage and reducing the time between overhauls during constant exhaust gas temperature operation. However, the cooling air temperature increases with increase in ambient temperature, thereby increasing the BMT.

The creep model layout has been described in chapter 3 of this research work. The major objective of this chapter is to estimate the turbine blade creep life for the service period along the various sections of the blade. The blade is divided into four equal sections, having five points from the RDS to the TDS at 25% interval, and afterwards the average temperatures were taken as output for the four equal blade sections. The stresses at this point were also determined, after which the required creep life were estimated. The required equations for determining the BMT and sectional stress of the blade, and the time to failure are as shown in equation 3-3, 3-6, and 3-8 respectively. The schematic layout for the creep life assessment model is as shown in figure 3-3 in chapter three. The LMP and stress values are sets of LMP creep generated from standard test where the material is tested under specified temperature and loadings.

The creep rate $d\varepsilon/dr$ is the slope of the curve, and the stages are as shown in figure 2-11.

5.2 Blade Cooling

The bleed air from the compressor helps in cooling the gas turbine internally and externally. The coolant is passed through cooling channels internally in the blade, and coolant air is ejected through slots meant to provide a film of air to help protect the external surface of the blade from hot gasses [162]. Being a preliminary estimation model in this work case, the turbine blade is assumed to be divided into sections, having a uniform blade temperature equal to the average temperature of each blade section.

In addition, the coolant and the blade temperature are expected to increase along the span of the blade [163].

- **Cooling effectiveness**

It is defined as the measure of the ability to cool the turbine blade. It will vary according to the cooling system in use, either convective or film cooling. The parameters of the combustors that are used in quantifying the gas temperature at entry to the first stage turbine are given by:

- **Convective Cooling Effectiveness** [164].

$$\varepsilon_c = \frac{T_{f-T_b}}{T_{f-T_{C1}}} \qquad 5-2$$

T_f =film average relative gas total temperature (K)

T_b = maximum allowable blade metal temperature (K)

T_{c1} = Initial coolant temperature (K)

- **Film Cooling Effectiveness**

$$\varepsilon_f = \frac{T_g - T_f}{T_g - T_{C2}} \qquad 5-3$$

T_g = free stream average relative gas total temperature (K)

T_f = Local film relative gas total temperature (K)

T_{c2}= Coolant exit temperature (K)

- **Coolant mass flow function**

$$M^* = \frac{m_{cb}\, c_p}{h_f\, s_g l} = \frac{T_{f-T_b}}{T_{C2-}\, T_{C1}} \qquad 5-4$$

m_{cb}= Coolant mass flow, the proportion of the total mass flow through the turbine which is diverted for blade cooling (kg/s)

C_p = Specific heat capacity at constant pressure (J/kgK)

h_f = Heat flux due to film temperature

s_g = Perimeter (m)

L = Length (m)

- **Overall cooling effectiveness**

$$\varepsilon = \frac{T_{g-}\, T_b}{T_{g-}\, T_{C1}} \qquad 5-5$$

With TET of 1505K, considering that the blade is convective and film cooled, the cooling effectiveness should be between 0.4 and 0.6 [165]. Consequently, the blade was assessed in the thermal model when the cooling effectiveness is 0.4.

5.3 Radial Temperature Distribution Factor

There are certain factors to be considered for accurate prediction of gas turbine components creep life. These include the complex geometry of most components, multi-dimensional stresses and strains arising from loads imposed degradation and non-uniform temperature distribution at the combustor exit. Such problems are usually avoided by assuming a uniform axial stress, time and temperature. The temperature distribution will be considered with a view to getting a more accurate result, by employing the radial temperature distribution factor RTDF in this research.

Turbine rotor blade life is determined with RTDF using circumferentially measured values. The turbine blades experience a circumferential average temperature in a given radial plane due to the rotation undergone. It was stipulated that the RTDF should be controlled to less than 20% by [166].

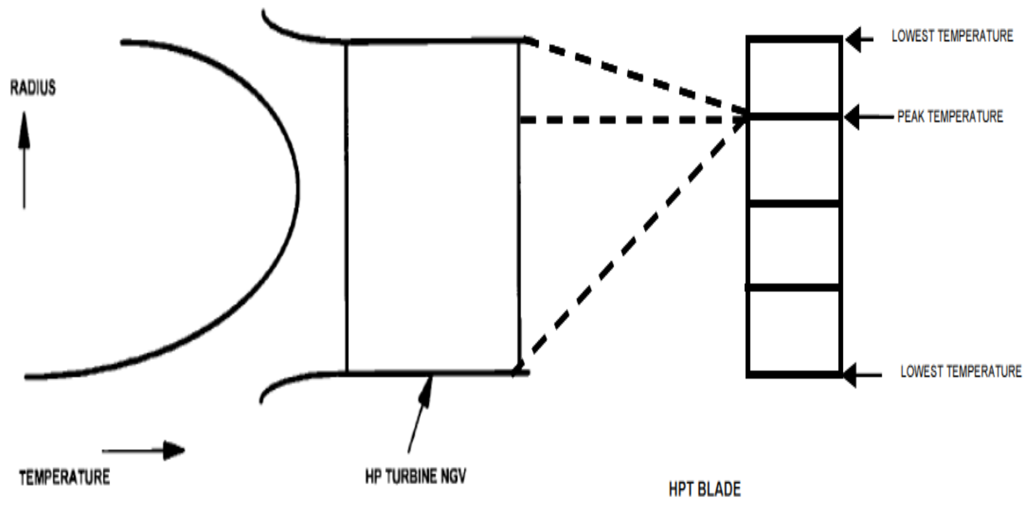


Figure 5-1: Combustor exit temperature profile- RTDF

RTDF is circumferentially mean outlet peak temperature minus mean temperature divided by mean combustor temperature rise [167].

$$RTDF = T_{max} - \left(\frac{TET}{TET - T_{in}}\right) \qquad 5-6$$

The maximum temperature location depends on the reason that the turbine blade rotation causes the peak gas temperature to shift from the middle of the span towards the tip as could be seen in figure 5-2 [162].

The RTDF is used to calculate the temperature variation at each section of the blade. As postulated by [7]. The maximum and minimum temperatures are calculated as follows:

$$T_{max} = T_{ri} + (\Delta T_{burner} * RTDF) \qquad 5-7$$

$$T_{min} = \frac{(5T_{ri} - 2T_{max})}{3} \qquad 5-8$$

T_{ri}: Rotor inlet relative gas temperature (K)

ΔT_{burner}: Temperature rise at the burner (K)

The derivation of these formulae is with the assumption that the maximum temperature will occur around 75% of the blade height (this is the gas temperature at which the blade section was predicted). The author adopted the following assumptions [7].

- There is linear rise in gas temperature from root to 75% blade height.
- The gas temperature is minimum at the root and tip of the blade.
- There is linear reduction in gas temperature from maximum to blade tip.
- The maximum temperature, T_{max}, occurs at the 75% height from the blade root. The assumption is based on the explanation given by [162] which states that due to the rotation of the turbine blade, the peak temperature will shift from the mid blade towards the tip region.
- The average temperature of those defined points should equal to the blade inlet temperature, T_{ri}.

These assumptions were considered because of the high operating temperature considered for the operation of the said engine. The highest temperature will tend to move from the middle towards the tip because of the rotation of the turbine blade, and the turbine blade tips are often the most susceptible to material failure due to high-speed leakage flow and associated large thermal loadings.

5.3.1 Effects of RTDF on Blade Creep Life

The inlet gas temperature of the blade is varied so as to enable variation in the metal temperature. This could be achieved by defining the RTDF, which could be defined as the ratio of the difference between the circumferential peak temperature, T_{max} and the gas mean temperature, T_{mean} to the combustor temperature rise, $T_{\Delta burner}$ as could be seen in equation 5-6. The blade inlet temperature T_{cin} for this model is taken as T_{mean}. T_{max} can be obtained as $T_{\Delta burner}$ from pythia. RTDF identifies the profile of the temperature distribution on the turbine blade. A distribution profile is produced from higher RTDF values, as could be seen in figure 5-3, having a substantial difference between the maximum and minimum gas temperature along the radius of the blade.

Contrastingly, a lower RTDF value will produce a uniform distribution with small temperature variation along the blade span. The metal temperature at the designated sections of the blade changes as the profile of the gas temperature changes, thereby creating a turbine metal temperature profile. The RTDF values is less than or equal to 0.2 as postulated by [168], and the user was able to vary the RTDF values in the model. Along the blade span, five gas temperature points were created and it characterised the profile of the gas temperature as $T_{gtip,}$ T_{gmid} and T_{groot} for the gas temperature at the blade tip, mid and root respectively; and $T_{g75\%}$ and $T_{g25\%}$ for the gas temperature at 75% and 25% distance from the root of the blade respectively. Assumptions made are as stated earlier in section 5.3.

In evaluating the blade creep life, a uniform value of the cooling effectiveness of 0.4 is used along the various sections of the blade for the various RTDF. This is because the highest value of temperature does not necessarily mean the value of stress will be highest at that point and also because creep is a function of temperature and stress. The projected life in hour is converted to the actual life by applying a factor of safety of 60%.

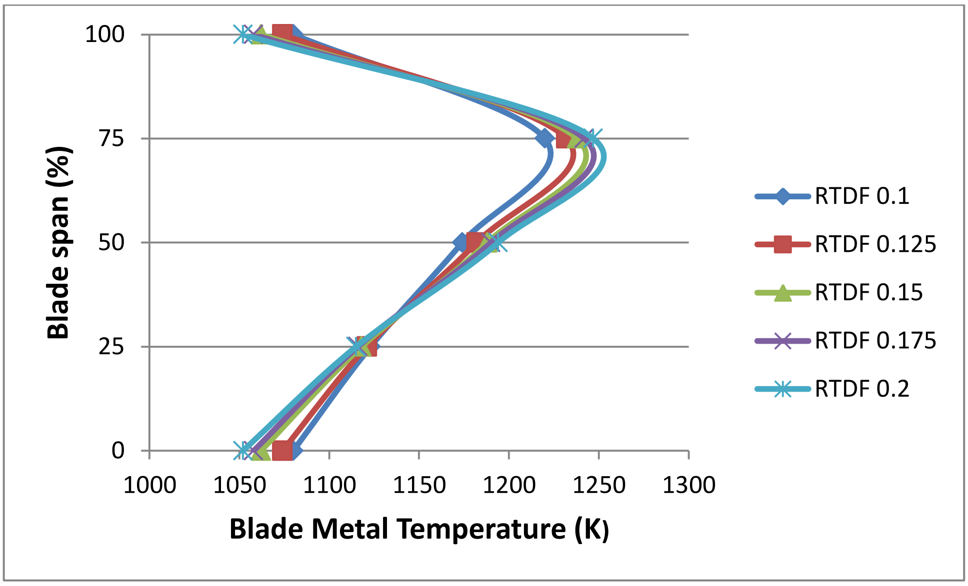


Figure 5-2: Blade span at varying metal temperatures

Figure 5-2 shows the effect of RTDF at different sections of the blade. Increasing the RTDF from 0.1 to 0.2 at the 75% point which is the point with the highest temperature shows an increase in BMT by about 2.41%, but the case is not the same for the root and the tip, as the RTDF increase in the same range, the BMT reduces by about 2.86%. It means that the blade temperatures increase at the root and tip are higher with lower RTDF when compared with other sections of the blade. This implies that it will be advisable to use higher values of RTDF to enable reduction in temperatures at the root and tip of the blade. The increased temperature on the turbine blade could cause tip clearance which can affect it efficiency. It is obvious from figure 5-2 that increase in BMT with about 2.78% decreases blade creep life by about 80.4%. This is an indication that creep life is dependent on BMT. Considering that the parent engine from which most of these data were obtained was designed originally as an aircraft engine, it shows that the figures are realistic [104].

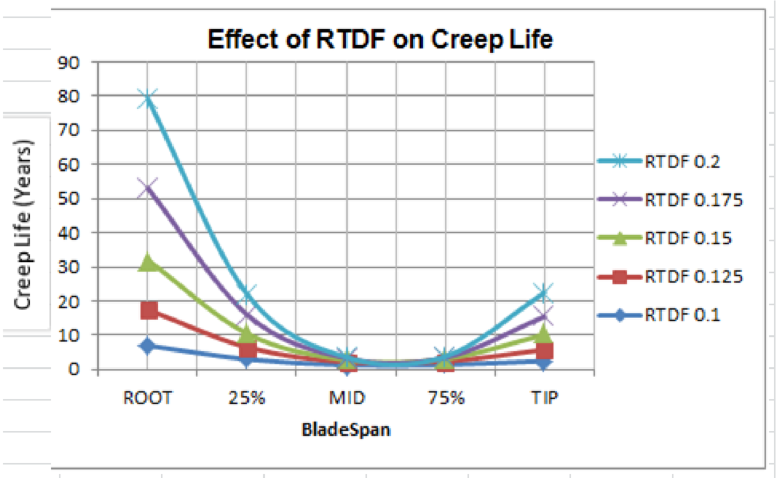


Figure 5-3: Effect of RTDF on creep life along blade sections

It could be noticed from figure 5-3 that there is an increase in the blade minimum life as a result of increase in RTDF at the root and 25%. Increase in RTDF from 0.1 to 0.175 at the root along the range increases the blade life by about 66.7%, and it is about 50% increase at the 25% section. These was not the case for the blade mid and 75% section, as an increase in RTDF here at the same range decreases the life by about 32% and 38% respectively.

Increasing RTDF reduces the BMT that is the reason for the increased metal temperature which in turn reduces the creep life at the mid and 75% section of the blade. In the same case the reduction in BMT as a result of RTDF variation, increases the creep life as could be seen at the root and 25% section of the blade. It shows that a 2.2% decrease in BMT could increase the creep life by about 67%, and 2.2% temperature increase could reduce creep life by about 38%. It is obvious from figure 5-3 that the difference at the root and at the 25% blade sections are much more than that at the mid and 75% section. The 75% section was affected more by the stress effect, thereby changing the minimum creep life.

It can be concluded that the RTDF have significant effect on the blade metal temperature. It has been shown that lower values of RTDF increases the blade metal temperature at the root and tip of the blade, while higher RTDF increases at the mid and 75% section. Hence, it is advisable that proper care be taken when choosing RTDF for such application, as carefulness could increase the creep life of the blade reasonably and in turn yield economic stability for the GT user.

5.4 Creep Factor

It is very important to carry out impact analysis with a reference point. For example, a remnant creep life of 25,000 hours is not well detailed to know how much the engine is being used. Stating that 25,000 hours constitute 50% less than expected would be understood that the engine in question has been operating under severe thermal and mechanical stresses. The 50% explains the enormity of the effect of operating the engine at off design condition. Such information will enable the owner or operator to improvise better means of optimisation or introduce a more preferred means of carrying out maintenance plans to reduce operating and maintenance cost to minimal.

In this research, the creep factor (C_F) was implemented to ascertain the effect of operating conditions on creep life, and measure how fast the creep life is being consumed compared with the reference operating point specified by the operator. The ratio of the calculated creep life remaining at the actual operating conditions and the remnant creep life calculated for the reference conditions is known as C_F [5, 7].

$$C_F = \frac{L_c}{L_{cRef}} \qquad 5-9$$

Here, L_c represents the calculated remnant life for actual operating condition, while L_{cRef} is the reference remnant life at user-defined reference operating condition (design point, baseline and nominal).

A realistic remnant life that is useful to the users will allow them to perform a realistic impact analysis, and the C_F value will allow the user to assess changes in the remaining creep life of components operating at conditions which deviate from the usual user operating conditions. C_F Will also help in eliminating dependency on the original engine maintenance OEM baseline operation, which is not always achievable when the user-defined normal operating conditions are far from the suggested baseline operation [5].

However, when:

- C_F=1, it means the engine is being operated at the reference condition with L_c=L_CRef
- C_F<1, it means the engine is being operated in a worse condition than its reference condition hence reducing the blade's remnant life.
- C_F>1, it means the engine is being operated under better conditions than its reference point, thus increasing the blade's remnant life.

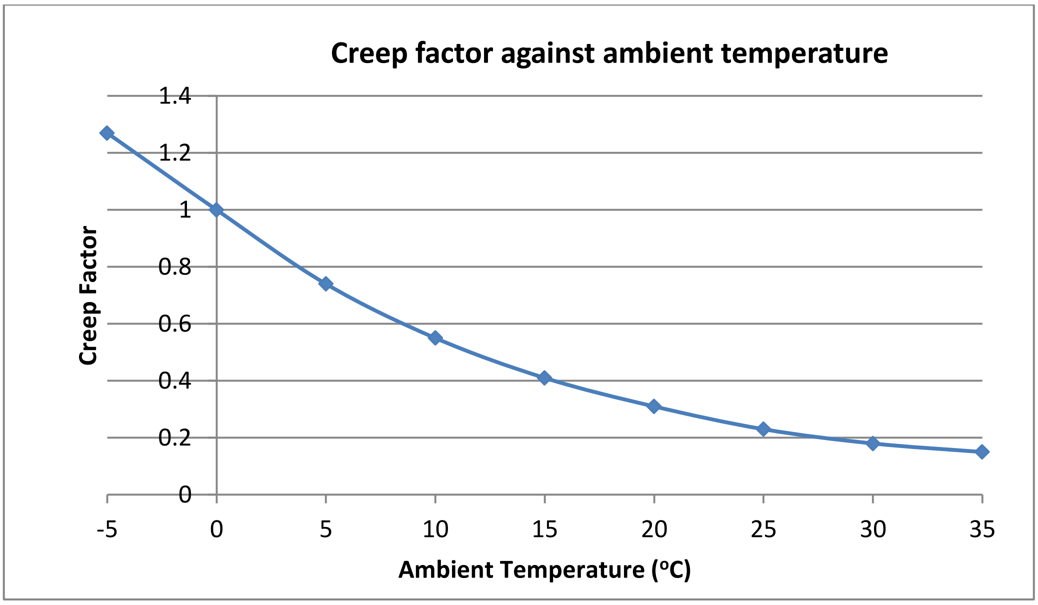


Figure 5-4: Effect of ambient temperature on creep factor

From figure 5-4, a CF of 1.3 indicates that the component life as increased by 30%, while a CF of 0.2 indicates that the blade life reduced by 80% from the reference point. These imply that the higher the temperature on the turbine blade, the faster life will reduce, and vice vasa.

5.5 Properties of Blade Materials/ Geometry

The high pressure turbine (HPT) is the component that most probably requires maintenance as a result of both high rotational speed (centrifugal force) and operating temperatures. The annulus geometry of blade can be found in Appendix E1.

In this research, a nickel-based super alloy known as Rene 80 with excellent creep-rupture strength, good elevated temperature ductility and superior hot corrosion resistance, is the blade material properties considered with a density of 15.9 slug/ft^3=8194.5kg/m^3 [169].

The chemical composition of Rene 80 and its related alloys are as can be seen below [170].

- **Rene 41**: Cr=18.00-20.00%, Ni=remainder, Mo=9.00-10.50%, Co=10.00-12.00%, Al=1.40-1.80%, Ti=3.00-3.30%, Fe=5.00% max.
- **Rene 80**: Ni=60.0%, Cr=14.0%, Co=9.5%, Ti=5.0%, Mo=4.0%, W=4.0%, Al=3.0%, C=0.17%, B=0.015% and Zr=0.03%.
- **Rene 95**: Ni=61.0%, Cr=14.0%, Co=8.0%, Mo=3.5%, W=3.5%, Nb=3.5%, Al=3.5%, Ti=2.54%, Fe=<0.3%, C=0.16%, B=0.01% and Zr=0.05%.

It can be obviously seen that the chemical composition are very close, and as such, the material properties are very close too, as can be seen below extracted from [171] report from NASA [172].

- **Rene 41**

Density= 0.298Ib/cu.in = 8248.6kg/m^3

Specific Heat = 0.11 Btu/Ib/Deg F = 460.5 J/KgK

Melting Point = 2430^{0}F = 1332.22^{0}C = 1605.7K

- **Rene 95**

Density = 0.297Ib/cu.in = 8220.9Kg/m^3

Specific Heat = 0.11 Btu/Ib/Deg F = 460.5 J/KgK

Melting Point = 2450^{0}F = 1343.3^{0}C = 1616.3K

- **Rene 80**

The three nickel-based alloy densities are all within same range. The specific heat capacity and melting point of Rene 80 was not gotten from any of the above listed references though, the figures can safely be put as could found in Appendix E2.

5.6 Effects of Ambient Temperature on TET

The ambient temperature of the operating environment as earlier discussed is 40^0C which is 25^0C away from ISA condition. An increase in ambient temperature of this magnitude would mean an increase in TET from it design point of 1505K to 1605K in other to recover the lose power output resulted from fouling. The TET increase is about 100K, which is a percentage deviation of about 6.2%. The linearity of figure 5-5 shows that ambient temperature is directly proportional to TET. Thus, a decrease in the blade creep life will be experienced as the hot section temperature increases. The degraded TET will now be given as:

$$Degraded\,TET = Deviation + clean\,TET \qquad 5-10$$

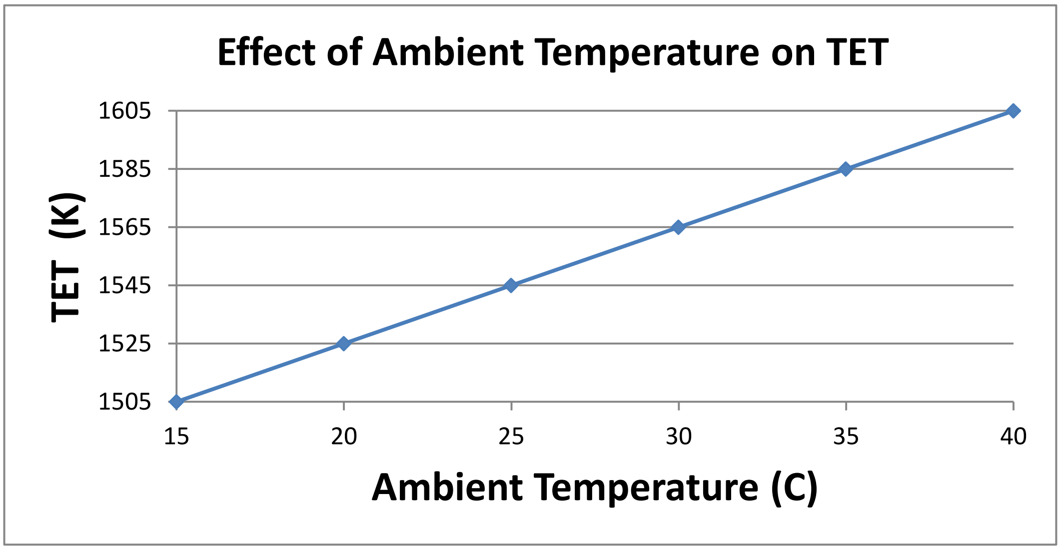


Figure 5-5: Effects of ambient temperature on TET

5.7 Gas Temperature of Blade Metal Sections

The temperature of the gas flow outside the blade influences the BMT and the coolant inlet and outlet temperatures [164].

Factors considered when determining the BMT are as stated below:

- Cooling air is injected into the blades close to their leading and trailing edges which are the areas of the blade exposed to the greatest thermal assaults.
- The HPT blades are cooled through multi-pass convection cooling and leading edge cooling.

The combined convection and film cooling effectiveness is as shown in equation 3-2.

The rotor blade cooling effectiveness, ε, is dependent upon the total gas and coolant temperature relative to those of the blades. The coolant gas temperature, T_{cin} is assumed to be the same as the temperature of the gas leaving the compressor. The HPT blade cooling effectiveness is assumed to remain constant, regardless of the likely degradation experienced by the blade. The function of the gas and coolant temperature and ε is given as shown in equation 3-3.

The hot gas makes its way through the Thermal Barrier Coating (TBC) onto the blade metal. The TBC is substantial in thickness resistance coating. The rotor inlet to interior temperature drop as a function of the TBC thickness is approximately equal to 150K [165]. Therefore, we can say that the gas temperature is equal to the difference between the turbine inlet temperature and the temperature drop due to TBC:

$$T_g = T_{TET} - \Delta T_{tbc} \qquad 5-11$$

Higher temperatures hit the stator, but it experiences lower stresses. While the rotors on the other hand, are subjected to relatively lower temperatures as a result of NGV cooling, but they experience higher stresses due to centrifugal loads.

At the design point of the engine blade, the TET was 1505K which is more than the Rene 80 temperature capability of 1373K (see figure 5-6). In addition, Rene 80 has a melting point of 1330^{0}C (1603K). This calls for the need for blade cooling and TBC application.

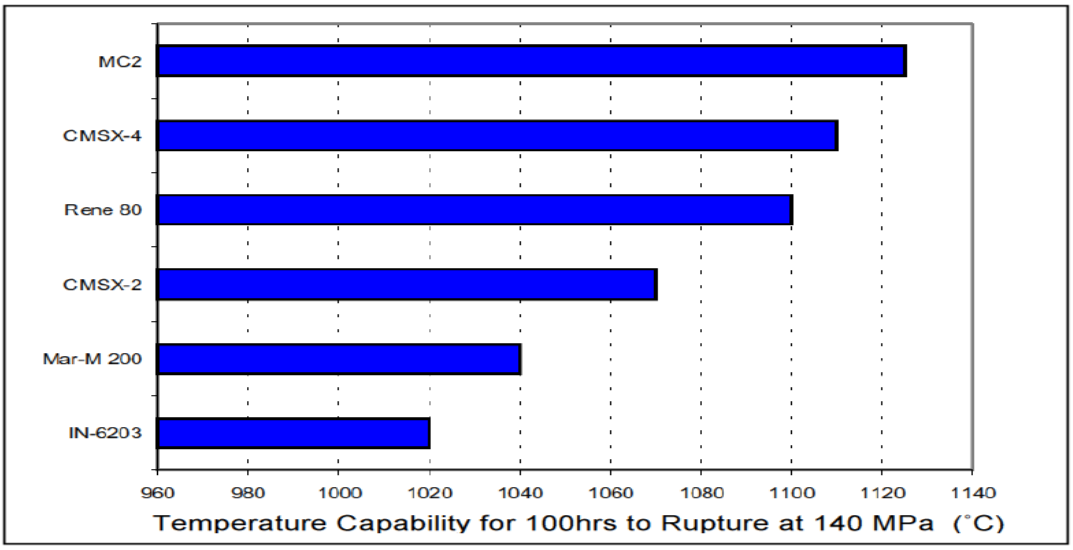


Figure 5-6: Temperature capability of blade materials [173]

- **Thermal Calculations**

Burner Entry Temperature = 762K

Burner Exit Temperature = 1505K

Burner Temperature Rise = 743K

Each blade section is treated as a blade individually. All blade sections are assumed to have the same coolant inlet temperature and cooling effectiveness. But this is not completely true because the temperature entering the first section is not the same as that leaving it. Also, the temperature entering the section differs from that exiting it. More so, the coolant inlet temperature can be assumed to be the same for a preliminary estimate [5].

The burner entry temperature, TET, and burner temperature rise for the clean and fouled engine behaviour was extracted from PYTHIA output results, as stated in section 5.9.1. From equation 5-6 and 5-7, the maximum temperature of the blade at 75% height and the minimum temperature at the tip and root with RTDF of 0.2 are given as follows;

$$T_{max} = T_{ri} + (\Delta T_{burner} * RTDF)$$

$$T_{max} = 1505 + (743 * 0.2) = 1653.6K$$

$$T_{min} = \frac{5T_{ri} - 2T_{max}}{3}$$

$$T_{min} = \frac{5(1505) - 2(1653.6)}{3} = 1406K$$

Assuming that the temperature of the blade changes from RDS to the 75% height increases linearly [6], the temperature at the 25% height and the mid height can be calculated by similar triangles and the result is as presented in figure 5-7 from Appendix E5-E9.

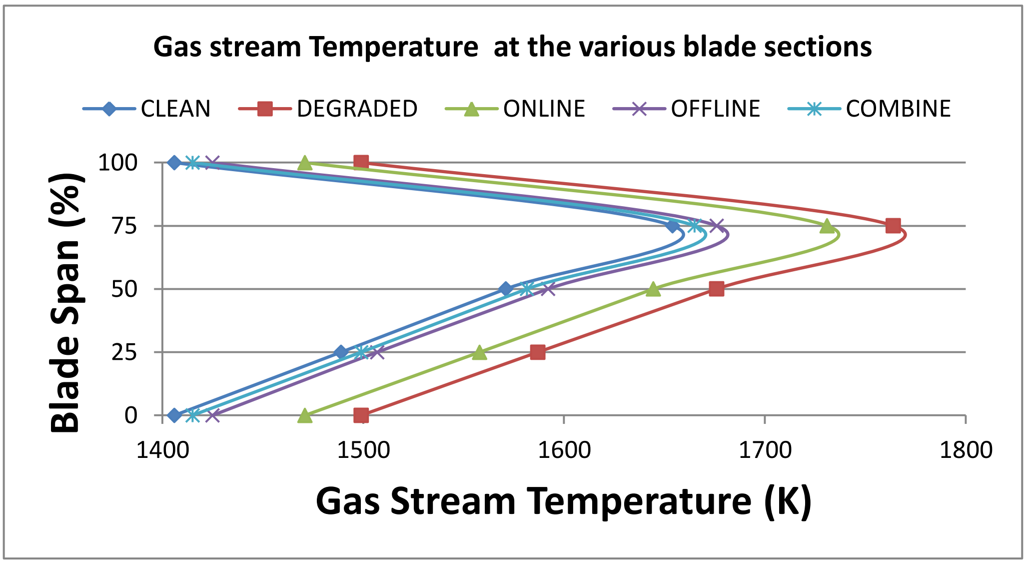


Figure 5-7: Gas stream temperature along the blade height

It is observed that the temperature of the fouled engine increases due to fouling, but it is highest at the 75% section of the blade. Subsequently, compressor washing was performed to determine the rate of recovery for the online, offline and combine online & offline compressor washing.

Table 5-1: Metal temperature at varying sections of the blade

Blade sections	Tip	75%	Mid	25%	Root
Gas Temp. (K)	1406	1654	1571	1489	1406

Table 5-1 indicates the values of gas stream temperatures at the various sections of the blade. Afterwards, average temperatures were taken to determine the various temperatures along the various blade heights. This is shown in figure 5-9 appendix E5-E9 for all the behaviours of the engine. Applying a cooling effectiveness of 0.4 into equation 3-3, with other values deduced from PYTHIA, the BMT will be determined thus;

$$T_b = T_g - \varepsilon\left(T_g - T_{cin}\right)$$

Gas temperature at tip, $T_{gtip} = (1530 - 150) = 1380K$

BMT at the TIP; $T_{btip} = 1380 - 0.4(1380 - 762) = 1132.8K$

Similar procedure is followed for the various sections of the blade and the results are as shown in Appendix E5-E9 for all cases.

Table 5-2: Average temperature at the varying blade sections

Blade sections	TDS-75%	75%-50%	50%-25%	25%-RDS
BMT (K)	1132.8	1182.3	1132.8	1083

In a similar way, the values from PYTHIA output file as stated in section 5.9.1 were inputted into equations 5-7 & 5-8 to determine the corresponding BMT at the various sections. It was repeated for the fouled and all wash cases as could be seen in figure 5-8.

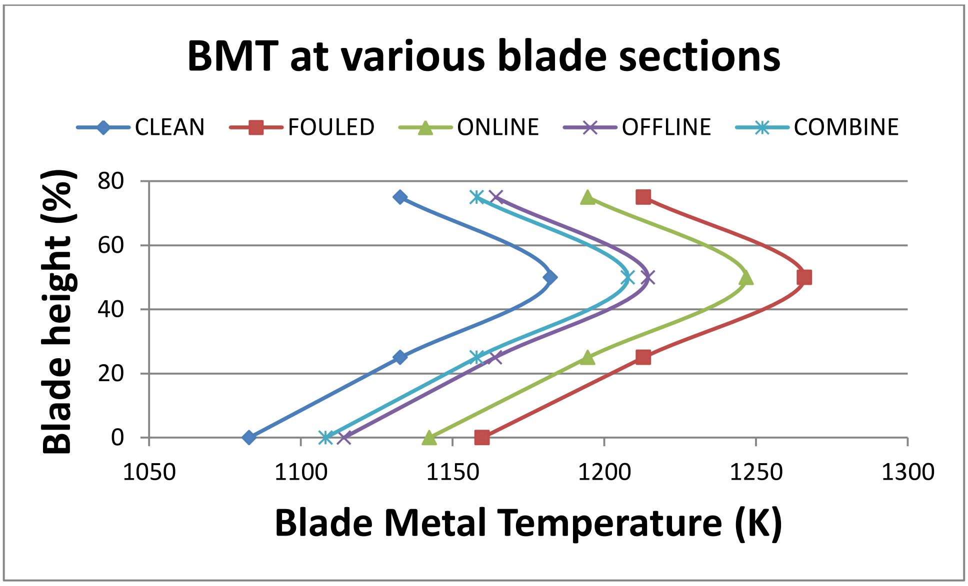


Figure 5-8: Blade metal temperature along blade height

5.8 Stress Determination of the HPT blade

The objective of the model developed for this research is to determine the variation of the metal temperature along the turbine blade sections as can be seen in figure 5-8, which has been coated with thermal barrier coating (TBC). The calculations here are similar to that of the thermal model. The blade was divided into four sections as shown in figure 3-5, and five gas temperature points were also created along the blade span to characterise the profile of the gas temperature. And the properties at the exit of the NGV were determined the same way the properties of the thermal model was determined.

There is usually time dependent deformation and temperature-induced changes in the microstructure of materials meant for service at very high temperature [174]. Rene 80 is a Nickel based super alloy that represents a class of material meant for application at high temperatures such as that faced by turbine blades. Creep is the main type of damage observed for turbine blades that are exposed to high temperature under varying stress condition for a protracted period of time.

Reducing the high cost of replacement or prematurely retired components, it is required that the remaining useful life of the components be determined. The churning out of design data is very expensive and time consuming.

- **Stress Model**

The direct centrifugal stress is employed in this research due to the mass of the blade, and the stress is evaluated from the root to the tip to enable the creep life calculation. The equations involve are as shown in equations 3-4, 3-5 and 3-6 respectively. The sectional stress can be determined with the application of equation 3-6;

- **Thermal Model**

It has been discussed in details in section 3.4.2 with equations 3-2 and 3-3 respectively.

- **Centrifugal Force**

This is the only force to be considered in arriving at the stress on the blade in this research work. It is due to the weight of the blade rotating (the stress acts through the blade, with the greatest occurring at the root where the full weight of the blade acts). The CF contributes the major stress in the blade, accounting for about 75%-90% of the total stress [5]. The blade is reduced to a rectangular block for the purpose of in-depth analysis and the block is divided into four sections as can be seen in figure 3-5. From the aerodynamic analysis as shown in Appendix E1.

D_{tip} = 0.9995m

D_{hub} = 0.9055m

Blade height, h = $(D_{tip} – D_{hub})/2$ = 0.0469m

Radius from CL to CG of the first section, R1 = 0.4586m

Radius from CL to CG of the second section, R2 = 0.4704m

Radius from CL to CG of the third section, R3 = 0.4821m

Radius from CL to CG of the fourth section, R4 = 0.4939m

The blade density, ρ =8194.5Kg/m^3 for the material properties of Rene 80, and the rotational speed of the blade, N = 8100RPM

By applying equation 3-6, the cumulative stresses at the different sections of the blade are as shown in Appendix E1.

The calculated values of stress (MPa) distribution along the various sections of the blade from the RDS to the TDS is as shown in Appendix E1, while an indication of the plots of stresses along the blade height is as in figure 5-10 . It can be observed that the blade is more stressed at the root as compared to the tip of the blade, and it is distributed cumulatively.

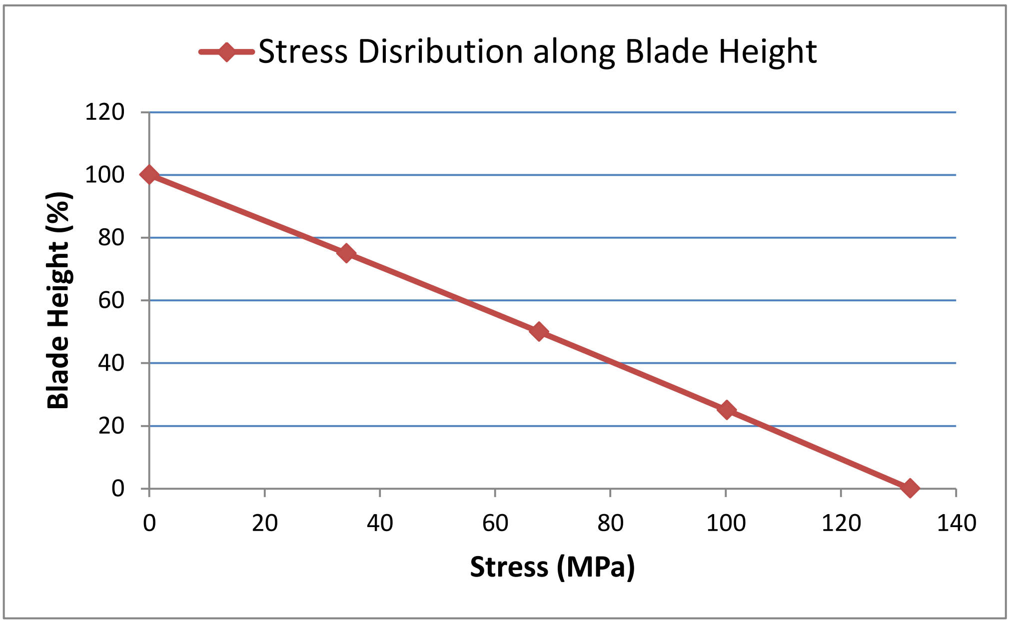


Figure 5-9: Stress Distribution along Blade Height

- **Larson-Miller Parameter**

The approach applied for the LMP is as described in section 3.4.2, and the equations are as explained in 3-7 and 3-8. According to Arrhenius law [150].

5.9 HPT Blade Creep Life Estimation

The reference point for the design was (TET=1505K, T_g=1355K, and T_b=1118K) which is equivalent to a creep factor of one, and figure 5-5 indicates the impact of ambient temperature on the creep factor of the blade. It is assumed that the power demand throughout this varying ambient temperature and TET is constant. It is expected that higher ambient temperature result in increased compressor delivery temperature, which amount to increased fuel flow and TET, and also lead to decrease in creep factor. Figure 5-5 shows the effect of ambient temperature on blade creep factor and can be seen that the creep factor dropped

It can also be seen here that the increase in ambient temperature increases the TET, which in turn causes the engine to foul. This is because the air density reduces with hot air at the inlet of the turbine and thus causes the mass flow to reduce as a result of constant area and velocity.

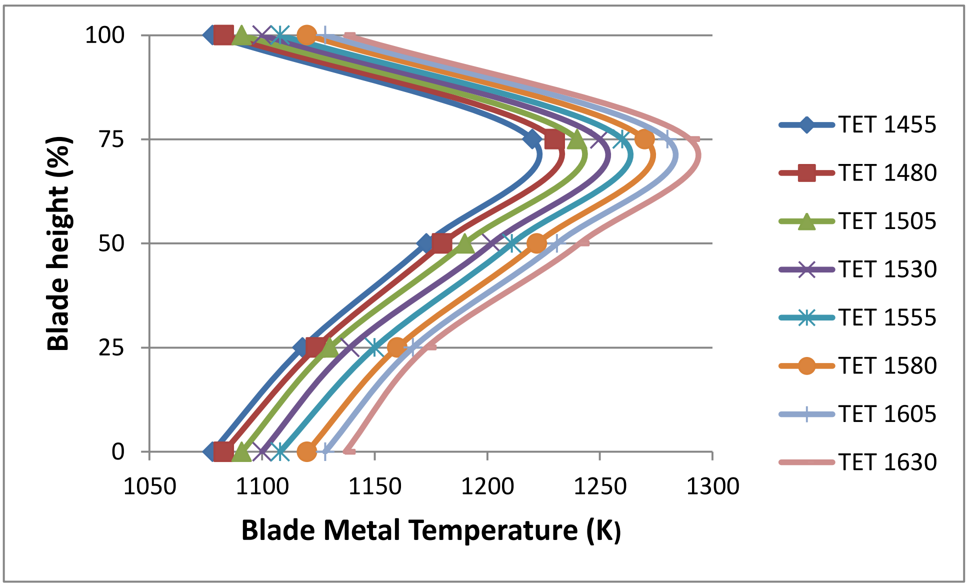


Figure 5-10: Effect of TET on blade metal temperature

Figure 5-10 shows the reaction of the blade temperature as a result of varying TET emanating from ambient temperature variation. An engine operating at constant PCN in a hot environment operates with lower N√T compared to a normal condition. In addition, the pressure at the exit of the compressor will decrease to enable constant NDMF as the temperature increases. This increase in the compressor exit temperature also increases the TET, and as a result, the BMT eventually increases as could be seen in figure 5-11. The blade experiences the highest temperature at the 75% position from the root because the blade rotation causes the peak gas temperature to shift from the middle of the span towards the tip. There was no much change in the centrifugal force because the PCN was constant, but the little changes were as result of stresses by the blade metal, and this is minimal compared to BMT. It can be deduced from here that the major factor affecting creep life is the change in ambient temperature which changes the BMT.

The parameters and corresponding values used for the creep life determination as could be found in Appendix E4 shows values of creep life at corresponding BMT due to ambient variations. These values are used to determine the blade life of HPT.

The cooling effectiveness was assumed, the compressor inlet temperature was gotten from performance simulation, the constant C is the constant used in the LMP model, though it usually ranges between 13 and 27, but 20 is assumed in this research work. The value of the LMP is gotten from calculation, and the TBC is assumed to be at the exit of the combustor.

Assumptions

- The temperature of the metal blade T_b is in K.
- The radius is the distance between the rotational axis and the section CG.
- The constant C is 20
- Thermal Barrier Coating is 150K
- Cooling effectiveness is taken to be 0.4.
- Factor of safety to be 60%

For the design point;

T_g=1355K, and T_b=1118K =2012R

In other to determine the LMP, we first determine the stress on the blade and read off the corresponding LMP from the chart in the appendix A. Considering the effect of centrifugal stress, Centrifugal force,

$$CF = m\omega^2 d_{cg} = \rho A h \left(\frac{2\pi N}{60}\right)^2 d_{cg} \qquad 5-12$$

Centrifugal stress,

$$\sigma_{CF} = \frac{CF}{A} \qquad 5-13$$

$$\sigma_{CF} = \rho h \left(\frac{2\pi N}{60}\right)^2 d_{cg} \qquad 5-14$$

$$\sigma_{CF} = 8194.5\ x\ 0.4586\ x\ 0.0470\ x \left(\frac{2\pi 81000}{60}\right)^2$$

$$\sigma_{CF} = 132MPa$$

The corresponding LMP for this stress is 50.4

$$t_f = 10^{\left(\frac{1000\ x\ 50.4}{2012} - 20\right)}$$

$$t_f = 112124\ hrs$$

Applying a factor of safety of 60%; **t_f = 67,275hours**

This means that the blade HPT will last about 67,275 hours before overhaul, which is over 7 years life at the entire blade.

5.10 Effects of BMT on Engine Creep Life

Haven calculated the blade metal temperature along the blade sections and the stress as well, the creep life can be determined with the application of equation 3-8. Applying similar procedure as that shown in section 5-9, but this time for the different sections of the blade. The result is as shown in figure 5-11, from Appendix E10 & E11.

$$t_f = 10^{\left(\frac{1000LMP}{T} - 20\right)}$$

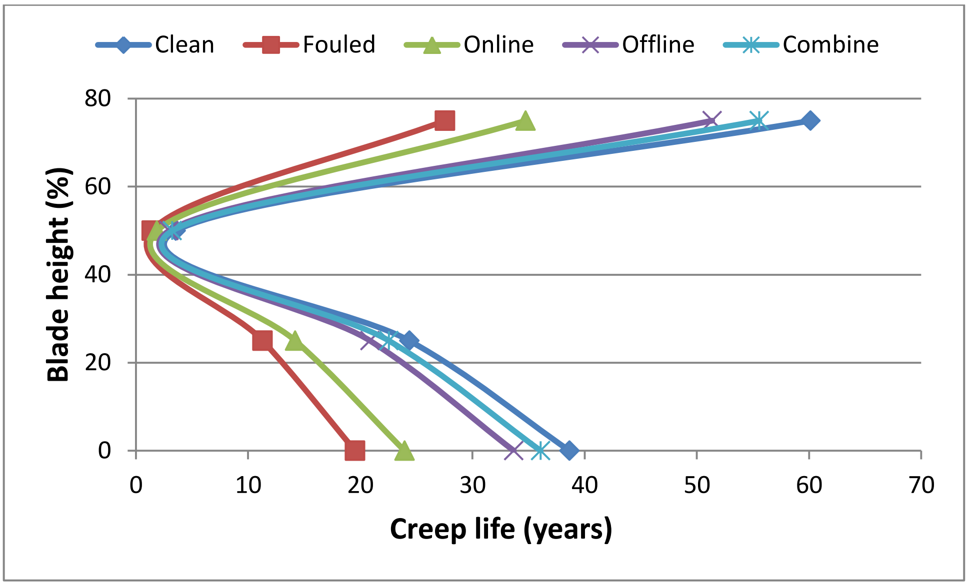


Figure 5-11: Effect of BMT on creep life at varying blade height

It is observed that the highest BMT occurred at the 75% section of the blade but that does not mean that the creep life is minimal at this point. Aside the time spent at highest temperatures, the major factors responsible for changes in the HPT blade creep life consumption include the RTDF, TET variations, stress and cooling effectiveness. Higher values will advertently results in higher creep life consumption. The minimal creep life is derived at the 50% section as could be seen from figure 5-11. This is an indication that creep life is a function of temperature and stress.

The number of operating hours for the clean case at the 50% section to failure is observed to be about 31,230 hours, which is about 3.5 years, assuming 8760 hours in a year. This estimation appears to be reliable when compared to that presented by [175] and [176], which shows the operation and maintenance life of industrial gas turbines for different application and fuel type.

5.11 Summary

This chapter has presented the effect of engine performance and operating environment to the creep life of an engine blade. The integrated creep life model and it sub-models have been explained. The algorithms, model design considerations, and the overall structure of the model have been presented and interlinked. The model incorporation with Pythia is also explained with values adopted from performance simulation output.

The effects and interlinks of thermal barrier coating, radial temperature distribution factor, ambient temperature, turbine entry temperature, gas stream temperature, and blade metal temperature have been explained. Also, the links between stress and thermal model with the time to failure have been clearly analysed.

The creep life was determined at the various sections of the high pressure turbine blade and it was observed that though the temperature was highest at the 75% sections, the creep life was minimal at the 50% section. This shows that creep is not only dependent on temperature but also the stress at the blade section. This was determined for the clean, fouled and various compressor wash cases as could be found in figure 5-11 and appendix E10, and it was observed that the clean engine has about 31 thousand operating hours at the 50% section of the blade while the degraded engine was discovered to be 1200 operating hours. This indicates that the blade will last about 3.5 years before major overhaul.

6. EMISSIONS MODEL

6.1 Introduction

Exhaust emissions from engine combustion processes have become a great concern to the public as a result of its health and environmental impact. There has been rapid change in the last decade for both regulations, controlling gas turbine emissions and in the technologies used to meet these regulations. Stationary gas turbines have become more reliable as prime movers in the oil and gas industries, and have acquired more ranges of application in combine cycle plants and in many areas of utility power generation. These and many more are the reasons why there is increasing demand on the GT maintenance engineer to keep the gas turbines at reduced pollutant emissions.

There are many types of pollutant emissions from gas turbine which include carbon monoxide (CO), carbon dioxide (CO_2), water vapour (H_2O), particulate matter (mainly carbon), unburned hydrocarbon (UHC), oxides of nitrogen (NO_x), oxides of sulphur (SO_x), and excess atmospheric oxygen and nitrogen which are considered as pollutants from combustion and are harmful to the environment and human health, as described by [115]. CO_2 and H_2O have not really been regarded as pollutant since they constitute the natural consequence of complete combustion. They both contribute to global warming and can only be reduced by burning less fuel.

Controlling GT pollutant emission has become a very important issue because of their harmful influence on environment and people in the community. The various pollutants and their corresponding effects are as shown in table 6-1 below.

Table 6-1: Gas turbine pollutants and their effects

POLLUTANTS	EFFECTS
Carbon monoxide (CO)	Toxic, ozone depletion
Unburned hydrocarbon (UHC)	Toxic
Particulate matter (C)	Visible
Oxides of Nitrogen (NO_x)	Toxic, precursor of chemical smog, depletion of ozone in Stratosphere
Oxides of Sulphur (SO_x)	Toxic and corrosive

Out of all the pollutants emissions, the naturally occurring CO_2 is calculated, while NO_x is considered by application of some emission control technologies, as the oxygen in blood can be reduced by CO, and even causes asphyxiation or death, and UHC are toxic and will combine with NO_x to form photochemical smog, as written by [115]. Also, photochemical smog can be produced by NO_x emissions and probably reduce the life of the plant by causing some damage and enlarge formation of acid rain.

The formation of pollutants can clearly be stated as dependent on the combustion pressure, temperature and mixing of the fuel and combustion air. The higher the pressure and temperature, the higher is the rate of reaction resulting in lower CO, but also in an increase in the formation of NO_x. The combustion pressure and temperature vary with engine load decreasing, when the load decreases. Therefore, we observe increasing level of CO and a decrease in the level of NO_x with the reduction in engine load.

6.2 NOx suppression using water and steam injection

It has been stated that the suppression of NO_x is very sensitive to combustion temperature. The amount of NO_x produced during combustion can be drastically reduced by introducing a heat sink to reduce the combustion temperature. The high specific heat of water has made it a good heat sink. The amount of NO_x can be reduced significantly by the injection of water into the primary zone. NO_x can be reduced by about 80% if equal amount of water and fuel (water to fuel ratio of 1.0) is ejected [177].

As effective as water injection could be in reducing NO_x, it has some side effects which include; CO and UHC increasing as a result of suppression of the combustion temperature. There is also increase in operation cost due to the cost of water treatment to improve the purity of water. In addition, the potential presence of corrosion of hot sections and therefore increased maintenance costs. There is also increase in fuel consumption resulting from the heat absorbed by the water. Though the power output is increased, the net effect is a reduction in thermal efficiency.

Water injection is still used for NO_x suppression despite the disadvantages, because it has been the most effective means to suppress NO_x emission substantially for many years. To be candid, about 35% of IGT nowadays uses water injection for NO_x suppression. Water injection is probably the most effective means of NO_x control when such engines uses liquid fuels. Operators also use water injection for power augmentation, but there is a loss in thermal efficiency, the increased power output is worthwhile in terms of increased production and revenue.

A similar impact in reducing NO_x is by steam injection, but the impact on thermal efficiency is more favourable because the latent heat of evaporation is normally supplied from the turbine exhaust heat. However, NO_x suppression with the use of water is more preferable. The injection of water to the flow stream has the effect of reducing the flame temperature. This effect is dramatically increased if liquid water is used, due to the latent heat of evaporation as the water become steam within the engine. Since the trend in higher GT compression ratio and, therefore, increased NO_x generation is likely to continue, research into a reduction of nitrogen oxides formation in gas turbine combustion chambers is warranted.

6.3 Determining Combustor Temperature

The conditions considered by convention are those for flow at constant pressure (isobaric) with no external loss of heat (adiabatic), with T^* as the absolute combustion temperature. At a standard value of 298.15K, the solution is based on equating energy enthalpy absorbed by the products in heating from this initial value of the combustion temperature T*, enthalpy $\mathbf{(H^{T^*})_P}$ released by the reactants $\mathbf{(-\Delta H_r^0)_R}$.

Hence, it is deduced that enthalpy absorbed by products = enthalpy released by reactants = difference in enthalpies of formation.

$$(H^{T*})_P = (-\Delta Hr^0)_R = -[(\Delta Hf^0)_P - (-\Delta Hf^0)_R] \qquad 6-1$$

Defining;

$$(H^{T*} + \Delta Hf^0)_P = (\Delta Ht^{T*})_R \qquad 6-2$$

To be the total enthalpy of products, then;

$$(Ht^{T*})_P = (\Delta Hf^0)_R \qquad 6-3$$

At standard initial temperature;= $(\Delta Hf^0 + H^T 1)_R$

Formation + sensible at initial temperature T_1 of reactants. Therefore,

$$(Ht^{T*})_P = (Ht^T 1) + (H^T 1) \qquad 6-4$$

That is total fuel plus sensible air. (Since ΔH_f^0 for elements =0) [174]. The details can be found in Appendix F1.

ΔHr^0= enthalpy of reaction

Since ΔHr^0 for C_3H_8 = -103.92KJ/mol, solution by interpolation gives:

$$T* = 2200 - 100\frac{(103.92 - 8.58)}{(133.11 - 8.58)} = 2200 - 100\frac{(95.34)}{(124.53)} = 2200 - 76.6$$

$$= 2123K\ approx$$

6.4 CO_2 Emissions Calculation

In a combustion chamber with complete combustion in the presence of excess air; the type, quality and quantity of fuel used will determine the amount of CO_2 to be emitted. The equation given below is used to calculate the emitted CO_2.

$$C_xH_y + nO_2 \rightarrow n_1CO_2 + n_2H_2O \qquad 6-5$$

Here, x and y are the carbon- hydrogen atomic ratio of the fuel.

The equation states that one mole of the fuel will react with n moles of O_2 to produce n_1 moles of CO_2 and n_2 moles of H_2O. For a molar balance performance:

$$n_1 = x$$

$$n_2 = 0.5_y$$

$$n = n_1 + 0.5n_2 = x + 0.25_y$$

Substituting n, n_1 and n_2 into equation 6-5:

$$C_xH_y + \left(x + 0.25_y\right)O_2 \rightarrow xCO_2 + 0.5_yH_2O \qquad 6-6$$

Stoichiometric:

$$C_3H_8 + ms(O_2 + 3.76N_2) = n_1CO_2 + n_2H_2O + n_6N_2 \qquad 6-7$$

C balance: n1=3

H_2 balance: n2=4

O_2 balance: n1 + 0.5 n2 = ms = 3+2 = 5

N_2 balance: n6 = 3.75ms =18.8

20% Lean m= 1.2ms = 6

$$C_3H_8 + 6(O_2 + 3.76N_2) = 3CO_2 + 4H_2O + n_5O_2 + n_6N_2 \qquad 6-8$$

New O_2 balance: n5 = m-ms = 6-5 =1

New N_2 balance: N6 = 1.2 (18.8) = 22.56

Hence, x mole of CO_2 will be produced by one mole of fuel, and one mole of fuel will weigh:

X x 12 + y x 1kg and one mole of CO_2 will weigh 44.01kg. Hence,

$$1kg\ of\ fuel = \frac{44x}{(12x+y)} kg\ of\ CO_2 \qquad 6-9$$

Or

$$1kg\ of\ fuel = \frac{44}{(12 + y/x)} kg\ of\ CO_2 \qquad 6-10$$

The molecular weight of CO_2 is 44.01g/mol as could be seen in the above equation.

$$m_{f} = \frac{E}{(\eta x FCV)} \qquad 6-11$$

$$e_{CO_2} = \left(\frac{44.01}{44}\right) x\ C \qquad 6-12$$

$$m_{CO_2} = e_{CO_2} x\ m_f \qquad 6-13$$

In operating a gas turbine, there is a tax liability associated with the amount of emission dissipate. The equation for determining this tax is as shown below;

$$Carbon\ Tax = mCO_2 * Tax\ rate \frac{\$}{Kwhr} \qquad 6-14$$

m_f = mass of fuel consumed

C = mass content of carbon in fuel (e.g kg C/kg fuel)

mCO_2 = mass of CO_2 emitted

E = Useful energy produced by the system

eCO_2 = CO_2 emission per unit mass of fuel e.g kg CO_2/ kg fuel

FCV = fuel calorific value

η = efficiency of the system based on the lower heating value of fuel

6.5 Empirical Correlations of NOx Emissions

Many correlations have been developed and validated by some research programmes and some of these parametric models for predicting NOx will now be discussed. In this research work, NOx and CO will be determined by the application of correlations proposed by Lefebvre [120]. The correlation proposed by Lefebvre suggests that;

$$NO_X = \frac{9 * 10^{-8} P^{1.25} V_c exp(0.01 T_{st})}{m_A T_{pz}} \frac{g}{kg} \; fuel \qquad 6-15$$

Where;

V_c = the combustion volume, m^3

P = the combustion pressure, kpa

T_{st} = the stoichiometric temperature, K

Ma = the combustion air flow, kg/s

T_{pz} = the average primary zone temperature, K

NO_X = calculated as an emission index in g/kg of fuel

The correlation has been developed for conventional spray combustors. It can also be used for lean pre-mixed vaporiser combustors so long as the primary zone temperature T_{pz}, which will be the maximum temperature attained during combustion, is substituted for T_{st} [156].

Figure 6-1 shows the reaction of NO_x increase against primary zone temperature. It is observed that an increase in primary zone temperature by about 10.2% increases NO_x emission by about 96%. It can also be noted that the highest amount of NO_x emitted is within standard range, which does not exceed 25ppmv.

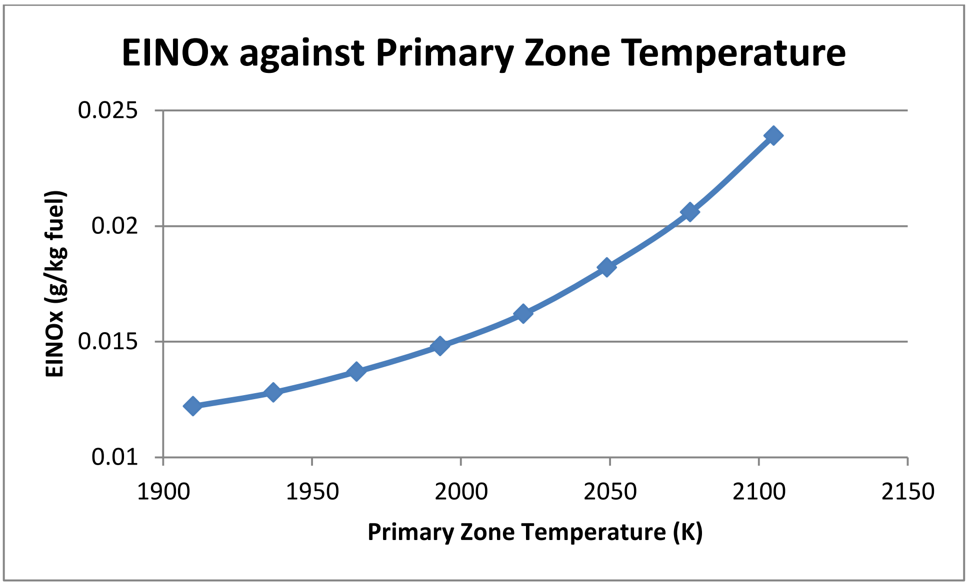


Figure 6-1: Effect of primary zone temperature on $EINO_X$

6.6 Empirical Correlations of CO Emissions

One of the incomplete combustion products which are important factors to influence gas turbine engine performance and combustion efficiency is known as carbon monoxide (CO). Where air-fuel mixture is lean and the temperature at the primary zone is low, CO dominates at low-power condition. The three major constituent of CO are; inadequate mixing and evaporation of air-fuel which determines the concentration of gas fuel, low to react completely, the liner wall cooling air is important to suppress and quench the chemical reaction, and insufficient burning rate is harmful to the degree of chemical reaction from gas fuel to carbon dioxide.

A correlation for CO proposed by [120] is summarised by [115] as follows:

$$\mathrm{CO} = \frac{86\mathrm{ma} * T_{pz} * \exp - \left(0.00345 T_{pz}\right)}{\left(V_c - V_e\right)\left[\frac{\Delta P}{P}\right]^{0.5} P^{1.5}} g/kgfuel \qquad 6-16$$

Where

V_c Is the combustion volume, m^3

V_e Is the volume occupied by the evaporated fuel, m^3

P is the combustion pressure, kpa

ΔP is the combustion non-dimensional pressure drop

ma Is the combustion air flow, kg/s

T_{pz} Is the average primary zone temperature

CO is calculated as an emission index in g/kg of fuel

And the volume employed (V_e) in fuel evaporation is obtained as:

$$\mathrm{V_e} = \frac{0.55 m_{pz} D_0^2}{\rho_{PZ} \lambda_{eff}} \qquad 6-17$$

In the case of CO, unlike NO_x that increases with higher temperature. The CO decreases with increase in primary zone temperature. Increase in primary zone temperature by 10.2% decreases the CO by about 44%. This research shows that 1.57% increase in temperature decreases CO by 7.18% and 16.7% increase in resident time increased the CO by about 83.3% as could be seen in the figure 6-2 below. This implies that CO is highly dependent on resident time in the combustor primary zone.

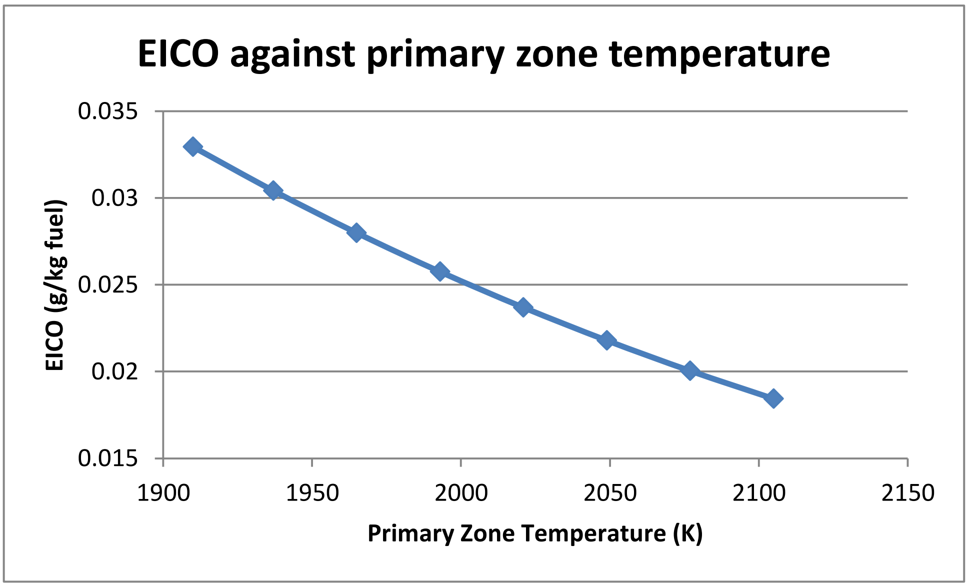


Figure 6-2: Effect of primary zone temperature on EICO

The decrease is because as the combustor is operating at fuel-rich resulting from higher temperature which calls for increase in specific fuel consumption, and there is insufficient oxygen to complete the reaction to CO_2, a large amount of CO is thus formed.

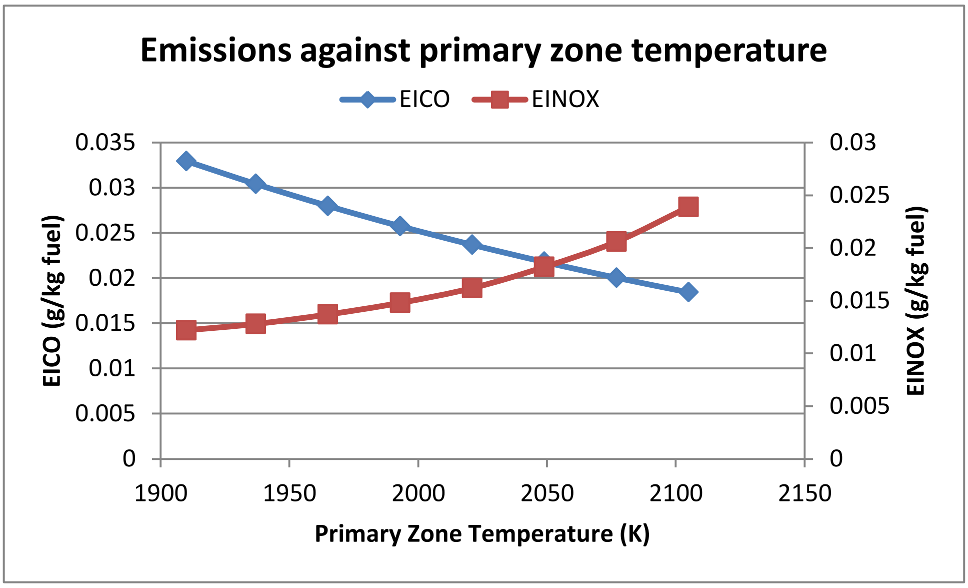


Figure 6-3: Effects of primary zone temperature on EINOx and EICO

In comparing the effect of primary zone temperature on emissions as could be seen in figure 6-4, it is observed that NO_x is more affected than CO for the same amount of temperature reduction. As earlier stated in figures 6-1 and 6-2, an increase in primary zone temperature of 10.2%, increases NO_x by 96% but decreases CO by 44%.

6.7 Effects of Compressor Washing on GT Emissions

Figure 6-4 and 6-5 indicates the $EINO_x$ and EICO respectively. Equations 6-15 and 6-16 were applied for the determination of $EINO_x$ and EICO respectively. The description of the parameters is as stated in section 6.5 and 6.6 respectively. The primary zone temperature is known to be higher than the TET, and was determined as shown in section 6.3 derived from appendix F1. The combustion temperature and pressure was gotten from PYTHIA performance simulation output results. The stoichiometric temperature was assumed to be the same as the primary zone temperature as proposed by [156].

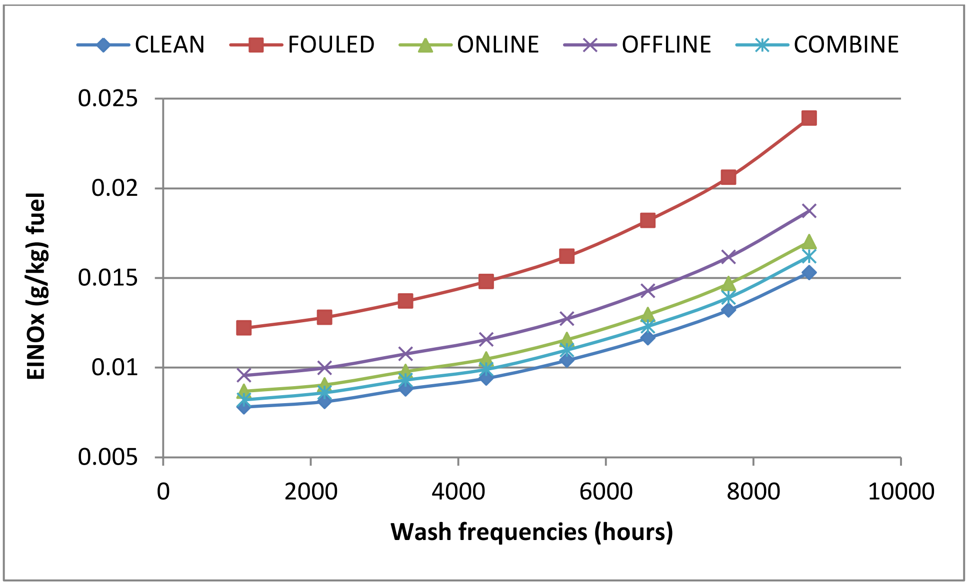


Figure 6-4: Effect of compressor washing on EINOx

For the baseline engine behaviour, the Pythia output was inputted into the correlation and the results were recorded for each section of the temperature. This was repeated after the fouling was implanted, and the process was repeated for each case of the compressor washing. The output results were plotted as could be seen in figure 6-4 and 6-5 respectively, and could be found in appendix F2 and F3.

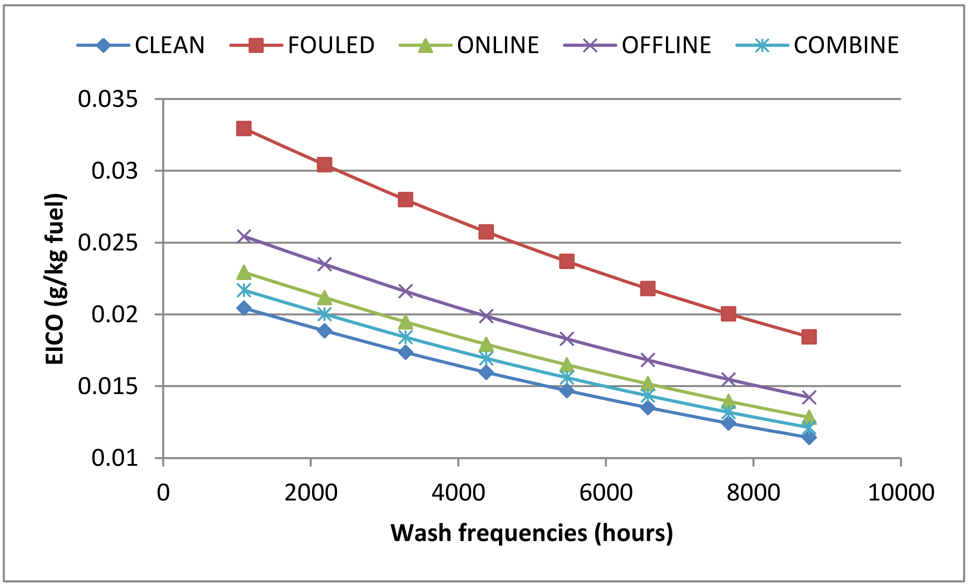


Figure 6-5: Effect of compressor washing on EICO

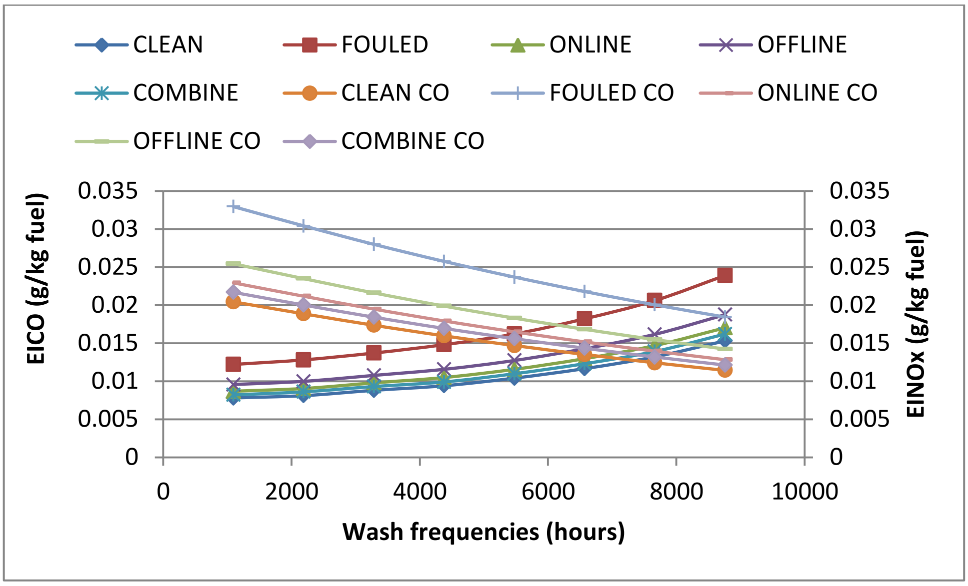


Figure 6-6: Effects of compressor washing on EINOx and EICO

Figure 6-6 is the combination of both behaviours. It can be clearly seen that as the compressor is being washed, the rate of pollutant emission is drastically reduced accordingly. It also shows that increase in fouling increases NO_x emission, but reduces CO emission.

6.8 Summary

This chapter contains some factors that affect the formation of pollutants such as NO_x and CO. Combustion temperature and pressure are predominant factors discussed. Other parameters that also contribute to the formation of these pollutants include; fuel-air ratio, fuel and air mixing, combustion geometry and residence time. Different correlations have been proposed and validated and serve as a very useful means of predicting emissions from gas turbines. Although not considered as toxic, CO_2 is also produced during combustion. It is a greenhouse gas which is thought to be responsible for global warming and as such considered. The prediction of CO_2 is relatively straightforward since the carbon-hydrogen ratio of the fuel is known.

7. ECONOMIC MODEL

7.1 Introduction

Engineering economics deals with the methods that enable one to take economic decisions towards minimising costs and/or maximising benefits to business organisations [141].

The financial evaluation of a project studies its performance and return (profitability) from the points of view of the industry (the utility and the firm), the owner and the investor [129].

Several researches have been conducted on electricity pricing, fuel price forecasting, power demand and supply forecasting [179, 180]. The three major factors include ambient conditions, electricity cost and fuel price. This research focuses on gas turbine performance degradation due to compressor fouling. Pythia gas turbine performance software was used to simulate the thermodynamic behaviour of the gas turbine at design and off design conditions. The fouled case was later washed at online, offline and a combination of online and offline. The power recovered at each case was recorded and converted from MWh to energy in kwh. This was linked with the lifing model used with the HPT blades to calculate the remaining life at each section of the blade, and then the cost of maintenance. An emission model was also included to evaluate the cost of the emission tax at each case.

To enable the determination of the effect of operating and ambient conditions on the electricity, and the GT plant cost, a GT was modelled as shown earlier in figure 3-1. More so, the operating environment of the GT plant which plays a significant role in the performance of the turbine is described therein.

Figure 3-7 shows an economic model developed for this study, in which the approach used for assessing the economic behaviour of all cases are employed. The results generated from turbo-match were used to develop the creep life which consisted of sub-models for creep, thermal model, stress analysis and thermal performance. The model output was combined with the data from the lifing and emission models. The results are then used to estimate the cost of electricity [181-183].

One of the main reasons for this research is to develop a procedure which integrates gas turbine activities, to enable it economic measures like fuel, operational & maintenance, and other costs and revenues to evaluate with time during the operation period. The economic metrics is a system level that provides a basis from which to evaluate long and short term GT profitability when optimising performance operation. Integrating local economic metrics along the entire operating time horizon is one method employed. Such method solicits the accuracy and efficiency to be balanced evaluating long-term economic measures and inter-relationships, and also needs various evaluation points.

7.2 Methods of Economic Analysis

Maximising profits per share consistence with good business practices are the financial objectives of power generation companies. The technic for economic analysis on this research is focused towards this objective. The applied approach in analysing these financial obligations is the life cycle cost (LCC).

The LCC is applied by calculating revenues, subtracting expenses, and computing earnings for each wash case over the period or over life of the engine blade.

7.3 Total Profit Equation (Objective Function)

Industrial gas turbines produce power, thereby generating revenue and profits for industries that deploy them. However, the sum of the operating costs associated with the operation of gas turbines are fuel, operation and maintenance, depreciation, returns paid to investors, and emission taxes.

The purchase cost of the gas turbine and the auxiliary units is known as its capital cost, and it also include the installation space of the engine. The cost of the safely disposal of the engine and its associated systems at the end of their lives is known as the abandonment cost. The total ownership or life cycle cost

(LCC) is the sum of the capital cost, the abandonment cost, and the operating cost.

The total profit (TP) generated is the difference between the total revenue (TR) and the life cycle cost (LCC). The higher the revenue and the lower the LCC, will yield better TP. This section of the research presents algorithm that enables the LCC of the engine model presented in this study for different wash cases.

7.4 Environmental conditions for power markets

There has been several research conducted on electricity pricing, fuel price forecasting, power demand and supply [184]. Three major factors which are functions of the time of the year such as fuel price, ambient conditions and electricity price have been considered.

In a deregulated power market, it would be theoretically expected that the electricity price and fuel price are stochastic or random in nature.

Operating a gas turbine without carrying out a proper scheduled routine check would be very disastrous to the engine user or owner. The operating environment affects the total power produced in that; if the engine is being operated for a long period under severe condition without proper maintenance schedule, it will eventually breakdown earlier than expected and as such will cause a very huge economic set-back to the GT owner. Keeping the engine for a longer life, an economic comparative study is carried out in this section for the various wash technique applied.

It is an algorithm capable of capturing the consequences of compressor fouling on GT operation and the economic effects of GT compressor water wash. The operating conditions have been explained earlier in chapter three of this research and the simulation results are as presented in chapter four in this research as well. The cumulative economic revenue and other cost associated with each of the compressor wash cases will be accounted for and compared. The cases are fouled, compressor online washed, compressor offline wash, and a combination of both online and offline wash. The necessary algorithms applied are as explained in the following session.

7.5 Total Revenue

Revenue is all gain made, including total sales, exchange of assets, increase in owners' equity and interest. Calculated before any expense is subtracted;

$$TR_T = \int EP(t) * P(t) dt \qquad 7-1$$

Where: TR_T is the total revenue over time T ($).

$EP_{(t)}$ the projected price of electricity at time t ($/kwh).

$P_{(t)}$ is the electricity power output (Energy gained) of the plant at time t (MWh) or (kwh).

7.5.1 Electricity Produced and Price

Energy produced is source of income to GT owner. For each wash case calculation, the total electricity produced is taken to be the sum of energy gained multiply by the cost of electricity in kWh for the period of operation.

$$Annual\ EP = \sum P_e * E_w \qquad 7-2$$

Where:

EP is Energy produced

P_e Price of electricity

E_w is Energy gained from compressor washing

7.5.2 Life Cycle Cost (LCC)

This is the sum of the total fixed cost and the total variable cost. It is also known as the ownership cost. The capital cost which comprises of the cost of equipment purchase, installation and transportation, and fixed operation & maintenance cost are known as the fixed cost. While the fuel cost, operation &

maintenance and repair cost are classified as variable cost. This is classified as operating cost [185].

Summation of the gas turbine and auxiliary systems purchase cost (capital cost), the abandonment cost of the GT and their associated systems at the end of their lives, and the cost of operation are all together known as the life cycle cost or total ownership cost. The difference between the TR and the LCC is known as the TP generated. Hence, lower LCC and higher revenue will mean increase in TP.

$$LCC\ (pa) = CC_{pa} + FC_{pa} + O\&MC_{pa} + ETC_{pa} \qquad 7-3$$

Where: CC is the capital cost per annum.

FC is the fuel cost per annum.

O&MC is the operating and maintenance cost per annum.

ETC is the emission tax cost per annum.

7.5.3 Evaluating capital cost

The capital, investment or initial cost will vary depending on the technology applied for generating power, as different technology will have varying operational capacities and efficiencies, with varying costs in terms of $/kwh [185].

7.5.4 Costs of Equipment

Equipment cost is the purchase price of the equipment plus its associated taxes, and other cost such as transportation to site or delivery delays.

7.5.5 Costs of Installation

This includes installation of the wash equipment, first set of spare parts and any tools to be used for servicing and repair.

7.5.6 Fuel Costs and Consumption

Fuel cost increase is a huge economic set-back to GT users, and it is expected to increase continually. This is an issue that is having a great effect on the operation of existing GT facilities and on the design of new ones.

Fuel cost is the most significant operation cost, and could be about 60% or more of the total operation cost over a typical service life. Considering the huge amount involved in GT operations, it is important to note that a small percentage improvements in fuel consumption will yield reasonable amount of savings over the life of the plant as will be shown later in this studies. See equation 3-20.

This is converted to energy units called Therm. It is the unit for the natural gas price. The mass of fuel can be converted to Therm and the price can be determined when the LHV of natural gas is known. See equation 3-21 and 3-22.

7.5.7 Operation and Maintenance cost

It is dependent on factors like fuel type, operation cycle and environment, and operation & maintenance quality. It is divided into fixed cost FC =staff, insurance etc and variable cost VC= wash fluids etc.

- **APPLICATION**

In other to effectively determine the optimum water wash interval, there is need to verify the total loss resulting from engine degradation, which include;

- Loss in revenue due to power loss from fouling
- Cost of increased fuel flow
- Cost of engine shutdown for washing
- Cost of wash kits
- Cost of fuel saved from engine shutdown
- Losses resulting from water heating and other associated cost contributing to total loss

- **Loss in revenue due to power loss from fouling**

The percentage power loss is converted to kilowatt hour (kWh) to enable calculation of the arear under the curve. The 0% in figure 4-10 represents 29MW which is original power of the engine. The cost of power per MWh is said to be £50 [186], which is $73 by conversion.

For the clean case, since the engine is operating at 29MW, therefore;

$$Energy\ clean = power\ output * 10^\wedge 3 * operating\ hrs \qquad 7-4$$

=29*10^3*8760=254040000kwh=254040MWh

For the fouled case, area under the degradation curve is considered. Appendix G.

$$Energy\ fouled = Area\ under\ the\ curve \qquad 7-5$$

=237464000kwh=237464MWh

Power loss will be integral of the area under the curve

$$Energy\ loss\ (MWh) = Clean\ engine\ energy - Fouled\ engine\ energy \qquad 7-6$$

=254040000-237464000=16576000kwh=16576MWh

Maximum revenue obtained is the product of the energy at each case and the electricity selling price.

$$Maximum\ revenue = Total\ energy\ \ loss * Electricity\ price \qquad 7-7$$

$$Cost\ of\ energy\ loss = 16576000 * \left(\frac{73}{100}\right) = \$12{,}100{,}480\ in\ 8760\ operating\ hrs$$

From this $12,100,480, it is assumed that ($7,260,288) 60% is as a result of fuel cost increased from fouling. This means that revenue worth $4,840,192 would be lost due to fouling during one year period of the engine operation without engaging any form of washing.

The same process was iterated by the application of equation 7-7 to determine the wash cases as could be found in Appendix G.

The maximum revenue that can be obtained after 8760 operating hours was determined for the various engine cases for comparison, and to determine the revenue that could be recovered as a result of compressor washing for the different cases. The revenue loss due to fouling and the recoverable revenue resulting from compressor washing can be achieved by the application of equation 7-8 and 7-9 respectively.

$$Revenue\ loss = Max.revenue\ clean - Max.revenue\ fouled \qquad 7-8$$

$$Revenue\ gained = Max.revenue\ wash - Max\ revenue\ fouled \qquad 7-9$$

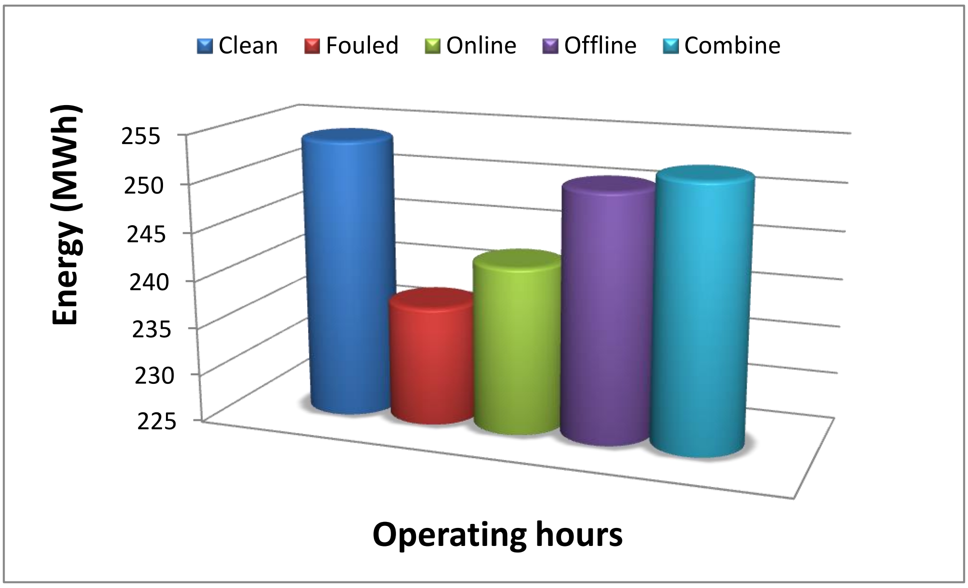


Figure 7- 1: Energy from compressor washing per annum

It can be observed from figure 7-1 that combine washing shows a better energy recovery than other wash cases. The revenue saved from the various wash cases are as can be seen in figure 7-2 expressed in million dollars (M$). It was observed that the maximum revenue lost was about 6.5%, while the recovery from the various wash cases yielded about 2.1%, 5.3%, and 5.9% for the online, offline and combine wash cases respectively. The results for all cases can be found in H.

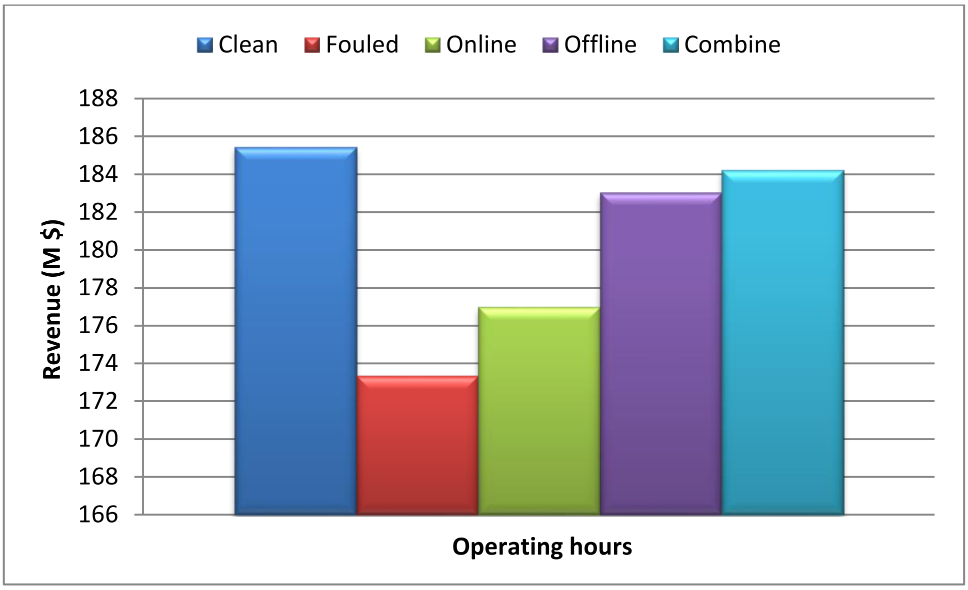


Figure 7- 2: Total revenue obtained from power due to compressor washing

- **Cost of increased fuel flow**

Fuel cost is assumed to be 60% of the revenue and can be determined by the application of equation 7-10, with the results as could be seen in Appendix H.

$$Fuel\ cost\ (\$) = 60\% * Maximum\ revenue\ for\ clean \qquad 7-10$$

From the performance output results of PYTHIA, the relationship between loss power and fuel consumption resulting from fouling decreases output power with an associated fuel flow increase. The fuel flow from performance design point was 1.722kg/s. Therefore, the CO_2 can be determined by the application of equation 7-11.

$$CO_2\ (kg) = \quad fuel\ flow * 3600 * operating\ hours \qquad 7-11$$

For Clean

$$CO_2 Emission = 1.7222 * 3600 * 8760 = 54311299kg$$

For Fouled

$$CO_2\ Emission = 1.9835 * 3600 * 8760 = 62551656kg$$

This means that 54311 tonnes of CO_2 is emitted for the clean case, and 62551 tonnes is also emitted for the fouled case. The other cases can be found in Appendix F4.

- **Emission Tax CO_2**

CO_2 emission increases with increase in fuel consumption from gas turbine. A simple illustration is shown in equation 6-8 for propane which is a family of natural gas. Therefore, for a complete combustion reaction it will take 44g of natural gas to yield 44g of CO_2, which means a unit mass of natural gas, will yield 1 units of CO_2. This indicates that 54311ton amount of excess as determined with the application of equation 6-11 to 6-13 will yield:

For Clean= $54311x\ 1 = 54311 ton\ of\ CO_2\ per\ year$

The US department of energy 'rule' set the rate of emission tax at $19.32 per ton [187]. Since the value is marginal, therefore, applying equation 6-14 for carbon tax becomes:

For Clean $= 54311x\ 19.32$

Carbon tax = $1049294

For Fouled= $62551\ x\ 1 = 62551 tonnes\ of\ CO_2\ per\ year$

For Fouled $= 62551\ x\ 19.32$

Carbon tax = $1208498

Since the combustion process for a gas turbine is more complex than the simple combustion reaction [115], this is assumed to be a rough estimate. The threshold beyond which the carbon tax applies has not been taken into consideration.

Figure 7-3 is the accumulated CO_2 tax per annum. It can also be observed that the combine wash is least affected. This is because in the combine wash, the emission reduction is more achieved and thus a significant change in the rate of emission reduction as compared to other cases. In a similar way, the offline wash achieved more reduction than the online case, as such, it is indicated clearly on the figure and thus more revenue is saved for the GT operator.

In a similar way, the iteration is made for other cases as well, and can be found in Appendix F4.

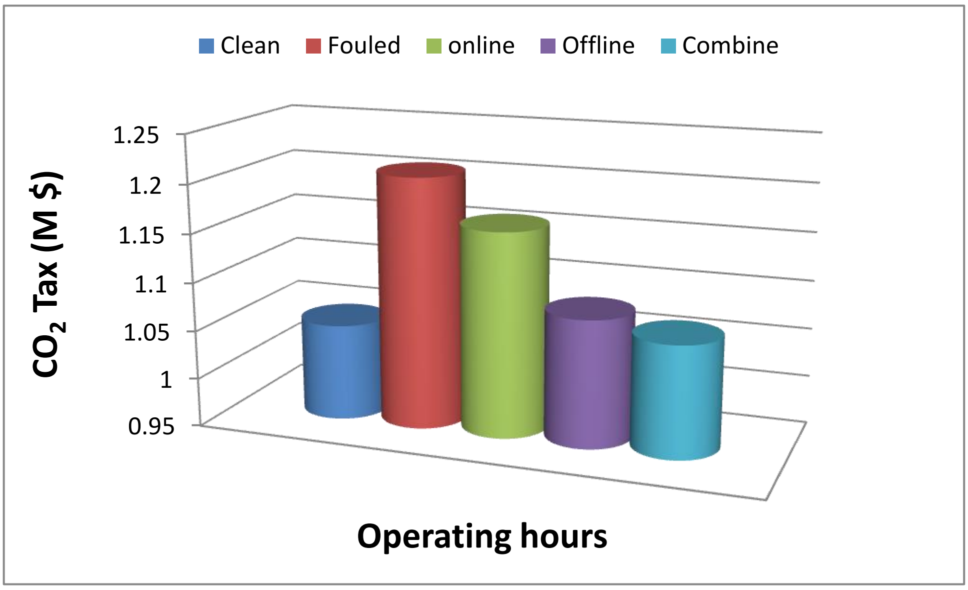


Figure 7-3: Emission CO_2 per annum

- **Emission Tax NOx**

The tax rate for NO_X emission as fixed by [188] is £38/ton which is equivalent to \$42.41. The specific NO_X emission is as could be found in Appendix F2, while the NO_x emission tax is derived by applying equations 7-12 to 7-15 in deriving each item required, as could be found in Appendix F5.

The fuel consumption was extracted from PYTHIA output result and computed accordingly to derive the emission NOx (kg), and the emission tax is achieved with the application of equation 7-15.

$$NO_X\ factor\ F\ \left(kg\frac{NOx}{ton}fuel\right) = Specific\ NOx\ emission\ in\ \left(\frac{g}{kwh}\right) * 1000 \quad 7-12$$

Or

$$NO_X\ factor\ F\ \left(kg\frac{NOx}{ton}fuel\right) = specific\ fuel\ consumption\ \left(\frac{g}{kwh}\right) \quad 7-13$$

$$NO_X(emission\ of\ NOx\ in\ kg) = F * fuel\ consumption\ in\ ton \qquad 7-14$$

$$NO_X\ Emission\ Tax\ (\$) = Emission\ NOx * Tax\ rate \qquad 7-15$$

The cost of NOx increase due to fouling is similar to that applied for CO_2, in this case, it is about 44% increase amounting to about $892984 in revenue reduction. The emission tax was derived for each wash case, and the results are presented in Appendix F5. The highest point was applied for this calculation since it is the point that yield the most revenue lost and used as the worst case, as could be seen in figure 7-4.

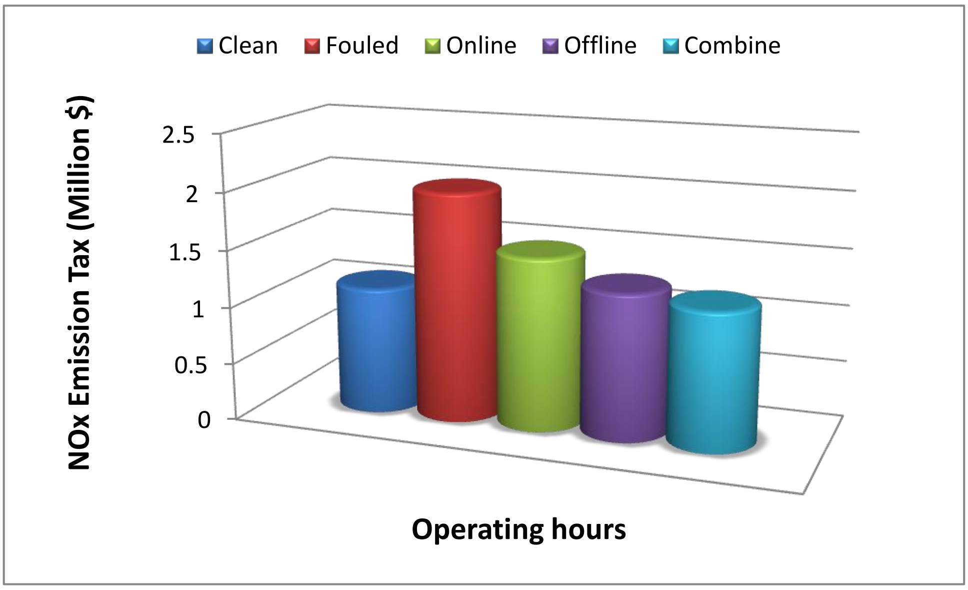


Figure 7-4: NOx Emission Tax per annum

It can be observed from figure 7-4 that compressor washing made a significant change in reducing NOx emission for all the wash cases. The combine case seems to reduce the highest portion of emission of about 40% off the fouled.

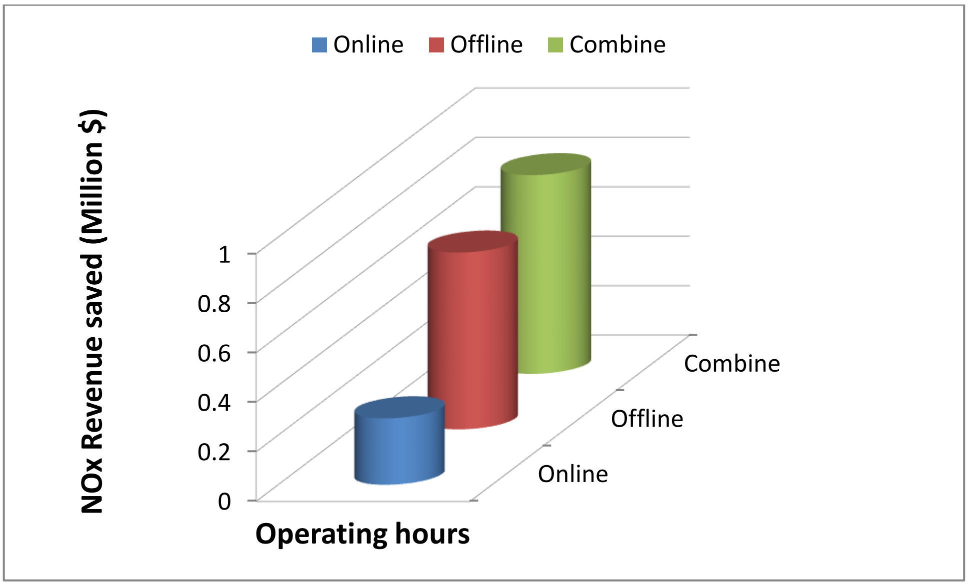


Figure 7-5: Savings for NOx due to compressor washing

In a similar way as could be seen in figure 7-5, the combine case of compressor washing saved more revenue for the GT user. Details are as presented in Appendix F5.

- **Emission Tax CO**

The tax rate for CO emission according to [122, 123] was $43 per ton. In this regards, converting the amount of emission as shown in Appendix F3 from g/kg to achieve the amount of CO emission, equation 7-16 was applied and the results obtained are as could be seen in figure 7-6 and 7-7.

$$CO\ Emission\ Tax\ (\$) = Emission\ CO * Tax\ rate \qquad 7-16$$

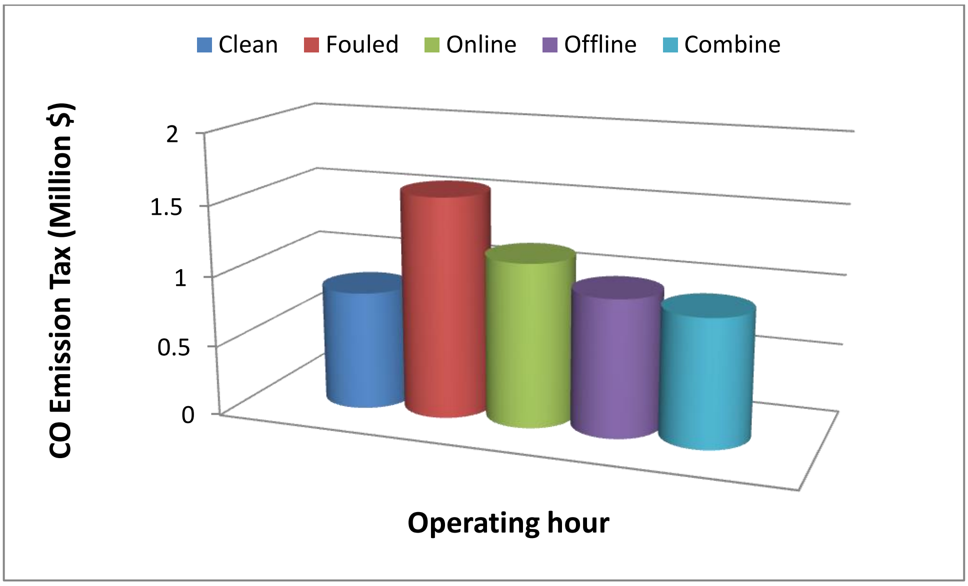


Figure 7-6: CO Emission tax per annum

It can be observed from figure 7-6 that the maximum CO emission achieved at each case differs. The fouled case is seen to be having the most CO emitted while the clean case is least from the engine as expected. On the other hand, the combine case achieved more savings than other wash cases as can be seen in figure 7-7. There was 46% increase in CO emission due to fouling and the three cases of compressor washing was observed to achieve 26%, 38%, and 42% for the online, offline and combine case respectively.

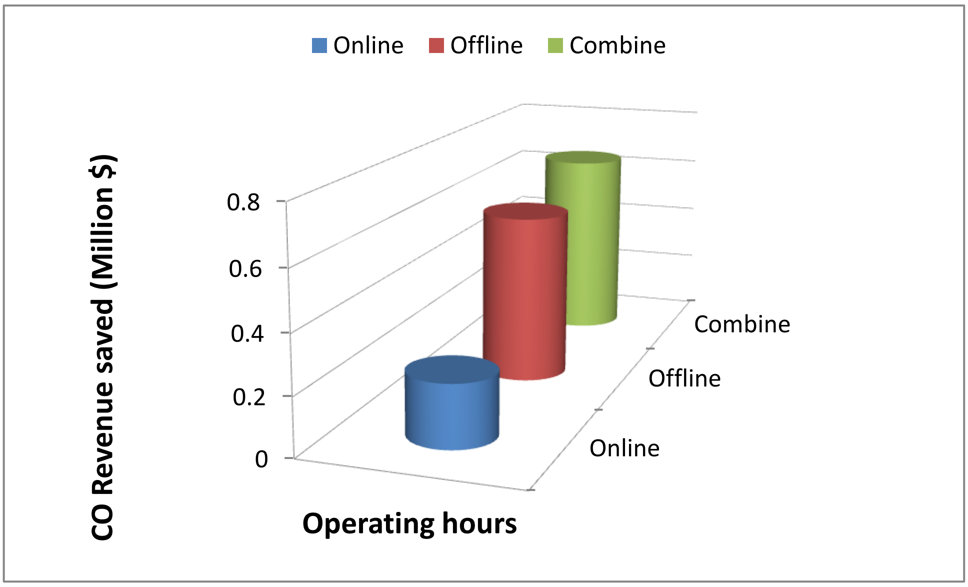


Figure 7-7: Revenue saved for CO due to online compressor washing

- **Cost of engine shutdown for washing**

OFFLINE

The unit was assumed to shut-down for 12 hours for each wash;

$$kwh = MW\ x\ 10^3\ x\ hours\ of shutdown\ interval$$

$$For\ one\ interval = 29\ x\ 10^3 x\ 12 = 348x10^3 = 348000kwh$$

$$For\ the\ four\ intervals = 1392000kwh = 1.392x10^6 Kwh$$

Total cost of loss power due to shut-down for offline water wash will be;

$$= \$\left(\frac{73}{100}\right) x\ 1392000 = \$\mathbf{1,016,160}$$

COMBINE

The unit was assumed to shut-down for 12 hours for each wash;

$$kwh = MW\ x\ 10^3\ x\ hours\ of shutdown\ interval$$

$$For\ one\ interval = 29\ x\ 10^3 x\ 12 = 348000kwh$$

$$For\ the\ two\ intervals = 696000kwh = 696x10^3 kwh$$

Total cost of loss power due to shut-down for offline water wash will be;

$$= \$\left(\frac{73}{100}\right) x\ 696000 = \$\mathbf{508,080}$$

This implies that \$1016160 and \$508080 were lost during shutdown for the compressor water wash process.

$$Net\ profit\ (\$) = Maximum\ revenue\ recovered - cost\ of\ washing \quad 7-17$$

- **Cost of wash kits**

Assuming the capital cost of online washing is \$156000 and the cost of manpower is assumed to be negligible [189]. Then the cost involved in carrying out online compressor washing would be as shown in table 7-1. It was assumed that there were 20 nozzles performing the water injection, and each nozzle was able to inject 7 litres of water, which gives a total of 140 litres of water.

The cost of wash fluid flowing per nozzle is $1.3 hence the total cost of fluid flowing through the 20 nozzles for the 10 minutes wash period is $182. The product of the cost of washing per wash and the number of times the washing is performed gives the cost of washing for the year. While the total cost of online washing is the accrue cost of washing and the capital cost.

The offline cost is the total worth of revenue that would have been gained during this period of shutdown. Each shutdown for offline wash takes approximately 12 hours. In a similar way, it was determined for the combine case and the total cost was accounted for as could be seen below.

ONLINE

Table 7-1: Cost of Online compressor wash

Capital cost of washing equipment per annum	156000
Quantity of fluid used per nozzle for a 10 minute wash (litres)	7
Number of nozzles in an online compressor washing setup	20
Total quantity of fluid required for a single online compressor wash (litres) **(Total quantity=Quantity of fluid*No. of nozzles)**	140
Cost of wash fluid flowing per nozzle	1.3
Total cost of fluid flowing through 20 nozzles for a 10 minutes wash ($)	182
Number of interval for 8760 hours in 7days frequency	52
Cost of wash fluid required for 8760 hours ($)	9464
Total cost of wash ($)	165464

The wash fluid used for the online washing is assumed to be demineralised water and the period of wash fluid injection for a wash interval or wash cycle is assumed to be 10 minutes.

$$Cost\ of\ online\ wash = cost\ of\ fluid * no\ of\ intervals \qquad 7-18$$

Cost of online wash=182*52=$**9,464**

$$Total\ cost\ of\ online\ wash = washing\ cost + equipment\ cost \qquad 7-19$$

Total cost of online wash = $156000+$9464=$**165,464**

OFFLINE

The recommended washing procedures include;

- Soak cycle which include water and detergent takes 20 minutes.
- Rinse cycle is water only and takes 15 minutes.
- Demineralised water price is assumed to be $200/ton and the detergent price was assumed to be $260/ drum [190].
- The dilution ratio is 1:6 for detergent to water.
- The solution flow rate is as recommended by the manufacturer to be 190 LPM (50 US gallons per minutes).

$$\text{Price of detergent} = \$\frac{260}{drum}\ x\ \frac{drum}{200l} = \$1.3/l$$

$$\text{Price of water} = \$\frac{260}{ton}\ x\ \frac{ton}{100l} = \$0.2/l$$

$$Cost\ of\ detergent\ consumed = time\ x\ detergent\ price\ x\ flow\ rate$$

$$= 20\ x\ 1.3\ x\ \left(190\ x(1/7)\right) = \$705.7$$

$$Cost\ of\ water\ consumed\ during\ soak\ cycle = time\ x\ water\ price\ x\ flow\ rate$$

$$= 20\ x\ 0.2\ x\ \left(190\ x\ (6/7)\right) = \$651.4$$

$$Cost\ of\ water\ consumed\ during\ rinse\ cycle = time\ x\ water\ price\ x\ flow\ rate$$

$$= 15\ x\ 0.2\ x\ 190 = \$570$$

For the four intervals:

$$Total\ cleaning\ material\ consumption\ cost = 4\ x\ (705.7 + 651.4 + 570)$$
$$= \$7708.4$$

COMBINATION

$$\text{Price of detergent} = £\$\ x\ \frac{drum}{200l} = \$1.3/l$$

$$\text{Price of water} = \$\frac{260}{ton}\ x\ \frac{ton}{100l} = \$0.2/l$$

$$Cost\ of\ detergent\ consumed = time\ x\ detergent\ price\ x\ flow\ rate$$

$$= 20\ x\ 1.3\ x\ \left(190\ x(1/7)\right) = \$705.7$$

$$Cost\ of\ water\ consumed\ during\ soak\ cycle = time\ x\ water\ price\ x\ flow\ rate$$

$$= 20\ x\ 0.2\ x\ \left(190\ x\ (6/7)\right) = \$651.4$$

$$Cost\ of\ water\ consumed\ during\ rinse\ cycle = time\ x\ water\ price\ x\ flow\ rate$$

$$= 15\ x\ 0.2\ x\ 190 = \$570$$

For the four intervals:

$$Total\ cleaning\ material\ consumption\ cost = 2\ x\ (705.7 + 651.4 + 570)$$
$$= \$3854.2$$

Total cost of combine online and offline wash=$3854.2+$9100=$12,954

Cost of fuel saved from engine shutdown

OFFLINE

$$Fuel\ saved = fuel\ flow\ x\ shutdown\ x\ fuel\ cost$$

$$Fuel\ flow = \frac{1.7745kg}{s}\ x\ 3600 = 6388kg/hr$$

$$Shutdown\ for\ four\ intervals = 4\ x\ 12 = 48hrs$$

$$Cost\ of\ fuel = \$1.329/kg$$

$$Fuel\ saved = 6388\ x\ 48\ x\ 1.329 = \$407516\ in\ 8394\ operating\ hours$$

COMBINE

$$Fuel\ saved = fuel\ flow\ x\ shutdown\ x\ fuel\ cost$$

$$Fuel\ flow = \frac{1.7483kg}{s}\ x\ 3600 = 6294kg/hr$$

$$Shutdown\ for\ two\ intervals = 2\ x\ 12 = 24hrs$$

$$Cost\ of\ fuel = \$1.329/kg$$

$$Fuel\ saved = 6294\ x\ 24\ x\ 1.329 = \$200749\ in\ 8538\ operating\ hours$$

The cost of fuel saved during engine shutdown for offline wash was about $407516, and $200749 for both cases respectively.

7.5.8 Creep cost estimation

The HPT blade life analysed in this research is the first stage along the section of the blade. The blade was assumed to be divided into four equal sections, and the creep life have been determined as explained earlier in section 5-10 with figure 5-12. The 50% section was observed to be the least and hence assumed to be the remaining life of the blade since it is the worst case. The creep life for the clean was observed to be a little above 31,230 thousand equivalent hours, and other cases were also determined as can be found in appendix H.

According to Jordan et al [191], for a set of blades for one stage in a modern gas turbine, a typical total cost, including manufacturing and replacement would be 10 MSEK (million Swedish Krona), which, when divided with the design lifetime of 40000 equivalent hours, results in a cost per equivalent hour of 250 SEK. The value SEK, is Swedish Krona, which is the currency used in Sweden. As such, converting this 250 SEK to US dollars amount to $29.5. Hence, each equivalent hours resulting from creep is assumed to be equal to $29.5, as determined and shown in appendix H.

The fouled case was also determined to be 12,403 equivalent hours, which gives a difference of 18,827 equivalent hours. This implies that revenue worth $555,396 would be lost from creep due to fouling.

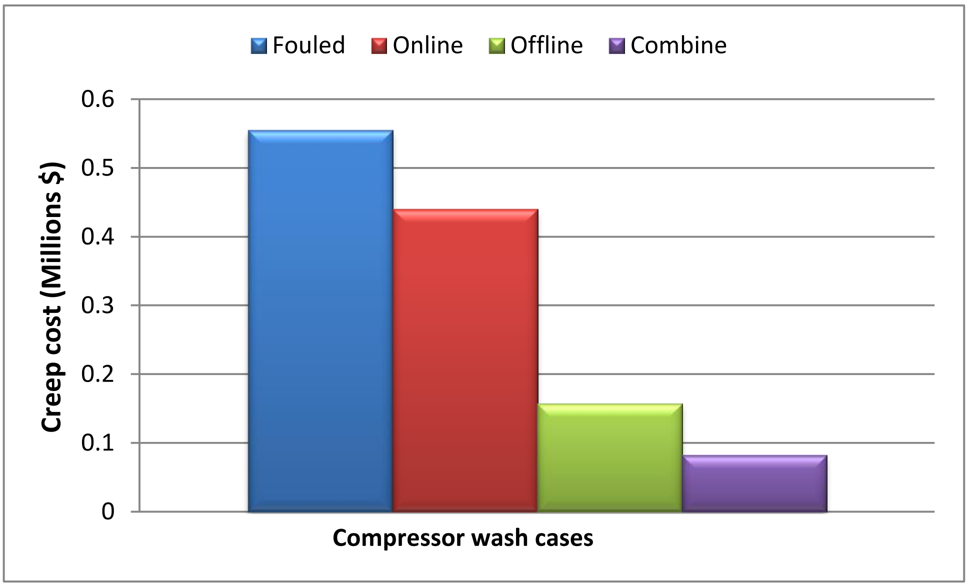


Figure 7-8: Revenue loss from HPT creep life due to fouling

It can be observed from figure 7-8 that the fouled has the highest revenue lost which is worth $555,396, and the combine case of compressor wash was seen to be far low ($82,412) when compared, this is because in this case the compressor washing was administered continuously online after each major offline wash, which prevented more heat from the HPT blade, as such greater life and reduced revenue loss.

7.6 Chapter summary

The financial involvement of each model from chapters four to six have been analysed in the economic model in this chapter. It can be observed that there is advantage in increasing TET to regain lost power from fouling due to the revenue of each kWh of energy sold. Also, the advantages of compressor washing have been shown indicating financial savings from each method of washing which is accompanied by its demerits as well. The financial implications of increased TET on the HPT blade due to life decrease and the implications resulting to emission to the environment have also been analysed due to its financial constrain from emission tax. The financial effects on creep have also been determined.

It can be deduced here that haven determined all revenue gained and loss from the various wash cases and the different models, the optimal cost need to be investigated so as to obtain the most viable economical point for compressor washing to be performed during the year. As such, it has become imperative to improvise a performance optimisation programme as could be seen in the next chapter.

8 PERFORMANCE OPTIMISATION

8.1 Introduction

Atmospheric pollutant and oil can cause fouling to the gas generator (compressor and turbine) rotor and stator blades during operation. Higher temperatures are required to maintain the needed power output, and such temperature has the capacity to reduce the creep life of the highly stressed hot components, which in turn leads to higher operating costs. Washing the compressor regularly with water can substantially reduce accumulation of deposits on the compressor blades [33], thereby extending creep life and reducing operating cost.

This chapter of the research aim at finding the optimum wash case for cleaning the compressor of an industrial gas turbine to enable the engine operates with minimal losses in the production processes. Different cases of compressor water wash have been applied in this research to restore performance and to determine the effect of washing on the rate of engine deterioration, cost of fuel, cost of electricity, loss in power output, and so on, with the help of an optimiser. An optimisation problem is one requiring the determination of an optimal value (maximum or minimum) of a given function called the *objective function*, subject to a set of stated restrictions, or *constraints*, placed on the variable concerned. In this research, we will need to maximise an objective function representing units of output in a gas turbine compressor washing situation, subject to constraints reflecting the performance, emissions, and creep life conditions of the gas turbine engine.

8.2 Linear optimisation methodology and assumptions

After realising the benefits of compressor washing, many gas turbine operators changed their opinion and start investigating the effect of the factors responsible for gas turbine performance degradation. These factors include: rate of fuel consumption, cleaning materials and its associated cost, time taken during off-line wash (shut-down), and the initial analysis is carried out as explained earlier in chapter three of this research. The idea is to use the output results obtained from turbomatch simulation code to find the relationship between these factors and the engine performance. Linear programming is a method of solving optimisation problem when the objective function is a linear function and the constraints are linear equations or inequalities, as can be seen in the following equations [192].

$$Minimise\ P = p_1x + p_2y + p_3z \qquad (objective\ function) \qquad 8-1$$

Subject to

$$\left.\begin{aligned} a_{11}x + a_{12}y + a_{13}z \le b_1 \\ a_{21}x + a_{22}y + a_{23}z \le b_2 \\ a_{31}x + a_{32}y + a_{33}z \le b_3 \end{aligned}\right\} \qquad (constraints) \qquad 8-2$$

$$x, y, z \ge 0$$

Introducing slack variables, we have

$$\left.\begin{aligned} a_{11}x + a_{12}y + a_{13}z + w_1 + 0 + 0 = b_1 \\ a_{21}x + a_{22}y + a_{23}z + 0 + w_2 + 0 = b_2 \\ a_{31}x + a_{32}y + a_{33}z + 0 + 0 + w_3 = b_3 \\ P - p_1x - p_2y - p_3z + 0 + 0 + 0 = 0 \end{aligned}\right\}$$

Where the objective function p_1x, p_2y, p_3z are the creep cost, emission cost, and performance cost accrue due to engine fouling respectively, and P is the optimum cost attained. While b_1, b_2, and b_3 are the total profits from online, offline, and combine wash respectively. The constraints (x, y, z) on the b_1 row are the cost from online wash for creep, emission and performance respectively, and a similar description is applicable to the row b_2 and row b_3 for the offline wash and combine wash respectively. Since there is now a total of n variables and m constraints, then at least (n-m) variables are equated to zero. The remainder form the basic variable column entries. Equating x, y, z to zero, then, the basic variables are w_1, w_2, w_3, as can be seen in table 8-1.

Table 8-1: Simplex method of three problem variables optimisation.

Basis	x	y	z	W1	W2	W3	b	Check
W1	a_{11}	a_{12}	a_{13}	1	0	0	b_1	
W2	a_{21}	a_{22}	a_{23}	0	1	0	b_2	
W3	a_{31}	a_{32}	a_{33}	0	0	1	b_3	
P	-p1	-p2	-p3	0	0	0	0	

The variables in the basis column are the variables heading the unity matrix. The procedures are as could be seen below;

i. Select the most negative entry in the index row to determine the *key column.*

ii. Divide the entries in the constant column (b) by the corresponding positive entries in the key column. The smallest positive ratio determines the *key row*.

iii. The entry at the intersection of the key column and the key row is the *key number* or *pivot*.

iv. Divide each entry in the key role by the pivot to reduce the key number to a *unit pivot*. The revised key row is called the *main row*.

v. Use the main row to operate on the remaining rows to reduce all other entries in the key column to zero.

vi. Repeat steps (I) to (V) until no negative entry remains in the index row.

In the application for this research, the rows represents the online, offline, and combine compressor wash respectively, while the column x, y, z represents the creep life cost, emissions, and performance for each wash respectively.

Inputting results from the economic model into equations 8-1 and 8-2, we have;

$$P = 555390_c + 4790876_e + 12100480_p \qquad Objective\ function$$

Subject to

$$\left.\begin{array}{r} 440120_c + 3840055_e + 165464_p \leq 66345907 \\ 152218_c + 3340389_e + 616352_p \leq 69102682 \\ 82413_c + 3177571_e + 320285_p \leq 70115392 \end{array}\right\} \qquad Constraints$$

It was observed that a maximum profit of $43,155,122 is attained. Details can be found in Appendix I.

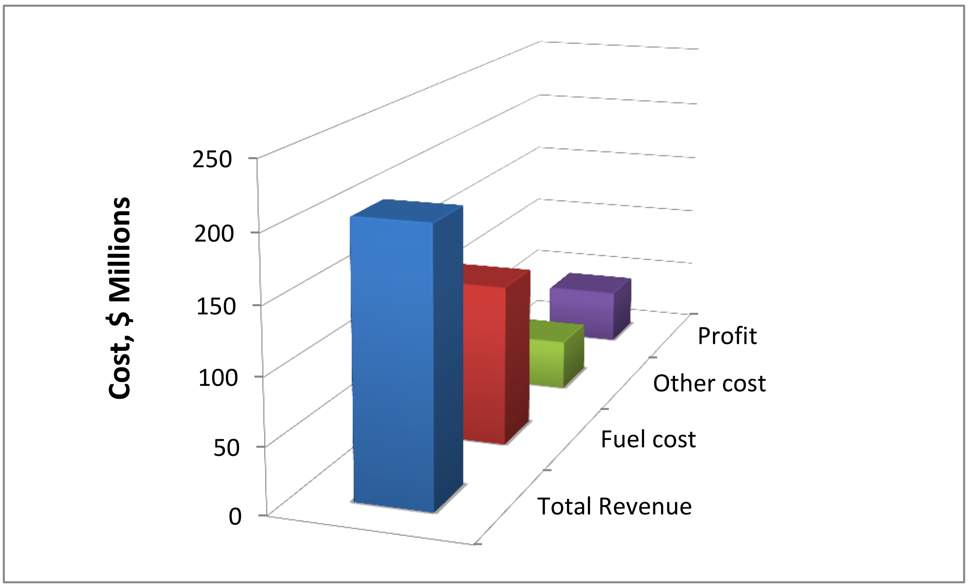


Figure 8-1: Total Revenue and Total cost for the applied cases

It can be observed that there is significant penalty in cost due to compressor fouling. Investigation for the three cases shows that the combine online and offline compressor wash yielded the highest profit, and from the optimisation application, the maximum profit was observed to be over $43M.

Figure 8-1 shows the various cost accrue from all cases due to fouling (emission tax cost, creep cost, fuel cost, and compressor washing cost, Total revenue from sale of power, and the optimum profit investigated.

8.3 Sensitivity Analysis

Sensitivity analysis examines how much a change in the value of a parameter of the plan will affect outcome. It is explored in terms of a development program yielding the necessary component for production sufficiently quickly without significant change in the output of the model or plan. Since prices of materials used for the compressor cleaning are not constant, fuel and electricity prices also varies, and rate of deterioration and shutdown time are also not fixed, the effect of each factor on the optimal compressor water wash interval will be shown by application of sensitivity analysis.

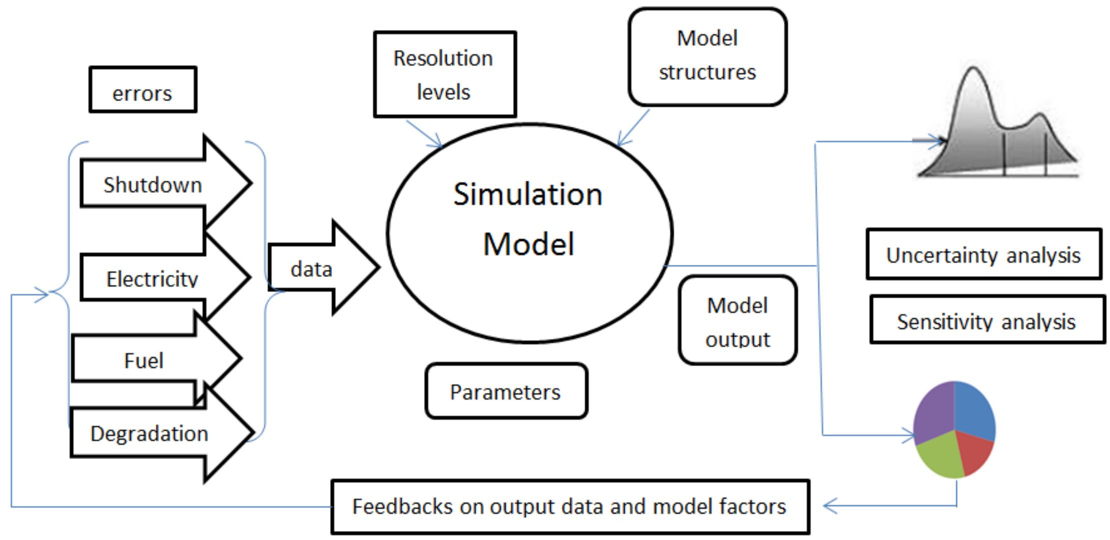


Figure 8.2: Model of Sensitivity Analysis

The figure 8.2 explains the relationship amongst the various factors. The different input variables, and their output sensitivities with various parameters and their rate of effects on the engine performance. The rate of sensitivity of each of the factors on the engine health is as explained below:

8.3.1 Effect of Changes in Shutdown Period

Increasing the number of shut-down hours reduces the net profit since each shut-down for compressor washing incur more cost. Shut-down hours were varied while other factors were kept constant, and the result is as shown in figure 8.3 below. The stipulated shut-down period could be longer than necessary for any of the under-listed reasons:

- The off-line water washing is repeated if the washing is not efficient.
- Technical or any other delay that may arise.
- Longer time for the turbine unit to cool down.
- Difficulty in starting the turbine unit.

Such delay is cost effective and hinders the financial benefits of operating a gas turbine. Shutting down for more than it stipulated period will mean the washing interval will become longer than required, as could be seen in the figure 8.3. The horizontal axis represent the operating hours, while the vertical axis represents the net profit in millions of dollars. It is obvious that too much delay in short-down time will result in loss to the engine operator. The large built-in marker line indicates the optimum interval for shut-down compressor water wash at the 12 hours wash duration.

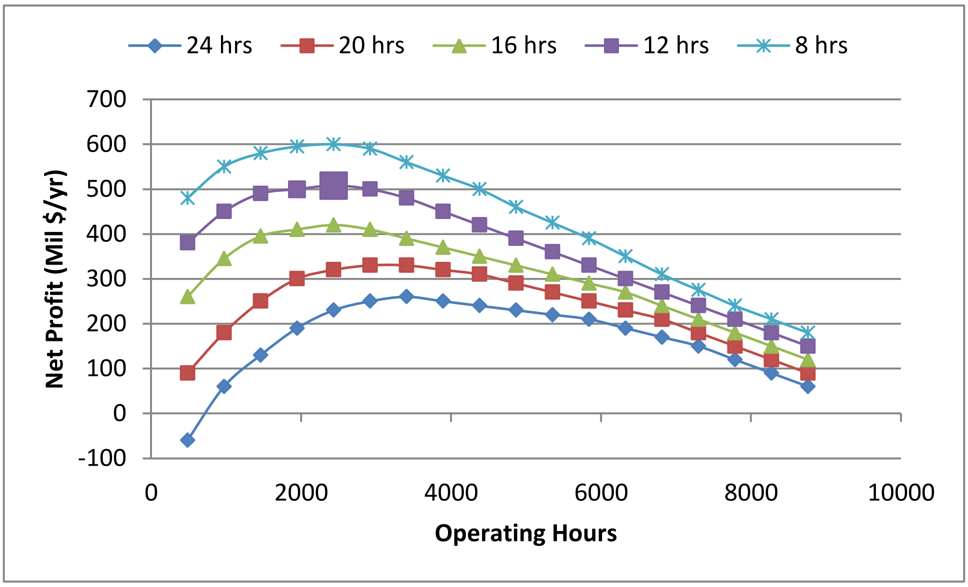


Figure 8.3: Effect of shutdown change on optimum wash interval

8.3.2 Effect of Changes in Rate of Degradation

The component water wash has a strong relationship with the rate of dirt deposition on compressor, as could be seen in figure 8.4. Increase in deposition rate, increases the number of washing intervals and decreases the net profit to minimal. The net profit increases at low rate of deposition, however, the net profit will increase and the parabolic curve will become straight as the deposition is closer to zero. In other word, increase in the deposition rate means shorter washing interval. The straighter the parabolic curve at low fouling rate, the increase there will be in profit. Where there is no fouling, washing is not required, as such, there is no optimum because the line will become straight like that of a clean engine since the fouling will be at 0%. In a similar way, the optimum interval for the degradation rate is at the 2% rate as indicated in figure 8.4 below.

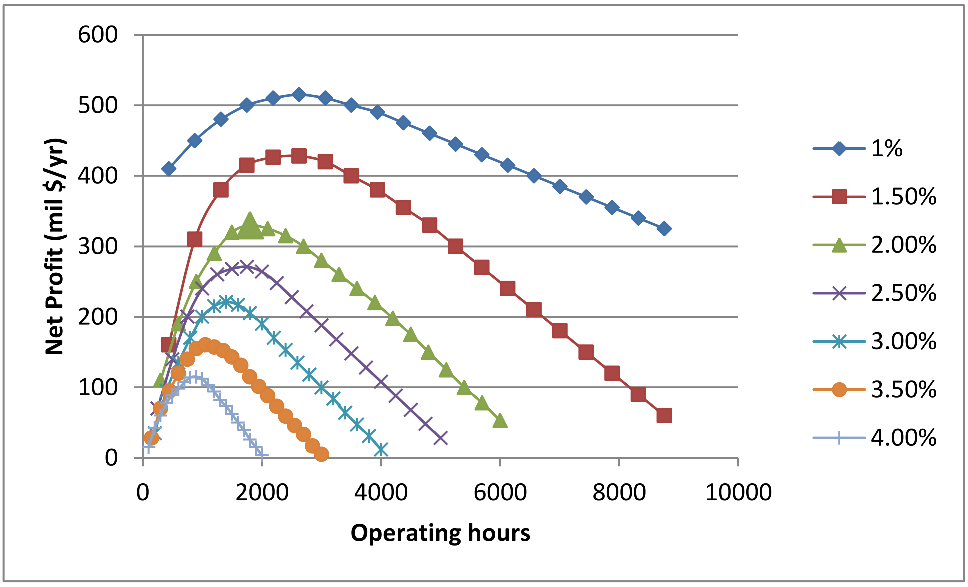


Figure 8.4: Effect of change in rate of degradation on optimum interval

8.3.3 Effect of Changes in Fuel Price on Optimal Wash

Increase in the price of fuel decreases the net profit and vice versa. The fuel price were varied while other factors were kept constant, it is obvious that increasing the number of wash intervals reduces the net profit which is inversely proportional to the change in fuel price. Change in fuel price has direct relationship with profit. Decrease in fuel price increases profit and shutting down for off-line water wash will be more expensive than consuming additional fuel. Keeping other losses constant, which will result in delays in washing when fuel is less expensive to give the optimum wash interval as could be seen in the figure 8.5. The optimum interval is indicated at $75/bbl, which shows that increase in fuel price during operation will result in reducing net profit.

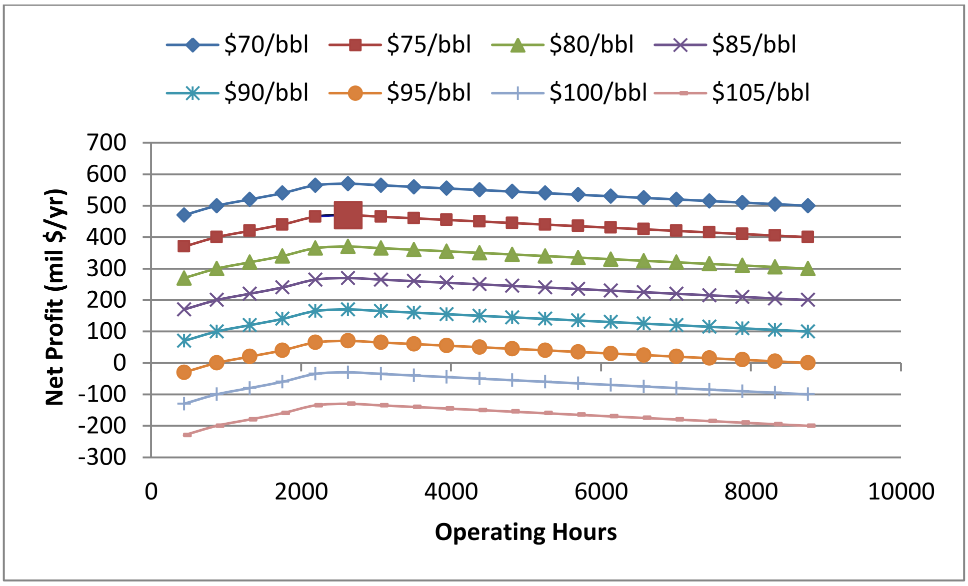


Figure 8.5: Effect of change in fuel price on optimal interval

8.3.4 Effect of Change in Electricity Prices

The net profit is proportional to the selling price of electricity. The selling price of electricity increase therefore forces the net profit to increase. The increments in the number of intervals will enable the net profit to increase until the optimum wash interval is attained and then reduces afterwards.

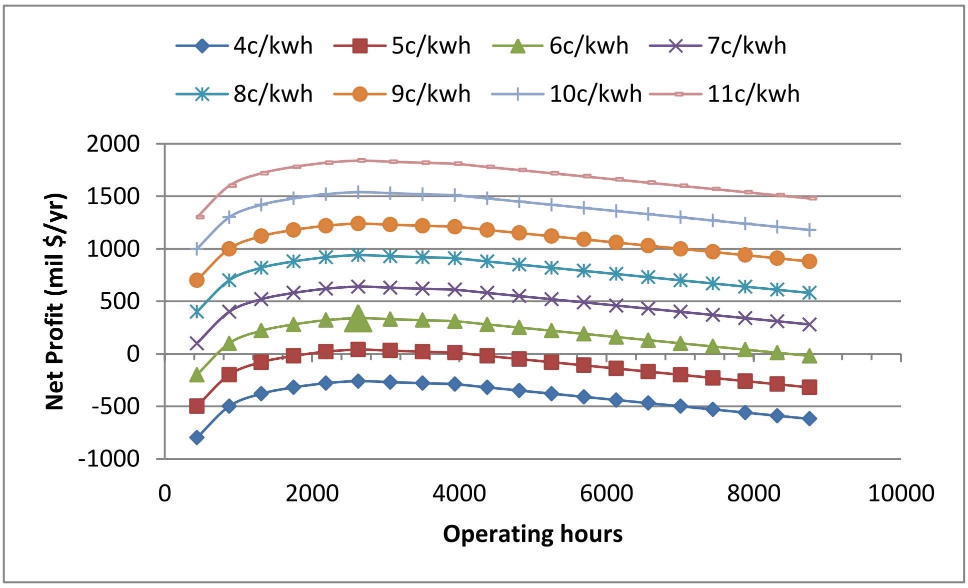


Figure 8.6: Effect of change in electricity price on optimal interval

The electricity price was varied, while other factors were fixed. The power loss due to degradation is costly, but shutting down resulting from off-line water wash will be more costly. The electricity price will not affect other cost for washing materials and fuel, and this will result in increased offline washing interval operating hours. Increase in electricity price increases optimum off-line washing interval to more operating hours and vice versa as could be seen in figure 8.6.

8.4 Chapter summary

The optimum wash interval is indicated at the point which gives the maximum net profit by analysing the variable factors with the net profit. This is indicated in figure 8.1. This is the point where the engine can run economically without damage to the equipment. The strong relationship between the numbers of wash intervals and the variable factors has great impact on the net profit, the variable factors affects the net profit more than the wash intervals, mostly rate of dirt deposits and cost of shut-down.

It can be summarised that the optimisation method is a very good approach for maximising profit and or minimising cost for gas turbine users. As such for the three wash cases applied in this research, it can be advised that the combine offline and online wash be utilised since it is the most economical of the three cases. This judgement is not universal, but based on the assumptions made in this research work, because in the situation where the gas turbine is always operated at constant TET, less fuel is used at the expense of reduced power. Also, if there is no compressor washing type applied to the gas turbine, then it will be advisable to operate the engine under constant power output. A lesser cost penalty is involved than constant TET, which is when the engine is operating at part load. This means the TET will be much less and has smaller effect on the turbine blade life. It is obvious that the gas turbine operating cost is very sensitive to the power setting and different ambient conditions, and the optimisation approach has proved that the operation plan can lead to a significant reduction in total cost.

9.0 CONCLUSION AND RECOMMENDATIONS

9.1 Conclusion

Compressor fouling and its possible solution (compressor water washing) have been in existence for some decades now to the engine operators. Proper selection for it application offers many technical and economic benefits to the end users. It has been presented that compressor fouling does not necessarily lead to total engine failure, but the continuous accumulation of dirt and deposits during engine operation reduces the efficiency and flow capacity of the compressor, as such affecting total output of the engine. Therefore cleaning the engine regularly, usually by water washing the compressor, is necessary to enhance production and profitability. The operating cost will adversely be affected if the compressor washing is carried out too often, due to increase in downtime for washing and unavailability of the engine in the case of offline washing. Similarly, if the washing is infrequent, there will be reduction in production and profit due to the decrease in engine performance resulting from fouling.

The possible mechanism responsible for gas turbine degradation have been stated and discussed in the body of this thesis, showing the different possible physical reasons. The design point was initially simulated successfully with it results very promising and close to the actual characteristics of the engine with error of about 0.01% from Turbomatch software application since the actual map could not be found on public domain for the sensitivity of commercial reasons. There was a reduction in flow capacity and isentropic efficiency, and power output resulting from compressor fouling due to deposits accumulation on the surface of the compressor blades. The application of Turbomatch and Pythia software which is in-house software available in school of engineering from Cranfield University, for the simulation of gas turbine performance, was used to predict the effect of compressor degradation on overall performance of the engine. The implementation was made both for design and off-design point as discussed earlier in chapter four. The off-design was carried out with varying ambient temperature so as to show the effect of increase in TET on the turbine blades, consequently estimating the creep life of the blades.

There is usually time dependent deformation and temperature-induced changes in the microstructure of materials meant for service at very high temperature. Rene 80 is a Nickel based super alloy that represents a class of material meant for application at high temperature such as that faced by turbine blades. Creep is the main type of damage observed for turbine blades that are exposed to high temperature under varying stress condition for a protracted period of time. The creep in this thesis was calculated with the application of Larson-Miller Parameter (LMP), to determine the creep life of the turbine blades, as discussed in chapter five in this thesis. The LMP chart for the particular above stated material was found in public domain, and the calculation shows that the creep life was determined for various sections of the HPT blade and it was observed that though the temperature was highest at the 75% section, the creep life was minimal at the 50% section. This indicates that creep is not only dependent on temperature but also on the stress at the blade sections.

The emissions from the gas turbine was determine by the application of mathematical correlations employed by Arthur H. Lefebvre [120], and was used to verify the effectiveness of compressor water wash on the rate of reduction of emission pollution. It was observed that the primary zone temperature has high impact on gas turbine NOx emission, and there was increase in primary zone temperature of about 10.2%, which increases NOx by about 90%, and it was also noted that the highest amount of NOx emitted was within standard which does not exceed 25ppmv. In a similar way, the CO emission was determined as could be found in equation 6-16. It was observed that an increase in primary zone temperature of about 10.2% decreases the CO by about 44%. This research shows that 1.57% increase in temperature decreases CO by 7.18%. All these have financial implications to the engine operator; as such there is need for an economic model to implement the financial benefit of the compressor wash frequency that will be more beneficial to the engine operator. Hence, the need to optimise the frequency of compressor washes such that the loss in profit or revenue due to fouling and washing is minimised. The revenue is directly affected by the engine downtime for compressor washing, and the total cost resulting from compressor washing can be determined by summing the cost of wash and the cost of lost revenue. The lost revenue per annum is determined by multiplying the lost revenue per engine wash by the number of engine washes per annum, and will increase with wash frequency.

There is also reduction in revenue due to compressor fouling resulting from engine degradation. The compressor fault index is used to determine this process due to fouling alongside the model of the engine. The engine model is used to determine the loss in maximum available power and the loss in thermal efficiency at various times during the fouling. The maximum power available from the engine is determined by running the model at some limiting condition such as the exhaust gas temperature or speed limit imposed by the manufacturer. The fouling index profile is determined by monitoring the compressor performance degradation resulting from fouling and the revenue lost resulting from fouling can be used to extrapolate the revenue lost per annum resulting from fouling. Infrequent washing will lead to increase in revenue caused by fouling and the addition of these two sources of revenue gives the total revenue. The optimised wash occurs when the total cost is at minimum and net profit at maximum.

The following factors influences total cost; rate of production, electricity unit price, fuel cost, emission tax, creep cost, downtime and washing cost. This means, increase in unit price of electricity, fuel cost and emission tax will tend to increase the wash frequency, while increase in downtime and costs associated with the compressor wash activity will tend to reduce the wash frequency. The wash frequency is also affected by the power demand. Higher power demand such as where the engine need to operate close or at the engine operating limit such as the EGT, will tend to increase the wash frequency. This process is most applicable to online compressor washing systems. It is obtained by generating the fouling profile of the compressor resulting from online washing and it is observed that offline washing is more efficient than online washing, where there are small offset in the fault indices after each online wash. This offset grows progressively larger after each wash due to residual fouling, and there is no downtime associated with it, and the availability is also high. This shows that frequency of online washing is also high. When the engine performance loss due to residual fouling is greater than the performance improvement due to offline wash, then this will be an indication as to when an offline wash will be beneficial.

The creep life when converted to creep cost based on cost per equivalent creep life hours for the fouled case shows a loss in revenue of about $555,390, but the compressor combine wash case shows revenue loss of about $82,412. This shows a wide difference in cost savings from compressor washing of about $472, 977.

In a similar way, the emission tax cost for the fouled case was about $3,840,055, while the emission loss cost was reduced due to compressor washing for the combine wash case to about $ 3,177,571, leaving a difference of about $662,484. The iteration was also done for other wash cases as well, and the combination of these with performance based on sale of power output as electricity were all incorporated into the optimisation model to yield the most optimum profit and advise on the amount of creep, emission, and performance required to arrive at this optimum point.

A computational model have been improvise and presented in this research for the maximisation of total profit and minimisation of total cost in operating gas turbine engine. The changes in ambient and operating conditions over a year period of the gas turbine were considered. The developed model minimises the total cost due to engine fouling resulting from creep, emission, and performance (power sold as electricity), and this was linked with cost accrue for each of these models based on the online, offline, the combine online and offline compressor wash cases. The total loss cost due to fouling was assumed to be the objective function, while the three wash cases for the creep, emission and performance were taken to be the constraints. A minimisation approach was applied to the total loss cost, and the recovery from the various compressor wash cases shows a whooping drop in the total loss cost and increase in the revenue recovered.

The sensitivity analysis shows that from the simulation of the compressor degradation used to find the optimum interval of compressor water wash, the net profit is affected by the change in fuel price, electricity price, rate of engine degradation, and shut-down time. It was confirmed from the analysis that the net profit is very responsive at high numbers of washing interval due to increase in expenses from shut-down. The maximum net profit indicates the optimum water wash, and the revenue loss is at minimum as discussed in chapter eight of this research.

9.2 Recommendations

It can be clearly seen from the current work that there is brilliant basis to suggest future work to augment the present study. These recommendations include;

- Incorporation of inlet filtration systems.
- Investigating the use of different types of fuel and economic benefits of using heavy fuel, hydrogen or bio-diesel fuel.
- Investigate heat production of the engine as a source of energy in addition to the electrical or mechanical energy.
- Investigating details on how failure mechanisms like corrosion, fatigue, oxidation, etc can be simulated and incorporated in the model.
- Investigate the use of different turbine blade material.

REFERENCES

[1] Chen, H. A., Multiscale Forecasting Methodology for Power Plant Fleet Management. Aerospace Engineering. Georgia Institute of Technology; 2005.

[2] Mohamed, W., Techno-Economic, Environmental and Risk Analysis (TERA) for Power Generation. Cranfield University; 2013.

[3] Abu, A. O., Integrated Approach for Stress Based Lifing of Aero Gas Turbine Blades. Cranfield University; 2014.

[4] Abdul Ghafir, M. F., Performance Based Creep Life Estimation for Gas Turbine Application. PhD Thesis, Cranfield University; 2011.

[5] Abdul Ghafir, M. F., Li, Y. G., Sing, R., Huang, k., and Feng, X., Impact of Operation and Health Conditions on Aero Gas Turbine hot section Creep Life Factor Approach. ASME, Turbo Expo, Glasgow; June 14-18, 2010.

[6] Eshati, S., Abu, A., Laskaridis, P., and Haslam, A., Investigation into the Effects of Operating Condition and Design Parameters on the Creep Life of High Pressure Turbine Blades in a Stationary Gas Turbine. 10th International Symposium on Compressor and Turbine Flow Systems-Theory and Application Areas Lodz; 26-28 October, 2011.

[7] Eshati, S., Abu, A., Laskaridis, P., and Haslam, A., Impact of Operating Conditions and Design Parameters on Gas Turbine hot section Creep Life. ASME, Turbo Expo, Glasgow; June 14-18, 2010.

[8] Wanis, M., Eshati, S., Pilidis, P., Ogaji, S., and Laskaridis, P., A Method to Evaluate the Impact of Power Demand on HPT Blade Creep Life. Procedings of ASME Turbo Expo, Power for Land, Sea and Air, Vancouver, Canada; 2011.

[9] Palmer., The TURBOMATCH Scheme for Gas-Turbine Performance Calculation (Unpublished User's Guid), Cranfield University. Cranfield, UK; 1999.

[10] Vassilious, P., Gas Turbine Performance Simulation and Diagnostics (Unpublished TURBOMATCH course note), Cranfield University; 2011.

[11] Ogaji, S. O. T., Pilidis, P., and Hales, R., TERA- A Tool for Aero-engine Modelling and Management. second world congress on Engineering Assest Management and the Fourth International Conference on Condition Monitoring, Harrogate, UK; 11-14 June, 2007.

[12] Khatib, H., Economic Evaluation of Projects in the Electricity Supply Industry. The Institute of Engineering and Technology, London, UK; 2008.

[13] Marsh, D. W., Economics of Electricity Utility Power Generation. Oxford University, UK; 1980.

[14] A, P., and G, M., The Future Role of Fossil Power Generation. E50001-G220-A137-X-4A00, Siemens AG Energy sector, Erlangen, Germany. 2011.

[15] Carniere, H., Willcox, A., Dick, E., Paepe, M. De., Raising Cycle Efficiency by Inter-cooling in Air Cooled Gas Turbine, Applied Thermal Engineering. 26 (16) 1780-1787. 2006.

[16] Nishada, K., Takagi, T., Kinoshita, S., Regenerative Steam Injection Gas Turbine system. Applied Energy 81, 231-246. 2005.

[17] Diankunchak, I.S., Performance Deterioration in Industrial Gas Turbine. Journal of Engineering for gas turbine and power, Vol. 114, No. 2, pp. 161-168, 1992.

[18] Meher-Homji, C. B., Chaker, M. A., and Motiwala, H. M., Gas Turbine Performance Deterioration. Proceedings of the 30th Turbomachinery Symposium, Turbomachinery Laboratory, Texas A & M University, College Station, Texas, pg.139-176, 2001.

[19] Lakshminarasimha, A. N., Boyce, M.P., and Meher-Homji, C.B., Modelling and Analysis of Gas Turbine Performance Deterioration. Journal of Engineering for gas turbines and power, Vol. 116, No. 1, pp. 46-52, 1994.

[20] Hamed, A., Tabakoff, W., and Singh, D., Modelling of Compressor Performance Deterioration due to Erosion. International Journal of Rotating Machinery, Vol. 4 Issue 4, pp. 243-248, 1998.

[21] Tabakoff, W., Lakshminarasimha, N., and Pasin, M., Simulation of Compressor Performance Deterioration due to Erosion. Journal of Turbomachinery 112 (1) pp 78-83, 1990.

[22] Merrington, G., Kwon, O.K., Goodwin, G., and Carlsson, B., Fault Detection and Diagnosis in Gas Turbines. Journal of Engineering for gas turbines and power, Vol. 113, No. 2, pp. 276-282, 1991.

[23] Ashley, D. S., Sarim, A. Z., Gas Turbine Performance at Varying Ambient Temperature. Applied Thermal Engineering, 31, 2735-2739. 2011.

[24] Basrawi, F., Yomada, T., Nakanishi, K., Naing, S., Effect of Ambient Temperature on the Performance of Micro Gas Turbine with Cogeneration

System in Cold Region. Applied Thermal Engineering 31 (6-7) 1058-1067. 2011.

[25] Naikan, V. N. A., Rao, P. N. S., An Optimal Maintenance Policy for Gas Turbine Power Plant. Journal of Engineering for Gas turbines and Power vol.130, 021810-1-5. 2008.

[26] Wilcox, M., Baldwin, R., Garcia-Hernandez., Brun, K., Guideline for Gas Turbine Inlet Air Filtration Systems. Gas machinery research council and southwest research institute. 2010.

[27] Song, T.W., Sohn, J.L., kim, T.S., Ro, S.T., An Analytical Approach to Predicting Particle Deposit Fouling in the Axial Compressor of the Industrial Gas Turbine. Proceedings of the institution of Mechanical Engineers, part A: A journal of power and energy, 219 pp. 203-212. 2005.

[28] Gulen, S.C., Griffin, P.R., Paolucci, S., Real-time, On-line Performance Monitoring of two Heavy Duty Industrial Gas Turbine. Presented at the international gas turbine and aero-engine congress and exhibition, Munich, Germany, May 8-11, 2000.

[29] Aker, G. F., Saravanamuttoo, H. I. H., Predicting Gas Turbine Performance Degradation due to Compressor Fouling using Computer Simulation Techniques. Trans ASME Journal of Engineering for gas turbines and power. 1994.

[30] Grewal, M.S., Gas Turbine Engine Performance Deterioration Modelling and Analysis. Ph.D. Thesis ed. Cranfield University; 1988.

[31] Basendwah, A.A., Pilidis, P., Li, I.Y., Turbine Off-line Water Wash Optimization Approach for Power Generation. GT2006-90244, Barcelona-Spain, 2006

[32] Meher-Homji, C.B., Compressor and Hot Section Fouling in Gas Turbine-Causes and Effects. Proceedings from the Ninth annual industrial energy technology conference, pp. 261-269, 1987.

[33] Diakunchak, I. S., Performance Deterioration in Industrial Gas Turbine. Journal of engineering for gas turbine and power, 114, pp. 161-168, 1992.

[34] Scheper, G.W., Mayoral, A.J., and Hipp, E.J., Maintaining Gas Turbine Compressors for High Efficiency. Power engineering, 82 (8). Pp. 54-57, 1978.

[35] Dyson, R.J.E., and Doel, D.L., CF6-80 Condition Monitoring- The Engine Manufacturer's Involvement in Data Acquisition and Analysis. AIAA-1984-1412, SAE, and ASME, Joint Propulsion Conference, 20th, Cincinnati, OH, June 11-13, 1984.

[36] Roemer, M.J., Testing of a Real-time Health Monitoring and Diagnostic System for Gas Turbine Engine. AIAA-1998-3603, AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, 34th, Cleveland, OH, July 13-15, 1998.

[37] Escher, P.C., PYTHIA: An Object-oriented Gas Turbine Gas Path Analysis Computer Programme for General Application. PhD thesis, SoE-Cranfield University-UK, 1995.

[38] Pillidis, P., and Palmer., Gas Turbine Theory and Performance. Msc Thermal power, course lecture note. 2010.

[39] Zwebek, A., One Giga Watt Single Shaft Industrial Gas Turbine Design. MSc Thesis. Cranfield University, UK. 1993.

[40] Lakshminarasimha, A. N., Boyce, M. P., and Meher-Homji, C. B., Modelling and Analysis of Gas Turbine Performance Deterioration. Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol. 116, No. 1, pp. 46-52. 1994

[41] Kurz, R., and Brun, K. Degradation in Gas Turbine Systems. Journal of Engineering for Gas Turbine and Power, Vol. 123, No.1, pp. 70-77. 2001

[42] Howell, A. R., and Calvert, W. J. A., New Stage Stacking Technique for Axial Flow Compressor Performance Prediction. 100:698-703. 1978.

[43] Aker, G., and Saravanamuttoo, H., Predicting Gas Turbine Performance Degradation due to Compressor Fouling using Computer Simulation Techniques. ASME Gas Turbine and Aero-engine Congress, ASME Paper No. 88-GT-206. 1988.

[44] Gulen, S., Griffin, P., and Paolucci, S., Real-Time Online Performance Diagnostics of Heavy-duty Industrial Gas Turbines. Journal of Engineering for Gas Turbines and Power. Vol, 124, No. 4, pp. 910-921. 2002

[45] Syverud, E., Aexial Compressor Performance Degradation and Recovery through Online Washing. 2007.

[46] Sanchez, D., Charcartegui, R., Becerra, J. A., Sanchez,T., Determining Compressor Wash Program for Fouled Gas Turbines. Proceedings of the institution of mechanical engineers, part A: A journal of power and energy, 219 pp 462-476, 2009.

[47] Meher-Homji, C. B., and Bromley, A., Gas Turbine Axial Compressor Fouling and Washing. Proceedings of the thirty-third turbomachinery symposium. 2004.

[48] Saravanamuttoo, H. I. H., A Preliminary Assessment of Compressor Fouling Methods. AGARI-LS-183, 2002.

[49] Sampath, S., Fault Diagnostics for Advanced Cycle Marine Gas Turbine using Generic Algorithm. School of engineering. Cranfield University, United Kingdom. 2003.

[50] Munde, F. C., Pillidis, P., Gas Turbine Compressor Washing: Historical Developments, Trends and main design parameters by online systems. Journal of engineering for gas turbine and power, Vol. 128 pp 344-353. 2006.

[51] Mattingly, J. D., Elements of Gas Turbine Propulsion. McGraw-Hill, Inc. 1996

[52] Fuks, N. A., The Mechanism of Aerosols. Moscow, USSR Academy of Science. 1955.

[53] Tarabrin, A. P., et al. An Analysis of Compressor Fouling and a Cleaning Methods of their Blades. ASME Paper No. 96-GT-363. 1996.

[54] Carazas, F. J. G. and Souza, G. F.M. D., Availability Analysis of Gas Turbine used in Power Plants. Int. J. of thermodynamics, Vol. 12 (1): p. 28-37. 2009.

[55] Boyce, M. P., Gas Turbine Engineering Handbook. 2nd ed. Houston: Gulf Professional publishing. 2002.

[56] Carlstrom, L.A. et al., The Operation and Maintenance of General Electric's Gas Turbines. Numerous maintenance articles/authors reprinted from power engineering magazines, General Electric's publications, GER 3148 December, 1978.

[57] Vesier, C., Understanding Reliability, Availability and Maintainability. Ronamax. 2004.

[58] Li, Y.G., Gas Turbine Diagnostics. Thermal Power MSc course note, Cranfield University: Cranfield, United Kingdom. 2011.

[59] Klaus, Brun., and Rainer, Kurz., Turbomachinery International Journal for Energy Equipment, Vol. 48, page 32, May/June, Norwalk, Connecticut, USA. 2007.

[60] Singh, R., Gas Turbine Application. Lecture note, school of engineering, Cranfield University, United Kingdom. 2011.

[61] Urban, L. A., Parameter Selection for Multiple Fault Diagnostics of Gas Turbine Engines. ASME paper 74-GT-62.

[62] Barringer, H. P., Availability, Reliability, Maintainability and Capability. Tripex chapter of the vibrations institute. Beaumont, Texas. 1997.

[63] Fluke., The Basics of Predictive/Preventive Maintenance”. Fluke corporation: Everett, WA, USA. 2005.

[64] Hoeft, R., Janawitz, J., and Keck, R., Heavy Duty Gas Turbine Operating and Maintenance Considerations. GE gas turbine library- GER 3620J. GE Power systems: Atlanta, GA USA. 2003.

[65] Saravanamuttoo, H. I. H., and Lakshminarasimba, A., Preliminary Assessment of Compressor Fouling. Turbomachinery International. Vol. 26, No. 7, pp. 14-18. 1993

[66] Langford, J. L. F., Contamination Removal Method. Patent 4, 0654, 322, USA. 1977.

[67] Ramsden, K. W., Axial Compressor Design and Performance. MSc Lecture note, Cranfield University, UK. 2011.

[68] Olhovsky, G. G., Power Gas Turbine Units. Moscow, Energy atomization, 303p. 1990.

[69] Brooks, F. J., GE Gas Turbine Performance Characteristics, GE Report. 2000.

[70] Upton., A. W. J., Axial Compressor and Turbine Blade Fouling. Some Cause, Effects and Cleaning Methods. Gas Turbine Operation and Maintenance Symposium edition, National Research Council, Canada. 1974.

[71] Diakunchak, I. S., Performance Deterioration in Industrial Gas Turbines. ASME International Gas Turbine and Aero-engine Conference, Orlando, Florida, ASME Paper No. 82-GT-66.1991.

[72] Mustafa, Z., Analysis of Droplets in Compressor Gas Turbines. PhD Thesis, Cranfield University, UK. 2006.

[73] Sire, and Stalder, J. P., Salt Perculation through Gas Turbine air Filtration Systems and its Contribution to total contaminants Level. Proceedings of the International Joint Power Generation Conference, Vol. 2, pp. 445-446. 2001

[74] Zwebek, A., Combine Cycle Performance Deterioration Analysis. PhD Thesis, Cranfield University, UK, 2002.

[75] Osborn, R., Design of an Industrial Gas Turbine Intake. American Air Filter Company. 1977.

[76] Kolkman, H., Performance of Gas Turbine Cpmpressor Cleaners. Journal of Engineering for Gas Turbines and Power. Transactions of ASME, Vol. 115, No3, pp. 674-677. 1993.

[77] Vigueras, Zuniga, M. O., Technoeconomic Analysis of Compressor Washing. MSc Thesis, Cranfield University, UK. 2003.

[78] Yee, R., and Myers, L., Enhanced tf40 Gas Turbine Design chnages to improve resistance to the Landing Craft Air Cushion (LCAC) operational enviroment, Naval Surface Warfare Center Carderock Division, C. S. C. A. M. C. USA, Vol. 3 at Atlanta Georgia USA; ASME USA, pp. 495-499.

[79] Mund, F., and Pilidis, P., A Review of Gas Turbine Online Washing Systems. Turbo Expo 2004.

[80] Syverud, E., and Bakken, L., Online Water Wash Test of GE J85-13, ASME Turbo Expo 2005.

[81] Caguiat, D., Zipkin, D. M., and Patterson, J., Compressor Fouling Testing on Rolls Royce/Allison 501-k17 and general electric LM2500 gas turbine engines, ASME Turbo Expo Vol. 2, No. B, pp. 933-942. 2002.

[82] Brumbaugh, D., Inlet Air Filteration Adapts to Evolving Gas Turbine Technology. Power Engineering Vol. 106, No. 10, pp. 51-54. 2002.

[83] Giampolo, T., The Gas Turbine Handbook. The fairmont Press, USA. 1997.

[84] Mathioudakis, K., and Tsalavoutas, T., Uncertainty Reduction in Gas Turbine Performance Diagnostics by Accounting for Humidity effects. Journal of Engineering for Gas Turbine and Power. Vol. 124, pp. 801-808. 2002.

[85] Thames, J., Stegmaier, J., and Ford, J. J.J., On-line Compressor Washing Practices and Benefits. ASME, June 4-8, Toronto, Ontario, Canada, pp.6. 1989.

[86] Fielder, J., Evaluation of Zero Compressor Wash Routine in RN service. ASME Turbo Expo, Vol. 3, pp. 543-547. 2003.

[87] Bagshaw, K. W., Maintaining Cleanliness in Axial Compressor. National Research Council edition, Canada. 1974.

[88] Ogbonnaya, E. A., Gas Turbine Performance Optimization using Compressor Online Water Washing Technique. 2011.

[89] Peltier, R. V., Swanekamp R. C., LM2500 Recoverable and non-recoverable loss. ASME 95-CTP-104. 2003

[90] Meher-Homji, C.B., Compressor and hot section fouling in gas turbine-causes and effects. Proceedings from the Ninth annual industrial energy technology conference, pp. 21-269, 1987.

[91] Marinai, L., Probert D., and Singh R., Prospects for aero gas-turbine diagnostics: A review, Applied Energy, Vol. 79, pp. 109-126, 2004

[92] Baker, J. D., Analysis of the sensitivity of multi-stage axial compressor fouling at various stages. Msc Thesis (Naval Post Graduate School, United State Navy). 1991

[93] GE Energy., Fundamentals of gas turbine construction and maintenance. Greenville South Carolina. 1997.

[94] Kurz, R., Brun, K., "Gas turbine tutorial-maintenance and operating practices effect on degradation and life. Urbolab.tamu.edu/uploads/files/papers/t36/T36-TUT05.pdf- United States. 2008.

[95] Zainul, A. M., GT Compressor water wash practices and effectiveness for TNB's combined cycle plants. Available at www.scribd.com/doc/27815265/Compressor-washing-performance(accessed 23rd November 2010.

[96] Boyce, M. P., Gonzalec, F., "A Study of on-line and off-line turbine washing to optimize the operation of gas turbine". ASME Transaction vol.129, 114-122. 2001.

[97] Wei, R. P. Y., and Gangloff, R. P., Fracture Mechanics: Perspectives and Directions. Twentieth Symposium, Vol 1. Philadelphia: ASTM International, 1989.

[98] Boutarek, N., Saidi, D., Acheheb, A., Iggui, M., and Bouterfaia, S., Competition between three damaging mechanisms in the fractured surface of an inconel 713 superalloy, Material characterisation, Vol. 59, No. 7, pp. 951-956, 2008.

[99] Naeem, M., Jazayeri, S. A., and Razamahdi, N., Failure Analysis of Gas Turbine Blades. IJAC-IJME International Conference, Nashville, Tenesse, 2008.

[100] Evans, R. W ., and Wilshire, B., Introduction to Creep, London: IOM Commmunication Ltd, 1999.

[101] Kassner, M. E., and Peres-Prado., Fundamental of Creep in Metals and Alloys. Oxford: Elsevier Science Ltd, 2004

[102] Webster, G. A., and Ainsworth, R. A., High Temperature Component Life Assessment, Vol. 1. London: Chapman & Hall, 1994.

[103] Parthasarathy, G., et al., Neural Network Models for Usage Based Remaining Life Computation. Journal of Engineering for Gas Turbine and Power, Vol. 130, 2008.

[104] Weber, B., Huitang, J., Pistor, R., and Lowden, P., Application of an Integrated Engineering Approach for LM1600 Blade Life On-linw Assessement. 05-IAGT-2.2, IN 16TH Symposium on Industrial Application of Gas Turbine, Alberta, Canada, 2005

[105] Carter, T. J., Common Failures in Gas Turbine Blades. Engineering Failure Analysis, Vol. 12, No. 2, pp. 237-247, 2005.

[106] Li, J., and Dasgupta, A., Failure Mechanism Models for Creep and Creep Rupture. IEEE Transactions on Reliability, Vol. 42, No. 3, pp. 339-353, Sept. 1993

[107] Tinga, T., De Wolf, W. B., Visser, W. P. J., and Woldendorp, S., Integrated Lifing Analysis of a Film-Cooled Turbine Blade. RTO-AVT Symposium on Monitoring and Manaement of Gas Turbine Fleets for Extended Life and Reduced Costs NATO Research and Technology, Manchester, UK. 2001.

[108] Betten, J., Sklepus, S., and Zolochevsky, A., A Microcrack Description of Creep Damage in Crystalline Solids with Different Behaviour in Tension and Compression. International Journal of Damage Mechanics, vol. 8, No. 3, pp.197-232. 1999.

[109] Cane, B. J., and Aplin, P. F., Creep Life Assessement Methods. Journal of Strain Analysis, Vol. 29, No. 3, pp. 225-232. 1994.

[110] Chaboche, J. L., Continuum Damage Mechanics: Present State and Future Trends. Nuclear Engineering and Design, Vol. 105, no. 1. 1993.

[111] Arebi, A. W. A., The Benefits of Compressor Cleaning on Power Output for Oil and Gas Field Application. (MSc Thesis), Cranfield University, UK. 2010.

[112] Rainer, K., Klaus, B., Meher-homji, C. B., Gas Turbine Degradation. 43rd Turbomachinery & 30th Pump Users symposia (Pump & Turbo), September 23-25, 2014.

[113] United Nations., The Kyoto Protocol. 1998.

[114] Stern, N., Stern Review: The Economics of Climate Change. Her majesty’s Treasury. 1999.

[115] Lefebvre, A. H., Dilip, R. B., Gas Turbine Combustion: Alternative Fuels and Emissions. 3rd edition, Taylor and Francis, London. 2010

[116] Goldberg, D. E., and Deb, K., A Comparative Analysis of Selection Scheme used in Genetic Algorithms. In: Foundations of Genetic Algorithms, Morgan Kauffmann Publishers, San Mateo, CA, 2000

[117] Singh, R., Gas Turbine Applications. (Unpublished course notes), Department of Power and Propulsion, Cranfield University, UK. 2011.

[118] Rizk, N. K., and Mongia, H. C., “Emissions Prediction of Different Gas Turbine Combustors”, AIAA, Vol. 94-0118. 1998

[119] Cookson., and Haslam, A. S., Mechanical Design of Turbo-machinery. (unpublished MSc lecture notes), Cranfield University. 2011.

[120] Lefebvre, A.H., Fuel Effect on Gas Turbine Combustor-Liner Temperature, Pattern Factor, and Pollutant Emissions. Journal of Aircraft, Vol. 21, no. 11, pp. 887-898. 1984.

[121] Quintero, R., Tcehno-economic and Environmental Risk Assessment of Innovative Systems FOR Short-Range Civil. (PhD Thesis), Schools of Engineering, Department of Power and Propulsion, Cranfield University, UK. 2003.

[122] Pavri, R., and Moore, D. D., Gas Turbine Emissions and Control. Atlanta, US: GE Power Systems, GER-4211. 2011

[123] US Department of Energy (DOE)., Fuel and Energy Source Codes and EMISSION Coefficients. Voluntary Reporting of Greenhouse Gases Program. Energy Information Administration (EIA). Retrieved (2008-04-15), available at http://www.eia.doe.gov/oiaf/1605/coefficients.html (accessed 17/10/2012).

[124] Energy Units., American Physical Society. (2011-05-05), available at http://www.aps.org/policy/reports/popa-reports/energy/units.cfm. 2011.

[125] Al-Abri, B., A Surface-Subsurface Model for the Techno-Economic and Risk Evaluation of Thermal EOR Projects. PhD Thesis, School of Engineering, Department of Power and Propulsion, Cranfield University. 2011.

[126] Marsh, W. D., Economics of Electric Utility Power Generation”. Oxford Engineering Science series, Oxford University. 1999.

[127] Junaibi, M., Techno-economic Study of a CHP Plant for Thermal EOR Operations. (MSc Thesis), Cranfield University. 2007.

[128] Harry, G. Stoll., Least Cost Electric Utility Planning. John Wiley & Sons, USA. 2006.

[129] Khatib, H., Economic Evaluation of Projects in the Electricity Supply Energy Market. in Hawaii International Conference, System Sciences, Proceedings of the 34th Annual, 3-6 Jan. 2001.

[130] NREL., A Manual for the Economic Evaluation of Energy Efficiency and Renewable Energy Technologies, National Renewable Energy Laboratory. 1995.

[131] Energy Market Authority., Long Run Marginal Cost (LRMC) Parameters for 1 January 2005 to 31 December 2006.

[132] OECD., Projected Costs of Generating Electricity. Organisation for Economic Co-operation and Department. 2010.

[133] GECOL., General Electric Company of Libya: Annual report, Libya. 2010.

[134] Energy Solution Center., Citing Internet Resource http://www.energysolutioncenter.org, (accessed 2014).

[135] Major, G., “Learning from Experiences with Small-scale Cogeneration, Centre for the Analysis and Dissemination of Demonstrated Energy Technologies”, Caddet Analysis Support unit. 1995.

[136] Educogen., Guid to Cogeneration, SAVE Program contract No XV11/4.1031/P/99-159, 2001.

[137] Panneerselvam, R., Engineering Economics. 2nd Edition, October, 2013.

[138] Meher-Homji, C. B., Gas Turbine Axial Compressor Fouling-A Unified Treatment of its Effects, Detection and Control, ASME Cogen Turbo IV, New Orleans, Louissiana, Also in International Journal of Turbo and Jet Engines, 9 (4), pp. 99-111. 1990.

[139] Meher-Homji, C. B., and Chaker, M. A., and Motiwala, H. M., Gas Turbine Performance Deterioration. Proceedings of the Thirtieth Turbomachinery Symposium, Tuebomachinery Laboratory, Texas A & M University, College Station, Texas, pp. 139-175. 2001.

[140] Kurz, R., and Brun, K., Degradation in Gas Turbine Systems, ASME International Gas Turbine and Aeroengine Congress, Munich, Germany, ASME Paper No. 2000-GT-345. 2000.

[141] Flashburg, L. S., and Haub, G. L., Measurement of Combustion Turbine Non Recoverable Deterioration. ASME International Gas Turbine and Aeroengine Congress, Cologne, Germany, ASME Paper No. 92-GT-264. 1992.

[142] http://www.weather-and-climate.com/average-monthly-rainfall-temperature-sunshine,maiduguri,Nigeria. Accessed August, 2012.

[143] Badeer, G. H., GE Aeroderivative Gas Turbines-Design and Operating Features, in GE Gas turbine reference library-GER3695E. GE power systems: Evendale-Ohio. 2002.

[144] LM2500+ Aeroderivative Gas Turbine. Available from: http://www.ge-energy.com/products_and_services/products/gas_turbines_aeroderivative/lm2500 plus aeroderivative gas turbines.jsp. [cited 2011 2 December].

[145] Aviation G., The Technological Evolution of a Popular Aero derivative Gas turbine. Press release. Available http://www.geaviation.com/aboutgeae/presscenter/marine/marine_20060511.html. [cited 2011 11 December]; 2006.

[146] Vigna, S. O., A Flexible Lifing Model for Gas Turbines: Creep and Low Fatigue Approach. (MSc Thesis), Cranfield University. 2006.

[147 Koff, B. L., Gas Turbine Technology Evolution-A Designer's Perspective. 14-17 July, Dayton, Ohio. 2003.

[148] Torbidoni, L., and Horlock, J. H., A New Method to Calculate the Coolant Requirements of a High-Temperature. Journal of Turbomachinery, ASME, Vol. 127, No. 1, pp. 191-199. 2005.

[149] Oskarsson, H. (2007), Material Challenges in Industrial Gas Turbines. Journal of Iron and steel Research, International, Vol. 14, no. 5, Supplement 1, pp. 11-14, 2007.

[150] Haslam, A., Mechanical Design of Turbo-machinery. Unpublished MSc course notes), Cranfield University, UK. 2005.

[151] Kaufman, J. G., Long, Z., and Ningileri, S., Application of Time-Temperature-Stress Parameters to High Temperature Performance of Aluminium Alloys. Warrendale, PA. 2007.

[152] Climate Leaders, Direct Emissions from Stationary Combustion Sources. *Greenhouse gas inventory protocol core module guidance.* United States Environmental Protection Agency (EAP). www.epa.gov/climateleaders. EPA430-K-08-003, May, 2008.

[153] Carbon Tax Centre 2007., Why setting the social cost of carbon is like sound parenting www.carbontax.org. Assessed 12/08/2011.

[154] Houssein, B. H., The Optimisation of the usage of Gas Turbine Generation sets for Oil & Gas Production using Genetic Algorithm. Unpublished PhD Thesis, Cranfield University, UK, 2010.

[155] MSc Lecture notes., SOE/1077, Cranfield University, 2000.

[156] Razak, A. M. Y., Industrial Gas Turbine: Performance and Operability, woodhead, 2007.

[157] Mund, F. C., Pilidis, P., Gas Turbine Compressor Washing: Historical developments, trends and main design parameters for engine systems, Journal of Engineering for gas turbines and power, vol. 128, no.2, pp. 344-353, 2006.

[158] Fabbri, A., Traverso, A., Cafaro, S., Compressor Performance Recovery systems: which solution and when. Proceedings of the Institute of Mechanical Engineers, Part A: Journal of power and energy, vol. 225, no.4, pp.447. 2011.

[159] Razak, A. M. Y., Carlyle, J. S., Methods, strategy and technology to reduce total equipment ownership cost. Aramco Conference Proceedings, pp. 69-84, October 1997.

[160] Viswanathan, R., Damage Mechanism & Life Assessment of High-Temperature Components. *ASME Internatinal, Metal Park, Ohio, 1998.*

[161] Reimer, M. M., Stress ans Strain. in Edward, P . J. (ed) Rules of Thumb for Mechanical Engineers, Gulf Publishing Company, 1997.

[162] Je-Chin, H., Sandip, D., Srinath, V. E., Gas Turbine Heat Transfer and Cooling Technology, Taylor and Francis, 2000.

[163] Horlock, J. H., Torbidoni, L., Turbine Blade Cooling: The Blade Temperature Distribution. Journal of power and energy, vol. 220, pp. 343-353, 2006.

[164] Laskaridis, P., Gas Turbine Lifing. Unpublished Fatigue and Fracture MSc lecture notes, 2010.

[165] Rubini, P., Turbine Blade Cooling. Unpublished MSc Lecture note, Cranfield University, 2009.

[166] Walsh, P. P., Fletcher, P., Gas Turbine Performance. Second edition, Blackwell Science, 2004.

[167] Guenette, G. R., Pappas, G., Epstein, A. H., The Influence of Non Uniform Spanwise inlet Temperature on Turbine Rotor Heat Transfer. AGARG Heat Transfer and Cooling in Gas Turbines, Voll. CP 527 1993, 12-16 October, Antalya, Turkey, pp. 5-11, 1993.

[168] Holland, M., Thakes, T., Rotor Blade Cooling in High Pressure Turbinees. Journal Aircraft, vol. 17, pp. 418-418, 1980.

[169] Mattingly, J. D., Heiser, W. H., Pratt, D. T., Aircraft Engine Design. American Institute of Aeronautics and Astronauts inc, Reston, 1996.

[170] Nickel Based Superalloy http://www.reade.com/products/Alloys/rene.html. accessed 2013.

[171] Rene Superalloys., Typical Chemical Purities (b), available at: metalsuppliersonline. accessed 2013.

[172] Ramaswany, V. G., A Constitutive Model for the Inelastic Multi-axial Cyclic Response of a Nickel Based Superalloy. Rene 80-NASA CR-3998, 2000.

[173] Leininger, J. C., Estimation of Creep and Fatigue Life for the Pre-design of Aircraft Engines. Unpublished MSc Thesis, Cranfield/Ecole Centrale de Nantes, 2003

[174] Marahleh, G., Kheder, A. R. I., Hamad, H. F., Creep Life Prediction of Serviced-Exposed Turbine Blades. Material Science, vol. 42, no.4, pp. 476-481, 2006.

[175] Uyioghosa, I., Orlando, M., Impact of Inlet Filter Pressure Loss on Single and Two-Spool Gas Turbine Engines for Different Control Modes. Journal of Engineering for Gas Turbines and Power. ASME, vol. 136, 2014.

[176] Boyce, M., Gas Turbine Engineering Handbook, 2nd ed., Gulf Professional Publishing, New York, 2002.

[177] Hung, W. S. Y., Accurate Method of Predicting the Effect of Humidity or Injected Water on NO_X Emissions from Industrial Gas Turbines, *ASME Paper 74-WA/GT-6,* 1974.

[178] Eric, M. G., Ogaji, S. O.T., Fuels and Combustion in Heat Engines. 2011.

[179] Breipohi, A. M., Electricity Price Forecasting Models, Power Engineering Society Winter Meeting, IEEE, vol.2,pp. 27-31, 2002.

[180] Temraz, H. K., Salama, M. M. A., Chikhani, A. Y., Review of Electric Load Forecasting Methods. Electrical and computer engineering, vol. 1, pp. 25-28, 1997.

[181] A, N., P, P., S, P., W, M., A Study of the Effect of Gas Turbine Emissions on the Economics of Nuturl Gas pipline transportation. Proceedings of the 20th International Conference on Nuclear Engineering ICONE20, Anaheim, California, USA, ASME, July 30-August 3, 2012.

[182] Wanis, M., Al-Abri, B., Pilidis, P., Nasir, A., Economic Evaluation of Industrial Gas Turbines for Electric Power Generation, GT2012-69495, Proceedings of *ASME TurboExpo2012*: Power for Land, Sea and Air, GT2012 June 11-15, Copenhagen, 2012.

[183] Wanis, M., Sethi, B., Pilidis, P., Hugo, P., Raja, S. R. K., A Method to Evaluate the Emissions of Gas Turbine for Power Generation, GT2012-69491, Proceedings of ASME TurboExpo: Power for Land, Sea and Air, June 11-15, Copenhagen, 2012.

[184] Temraz, H. K., Salama, M. M. A., Chikhani, A. Y., Review of Electric Load Forecasting Methods, Electrical and Computer Engineering. IEEE, Canada Conference, vol 1. 25-28, 1997.

[185] Marsh, W. D., Economics of Electric Utility Power Generation, Oxford Engineering Science series, Oxford University, 1980.

[186] Uyioghosa, I., P. Pilidis., Daniel, G., Orlando, M., Economic Evaluation of Online Washing for Heavy Duty Gas Turbine Operation. IDGT Power Engin eering, 2015.

[187] James, H., Carbon Tax Centre: Why Setting The Social Cost of Carbon is Like Sound Parenting. (www.carbontax.org accessed 21/05/2015), August 4, 2011.

[188] Andrew, K., Luke, R., and Fearghal, K., NO_X Taxation: A Sample review of examples of NO_X taxation systems. AP EnvEcon Ltd NouaUCD, Belfield, Dublin 4, Ireland. January, 2009.

[189] Roupa, A., System Design and Techno-Economic Analysis of On-line Compressor Washing in an Industrial Gas Turbine. MSc Thesis, School of Engineering, Department of Power and Propulsion, Cranfield University, 2011.

[190] Web. Power Technology., [Web Page]. Accessed and available http://www.power-technology.com/contractors/cleaning/. 2004

[191] Kristin, J., Mohsen, A.,Magnus, Genrup., Variation in Gas Turbine Blade Life and Cost due to Compressor Fouling-A Thermoeconomic Approach. Internal Journal of Applied Thermodynamics, vol.5, no.1,pp.37-47, 2002.

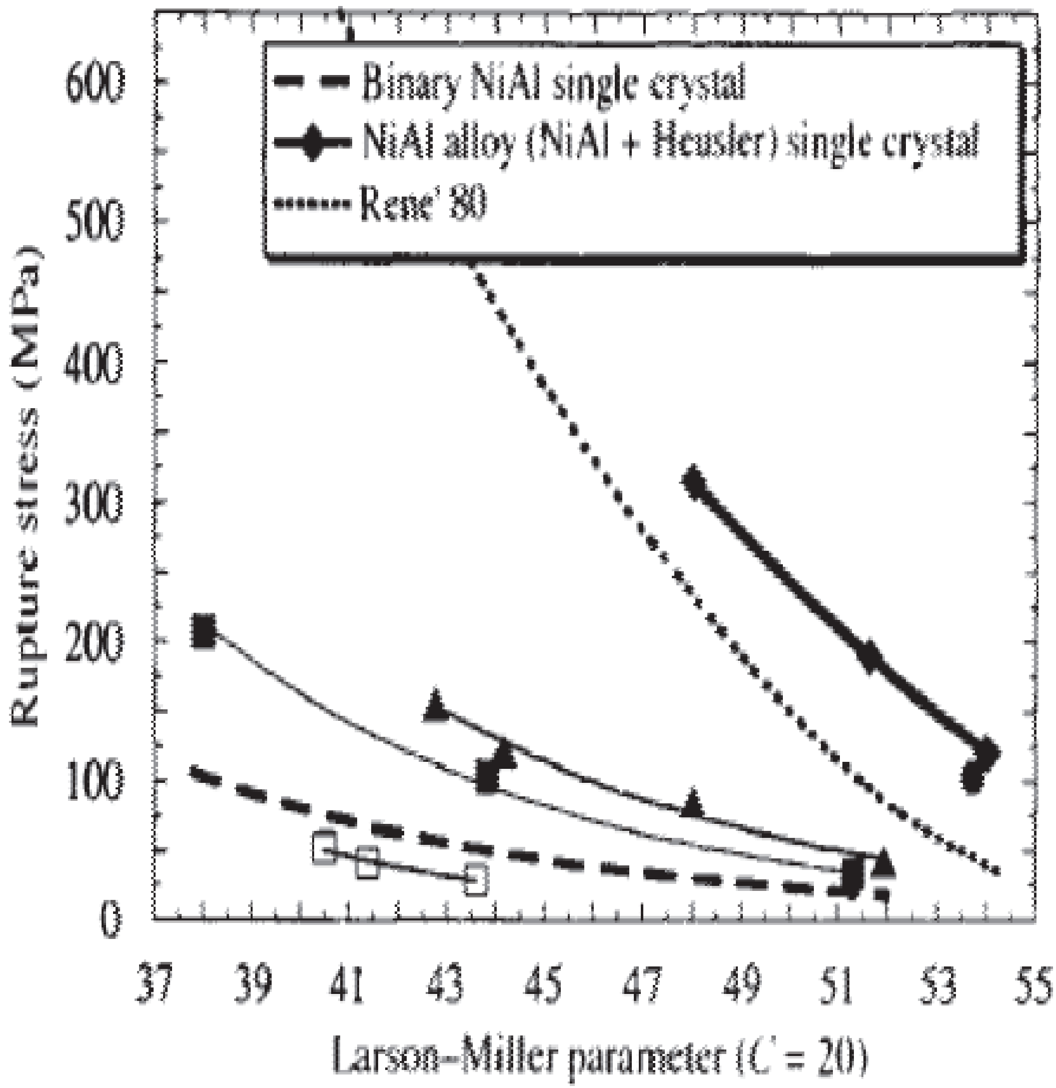
[192] Stroud, K. A., Dexter, J. B., Advanced Engineering Mathematics. 5th ed, 2011

[193] Venkataraman, P., Applied Optimization with MATLAB Programming. 2nd ed,2009.

APPENDICES

Appendix A

Creep rupture properties of Rene 80



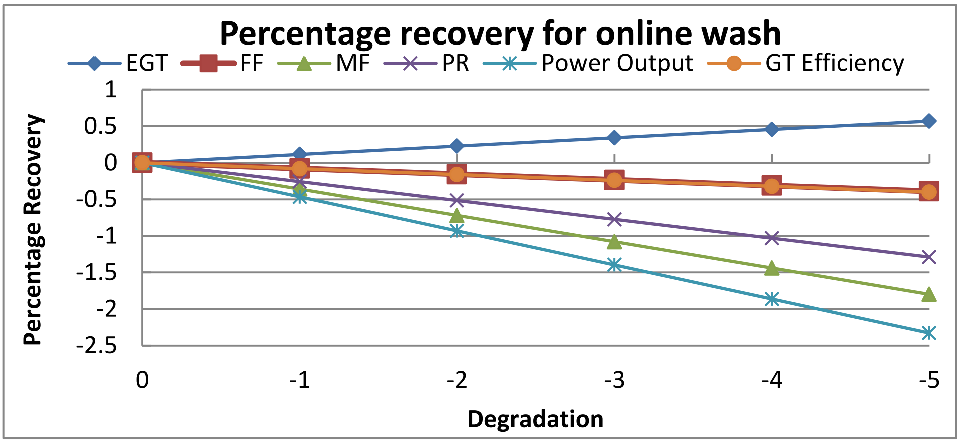
$$t_f = 10^{\left(\frac{1000LMP}{T} - 20\right)}$$

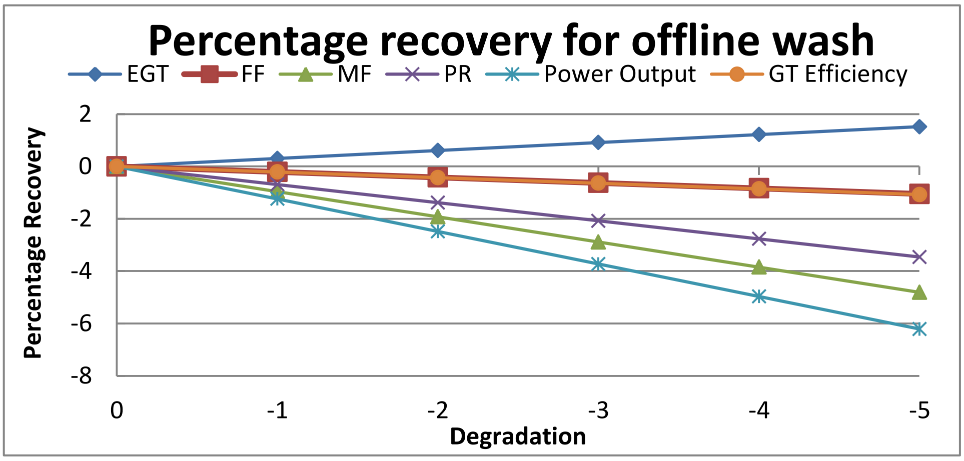
The temperature T is Absolute temperature in Rankine.

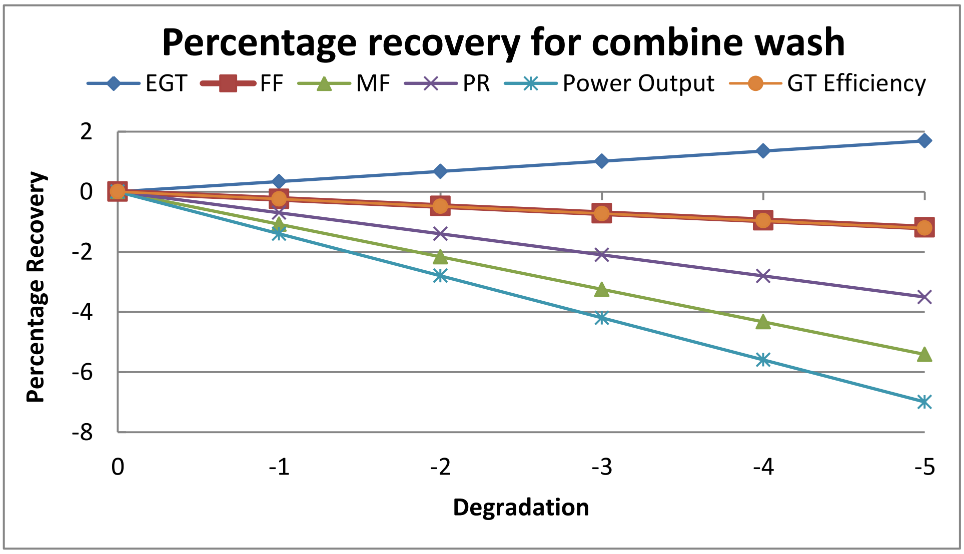
http://www.ewp.rpi.edu/hartford/~morens/EP/References/NiAl.pdf

Appendix B

PERFORMANCE PERCENTAGE DEVIATION

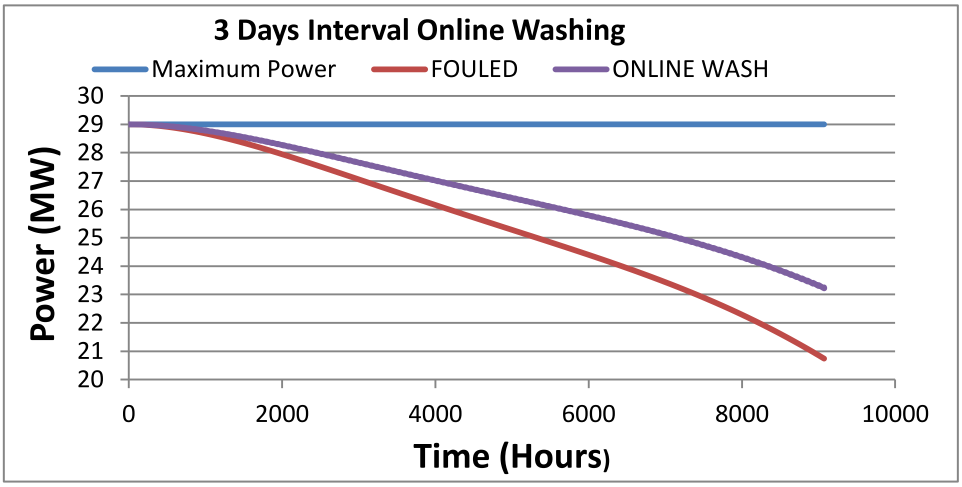


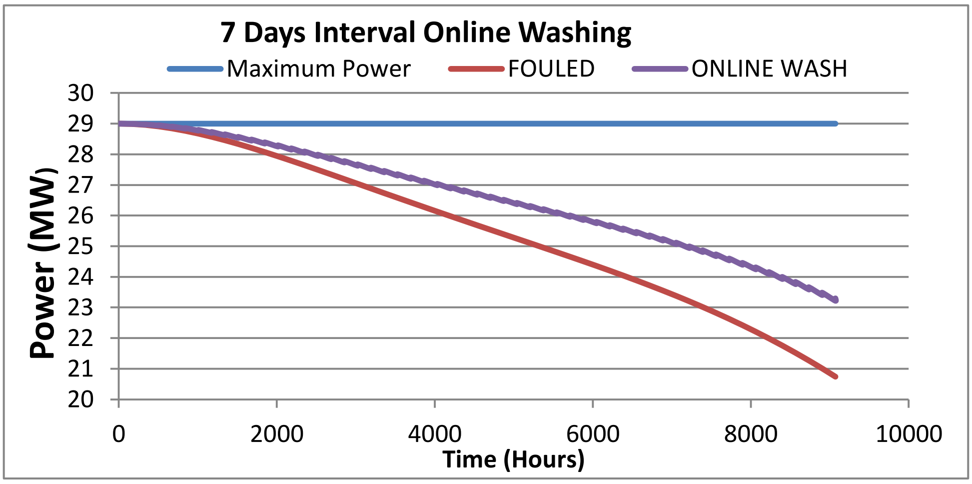


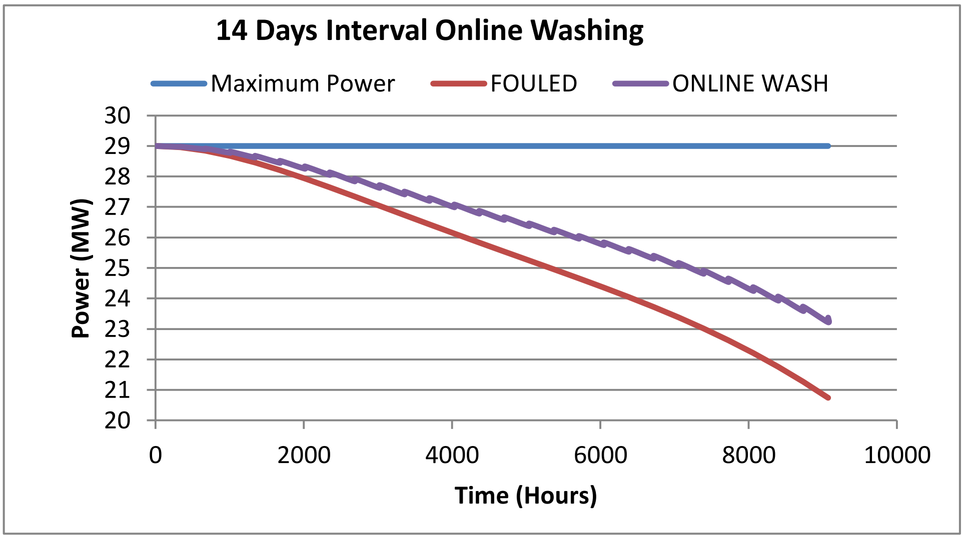


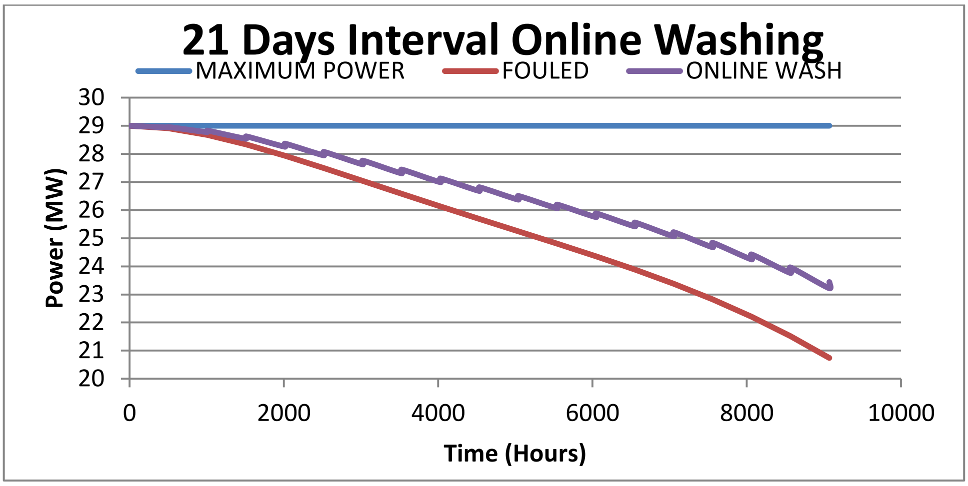
Appendix C

Effects of online compressor washing on GT performance



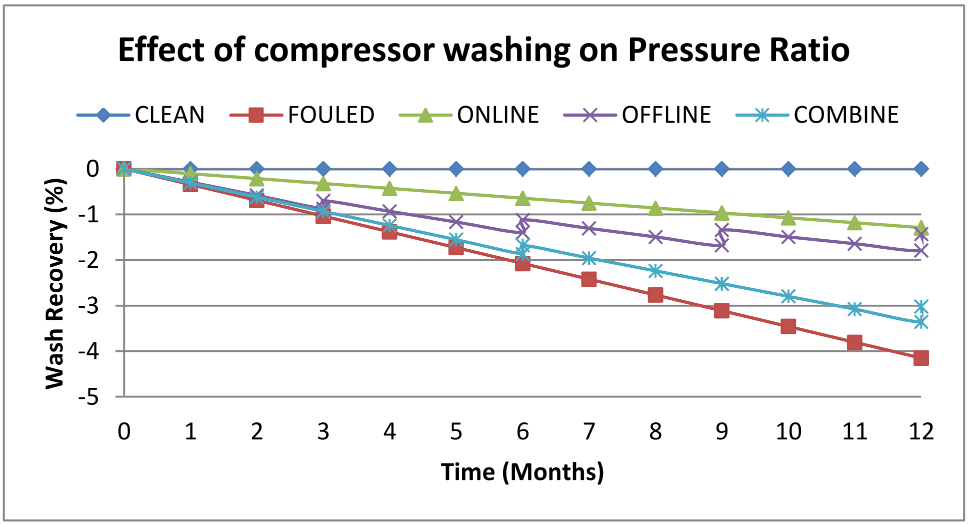


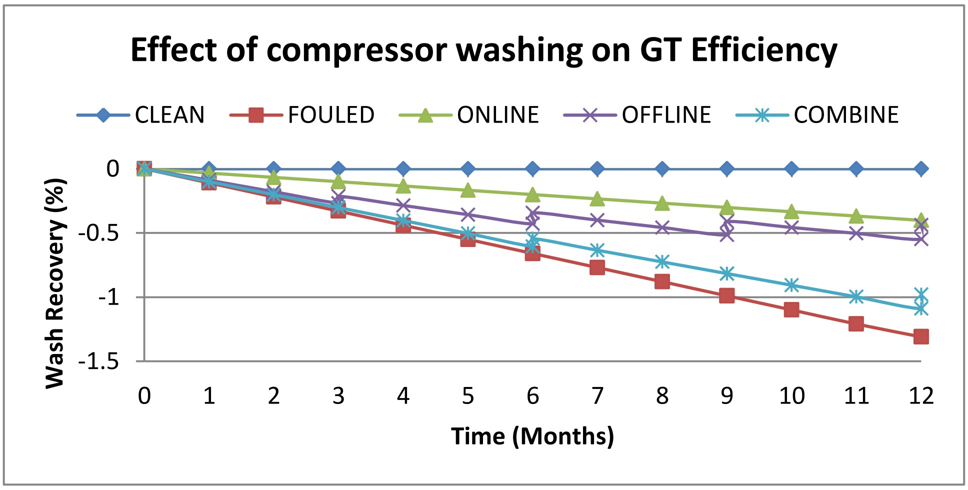


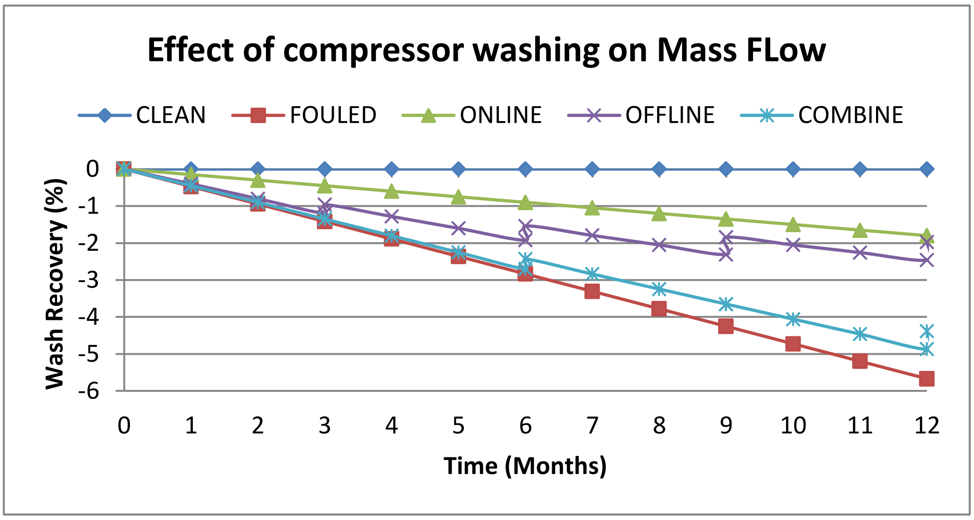


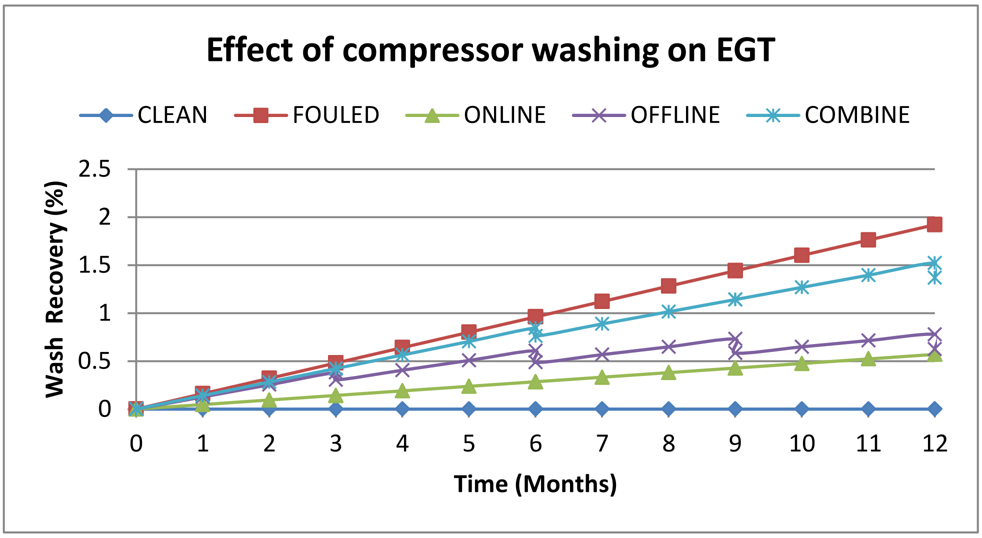
Appendix D

Recovery rate for compressor washing









Appendix E

Appendix E1 (Inlet annulus geometry of blade) [67]

Dmean (m)	0.9525
H (m)	0.0469
D tip (m)	0.9995
D hub (m)	0.9056

Appendix E2 (Stress Distribution values)

Blade height (%)	Stress (Mpa)
100	0
75	34.22
50	67.63
25	100.23
0	132.01

Density = 8194.5 Kg/m^3

Specific Heat Capacity = 460 J/KgK

Melting Point = 1330^0C = 1603K

Appendix E3 (Average stress)

STRESS	LMP
34.22	52.9
50.925	52.6
83.93	52.1
116.12	50.2

Appendix E4 (HPT blade creep life estimation)

Cooling effectiveness		0.4
Compressor inlet temperature		762
Constant C		20
LMP		50.24
Rankine		1.8
TBC	150k	

Appendix E5 (Clean blade life)

BLADE SPAN	Tg	TBC	BMT	1.8	LIFE
75	1530	1380	1125.2	2025.36	1848992.97
50	1612.5	1462.5	1174.7	2114.46	60497.9926
25	1530	1380	1125.2	2025.36	593192.807
0	1447	1297	1075.4	1935.72	1555265.72

Appendix E6 (Fouled blade life)

Tg	TBC	BMT	1.8	LIFE
1555	1405	1140.2	2052.36	834407.7241
1645	1495	1194.2	2149.56	23827.68342
1555	1405	1140.2	2052.36	271727.9813
1468	1318	1088	1958.4	773535.7538

Appendix E7 (Online blade life)

Tg	TBC	BMT	1.8	LIFE
1547.5	1397.5	1135.7	2044.26	1057029.1
1635.25	1485.25	1188.35	2139.03	31411.243
1547.5	1397.5	1135.7	2044.26	342698.65
1461.7	1311.7	1084.22	1951.596	952220.23

Appendix E8 (Offline blade life)

Tg	TBC	BMT	1.8	LIFE
1535	1385	1128.2	2030.76	1574304.637
1619	1469	1178.6	2121.48	50088.48763
1535	1385	1128.2	2030.76	506596.6271
1451.2	1301.2	1077.92	1940.256	1350742.176

Appendix E9 (Combine blade life)

Tg	TBC	BMT	1.8	LIFE	STRESS	LMP
1532.5	1382.5	1126.7	2028.06	1705947	34.22	53.2
1615.75	1465.75	1176.65	2117.97	55039.121	50.925	52.4
1532.5	1382.5	1126.7	2028.06	548129.86	83.93	52.2
1449.1	1299.1	1076.66	1937.988	1449281.3	116.12	50.7

Appendix E10 (Creep life in hours)

BLADE SPAN	CLEAN	FOULED	ONLINE	OFFLINE	COMBINE
75	1109395.78	500644.6345	634217.48	944582.7822	1023568.17
50	36298.7956	14296.61005	18846.746	30053.09258	33023.4728
25	355915.684	163036.7888	205619.19	303957.9762	328877.919
0	933159.432	464121.4523	571332.14	810445.3054	869568.804

Appendix E11 (Creep life in years)

BLADE SPAN	CLEAN	FOULED	ONLINE	OFFLINE	COMBINE
75	126.643354	57.15121398	72.399255	107.8290847	116.845682
50	4.14369812	1.632033111	2.151455	3.430718331	3.76980283
25	40.6296443	18.61150557	23.47251	34.69839911	37.5431414
0	106.525049	52.98190095	65.220564	92.51658738	99.2658452

Appendix F (Emissions)

Appendix F1 (Combustor Temperature)

Temperature K		2100			2200
Product	n	H_tT	n H_tT	H_tT	n H_tT
CO_2	3	-296.02	-888.06	-289.95	-869.85
H_2O	4	-164.00	-656.00	-158.79	-635.16
O_2	1	62.990	62.99	66.80	66.80
N_2	22.56	59.75	1347.96	63.37	1429.63
Σ			-133.11		-8.58

Appendix F2 (NOx calculation)

0.0078	0.0081	0.0088	0.0094	0.0104	0.01165	0.0132	0.0153	CLEAN
0.0122	0.0128	0.0137	0.0148	0.0162	0.0182	0.0206	0.0239	FOULED
0.00868	0.00904	0.00978	0.01048	0.01156	0.01296	0.01468	0.01702	ONLINE
0.00956	0.00998	0.01076	0.01156	0.01272	0.01427	0.01616	0.01874	OFFLINE
0.0082	0.0086	0.0093	0.0099	0.01098	0.0123	0.0139	0.0162	COMBINE
1095	2190	3285	4380	5475	6570	7665	8760	TIME
1910	1937	1965	1993	2021	2049	2077	2105	TPZ

Appendix F3 (CO calculation)

0.02043	0.01886	0.01735	0.01596	0.01469	0.01351	0.01242	0.01142	CLEAN
0.03295	0.03042	0.02799	0.02575	0.02369	0.02179	0.02004	0.01843	FOULED
0.02293	0.021172	0.019478	0.017918	0.01649	0.015166	0.013944	0.012822	ONLINE
0.025438	0.023484	0.021606	0.019876	0.01829	0.016822	0.015468	0.014224	OFFLINE
0.021682	0.020016	0.018414	0.016939	0.01559	0.014338	0.013182	0.012121	COMBINE
1095	2190	3285	4380	5475	6570	7665	8760	TIME
1910	1937	1965	1993	2021	2049	2077	2105	TPZ

Appendix F4 (CO_2 Emission Tax)

	Cost of fuel flow increase kg/s	COS2 (kg)	CO2 (ton)	Appendiz F4 Carbon Tax ($)	Emission Reduced (kg)	Revenue saved ($)
CLEAN	1.7222	54311299	54311.299	1049294		
FOULED	1.9835	62551656	62551.656	1208498	8240.4	159204
ONLINE	1.9051	60079234	60079.234	1160731	2472.1	47761.1
OFFLINE	1.7745	55960632	55960.632	1081159	6592.3	127363

COMBINE	1.7483	55134389	55134.389	1065196	7416.3	143283

Appendix F5 (NOX Emission Tax)

Cases	fuel flow	NOX (g/kg) fuel	NOX (kg)	Nox Tax ($)	Emission reduced (kg)	Revenue Saved ($)
CLEAN	1.7222	0.0153	26350	1117489		
FOULED	1.9835	0.0239	47406	2010474	21055.99	892984.5
ONLINE	1.9051	0.01874	35702	1514104	6316.797	267895.4
OFFLINE	1.7745	0.01702	30202	1280866	16844.792	714387.6
COMBINE	1.7483	0.0162	28322	1201156	18950.391	803686.1

Appendix F6 (CO Emission Tax)

Cases	fuel flow	CO (g/kg fuel)	CO (kg)	CO Tax ($)	Emission reduced (kg)	Revenue saved ($)
CLEAN	1.7222	0.01142	19668	845704		
FOULED	1.9835	0.01843	36556	1571904	16888.381	726200.4
ONLINE	1.9051	0.014224	27098	1165220	5066.5143	217860.1
OFFLINE	1.7745	0.012822	22753	978363	13510.705	580960.3
COMBINE	1.7483	0.012121	21191	911219	15199.543	653580.3

Appendix G (Power output cost)

Cases	Area under curve	Energy (MWh)	Revenue ($/kwh)	Maximum revenue ($)	Energy loss (MWh)	Energy loss cost ($)	Fuel cost ($)	Revenue loss/gain
CLEAN	254040000	254040	185449200	18544920	0	0	111269520	0
FOULED	237464000	237464	173348720	17334872	16576	1210048	104009232	12100480
ONLINE	242436800	242437	176978864	17697886	11603	847034	106187318	3630144
OFFLINE	250724800	250725	183029104	18302910	3315	242010	109817462	9680384
COMBINE	252382400	252382	184239152	18423915	1658	121005	110543491	10890432
LOSS	16576000	16576	12100480	1210048			7260288	

Appendix H (Creep cost estimation)

Cases	Creep life (hrs)	Creep life (yrs)	Creep cost ($)	Revenue loss ($)

CLEAN	31229.8777	3.565054528	921281.39	0
FOULED	12403.094	1.415878307	365891.27	555390.1189
ONLINE	16310.562	1.861936297	481161.58	440119.8132
OFFLINE	25900.4553	2.956672975	764063.43	157217.9608
COMBINE	28436.2246	3.246144364	838868.63	82412.76449

Appendix I (Economic Optimisation)

TABLE A

	CREEP COST	EMISSION COST	PERFORMANCE COST				TOTAL PROFIT		
Basis	C	E	P	w1	w2	w3	b	Check	
w1	C1	E1	P1	1	0	0	b1		ONLINE
w2	C2	E23	P2	0	1	0	b2		OFFLINE
w3	C3	E3	P3	0	0	1	b3		COMBINE
P	PC	PE	PP	0	0	0	0		PROFIT

TABLE B

C1	440120	E1	3840055	P1	165464	b1	66345907	PC	555390
C2	152218	E2	3340389	P2	616352	b2	69102682	PE	4790876
C3	82413	E3	3177571	P3	320285	b3	70115392	PP	12100480

Form a simplex tableau

TABLE C

Basis	C	E	P	W1	W2	W3	b	check	
w1	440120	3840055	165464	1	0	0	66345907	7.1E+07	
w2	152218	3340389	6E+05	0	1	0	69102682	7.3E+07	KEY ROW
w3	82413	3177571	320285	0	0	1	70115392	7.4E+07	
P	555390	4790876	12100480	0	0	0	0	1.7E+07	
			KEY COLUMN						

ROW 1	400.968833	ROW 2	112.1156	ROW 3	218.915628	PIVOT	616352

TABLE D

Basis	C	E	P	W1	W2	W3	b	check

w1	440120	3840055	165464	1	0	0	66345907	7.1E+07	
w2	0.24696602	5.41961249	1	0	1.6224E-06	0	112.11561	118.782	KEY ROW
w3	82413	3177571	320285	0	0	1	70115392	7.4E+07	
P	555390	4790876	12100480	0	0	0	0	1.7E+07	

KEY COLUMN

PIVOT 1

TABLE E

Basis	C	E	P	W1	W2	W3	b	check
w1	399256.015	2943304.24	0	1	-0.268457	0	47794809	5.1E+07
P2	0.24696602	5.41961249	1	0	1.6224E-06	0	112.11561	118.782
w3	3313.48847	1441750.41	0	0	-0.5196462	1	34206443	3.6E+07
P	-2433017.4	-60789037	0	0	-19.632418	0	-1.357E+09	-1E+09

KEY COLUMN

ROW 1	16.2384876	ROW 2	20.687	ROW 3	23.7256	PIVOT	2943304

TABLE F

Basis	C	E	P	W1	W2	W3	b	check	
E1	0.13564891	1	0	3.4E-07	-9.121E-08	0	16.238488	17.3741	KEY ROW
w2	0.24696602	5.41961249	1	0	1.6224E-06	0	112.11561	118.782	
w3	3313.48847	1441750.41	0	0	-0.5196462	1	34206443	3.6E+07	
P	-2433017.4	-60789037	0	0	-19.632418	0	-1.357E+09	-1E+09	

KEY COLUMN

TABLE G

Basis	C	E	P	W1	W2	W3	b	check	
w1	0.13564891	1	0	3.4E-07	-9.121E-08	0	16.238488	17.3741	
w2	-0.4881985	0	1	-2E-06	2.1168E-06	0	24.109302	24.6211	
w3	-192258.38	0	0	-0.4898	-0.3881451	1	10794597	1.1E+07	KEY ROW

P	5812949.23	0	0	20.6533	-25.176949	0	-369530712	-4E+08	
	KEY COLUMN								
ROW 1	119.709679	ROW 2	-49.3842	ROW 3	-56.1463	PIVOT	-192258.38		

TABLE H

Basis	C	E	P	W1	W2	W3	b	check	
w1	0.13564891	1	0	3.4E-07	-9.121E-08	0	16.238488	17.3741	
w2	-0.4881985	0	1	-2E-06	2.1168E-06	0	24.109302	24.6211	
Cw3	1	0	0	2.5E-06	2.0189E-06	-5E-06	-56.1463	-55.146	KEY ROW
P	5812949.23	0	0	20.6533	-25.176949	0	-369530712	-4E+08	
	KEY COLUMN								

TABLE I

Basis	C	E	P	W1	W2	W3	b	check
Ew1	0	1	0	-6E-09	-3.651E-07	7.1E-07	23.854672	24.8547
Pw2	0	0	1	-6E-07	3.1024E-06	-3E-06	-3.3012377	-2.3012
Cw3	1	0	0	2.5E-06	2.0189E-06	-5E-06	-56.1463	-55.146
P	0	0	0	5.84295	-36.912549	30.2351	-43155122	-4E+07

TABLE J

EMISSION	23.854672	PERFORMANCE	-3.30124	CREEP	-56.1463	Pmax	-43155122
Pmax	43155121.8						
creep	555390	emission	4790876	performance	12100480		